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Set-up of a new method for tests on switches addressed to the homologation of special railway vehicles: from planning to testing

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Alla mia famiglia

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CHAPTER 1
INTRODUCTION

The argument faced in this Ph.D. thesis is a derivation of a commercial contract between Trenitalia Cargo (TI), Italcertifer (ITCF), a railway certification and research institute, the University of Florence (UNIFI) and the Politecnico di Milano (POLIMI). The aim of this research work is to investigate, through a complete experimental "on-track" campaign, the behaviour of SAADKMS rail vehicles while routing switches. Such vehicles are a particular kind of wagons employed for the combined transportation of heavy road vehicles used for traffic through the Italian boundaries toward Germany, Austria and Swiss. In Italy the traffic of these wagons is restricted by some circulation limitations. The final goal of this thesis is to define an exhaustive test layout for the switch behaviour investigation of these wagons, attaining a normative validity collecting sufficient informations about the dynamic behaviour that could be used for the removal of circulation limitations.

Following a brief introduction to the arguments handled in this thesis is discussed.

The first chapter illustrates the SAADKMS vehicle, which is the main "actor" of this work, focusing on the most critical parts that mainly influence its dynamic.

In the second chapter some mixed notions about the railway engineering are introduced, in order to give some instruments to the reader for a smarter understanding of the following arguments.

The third chapter resumes an overview of the historical activities performed on such wagons that lead to the definition of the procedure for the testing on switches.

In chapter four an analysis of the results of the multibody models of the vehicle is discussed, and constitutes the basis for the definitions of the track layout and vehicle condition on which perform the tests.

In chapter five are described the instrumentation layout with the characteristics of each component and the reason for their employment, the calibration method and the calibration results.

Then the last chapter contains the description of the whole experimental campaign, focusing on the description of the strategy adopted, and the comments of the results.

Finally a summary of all the activities performed, with some conclusive considerations on the results and the normative validity of the tests are going to be argued.

All the results achieved in the course of this thesis have been presented to ANSF (The Italian National Safety Authority) in order to obtain the authorization for the prosecution of the experimental tests aimed at the removal of the circulation limits.

1.1 The railway normative world

An efficient railway net of connections inside a country boundaries and toward neighbour states, is a very contemporary idea. Many projects about high speed trains that could run across the borders are growing up, and more and more technical specifications for the interoperability requirements are requested by train operators. For example, historically each country employs aerial cableways that supply electricity at different voltages, thus new locos hare geared with multiple transformer for the conversion of the energy to the motors. This “unification” process is aided by TSI (Technical Specifications for Interoperability) standards, that have the task to define homogeneous rules about weight, dimensions, limit gauges, length and circulation rules. The aim of the ERA (European Railway Association) is the achievement of an high level of harmonization and standardization of the European railways, improving the role of the railway traffic across the Europe. All the rules introduced with TSI have the goal to guarantee safety and, at the same time, permit a fluent train traffic harmonizing both the vehicles and the rails.

Figure 1.1. RoLa commercial routes

In general the European normative environment is regulated by another institution, the CEN (European Committee for Standardization) that at a lower level, defines a series of advices that shall be complied by local national rules. Some arguments faced by these standards are drawn by ERA in the TSI, but all the others regulations pertaining to most of the railway world issues are their own competence. In the course of this thesis the reference EN standards pertaining the on-track tests addressed to the homologation of a vehicle, is the EN14363 [1], that constitutes the evolution of the UIC 518 [2], previously released by the UIC (International Union of Railways). Despite these standards, each country has its own internal rules, although compliant with the TSI ,that introduce some modifications to some specific arguments, making more complex the process of homogenization across the Europe.

1.2 The aims of the thesis

When talking about interoperability not only high speed trains [3] are included, but also freight wagons and standard vehicles, indeed another important and potentially efficient railway business is the commercial traffic of goods [4]. A particular kind of freight wagons is the SAADKMS vehicle. A convoy constitutes by these vehicles is the so called "RoLa" which is the acronym of "Rollende Landstraße" (Rolling Roads).

As introduced SAADKMS are special freight wagons employed for the combined transportation of trucks on railway lines. Their commercial routes are mainly across countries boundaries as depicted by figure 1.1. The attention on this type of transportation is very high, because it decreases the occupancy of roads, pollution, driving hours for drivers, and bypass the rules about the allowed time tables for trucks travelling. Moreover they constitute a more energetically efficient transportation than a single truck. All these positive aspects are opposed by some drawbacks, first of all their unconventional dynamic behaviour affected by all the friction parts and their very small wheels used to decrease their height because of the necessity to stay under the limit gauge allowed by the infrastructures.

The investigation about their dynamic, indeed, consequently to some past derailments reported in table 1.1, are the core of the project on the basis of this thesis. Such investigation is addressed to their behaviour on switches and on the influence of some of their parts. This investigation project began in the early noughties and stopped few years later, thus the activity described in this thesis is the continuation, or better, a new begin of such project. Despite these few words, all this activity could seems simple and of fast realization but there are many normative, logistical, technical and safety aspects that shall be observed coordinating many entities unravelling through the multitude of rules of the railway world.

In conclusion this thesis have the final target to define a valid test layout in terms of instrumentation of the wagons and testing scenarios, accordingly with the general rules of the standards but facing an argument that is not specifically treat in it: tests on switches. Last but not least is the aim to verify if these wagons are able to run safely on switches, satisfying the requests and expectations of ANSF (Agenzia nazionale per la sicurezza fer-

ID	Traction	Cars	Load	Bogie	Switch Geometry	Track	Direction
1	pulled	2	empty empty	rear front	S46/150/0.12 S50/245/0.10	diverging track diverging track	trailing trailing
2	pulled	2	empty empty	rear front	SI 60/170/0.12	diverging track	
3	pushed	3	empty empty empty	rear both front	S46/150/0.12	diverging track	facing
4	pulled	2	empty empty	rear both	S46/150/0.12 S46/150/0.12	diverging track diverging track	trailing trailing
5	pulled	3	empty empty empty	rear both front	S60/170/0.12 SI 60/170/0.12	diverging track diverging track diverging track	facing
6	pushed		empty		S50/170/0.12	diverging track	trailing
7	pushed		empty		S50/170/0.12	diverging track	trailing

Table 1.1. Past derailments

roviaria). ANSF is the bureau of the Transportation Ministry dedicated to the regulation of Italian railways, that have a tight connection with the European standardization entities. The development of this project has been of course subordinate to the coordination of the synergic work of many actors (figure 1.2) :

- Italcertifer that is an Italian "Notified Body", which, in this work, has the role to coordinate the whole project and relate and manage the relationship with ANSF;
- the Politecnico di Milano, that is the laboratory in charge for the instrumentation of the vehicles and signal acquiring and processing;
- Trenitalia Cargo which is the train operator that supplies the train drivers, the wagons and the logistic support;
- RFI, that is the owner of the line and the granter of the line interruptions;
- the University of Florence that, cooperating with Italcertifer, manages the whole project supplying technical and management contribution.

The last target is to include this study case in an informative annex of the next EN14363 leaflet, that in its last draft, introduces the argument of tests on switches. Such draft is based on a work performed by the German work group of DB (Deutsche Bahn), regarding the routing of a single specific switch. The work performed in this thesis is completely independent by that, but has many common points, being a valid integration for a more complete normative scenario.

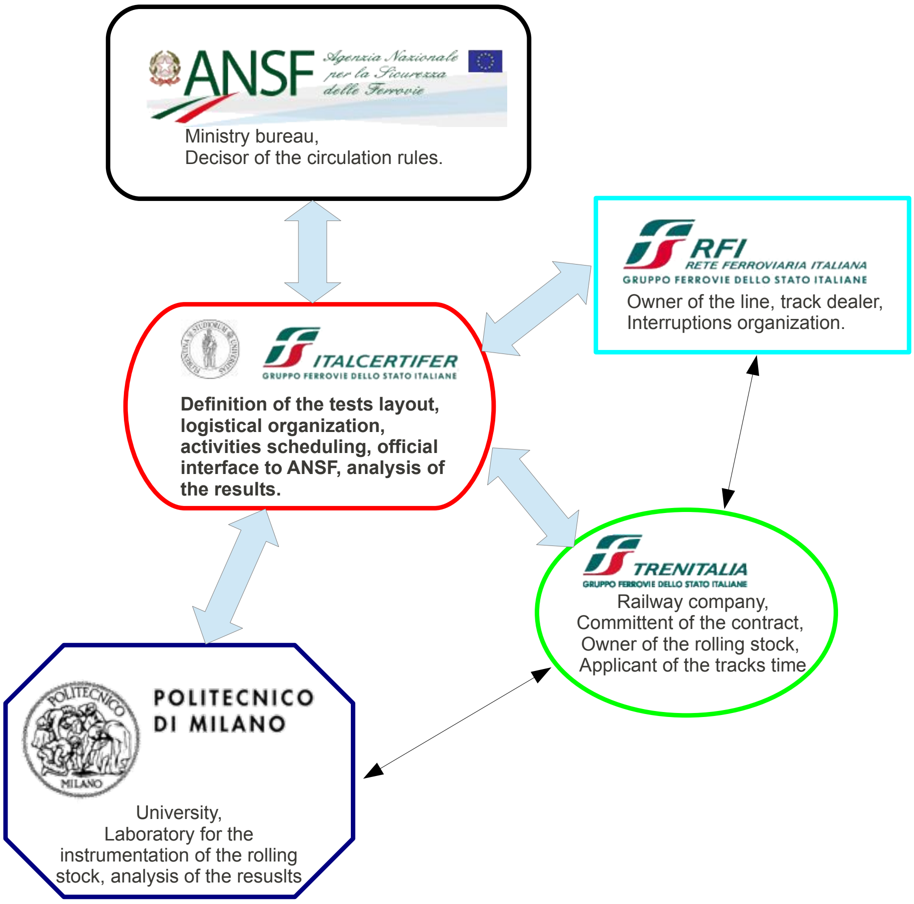


Figure 1.2. Relationship between institutions

CHAPTER 2

THEORY PILLS

In this chapter some concepts about the railway engineering will be introduced in order to get the reader into the railway issues that are going to be used in the later part of this work. Since the study of this thesis is strictly correlated with European norms and leaflet the technical aspects are reported using the norms nomenclature and with many quotes to them.

2.1 Equivalent conicity

The vehicle running behaviour is highly influenced by the contact geometry, that depends on both the wheels profiles and the rail profiles, since it is the interface where forces are exerted.

"Equivalent conicity" (γ_e) plays a foundamental role since it permits the appreciation of wheel-rail contact geometry on large-radius curve and tangent track.

The "UIC 519" [5] leaflet describes the method to determine the equivalent conicity for a couple of wheelset-rail. In order to understand the physical concepts that are behind the value of γ_e the it's useful to start by the kinematic study of a wheelset with constant conicity profiles of the wheels.

The equation of motion of a free wheelset on a track, without inertia, is described by the differential equation:

$$\ddot{y} + \frac{V^2}{er_0}\Delta r = 0 \tag{2.1}$$

where:

- e is the distance between contact points (about 1500 mm for standard gauge);
- r_0 is the radius of the wheel when the wheelset is centred on the track;

- V is the speed on forward movement of the vehicle;
- Δr is the difference of rolling-radii between right-hand and left-hand wheels

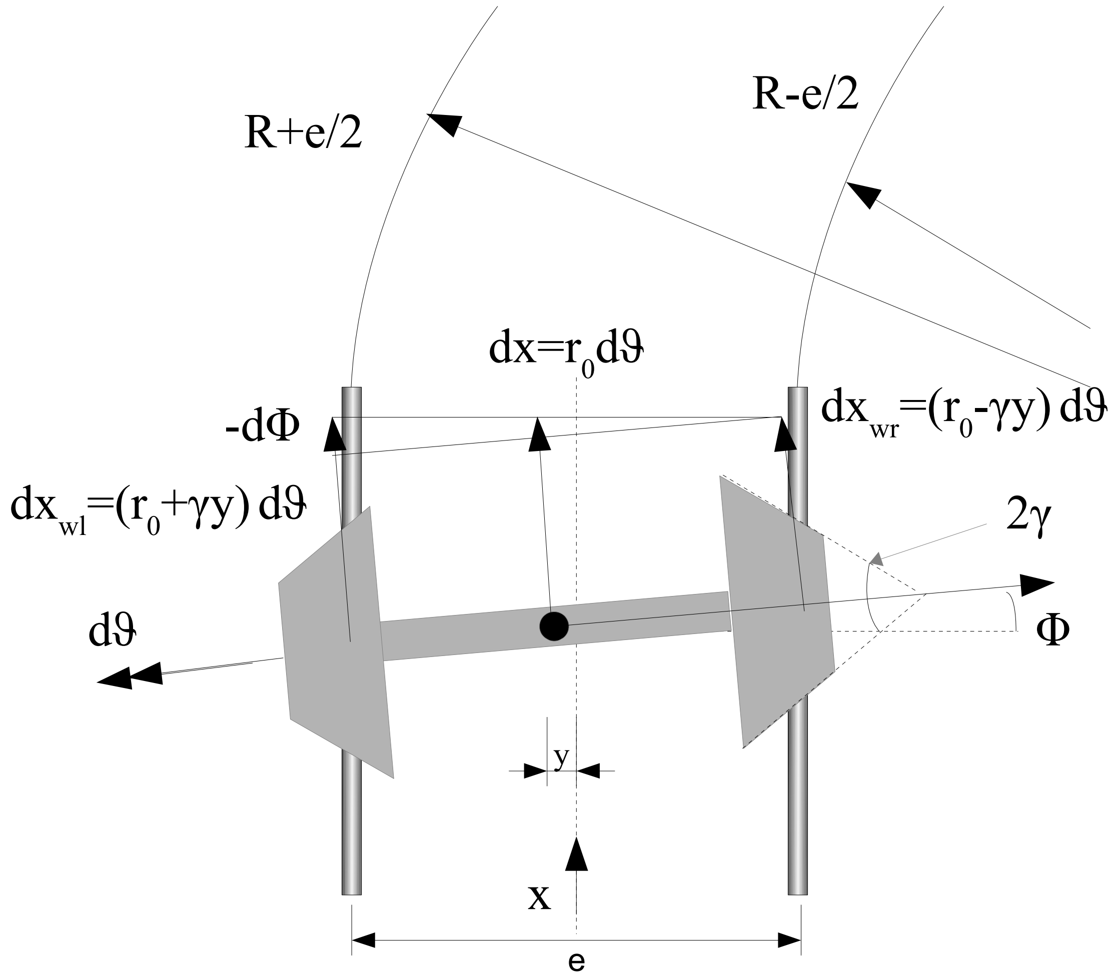


Figure 2.1. Schematic representation of a wheelset trajectory

since the velocity V could be assumed constant, it's determined by:

$$V = \frac{dx}{dt}$$

hence, for the derivative rule for the composed function:

$$\frac{dy}{dt} = V\frac{dy}{dx} \quad \text{and} \quad \frac{d^2y}{dt^2} = V^2\frac{d^2y}{dx^2}$$

and, since the wheels have a constant conical profile of angle γ:

$$\Delta r = 2y \tan\gamma$$

thus the differential equation 2.1 become a second order differential equation with constant coefficients:

$$\frac{d^2y}{dt^2} + \frac{2\tan\gamma}{er_0}y = 0 \tag{2.2}$$

The solution of this equation is a sinewave with a wavelength of λ:

$$\lambda = 2\pi\sqrt{\frac{er_0}{2\tan\gamma}}$$

Of course real profiles don't have a constant conical profile γ, so, in order to still use the linear differential equation 2.1 $\tan\gamma$ can be replaced by $\tan\gamma_e$ which is called the "equivalent conicity".

$$\tan\gamma_e = \left(\frac{\pi}{\lambda}\right)^2 2er_0 \tag{2.3}$$

Thus, by UIC 519 definition:

≪*the equivalent conicity is equal to the tangent* $\tan\gamma_e$ *of the cone angle of a wheelset with coned wheels whose lateral movement has the same kinematic wavelength as the given wheelset (but only on tangent track and on large-radius curves)*≫

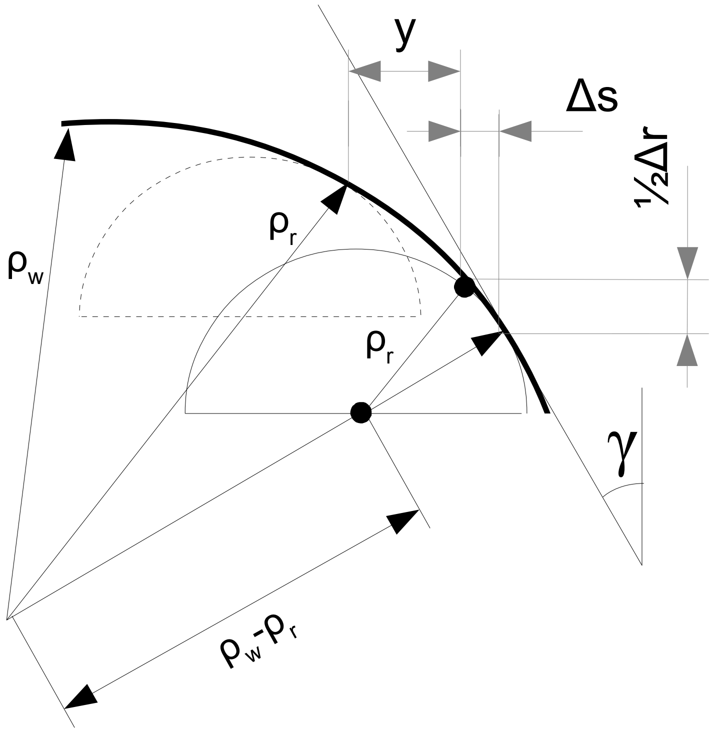


Figure 2.2. Local wheel-rail profiles coupling

Thus, assuming locally a constant curvature wheel ρ_w and rail ρ_r profile 2.2, it's possible to calculate the displacement ΔS of the contact point on both, the wheel and the rail, in relation to a lateral movement of the wheel y.

$$\begin{aligned}\Delta S_w &= \frac{\rho_r}{\rho_w - \rho_r} y \\ \Delta S_r &= \frac{\rho_w}{\rho_w - \rho_r} y\end{aligned} \tag{2.4}$$

if the lateral displacement is very small (0.2 mm [5]), and ρ_w is infinite (constant conicity γ), $\Delta S = 0$ and:

$$\Delta r = 2y \tan\gamma \tag{2.5}$$

Since the wheels are rigidly connected by the axle, in relation with figure 2.1, the tangential speed of the outer wheel should be higher than that of the inner wheel to avoid sliding. In order to avoid sliding the arc described by each wheel, rolling at the θ velocity, and the track distance must be the same, this concept is expressed by the following formula:

$$\frac{r_0 + \tan\gamma y}{r_0 - \tan\gamma y} = \frac{R + e/2}{R - e/2} \tag{2.6}$$

In the plane xy $d\Phi = dy/dx$, and the portion of track covered ds is equal to $ds = -Rd\Phi$. For small angles $ds \approx dx$ thus:

$$\frac{1}{R} \approx -\frac{d\Phi}{dx} = -\frac{d^2y}{dx^2} \tag{2.7}$$

substituting the 2.7 into the 2.6 we obtain the following equation:

$$\frac{r_0 + \tan\gamma y}{r_0 - \tan\gamma y} = \frac{1 - \frac{e}{2}\frac{d^2y}{dx^2}}{1 + \frac{e}{2}\frac{d^2y}{dx^2}} \tag{2.8}$$

from which we obtain the 2.2. In physical therms a wheelset with a constant conical profile moved from the central position, describes a sinusoidal trajectory that depends on the initial position and on the conicity of the profiles.

In conclusion a profile with variable values of conicity as depicted in figure 2.3, describes many waveforms depending on the value of γ_e that depends on the initial lateral displacement.

From this some considerations could be done, for example that a low conicity profile have a longer wavelength and than a better stability behaviour, moreover an high conicity improve the steering capability of a vehicle. Usually the conicity tends to increase with the wear of the profile till values of 0.4 where some stability issues may occur.

2.2 Derailment criterion

[6]

Usually when people think about the safety of a train imagine movies where trains running at high speed drop down the binaries with catastrophic consequences. Although

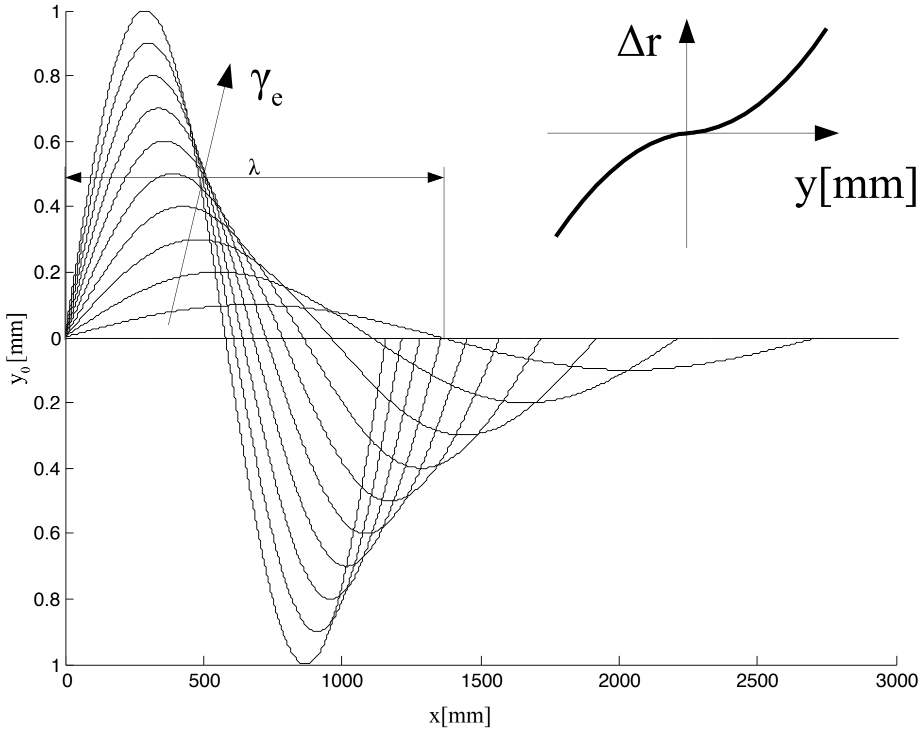


Figure 2.3. Waveform of a wheelset trajectory

this is true as many railway accidents demonstrate, also the low speed derailment is an interesting phenomenon to investigate because usually vehicles must run on old, secondary binaries that present higher superelevations and worst maintenance conditions. Obviously if the derailment occurs at high speeds the damages are more consistent because of the amazing linear momentum of the train, but derailment at low speeds are more frequents because each vehicle must run at low ($< 30km/h$) before reach high speeds. Moreover different parameters affect the safety of the train at high and low speeds: for the first the coupling between the track wave length and the resonance frequency of the vehicle is fundamental, the second the track cant and vertical irregularities are of primary importance. In addition, at low speeds, the lateral forces are more constant and persistent because of the higher friction coefficient at the contact and between the mechanical components of the train.

In conclusion, since the most catastrophic event that can occur during the run of a railway vehicle is the derailment, many studies has been performed in this way, in order to investigate the parameters that influences this phenomena. For these reasons it's important to constantly monitoring the Y and Q forces which are the input of the train mechanical system.

In this section are described three limits that the EN 14363 [1] leaflet consider for the running safety.

2.2.1 Nadal criterion

Despite to complex models that take into account the difference of friction coefficient between the flange and the rolling table or the duration of the wheel climb, the oldest and simplest approach is used as reference to evaluate the safety of the wheelset dynamic.

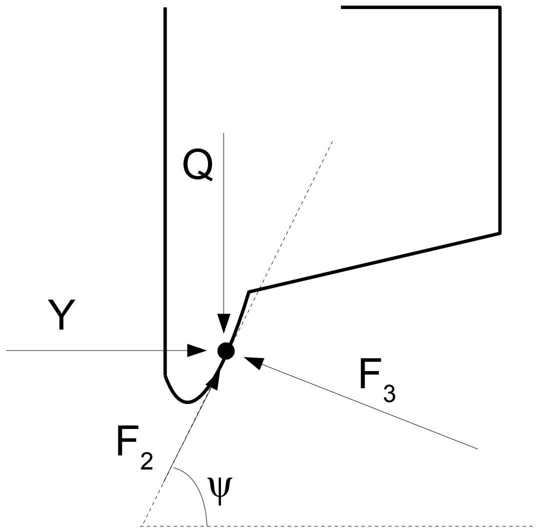


Figure 2.4. Forces at contact point on flange

This approach is defined by Nadal and is based on geometrical consideration considering the wheel profile. Considering the wheel with vertical and lateral forces applied, making the equilibrium at the flange we have:

$$F_3 = Q\cos\psi + Y\sin\psi = Q(\cos\psi + \frac{Y}{Q}\sin\psi)$$

$$F_2 = Q\sin\psi - Y\cos\psi = Q(\sin\psi - \frac{Y}{Q}\cos\psi) \quad \text{when} \quad Q\sin\psi - Y\cos\psi < \mu F_3$$

$$F_2 = \mu F_3 \quad \text{when} \quad Q\sin\psi - Y\cos\psi \geq \mu F_3$$

The ratio **Y/Q** can be expressed as:

$$\frac{Y}{Q} = \frac{\tan\psi - \frac{F_2}{F_3}}{1 + \frac{F_2}{F_3}\tan\psi}$$

in saturated condition the previous relation become the famous Nadal criterion:

$$\frac{Y}{Q} = \frac{\tan\psi - \mu}{1 + \mu\tan\psi} \tag{2.9}$$

The Nadal criteria is known to be conservative for small angles of attack of the wheel and quite realistic for high angle of attack. Figure 2.5 shows the influence of the friction coefficient and the wheel flange angle on the derailment limit.

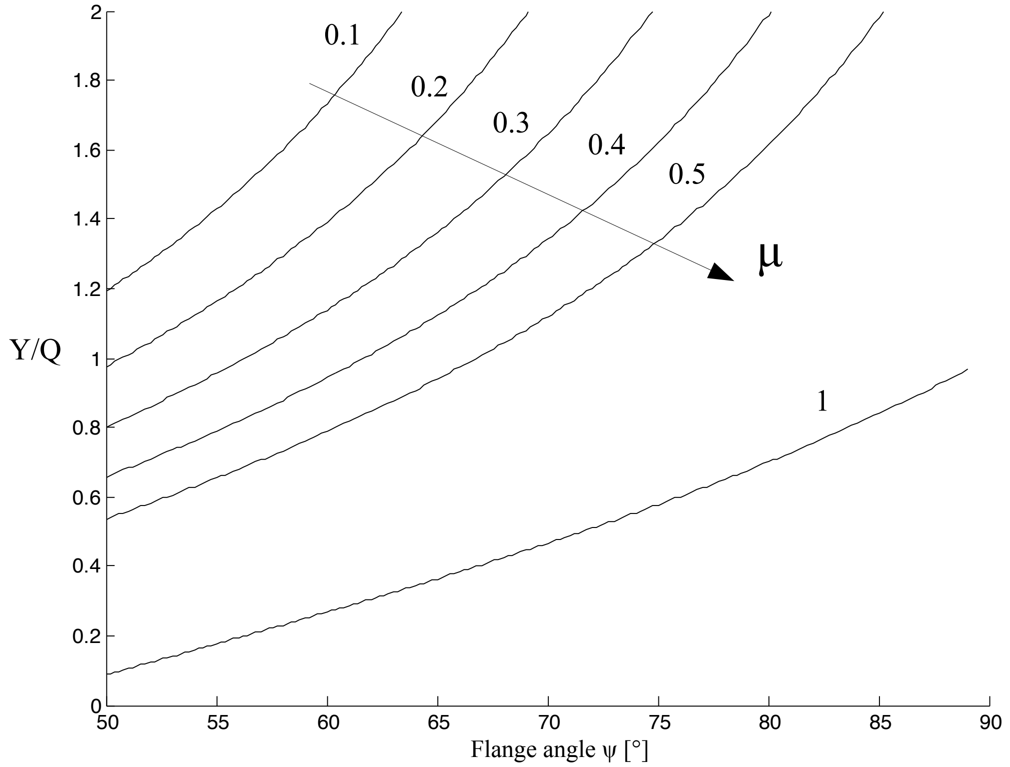


Figure 2.5. Influence of the flange angle and friction coefficient on the Nadal limit

2.2.2 Proud'homme criterion

A possible cause of derailment could be the track strength to lateral loads exerted by the train traffic. If a track is laterally overloaded the panels could shift causing high misalignment that could lead to train derailment. Usually this phenomena is gradual, but if the load limit is exceded a rapid increase of the panel shift occurs depending on the number of repeated load. Considering the elasto-plastic material behaviour, the increment of shift is linear with the load cycles over the limit, because of the elastic plus residual deformation; for load cycles under the limit only the elastic deformation occurs and the residual ones tends to zero. A load limit for lateral forces on the track has been defined by Prud'Homme as following:

$$\sum Y_{max,2m} = k_1(10 + 2 * Q_0/3) \tag{2.10}$$

where k_1 is equal to 0.85 for freight wagons, and to 1 for all the other kinds. The signal $\sum Y$ filtered with a low pass filter of 20 Hz, is obtained with a sliding mean with a window length of 2 m and a step length of 0.5 m.

2.2.3 Instability criterion

The instability of a vehicle bogie could cause derailment due to high lateral forces that the vehicle exerts on the track. The instability appears as a sinusoidal movement of the wheelsets, that is hindered by the wheel conicity with its gravitational recall and by the flange contact. The [1] prescribes a method for the evaluation of the stability condition of the wheelset, observing the *rms* signal of the **Y** forces:

$$\sum Y_{rms} = \frac{\sum_Y max, 2m}{2}$$

This signal is filtered with a band pass filter around the instability frequency $f_0 \pm 2Hz$ (usually supplied by the manufacturer) and obtained with a sliding mean with a window length of 100 m and a step length of 10 m.

The *rms* signal gives the trend of the signal, if it's increasing o decreasing indicating a durable unstable behaviour of the wheelset (figure 2.6).

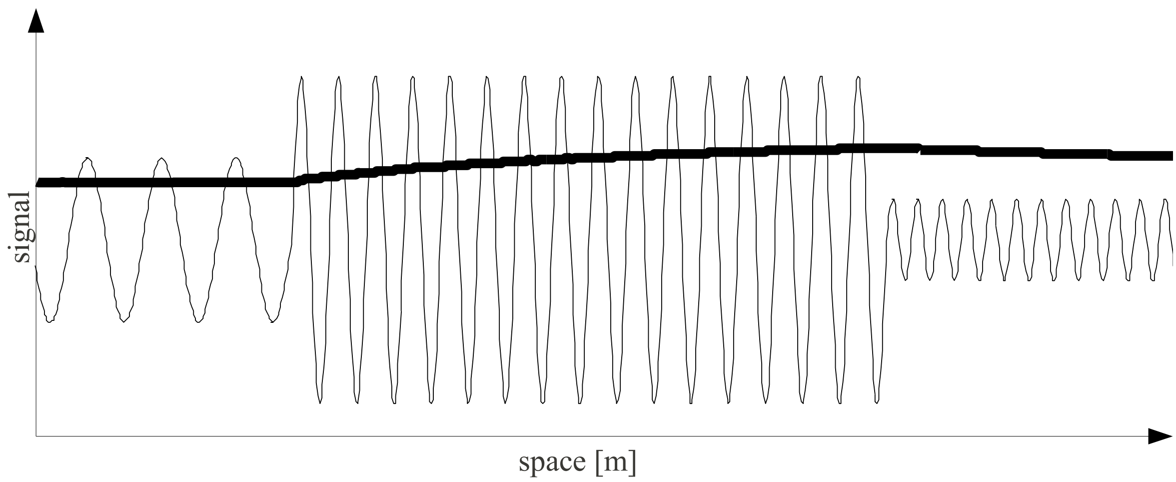


Figure 2.6. Rms example of a signal

2.3 Lateral dynamic

A train vehicle can approach two different kind of track topology, straight track and curve track [7]. For the first type the main issues are related with the running stability, for the second there are three related issues to take into account:

- the mechanical limit, due to force exerted between wheel and rail;
- the physiological limit, due to the maximum, continuous acceleration that people can experience;
- the geometric limit, due to the maximum deformation of the suspensions that determine the maximum gauge of the vehicle.

The geometry of a curved track is characterized by two main quantities, the radius of curvature and the cant. Although for the first quantity not many words should be spent to describe its meaning, for the second some considerations about its physical meaning and influence on the dynamic of the train shall be done.

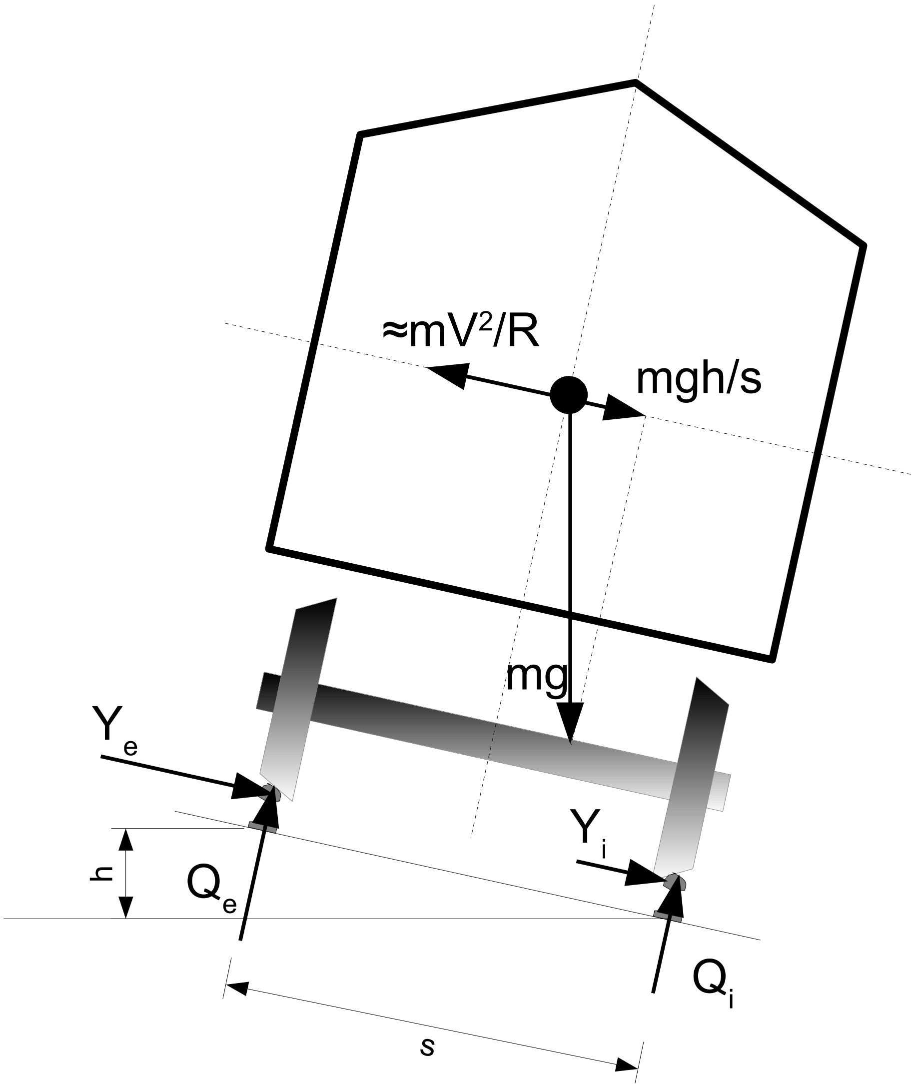


Figure 2.7. Forces on a train on a canted curve

Cant is the height of the external rail in the curve, it's expressed in millimetres, and of course it determine the inclination of the track with respect to the horizontal plane, since the nominal gauge is fixed to 1500 mm.

The presence of superelevation is fundamental for two of the three points listed above, because it lead to a reduction of the acceleration experienced by the passengers and of the forces exchanged between rail and the vehicle.

