DELIVERABLE

D0602. FINAL CONSOLIDATED REPORT - CHAPTER 5

Related Milestone

031312
FP6-31312
ACRONYM
URBAN TRACK
TITLE
Urban Rail Transport
PROJECT START DATE
September 1, 2006
DURATION
48 months

Subproject
SP6
Work Package
WP6.2

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Date of issue of this report
15 November 2010

Project funded by the European Community under the SIXTH FRAMEWORK PROGRAMME PRIORITY 6 Sustainable development, global change & ecosystems

TIP5-CT-2006-031312
URBAN TRACK
Issued: 15/11/2010

Quality checked & approved by project co-ordinator André Van Leuven
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5 FUNCTIONAL REQUIREMENTS FOR URBAN RAIL TRACKS

The following chapter 5 is presenting Sub-project 5 (SP5) targeting functional requirements for Urban Rail tracks.

SP1 and SP2 – see chapters 1 and 2 - focused respectively on new track construction and on track renewal or maintenance. The purpose of SP5 performed by D2S, Alstom, TTK and Polimi was to identify where the axes are for improvement and for further development of track components, construction methodology and system design, and to set the basis for further evaluation of improvements and development that resulted from SP1 and SP2 and which have been validated in SP3 (see chapter 3).

The sub-project 5 was made of two work packages:

- WP5.1, presented in chapter 5.1, was targeting the definition of functional requirements and more especially the determination and analysis of the forces applied to the track and the subsequent stress and displacement in rail. Another objective was to assess the influence of the track condition (geometrical and mechanical) on the vehicle dynamic and the subsequent impact on stress in rail.

- WP5.2, presented in chapter 5.2, concentrated on the definition of functional specifications regarding Tram, Light Rail and metro tracks. This objective could be achieved through the following approach:
  - by defining the significant parameters for identifying the level of duty conditions (loads): track characteristics and conditions, vehicle characteristics and conditions, traffic type and density.
  - by identifying the range of duty conditions that exists on segments of the track infrastructure of participating networks taking into account the influence of vehicle characteristics and condition and traffic density.
  - by selecting, through consultation with network operators, those duty conditions and track parameters that could provide the most marked decreases in LCC through the implementation of improved products and processes. In addition to the use of improved products in track renewals, the objective was also to increase the residual life of existing track through the use of better maintenance practices.

Updated functional specifications for a track infrastructure in a new network were made available, also based upon:

- sensitivity analysis and parametric studies which evaluate the influence of track parameters (such as gauge, gauge widening, rail type, rail support stiffness, gauge tolerance, …) on track stability, on noise & vibrations, on construction costs and on LCC in general;
- the findings in ongoing EC research projects such as CORRUGATION, INMAR, TURNOUTS, SPURT, MODTRAIN, WIDEM, SILENCE, QCITY.

At the end of the sub-project, it has also been decided to carry out a thorough analysis of existing railway standards: since track alignment, track irregularities are defined in railway standards primarily from mainlines networks, the purpose was to review these standards and to point out their relevance regarding urban rail systems, as well as the needs for adaptation in order to better cope with urban rail
requirements. This analysis is presented in separate EXCEL files and the outcomes were used as an input for the works of the Urban Rail Platform set up by UITP and UNIFE to develop recommendations for technical harmonisation of urban rail systems throughout Europe.

5.1 **DEFINITION OF FUNCTIONAL TRACK REQUIREMENTS FOR TRAM & METRO**

Within WP5.1, the work was developed through models and simulations on one hand and site measurement on the other hand, for both tram and metro tracks. Vehicle dynamic simulations were carried out based on multi-body method in order to have a complete view of the track response, to assess the influence of passing vehicles and to evaluate the sensitivity of the wheel rail interaction for various track systems. The subject of rail stress has been discussed in terms of fatigue behaviour under repeated cyclic loading. The dynamic forces were generated by railway vehicles and also by track irregularities. Fracture mechanics was applied to assess the behaviour of the rail. A new method for simulating a high number of passes was implemented.

(a) **CONCLUSIONS ON TRAMWAY TRACK FORM**

A wide range of subjects was dealt with in the present document, all of them related to the forces applied to the track and the resulting stresses in rail, and leading to conclusions on track geometry, track roughness and vibration mitigation.

Complete models including track and vehicles were setup allowing for simulations taking into account the actual conditions of the track.

(i) **Track geometry**

The track geometry was studied in relation with its mechanical response and the resulting forces and stresses undergone. Following the measurements in Madrid of track for gauge, vertical alignment, and cant and horizontal alignment, an approach based on 95% compliance can be applied to conclude that the track V1 of the ML-3 line is compliant to the EN 13231-1 standard acceptance tolerances for these track parameters.

The results of the track geometry study can be proposed as a contribution for the development of urban track standards.

(ii) **Rail fatigue**

The rail fatigue was studied with two practical purposes: to be able to make a commitment on the lifetime of the rails installed and to optimise the rail replacement frequencies for the track maintenance.

The development of that study was the opportunity to implement the stationary method, which enables to undertake fast simulation of a high number of cycles.

(iii) **Track roughness**

The track roughness measured on Madrid network is particularly good in comparison with what could be found on some other networks.

The reason for this good track roughness is somewhat difficult to find even after analysis of measurement collected.
It is difficult to draw a conclusion at the moment.

(iv) Vibration mitigation

The approach pertaining to vibration mitigation has to consider the whole vibration transmission path and to be adapted to the vehicle that will be operated.

And the vibration performance of a transportation system must be defined in relation to its compliance with a level of vibration measured at building locations and not by empirical rules based on outdated data.

The measurements made on many other sites for several tram designs and manufacturers are correlating what could be observed in Madrid:

- A very high performance system (floating slab) is rarely required,
- A continuously supported rail system is sufficient in many cases.

(b) CONCLUSIONS ON METRO TRACK FORM

The study of the Madrid metro track form based on the REMS system was finalised following the calibration and adjustment phases carried out during laboratory testing.

5.1.1 Modelling and simulations

The determination and analysis of track forces and rail stress data addresses two types of track forms: tramway and metro.

5.1.1.1. Tramway Track form

As a basis, a typical description of the tramway track form was provided. Then modelling and simulation works of selected tramway track forms were carried out.

The linear dynamic response, under a permanent regime, of a multi-layer half space subjected to a mobile load applied to its free surface, was studied. The purpose was to validate the quasi-static approach, as the influence of speed is not major in urban railway context.

Following this validation, two modelling approaches were investigated:

- Semi Analytical
- Numerical

The results were quite similar between the two approaches with an advantage for the semi-analytical as the simulation time is much reduced.

5.1.1.2. Metro Track form

The first step in the study of the metro track form consisted in setting-up the mathematical model and in particular the train-track interaction model and the wheel-rail contact model.

The metro of Madrid is the reference of the current study. Vehicle and track characteristics were input in the models.

Finally the numerical results were output. Displacements and forces on track were computed.
5.1.2 Contact forces on tramway track

5.1.2.1. Introduction

The rails installed on a track typically undergo forces of different nature: static, quasi-static (e.g. in curves), and dynamic.