Making the equilibrium of the forces acting on a vehicle while approaching a curved track, referring to figure 2.7, in the lateral direction:

$$Y_e + Y_i = m(\frac{V^2}{R} - g\frac{h}{s})$$

This indicates that a part of the lateral contact forces are lower than in the case the same track without cant. Moreover, the acceleration experienced by passengers in the lateral direction is equal to:

$$a_{nc} = \frac{V^2}{R} - g\frac{h}{s} \tag{2.11}$$

where with a_{nc} is indicated the "*not compensated acceleration*" by the lateral component of the gravity. Usually leaflet uses the "*cant deficiency*" cd that is, not more, than the a_{nc} expressed through the cant that the rail should have in addition to the actual, in order to compensate all the centrifugal acceleration on the coach.

$$cd = \frac{sV^2}{gR} - h = a_{nc}\frac{g}{s}$$

Usually the limit for the lateral acceleration exerted on the human body is $1m/s^2$, thus the allowed speed of the vehicles on the tracks is related on the cant, the curve radius and the allowed a_{nc}, in addition to the mechanical limit of the vehicle.

2.4 Switches

[6]

Switches (figure 2.8) represent a special kind of track topology since they are necessary for the vehicle to "switch" from a route to another. They can be automatic or manual on the basis of their actuating system. Moreover they introduce a change of stiffness along the track because of the different sleepers spacing and length which can cause different rail stiffness between the left and the right side the vehicle during the switch crossing. The core of the switch could be manufactured with a different material, typically manganese that introduce a singularity in the track stiffness due to the harder material and to the mono-block geometry that increase the bending stiffness (proportional to EI)of the track. This induces an high frequency impact into the rolling stock system. The dynamic of the vehicle is heavily affected by the particular geometry of the switch since when it approaches the frog the wheel-rail contact patch moves laterally in order to find a new equilibrium position, generating a longitudinal force, thus a yaw torque. When the wheelset moves from the wing to the nose rail, the different heigh of the two rails creates a vertical impact. The concomitant presence of both the impacts in such a short length and the steep change of curvature of the rails make the switch a critical route for every vehicle, that must be investigate in order to avoid derailments.

In Italy switches have a specific nomenclature which indicates the nominal weight per unit length, the radius of curvature and the tangent of the angle between the straight and diverging track [8]; for example "*S60 UNI/400/0,094*" means that one metre of rail weights

60 kg, the curve radius is 400 m and the tangent of β is 0.094.

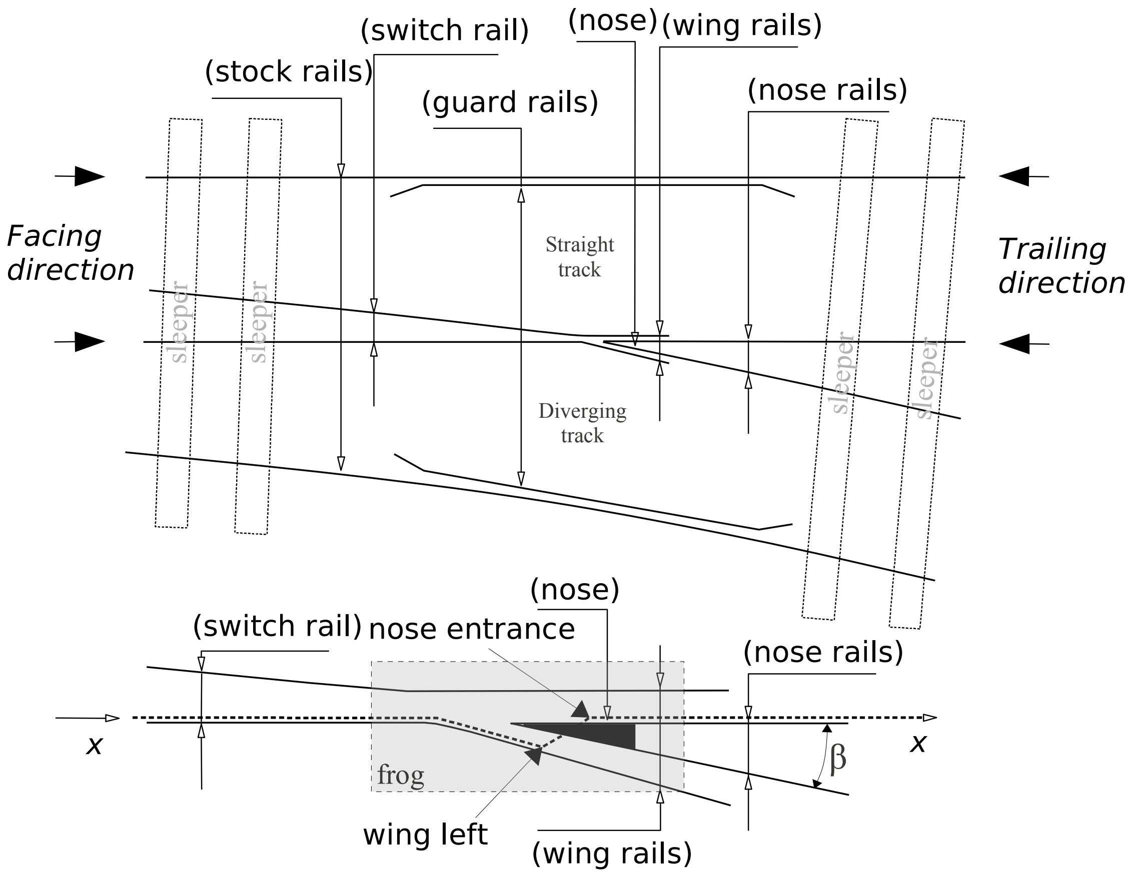


Figure 2.8. Switch topology and nomenclature

CHAPTER 3

DESCRIPTION OF THE VEHICLE

This chapter describes the geometry of the vehicle under investigation, focusing on the main critical parts that characterize its dynamic behaviour in terms of stability, curve approaching and negotiation. This section is included in this work in order to emphasize what are the singular construction solutions that portray this vehicle.

As previously indicated SAADKMS vehicles own very unusual design solutions with respect to standard railway vehicles. Their peculiarities lead to a not well-known dynamic behaviour when the vehicle approaches a curved track, or when it's included in a long composition with other similar vehicles in mixed load conditions, some wagons empty and some others loaded. Their particular design is dictated by their main task that is to involve the conveying of trucks along the railway. Such combined transport have the strong constrain of the limit gauge when running over the line loaded with heavy trucks. Indeed in order to warranty the gauge [9] limit of the lines, SAADKMS vehicles are geared with two bogies, each one with four wheelsets. Wheels have a reduced diameter, 380 mm at factory production, 350 mm at life limit, with respect to typical freight vehicles that have a diameter of about 900 mm . The four wheels are necessary in order to distribute the load of the truck placed on the vehicle and reduce the pressure on each wheelset, this because of the reduced wheel diameter. In figure 3.1 a graph of the trend of the Hertzian [10] contact pressure and area against the wheel diameter is showed.

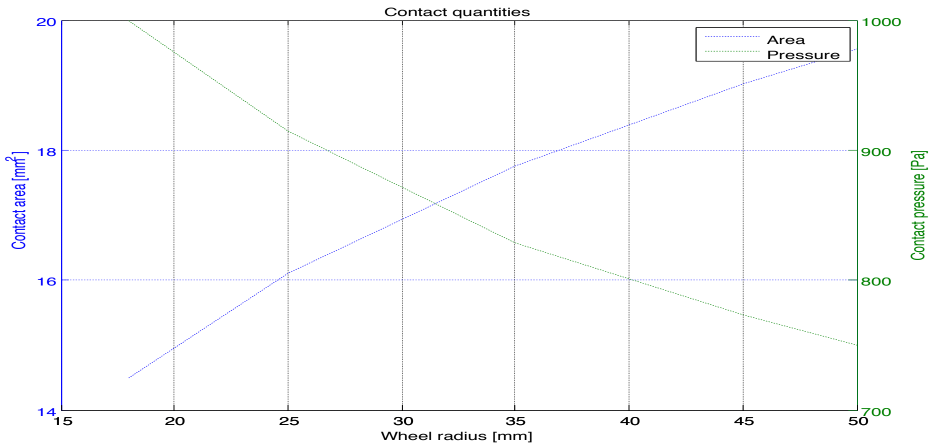


Figure 3.1. Hertzian contact pressure and area

The wheelsets are linked two on two through a semi-bogie by a cylindrical joint on the top of the axlebox (figure 3.6). This permits a little misalignment between the axis of the wheelsets, that improve the steering performances partially freeing the angle of attack of each wheelset. The primary suspensions, that link these two semi-bogies to the main bogie frame, are realized with two toroidal rubber elements. These kind of suspensions are characterized by a very compact geometry, with stiffness and damping capabilities in the same element, but also by a very large hysteresis cycle, that does not permit an easy measurement of the forces exerted through their deformations. Moreover they show a not constant mechanical characteristic in time due to the rubber deterioration.

The main bogie frame has the typical "H" shape with a central spherical joint that realizes the connection with the upper flat-wagon. The contact surfaces are covered with a friction material, that damps the yaw, rolling and pitch motions of the flat-wagon (figure 3.2).

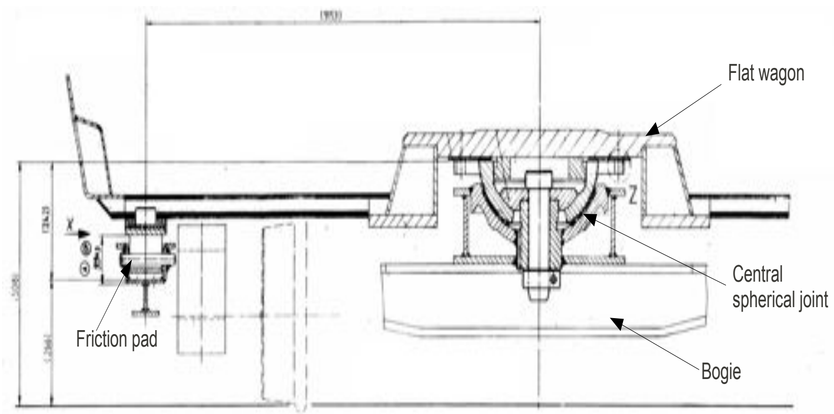


Figure 3.2. Lateral section of the vehicle

Yaw motions are also hindered by two friction pads acting between the bogie frame and the flat-wagon, the contact of these two parts is guaranteed by two spring coils for each

Length bumper-to-bumper with mobile header without mobile header	 19.995 m 18.590 m
Running gear	2 bogies, 4 wheelsets
Wheel distance	750 mm
Semi-bogie distance	700 mm
Bogie base distance	2200 mm
Wheelset base distance	13.500 m
Wheel diameter design diameter minimum diameter	 380 mm 335 mm
Flat wagon height	436 mm
Empty weight with mobile header without mobile header	 18.3 t 17.0 t
Max load	59 t
Max axle load	7.5 t
Breaking mass	43 t
Rango	A
Maximum speed	100 km/h
Minimum track radius	150 m
Limit gauge	UIC 505.1

Table 3.1. Main wagon characteristics

side of the wagon (figure 3.9) As follows by the previous description this kind of vehicle has all the characteristic of a fully suspended wagon but in a very height reduced and light railway car.

In 3.1 are indicated the main characteristics of the wagon.

The next sections describe the geometrical and mechanical characteristics of each part that composes the vehicle including their nominal and measured on the vehicle under exam mechanical characteristics. The particular geometry of the vehicle influences the choices adopted during the planning of the tests layout and logistic because not all the available lines and plants are able to receive these wagons.

3.1 Wheelsets

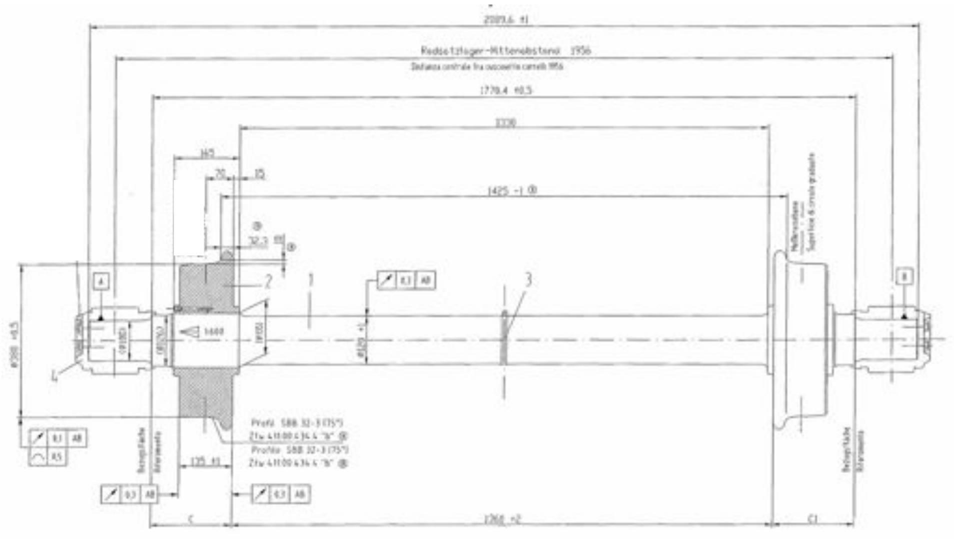
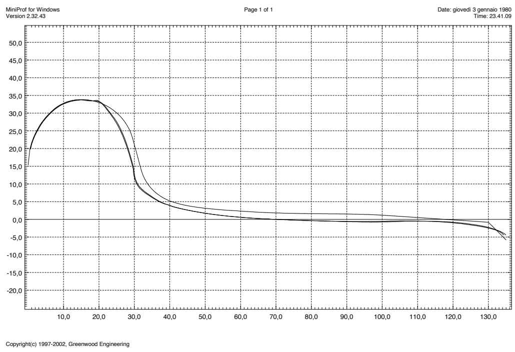


Figure 3.3. Wheelset dimensions

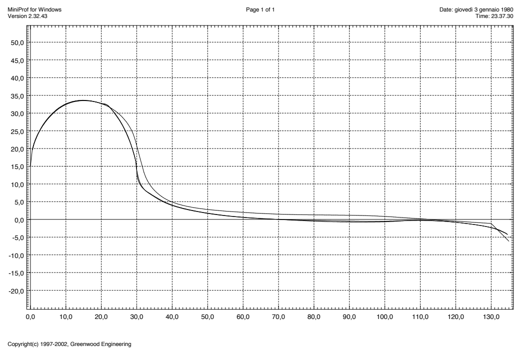
Wheelsets are composed by a full body axle with an external diameter of 120 mm and two wheels mounted through a forced coupling (figure 3.3). As introduced before the wheels diameters are very small and thus, by the hertzian theory, a very high contact pressure acts on the wheels. This phenomenon leads manufacturers to double the number of wheelsets with respect to a typical wagon configuration, in order to distribute the contact pressure on more contact surfaces. The un-wear wheel profile is an *SBB 32-3 75°* where *75°* indicates the slope of the tangent of the internal flange surface.

The instrumented wagon wheels have worn profiles, which are measured with a *miniprof*. After their measure, the wheels profiles of a wheelset are coupled with the wheelset gauge to determine the equivalent conicity (γ_e) of each wheelset that describes the motion wave shape when the straight trajectory is perturbed by a lateral displacement. Many information about the behaviour of a straight and curved track may be achieved by this quantity.

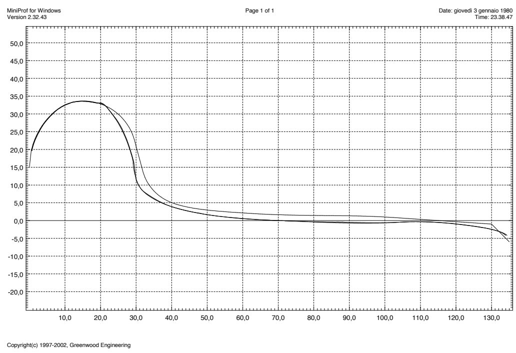
The worn profile of the first wheelset is depicted in figure 3.5(a) and it's clear that the wheel material has been moved toward the flange by the interaction forces between the wheel and the rail, that is a clear consequence of the extremely aggressive behaviour of the vehicle especially during the curve approach. This plastic deformation is lower in the central wheels because they don't ever lead the wagon.



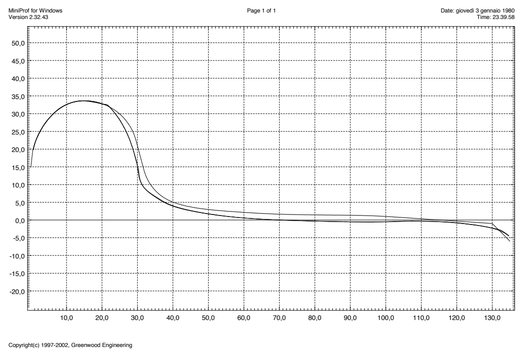
(a) Wheels 11-12



(b) Wheels 21-22



(c) Wheels 31-32

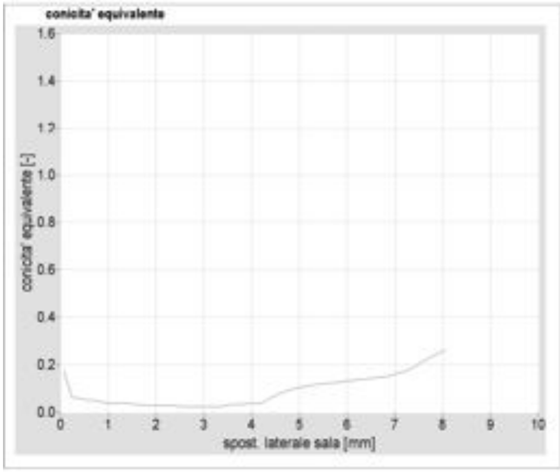


(d) Wheels 41-42

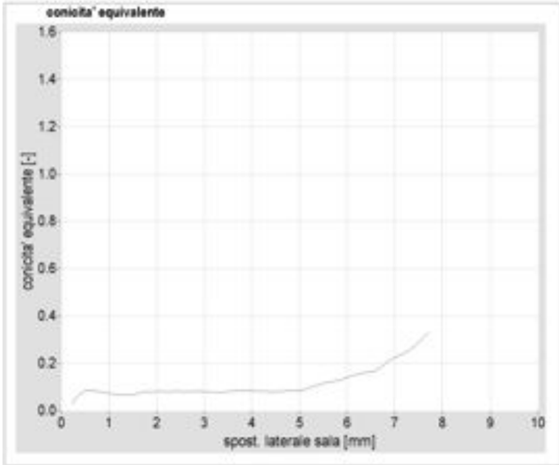
Figure 3.4. Forces estimations of wheelset 1

3.1.1 Equivalent conicity

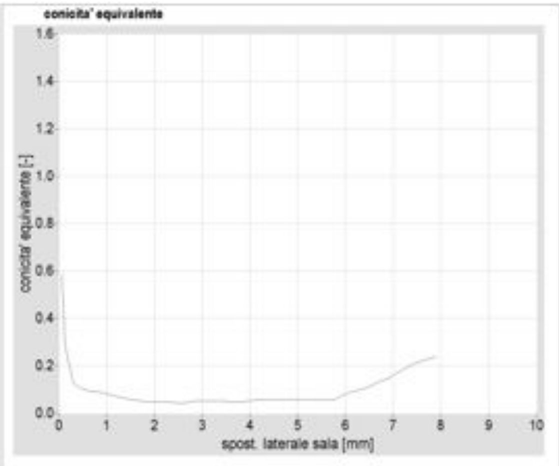
The equivalent conicities of each wheelsets have been calculated coupling the measured profiles and gauges with the nominal one of the UIC60 profile. The values of the equivalent conicity are low although the profiles are wear, but they are representative of the wagons fleet, indeed the instrumented wagon had been chosen on the basis of its wheel profiles between 50 wagons.



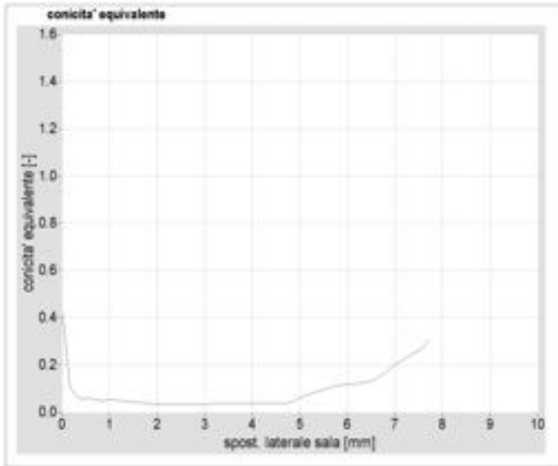
(a) γ_e Wheelset 1



(b) γ_e Wheelset 2



(c) γ_e Wheelset 3



(d) γ_e Wheelset 4

Figure 3.5. Equivalent conicity of the begie

3.2 Axlebox connection

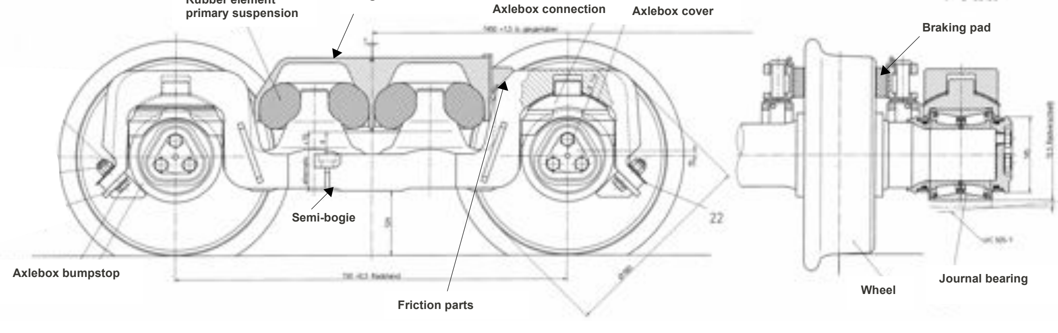


Figure 3.6. Lateral section of the bogie

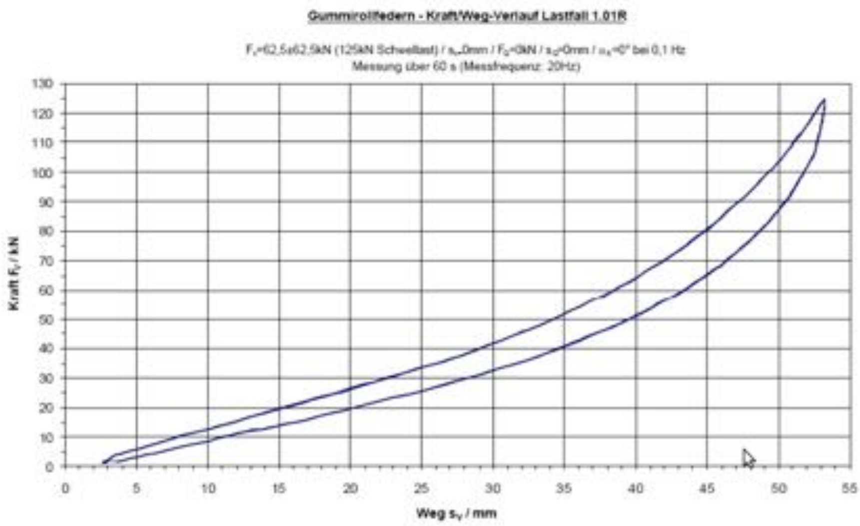
Each axlebox is connected to the semi-bogie through a spherical joint with a central pin that limits the rotations along the lateral and longitudinal axes. This results in such a trapezoidal articulated mechanism that links the wheelsets of a semi-bogie permitting misalignments between the axis of the wheelset giving some steering properties to the vehicle. Moreover the friction surfaces between the semi-bogie and the axleboxes introduces a rotational stiffness and damping that are necessary in order to avoid an excessive movement freedom to the axles that could lead to an unstable behaviour. Each wheelset revolute joint is realized with an opposite conical bearing that is able to equilibrate lateral and radial forces on the wheels.

3.3 Suspensions

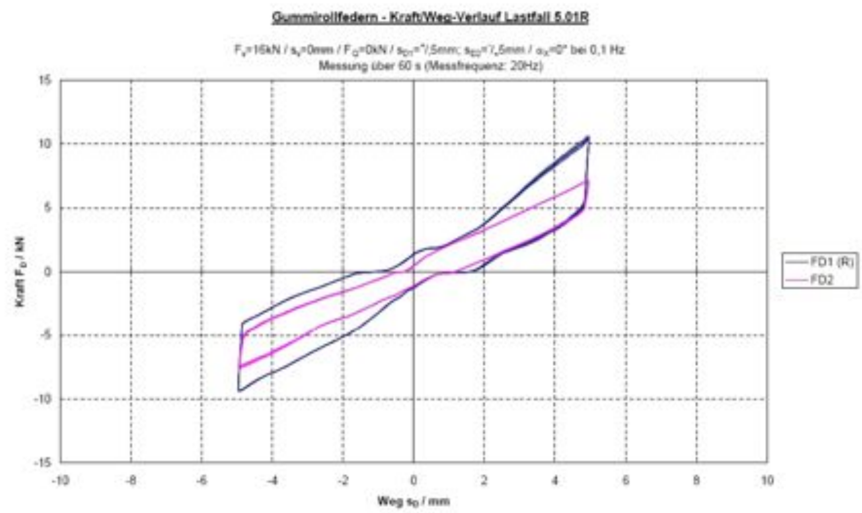
This vehicle has two level of suspensions, which is rather unusual for a freight wagon, but in this case necessary to confer to the vehicle the desired performances of guidance and stability. In freight wagon the secondary suspension is often omitted because of the lack of passengers, thus the "comfort" of the load is not of primary importance. However because of the very small geometry of wheels the stability of the vehicle is aided by the presence of the friction pads and also the vertical load transfer is smoothed by the vertical coil springs.

The primary suspension is realised with rubber elements, with a toroidal section located in the central pins of the semi-bogie. These elements provides both stiffness and damping to the system, with a small space requirement, so they are the perfect candidate for this kind of wagon, that bore with the necessity to reduce its encumbrance.

Otherwise, on the opposite hand, they have the drawback to be highly non-linear, and quite sensitive to the weather and the time life, changing their nominal characteristics, hardening. These kind of elements have a real characteristic that is quite far to be linear as a flex-coil spring, and they must be characterised by laboratory tests in order to know their real behaviour.



(a) Vertical characteristic



(b) Torsional characteristic

Figure 3.7. Mechanical characteristic of the 1st stage of suspension

Some tests have been performed by the manufacturer at different force frequency, and the results present a large hysteresis loop due to the damping characteristic of the rubber. The test are performed on a couple of springs applying the force in the mean position between them.

The test rig for the characterisation of the spring is able to apply vertical (figure 3.7(a)), lateral and a torque forces, and for each component three loops are investigated, one which cover the complete force range, one around the lower part of the complete loop, and one around the upper part at the frequency of 3 Hz.

The lateral force of the spring has been measured in the same way, applying different preloads and performing many hysteresis loop. Obviously the damping characteristic rise with the test frequency increasing the area of the hysteresis loop that indicates the amount of energy dissipated by the system.

The torsional loop is obtained applying a torque on the top of the linkage system of the couple of springs. The hysteresis loop is showed in figure 3.7(b).

3.4 Bogie frame

The bogie frame has a typical "H" shape [11], that differently by usual frames, has a plane geometric extension, not a curved shape like usual. This characteristic is leaded by the low height of its position, that don't need further reductions (figure 3.8). On its center an element that constitutes the connection between the bogie and the flat-wagon for the brake and traction forces, is realised with a semi-spherical friction surface, that allow the rotation in the three direction avoiding the translations. Moreover the bogie frame realizes the support of all the mechanical parts of the wagon such as the braking system.

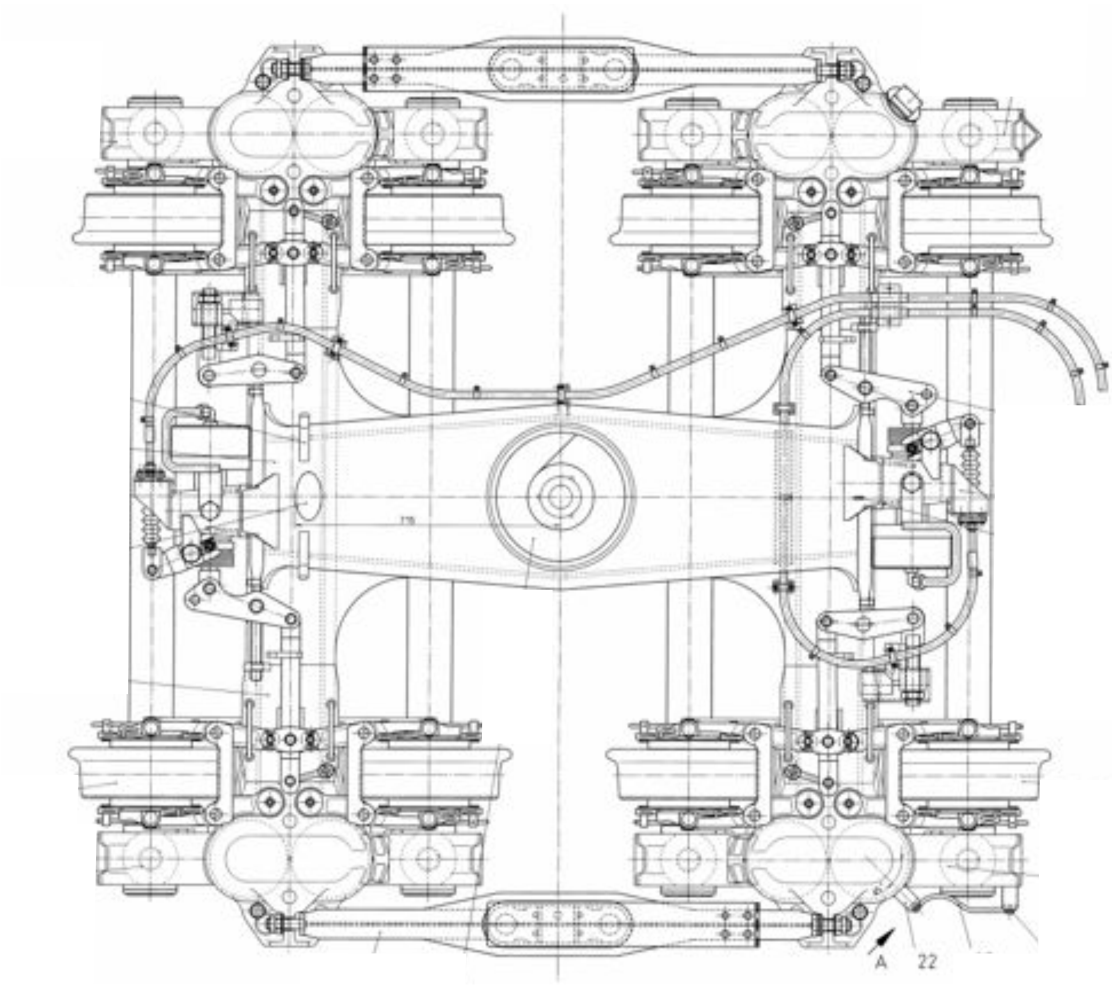


Figure 3.8. Bogie frame over view

3.5 Secondary suspensions

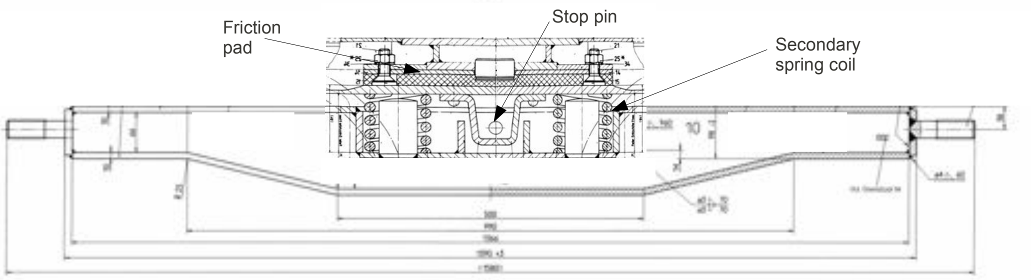


Figure 3.9. Secondary suspensions and friction pads

As previously mentioned the elements that mostly characterize the dynamic behaviour of this wagon are the secondary suspensions. Indeed usually the freight wagon, because of the necessity to reduce costs, for the low speeds and for the lack of necessity to improve comfort are not provided of such level of suspension. These suspensions give to the wagon a torsional stiffness and damping in the yaw direction, since they are realised with two coil springs pushing two plates between them. The coil spring have the function to preload the connected surfaces of the pads, in order to guarantee a friction force also when the wagon is empty. As following consequence of the use of friction elements, it's clear that the friction force increase with the load on the wagon, that leads to a not symmetric behaviour in curve (on side is more loaded then the other), and a behaviour that is quite independent by the relative speed between the pads. Otherwise these elements give to the bogie a better stability along a straight track because they increase the rotational stiffness of the bogie limiting hunting motion. The friction material revealed a highly non constant behaviour, dependant on many factors, the humidity, the temperature, the wear of the friction surfaces etc. For this reason it's very difficult to determine the real behaviour of these elements and it's development in time. Some tests reveal the behaviour of these elements at different boundary condition of temperature and time-life, showing a rapid worsening of the performances. Moreover their characteristic changes with the vertical load on the vehicle, because the central joint equilibrate only a part of the total amount of load on the wagon, transferring the remaining on the pads that increase their preload, thus the tangential force that they exert avoiding yaw motions.

The influence of the mechanical characteristic of these elements have been investigated during the on-track test performed in this work.

3.6 Flat wagon

The flat-wagon is a very simple structure, because it has the task to realize a long "corridor" when coupled with other vehicles, on which the trucks can move finding a location along the convoy. Each flat-wagon don't have the frontal part, and the bumpers are at a lower position then that prescribed in the [4], that is 940-1065 mm from the level of the iron at an horizontal distance between them of 1750 mm. For this reason that wagon can be coupled only with vehicles of the same family. Only the extreme wagons could be geared with a mobile head for the coupling with an usual, interoperable vehicle. The lateral parts of the wagon are the holding ones because they must carry the whole load of the truck on it, moreover the central part between bogies have not to be loaded because it's not designed to carry static loads during the exercise, this consideration leads the load method choice the on-track tests. On the flat wagon there are some block to constrain the trucks on it, and there are some flap for the wheel inspection. In the front part of each wagon there are the bumpers that are at the same level of the wagon floor which, as said, is much lower than standard ones, so they can be coupled only with wagons of the same family. For this reason each wagon can be geared with a mobile-head which is supplied with couplers at the interoperability height and all the pipes of the pneumatic braking system.

3.6.1 Mobile head

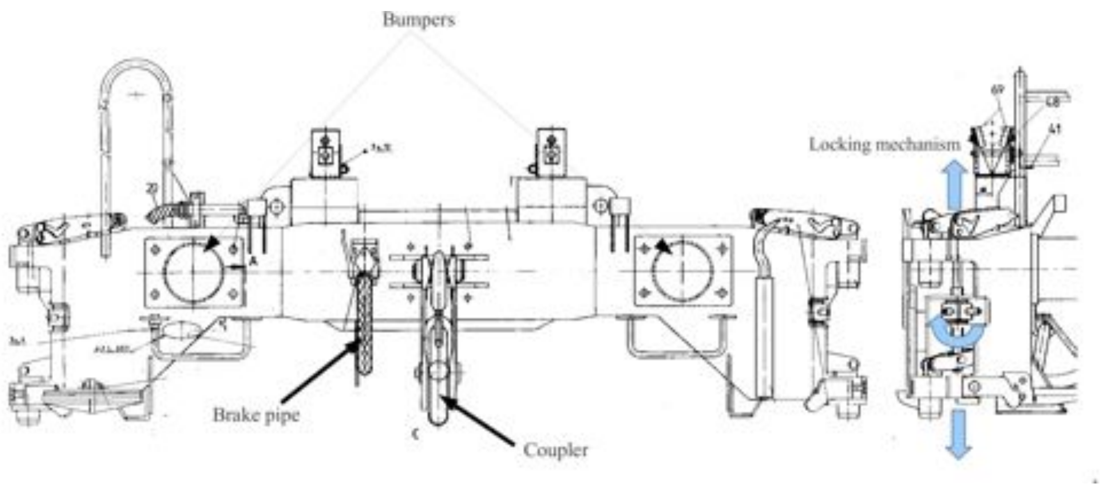


Figure 3.10. Mobile head

The mobile head is necessary to couple SAADKMS vehicles with other, different, vehicles, in deeds it must be mounted on the extreme two wagons and it is geared with bumpers an couplers at the interoperability height. When mounted on the wagon, this part can rotate around one of the lateral hinges to permit the trucks entering in the convoy through an apposite ramp used to climb the vehicles. Once the load is complete, the head must be keep in the closed position and locked by the apposite mechanism. Also the brake pipes terminals are located on this component for the connection with other vehicles. The locking system is realized with a worm drive.

3.6.2 Limit gauge

Each rolling-stock vehicle must respect a certain limit gauge which allows it to run on a certain infrastructure. Each line has a specific code that declare what is its limit gauge and what are the vehicles that can run on it without issues with the profile of galleries and track elements. There are two limit gauges that each vehicle must comply, the upper and the lower limit gauge. For these kind of vehicle the upper limit gauge is delimited by the truck on it, in deeds they can run only in some lines of the north Italy when loaded with the heaviest and biggest trucks. The lower limit gauge, although is not a problem for these vehicles in normal conditions, in our applications it's a real limit because of the telemetry system mounted on the axles, that is very close to the limit, because it exceed the vehicle lower limit gauge allowed. The lower limit gauge for a freight wagon in the central part is at 100 mm over the plane of the iron and the limit gauge for the track is at 80 mm from the same point, so there are about 20 mm of free space where not any part is allowed to be. When the telemetry is mounted this reaches 83 mm from the plane of the iron, although it's not into the infrastructure limit gauge it's very close to it with the risk of telemetry destroying during the tests. For this reason, along all the covered line, a first run with a telemetry simulacrum is done, in order to warranty the telemetry safety during the tests. The lower limit gauge as depicted in the "TSI wagon" is reported in figure 3.11.

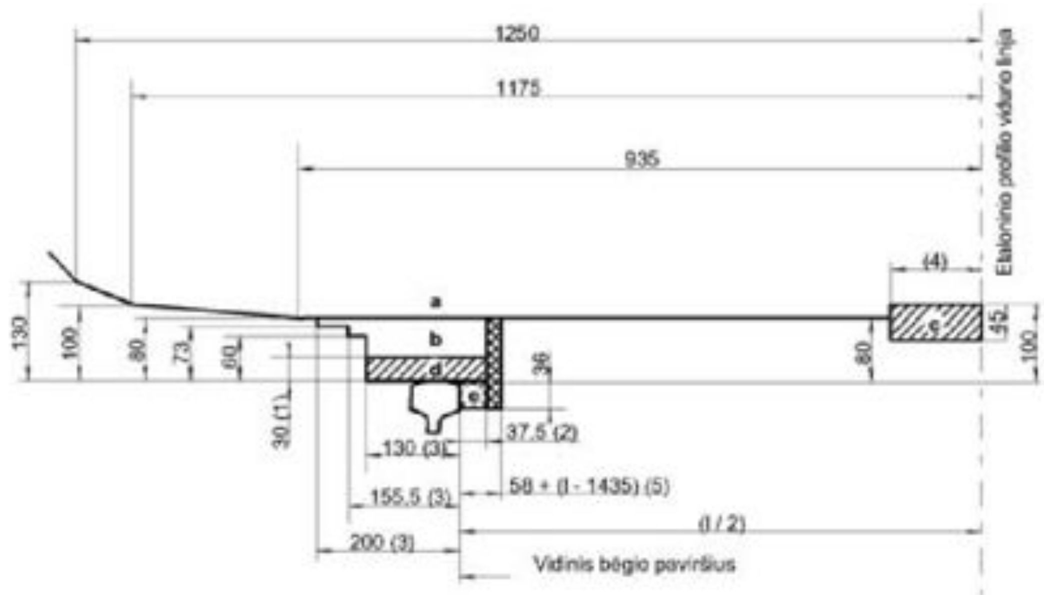


Figure 3.11. Low limit gauge as indicated in the TSI wagon

where the zones indicated with letters are:

a) space for parts far from wheels;

b) space for parts near wheels;

c) space for brushes;

d) space for wheels and parts in contact with binaries;

e) space for wheels;

f) space for parts located outside the wheelsets.

3.7 Braking plant

Braking on these wagons id provided by a pneumatic actuator acting on a cantilever system which apply the braking efforts directly on the solid web of the wheels that are then subject to very high thermal and mechanical loads because of the reduced radius and thus of the high contact force in order to supply the same braking torque as a normal vehicle. During tests the braking system for the measurement wagon is excluded in order to avoid high thermal load on both the instrumentation and the axles. The braking characteristic of a vehicle is given in terms of braking-mass which describes the braking effort percentage in relation to the load on the wagon. This quantity is necessary for trains because trains with a braking mass below 75% are subject to some running restrictions due to the minimum space requisition for stopping during a safety braking. During tests the instrumented bogie is excluded by braking in order to avoid the

3.8 Running conditions

Finally some words about the running conditions of these kind of wagons must be spent, in order to understand what are the restrictions to which it's subject. The circulations rules for a vehicle are released by ANSF that can give a complete circulation admission or a partial circulation depending on the the results of homologation tests and on the documents that assess the behaviour of a vehicle. On the reason of the past derailment these wagons are subject to some limitation for what concern the speed on switches and the traction condition. Moreover till the beginning of 2012 these wagons were considered like special transportation and they request a special document (TES) each time they went in line with some limitations about the maximum speed allowed in line, especially over switches. These limitations where the followings:

- maximum speed is 100 km/h or the maximum allowed by the "Rango A" of the line;
- maximum speed on 30 km/h switches is 10 km/h;
- maximum speed on 60 km/h switches is 30 km/h;
- maximum speed on 100 km/h switches is 60 km/h;

Finally the limitations about the traction and the load conditions, that are that the convoy can only be pulled and not pushed and that, in loaded conditions the extreme wagons must be always loaded.

CHAPTER 4

HISTORICAL REVIEW

In this chapter an overview on the origins of this project and to the motivations that make it came in my hands will be discussed, dealing with the results of previous tests and instrumentation layouts.

Originally this project involves many railway companies because the traffic that this kind of vehicle performs interest many countries around the Italian boundaries, such as Swiss and Germany.

Since the early nineties these wagons called “Rolling roads” are employed by railway companies to perform commercial transport between nations, and they focus on their selves many attentions for their traffic capability and for the possibility to create a strong commercial route with the efficiency of the railway system and the local distribution of the road transportation, with reduction of work hours for drivers, fuel and road occupation. However these kind of wagon didn’t never had a real homologation because of their particular design solutions and heavy differences with respect to the standard vehicles, they had been considered as an extraordinary transport.

The project with the aim to understand the running dynamic of these wagons had origin in the nineties when some derailments occur along the “Domodossola-Brennero” line while the SAADKMS long convoy were routing some switches. A technical commission was instituted to define the guide lines for the investigation of the behaviour of these wagons and some inspections on the location of the derailments were performed in order to understand if some specific event occur at the time of the accidents.

The lines under examination were usually routed by these convoys and their track geometry was modified by the several transits of SAADKMS vehicles and the first evidence observed by technicians was the excessive wear of the track, on the side where the wheel flange contact occurs with the rail as depicted by figure 4.1.

Tracing the origin of the first official document of this activity, the first MOM was about

(a) Track wear 1 (b) Track wear 2

Figure 4.1. Track wear due to the wheel flange contact

a meeting in 11/10/1991 where were introduced the limitations in the TES due to the previous derailment.

Some other meetings had place where Trenitalia, SBB, and RFI discussed about the definition of a measurement system in order to record the contact forces and the running parameters. With the instrumentation techniques available at that time this measures resulted quite difficult leading to many delays in the project.

4.1 2001-2002 activities

A first approach about the measure of the quantities useful to understand the dynamic of the vehicle had been the determination of the axlebox temperature due to lateral forces on bearings along a specific route. These tests had take place on the Airolo line in August 2002 [12] and some critical aspects about the wear of the wheels and rails and the safety of the running emerged.

About the derailment coefficient the results confirm that it's not strictly dependant from the line where they had been measured (Swiss or Italy), and that the axlebox temperature is not affected by the load but it's strictly influenced by the speed of the convoy. Although the derailment occurs only on the Italian railway lines and the main reason had been individuated in the lack of lubrication between flange and rail along the line. This lack is not verified on the Swiss lines because of their maintenance and efficiency. In deeds with an appropriate lubrication an higher Nadal's coefficient could be reached avoiding derailment.

Another particular condition observed on SAADKMS vehicle in service on the Italian lines is the excessive wear and fatigue damages due on the wheel flange caused by the anomalous contact between the flange crest and the top of the rail at the entry of switches because of the modified geometry of the rail shape generated by the heavy flange-rail contact forces as reported in figure 4.1. the other possibility individuated was related to the

crack length > 60mm	
diameter [mm]	samples [%]
335-340	25
340-345	40
345-350	20
350-355	12
355-360	3

Table 4.1. Crack occurrences related to wheel diameter

lack of lubrication on the Italian lines that is the cause of an excessive wear and overheating of the contact surfaces that could start the crack as showed in figure 4.2. As showed the crack generation and then propagation starts on the inner side of the flange where the contact with the rail occurs.

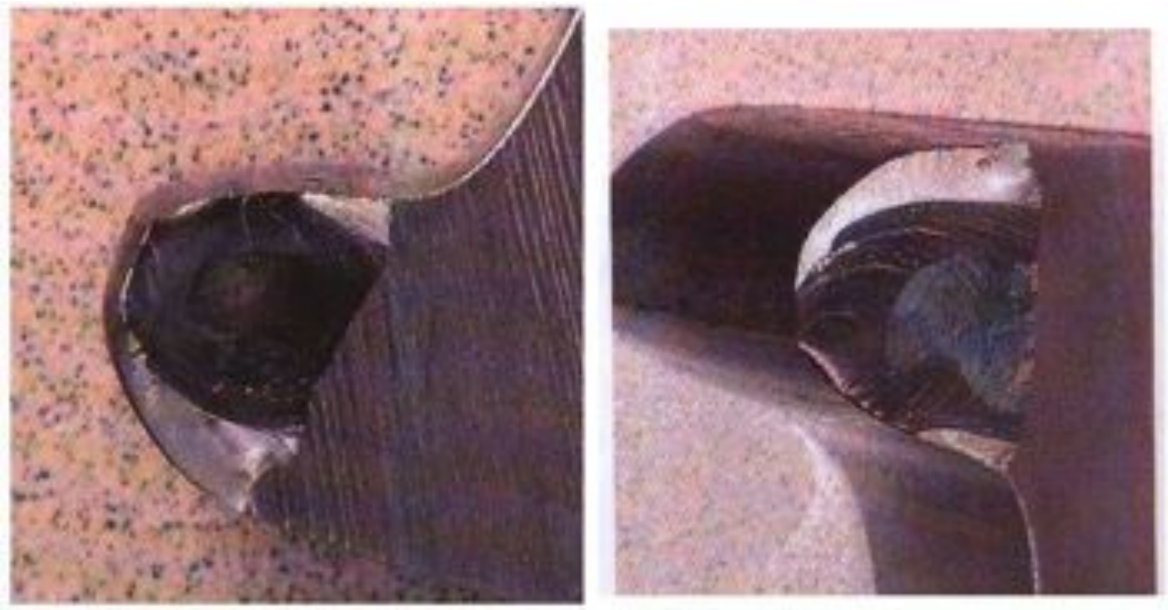


Figure 4.2. Flange wear

Related to this last issue, OBB performed a statistical investigation about the condition of wheels where the presence of crack occurred, obtaining that the cracks start on the inner side of the flange in the 98.6% of the samples. This result is summarized in table 4.1:

These results confirm the the crack entity is strictly related to the wheel life and to the difficulty of heat drainage by the wheels. Moreover all the damaged wheels belonged to wagon in service between the Swiss and Italian lines, thus the systematically use of the pneumatic braking instead of the electrical one on the FS lines was one of the cause of crack propagation in addiction to the poor lubrication of the lines.

4.1.1 Results of the axlebox temperature in line tests

These tests were performed along the Gottardo downhill (from Airolo to Biasca) in order to investigate the influence of braking on the temperature on the mechanical parts.

The instrumentation layout used for these test were composed by thermocouples placed in proximity of the bearing (LU), on the bolts of the axlebox cover (LB), on the axlebox (RU) and in proximity of the braking surface of the wheel. The schematic altimetric summary of the track is reported in figure 4.3, the route was ran in both the directions, downward and upward.

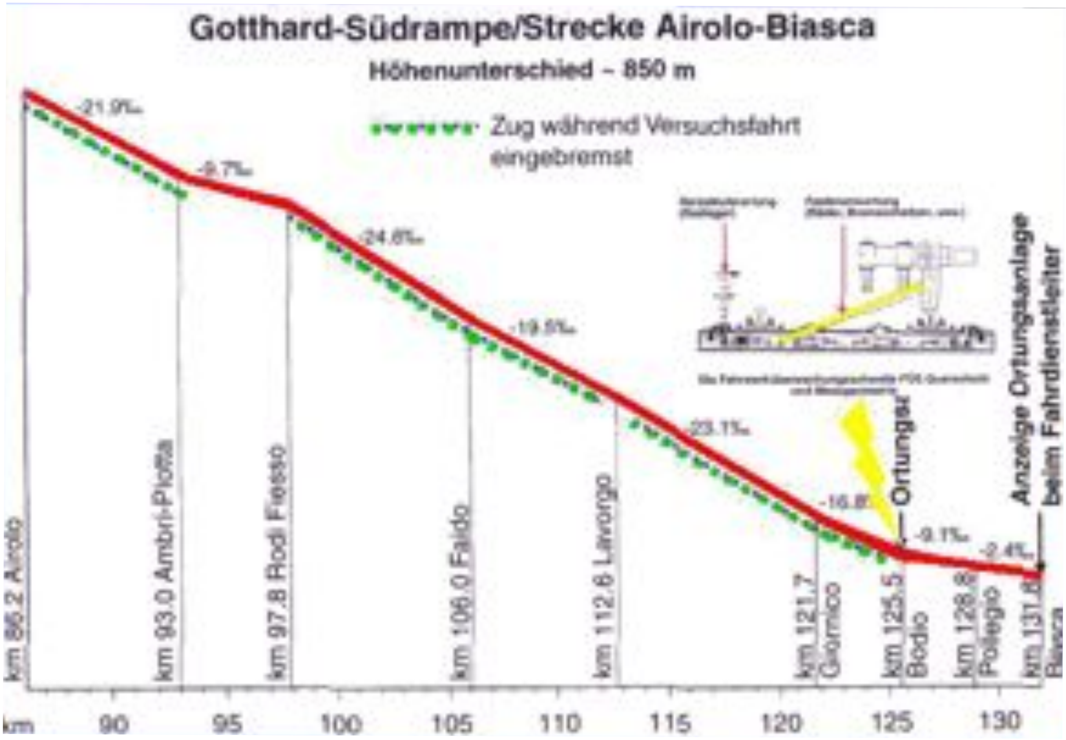


Figure 4.3. Airolo altimetric scheme

By these tests it had observed that the temperature on the axleboxes cover was about 50-55° against the 65-70° on the bearings that is the limit temperature during the exercise for these kind of bearings. Since only the external part is accessible by the ultrasound thermometers this temperature ratio is necessary to establish a limit on the external temperature in order to avoid the compromise of the bearings. Moreover, along the same slope track, the temperatures measured on the wheels were significantly higher, about 110-115° for both the directions because downward the pneumatic braking was completed by the electric braking from locos, reducing the mechanical stresses on the wheels. Another important result had been that the thermal inertia of the system was quite high and the reduction of the temperature once the train stopped was about 1° every 5 minutes, this give the time to the operator to measure with enough precision the temperature of the wheels. After these tests the check of the axleboxes temperature when the vehicle stops had become mandatory on the Italian lines.

4.2 On-track SBB tests

In the early noughties during a work performed by DB, SBB and OeBB in collaboration with Prof. Riessberger of the University of GRAZ [13], the wheel-rail forces where measured (not final report about this is available but only some results) through a measurement system similar to that used in this thesis.

These tests where performed on specific tracks with the aim to measure the derailment coefficient and thus the Y and Q forces while the vehicle was routing some curves. The results of these tests had been the benchmark experimental results used for the validation of the mathematical models realized till now. The technology at that time was not powerful enough for the on-line visualization of the recorded data, thus only some plots of the acquisitions are available with a small drawing scale that induce to a bad evaluation of the exact value of the parameters. An example of such recording data is reported in figure 4.5 and the list of the tests performed are in table 4.2.

(a) Telemetry

(b) Load of the wagon

Figure 4.4. Track wear due to the wheel flange contact

The telemetry was placed into the hollow axles appositely realized for the tests, and all the cables goes inside from the strain-gauges on the axle to the pick-up. In figure 4.4 is showed the instrumentation mounted and the load method of the vehicle, in particular the accelerometers and the telemetry metallic shield applied to protect it by external factors. The instrumented axles are the extremes of each bogie, thus the 1, 4, 5 and 8th. The load was realized with many binaries instead of a real truck. The aim of these test was to measure the behaviour of these kind of vehicles along some different kind of curve track, in particular that reported in table 4.2, measuring all the safety parameters and force components.

These results, till now, have been the only reference to validate the numerical models, involving only a qualitatively validation of the models because the were available only on paper prints with poor quality and small scale. Although some consideration about the wagon behaviour are possible to be done.

From picture 4.5 it's possible to evaluate the vertical and the lateral dynamic of the vehicle, with the vertical load transfer when the curve is approached and the lateral forces

Track	Radius [m]	Speed [km/h]	Superelevation [mm]
Curve 1	300	75	120
Curve 2	350	80	134
Curve 3	360	80	134
Curve 4	590	110	136
Curve 5	600	100	136

Table 4.2. Tracks of experimental SBB tests

exerted on the external wheel. To notice is that since a centralized acquisition system was available at the time of these tests in figure 4.5 only the force parameters are recorded, while probably the accelerations 6.5 measured were recorded by another acquiring system and synchronized with the so called “dash-dot” signal from the odometer.

No further comments are introduced here because a more detailed analysis of the test results is introduced in the “Virtual modelling” 5 chapter.

4.3 From 2002 to 2009

Despite the initial enthusiasm around this activity, because of the particular rolling stock solutions and for the quite brand new and singular solutions of this vehicle, in addition to the interest of many railway institutes of the central Europe, the activities related to SAADKMS stopped till 2004 when Trenitalia commissioned a research activity to CESIFER that was the certification and research Italian institute at that time.

The aim of the research was the same as before: the investigation of the dynamic dealing on switches, so a new team was instituted to face this question. Initially DB was put in charge for the instrumentation of wheelset and for the realization of the static tests on the vehicle, because they were the only railway institute owner of the know how about the instrumentation of the axle instead wheels, that was instead the Trenitalia trademark. The static tests had been performed and the results provided to CESIFER as the contract indicates, but the instrumentation of the wheelsets never took place, and after many years during which the activities proceeded very slowly, CESIFER was reconstituted to ANSF (Agenzia Nazionale per la Sicurezza Ferroviaria) and the “actors” of the project changed again.

The contract was taken over by ITALCERTIFER (ITCF), that should have been a new third party society with the task to asses new and old vehicles and lines, following the whole assessment and verification processes for the ANSF approval.

At that time ITCF didn’t have a laboratory with which perform the tests and the Politecnico di Milano (POLIMI) was charged and accredited as laboratory for the instrumentation of the wheelsets and wagons. Moreover the POLIMI and the University of Florence (UNIFI) was charged to perform some numerical simulations on the running behaviour of SAADKMS vehicles. From this point starts the work performed during this thesis.

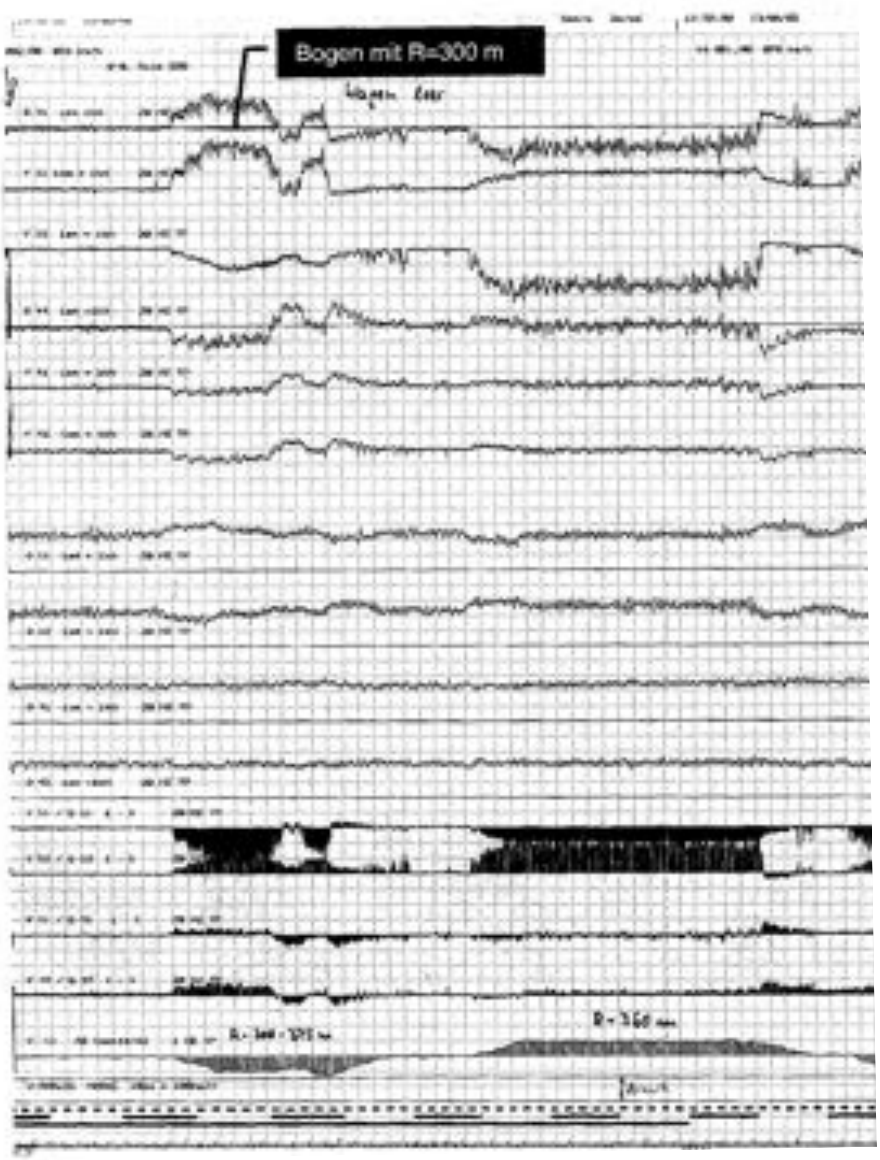


Figure 4.5. SBB test recording example

CHAPTER 5

VIRTUAL MODELLING

In last years the calculation power of PCs has been rised very rapidly, and with it the possibility to realize very accurate dynamic models of each mechanical system. In the railway field there are many multibody software that, through the definition of a dynamic model, are able to reproduce the behaviour of a complex system such as an entire train. It's possible to summarize the function of these software in three main blocks:

- the first block includes the equations that describes the interaction between the multibody model and the external environment (wheel-rail contact);
- the second block concerns equations that describes the interaction between the bodies that compose the mechanical system (train parts);
- the third block contains the post processing tools (signal processing, scope, filtering...)

These software give the possibility to model all the component of a vehicle such as non-linearities, friction components etc. making the final model as more accurate as possible for the description of the real problem.

For what concerns these wagon a virtual model that reproduces the main parts of the vehicle as been realized and, after a first validation based on old experimental results acquired during a previous on-track campaign in Swiss by DB, a set of simulations have been performed for the evaluation of the behaviour of the vehicle on many track layouts and traction configurations.

This had been necessary in order to focus the experimental campaign only on some train configurations because of its cost in terms of time, money and resources employed.

The simulations are performed in two subsequently steps, the first simulates the entire convoy in order to individuate a global behaviour of the train, with a particular attention to the forces exerted by each wagon to the next, in coasting condition and "*fast breaking*" condition. The second step concerns the simulation of a single vehicle with boundary of forces conditions obtained from the previous simulations, aiming to a more accurate evaluation of the local wheel-rail contact interaction.

However SAADKMS vehicles are characterized by small wheels and particular geometries, the modelling of these parts is not critical. The crucial points, rather, are the friction pads of the secondary suspensions, that influences the dynamic of the train both in curve and straight tracks, introducing a non-linear stiffness in the system that is not easy to model and sensible to many variables that heavily affects the dynamic of the entire convoy.

These devices, as mentioned works on the Coulomb principle, due to the friction between the part connected to the bogie and the other connected to the flat-wagon, that dumps the yaw motions. The contact force is guaranteed by two spring coils that confer to the secondary suspension also an anti-roll property.

The reasons that lead the manufacturer (*Bombardier Transportation*) to adopt these kind of components are firstly the necessity to maintain a low and tight gauge of the wagon because of its particular utilization, the second reason is the necessity to combine in an unique component both the secondary suspension and the anti-yaw devices.

The pads behaviour, in terms of total yaw torque, is modelled as a function of the ω_{rel} sign and the vertical loads on the pads (Z_{left} , Z_{right}) (5.1).

$$T = -sgn(\omega_{rel}) * b * (Z_{left} + Z_{right}) \tag{5.1}$$

where b is the distance between each pad and the central joint of the bogie. The function $sgn(\omega_{rel})$ doesn't have a discontinuity in 0, but a slope transition in order to avoid numerical issues; the sloper the transition part, the better the coulombian friction approximation is [14].

Thus ,this kind of model represents a good description of the coulombian friction in transitory, but it has some limitations when observing the steady state behaviour (full curve). These limitations mainly derive by the fact that at the end of the transition track, the force T gradually goes to zero, because there are no force component that works in opposition to this torque, then the bogie tends to return in the central position and a strange behaviour of the lateral contact forces could be observed. So the full curve running behaviour is not to be considered. Finally, in the real wagon there are non component that confer yaw stiffness to the train, and the high aggressive dynamic is to attribute mainly at the presence of the friction pads.

Before accepting the results obtained by a virtual model it has to be validated on the basis of some experimental results, in order to tune some parameters matching the behaviour of the model with the real vehicle. For this application some earlier experimental results are available, obtained during some tests in a previous on track campaign. In the next section some of the validation results are going to be presented to keep confidence about the results obtained by simulations on other kind of tracks useful for the target investigation.

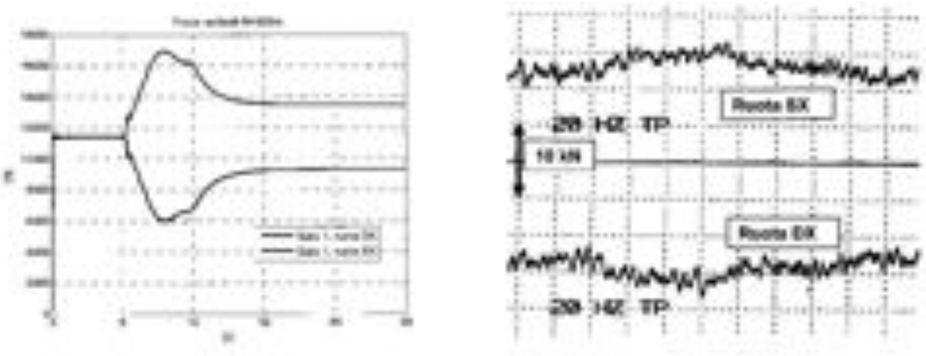
5.1 Validation of the multibody model

[15]

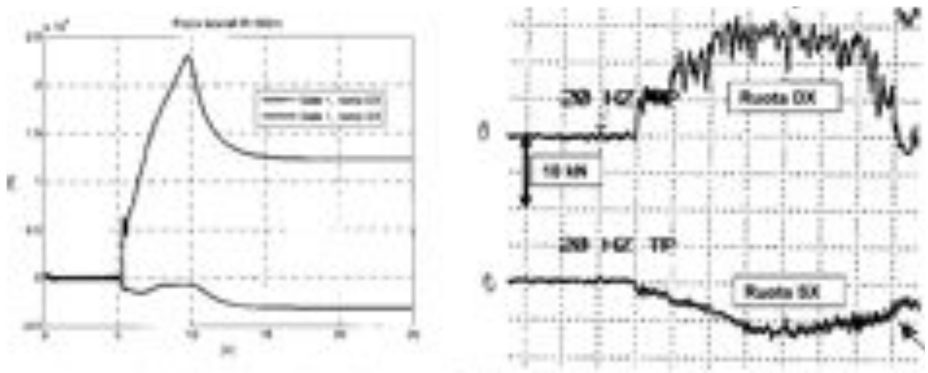
The data available for the validation of the model is relative to some curve routing, although the values of the measurements are not of easy evaluation because of the poor quality of the recording support that was a "continuous paper printer". Some results are showed in order to keep valid the results of the next simulations.

These results are presented with a few description about the behaviour of the vehicle, but they should be useful to verify that the model reproduces the physic of the wagon.

5.1.1 Curve R=300 m, V= 75 km/h



(a) Q forces, WS_1 simulation and record



(b) Y forces, WS_1 simulation and record

Figure 5.1. R=300 m curve, Q and Y forces validation

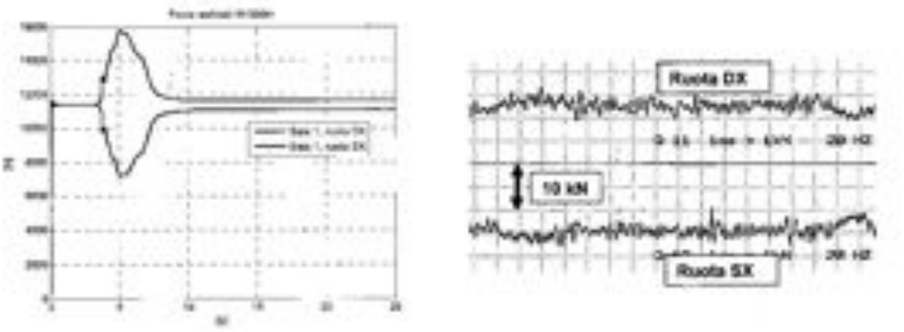
The track is composed by an initial transition and a full curve stretch. The real transition geometry is unknown, thus a typical geometry has been modelled with a clothoids curvature for what concern the planimetric transition and a linear superelevation for what concern the altimetric transition. Only the Y and Q force values along the curve should be considered for a direct comparison.

Also in this case a lack of informations about the transition stretch occurs and the same method for the geometry generation has been utilized.

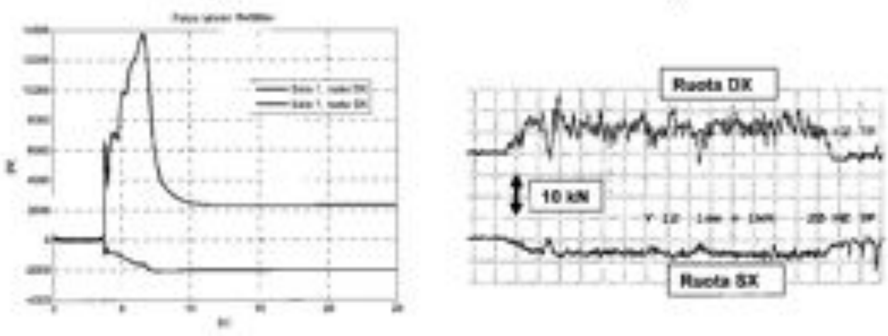
From picture 5.1 the external wheel of the first wheelset reaches a value of about 13 kN while the internal of about 3 kN. From the test records it's possible to observe a similar behaviour but the values reached are a little bit higher. For what concern the vertical forces the load transfer is quite symmetric equal to about 3 kN; also in this case the model reproduces a similar trend of the lateral forces.

5.1.2 Curve R=590 m, V= 110 km/h

Also in this case a lack of informations about the transition stretch occurs and the same method for the geometry generation has been utilized.



(a) Q forces, WS_1 simulation and record



(b) Y forces, WS_1 simulation and record

Figure 5.2. R=590 m curve, Q and Y forces validation

From picture 5.2 the external wheel of the first wheelset reaches a value of about 4.5 kN while the internal of about 2 kN. From the test records it's possible to observe a similar behaviour but the value reached are a little bit higher. For what concern the vertical forces the load transfer is quite symmetric equal to about 1 kN; again, the model understimate the forces values of about 1-1.5 kN, but this mismatch could be attributed to the measure uncertainty and to the difficulty to model all the friction parts present in the vehicle that determines it's particular behaviour.

Despite these mismatches the model has to be considered validated, indeed it's able to reproduce the behaviour recorded on-track. Finally it would be be useless try to find a better quantitative matching between the results because of the uncertainty of the recording condition that could lead to a not correct definition of some parameters.

5.2 Curve simulations

Once a model is validated some simulation on different kind of track can be performed, individuating what are the most critical for the dynamic of the vehicle, focusing the experimental investigation on them. Some results of simulations about these curves and switches tracks are following explained, in order to foreseen the attended behaviour during the on-track tests.

During a curve approach there are some considerations to do about the general behaviour of a train. In fact depending on the wheel-rail forces exerted at the contact, the wheelsets can assume a certain configuration that determines the capacity of the train to have a good curve inscription or not. Typically the first wheelset of each bogie, and in particular the first of each wagon, is the more critical because it's the first that approaches the curve and then the guiding wheelset that brings/experiences the bigger lateral force amount of the bogie. Such force depends on the angle of attack of the wheel with respect to the rail tangent ϕ, and on the flange angle at the contact point β.

The total lateral force Y is composed by two parts as the follow equation 5.2 shows:

$$\begin{aligned} Y &= Y_{wf} - Y_R \\ \text{with} Y_R &= Zf\left(\frac{\phi_T}{\phi} + 1\right)\frac{a}{R} \end{aligned} \tag{5.2}$$

where Y_{wf} is the lateral force on the wheel flange and Y_R is the lateral creep force on the rolling surface.

Thus when the flange goes in contact with the rail the lateral forces rise steeply, increasing the Y/Q ratio. Usually the flange contact occurs when a vehicle routes a curve, but it should be followed by a vertical load transfer due to the centrifugal force and the superelevation of the rail. The simulations show how this wagon has a quite constant behaviour during a curve approach indeed it presents the same behaviour trend while routing a curve with different radius.

Because of the presence of the friction pads the wheelsets present some difficulties during the curve inscription due to an high lateral force on the wheel-rail interface necessary for the bogie/flat-wagon rotation. This leads to an high derailment coefficient (Y/Q). Fortunately, in simulation there are some parameters that can be take under control useful for the derailment evolution monitoring.

Since all the simulated curves are right-curves, the wheels with the numeration i_1, where i is the wheelset number, are the inner wheels, and the i_2 wheels are the outer wheels as showed in figure 5.3.

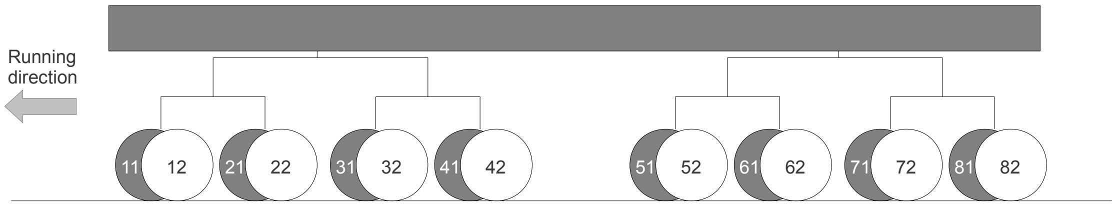


Figure 5.3. Wheel numbering convention

R [m]	170	250	400	600
sup [mm]	140	150	150	160
racc [m]	56	60	60	80
$a_{nc}= -0.6m/s^2$	30 km/h	37 km/h	47 km/h	62 km/h
$a_{nc}= 0.6m/s^2$	60 km/h	73 km/h	92 km/h	115 km/h
$a_{nc}= 0.85m/s^2$	64 km/h	78 km/h	99 km/h	123 km/h

Table 5.1. Curve track

The influence of the a_{nc} is evaluated simulating three running velocities on each track type, that lead to a centripetal, low positive and high positive a_{nc}.

All the scenarios simulated are indicated in table 5.1.