The current section will focus on quasi-static and dynamic forces as they are in most cases at the origin of rail degradation.

The source of these forces lies, on one hand, in the dynamic behaviour of the rolling stock and, on the other hand, in the track and rail conditions.

The phenomena occurring within the wheel-rail contact area because of geometrical imperfection of wheel or rail and because of resonance phenomena due to the various stiffnesses involved in the wheel suspension and in the rail supporting.

One of the roles of the track is to guide the rolling stock so that it follows a determined path, which is necessarily made of straight alignment, transitions and curves. These two latter types of section induce a dynamic behaviour from the rolling stock.

The study developed below will focus on the influence of track alignment on train dynamics with the objective of determining the technical-economical optimum for track geometry and tolerances and for track alignment.

5.1.2.2. Track irregularities and contact forces

The objective of this study is to analyse the impact of the track irregularities on the dynamic behaviour of the rolling stock (tramway). Four track geometry parameters are particularly under interest: The gauge, the alignment (or horizontal level), the top (or vertical level), and the cant.

Track geometry irregularities used for the entire analysis are extracted from in-field measurements. All measured irregularities are then numerically increased or decreased. The variation of these four parameters is chosen in order to check whether EN 13231-1 (new construction) is applicable to urban transportation such as tramways.

All parameters are analysed individually, in other terms no combination of defects has been studied.

For each parameter and for each numerical configuration, safety criteria, passenger comfort and wheel/rail contacts loads are assessed and analysed with respect to the amplitude of track geometry irregularities.

The results of this study allow confirming the applicability of EN 13231-1 tolerances used for the track works of a new line in an urban context and with the Citadis tramway.
(a) **NUMERICAL MODEL VALIDATION**

The numerical model used for the study has been correlated from measurements done on a French network. The validation process is the same as the one described in section 5.1.2.3.b.

(b) **GAUGE IRREGULARITIES VS. CONTACT FORCES**

The initial gauge defects extracted from measurements are displayed in the picture below.

![Figure 5.1.1 Measured gauge irregularities](image1)

This defect is then amplified and/or lowered as shown in the picture below:

![Figure 5.1.2: Amplification/Reduction of gauge irregularities amplitude](image2)
As can be seen in the above picture, the case where initial measurements are multiplied by 2 is, in certain areas above the limit given in EN 13231-1. Other cases are mainly compliant with gauge irregularities tolerances of EN 13231-1.

As previously mentioned, the followings items are further analysed: Safety parameters, Lateral and Vertical dynamic loads at wheel/rail contact, maximum lateral and vertical accelerations

(i) Safety Parameters:

Safety parameters are derailment parameter \((Y/Q ; Y, Q \text{ respectively the transversal ans vertical load}),\) Prud’homme criteria \((\Sigma Y_{2m})_{\text{limit}} = \alpha(10+P_0/3)\), dynamic wheel unloading \((dQ/Q ; dQ \text{ the dynamic vertical load})\). The safety parameters are checked for each dynamic calculation with respect to the criteria given in the table below.

<table>
<thead>
<tr>
<th>SAFETY PARAMETERS</th>
<th>CRITERIA</th>
</tr>
</thead>
<tbody>
<tr>
<td>(Y/Q)</td>
<td>(Y/Q \leq 0.94)</td>
</tr>
</tbody>
</table>

Prud’homme

\[ \Sigma Y_{\text{limit}} = \alpha \left(10 + \frac{P}{3}\right) \]

\(\alpha = 1\)

\(P = \text{axle Static Load}\)

\(dQ/Q\)

\(dQ/Q \leq 0.6\)

Table 5.1.1 Safety Parameters - Criteria

As can be seen in the table 5.1.2, the safety parameters are in any case under the limit given above.

<table>
<thead>
<tr>
<th>Increasing Gauge</th>
</tr>
</thead>
<tbody>
<tr>
<td>Safety Parameters: Maximum Values calculated</td>
</tr>
<tr>
<td>(Y/Q) Max</td>
</tr>
<tr>
<td>----------</td>
</tr>
<tr>
<td>0.04</td>
</tr>
<tr>
<td>0.03</td>
</tr>
<tr>
<td>0.05</td>
</tr>
<tr>
<td>0.05</td>
</tr>
<tr>
<td>0.11</td>
</tr>
</tbody>
</table>

Table 5.1.2 Safety Parameters - Maximum Values
(ii) **Dynamic loads on the wheel/rail contact:**

The picture below displays the influence of track gauge irregularities on transversal loads.

![Figure 5.1.3: Effort Y - Influence of gauge irregularities](image1)

As can be seen, only the last case (i.e. initial times 2, this case corresponds to the case where gauge is above the limits specified in EN 13231-1) gives rise to high levels of lateral loads.

The picture below displays the influence of track gauge irregularities on vertical loads.

![Figure 5.1.4: Loads Q – Gauge irregularities](image2)
As can be seen, there is no influence of track gauge irregularities, in the variation range studied, on the vertical loads.

(iii) Lateral and vertical accelerations:

Only the results calculated at axle levels are shown because the values are more significant.

The picture below displays the influence of track gauge irregularities on axle lateral rms acceleration.

![Axle Lateral rms acceleration: Influence of Gauge Irregularities](image)

Figure 5.1.5: Lateral rms acceleration - Influence of gauge irregularities

As can be seen, only the last case (i.e. initial times 2, this case corresponds to the case where gauge is above the limits specified in EN 13231-1) gives rise to high levels of lateral accelerations.

The results obtained for the axle vertical rms acceleration show that, as per vertical load, the gauge irregularities have almost no influence on this parameter.

The same analysis is performed for all other parameters; only the main results are given in following sections.
(c) ALIGNMENT IRREGULARITIES VS. CONTACT LOADS

The initial alignment irregularities extracted from measurements are displayed in the picture below (black curve). Other curves correspond to reduction (green curve) or amplification (other curves) of the initial alignment signal.

![Amplification/Reduction of Alignment Irregularities Amplitude](image)

As shown in the picture above, the cases where initial measurements are multiplied by 2.6 and 2.9 are, in certain areas above the limit given in EN 13231-1. Other cases are fully compliant with alignment irregularities tolerances of EN 13231-1.

As per gauge irregularities, followings items are further analysed: safety parameters, lateral and vertical dynamic loads at wheel/rail contact and maximum lateral and vertical accelerations.

(i) **Safety Parameters:**

The table 5.1.3 below shows the main results for safety parameters:

<table>
<thead>
<tr>
<th>Safety Parameters: Maximum Values calculated</th>
<th>Alignment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Y/Q Max</td>
<td>dQ/Q</td>
</tr>
<tr>
<td>0.16</td>
<td>0.1</td>
</tr>
<tr>
<td>0.09</td>
<td>0.09</td>
</tr>
<tr>
<td>0.34</td>
<td>0.15</td>
</tr>
<tr>
<td>0.45</td>
<td>0.23</td>
</tr>
<tr>
<td>0.51</td>
<td>0.27</td>
</tr>
</tbody>
</table>

Table 5.1.3: Safety parameters - Maximum values calculated

All Safety parameters are under the criteria. So no alignment configuration tested is critical.
(ii) Dynamic loads on the wheel/rail contact:

c.ii-1 Lateral Loads:

The picture below displays the influence of track alignment irregularities on lateral loads.