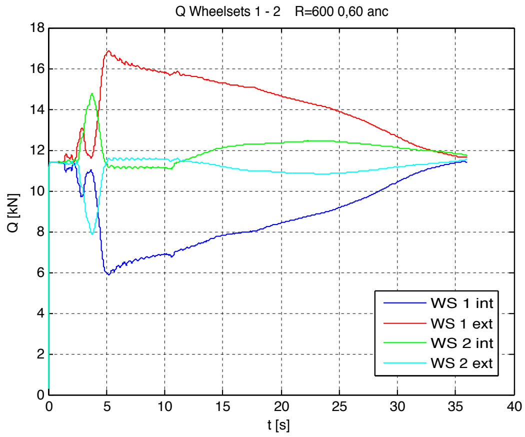
However only some results about simulations in curved track are showed with the aim to individuate the general behaviour of these vehicles, reserving a particular attention to the values of some parameters calculated.

Previously some remarks are necessary about the influence of the traction mode, indeed there are not consistent difference when the convoy is pulled or pushed, either when the rapid braking is simulated because the reduced length of the convoy doesn't introduce such delay in the pneumatic wave to induce to significant longitudinal dynamic effects. At the light of these facts different conditions are showed and described to see the local effects of the simulation conditions.

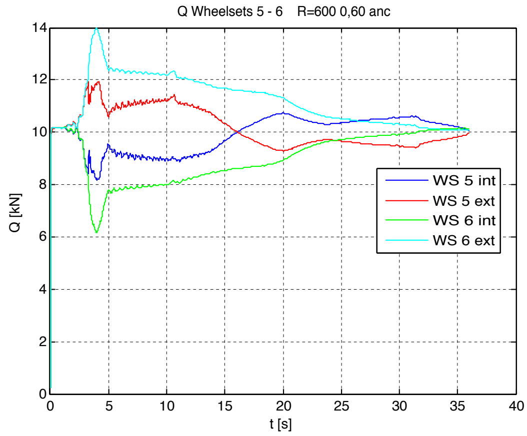
5.2.1 Curve - R=600 m, a_{nc}=0.6 m/s^2, braking

The calculated quantities in these simulations are the vertical and lateral forces on the first two wheelsets of each bogie and the related derailment coefficient [14].

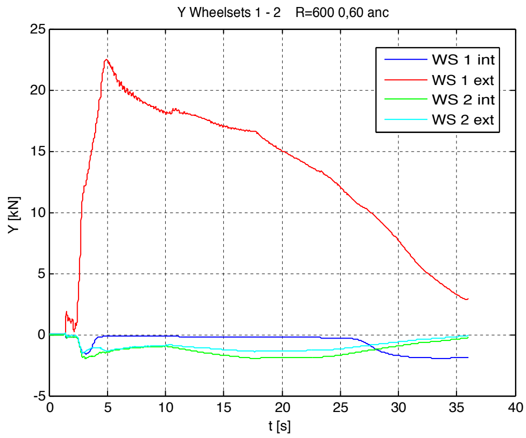
In figure 5.4(a) and 5.4(b) the vertical load on the wheels indicates the load repartition of the vehicle due to the suspension deformation for the compensation of lateral acceleration. The first wheelset shows an higher vertical load, due to the presence of the mobile head. In addition at the curve entrance the external wheel loads increase due to the a_{nc} that generates a roll moment on the flat-wagon. At about 10 s the braking occurs and the curve gradually become over compensated, causing an opposite transfer of load ($approx$35 s). Observing the lateral forces (figure 5.4(c),5.4(d)) it's clear that the external wheel 12 reaches the flange contact while entering the curve, and the (internal) wheel 51 flange goes in contact on the second bogie denoting an over steering behaviour of this bogie. The de-



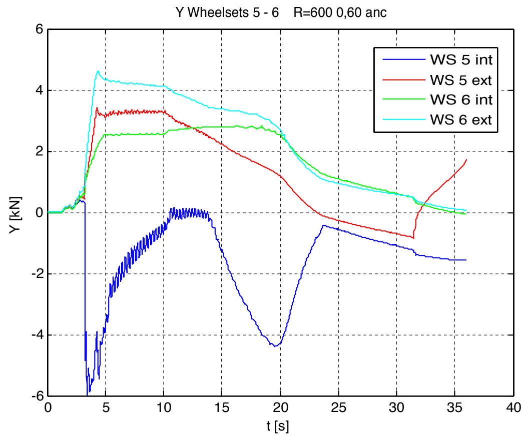
(a) Q forces, $WS_{1,2}$



(b) Q forces, $WS_{5,6}$



(c) Y forces, $WS_{1,2}$



(d) Y forces, $WS_{5,6}$

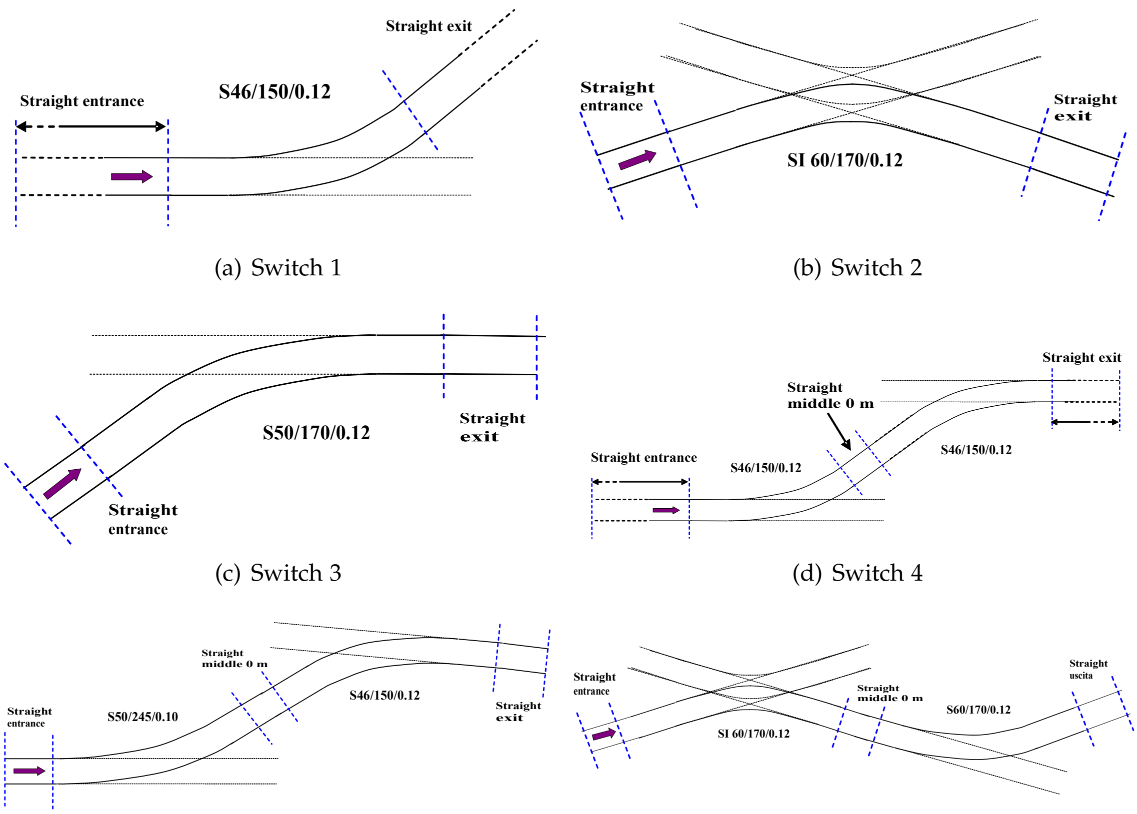
Figure 5.4. R=600 m curve, Q and Y forces

railment coefficients are respectively about 1.39 and 0.7 for the wheels 12 and 51. Using the Nadal's criterion in this case, whit $\mu = 0.35$ and $\tan\gamma = 2.6(6858')$ the limit should be 1.2. The derailment doens't occur and the relative displacement between the wheelset and the rail is constant, preventing the derailment attitude of the vehicle. As known the Nadal's criterion is really conservative because it considers that all the adhesion saturates in the tangential direction, without considering the longitudinal creep forces.

The curve routing do't really represent a dangerous scenario for this kind of vehicle, but these simulations are preparatory for the switch simulations.

5.2.2 Switches

In this analysis only the empty condition has been investigated, although the load on the vehicles changes significantly the mechanical characteristic of the friction pads. This decision had been leaded by the reason that when the vehicle is loaded the central joint equilibrates only a part of the total amount of the vertical load, increasing rather the preload on the lateral pads, which in this new condition can exert an higher yaw torque of about three times than in the empty condition. This improves the stability of the vehicle at high speeds, but worsening its "aggressive behaviour" in curve. Moreover the full-load condition is less critical than the empty one because the higher vertical component on the wheels determines a lower Y/Q coefficient due to the smaller arise of the later forces than verticals [15]. Under these premises the geometries of the analysed switches are reported in figure 5.5 and the results for the switches number 2 and 4 are showed because of their major criticality due to the geometry of the double switch and that of the double switch.



(a) Switch 1 (b) Switch 2

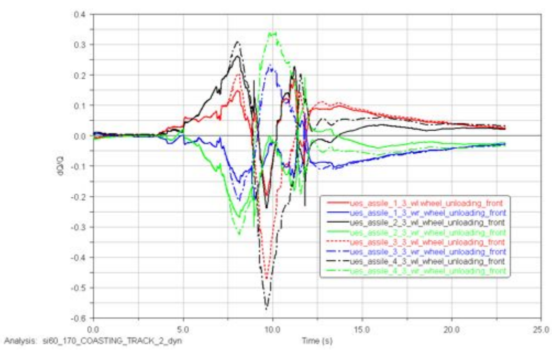
(c) Switch 3 (d) Switch 4

(e) Switch 5 (f) Switch 6

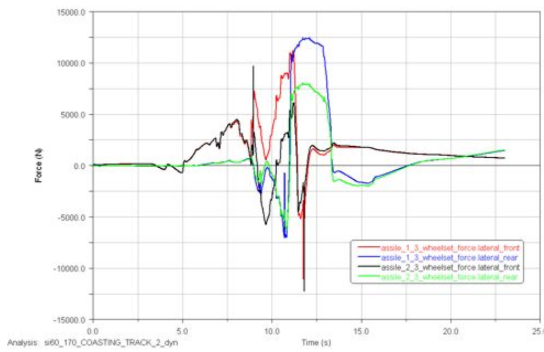
Figure 5.5. Switches geometry

5.2.2.1 Switch 2 -30 km/h

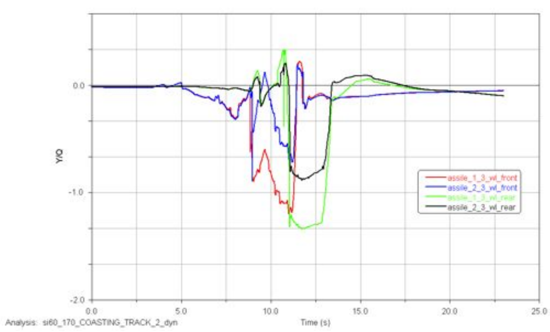
This type of switch is the so called "english switch". It has been modelled without any curvature transition, in order to reproduce a more real track geometry, introducing a curvature discontinuity in the topology of the track. This steep change generates an impact at the entrance of the switch, but for its short duration in space, it's not safety critical, because the inertia of the train tends to attenuate the transitory generated. After the entrance the Y/Q ratio on the first external wheel tends to increase till 0.75, because of the rise of the Y force and the quite small vertical load transfer due to the unload condition. The first wheel of the second bogie presents a similar behaviour reaching a lower value of the derailment coefficient. All these quantities are processed with a 20 Hz, 6^{th} order low pass filter, as prescribed by the EN14363. The vehicle reaches a value of Y/Q coefficient near 1.05 that is enough to considerate this track geometry critical (figure 5.6(c)). The simulation results in braking condition shows that the braking efforts don't generate any significant change in the forces exchanged through the contact interfaces, because of the short length of the convoy that doesn't introduce such a relevant delay in the pressure propagation through braking pipes to influence the dynamic of the vehicle. Moreover when the speed reduces for the braking the lateral forces that compensates the lateral acceleration fall down while the vertical load transfer is reduced without a significant variation of the derailment coefficient. Some other informations can be achieved by these simulations, because of the possibility to calculate more quantities than in the reality as the angle of attack of all the wheelsets. These measures shows the steering capabilities of the wheelsets, and in figure 5.6(d) is reported its trend. Its negative value means that the external wheels are rotated backward with respect to the kinematic steering configuration that is the proof of the understeering behaviour of the first bogie, as said, due to the presence of the friction pads that hinder the rotation when the equilibrium position is reached. In conclusion this kind of track reveals critical for these vehicle because of the high curvature radius and the lack of superelevation. A detailed experimental investigation shall be performed on this kind of track.



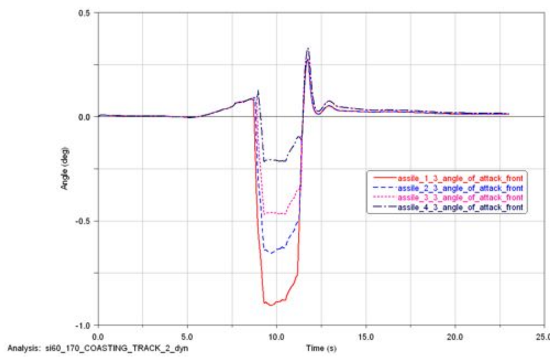
(a) $\Delta Q/Q$ unload, $WS_{1,2}$



(b) Y forces, $WS_{1,2}$



(c) Y/Q ratio, $WS_{1,2}$

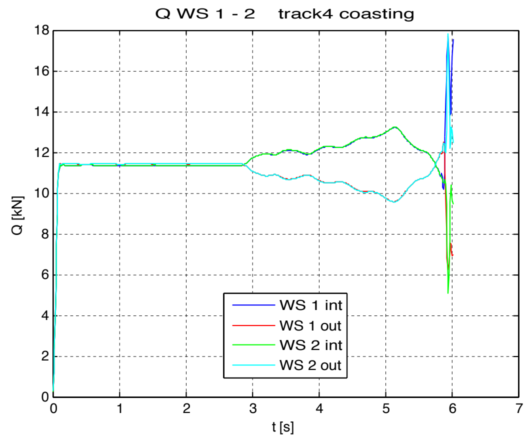


(d) angle of attack, $WS_{1,2}$

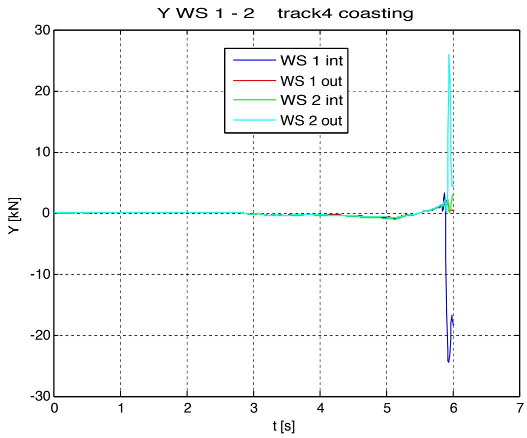
Figure 5.6. Safety parameters, Switch 1

5.2.2.2 Switch 4

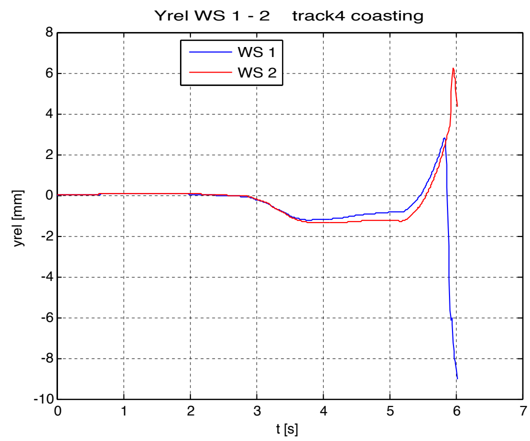
Another critical switch is the 4th, which presents a double curve, one opposite to the other, without a straight transition between them. The curvature radii are the same, 170 m, with a tangent of 0.12.



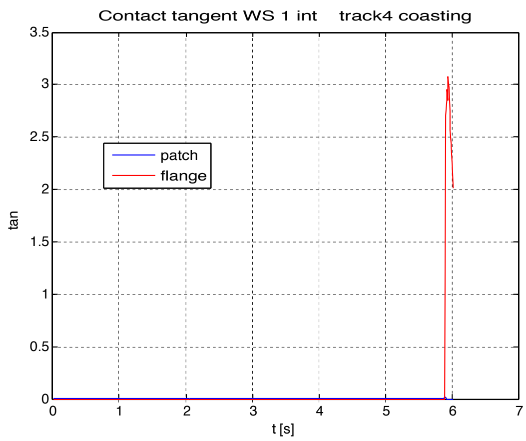
(a) Q forces, $WS_{1,2}$



(b) Y forces, $WS_{1,2}$



(c) lateral displacement, $WS_{1,2}$

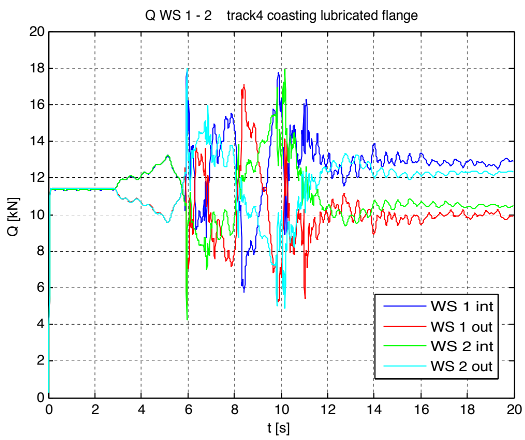


(d) contact angle, $WS_{1,2}$

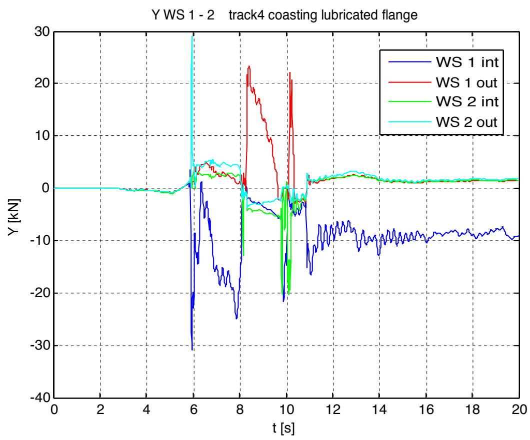
Figure 5.7. Safety parameters, Switch 4

In figure 5.7 some quantities are plotted, where Y, Q and the kinematic parameters are showed to understand the virtual behaviour of the first axle of the wagon. In this case the first two wheelsets of the forward bogie experience a gradual increase of the vertical force due to the increasingly load transfer along the wagon, due to the presence of the frontal coach ($\approx 3s$). When the second bogie enters the curve the first begins the approach to the second curve and the gradient of the vertical forces is very steep with a change of sign ($\approx 5.7s$). At the same time the lateral forces on the external wheel steeply increase for the wheel-flange contact reaching, generating an Y/Q coefficient of about 1.5 causing the flange climbing. Observing the kinematic quantities of the wheelset this is clear, because the lateral displacement increases continuously exceeding the allowed lateral clearance between the wheel flange and the rail, and also the contact tangent increases till a maximum from which starts decreasing: this means that the flange crest is crossed.

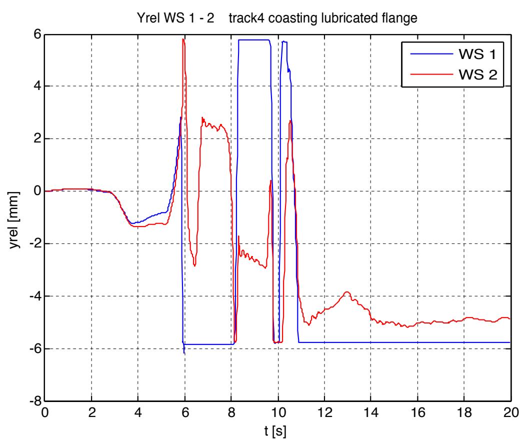
Performing the same simulation with a different friction coefficient on the flange, the first part of the switch is again the more critical, but the lower friction coefficient (0.2) makes the vehicle pass along the switch without the derailment occurs (figures 5.8). In this case the kinematic quantities of the first wheelset shows that the lateral displacement and the contact tangent remain constant, even if at high values, showing the absence of wheel flange climb attitude.



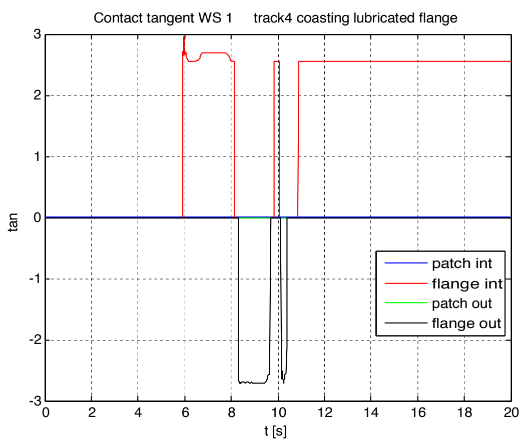
(a) Q forces, $WS_{1,2}$



(b) Y forces, $WS_{1,2}$



(c) lateral displacement, $WS_{1,2}$



(d) contact angle, $WS_{1,2}$

Figure 5.8. Safety parameters, Switch 4-lubricated wheel flange

These simulations have been necessary for the delineation of the on-line tests layout, in order to focus the campaign only on some switch configurations, investigating different conditions such as different friction coefficients at the contact and at the friction pads, and with the train pulled and pushed.

Once individuated the switch scenarios, the main issue remains the concrete location for the tests, that shall present an adequate variation of switches and an opportune availability of binaries for the execution of the tests.

5.3 Conclusions

These simulations are the prelude to the on-track tests, useful to understand what are the safety values that could be achieved on-line without compromising the safety of the tests. Moreover these results are the basis for the selection of the better track configuration, optimizing both the time and the logistic costs. For example the braking conditions for the short convoy will not be investigated because the results show that in all the configuration of track and traction position, the braking efforts are not so consistent to affect the dynamic of the convoy for the limited number of vehicles. In fact because of the short length of the convoy it does not introduce significant delay in the propagation of the pressure wave along the braking pipes. This avoid the generation of the "accordion effect" along the train [16] that might significantly affect the long train dynamic during the service. Moreover for this convoy configuration the difference between pulled and pushed traction is not very consistent, in reason of the limited number of vehicles and the homogeneous load of the vehicles, but this configuration is going to be tested in order to achieve more informations on the behaviour of the train. In the next chapter the experimental results are going to be presented with the main focus on the tests on switches and on the reasons that lead to the definition of such tests layout.

CHAPTER 6

INSTRUMENTATION OF THE WAGON

In a generic mechanical system, in order to determine its dynamic behaviour, it is necessary the knowledge of the equation that links the system degrees of freedom to the forces applied to it. In experimental applications, usually, the inverse problem occurs, in deed it can be measured some quantities related to the degrees of freedom to determine the transfer function of the system.

Therefore in rolling-stock applications the most important quantities for the determination of the running behaviour of a vehicle are the wheel-rail contact forces that, of course, are not a known input of the system. For the wagon under exam a system that permits to determine such forces has been studied and applied. The instrumentation method is quite innovative because of the small spaces available under and on the wagon and the lack of possibility to evaluate contact forces through the wheel web deformation.

A preliminary overview about the instrumentation methods [17] for the contact forces evaluation is useful to understand what is the state of the art. These methods are essentially collected in three main categories:

- **wheel deformation method**: this kind of method uses strain-gauge bridges, connected to a telemetry system that acquires the signals and supply the energy to them. The strain-gauges are placed on the wheel web, and the determination of the forces acting on the contact surface occurs through the measurement of the wheel deformation. This method is the most used for vehicles geared with "usual" wheels i.e. with a big ratio between diameter and thickness, that allows a measurable bending of the wheel web. The output signal is related to the mechanical properties of the wheel and a preventive calibration of the system is necessary for the determination of the relationship (usually linear) between the force applied and the web deformation. Only the **Y** forces on each wheel can be determined with this method, **Q** forces must be measured through the primary suspension deflection, in deed the diametral position where the strain gauges are applied is the more sensitive to the **Y** forces.

- **axle deformations method**: this method is used to estimate the contact forces measuring the axle deformation, in terms of bending and torsional strains. Historically this practice was used on hollow axles because of the easiest way for cable crossing fro the central part of the wheelset to the outer. Although it can be implemented also on full body axles making an hole on the wheel web (this method is used in this thesis). Acquiring the strain-gauge bridges signals, the **Y**, **Q** and **T** forces can be obtained through an heavy post-processing of the signals;
- **mechanical component deflections method**: this method is the older and determines the **H** force (lateral force on the whole wheelset) through the relative displacement between the axle and the axlebox: **Q** forces are measured through the primary suspension deflection, so not any information about the single **Y** force can be retrieved.

Because of the mono-block shape of the SAADKMS wheels the 2nd method is used, since no significant web deformation can be measured during the on-track tests.

The wheelsets used for the instrumentation are the same of the series production, employed during the life of the vehicle, opportunely modified and instrumented to become a measurement instrument with its own precision, sensitivity and resolution. Along each axle, 8 measurement sections (figure 6.1) had been individuated (7 for bending and 1 for torsion). Through a fem model of the wheelset are evaluated the positions where the signals due to the applied forces are higher. On each section, for the bending evaluation, two half strain-gauge bridges, mounted at 90°, measure the rotating signal that reaches its maximum when the plane that contains the bridge includes also the contact point. Two half bridges are used to achieve a more constant signal on a complete wheel revolution. Wheels have been holed in order to cross cables from the centre of the axle to the external part where the telemetry pick up is installed. The radial position of the hole is selected through a fem model of the wheel, where the tensions due to the contact forces are lower. In the section of the torsion measurement, a full bridge with stain gauges at 45° is installed for the evaluation of the tangential stress induced by the torque force on the axle.

Before proceeding with the description of the calibration of the wheelsets, some concepts about the strain-gauge theory shall be done for the comprehension of the strain-gauge positioning.

6.1 Strain-gauges application

Strain gauge bridges are electrical circuits with resistances on each branch [18], that vary their value in relation to their deformation, giving a resultant tension signal that depends on the value of each resistance and on the power supplier tension.

The theory of strain gauge bridges, the resultant tension signal directly follows equation 6.1:

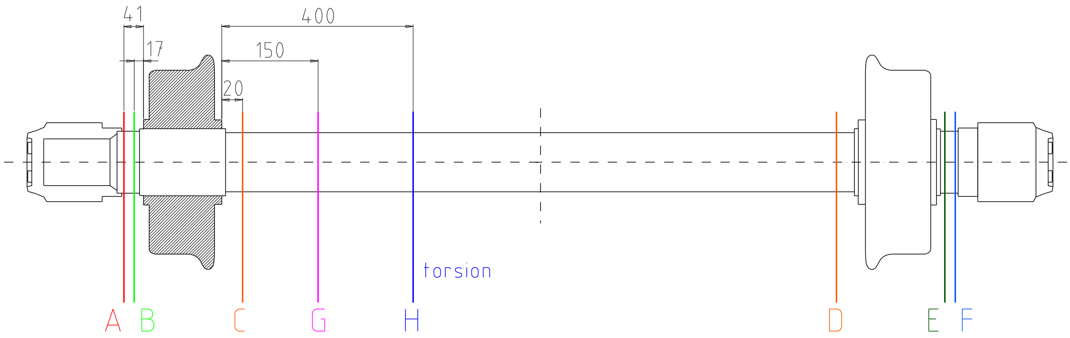
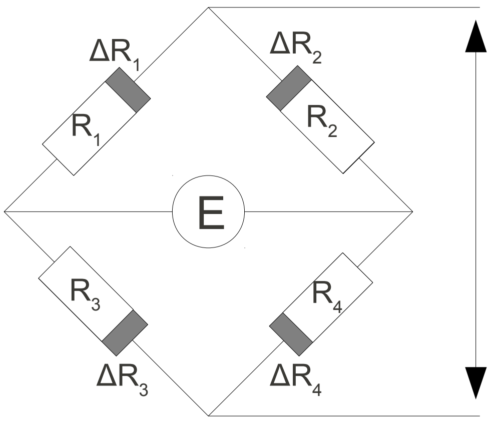


Figure 6.1. Axle measurement sections



$$\Delta V = E\frac{R_1 R_2}{(R_1+R_2)^2}\left(\frac{\Delta R_1}{R_1}-\frac{\Delta R_2}{R_2}+\frac{\Delta R_3}{R_3}-\frac{\Delta R_4}{R_4}\right) \tag{6.1}$$

Figure 6.2. Wheatstone bridge

where ΔR_i is the difference between the real value of each resistance and the nominal one. For simplicity the position of the strain-gauge (1, 2, 3, 4) as depicted in the figure 6.2 will be always the same during in this thesis.

Observing this equation three remarks immediately follow:

- the relation of the tension V with the resistances is linear;
- resistance values on adjacent branches have opposite signs, so their are subtracted in the final calculation of tension;
- resistance values on opposite branches have equal signs, so they are summed in the final calculation of tension.

These considerations explain the motivations that lead to adopt opposite branches at 180° for bending measurements, in fact in this way the variations of resistance due to strains are summed in the V calculation, amplifying the signal.

For torque evaluation, the measurement bridge is composed with 4 strain-gauges with a 45° pose with respect to the axles axis. This directly descend by the Mohr circle in the case of pure torsion stress. Figure 6.3 shows that the directions of the principal stresses are at 45° with respect to the axis of the examined beam.

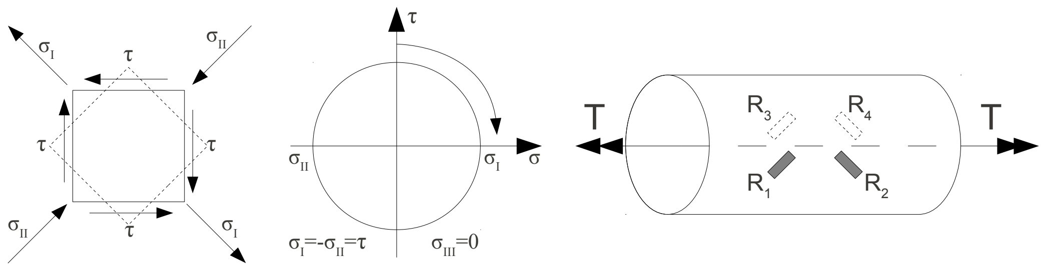


Figure 6.3. Pure torque stresses on a round section beam

From theory the strain due to a torque force applied on a circular section beam is equal to:

$$T = GJ_p\Theta \qquad , \qquad \gamma = \Theta r \qquad , \qquad \gamma = 2\epsilon_I = -2\epsilon_{II} = 2\epsilon$$

where r is the radii of the beam section, then:

$$T = GJ_p\frac{\gamma}{r} = GJ_p\frac{2\epsilon_I}{r} = 2G\frac{J_p}{r}\frac{1}{k}\frac{\Delta R_1}{R_1}$$

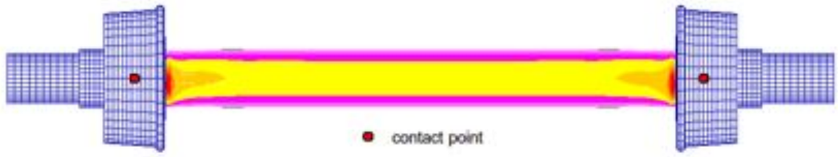
The configuration illustrated in figure 6.3 is able to compensate the effects of bending stresses in the measurement section because the adjacent branches of the bridge are at opposite diametral position, so their signals are subtracted, avoiding the bending effects. In conclusion this should be the best measure for the torque evaluation because it's not affected by other force inputs. The position of the strain-gauge bridge along the axle is not constrained because the torque due to the traction efforts is constant between the wheels.

6.2 Modelling of the wheelset structure

As introduced, the measurement sections are chosen by a preliminary F.E.A. on the wheelset model for the evaluation of the best positioning of the sections, compatibly with the geometry of the system and the bulk of the other mechanical parts (brakes, pipes and leverage). Meshing the wheelset model with regular hexaedric elements is the best solution to obtain a regular mesh on the model and because through the nodal displacement of the elements on the surface is quite simple to simulate the tension signal generated by the strain-gauge bridges. These signals can be used to observe what is the entity of the signals and its shape varying the number of strain-gauge on an axle section, in order to evaluate what is the best combination between this number and the mean value of the signal.

Moreover a preliminary analysis on the forces signals can be performed using the linear behaviour of the wheelsets, in deeds their deformations are quite proportional to the applied forces.

The general approach at the basis of this method is to solve a closed form system of equations obtained by the solution of the loaded beam problem as depicted in figure 6.4. The correlation between strains and applied forces are determined experimentally.



(a) F.E.M. lateral strains, centre of the axle, Y and Q forces applied, upper-view

(b) F.E.M. lateral strains, journal of the axle, Y and Q forces applied, upper-view

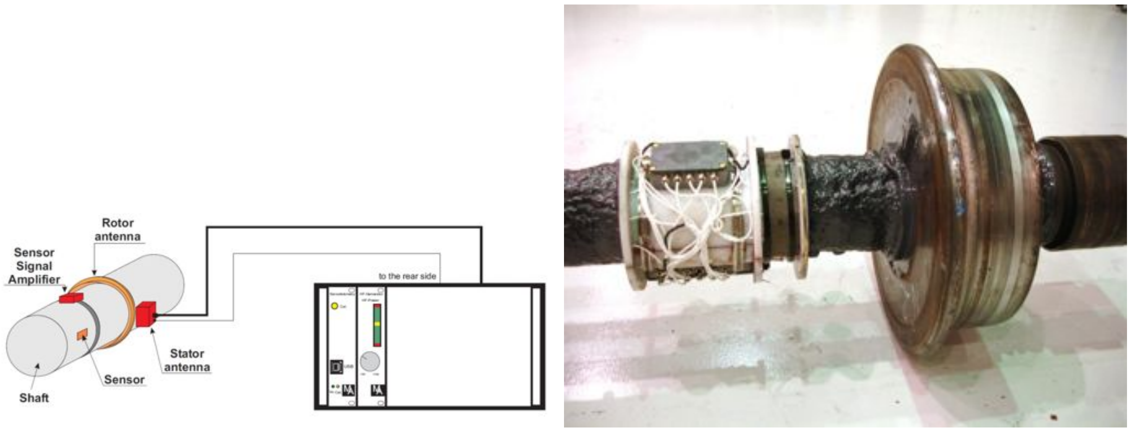
Figure 6.4. F.E.M. results - strains

Once the strain-gauges are applied in the selected positions, the experimental calibration of the wheelset is necessary to correlate the input forces to the measured signals. To perform this operation the measurement chain shall be completed with the telemetry and the pickup.

6.3 Signal acquiring

Once the signals are generated by the strain-gauge bridges, each signal flows through cables to a 12 channels digital telemetry that acquires the analog signals from the bridges and convert them to digital signals. Since there are 14 half bridges (each one produces a signal) and a full bridge, only 5 bending sections and the torque section can be acquired simultaneously. The other channels are redundant and realize the possibility to have a fail safe system switching from a bridge to another if one breaks.

An antenna, which have the double task to supply the power to the strain-gauges and to retrieve the signals, transmits them to a not-contact inductive receiver that is the link between the telemetry and the acquiring system (figure 6.5).



(a) Scheme of the acquisition chain (b) Real application of the telemetry

Figure 6.5. Telemetry scheme and instrumentation

With this equipment each wheelset constitutes a stand-alone measurement instrument, but as said it must be calibrated in order to correlate the force input to the signal output through the transfer function of the mechanical system. This operation is performed firstly on a test rig of the *POLIMI* that is able to statically apply forces in all the directions on the complete bogie and than is performed on a dynamic roller rig of *Lucchini* to make a dynamic calibration of the single wheelset while it's rotating under known applied forces.

6.3.1 Signal reconstruction

The force signals reported in figure 6.12, as said, are not the output of the acquiring system, but they are the results of the post-processing algorithm following described [19], for the determination of the force signals in the time domain. To do this operation, first of all, some consideration about the dimension of the matrices used must be done.