![Mean Values of Lateral Loads Y](image)

Figure 5.1.7: Mean values of Lateral loads – Horizontal alignment

In the above picture, it can be noted that, while alignment irregularities are under the levels specified in EN 13231-1 (i.e. [-5;+5]mm on a 10 m chord), the values of lateral loads are low and have almost the same value. These cases correspond to plain bars. As the EN 13231-1 threshold is overcome, the lateral loads amplitude increase. These cases correspond to bars with stripes.

**c.ii-2 Vertical Loads:**

It was found that there is no influence of track alignment irregularities (in the range under analysis) on vertical loads.
(iii) **Vertical and lateral acceleration:**

Only the results at axle levels are shown.

c.iii-1 **Lateral Acceleration:**

The picture below displays the influence of track alignment irregularities on axles rms transverse acceleration.

![RMS Transverse acceleration per axle](image_url)

In the above picture, it can be noted that, while alignment irregularities are under the levels specified in EN 13231-1 (i.e. [-5;+5]mm on a 10 m chord), the values of lateral acceleration are low and remain equivalent. These cases correspond to plain bars. As the EN 13231-1 threshold is overcome, the lateral acceleration increases. These cases correspond to bars with stripes.
(d) **TOP IRREGULARITIES VS. CONTACT LOADS**

The initial top irregularities extracted from measurements are displayed in the picture below (black curve). Other curves correspond to reduction (green curves) or amplification (other curves) of the initial top signal.

**Amplification/Reduction of Top Irregularities Amplitude**

As shown in the picture above, the case where an initial measurement is multiplied by 3.5 is, in certain areas above the limit given in EN 13231-1. Other cases are mainly compliant with top irregularities tolerances of EN 13231-1.

(i) **Safety Parameters:**

The table below shows the main results with respect to safety parameters:

<table>
<thead>
<tr>
<th>Safety Parameters: Maximum Values calculated</th>
<th>Top Level</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
</tr>
<tr>
<td>Y/Q Max</td>
<td>dQ/Q</td>
</tr>
<tr>
<td>0.02</td>
<td>0.07</td>
</tr>
<tr>
<td>0.02</td>
<td>0.03</td>
</tr>
<tr>
<td>0.02</td>
<td>0.04</td>
</tr>
<tr>
<td>0.02</td>
<td>0.23</td>
</tr>
<tr>
<td>0.02</td>
<td>0.25</td>
</tr>
</tbody>
</table>

Table 5.1.4: Safety Parameters: Maximum Values calculated

All Safety parameters are under the criteria. So no top-level configuration tested is critical.
(ii) Dynamic loads on the wheel/rail contact:

  d.ii-1 Lateral Loads:
  Lateral loads do not vary significantly with top-level amplification. Thus, in the variation range studied, no significant influence has been demonstrated between top level and lateral loads.

  d.ii-2 Vertical Loads:
  Vertical loads do not vary significantly with top-level amplification. Thus, in the variation range studied, no significant influence between top level and vertical loads has been demonstrated.
(iii) **Vertical and Lateral acceleration:**

**d.iii-1 Lateral acceleration:**

In any configuration tested (see picture below), lateral accelerations have very low values, which means that the influence of top-level irregularities on lateral RMS acceleration is not significant.

![Axle Lateral rms acceleration](image)

Figure 5.1.10: Axle vertical acceleration

**d.iii-2 Vertical acceleration:**

Vertical accelerations vary when the amplitude of the top-level irregularity is close or above the limit mentioned in EN 13231-1 ([−6; +6] mm). Nonetheless, the values calculated remain low as can be seen in the following picture.

![Axle vertical rms acceleration](image)

Figure 5.1.11: Axle vertical acceleration
(e) **CANT IRREGULARITIES VS. CONTACT LOADS**

The initial cant irregularities extracted from measurements are displayed in the picture below (black curve). Other curves correspond to reduction (green curves) or amplification (red or orange curves) of the initial cant signal.

![Amplification/Reduction of cant irregularities amplitude](image)

**Figure 5.1.12: Amplification / Reduction of cant irregularities**

In the picture above, the cases where initial measurements are multiplied by 3.8 and 4.5 are, in certain areas above the limit given in EN 13231-1. Other cases are mainly compliant with cant irregularities tolerances of EN 13231-1.

(i) **Safety Parameters:**

The table below shows the main results for parametric study on cant irregularities with respect to safety parameters:

<table>
<thead>
<tr>
<th>Safety Parameters: Maximum Values calculated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cant irregularities</td>
</tr>
<tr>
<td>Y/Q Max</td>
</tr>
<tr>
<td>---------</td>
</tr>
<tr>
<td>0.02</td>
</tr>
<tr>
<td>0.03</td>
</tr>
<tr>
<td>0.04</td>
</tr>
<tr>
<td>0.04</td>
</tr>
<tr>
<td>0.04</td>
</tr>
</tbody>
</table>

**Table 5.1.5: Safety Parameters - Cant irregularities**

All safety parameters are under the criteria. So no cant irregularities configuration tested is critical.
(ii) Dynamic loads on the wheel/rail contact:

e.ii-1 Lateral Loads:

Lateral loads only vary significantly when cant irregularities correspond to higher levels than those given in EN 13231-1 (see picture below).

![Mean Value of Lateral Loads Y](image)

The case corresponding to the where the initial cant irregularities amplitude is multiplied by 2.6 is satisfactory with regards to lateral loads. This latter case corresponds to a cant irregularity signal equivalent to EN 13231-1.

e.ii-2 Vertical Loads:

Lateral loads do not vary with cant irregularities (See Picture below).

![Mean Values of Vertical Loads Q](image)

Figure 5.1.13: Mean value of lateral loads

Figure 5.1.14: Vertical loads - Influence of cant irregularities
(iii) **Vertical and Lateral acceleration:**

**e.iii-1 Lateral acceleration:**

As can be seen in picture below, only the last two cases (i.e. initial times 3.8 and initial times 4.5, this case corresponds to the case where Cant irregularities is above the limits specified in EN 13231-1) gives rise to higher levels of lateral accelerations.

![Axle lateral rms acceleration Influence of Cant Irregularities](image)

Figure 5.1.15: Lateral acceleration - Influence of cant irregularities

**e.iii-2 Vertical acceleration:**

The values of vertical accelerations obtained for this parametric study are very low. Moreover, no significant variation was observed with varying cant irregularities. Thus, no influence of cant irregularities on axle vertical acceleration is found in the studied variation range.
**ANALYSIS**

Four geometrical parameters have been studied: Gauge, Alignment, Top level, Cant. Those parameters have been studied in terms of irregularities, in other terms deviation from theoretical values.

The objective of this study was to check whether the EN 13231-1 (for new track construction) could be applied to typical tramway applications. In order to answer this question, four parametric studies have been performed, one for each geometrical parameters. Basically, the parametric studies consisted in amplifying or reducing the levels of irregularities and simulating the influence of such an operation on safety parameters, vertical and lateral loads, vertical and lateral accelerations.