The telemetry output is a 13x *ns* matrix [T] where 13 are the half bridge strain-gauge signals acquired (SG_{ij}) and *ns* are the number of samples acquired, thus the matrix has the following shape:

$$[T] = \begin{bmatrix} SG_{1,1} & SG_{1,2} & \dots & SG_{1,ns} \\ SG_{2,1} & SG_{2,2} & \dots & SG_{2,ns} \\ SG_{3,1} & SG_{3,2} & \dots & SG_{3,ns} \\ \vdots & \vdots & \ddots & \vdots \\ SG_{13,1} & SG_{13,2} & \dots & SG_{13,ns} \end{bmatrix} \tag{6.2}$$

The signal on each section is reconstructed multiplying a matrix [S] that couples the strain-gauge signals of the same measurement section. This coupler matrix is defined as follow (6.3):

$$[S] = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 & 0 \\ 0 & 0 & 0 & 0 & 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \tag{6.3}$$

Thus the bending moment signal (M_{sec}) is obtained squaring each single half bridge signal, then summed with that on the same section and finally the square root of the result is made as indicated by equation 6.4:

$$M_{sec} = \sqrt{b_1^2 + b_2^2} \tag{6.4}$$

For example using the signals acquired by the telemetry during the dynamic calibration (figure 6.14), the bending moment on the section F is a rotating vector (related to a rotating reference system with the axis on the strain-gauge bridges of the section) defined in the time domain as figure 6.6 depicts.

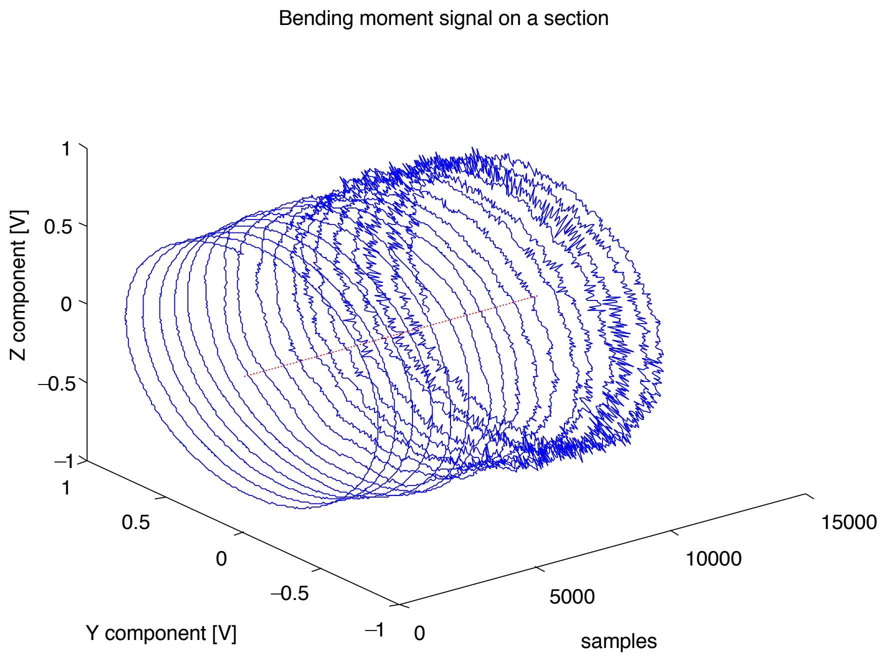
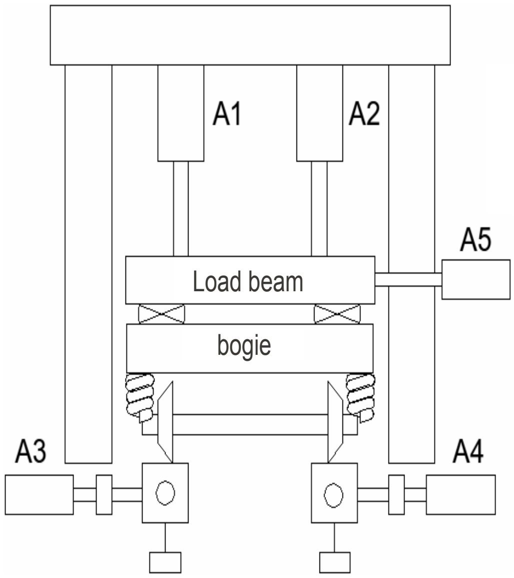


Figure 6.6. Rotating bending moment construction

Finally the force value on each wheel is derived using the calibration matrix obtained during the static calibration routine.

6.3.2 Static test rig layout

The *POLIMI* test rig is composed by a portal structure with 6 actuators that acts directly on the central beam of the complete bogie. The quantity to be measured is the contact force at the wheel-rail interface, distinguishing all its components; thus the actuators feedback set point is the force measured by a 7 axis scale placed under one of the wheels as showed in figure 6.7. The wheelset under calibration is constrained with a wheel on a mobile rail section (over the scale) and with the other wheel on a fixed rail section; this constrain condition realizes an isostatic configuration that permits the direct measure of the forces applied.



(a) Scheme of the POLIMI rig

(b) Rig scale

Figure 6.7. Polimi roller rig scheme and scale

The vertical actuators apply forces on the bogie central beam, while the longitudinal and lateral forces acts directly in correspondence with the wheel-rail interface through rods geared with load cells at their ends.

Once the bogie is positioned on the test rig a calibration routine starts and the acquisition system elaborates the input data keeping in relation the force inputs with the telemetry output. The routine consists in a load history applied to the bogie, the force set points are illustrated in the figure 6.8 and their value are chosen in order to cover all the possible ranges of forces that the vehicle could experience during its running life (this information had been retrieved through the simulation results).

The procedure is repeated for each wheel moving the bogie along the rig and rotating it calibrating all the wheels.

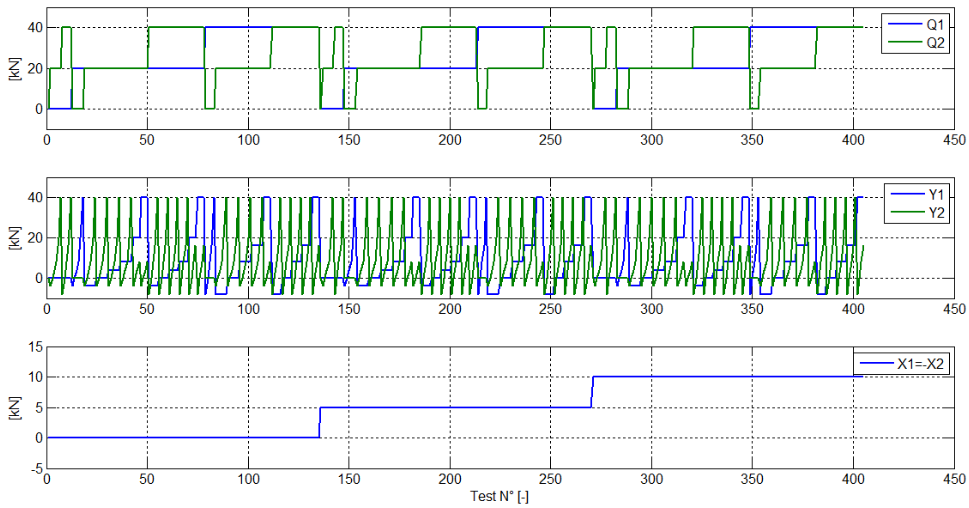


Figure 6.8. Static load history applied to each wheelset

6.3.3 Calibration method

Each time we have an experimental data set, we can try to find the best fitting mathematical model. When a set of parameters appear to have a linear correlation between input and output and the number of observations is higher than the basis functions, it's possible to approach the problem with the **Linear least squares** method. In deed it keeps in relation the idealized measurements of the model (strains, ϵ) with the unknown input parameters (Forces, F) through a linear function.

In general a linear relation between two quantities is expressed as in equation 6.5:

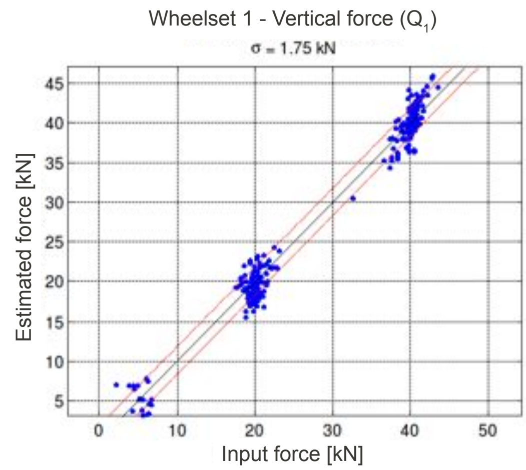
$$x \approx [A]y \tag{6.5}$$

This method is used when we have an overdetermined system, thus more equations then unknowns, so it is not possible to exactly solve the system. A solution is to solve the system in the least square way as in equation 6.6:

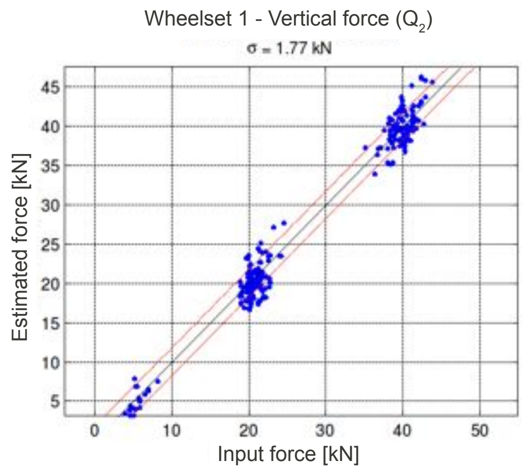
$$\min_{y} ||[A]y - x|| \tag{6.6}$$

The minimization problem just described has an unique solution, because of the linear independents row of the matrix $[A]$ in our case. In the case under exam $x = [F]$, $y = [\epsilon]$ and $[A]$ is the "**calibration matrix**". The correlation between the wheelset strains and the applied forces $[F]$ is linear and the $[\epsilon]$ is a full row rank matrix, thus the coefficients A_{ij} are the solutions of the equation 6.7:

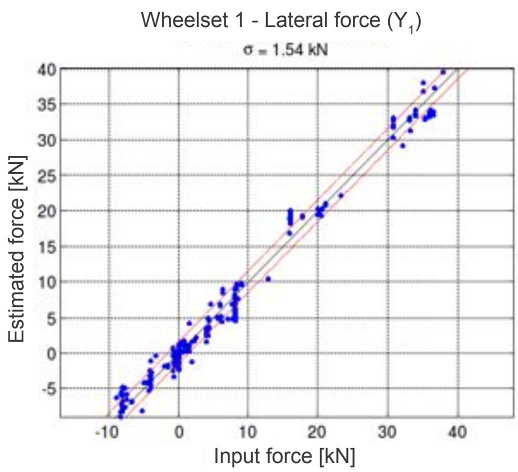
$$\begin{aligned} &[A] = [F][\epsilon]^T\left([\epsilon][\epsilon]^T\right)^{-1} \\ &[A] = [F][\epsilon]^+ \quad \text{where } [\epsilon]^+ \text{ is the } [\epsilon] \text{ pseudo-inverse matrix} \end{aligned} \tag{6.7}$$



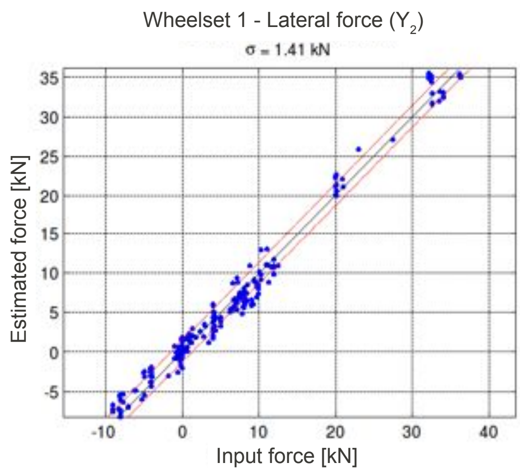
(a) Q_1 force estimation



(b) Q_2 force estimation



(c) Y_1 force estimation



(d) Y_2 force estimation

Figure 6.9. Forces estimations of wheelset 1

σ [kN]						
	Q_1	Q_2	Y_1	Y_2	X_1	X_2
Wheelset 1	1,75	1,77	1,54	1,41	0,39	0,43
Wheelset 2	3,13	3,11	2,17	2,5	0,33	0,42
Wheelset 3	2,92	2,85	2,74	2,75	0,48	1,53
Wheelset 4	1,46	1,33	1,36	2,38	0,24	0,44

Table 6.1. σ of each wheelset for the force measurement

where:

- $[F]$ is a 5xn (force components x number of tests) matrix;
- $[\epsilon]$ is a mxn (number of channels (strains) x number of tests) matrix (it's pseudo-inverse is an nxm matrix);
- $[A]$ is a 5xm matrix;

Now it is possible to evaluate the difference between the estimated force F_{est} and the real force F_{real} applied to the system, for the quantification of the quality of the measure.

For this purpose the standard deviation σ (6.8) is a good estimator that gives the uncertainty of the measure quantifying the dispersion around the mean of the data.

$$\sigma = \sqrt{\frac{1}{n}\sum_{i=1}^{n}(x_i - \overline{x})^2} \tag{6.8}$$

6.3.4 Experimental calibration results

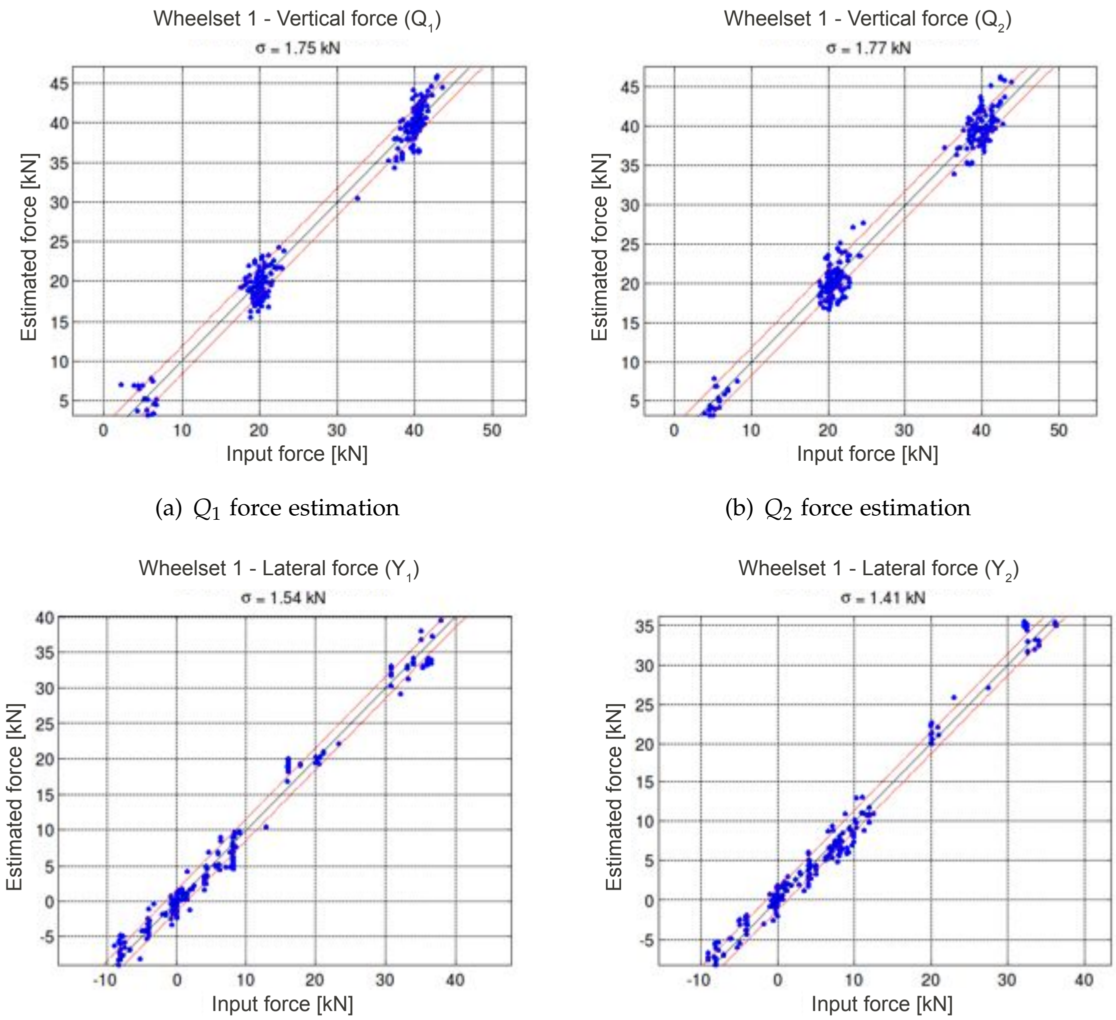
As explained above, the calibration routines have the purpose to define the relation between the measured strains and the contact forces and the "calibration matrix" is the mathematical entity that fulfil this task. The pictures 6.10 show the dispersion of the measurements with respect to the forces set points of the first wheelset.

The calibration has been performed for each wheelset for the attribution of the measurement uncertainty to all of them. In table 6.1 all the standard deviations are indicated.

As figure 6.10 shows the linearity of the problem is supported by experimental results. A more accurate measure of forces could be reached using a more thick force set point, in order to have a bigger number of experimental tests, since, as the equation 6.8 shows, the standard deviation entity is in inverse proportion to the number of samples. Therefore the [1] don't introduce any evaluation or necessity to declare the measurement uncertainty, and the values reached with the tests are considered acceptable.

However to remark is that the static Q forces on each wheel are about 2 t when the wagon is empty, so the force estimation error, considering the calibration results, could reach the 15 % of the real value. The same consideration could be done for the Y forces.

Both the "POLIMI" and the ANSF retain this result acceptable, considering the hardware (12 ch. telemetry) available and the alternatives present on the market for this application.



(a) Q_1 force estimation

(b) Q_2 force estimation

(c) Y_1 force estimation

(d) Y_2 force estimation

Figure 6.10. Forces estimations of wheelset 1

6.3.5 Dynamic calibration

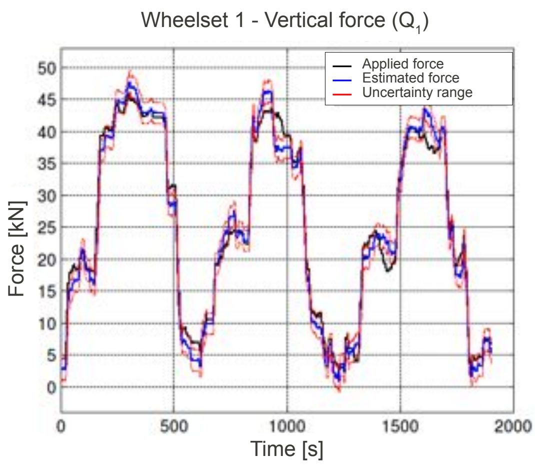
Figure 6.11. Lucchini BU300 test rig

Usually the static calibration is sufficient to reach an acceptable confidence on the results of the measurement system, but in this case also a dynamic calibration has been performed to evaluate the single wheelset behaviour varying the angle of attack of the wheels and the position of the contact point. This kind of calibration is also useful to verify the measurement chain, from the strain-gauge to the acquisition system when the wheelset rotates, making possible a check of the cabling and all that physical stuffs that could not be investigated during a static test [20].

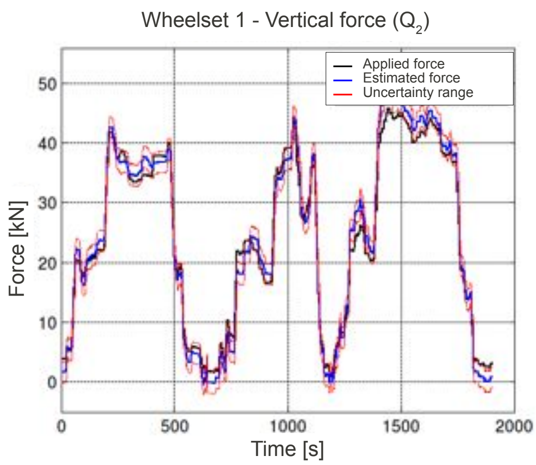
The test rig used for this operation is the *BU300* (6.11) located at the *Lucchini* plant in Lovere. It has, obviously, a similar architecture with respect to the static test rig, indeed it has a portal geared with actuators that apply forces to the wheelset and two rollers that keep the wheelset in rotation and, through their web bending, measure the contact forces using strain-gauges. Thus the scale of the static rig is substituted by the rollers whose deformation measurements are the feedback for the actuator control.

Figure 6.12 depicts how the quality of the measure is acceptable also during a dynamic test of the wheelset, in deed the gap between the red lines indicates the maximum and the minimum value of the measure in relation with the standard deviation, and it is possible to observe that they are very close to the applied forces history.

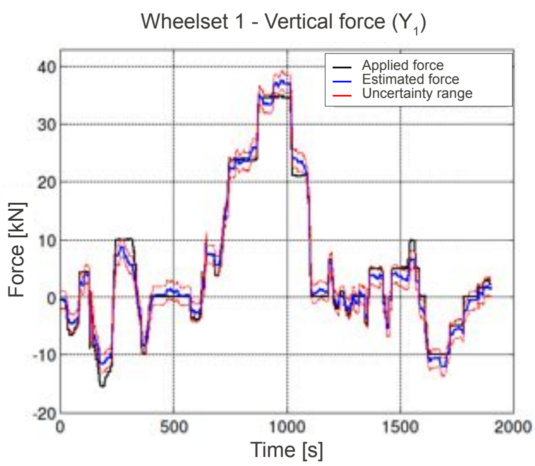
On the same rig another test for the evaluation of the Y/Q limit allowed by the geometry of the wheelset has been performed. The procedure adopted is based on the lifting of one wheel, just enough to lose its contact with the rail and pushing the other toward the flange contact just before the derailment occurs, as depicted in figure 6.13.



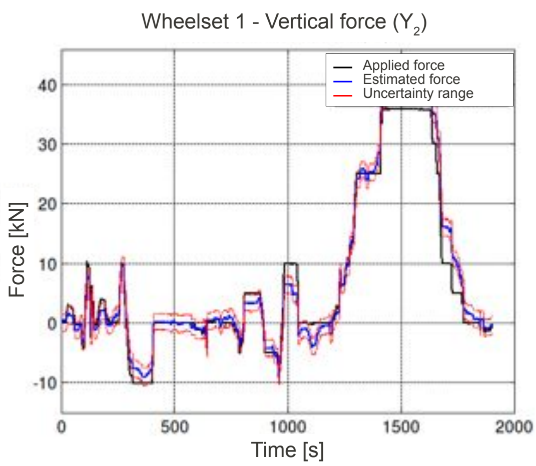
(a) Q_1 dynamic force estimation



(b) Q_2 dynamic force estimation



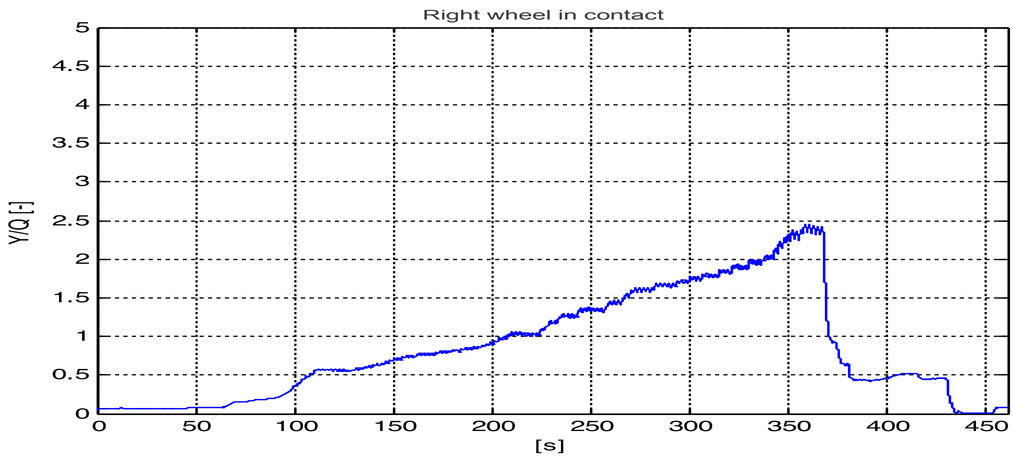
(c) Y_1 dynamic force estimation



(d) Y_2 dynamic force estimation

Figure 6.12. Dynamic forces estimations of wheelset 1

(a) Position sequence



(b) Y/Q measured limit

Figure 6.13. Experimental limit for the Y/Q factor

As showed in the picture 6.13 at the value of $Y/Q = 2.5$ the derailment has not took place, but during tests the limit calculated with the Nadal's formulae is used, because of its normative validity. Moreover during the laboratory tests the boundary conditions are controlled and vary very slowly, a condition that is not verified during on-track tests [21]. The signal output of each strain-gauge bridge on the same measurement section, are depicted in the figures 6.14 and, as said, constitutes the input of the telemetry that shall be post processed to obtain the force signal.

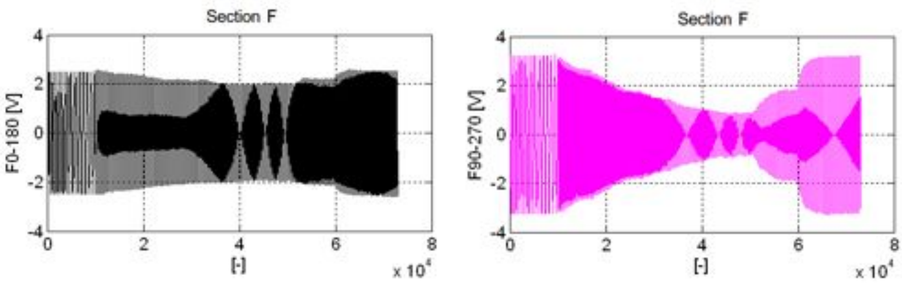


Figure 6.14. Signals from section F

6.4 Other signals

The EN 14363 defines two kind of signals to manage during the running tests to evaluate the dynamic behaviour of the vehicles i.e. force and acceleration signals. The forces are measured through the wheelset deformations as indicated, for the accelerations some accelerometers have been placed along the wagon, positioning them in accordance with standards. Indeeds [1] prescribes the monitoring, during the tests, of the following quantities for the "complete method" adopted in this case:

- guiding forces **Y**, lateral measuring direction;
- wheel force **Q**, vertical measuring direction;
- sum of guiding forces $\sum Y$ of a wheelset for the safety against track shifting;
- quotient **Y/Q** of guiding force/wheel force for the safety against the derailment due to the climbing of the wheel flange onto the rail.
- acceleration at the bogie frame, in correspondence with axle-boxes;
- acceleration at the carbody, in correspondence with the bogies and at the centre of the wagon.

Mems accelerometers are placed on the semi-bogie, the bogie frame and the flat wagon for the measuring of the accelerations in the principal directions.

The lateral acceleration spectrum is the best way to know if the wagon reaches an unstable behaviour since, as each mechanical system, it has its own frequency in correspondence of which its displacements are amplified leading to a dangerous and uncontrollable dynamic.

Further measurements are performed to evaluate the traction conditions, thus some load cells have been collocated between the frontal bumpers, on the rotating head next to the loco, and the head body. Moreover an instrumented coupler between the measure wagon an the coach is installed to evaluate the traction force through the coupler strain.

Another useful informations for the dynamic behaviour investigation are the relative rotation between the bogie frame and the flat-wagon and the rotation between the bogie frame and the four semi-bogies. These quantities are acquired through some potentiometers which measures the relative displacements between these parts.

Table 6.2 depicts what are the signals added to the wheel-rail contact force to investigate the running behaviour of the wagon. All these measurements have been preventively arranged in order to obtain as more informations as possible on the behaviour of the convoy.

We can observe that other useful measures as the angle of attack of the wheelset should give a very important contribute to the analysis of the running behaviour of the vehicle. However this kind of measure is very difficult to realize because the axleboxes are not protected with a cap and there is not the possibility to perform a continuous monitoring of this quantity with a fixed instrument.

6.5 Acquisition system

All the signals above mentioned are acquired by an architecture that is composed by two main parts:

- the remote station: this part has the task to acquire and store all the signals;
- the local station: this part has the task to post-process, visualize and elaborate the desired signals.

The remote station is a laptop close to the telemetries and it directly acquires their digital signals and all the other analog signals installed on the wagon through a National Instrument® system . A trigger signal synchronizes all the signals in order to avoid delays and phase displacements. An ethernet cable links the remote and the local stations, for the data transfer in both directions, indeed the local station can connect to the other through a remote desktop connection. The local station is a PC placed on the coach and directly connected to the remote station with the ethernet cable and with a power-line that supplies energy to all the measurement instruments (220V, 24V, 10V, 5V). This station is also put in parallel with an UPS that avoid energy, and then acquisition, fails. On both the stations a LabView® software is installed for the acquisition of the signals and their elaboration.

Figure 6.15 reports a scheme of the whole acquisition system.

This configuration with two main parts has been used because the whole amount of data is huge and a commercial pc has not enough power to acquire, store, post-process and visualize all the signals, so that tasks are divided onto two dedicated machines. Because the telemetries drivers works only on a windows system a Microsoft® platform is used for both the stations.

The "safety" signals are on-line monitored during the running tests in order to have the possibility to evaluate constantly the behaviour of the vehicle, and to transmit the right command to the train driver that must be able to brake or give the right tractive effort by the loco.

For the graphical representation of these signals [1] prescribes some filtering methods in order to reject high frequency component that are not consistent energetically: respectively a 10 Hz and 20 Hz low-pass filter for lateral and vertical parameters.

Other four signals can be observed in relation to the test performed, for example during the track tests at the homologation speed the measured and calculated non-compensated acceleration is a good parameter for the evaluation of the quality of the "Y" and "Q" measures.

n°	quantity	direction	description	side	instrument
1-52			Wheelsets strain gauge signals		
53	acceleration	long	Semi-bogie , WS 1,2	L side	(MEMS1)
54	acceleration	lat	Semi-bogie , WS 1,2	L side	(MEMS1)
55	acceleration	vert	Semi-bogie , WS 1,2	L side	(MEMS1)
56	acceleration	long	bogie , between axleboxes WS 1,2	L side	(MEMS2)
57	acceleration	lat	bogie , between axleboxes WS 1,2	L side	(MEMS2)
58	acceleration	vert	bogie , between axleboxes WS 1,2	L side	(MEMS2)
59	acceleration	long	bogie , between axleboxes WS 3,4	L side	(MEMS3)
60	acceleration	lat	bogie , between axleboxes WS 3,4	L side	(MEMS3)
61	acceleration	vert	bogie , between axleboxes WS 3,4	L side	(MEMS3)
62	acceleration	long	Semi-bogie , WS 1,2	R side	(MEMS4)
63	acceleration	lat	Semi-bogie , WS 1,2	R side	(MEMS4)
64	acceleration	vert	Semi-bogie , WS 1,2	R side	(MEMS4)
65	acceleration	long	bogie , between axleboxes WS 1,2	R side	(MEMS5)
66	acceleration	lat	bogie , between axleboxes WS 1,2	R side	(MEMS5)
67	acceleration	vert	bogie , between axleboxes WS 1,2	R side	(MEMS5)
68	acceleration	long	bogie , between axleboxes WS 5,6	L side	(MEMS6)
69	acceleration	lat	bogie , between axleboxes WS 5,6	L side	(MEMS6)
70	acceleration	vert	bogie , between axleboxes WS 5,6	L side	(MEMS6)
71	acceleration	long	bogie , between axleboxes WS 7,8	L side	(MEMS7)
72	acceleration	lat	bogie , between axleboxes WS 7,8	L side	(MEMS7)
73	acceleration	vert	bogie , between axleboxes WS 7,8	L side	(MEMS7)
74	acceleration	long	flat-wagon on central bearing instr. Bogie		(MEMS8)
75	acceleration	lat	flat-wagon on central bearing instr. Bogie		(MEMS8)
76	acceleration	vert	flat-wagon on central bearing instr. Bogie		(MEMS8)
77	acceleration	long	flat-wagon, central bearing instr. Bogie		(MEMS9)
78	acceleration	lat	flat-wagon, central bearing instr. Bogie		(MEMS9)
79	acceleration	vert	flat-wagon, central bearing instr. Bogie		(MEMS9)
80	acceleration	long	flat-wagon on central bearing second bogie		(MEMS10)
81	acceleration	lat	flat-wagon on central bearing second bogie		(MEMS10)
82	acceleration	vert	flat-wagon on central bearing second bogie		(MEMS10)
83	acceleration	long	bogie , between axleboxes WS 9,10	L side	(MEMS11)
84	acceleration	lat	bogie , between axleboxes WS 9,10	L side	(MEMS11)
85	acceleration	vert	bogie , between axleboxes WS 9,10	L side	(MEMS11)
86	acceleration	long	bogie , between axleboxes WS 15,16	L side	(MEMS12)
87	acceleration	lat	bogie , between axleboxes WS 15,16	L side	(MEMS12)
88	acceleration	vert	bogie , between axleboxes WS 15,16	L side	(MEMS12)
89	acceleration	long	bogie , between axleboxes WS 23,24	L side	(MEMS13)
90	acceleration	lat	bogie , between axleboxes WS 23,24	L side	(MEMS13)
91	acceleration	vert	bogie , between axleboxes WS 23,24	L side	(MEMS13)
92	acceleration	lat	axlebox del1 WS	L side	(PIEZO1)
93	acceleration	lat	axlebox del4 WS	L side	(PIEZO2)
94	acceleration	lat	bogie , between axleboxes WS 17,18	L side	(PIEZO3)
95	rotation		flat-wagon / instr. Bogie		(CELESCO)
96	displacement	long	bogie / semi-bogie front	L side	(POT1)
97	displacement	vert	bogie / semi-bogie front int.	L side	(POT2)
98	displacement	vert	bogie / semi-bogie rear int.	L side	(POT3)
99	displacement	vert	bogie / semi-bogie rear ext.	L side	(POT4)
100	displacement	long	bogie / semi-bogie front	R side	(POT5)
101	displacement	vert	bogie / semi-bogie front int.	R side	(POT6)
102	displacement	vert	bogie / semi-bogie rear int.	R side	(POT7)
103	displacement	vert	bogie / semi-bogie rear ext.	R side	(POT8)
104	displacement	lat	bogie / semi-bogie front	R side	(POT9)
105	displacement	lat	bogie / semi-bogie rear	R side	(POT10)
106	displacement	long	bumpstop coach / wagon	L side	(POT11)
107	displacement	long	bumpstop coach / wagon	R side	(POT12)
108	displacement	long	bumpstop head	L side	(POT13)
109	displacement	long	bumpstop head	R side	(POT14)
110	displacement	long	bumpstop wagon		(POT15)
111	displacement	long	bumpstop wagon 2		(POT16)
112	force	long	bumpstop bolt sup.	L side	(CELLA1)
113	force	long	bumpstop bolt sup.	R side	(CELLA2)
114	force	long	bumpstop bolt inf.	L side	(CELLA3)
115	force	long	bumpstop bolt inf.	R side	(CELLA4)
116	distance	long	1 WS		(PROXI)
117	Speed	long	1 WS		(PROXI)
118	force	long	coupler coach-wagon		(SG)

Table 6.2. Other signals

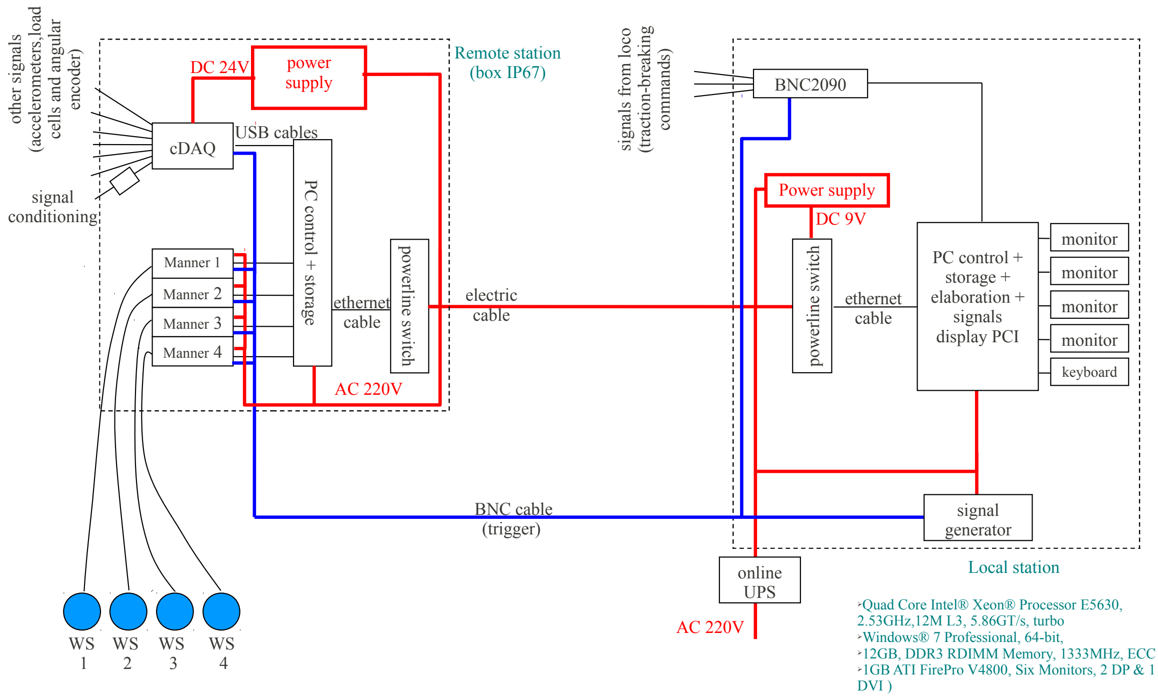


Figure 6.15. Scheme of the acquisition system

CHAPTER 7

TRACK TESTS - DEFINITION AND RESULTS

The railway world is governed by many simple, rigid rules, and each railwayman usually knows only that which are strictly pertaining to its competences. This creates a very solid and secure system because the rules are clear and not to be interpreted or infringed. Although tests are something not conventional, outside from normal circulation rules, with many borderline conditions to be observed and some waivers from conventional rules that involves people from different companies, with different competencies and at different levels. At the light of this clarification, it's easy to prospect some difficulties when approaching and performing some new kind of tests, such as that in the purpose of this thesis, because of the connections between all the rules, of course with the preventive information and agreement with all the railway institution involved.

As introduced the complete test campaign has been defined with the goal to understand as more as possible about the behaviour of these wagons, in particular their dynamic during the switches routing because of some past derailment that occur inside the Italian boundaries 1.1. Besides, since the normative survey is not clear about the dynamic characterization of a vehicle on switches, a type-procedure has been defined, starting from the guide lines introduced in the reference leaflet EN14363, and following its philosophy. Thus the real aim of the work is the identification of a methodology, on the basis of railway experiences, to perform these tests on particular wagons such as SAADKMS, taking into account of the practical issues that can occur on the line.

Since these kinds of wagons have never been characterized with a complete on track tests homologation, over all because of the difficulty of their instrumentation, the lack of experimental data makes it difficult to understand what is the real influence of each component on their running behaviour. Although they don't need to perform a complete homologation to be kept in service, because they already own the circulation permissions accorded by RFI and ANSF, with the limitations indicated in section 3.8.

Despite of this condition, for completeness of the investigation process, it had been decided, in accordance with ANSF, to perform some tests following the EN14363 philosophy with the purpose to really know what are the ranges of safety parameters values under which they run safely, which are rationally higher than for an usual vehicle for the reasons mentioned before in chapter 3.8.

Thus two macro set of tests had been performed, the first following the [1] for what concern the homologation tests, the second, using the same leaflet rules for the switch routing behaviour investigation. Keeping track of this, two set of tests with two different convoy configurations have been defined:

- short composition convoy (3 SAADKMS vehicles)
  - EN 14363 homologation tests philosophy;
  - switches routing investigation;
- long composition convoy (21 SAADKMS vehicles)
  - commercial route investigation;
  - switches routing investigation;

Although the tests with the first convoy configuration had been concluded in the course of this thesis, for the second only the definition of the test scenarios have been individuated dealing with ANSF the conclusion of the whole campaign in the first half of 2013. Later in this work are described the configurations for each test in terms of track topology and vehicle load and traction condition with a focus to the motivations that leads such choices. To keep in mind that all these test configurations have been agreed by ANSF, Polimi, Italcertifer and Unifi.

7.1 Short convoy tests

Firstly, for the short convoy, some words must be spent on the choice of the composition layout and of the tracks for the tests.

The convoy is composed by 5 vehicles, a coach with the "local station" of the instrumentation system on board, a baggage and two electronic locos. The three SAADKMS vehicles are at the centre of the convoy and the instrumented wheelsets belong to one of the extreme bogies, close to the coach, in order to monitor the forces of the first guiding wheelset which is the most critical. Since also the first trailing wheelset of the second bogie,in the other running direction, is to be investigated because of it's criticality emerged by the results of the simulations, with this convoy layout both the extreme bogies results instrumented. The first wheelset instrumentation is also a mandatory prescription of [1]. The composition is depicted in figure 7.1.

Finally, because the the traction unit between the instrumented wagon and the next coach had been kept slack, during the tests when the convoy is dragged, the influence of the coach on the wagon is minimized.

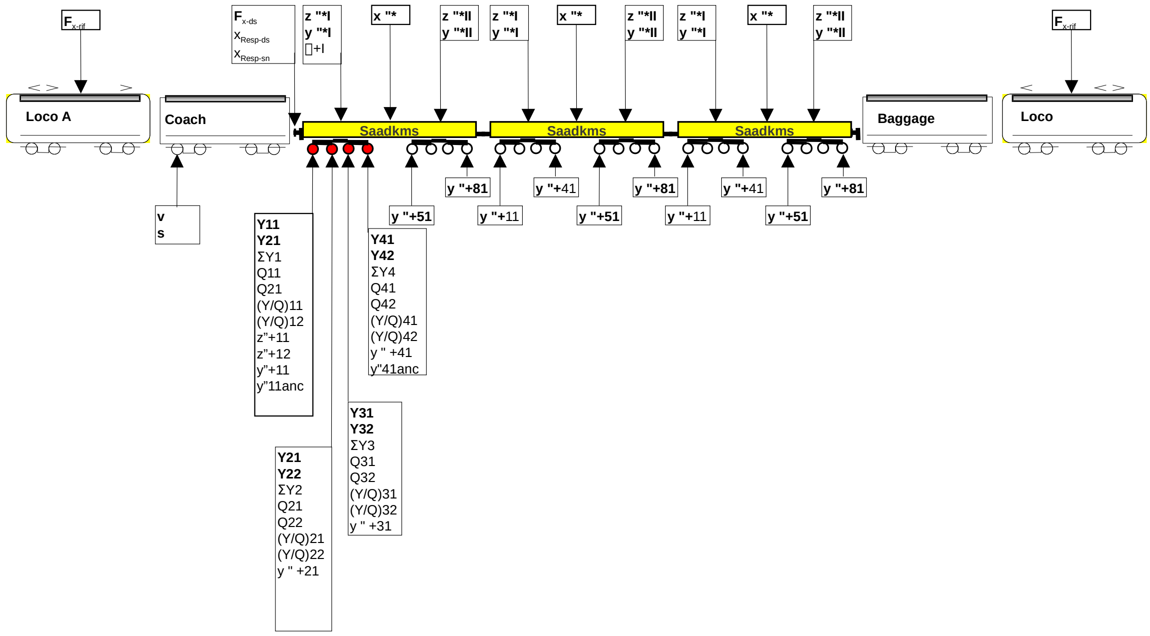


Figure 7.1. Short convoy layout

7.1.1 EN14363 homologation tests philosophy

This leaflet represents the reference document each time a vehicle have to perform some on-track tests. It contains a lot of rules to follow for the definition of the behaviour of a vehicle in order to establish if it's running safe under certain conditions that could regard the wheel profile or the failure of the pneumetic suspensions if present. About the on-track tests for the homologation of a vehicle, it establishes what are the characteristics of the tracks to be routed. Such tracks shall be divided into sections characterized on the basis of their geometry (radii and quality). A minimum number of sections have to be achieved in order to perform a statistical processing of the results, obtaining an unique value to compare with the imposed limits.

Thus the test route shall be chosen taking into account of the number of sections collected, respecting the prescriptions on the cant deficiency, radius and speed indicated in the leaflet. Moreover for a valid statistical analysis the quality level of the track geometry should respect a specific distribution indicated in the leaflet. For these reasons, after an accurate selection of the available tracks, considering the logistical issues and the availability of the tracks granted by RFI, two test tracks have been selected: the Firenze-Arezzo and the Firenze-Empoli lines [22].

The track classes defined in the leaflet are four, each one selected with the aim to investigate a characteristic behaviour of a vehicle, and are listed in table 7.1.

As indicated in table 7.1 each track topology shall be routed under certain conditions, for example, the maximum allowed speed is the "homologation" speed increased by 10% ± 5 km/h and the cant deficiency (*cd*) must be included between 0.75 and 1.1 ± 0.05 of

Test zone	Objective	Anticipated vehicle characteristic
Straight track and curves with very large radii	Testing of the vehicle in the area of the vehicle permissible speed	Low quasi static foces, but larger dynamic content in all evaluation variables
Large radius curve	Testing of the vehicle in the area of the vehicle permissible speed and cant deficiency	Superposition of quasistatic and dynamic contents of all evaluation variables
Small-radius curves (400m≤R≤600m); very small-radius curves (250m≤R≤400m)	Testing of the vehicle in the area of the vehicle permissible cant deficiency	Larger quasistatic guiding forces, wheel forces and accelerations, dynamic content generally decreases

Table 7.1. EN 14363 track classes

the admissible cant deficiency cd_{adm}. On the Italian infrastructure there are some signalling rules to take into account to determine the cd_{adm} in relation to the kind of vehicle, called "Rango".

The "Rango" of a vehicle indicates the maximum anc allowed on the track. Obviously it's related to the vehicle speed and the track cant. SAADKMS vehicles must respect the "Rango A" conditions on each track, thus a maximum a_{nc} of $0.6 m/s^2$ is allowed. As mentioned in 2.3 the relation between a_{nc} and cd is linear so "Rango A" allows a maximum cant deficiency of 130 mm.

As said on each track class shall be collected a certain number of sections in order to have a statistical consistent data set for a statistical elaboration of the measured quantities. The data acquired shall be aligned with the track kilo-metric progressive and divided in as more as possible sections, in accordance with the minimum and maximum length allowed for each class of track (figure 7.2).

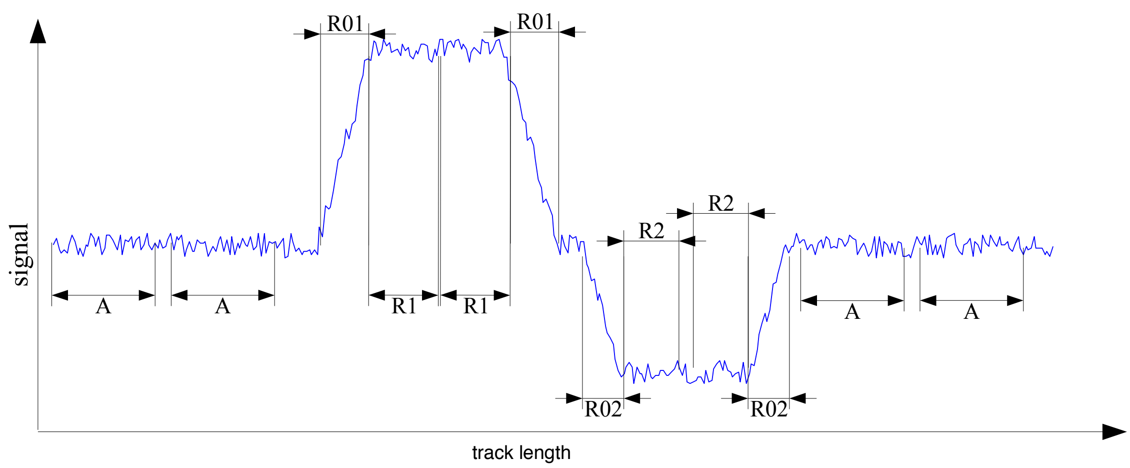


Figure 7.2. Section selection

On each section a statistical evaluation is requested, defining the values at 99.85% and 0.15% of each signal related to its theoretical normal distribution in the section. For some parameters also the values at 50% (quasi-static signal) shall be elaborated. Once rounded up all the statistical values for each section, another statistical elaboration, considering each track class, shall be performed in order to calculate the final values at 99.85% and 0.15% of the whole distribution. Thus, the maximum value for the quantity under investigation is determined using a normal distribution with a confidence interval indicated in the [1].

Finally the evaluated quantity shall be used to compare each parameter with respect to the assessment limits.

7.2 Safety against derailment at low speed

Before entering on the full-line, a vehicle shall performs some preliminary checks in order to evaluate the vehicle attitude to avoid derailment, in other words the low speed safety of the vehicle while running small radii curves. These tests are collected in three methods, one excluding the others, that could be used on the basis of the tools available from the tester. The common test to all the procedures is the weighing [23]. In this application the "Method 2" of the section "Safety against derailment" of the leaflet is used, so a twist test and a flat track test have been performed.

7.2.1 Weighing

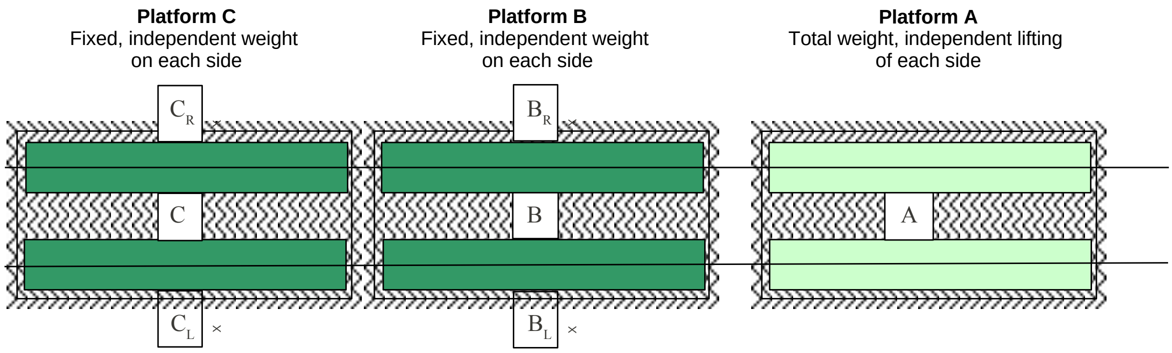


Figure 7.3. Weight rig layout

First of all the vehicle must be weighted to evaluate the mass on each wheel and if there are some kind of non symmetric distributions of the masses along the train. This kind of measure is realized with a dedicated test rig available at the Osmannoro-RFI plant. The weigh is one of the mandatory parameters for the characterization of a railway vehicle, and it must be declared before of its introduction on the line.

The test rig is composed by three balance platforms (A, B and C) that are able to weight the vehicle [24]. Platform A can lift each side of the vehicle independently, measuring the weight of both the vehicle sides, platforms B and C are not able to lift the wheels but they can weigh each side independently (figure 7.3). Once the train is placed on the measure platform a manual routine starts to lift the wheels of one side of the vehicle in order to eliminate all the mechanical clearances and to unjam all the suspension friction components. The weigh measures are obtained on each wheel, for the first axle by a direct measurement, for the others, because of the short base length of the semi-bogies, the weight is calculated through the difference with the measures of the toward wheels.

7.2.1.1 Results

Both the test conditions have been measured, the empty and the full load conditions. For the second one some arrangements have been adopted because of some gauge issues.

Empty weight The results of this static test are depicted by picture 7.5(a) and shows a quite symmetric distribution of masses on both the wagon sides. The difference of weight between the bogies is due to the presence of the mobile head that weight about 1.2 t, and the difference between the left and the right wheel of the third wheelset is related to the presence of the “local station” instrumentation located in a box on the left side of the train in correspondence of that wheelset. On each wheel four cycles of weight have been performed in order to reduce the measure uncertainty. The maximum difference of weight between wheels is in the worst case of about 6.0%, that probe a quite homogeneous distribution of mass along the train (figure 7.5(a)).

Figure 7.4. Load condition emulation

Full load weight The full load condition has been emulated, as said, because of line gauge and logistic issues. In fact trucks, to be loaded on the wagons need a special ramp placed at the end of the SAADKMS convoy that allows the vehicles to climb the height gap. This operation implies the presence of rails at the level of the ground with enough space behind the end of the convoy, at least long as the truck and the ramp, to permit the climbing manoeuvres. In Tuscany these typologies of track layout are available in plants where car-transporter wagons are loaded, but because of the different heigh of the load plane it's not possible to use them for this operation. Moreover along the Tuscany's lines the limit gauge is such low to don't allow the passage of SAADKMS loaded with 44 t trucks [9].

Even if the resultant centre of gravity and inertia distribution are not the same than in the case of a real truck load, the solution adopted to simulate the load is to put gravel bags, usually used to ballast coaches, on the wagon floor and fixed with ropes and ground clothes as showed in figure 7.4. The fixation and the positioning of the loads had been an operation to perform with care in as for the central part of the vehicle is not designed to carry excessive weigh.

The procedure for the weight of the wagon has been the same as for the empty one and from the results a really symmetric and homogeneous distribution of masses. The choice to realize an homogeneous load along the vehicle is leaded by the reason that when a real truck is loaded, there are some markers on the wagon indicated by the manufacturer to maintain the load more uniform as possible.

The results of the measure are reported in figure 7.5(b). As visible the maximum difference between the two bogies is of about 6.4 % and the two sides of the vehicle are perfectly balanced.

7.2.2 Twist test

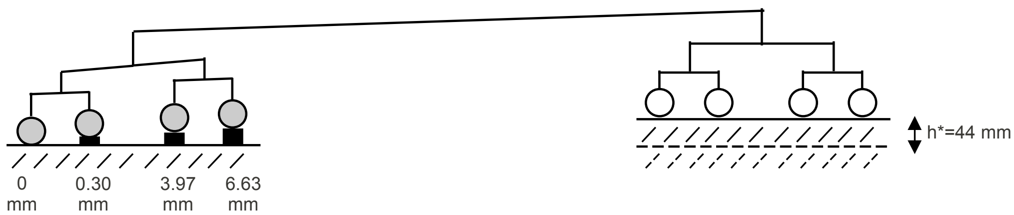
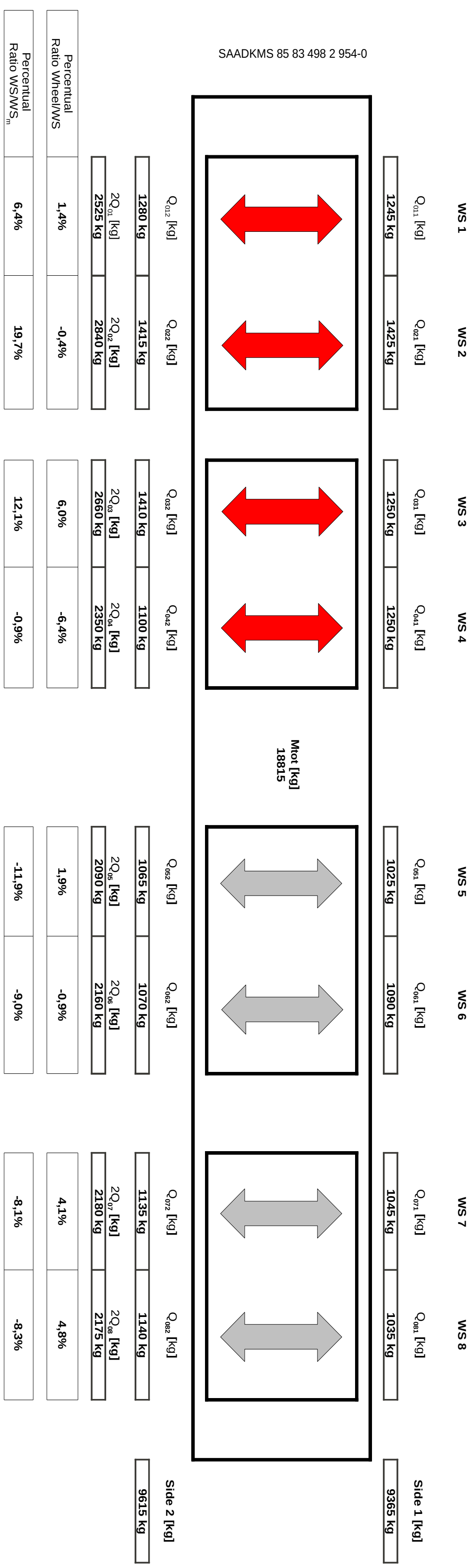


Figure 7.6. Twist gauges configuration

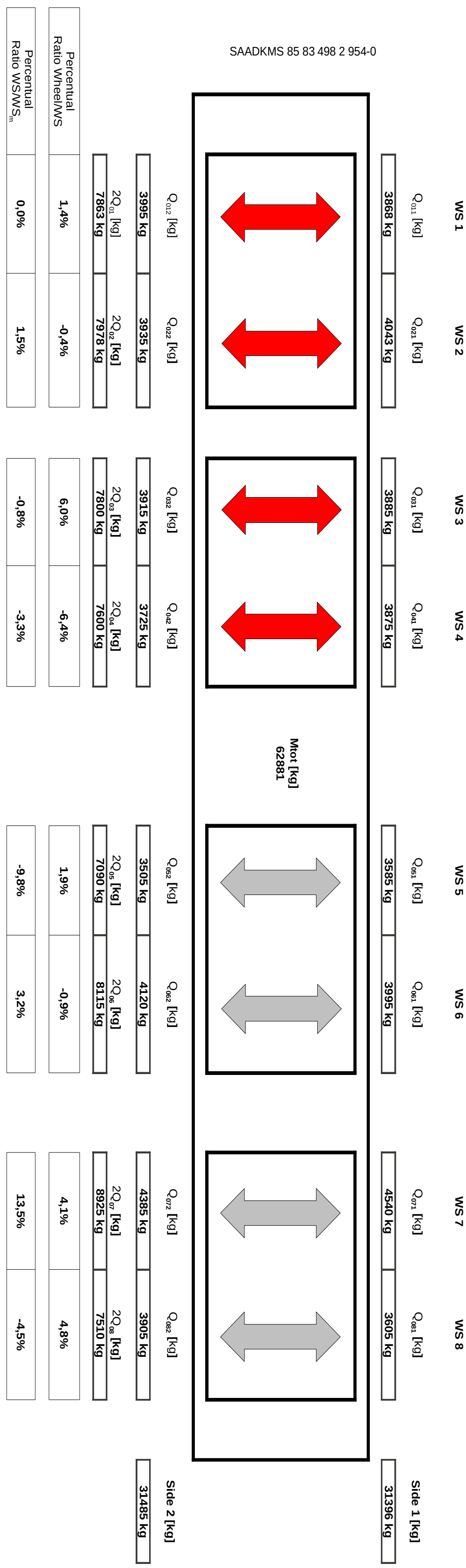
The twist test is one of the two measures that shall be performed on a railway vehicle, approaching method 2 of EN14363. It consists in the application of both wheels and bogie twist in order to measure the maximum unload on each wheel that is a good parameter to evaluate the capacity of a train to run on a small radius curves avoiding derailment [25].

Observing the norm [1] the maximum wheel unload shall not be greater than 60%, thus on each wheel at least the 40% of the wheelset load shall weigh, because otherwise the vehicle is not considered running safe.

The measure is performed with the same rig as the weighing, because of the possibility to completely lift one of the bogie with platform A, that is used to realize the bogie-base twist. The wheelset-base twist is performed using some calibrated gauges placed under the wheels, following the physical and mechanical consideration behind this test and defining

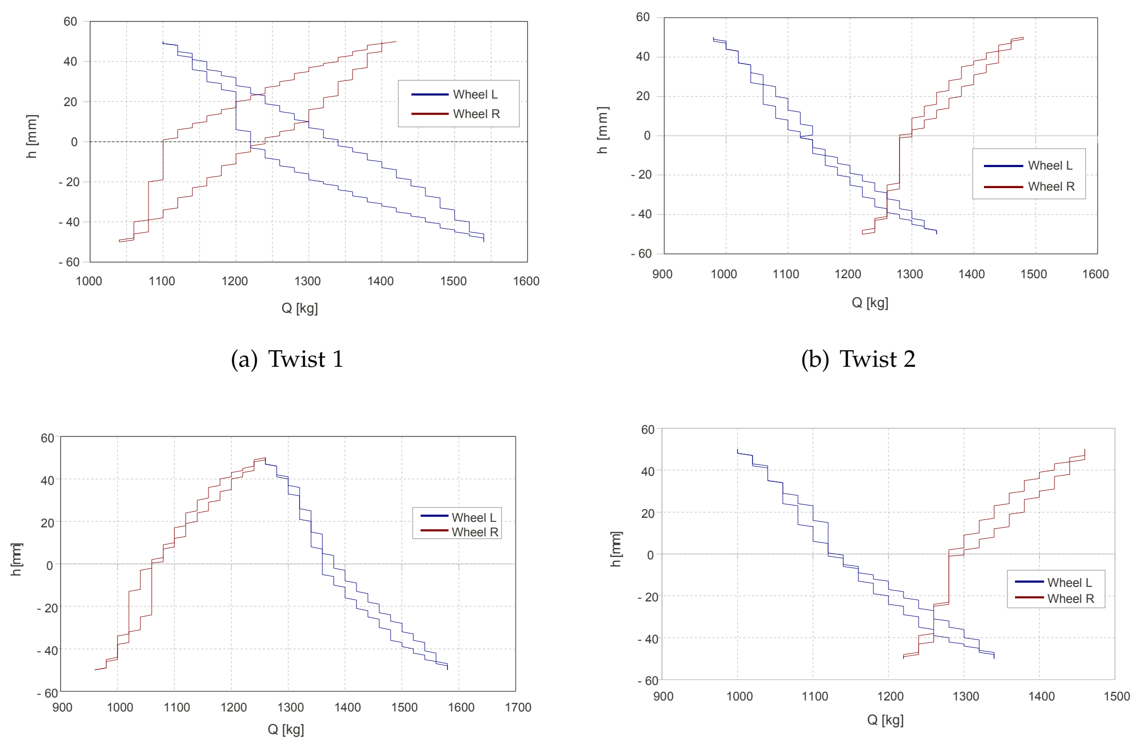


(a) Empty weight



(b) Load weight

Figure 7.5. Weight results



(a) Twist 1 (b) Twist 2

(c) Twist 3 (d) Twist 4

Figure 7.7. Twist results

a brand new procedure. In fact, the vehicle presents 8 wheelsets twist test is not envisaged by [1], leading to a twist configuration depicted in figure 7.6.

The twist slopes for the bogie-base $g^{\#}$ and the wheelset base g^{+} twist have been calculated in accordance with the Annex A.6 through the following equations:

$$g^{\#} = \frac{15}{14.425} + 2 = 0.00304\% \quad 4m < a^{\#} < 20m$$

$$g^{+} = 7 - \frac{5}{1.45} + 2 = 0.00355\% \quad a^{+} < 4m$$

The results are obtained applying a lift cycle on each wheel, measuring the load when in the nominal position, in order to obtain the results of the unloads on each wheel when the twists are applied.

For the first wheelset the maximum unload obtained is equal to 24.90% on the left wheel that represents the worst unload configuration. Since this value is below the 60% limit indicated, the vehicle can be considered safe against the maximum allowable twist that could be encountered on the track.

The results of each twist test are reported in figure 7.7 and it clearly shows the behaviour of the non-linear friction elements, in deeds when the static friction of the contact elements is overcome by the vertical load in the correspondent wheel there is a steep variation in the distribution of the wheel load.

7.2.3 Flat-curve test

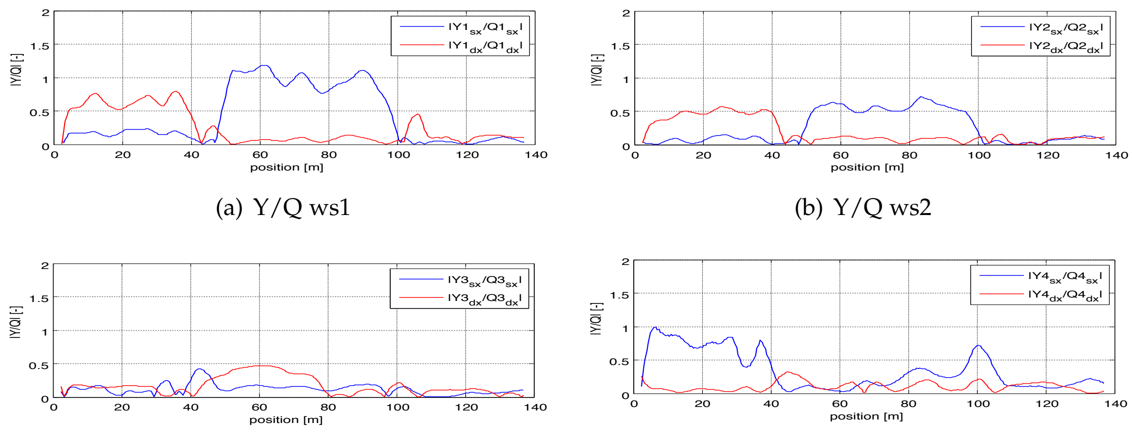
Using the second method of [1], the flat curve is the other necessary test to characterize the vehicle safety against derailment at low speed. This test consists in the routing of a double curve without superelevation, with no straight transition in the middle as prescribed in the [1] §4.1.3.3.3. The test must be repeated at least three times with a quite constant speed not exceeding 10 km/h with the mandatory condition of the dry rail.

These test started 15 m before the first curve, on a straight stretch, in order to achieve the sufficient constant speed at the entrance of the first curve. Both the load configurations had been tested.

7.2.3.1 Results

The results of these tests consist in the determination of the Y/Q ratio of the vehicle in certain sections of the track and in the determination of the angle of attack of the first wheelset in full curve. For these kind of vehicles the determination of the angle of attack is not possible through a continuous measure, because the rotating part of the axlebox is not covered by a cap, making impossible the fixation of a specific instrument. Otherwise two plates with strain-gauges should be used for its determination through the plates deformations.

Empty condition The results are relative to the mean Y/Q of the results of the four tests (figure 7.9). Since the test are highly repeatable this operation doesn't introduce an error in the determination of the assessment value.



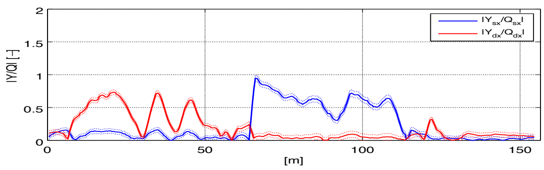
(a) Y/Q ws1 (b) Y/Q ws2

(c) Y/Q ws3 (d) Y/Q ws14

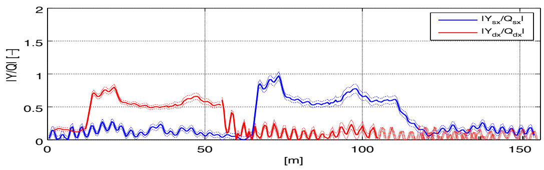
Figure 7.8. Y/Q results of the flat curve in empty condition

The limit value defined as the [1] is $Y/Q_{lim} = 1.44$, so the value reached during the flat curve is below that limit and the vehicle could be considered safe when routing small radius curves.

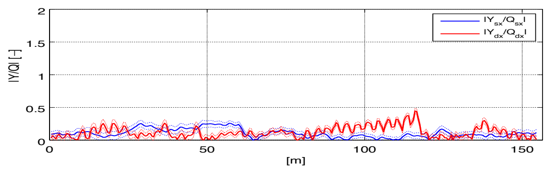
Load condition The test procedure is the same than in the empty condition and the speed is approximatively of 10 km/h. The results shows that the derailment coefficient recorded is quite always below the unity, declaring the vehicle safe for what concern the safety against the derailment at low speed.



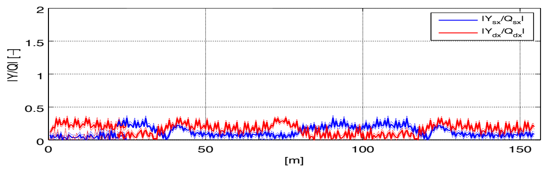
(a) Y/Q ws1



(b) Y/Q ws2



(c) Y/Q ws3



(d) Y/Q ws14

Figure 7.9. Y/Q results of the flat curve in empty condition

Once these tests are concluded and successfully passed, a vehicle is suitable to approach the full-line or for the other tests not mentioned in the leaflet.

7.3 Switches investigation

As mentioned before, there is a normative vacuum about the test to be performed on a vehicle over switches for it's assessment. The flat curve is idealized as the worst track geometry that a vehicle can encounter along it's route, so, in the majority of cases a test on it is enough to assess that a vehicle is safe while running small curves or switches. However when a particular convoy such that composed by SAADKMS vehicles is under investigation the flat test curve is not sufficient to determine its safety on switches. For this reason this thesis have also the purpose to define some guide lines useful for the selection of a test track layout to perform such tests.

There are some things to take into account for the definition of such procedure because the switch investigation shouldn't be a single test like on the flat curve. In deed the flat curve is not a track shape that a vehicle really encounter along its way, but it simulates the most critical condition that a vehicle can find, and its aim is to define a single parameter for the evaluation of the safety of a vehicle. Contrariwise many kind of switches, different for their tangent, radius and allowed running speed are on the way of every single vehicle and the behaviour investigation should not drain in a single test. Thus this test type stay in the middle course between the tests on the open track, where only some indications about the radius, geometry and speed are indicated in the leaflet, and a characterization test where the track is fully defined by the norm and a value shall be kept out from the tests.

Moreover many configuration of switches sequence could be encountered, and the most significant of them should be investigated. This suggest to concentrate the tests on a large manoeuvre plant where many routes structuring are possible.

About the running speed, these vehicles have some limitations, thus some velocity steps, rising from the actual to the goal speed, have been defined in order to allow the next step if the parameters of the actuals are evaluated safe during tests. The speed steps are, from 10 to 30 km/h: 5, 5, 5 and 5 km/h. Finally since these wagons run through the same route, it has been considered effective the choice of some switches along their fixed route, keeping attention to the question about the maintenance condition of the line, in order to avoid the selection of new or with the same "service life" switches.

Under the conditions over listed, this set of on-track tests are the core of the whole project, because of their focus on the reason behind the beginning of this work. In order to obtain the more complete as possible informations in terms of track geometry, running conditions and matching of the test switches with that usually routed by SAADKMS vehicles, after a detailed analysis of the possible scenarios available on the Italian lines, the best choice revealed the Roncafort plant, one of the terminal station for SAADKMS vehicle, where the convoys are composed and loaded to cross the borders toward Germany, Swiss and Austria.

One of the useful characteristic of this plant for our investigation is the presence of many switches with an analogue geometry of switches where these trains have been derailed, and on which the simulations had been performed and revealed critical conditions for the vehicle. Moreover the Roncafort plant supplies the real possibility to run all over the plant because of it's low activity during the day (most of the arrivals and departures are along the night).

The switches life evaluation had not been easy, but with the collaboration of the plant managers some qualitative informations had been achieved about the more used routes within the plant. Another issue related to the logistic and the infrastructure is that nowadays most of the "goods stations" are automatic and the central control system has some pre-determined routes to make the rolling stock goes from point A to point B. Usually this logic follows the principle of the minimum energy, thus the pre-selected routes are planned in order to cross the minimum number of switches. Excluding the automatic system it's possible to set the desired route.

Finally, since the tests shall be repeated with the long convoy, this plant is provided by long sections of binaries at its extremities, where a full convoy can be placed, permitting the implementation of the tests in the next configuration.

7.3.0.2 Assessment limit

Although the leaflet indicates that the Y/Q_{lim} shall not exceed 1.2, in accordance with ANSF considering the simulation and calibration tests, this limit had been accepted for the assessment of the behaviour of these vehicles. Moreover, as said, since the approach for the evaluation of the wagon behaviour in switches is a mix of on-track tests and characterization tests, the assessment limit defined for the flat curve had been used, indeed as the leaflet [1] states:

Following investigations by European railway administrations the limit value for $(Y/Q)_{lim}$ was defined as 1,2 for a flange angle of 70° (this corresponds to μ = 0,36 from A.2). For different flange angles the limit value is calculated by:

$$\frac{Y}{Q}_{lim} = \frac{tan\gamma - 0.36}{1 + 0.36\ tan\gamma}$$

Where 0.36 is the hypothetical friction coefficient.