These four parametric studies enable to conclude that for:

- **Safety parameters**: the safety parameters are fully and largely compliant with the safety criteria. In other term, the values given in EN 13231-1 (for new track construction) present no safety risk.

- **Vertical and Lateral Loads**: as long as track irregularities are below EN 13231-1, no significant influence of those defects has been observed on vertical and lateral loads.

- **Vertical and Lateral Accelerations**: since track irregularities are below EN 13231-1, no adverse influence of those defects has been calculated on axle vertical and lateral accelerations.

In the light of those observations/results, it can be concluded that no adverse situation is expected in the case EN 13231-1 (new construction) is applied. Thus, EN 13231-1 (new construction) could be applied to tramway application.
5.1.2.3. Track alignment criterias optimization

The objective of this study is to analyse the impact of the track alignment criteria on the wheel rail contact and forces

(a) **PRINCIPLE OF THE STUDY**

(i) **Model validation**

The validation of the model used for this study was conducted on the basis of tests conducted on a French network.

(ii) **Parametric study**

a.ii-1 **Specification of the parametric study**

- The parametric study allows testing a number of configurations of track design by varying the different components of the track one after another. For each configuration, a number of parameters are evaluated to valid whether or not a configuration.
- The following configurations will be a parametric study.
- Studies of track alignment criteria will focus on following cases:
  - Length of Clothoid (= Transition) (for Radius of curvature \( R = 25, 50, 100, 200, 300, 400, 500 \) m)
  - Length of straight alignment between two cloths in case of reverse curves (for Radius of curvature \( R = 25, 50, 100, 200, 300, 400, 500 \) m),
  - Overlapping (combination of Vertical and Horizontal Curves),

a.ii-2 **Evaluation parameters**

The security parameters are 3 and will be systematically checked for each dynamic calculation:

- The derailment indicator \( Y/Q \leq 0.95 \)
- The sum wheelset calculated on each wheelset \( (\Sigma Y_{2m})_{lim} = \alpha(10 + P_{0}/3) \)
- The wheel unloading behaviour \( dQ/Q \leq 0.6 \)

The classic parameters to evaluate are accelerations in the different carbody and are used to calculate the Jerk.

The vertical and lateral loads will be evaluated at the contact for all wheels from the rolling stock (not presented in this document).

a.ii-3 **Validation of all tested configurations**

All tested configurations will be validated if all security parameters are respected, then the comfort criteria and effort to the contact patch can be optimized.
(b) PRELIMINARY STUDIES OF THE DYNAMICS CALCULATIONS

(i) General validation of the model on vampire software

The parametric study is performed using VAMPIRE dynamic simulations software. To ensure the validity of the results, a validation of the model is preferable. This is done on the results of the measurement campaign of the network. An initial check was conducted on the basis of accelerometer placed on the vehicle (on the floor on the carbody M2). The comparison between numerical predictions and simulation results show a very good correlation of the model.

![Comparison measurement / simulation acceleration in Carbody M2](image)

Figure 5.1.16: Comparison measurement / simulation acceleration in Carbody M2

Measurements of track defects measured on the site of The network (gauge, alignment, top, cant) are located every 25 cm and are implemented in the dynamic simulation program to simulate the dynamic behaviour of the tramway as realistic as possible.

Results of amplitudes and signal dynamics form remain satisfactory as seen in the graph above.
(ii) Length of curvature taken in dynamics calculations

b.ii-1 Objective

To determine the minimum length of curvature for radius $R = 25$ to $500$ m from dynamic calculations in order to avoid any interaction on others parameters for example when we study the influence of length of the clothoid. These minimum length of curvature will be kept in all future calculations.

b.ii-2 Summary of minimum curvature length for all radius

Below trends representing the minimum arc length for each radius of curvature for Citadis 302_402.

<table>
<thead>
<tr>
<th>Radius of curvature (m)</th>
<th>Curvature length 302_402 taken in calculations (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>63</td>
</tr>
<tr>
<td>50</td>
<td>68</td>
</tr>
<tr>
<td>100</td>
<td>79</td>
</tr>
<tr>
<td>200</td>
<td>95</td>
</tr>
<tr>
<td>300</td>
<td>112</td>
</tr>
<tr>
<td>400</td>
<td>122</td>
</tr>
<tr>
<td>500</td>
<td>130</td>
</tr>
</tbody>
</table>

Table 5.1.6: Radius vs Curvature

The length of curvature according the radius will be taken into account in the construction of track file for dynamic calculations in order to study the influence of the clothoid length, the length of alignment between reverse curve, overlapping (without the impact of the circular curve length).
(c) APPLICATIONS TO OPTIMIZATION OF DESIGN CRITERIA

(i) Minimum length of clothoid (transition) in curve without cant

Objective

Determine the impact of the lengths of clothoid on the behaviour of rolling stock for radius of curvature from R = 25 m to R = 500 m. These simulations will also verify the behaviour of RS in the minimum values calculated according to the specification of track alignment criteria for Citadis 302_402 and will help to validate or optimize the calculation method following.

The maximum speed (in circular curve without cant) is calculated with respect to the maximum value of lateral acceleration of 0.68 m/s\(^2\) following the formula. This formula reflects the maximum allowable lateral acceleration of 0.68 m/s\(^2\) in curve of constant radius.

\[ V = 3.6\sqrt{0.68 \cdot R} \]
Where \( V \) in km/h and \( R \) in m

In order to determine the optimum length of each clothoid (in circular curve without cant), the following formulae are used in conjunction with the required speed and radius of curve (s).

\[ L = 2.5 \frac{v^3}{R} \]
\((v \text{ in } \text{m/s}, R \text{ in } \text{m}, J \text{ in } \text{m/s}^3)\)

The results of this formula will be compared with those of the dynamic calculations in order to validate or optimize the length of clothoid.

Safety assessment in clothoid

All tested configurations respect the safety criteria (derailment, sum of lateral effort calculated on each wheelset, wheel unloading) with different lengths of clothoid considered in dynamic calculations.

- The derailment indicator \( Y/Q \leq 0.94 \)
- The sum of lateral effort calculated on each wheelset \( (\Sigma Y_{2m})_{\text{lim}} = a(10 + P_0/3) \)
- The wheel unloading behaviour \( dQ/Q \leq 0.6 \)

We can see the degradation risk increase for values inferior to calculated values by the formulae below

\[ L_K = 2.5 \frac{v^3}{R} \]

All these configurations (except the length of clothoid for R25 and R50 with 4 m) respect safety criteria and can be studied of point of view comfort passengers.
Figure 5.1.17: Minimum clothoid length vs Radius

This chart can be used to define the lengths of clothoid optimized for a certain radius of curvature. We respect the maximum Jerk of 0.4 m/s$^3$ and the maximum lateral acceleration of 0.68 m/s$^2$.