Another value could be used but in the viewpoint to use the leaflet indications to define a new test environment, this way to define the assessment limit had been considered suitable for these tests.

Moreover an elaborations of the recorded signals with a sliding mean window of 1 m would be more appropriate for such small wheel, because it's possible to imagine that the derailment occurs in about 1 wheel revolution, but for the same reason as above the "conventional limit" has been used.

7.3.1 Switches life estimation

Through an analysis of the summary of the plant train traffic during the last four months, some informations about the utilization of some preferential routes for the handling of the trains can be achieved. In deeds the traffic in the plant is mainly from the open-line to the interport for the loading-unloading operations. Moreover since the routes are preselected, always the same roads are utilized for usual rolling stock handling. From the matching between Trenitalia (owner of the inter-port) and RFI (owner of the good station) data about the traffic in the whole plant, a qualitative estimation about the life condition of the switches has been obtained as depicted by graph 7.10.

The data has been supplied as time tables of the trains with the arrival and departure times, the binaries occupied and the operations performed on them. That data makes possible to assign a weight to each switch, making a ranking of the most and less used switches.

The 100% value corresponds to about 1080 passages in four months. Although only the data related to this temporal arc was available, the activity in the plant can be considered quite constant in time, so it could be extended to a longer period. This kind of survey had been useful to know that the condition of selected switches is not the same, giving to the results a more significant relevance.

Once individuated the plant for the tests, a theoretical "best" route in terms of number of switches crossed on each run for variety of curve sequence had been selected. Since all the switching control system is centralized and automatic it has been necessary excluding the automatic control in order to configure manually the desired route along the plant. Two ways have been ran, one for "30 km/h switches", with small curve radius (R=170 m) and one for the "60 km/h switches" with larger curves (R=400 m). The most "winding"

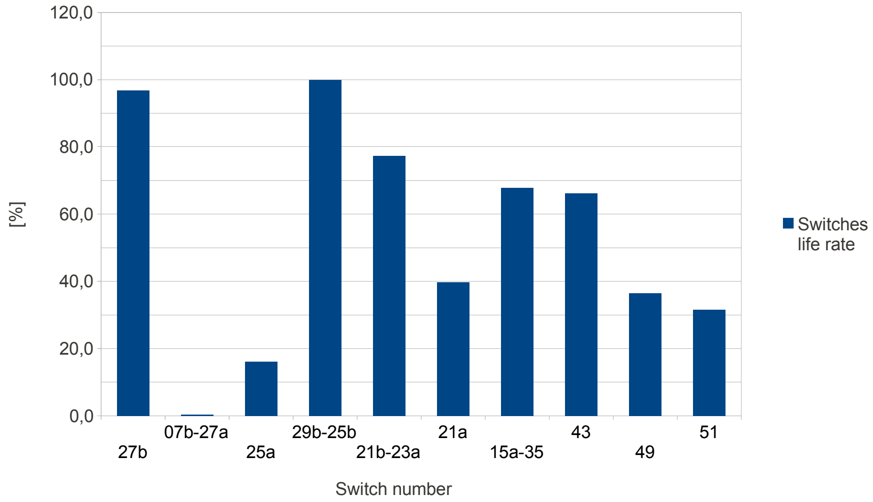


Figure 7.10. Switches life condition

switches sequence achievable is depicted by figure 7.11, where in black is put in evidence the selected routes for the tighter switches, in green that for the larger. As possible to see a variety of left, right, "english" and some double switches are present for the evaluation of the curve and counter-curve behaviour.

Once individuated the test routes, the convoy conditions have been defined, generating a matrix of target tests for the achievement of the final result. Such matrices are reported in tables 7.2 and 7.3, where (FW) is the forward direction, from Trento toward Brennero, and (BW) is the backward one.

As reported in tables 7.2 and 7.3 four conditions has been investigated:

- load;
- empty;
- dry;
- lubricated;

These terms refers to the conditions of the wagon load and the friction pads lubrication condition. These set of tests have been selected because they are of quite practical implementation since, i.e. the substitution of the rubber elements with others at different ageing, or the substitution of the central bearing friction material between bogie and wagon results impossible in practice, although they should enrich the investigation process. This kind of elements could be investigated by simulations where the implementation different elements in the wagon is certainly easier.

For what concerns the experimental results achieved in this experimental campaign, only the most significant results are showed and discussed in this thesis because of the

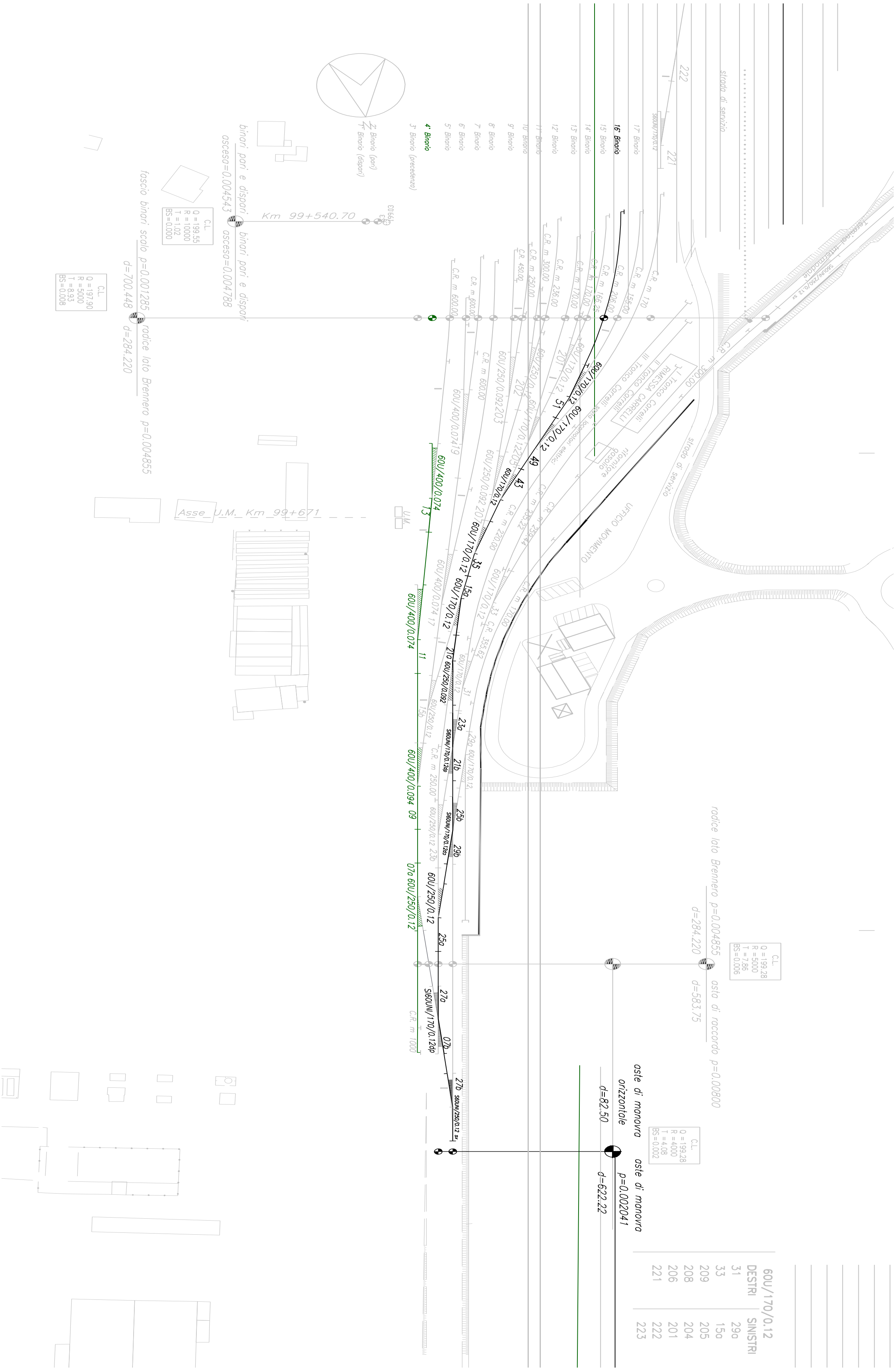


Figure 7.11. Roncafort plant - Test route

	Y/Q_{max}							
	"dry" condition							
	Load				Empty			
	Pushed		Pulled		Pushed		Pulled	
	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)
10 km/h	1,25	1,40	1,25	1,39	1,20	1,50	1,40	1,41
15 km/h	1,00			1,25	1,10			1,40
20 km/h	1,00			1,20	1,10			1,40
25 km/h	1,00	1,35	1,20	1,25	1,10	1,25	1,25	1,40
30 km/h	1,20	1,35	1,25	1,28	1,25	1,25	1,25	1,25
	"lubricated" condition							
10 km/h	1,00			1,20	0,85			1,35
15 km/h								
20 km/h	0,95			1,10	0,80			1,25
25 km/h								
30 km/h	0,90			1,10	0,75			1,25

Table 7.2. 30 km/h switches tests

	Y/Q_{max}							
	"dry" condition							
	Load				Empty			
	Pushed		Pulled		Pushed		Pulled	
	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)	1a WS (FW)	4a WS (BW)
30km/h	0,90			1,20	0,95			1,10
40km/h	0,90			1,20	0,95			1,10
50km/h	0,90			1,25	0,90			1,00
55km/h	0,90	1,25	1,00	1,25	0,95	1,10	1,00	1,10
60km/h	1,00	1,40	1,00	1,50	0,95	1,15	1,00	1,10
	"lubricated" condition							
30km/h	0,75			0,90	0,70			0,90
40km/h								
50km/h	0,75			0,95	0,75			0,90
55km/h								
60km/h	0,75			0,95	0,75			0,85

Table 7.3. 60 km/h switches tests

huge amount of data. Finally a summary of all the results is showed for the delineation of the overall behaviour of the vehicle.

For each selected test the same quantities are showed:

- wheel forces;
- Y/Q ratio on each wheel;
- bogie-wagon rotation displacements;
- bumpers displacements;

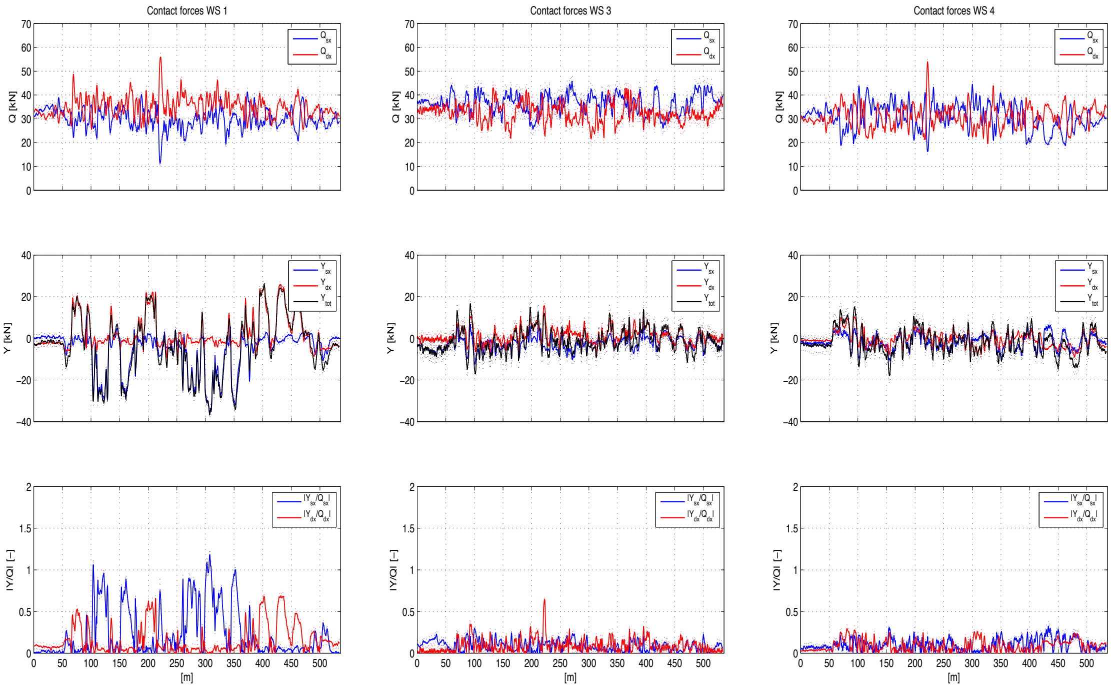
all these measurements are necessary to recovery what are the longitudinal, lateral and vertical dynamics of the wagon. Each test starts and finishes with a straight track scratch.

7.3.2 Switches tests results

The first condition analysed is the loaded one, with a 43.380 t truck. The load operation has been performed in the Roncafort plant where an apposite equipment is available for the truck climbing on the wagon. Once loaded the truck has been locked in the "manufacturer established" position, for the best mass distribution achievement.

For this tests two speed steps are completely described, the 10 km/h and the 30 km/h which are the actual and the aim set point velocity. Regrettably telemetry n°2 had some calibration problems during the tests and only after an heavy post-processing by the laboratory its result could be obtained, thus they are not showed in this thesis.

7.3.2.1 Test 102/131 - 10 km/h, R=170 m, Loaded condition



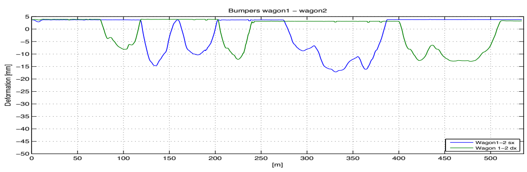
(a) $(Y/Q)_1$ (b) $(Y/Q)_3$ (c) $(Y/Q)_4$

Figure 7.12. Y/Q forces (test 102)

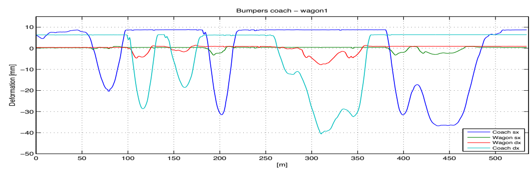
In test 102 the first wheelset is the leading one, and the travel direction is toward Trento. First of all a simple verify of the measured quantities are the values of the Q forces on each wheel that, when the vehicle is loaded, are about 3.6 t as showed in picture 7.12, in the first part, corresponding to the straight track. From the bogie-wagon relative displacement graph a quite symmetric steering behaviour can be observed, in deed the relative displacements on the right and left curves are fairly of the same entity but on left curves the bogie tends to find its equilibrium position at smaller displacements. Larger radius curve are easily recognisable by the smaller displacement of the bogie (about 10 mm) against the 15 mm for the 170 m radius switches.

For what concern the Y and Q forces and their ratio the first wheelset is the more critical as introduced during the simulations discussion. When the wagon approaches the first switch (27a, left curve) the right wheel experiences a loading due to the roll movement of the truck, while the left wheel for the equilibrium of the vertical forces results unloaded.

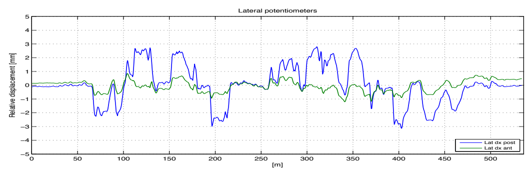
Near the entrance and the exit of each switch, the steeply change of curvature introduces a peak of lateral acceleration on the external wheel while in the middle the equilibrium condition is reached. By these pictures it's possible to observe a small difference in the Y forces on the left and right wheels while crossing respectively left and right switches of the same geometry; this behaviour is possibly leaded by the not perfectly symmetric steering characteristic of the wagon as deduced by the relative displacement between the bogie and the flat-wagon. Moreover the steering performances of the wagon is determined by the 6 degrees of freedom of the semi-bogie with respect to the bogie. In particular the measure of the lateral relative displacement between them (figure 7.13(c)) shows how the neck-crossbeam tends to adapt to the track geometry with small rotations around the primary suspensions. Also in this case the rear semi-bogie experiences smaller displacements and, as expected, the third and the forth wheelsets show a lower level of Y forces. Then the Y/Q ratio decreases, because they are already leaded by the first semi-bogie wheels that equilibrate the most part of lateral forces.



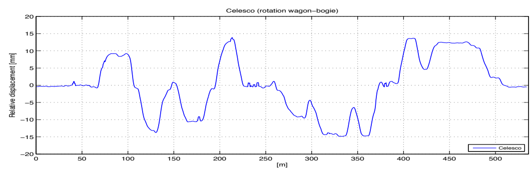
(a) wagon 1-2 bumpstops def.



(b) coach-wagon bumpstops def.



(c) semi-bogie/bogie rotation

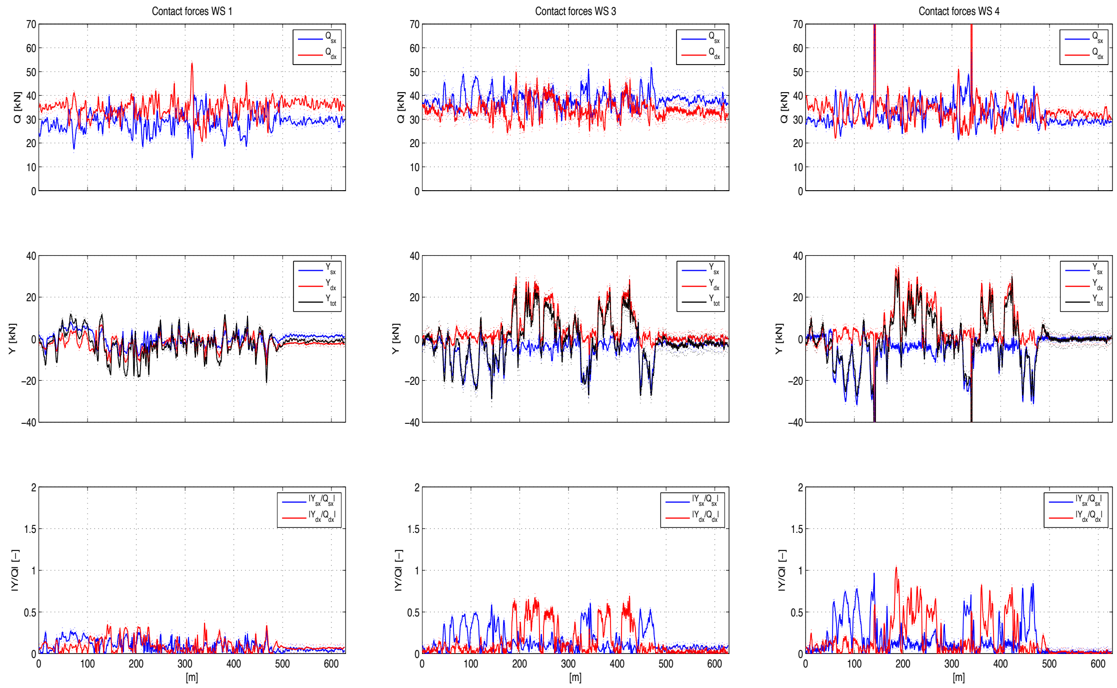


(d) bogie/wagon rotation

Figure 7.13. Other parameters (test 102)

Longitudinally the forces between wagons are measured indirectly through the bumpers deformations, these parameters represent the boundary force conditions to input in the multibody model of the single vehicle for the validation and the investigation of particular conditions, impossible to reproduce in the reality. This activity is programmed in a future work. As showed in figure 7.13(b) bumpers on the wagon are stiffer than that on the coach.

Following the results about the test 131 with the measurement bogie on the rear part of the wagon, thus with the forth wheelset leading, are showed to emulate to show what is the different behaviour of the rear bogie in the same load and speed conditions.

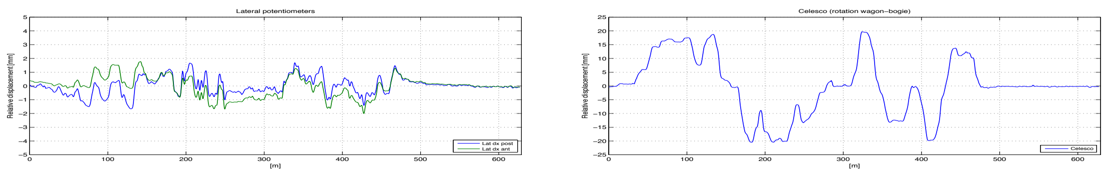


(a) $(Y/Q)_1$ (b) $(Y/Q)_3$ (c) $(Y/Q)_4$

Figure 7.14. Y/Q forces (test 131)

The Y forces measured on the forth leading wheelset are of the same entity than on the first bogie but the vertical load transfer is more consistent now because the frontal part of the wagon is already inserted in the curve and the vertical load transfer has already took place. Moreover the oversteering characteristic of the rear bogie, obtained during the simulations is verified by these experimental results, in deeds the relative lateral displacement of the rear bogie is higher of about 25% on each switch (figure 7.15(b)).

In conclusion, although the train run along the defined route in a different direction, so the curves are in opposite sides, many considerations could be done about its behaviour such as the oversteering characteristic of the rear bogie, with respect to the forward, and the attitude of the semibogies to adapt to the track curvature (figure 7.15(a)). The vertical load transfers of the rear bogie are greater than on the first bogie because the front part of the train is already inserted in the switch and the vertical load transfer on the outer side has already begun.

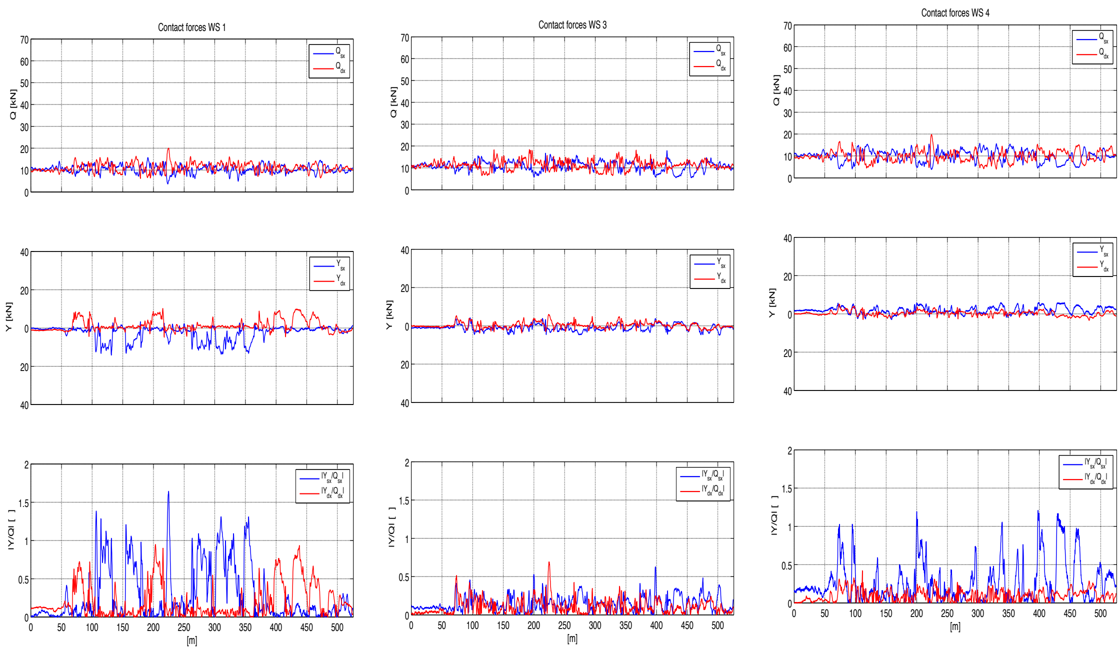


(a) semi-bogie/bogie rotation (b) bogie/wagon rotation

Figure 7.15. Other parameters (test 131)

7.3.2.2 Test 138/137 - 10 km/h, R=170 m, Empty condition

During these tests, a similar behaviour than in the loaded condition is observed, but the vertical loads on each wheel are of small entity along the curves because of the lower load on the wagon and the lower position of the centre of mass of the system, that reduces the entity of the moment due to the not compensated acceleration. For these reasons the values of the Y/Q ratio are usually higher than in the previous condition. The lateral forces are about 1/3 of that in loaded conditions, because of the lack of load on the wagon (about three times than in the empty condition). All these factors contribute to the final safety factor Y/Q that is about 1.1 on the smaller radius curves (figure 7.16).



(a) $(Y/Q)_1$ (b) $(Y/Q)_3$ (c) $(Y/Q)_4$

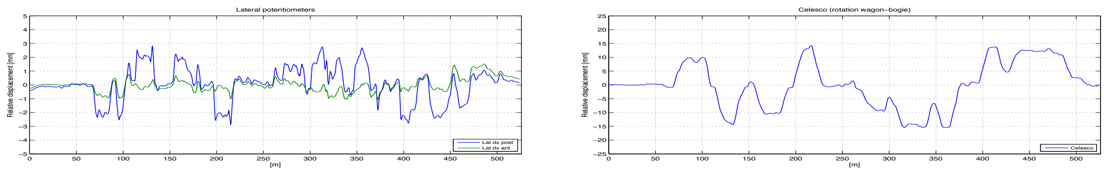
Figure 7.16. Y/Q forces (test 138)

Most of the lateral forces are equilibrated by the first two wheelset toward the direction of motion as in the previous described case. The relative rotation between bogie and wagon are quite the same than in loaded conditions and the major steering characteristic of the rear bogie is verified again.

The longitudinal dynamic of the convoy deduced by the bumpers elongations, is the same than in the loaded condition, because however the wagon is empty, the tests are performed at constant speed because from the simulation results the braking manoeuvre doesn't introduce to a particular dynamic effects in such a short convoy.

In test 137, where the rear bogie is monitored, the leading wheelset experiences some hurts at the exit of two switches; from the registrations the first seems to be exerted by an huge track irregularity due to the double switch geometry, especially by the sudden change of curvature without straight track between the curves. However this condition exerts big lateral and vertical forces on the wheelset, it's duration is quite small and it's not to be considered safety dangerous, but this condition should be avoided during the normal life of the vehicle.

The lateral displacement of the semi-bogie along the curves (figure 7.17(a)) gives a nice information about the steering capabilities of the wheelsets in fact since the potentiometers is placed toward the external side with respect to the rotation centre of the semi bogie (in the middle of the rubber elements) the measures are of opposite sign for the semi-bogies, thus it indicates that they rotate in the same direction. Another thing to take into account is that these displacements are very small because of the high stiffness of the rubber elements.



(a) semi-bogie/bogie displacements

(b) bogie/wagon displacements

Figure 7.17. Other parameters (test 138)

7.3.2.3 Test 110/135 - 30 km/h, R=170 m, Load condition

These tests are performed, as said, on the same route than the others, but at higher speed. This new condition introduces some dynamic effects on the wagon that are visible directly on the forces measured.

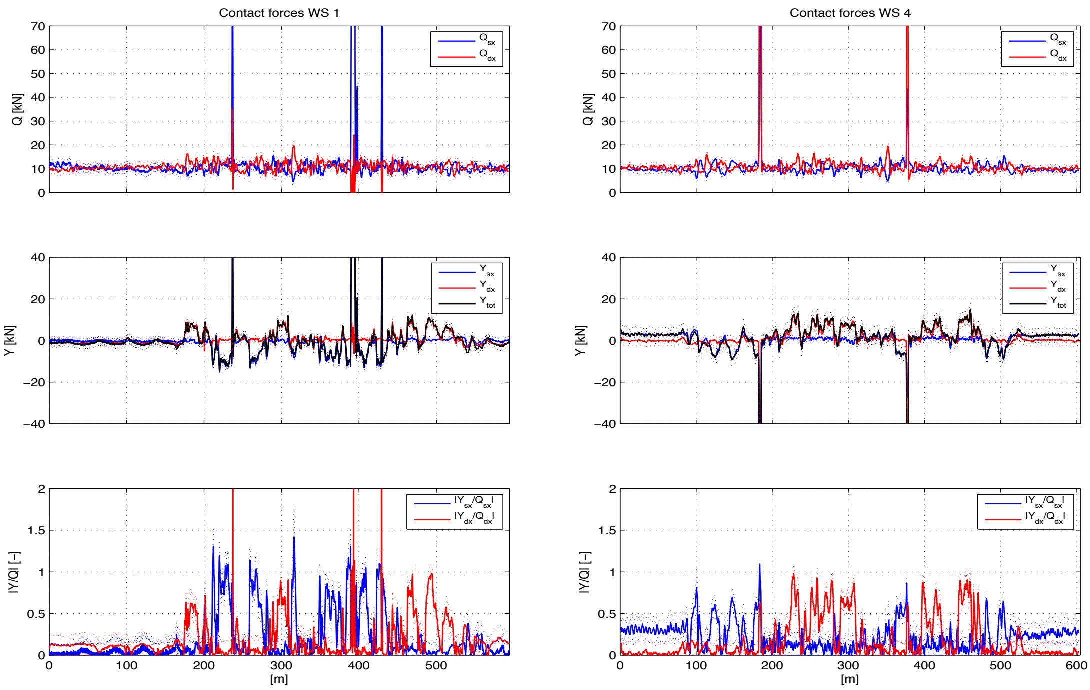
The qualitative behaviour of the vehicle is the same, also the values of Y and Q forces are of the same entity except for the exit of central double switch 29b-25b, where the vertical load transfer is hindered by the second bogie that is still in the full curve. The Y forces load the internal wheel because the bogie has not completed the steering manoeuvre. The Y/Q value is high (≈ 1.45) on this track but it's duration is very short, and it should not induce to a dangerous behaviour, although this situation should be avoided.

When the forth axle is the leading two vertical and lateral hits occur at the entrance of the double "S" switches, when the first bogie is already inserted in the second curve and the second bogie rotation is contrasted by the wagon that links the two bogies that are turned in the opposite direction. This switch configuration is the most critical for this kind of vehicle.

7.3.2.4 Test 153/148 - 30 km/h, R=170 m, Empty condition

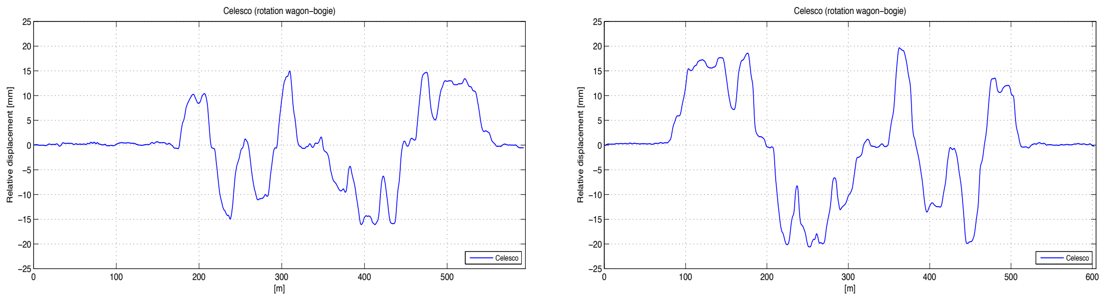
As for the lower speeds, the same tests have been performed in empty condition, in trailing and leading conditions. When the first axle is the leading one, some peaks of vertical and lateral forces at the exit of some switches occur, due to the lack of curvature transition which induce an impulsive acceleration, significant at this speed, causing a sudden load transfer (of small entity than in the load case) and a consistent lateral load transfer that increase the derailment factor.

On the forth wheelset wheels the lateral and vertical forces reach a lower level, but their ratio is approximatively around the unity because of the relative percentage of variation between them.



(a) $(Y/Q)_1$ (153)

(b) $(Y/Q)_4$ (148)



(c) Bogie-wagon relative displacement (153)

(d) Bogie-wagon relative displacement (148)

Figure 7.18. Y/Q forces (test 153-148)

As seen, when the wagon is empty the dynamic effects due to the track geometry are more significant, as expected, in deeds the vertical load transfer when the switch is crossed are numerically lower, but the relative diminution of the lateral forces is quite proportional. All the dynamic effects introduced by the lateral acceleration and the flange contact are more consistent. The relative displacement between the wagon and the bogie are identical to that reached in the loaded configuration which means that the vertical load is not affected by the vertical load. Another interesting thing is that when the wagon crosses the double switches, the opposite rotation of the bogies, due to the curve inscription, leads to peaks of lateral and vertical forces. This condition occurs because the flat-wagon constraints the rotation of the bogies generating these force peaks, that although don't increase the derailment coefficient value.

7.3.3 Lubricated tests

The "lubricated tests" terminology means that the friction pads surfaces are lubricated through the application of dumper grease,for the investigation of the influence of the elements that mostly affect the lateral running performances. The lateral pads have been lubricated lowering the below part in contact with the coil springs, using a pry and a paintbrush to apply the grease between the surfaces. The lubrication operation has been performed manually (figure 7.19). These actions are the most practical and reasonable to perform on a SAADKMS vehicle because they don't require any part substitution, and gives some useful indications on what could be the usual operations that could be performed during maintenance.

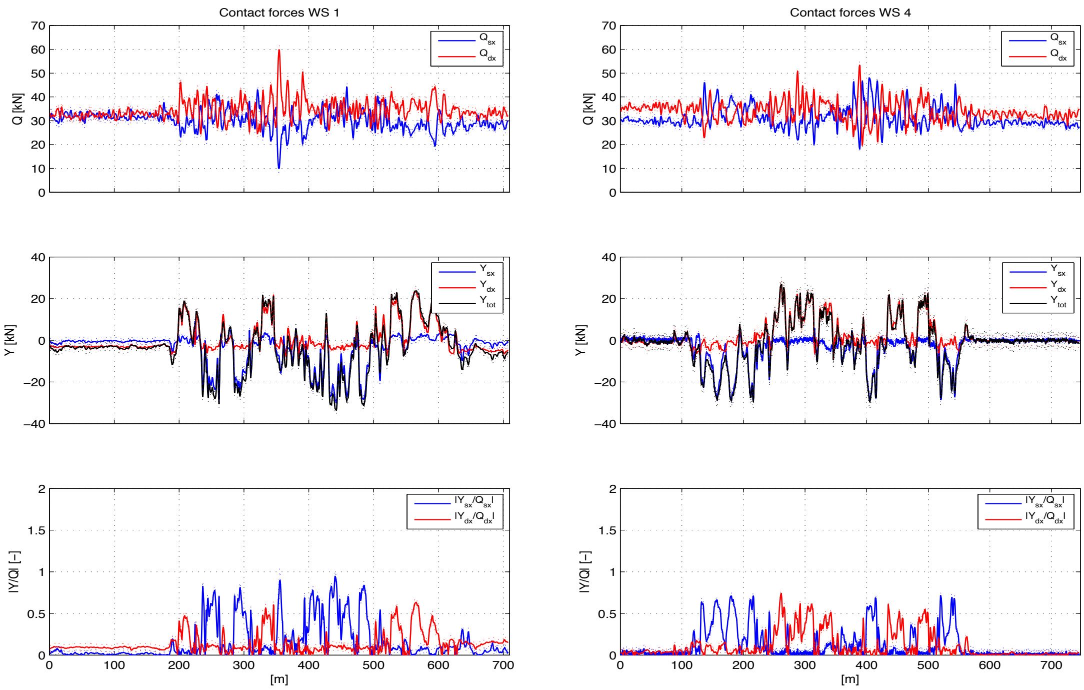
Figure 7.19. Lubrication operations

Most of the tests performed in "dry" conditions have been repeated with some reduction about the speed steps, and the traction conditions. The summary of the tests are reported in table 7.2.

Since the 30 km/h is the target speed, in order to evaluate the influence of the lubricated parts, the same tests at 30 km/h with respect to the dry condition are discussed in this section, remarking the differences observed.

7.3.3.1 Test 224/223-238/237 - 30 km/h, R=170 m, Load and empty condition

This condition is characterized by lower values of the derailment coefficient on the first wheel of the leading bogie, in facts, although the vertical load transfer are quite of the same entity with respect to the previous case, the lateral forces are about 15 % lower (figure 7.20). A different behaviour is also observable on the relative rotation of the bogie with respect to the wagon where the curve on the graph 7.21 is smoother both during the transitions from straight to curve and along the full curve. This dictates a lower reaction torque of the bogie, thus a lower entity of lateral forces when the vehicle approaches curves.