### Conclusion: Length of the minimum clothoid (without cant), optimisation

<table>
<thead>
<tr>
<th>RADIUS OF CURVATURE</th>
<th>Minimum length of clothoid (dynamic simulation)</th>
<th>Minimum length of clothoid (according to the specification)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>7</td>
<td>12 (7 exceptionally)</td>
</tr>
<tr>
<td>50</td>
<td>9.5</td>
<td>12 (10 exceptionally)</td>
</tr>
<tr>
<td>100</td>
<td>13.5</td>
<td>14</td>
</tr>
<tr>
<td>200</td>
<td>19</td>
<td>20</td>
</tr>
<tr>
<td>300</td>
<td>23</td>
<td>24.3</td>
</tr>
<tr>
<td>400</td>
<td>27</td>
<td>28</td>
</tr>
<tr>
<td>500</td>
<td>30</td>
<td>31.34</td>
</tr>
</tbody>
</table>

Table 5.1.7: Radius vs minimum length of clothoid

Optimized

It is found that the calculated clothoid length following the formulation $L = 2.5 \frac{V^3}{R}$ is higher than those derived from dynamic simulations.
(ii) **Length of straight alignment between 2 clothoids in case of reverses curves**

**c.ii-1 Objective**

- Check and optimize the value of 12 m imposed by the criteria of the specification and issued of railway practices (EN 13803-1 et 2 design of track with abrupt changes of curvature) $L_{AD \text{ mini}} = 0.2 \cdot V$ with $V$ the speed (km/h).
- Check and validate the value of 0 m (inflection point) in relation to the behaviour of Citadis in these configurations.

The minimum lengths of clothoid (transition curves), which were described and validated in the previous chapter for maximum speeds in curves without cant, are retained in this study. The dynamic calculations were made in taking into account straights of 0, 3, 6, 9 and 12m between reverses curves of radius $R = 25$ to 500 m. In all configurations the acceleration for calculation of the jerk was evaluated in the area marked in brown to know the influence of the alignment between reverses curves (S-CURVE).

**c.ii-2 Conclusion: Length of the minimum alignment between s-curves**

<table>
<thead>
<tr>
<th>Radius of curvature (in m)</th>
<th>Length of minimal alignment between s-curves (<em>DYNAMIC SIMULATION</em>)</th>
<th>Length of minimal alignment between s-curves (<em>SPECIFICATION CITADIS</em>)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>3</td>
<td>12</td>
</tr>
<tr>
<td>50</td>
<td>3</td>
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<td>12</td>
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<tr>
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<td>12</td>
</tr>
<tr>
<td>400</td>
<td>3</td>
<td>12</td>
</tr>
<tr>
<td>500</td>
<td>3</td>
<td><strong>Optimized</strong></td>
</tr>
</tbody>
</table>

Table 5.1.8: Minimum length of alignment in s-curve

The minimum length of alignment between S-curves will be 3 m whatever the studied radius of curvature: respect of the maximum acceleration of $0.68 \text{ m/s}^2$ and a maximum Jerk of $0.4 \text{ m/s}^3$. 
(iii) Minimum length of clothoid for curves with cant

c.iii-1 Objective

Determine the minimum length of clothoid “for radius of curvature with CANT” from dynamic simulations in order to verify if the formulation used until today in our design criteria is correct to calculate the limit value of the rate of change of cant deficiency \((dI/dt)_{\text{lim}}\) in mm/s. This value \((dI/dt)_{\text{lim}}\) is a parameter which is not a normalized data for tramways.

The speed used in dynamic simulations is the maximum speed using the equilibrium cant \(dth\) which is the sum of cant with the cant deficiency. We respect a maximum acceptable lateral acceleration of 0.68 m/s²

\[
V_{\text{max}} = \left(\frac{R \cdot dth}{11.8}\right) \times 0.5 \quad \text{(with equilibrium cant } dth \text{ in mm and } R \text{ in m, } V \text{ in km/h)}
\]

and \(dth\) (equilibrium cant in mm) = applied cant + cant deficiency.

To dimension the length of clothoid in curve with cant, we use the 3 formulations according to the standard PrEN 13803-1:

\[
L_D \geq \frac{V}{3.6} \Delta D \left(\frac{dD}{dt}\right)_{\text{lim}}^{-1} [\text{m}]
\]

\[
L_D \geq \Delta D \left(\frac{dD}{ds}\right)_{\text{lim}}^{-1} [\text{m}]
\]

\[
L_K \geq \frac{V}{3.6} \Delta I \left(\frac{dI}{dt}\right)_{\text{lim}}^{-1} [\text{m}]
\]

**Formulae with the Rate of change of cant:** \(\frac{dD}{dt}\)

\(\checkmark\) \((dD/dt)_{\text{lim}} \leq \frac{V}{L_D} = 50 \text{ mm/s}\)

**Formulae with the cant gradient:** \(\frac{dD}{ds}\)

\(\checkmark\) \((dD/ds)_{\text{lim}} = 3 \text{ mm/m}\)

**Formulae with the Rate of change of cant deficiency:** \(\frac{dI}{dt}\)

\(\checkmark\) \((dI/dt)_{\text{lim}} = \text{numerical data not normalized for tramways}\)

The minimum length of the transition curve (clothoid) will be the largest of these 3 formulations in order to ensure the 3 limits defined above (respect of 3 conditions of the standard PrEN 13803-1).


c.iii-2 Used Methodology

The main objective of this study will be to determine a limit of Rate of change of cant deficiency \((dI/dt)_{\text{lim}}\) from this methodology by studying different radius of curvature, cant and maximum speed associated.

The dynamic simulations will verify and validate the procedure described below in defining a limit value for the Rate of change of cant deficiency: \(\frac{dI}{dt}\) (using as criterion the maximum jerk to 0.4 m/s³ as in the case of curve without cant, the jerk defines the comfort felt by passengers). From the dynamics
calculations and according to EN 12299, the maximum jerk in clothoid will be calculated as in the case of the study of curves without cant (see in the first chapter of this document).

Until today, the limit of the Rate of change of cant deficiency used was 0.61 mm/s, data provided by EGIS-Rail recommendations. We will check this information in insuring that we respect a maximum jerk of 0.4 m/s³.

Results of dynamic calculations will be compared with analytical calculations to verify the method used and determine / validate a limit of the Rate of change of cant deficiency (dI/dt)lim.

c.iii-3 Conclusion: Accepted limit value of rate of change of cant deficiency used (dI/dt)lim

Taking as limit value a rate of change of cant deficiency \((dI/dt)\)lim = 61 mm/s, we respect a maximum lateral acceleration of 0.68 m/s² in the track plan and a maximum jerk of 0.4 m/s³ in the track plan (conception criteria for the track).

The two methods (analytical and numerical) give identical values at the track plan, which means that the methodologies used to calculate the Jerk by these methods are correct.

For the jerk calculated to the floor, there is a slight difference because the analytical method is simplified. It’s clear that the dynamic simulation results are more accurate to the floor because we take into account the realistic dynamic behaviour of the train when it passes along the curve.

We therefore validated all analytical and dynamics calculations and have shown that the methods were equivalent. The value of 0.61 mm/s for the rate of change of cant deficiency is justified.