(a) $(Y/Q)_1$ leading WS1 (b) $(Y/Q)_4$ leading WS4

Figure 7.20. Y/Q forces (test 223-224)

When the forth wheelset is the leading of the second bogie, the peaks of acceleration along double switches take place again, indeed the influence of the first bogie inserted in the counter curve is critical also in this case because the flat wagon is such a mechanical link and the steering of the bogies is constrained; the reduced friction on the pads don't really improve the steering of the second bogie in this case.

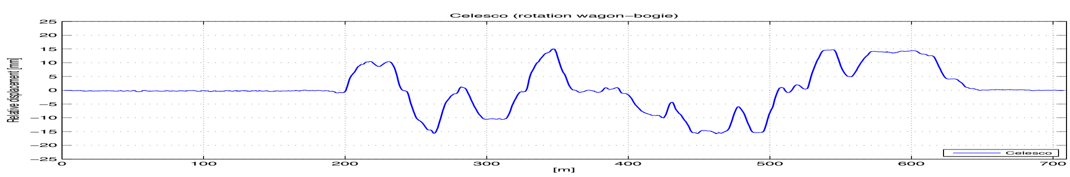


Figure 7.21. Other parameters (test 224)

The lubrication of the friction pads is not an usual operation performed during the maintenance of the vehicle, but it could be considered because of it's improvement of the steering performances of the wagon.

7.3.4 Results summary

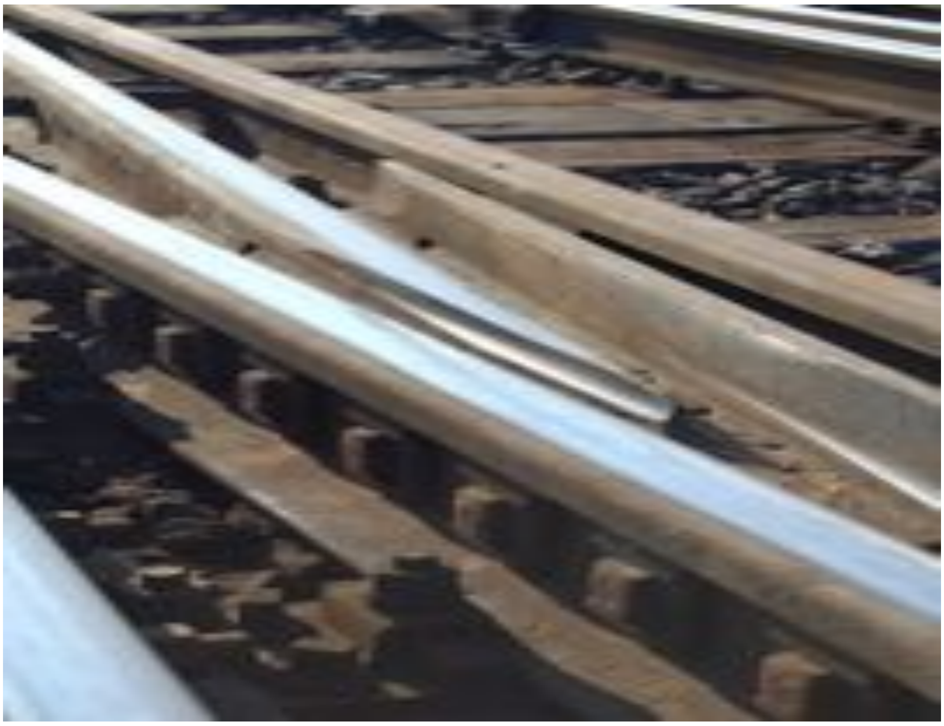
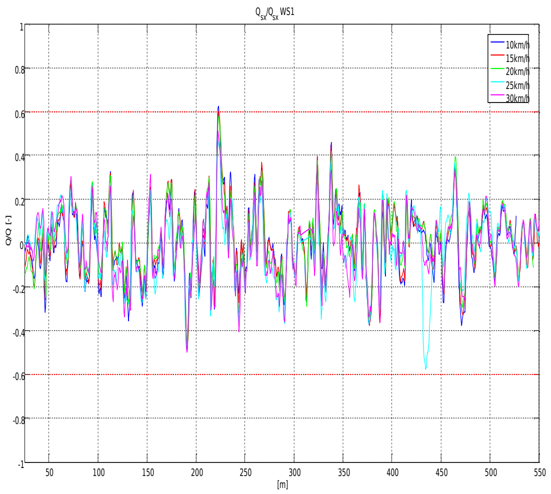


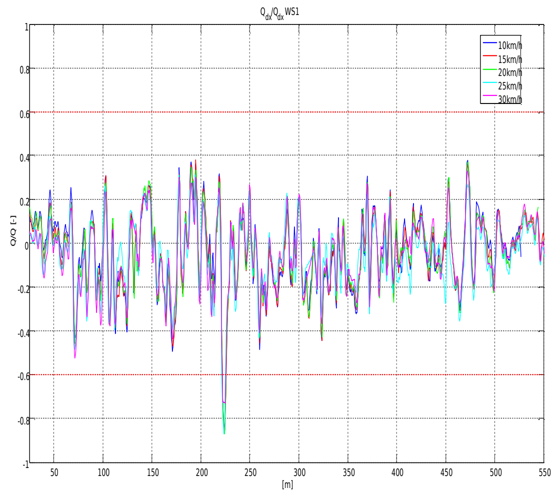
Figure 7.22. Wing rail wear

Since the amount of experimental evidences is huge, but a detailed analysis of some of the test conditions was due, a summary of the results about the most critical condition is reported. The most critical condition revealed is that with the empty wagon pulled. The results are relative to the wheels vertical unload and the derailment coefficient of the first wheelset.

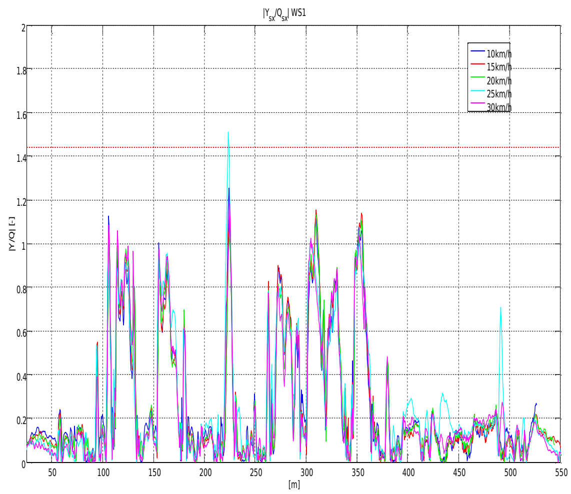
The trend of the derailment coefficient value, starting from 10 km/h, decreases till about 20 km/h when the dynamic effects generated on the wagon tend to reduce the vertical load transfer on the external wheel, rising the Y/Q value. From picture 7.23 the value of the derailment coefficient is always quite below the unit, except when the wagon cross the straight switch n° 23a-21b. Along this particular double switch routed in the straight direction a vertical overload of the left wheel occurs, probably for the particular geometry of such "english" switch that could be affected by an excessive wear of the wing rail (figure 7.22), bringing to a multiple contact of the wheel rolling surface and flange. Although a more accurate survey of the switch geometry should be performed, in order to attribute this critical behaviour to a specific cause, and to evaluate if it could reveal a systematic problem of this switch topology or due to a singular event on this switch.



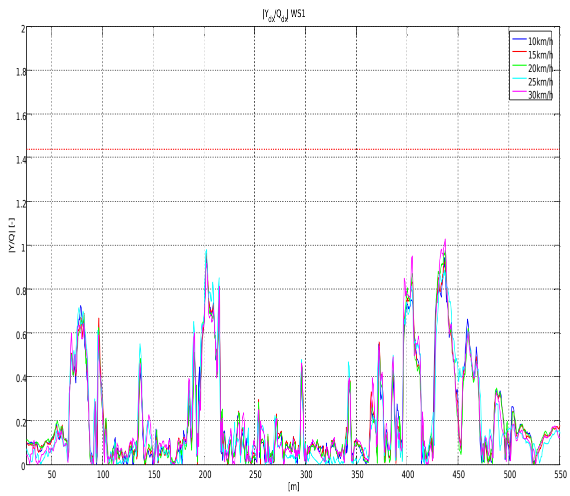
(a) $(\Delta Q/Q)_L$ WS1



(b) $(\Delta Q/Q)_R$ WS1



(c) $(Y/Q)_L$ WS1

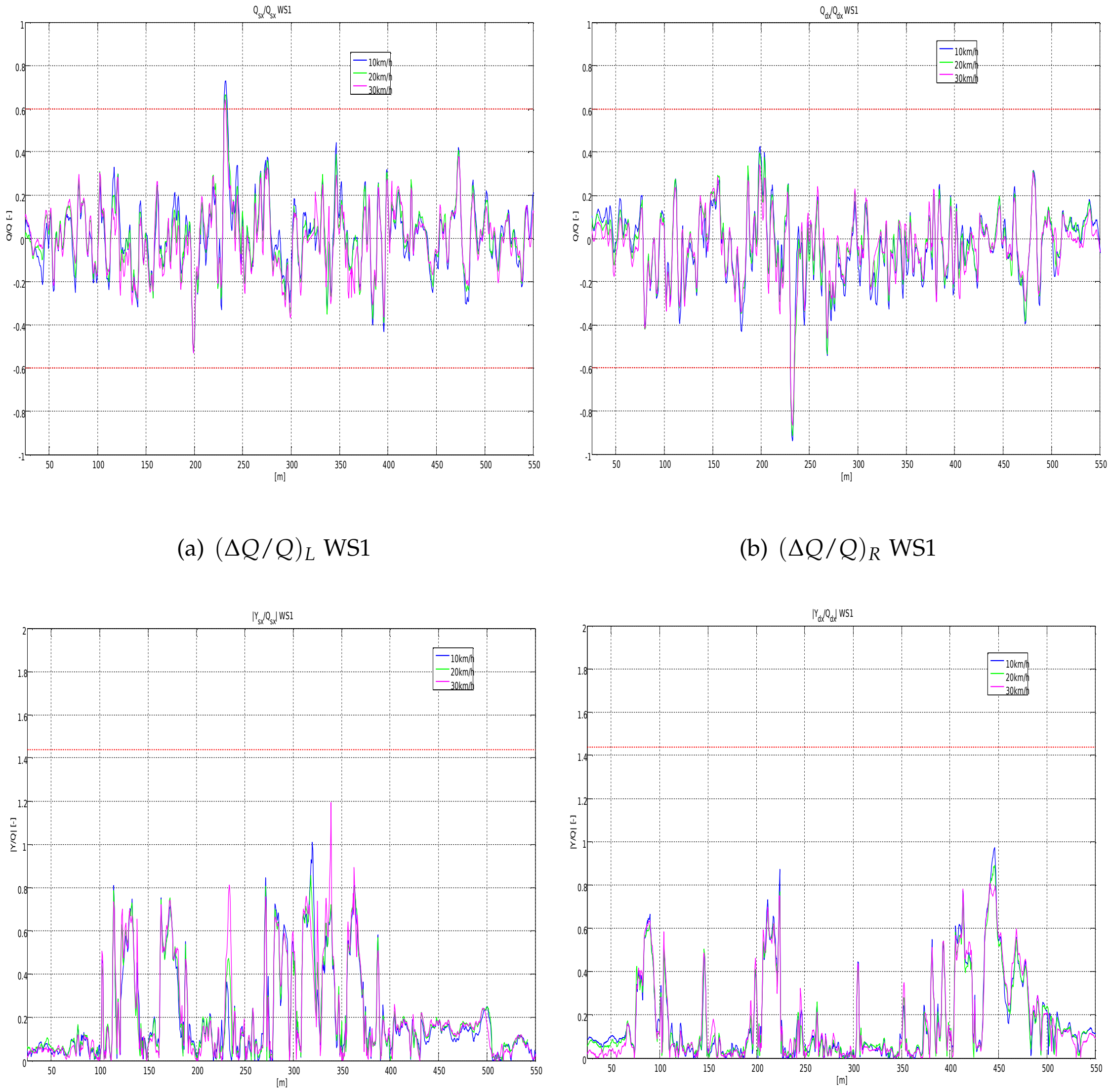


(d) $(Y/Q)_R$ WS1

Figure 7.23. $(Y/Q)_{WS1}$ 30 km/h

Except for that particular switch the derailment coefficient is always below the limit of 1.44 and the percentage unloading of the wheel below the 60%, a limit used for the static tests assessment. At the light of these values the SAADKMS vehicle could be considered safe when routing switches at 30 km/h.

The "lubricated" condition shows a consistent improvement of the performances of the vehicle in terms of forces exerted on the rail, the Y/Q ratio decreases because of the diminution of the reaction torque of the bogie rotational stiffness, while the vertical unload is less sensitive to the reduction of the friction coefficient at the pads interfaces, as reasonably expected, and its values are quite the same.



(a) $(\Delta Q/Q)_L$ WS1 (b) $(\Delta Q/Q)_R$ WS1

(c) $(Y/Q)_L$ WS1 (d) $(Y/Q)_R$ WS1

Figure 7.24. $(Y/Q)_{WS1}$ 30 km/h, lubricated pads

These results confirm that the lubrication of the pads surfaces is a good solution for the reduction of the derailment attitude of these type of wagon on switches. The wheel flange lubrication is a solution already adopted that undoubtedly improves the wagon behaviour in terms of lateral forces exerted and than of safety against derailment.

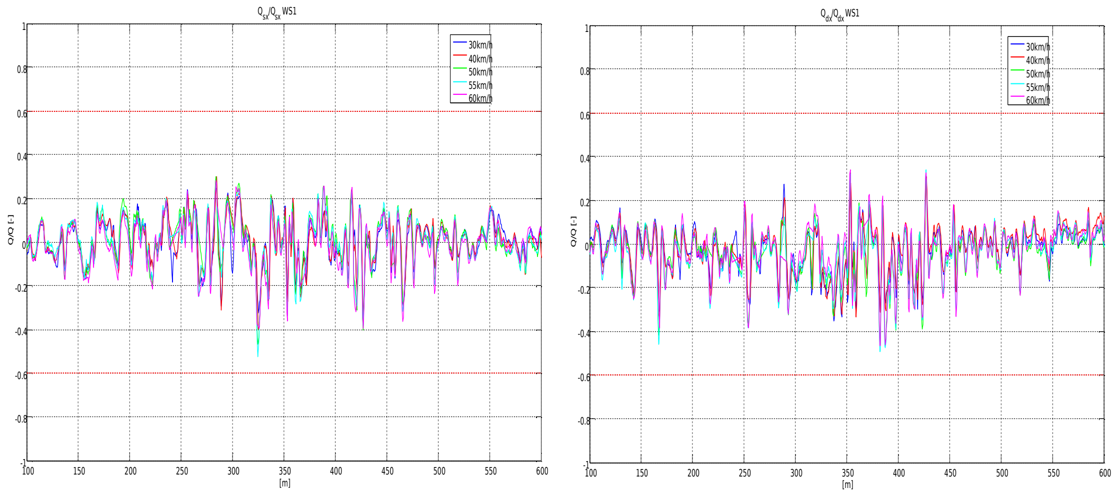
7.4 Switch at 60 km/h

For the tests on switches at 60 km/h, the route individuated inside the "Roncafort" plant is composed by two consecutive switches one to the right and one to the left with a radius of 400 m and a tangent to the straight track equal to 0.074. The test procedure is the same as used for the 30 km/h switches, the same route ran at different speeds, from 30 km/h to 60 km/h, with the following step increments: 10, 10, 10, 5, 5 km/h.

The behaviour of the wagon in terms of forces on the wheels is not much different varying the speed because of the larger radius of curvature that represents a less critical condition for the vehicle.

A synthesis of the results on the first wheelset in terms of $\Delta Q/Q$ and Y/Q is reported for the empty condition when the wagon is pushed, which results the most critical condition for this kind of convoy.

Observing the vertical unloading of the wheels, not any critical conditions occur, and the value is always below the 60 % along the whole track on the first wheelset (figure 7.25).



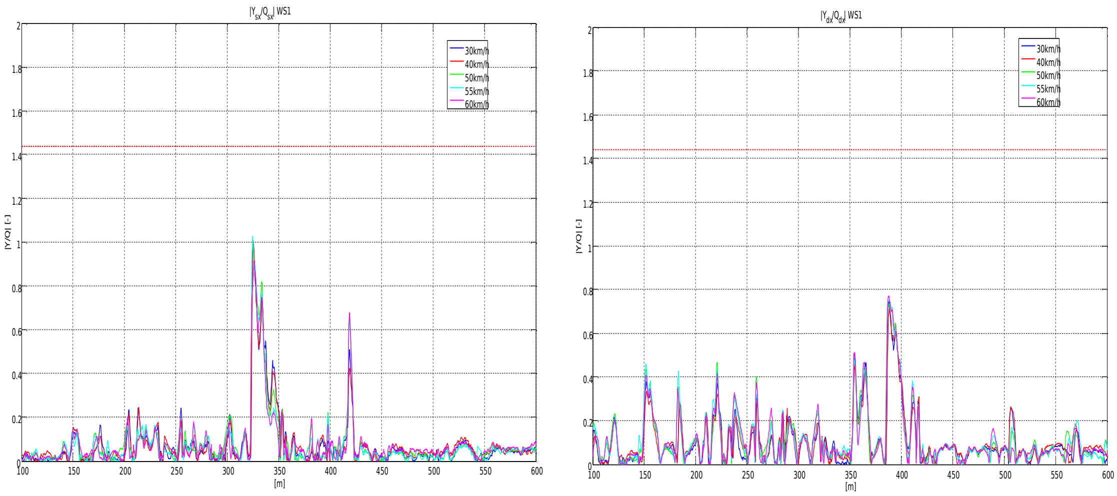
(a) $(\Delta Q/Q)_L$ WS1

(b) $(\Delta Q/Q)_R$ WS1

Figure 7.25. $(\Delta Q/Q)_{WS1}$ 60 km/h

Moreover the derailment coefficient is always below the unit and not a relevant speed influence is observed. When the first wheelset leads the bogie, along the first switch the maximum value reached is of about 1 at the entry of the switch, but decrease along the curve when the final asset is reached and the second bogie goes into the curve (figure 7.26)

As said this kind of track geometry doesn't represent a critical condition for these wagon.



(a) $(Y/Q)_L$ WS1

(b) $(Y/Q)_R$ WS1

Figure 7.26. $(Y/Q)_{WS1}$ 60 km/h

7.5 Conclusions

In this chapter a discussion about the factors that leaded to the definition of the test campaign for the assessment of the switch crossing behaviour of SAADKMS vehicles have been discussed and motivated. This kind of investigation has the aim to understand what are the phenomenons that occur while the vehicle route different type of switches in different configurations, and it's the most complete survey about this argument. All the results achieved don't induce any particular motivation to preclude the continuation of the test, with the long composition, because the safety parameters are, in general below the limits, except in some particular location of the route selected, that reasonably don't constitute a real danger for the normal manoeuvre operations and have to be attributed to some line singularities. Moreover the influence of the pads lubrication on the dynamic behaviour of the vehicle represents an improvement of the steering performances that lead to a lower level of lateral forces.

The procedure individuated for the tests permits to investigate both the general and the local behaviour of the wagon on switches realizing a complete campaign that can be used for multiple purposes making a relatively short experimental investigation. Moreover the fact that the tests have to be performed on a train is always to keep in mind, because the operations to perform on a vehicle, either the most simple, such as the inversion of the direction of the train, results logistically and operationally very expensive. This method could be adopted as a guide line for the switches investigations, because results more complete and don't need a special infrastructure as characterization tests.

The continuation of the tests has been discussed with ANSF, that has been fully satisfied about the decisions regarding the tests procedure and the measurement performed. Thus the rules for the continuation of the tests in long configuration, in pulled condition, with braking efforts along the line in specific positions of the test routes shall be decided and discussed in the following chapter.

CHAPTER 8

CONCLUSIONS

The work at the centre of this thesis, although it refers to a project began many years ago, is extremely actual and tackle problems never faced in a systematic way till now. A recent update presented by the German group at the international CEN committee "WG10" about the rolling stock dynamic, and introduced in the draft of the next EN14363, concern the tests on switches for vehicles with small diameter wheels. In the draft version it's included in an informative annex E, and individuates an unique track where performing the tests. The switch individuated has a well defined geometry, a radius of 450 m and other quotes defined in the leaflet. However this kind of geometry is related to the tests performed by DB and the sentence *"The specified set of data is related to the DB type of crossing described in UIC 510-2. For assessment of vehicles with small wheels in other types of crossings modified parameters should be used."* means that also other types of track geometries could be considered. The other indications about the tests are that the speed should be constant in order to minimize the traction efforts influence on the dynamic of the vehicle, the second is that the switch shall be routed by consequently speed steps.

The assessment values are about the **H** force of the guiding wheelset or the the lateral forces on its wheels, the other method is about the measurement of the nose impact through a paint on the nose entrance.

The assessment limit values for the values of the guiding forces on a wheelset are:

$$\sum Y_{lim} = 0,25 \cdot P_0 + m0 \cdot I \cdot g/2b_A$$

where P_0 is the weight of the wagon, I is the cant deficiency, g is the gravity, b_A is the wheels contact points distance and m_0 is the mass on the wheelset. Since the cant shall be zero on the switches, the limit of the guiding forces is $1/4$ of the total weight (4 axles vehicle).

The guide lines defined in the course of this thesis are pertaining to that prescribed in the new draft of the norm but with different goals. In deed for such test campaign the experimental results could not drain in a single test switch, because of the variety of scenarios that the vehicle can encounter, trying to analyse most of them.

Thus such methodology is focused on the determination of the behaviour of the vehicle in a large number of switches succeeding in different combinations, but taking into account of the logistical and practical issues that could permit a bearable test campaign.

For this reason the limit value used is defined on the basis of the Nadal's criterion opportunely calculated on the basis of the geometry of the wheel, and verified by experimental and numeric data as defined during the calibration tests reported in the [1].

Moreover, in the new draft of the leaflet there isn't any indication about the test condition of the mechanical components, except for the pneumatic suspensions if present. In this case the influence of the friction pads on the dynamic of the vehicle was suspected of primary importance, thus also a testing campaign intervening on these parts has been set up.

All the measured quantities had been defined in order to achieve all the informations on the dynamic of the parts of the vehicle, the semi-bogie, the bogie, the friction pads and also the accelerations at the extreme parts of the primary suspensions with the future purpose to determine the dynamic transfer function of the rubber elements. The motions

The only quantity that has not been possible to measure continuously is the angle of attack, because of the lack of fixed parts on the axleboxes that don't have any cap or parts where link an instrument for such measure. This quantity would have been the last measure for the reconstruction of the full wagon lateral dynamic and rotation in the horizontal plane, moreover it could have added an important information on the local dynamic of the wheelset. At support of the validity of the method proposed is the fact that this test campaign is a mix of the "on-track" tests and the "specific" tests as prescribed in the [1], and the angle of attack is not a required measure while the train is running.

The continue measure of this quantity should be prescribed in the final part of the new version of the leaflet, for a general vehicle when possible.

This test campaign have to be extended to the "long convoy" composed by 21 vehicles, under the actual load specifics, where the extreme cars shall be loaded. The instrumented car shall be placed in the ninth position along the convoy, because from simulations results the more critical in relation to the braking manoeuvre when the adjacent vehicles are loaded and the instrumented unloaded. This configuration leads to a delay in the pneumatic signal to the brakes along the convoy and when the previous car is loaded, an "accordion effect" occurs and the measure vehicle is squeezed between the two halves of the convoy and, if in curve the derailment is aid [16]. In long composition both the conditions are going to be investigated, braking and coasting, varying the position along the route where the braking occur in order to evaluate the influence of the switch geometry on the wagon forces.

This part of the tests has already been authorized by ANSF at the light of the results in terms of procedure and measures reported in the course of this thesis.

Finally the procedure defined in this work is going to be presented in the EN work-group on the rolling stock dynamic, for its evaluation and inclusion in the informative annex of the next leaflet version 8.

ANNEX 1

Annex E
(informative)

Assessment of the behaviour of vehicles with small wheels in curved crossings

E.1 Purpose

In crossings, unguided gaps exist, depending on crossing angle, height of check rails, wheel diameter and flange height of the wheel. In some European networks obtuse crossings are located also in curves. In such a situation lateral forces on the wheelset can occur, leading to a risk of misdirection of the wheelset. This risk can be described by the angle of attack and the lateral forces, or by the angle of attack at the entrance to the gap and the impact of the wheel on the crossing nose.

The assessment conditions are based on a description developed by ERRI (see Bibliography /UIC 510-2/ and /ORE-C9/).

E.2 Area of application

Investigations according to this specification shall be performed only for vehicles

— with minimum permitted wheel diameters $\boldsymbol{d} \leq 840$ mm

— which are intended to be operated in specific networks with obtuse crossings located in curves

NOTE It is assumed that the following relation between flange height and wheel diameter is respected.

Minimum permitted wheel diameter	Nominal flange height
$\boldsymbol{d} > 760$ mm	$\boldsymbol{S_h} \geq 28$ mm
$630 < \boldsymbol{d} \leq 760$ mm	$\boldsymbol{S_h} \geq 30$ mm
$\boldsymbol{d} \leq 630$ mm	$\boldsymbol{S_h} \geq 32$ mm

E.3 Description of the crossing geometry

The vehicle's guiding behaviour is assessed for curved crossings with the parameters given in the following table (see also figures E.1 and E.2):

Table E.1 — Parameters of curved crossing

Curve Radius	$\boldsymbol{R}$ = 450 m
Cant	$\boldsymbol{u}$ = 0 mm
Crossing angle	α = 1:9
Track gauge of the crossing (see dimensions $\boldsymbol{A_1}$, $\boldsymbol{A_2}$, $\boldsymbol{A_3}$ and $\boldsymbol{A_4}$ in figure E.1)	$\boldsymbol{TG} = 1435_{-2}^{+4}$ mm
Flangeway clearance	40_{-4}^{+4} mm
Dimensions for nose protection (see dimensions $\boldsymbol{C_1}$, $\boldsymbol{C_2}$, $\boldsymbol{C_3}$ and $\boldsymbol{C_4}$ in figure E.1)	1395_{-2}^{+3} mm
Dimensions for running clearance (see dimensions $\boldsymbol{B_1}$ and $\boldsymbol{B_2}$ in figure E.1)	≤ 1356 mm
Location of measurement (see dimensions $\boldsymbol{a_1}$, $\boldsymbol{a_2}$, $\boldsymbol{a_3}$ and $\boldsymbol{a_4}$ in figure E.1)	750 mm
Location of measurement (see dimensions $\boldsymbol{b_1}$, $\boldsymbol{b_2}$, $\boldsymbol{c_1}$ $\boldsymbol{c_2}$ $\boldsymbol{c_3}$ and $\boldsymbol{c_4}$ in figure E.1)	80 mm
Height of check rail above rails	$\boldsymbol{h_{cR}} = 45^{+10}$ mm
Height reduction of the nose over a length of 200 – 500 mm approximately (see dimension $\boldsymbol{y}$ in figure E.2)	$\Delta \boldsymbol{h_n}$ = 8 mm approximately
Width reduction of nose over a length of 150 mm approximately (see dimension $\boldsymbol{x}$ in figure E.2)	$\Delta \boldsymbol{b_n}$ = 3 mm approximately

NOTE The specified set of data is related to the DB type of crossing described in UIC 510-2. For assessment of vehicles with small wheels in other types of crossings modified parameters should be used.

E.4 Test conditions

E.4.1 General

There are two methods specified in the following chapters to demonstrate safe behaviour of vehicles with small wheels in curved crossings according to E.3. Only one of the two methods is required to be carried out.

The tests shall be conducted with the empty vehicle in a curve with a nominal radius of 450 m with an installed cant of 0 mm. They shall be conducted with different speeds between 10 km/h and the vehicle's maximum service speed up to a maximum of 60 km/h, the test with the highest speed shall be performed three times.

The tests shall be performed with minimum traction forces to ensure a roughly constant speed and be repeated also in failure conditions unless the failure study proves that there is no negative effect.

The back to back distance and the flange thickness of the outer guiding wheel shall be documented.

The wheel diameter shall be between the nominal diameter $\boldsymbol{D}$ and the minimum permitted diameter $\boldsymbol{d}$.

E.4.2 Method 1: Lateral forces and angle of attack

In Method 1 the assessment shall be based on the angle of attack and either the lateral axle-box forces $\boldsymbol{H}$ or the sum of the lateral wheel forces $\Sigma \boldsymbol{Y}$ in the full curve. The $\boldsymbol{H}$-forces and $\Sigma \boldsymbol{Y}$-forces (if measured with instrumented wheelsets) shall be processed as specified in chapter 6.6.2.1. Alternatively it is possible to measure the ΣY-forces with track mounted equipment.

NOTE In this method it is not necessary to have physical crossing installed in the curve.

The running clearance between wheel and rail shall be within the range 10 mm <= (TG - SR) <= 20 mm during the operation of the vehicle in the full curve. The test shall be performed in dry rail condition to respect high lateral forces in the assessment.

E.4.3 Method 2: Examination of the impact on the crossing nose

In Method 2 the assessment shall be based on the angle of attack and the examination of the impact on the crossing nose. The assessment of the impact of the wheel on the nose should be done using paint to visualise the running path of the wheel in the relevant section of the crossing.

In that case it is necessary to assess the vehicle in the gap given by the crossing geometry above in wet and dry rail conditions to include the effect of different friction conditions.

E.4.4 Limit values

The limit value for the axle box H-forces is specified depending on the nominal axle load P_0:

$H_{y,lim} = 0{,}25 \cdot P_0$

The limit value for ΣY-forces – measured either by local measuring points or by instrumented wheelsets – is:

$\Sigma Y_{lim} = 0{,}25 \cdot P_0 + m_0 \cdot I \cdot g / 2b_A$

Table E.2 gives the limits for the angle of attack depending on the minimum permitted wheel diameter

Table E.2 — Limit values for angle of attack α_{lim}

Minimum permitted wheel diameter	Maximum angle of attack α_{lim}	For information: Required flange heights (see table xxx)
840 mm ≥ d > 760 mm	15,6 mrad	Sh ≥ 28 mm
760 mm ≥ d > 680 mm	18,5 mrad	Sh ≥ 30 mm
680 mm ≥ d > 630 mm	17,9 mrad	
630 mm ≥ d > 550 mm	18,1 mrad	Sh ≥ 32 mm
550 mm ≥ d > 470 mm	16,8 mrad	
470 mm ≥ d > 390 mm	15,0 mrad	
390 mm ≥ d ≥ 330 mm	14,6 mrad	

There is no specific limit value defined for the assessment of the impact on the nose.

E.4.5 Assessment

There are two different combinations of assessment criteria for the vehicle's behaviour using the respective limit values given in E.4.4. Only one of the combinations is required for the acceptance.

1) Angle of attack and ***H***-forces or ***ΣY***-forces

2) Angle of attack and examination of the impact on the nose

Alternatively an assessment using the ***H***-forces or ***ΣY***-forces together with the information about the impact on the nose is possible.

For the assessment of the impact on the nose there should be no abrasion of paint on the tip of the nose, but only on the nose flank.

Instead of ***H***-forces or ***ΣY***-forces measured in the test curve or test crossing, results from on-track tests under similar test conditions (Curve Radius around 450 m, cant deficiency around 94 mm, dry rails) may be used for comparison with the limit values.

E.4.6 Dispensation

If there is an already tested Reference Vehicle, dispensation shall be granted to a vehicle, if

— the results of the Reference Vehicle were 10% below the limit values and

— the modifications of the following parameters remain in the ranges given in Table L.1 for dispensation from on-track testing.

  — Distance between bogie centres / vehicle wheel base

  — Secondary Suspended Mass (vehicle tare for freight stock)

  — Bogie wheel base

  — Axle guiding

  — Yaw resistance of bogie

  — Moment of inertia of whole bogie

Dispensation shall also be granted for minimum permitted wheel diameters 760 mm ≤ ***d*** < 840 mm if a minimum permitted flange height of

$$S_{h.min} = 32mm + \frac{26mm - 32mm}{840mm - 760mm} \times \; d - 760mm \; = 89mm - 0.075 \times d$$

is applied.

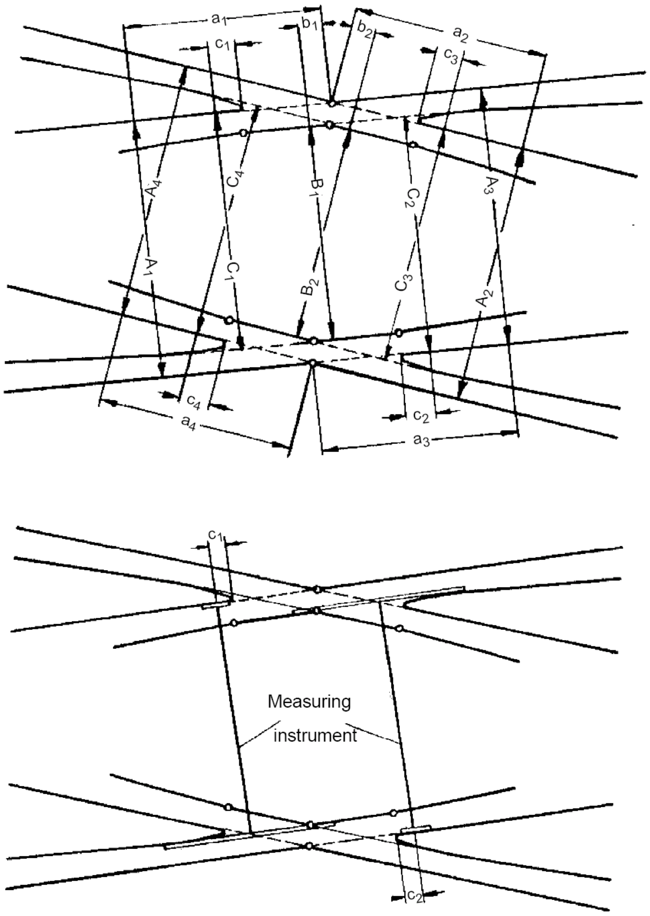
E.4.7 Simulation

The proof of running safety in a curved crossing may be performed also by simulations in a full curve without crossing using a validated model (see Annex K). Therefore ***H***-forces (or ***ΣY***-forces) and angles of attack shall be calculated for a 450 m-curve without cant for a running clearance between wheel and rail within the range 10 mm <= (***TG*** - ***SR***) <= 20 mm.

The friction coefficient shall be varied between 0.05 and 0.45 in steps of 0.1. The following speeds shall be investigated: 10 km/h, 25 km/h, 40 km/h and 60 km/h.

The assessment shall be made against the criteria given in E.4.4.

Measuring plane located 0,014 m beneath the running surface



a_1, a_2, a_3, a_4 = 0,750 m
b_1, b_2, c_1, c_2, c_3, c_4 = 0 - 0,080 m

Figure E.1 — Geometry of curved crossing for testing of running safety

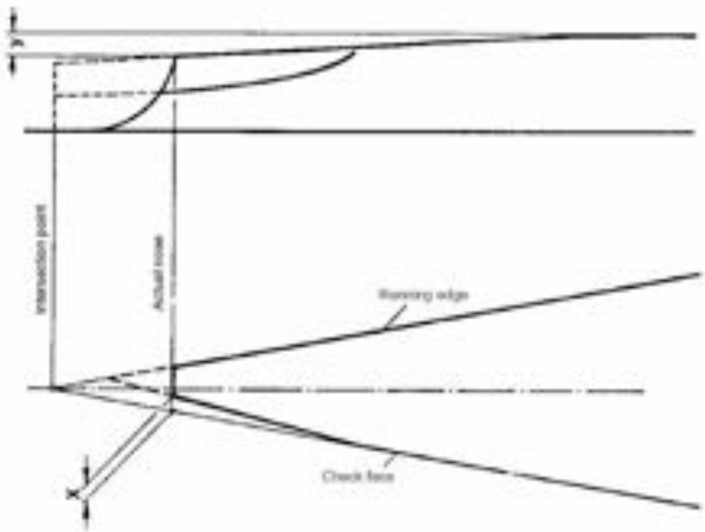


Figure E.2 — Detail of the crossing nose

Replace y by Δh, and x by Δb,

Figure E.3 — Example for curved crossing (DB Systemtechnik, Minden)

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