See details of calculations below.

c.iii-4 Details of calculation of limit value of rate of change of cant deficiency (dI/dt)lim

Below the cases studied by the analytical method (developed in using PrEN 13803-1) and the dynamic method on VAMPIRE (calculations of acceleration and jerk is then calculated in according to PrEN 12299), the values demonstrate the equivalence between these 2 methods and give us the limit value of Rate of change of cant deficiency (dI/dt)lim ensuring a maximum Jerk = ∆aq de 0.4 m/s³ to the track plan.
### Radius of curvature and length of clothoid associated

<table>
<thead>
<tr>
<th>R (m)</th>
<th>Lk (m)</th>
<th>D (mm)</th>
<th>I (mm)</th>
<th>V (Km/h)</th>
<th>aq (m/s²)</th>
<th>ai (m/s²)</th>
<th>$\Delta a_q$</th>
<th>$\Delta a_i$</th>
<th>$\Delta a_{aq}$</th>
<th>$\Delta a_{ai}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>15.35</td>
<td>20</td>
<td>104</td>
<td>32.4</td>
<td>0.68</td>
<td>0.80</td>
<td>0.399</td>
<td>0.467</td>
<td>0.392</td>
<td>0.429</td>
</tr>
<tr>
<td>100</td>
<td>17</td>
<td>20</td>
<td>104</td>
<td>32.4</td>
<td>0.68</td>
<td>0.80</td>
<td>0.360</td>
<td>0.421</td>
<td>0.361</td>
<td>0.399</td>
</tr>
<tr>
<td>100</td>
<td>18.73</td>
<td>20</td>
<td>104</td>
<td>32.4</td>
<td>0.68</td>
<td>0.80</td>
<td>0.327</td>
<td>0.383</td>
<td>0.329</td>
<td>0.368</td>
</tr>
<tr>
<td>100</td>
<td>19.71</td>
<td>40</td>
<td>104</td>
<td>34.9</td>
<td>0.68</td>
<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.336</td>
<td>0.400</td>
</tr>
<tr>
<td>100</td>
<td>21.03</td>
<td>60</td>
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<td>37.3</td>
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<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.335</td>
<td>0.422</td>
</tr>
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<td>104</td>
<td>56.1</td>
<td>0.68</td>
<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.336</td>
<td>0.393</td>
</tr>
<tr>
<td>300</td>
<td>34.14</td>
<td>40</td>
<td>104</td>
<td>60.5</td>
<td>0.68</td>
<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.335</td>
<td>0.408</td>
</tr>
<tr>
<td>300</td>
<td>36.43</td>
<td>60</td>
<td>104</td>
<td>64.6</td>
<td>0.68</td>
<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.334</td>
<td>0.421</td>
</tr>
<tr>
<td>500</td>
<td>39.22</td>
<td>10</td>
<td>104</td>
<td>69.5</td>
<td>0.68</td>
<td>0.80</td>
<td>0.335</td>
<td>0.392</td>
<td>0.335</td>
<td>0.386</td>
</tr>
</tbody>
</table>

### Cases where Lk is greater to the two others expressions, by default we have taken 61 mm/s to verify the maximum Jerk associated (value taken today by ALSTOM to define the minimum length of clothoid in curve with cant)

$$L_k \geq \frac{V}{3.6} \frac{\Delta I}{\Delta t_{lim}} [m]$$

### Table 5.1.9: Detail of calculations of limit value of rate of change of cant deficiency

Calculs (1) | Simulations (2) | Limit value of Rate of change of cant deficiency ($\frac{df}{dt}_{lim}$ in mm/s)
---|---|---
0.399 | 0.467 | 0.392 | 0.429 | 61
0.360 | 0.421 | 0.361 | 0.399 | 55
0.327 | 0.383 | 0.329 | 0.368 | 50
0.335 | 0.392 | 0.336 | 0.400 | 51.2
0.335 | 0.392 | 0.335 | 0.422 | 51.2
0.335 | 0.392 | 0.336 | 0.408 | 51.2
0.335 | 0.392 | 0.334 | 0.421 | 51.2
0.335 | 0.392 | 0.335 | 0.386 | 51.2
5.1.3 Rail stress and deformation

5.1.3.1 Introduction

A rail is subject to strains and deformations due to the sheer number of trains passing over it. Each time a wheel passes over may be considered as a load cycle. The accumulation passing wheels of a rolling stock are exerting a repeated loading on that rail. And as a response to this undergone cyclic loading it has been observed that the rail material is reaching an asymptotic regime, which can be:

- Pure elasticity,
- Elastic shakedown,
- Plastic shakedown,
- Ratchetting.

Identifying which one of the above regimes is reached, provides an indication about the subsequent consequences due to the numerous passes of trains and constitutes a first step in determining the expected life duration of the rail.

Several parameters are determining the trend the material will follow and the nature of the asymptotic regime. So we will firstly present the calculus method known as stationary method as well as the parameters identified as relevant ones. Then the implementation of the stationary method with the FEM software Castem will be detailed. Finally, the results of the parametric study will be given.

They will lead into a conclusion concerning the effect of every parameter on the nature of the asymptotic regime given the model used.
5.1.3.2. Theoretical bases for the study

(a) **WORKING HYPOTHESES**

We studied a rail subject to repeated loading by a train wheel. The component material of the rail is considered to be homogenous and isothermal at all times. It is modelled by an elastoplastic material. We worked in the context of small disruptions (HPP). Several material behaviour laws may be considered; we will use Von Mises type criteria, which are well adapted to metals.

We worked in 2 dimensions first of all. The rail is seen as a semi-infinite solid object. We will return to this hypothesis during the parametric study.

The wheel/rail contact force is modelled using Hertz’s theory. The normal force is given by

\[ p(x) = p_{\text{max}} \sqrt{1 - \frac{x^2}{b^2}} \]

where \( p_{\text{max}} \) is the normal maximum pressure and \( b \) the half-length of the contact.

The tangential force is equal to \( f \cdot p(x) \).

Vertical movement is blocked on the bottom of the rail.

Here is the list of parameters that in principle enable us to define our problem:

- Young modulus \( E \)
- Poisson’s ratio \( \nu \)
- Density \( \rho \)
- Elasticity limit in simple traction \( \sigma_Y \)
- Hardening law and associated parameters (hardening modulus, etc.)
- Half-length of the wheel-rail contact \( b \)
- Normal maximum pressure \( p_{\text{max}} \)
- Coefficient of friction \( f \)
- Train speed \( V \)

Since all the rails are made of steel, we choose to set \( E \) at 210GPa and \( \nu \) at 0.3.

(b) **THE STATIONARY METHOD**

To model the passage of a wheel over the rail in the standard manner, we must resolve a dynamic problem by having the elliptic force moved from one end of the rail to the other, using time steps. We must repeat this operation until we find the asymptotic response of the rail. This would require a considerable calculation time.
Figure 5.1.18: Stationary problem

For structures subject to a moving load at a fixed speed, a method called the stationary method was developed by MM. Dang Van and Maitournam [1 to 4]. It applies perfectly to the case being studied of an elliptic load on a rail. It involves going to the marker related to the load and adding a volume drive inertia force to the material. The advantage lies in removing the dependency on time in the equations. In fact, the temporal drift of a vector $A$ is written

$$ \frac{\partial A}{\partial t} = V \cdot \text{grad} A = V \frac{\partial A}{\partial x} $$

The problem is then resolved by imposing a static force on the rail and by resolving the problem’s equations from the right to the left. The initial conditions in terms of plastic deformation and internal variables are given by the plastic deformations and internal variables in $x = +\infty$.

The problem’s equations are the following:

Balance

$$ \text{div} \sigma = \rho V^2 u_{xx} \quad \text{in} \ \Omega $$

Boundary conditions

$$ \sigma n = T^{d} \quad \text{on} \ \partial S_T $$
$$ u = u^{d} \quad \text{on} \ \partial S_u $$

Initial conditions

$$ \varepsilon^p (+\infty, y) = \varepsilon^{p0} (y) $$
$$ \alpha_k (+\infty, y) = \alpha_k^0 (y) $$

Behaviour

$$ \varepsilon = \varepsilon^{el} + \varepsilon^p $$
$$ \sigma = \sigma^0 + L : \varepsilon^{el} $$

$$ \varepsilon_{el} = -\lambda \frac{\delta f}{\delta \sigma} $$
$$ \alpha_{k,el} = \lambda \frac{\delta f}{\delta A_k} $$

$$ f, \alpha_k \text{ and } A_k \text{ depend on the hardening model selected. Nevertheless, we can provide a general form for the plasticity criterion } f: \ f (\sigma, \alpha_k) = \|\sigma - X\|_q - R(p) - \sigma_q \quad \text{where } X \text{ is the centre of the elasticity zone, depending on } \alpha_k \text{ and } p \text{ the cumulated plastic deformation.} $$

(c) **ADDITIONAL HYPOTHESES**

It is possible to carry out a few additional approximations that will simplify our study. To do so, let us examine the variation formulation of the elastic problem:

Find $u \in C(\bar{u}^e)$ so that for any field $w \in C(0)$ we have
\[ \int_{\Omega} \varepsilon(u) : L : \varepsilon(w) dV = \int_{S_t} T^d w dS + \int_{\Omega} -\rho V^2 \frac{\partial^2 u}{\partial x^2} w dV \]

and compare the scales of the terms in the first and last integral thanks to the ratio \( r \) (\( C_s \) denotes the velocity of the mechanical waves in steel, \( h \) the characteristic size of the mesh and \( \Delta x \) the typical distance between 2 columns, see 5.1.3.3(a)).

\[ r = \frac{\varepsilon(u) : L : \varepsilon(w)}{\rho V^2 \frac{\partial^2 u}{\partial x^2} w} \quad \frac{E/h^2}{\rho V^2/\Delta x^2} \quad \left( \frac{C_s}{V} \right)^2 \left( \frac{\Delta x}{h} \right)^2 \]

But \( \Delta x \approx h \), so \( r \approx \left( \frac{C_s}{V} \right)^2 \) and as \( C_s \approx 6000 \text{km/h} \) for steel, we have \( r \gg 1 \).

We therefore choose to deal with the problem quasi-statically. The variation formulation becomes

Find \( u \in C(u^d) \) so that for any field \( w \in C(0) \) we have

\[ \int_{\Omega} \varepsilon(u) : L : \varepsilon(w) dV = \int_{S_t} T^d w dS \]

We find the usual formulation. The difference is that the problem is resolved numerically the closer from the right to the left in the mesh (see 5.1.II.1).

In addition we suppose that the plastic deformations and the initial constraints in the rail are zero. However, the algorithms written under Castem provide the user with possibility of entering fields that are not zero.

We use a cinematic linear hardening law. The plasticity criterion is \( f = \| \sigma - X \| - \sigma_Y \) with \( X = H \varepsilon^n \).

The hardening modulus \( H \) is a parameter in our study.

We therefore give the parameters retained for our study and the interval of the values on which we work and which correspond to realistic values. We will see in 5.1.II.1. that we must add a parameter related to the mesh.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>( H ) (GPa)</th>
<th>( \text{SigmaY tensile strength(MPa)} )</th>
<th>( P_{\text{max}} ) (Mpa)</th>
<th>( f )</th>
<th>( b ) (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>minimum</td>
<td>20</td>
<td>200</td>
<td>300</td>
<td>-0.6</td>
<td>2</td>
</tr>
<tr>
<td>maximum</td>
<td>100</td>
<td>900</td>
<td>2000</td>
<td>0.6</td>
<td>6</td>
</tr>
</tbody>
</table>

Table 5.1.10: Range of parameters

(d) **CALCULATION METHODS**

The stationary method requires specific meshing that limits the rail in vertical columns numbered from 1 to \( N \) from right to left.

The algorithm carries out several elastoplastic calculations in order to find the stationary solution. A calculation is performed as follows:

The plastic deformations and the internal variables are given in column 1 (\( x = +\infty \)) either by the user in the first calculation or via the result of the previous calculations.
For column \( n+1 > 1 \), the plasticity criterion is calculated at each Gauss point. If it is respected \( (f < 0) \) we have \( \varepsilon_{n+1}^{p} = \varepsilon_{n}^{p}, \alpha_{k,n+1} = \alpha_{k,n} \). If the criterion is not met \( (f > 0) \), we calculate the increment in the plastic deformations and the internal variables with the behaviour laws in I.2.; we have \( \varepsilon_{n+1}^{p} = \varepsilon_{n}^{p} + \Delta \varepsilon_{n}, \alpha_{k,n+1} = \alpha_{k,n} + \Delta \alpha_{k,n} \). See [1] [2] and [11] for more details.

The algorithm converges when 2 stoppage conditions are verified:

The plasticity criterion is respected at all points.

The plastic deformations are stationary. This means that we find the same plastic status upstream and downstream of the wheel: \( \varepsilon_{\infty}^{p} (x = -\infty) = \varepsilon_{\infty}^{p} (x = +\infty) \)

We must then specify what the algorithm does between 2 elastoplastic calculations. There are two possibilities.

(i) **Passage by passage method (indirect)**

This method calculates the result of each wheel passage.

Each passage may require several elastoplastic calculations between which we conserve the previous plastic deformation field then carry out a new elastic prediction. The plastic deformations in the column 1 in calculation \( i+1 \) are those found in calculation \( i \) in this same column. The operation is repeated until the plasticity criterion is respected everywhere.

Then, between two passages, we recover the plastic deformation in \( x = -\infty \) and impose it in all the other meshing columns. We therefore have the rail as it actually is between 2 passages. Then we carry out a new elastic prediction, which is the starting point for the calculation of the next passage. The algorithm converges when the plastic deformations in \( x = -\infty \) are identical between 2 successive wheel passages.

This method has the advantage that it is exact outside the approximation of the finite elements. The proof was provided by MM. Dang Van and Maitournam [1]. However, it does require a certain calculation time.

(ii) **Direct method**

This method does not differentiate between the wheel passages.

It involves carrying out the elastoplastic calculations in succession, but this time by imposing in \( x = +\infty \) for the calculation \( i+1 \) the plastic deformations found in \( x = -\infty \) at the end of calculation \( i \).

At the end of each calculation we check to see whether the plastic deformations \( x = -\infty \) and in \( x = +\infty \) are identical. If this is the case we then check the plasticity criterion. If this criterion is respected everywhere, the algorithm converges. If either of the two stoppage conditions is not checked, then we move onto the next calculation.

This method requires reduced calculation time. It leads to the same result as the indirect method concerning the nature of the asymptotic regime. However, the scale of the plastic deformations and other solution fields is not the same as with the indirect method whose accuracy has been demonstrated.
(iii) Asymptotic regime

Once the stationary solution has been found with one of the methods above, we only need to determine the nature of the asymptotic regime achieved.

The case of a purely elastic regime is clear since the plasticity criterion is checked from the start. The algorithm does not carry out an elastoplastic calculation.

For elastic shakedown and plastic shakedown, we take the plastic deformations' solution field, extract the plastic deformations from column 1 and then compare them with the plastic deformations in the other columns. If they are still identical, this means that they do not change during a load cycle; this is elastic shakedown. If there is at least one column for which they vary, this means that they change during a load cycle but finish in the same state as at the start of the cycle; this is plastic shakedown.
5.1.3.3. Implementation under Castem

(a) GENERAL
Castem, also called Cast3M, is a partial equation resolution application that uses finite elements developed by the CEA.

Castem is able to mesh a structure whose geometry is defined by the user. It also carries out post-processing for results.

It is a very open code which is well suited to research work. Nevertheless, the interface with the user is almost zero since it does not have its own editor, it is executed in a command window and is not simple to use.

(b) MESHING
To represent a portion of rail in 2D, we use a simple rectangle. We must then choose several things: the dimensions of this rectangle, the size of the elements to ensure that the meshing is not too rough and the type of element.

Since we must create a meshing composed of vertical columns, we opt for 8-point rectangular elements (QUA8). We could have taken 4-point elements only (QUA4).

The piece of rail modelled must be large enough in relation to the size of the wheel-rail contact (b). At the same time, the meshing must be fine enough under the contact zone.

Initially, we used the following meshing (the load is also represented, the “different sizes” of the nodal forces are related to the use of QUA8 elements):

![Initial meshing and load](image-url)
This meshing was sufficiently fine but not large enough, which altered the results. In addition, we need independent meshing for parameter \( b \) to be able study the effect of \( b \) on the nature of the asymptotic regime. This leads us to introduce a new parameter that we note \( a \). To avoid overloading the computer’s memory, we have created a meshing with variable density, which is larger but which keeps almost the same number of elements.

![Meshing used and load](image)

This meshing contains 100 columns with 50 elements each.

We can already see that \( b \) and \( a \) must be relatively close. We will return to this in the parametric study.

(i) **Boundary conditions**

As indicated in 5.1.I.1., the movement is blocked according to \( y \) for the P2P3 straight line. To enable the computer to find a solution (uniqueness problem) the structure must be blocked according to \( x \). Several solutions may be considered: block the straight line P2P3 according to \( x \) (and still \( y \)), block the straight line P1P2 according to \( x \), or block point P2 according to \( x \). These alternatives are discussed in (c)(iii).

(c) **SPECIFIC PROCEDURES**

(i) **Plastic increment**

Let’s suppose that we are at a point in the calculation where one of the Gauss points does not respect the plasticity criterion. We must then calculate the increment in the plastic deformations and internal variables for this point by a radial return algorithm.

There are two problems with Castem. It is difficult to access the Gauss points individually; we must handle the fields defined over a column and there is no radial return function included. Two similar procedures are provided in Castem but they are not very well documented and it is difficult to find out what they calculate effectively. This is why we have preferred to write a radial return algorithm ourselves. We may describe it as follows:
The entry fields are the characteristics of the material, the elastic prediction for the deformations \( \varepsilon^{\text{elas}} \), the plastic deformations and internal variables in the previous column. For the cinematic linear hardening, the computer calculates:

\[
\varepsilon^{\text{elas}} = \varepsilon^{\text{préd}} - \varepsilon^p
\]

\[
S^{\text{elas}} = \text{dev}\left(\sigma^0 + \lambda \text{tr}(\varepsilon^{\text{elas}}) \mathbb{I} + 2\mu\varepsilon^{\text{elas}}\right)
\]

Then

\[
\Delta \varepsilon^p = \frac{1}{H + 2\mu} \cdot \left(\left\| S^{\text{elas}} \right\| \sqrt{\frac{2}{3}} \sigma^p \right) \cdot \frac{S^{\text{elas}}}{\left\| S^{\text{elas}} \right\|}
\]

Which enables us to update the plastic deformations and internal variables.

The simultaneous handling for all the points in the column makes the coding of these relatively simple calculations quite delicate.

(ii) Stoppage tests

Let us return to the operation of the stoppage tests.

c.ii-1 Steady state

This involves comparing two deformation fields \( \varepsilon^p_1 \) and \( \varepsilon^p_2 \) defined on the same geometric support (one column) with a certain tolerance \( \text{tol} \), typically 1%. For this, we start by setting to 0 the deformations lower than \( 10^{-5} \). This enables us to avoid any problems at points with a relatively low level of deformation. Then we calculate \( \left\| \varepsilon^p_1 - \varepsilon^p_2 \right\| \) which we will compare to a reference value. This is defined by

\[
(\text{tol} \times \sqrt{\frac{2}{3}} \sigma^p \max \left( \left\| \varepsilon^p_1 \right\| + \left\| \varepsilon^p_2 \right\| \right))
\]

If there is a point where the difference in the two fields is greater than the reference value, the test is false.

c.ii-2 Plasticity criterion

On input we provide the characteristics of the material, an internal variable field \( (X) \) and a tolerance \( \text{tol}' \). We create a reference value equal to \( \text{tol}' \times \sigma_Y \), we calculate the criterion then we check at each point that

\[
f = \left\| \sigma - X \right\| - \sigma_Y \leq \text{tol}' \times \sigma_Y
\]

If a point violates this equality, the test is false.

(iii) Limits

c.iii-1 Boundary conditions

The Boundary conditions in movement have a significant impact on the program's behaviour. Here we discuss the different possibilities for the blockage according to \( x \).
If we block the movement according to y and x for the straight line P2P3 we quickly encounter edging problems. In fact, points P2 and P3 are places where stresses are concentrated and they plastify when the load is increased, which leads to undesirable behaviour for the program – bad asymptotic regime at medium load, non convergence at high load.

Blocking P2 according to x is a good alternative. We can see in figure 7 that the structure is deformed significantly more according to x than on the previous figure. However, we encounter problems when we
take non-zero tangential stresses. In fact, it is only point P2, which "retains" the structure. The stresses are high at this point and for $f > 0.2$ it plasticizes well before the part under the load plasticizes, as shown below.

![Figure 5.1.24: Effect of the conditions on the limits (plastic deformations)](image)

To get around this problem, we choose to block the whole straight line P1P2 according to x. The deformations in figures 7 and 8 are very close, but the stresses are highly distributed and are no longer concentrated on P2. Except otherwise stated, this boundary condition has been used.

In addition, in order to avoid other edge effects, we “exclude” the two columns for each end. This means that if there are N columns we assimilate $x = +\infty$ in column 3 and $x = -\infty$ in column $(N - 2)$.

c.iii-2 Accuracy

The two stoppage tests have a certain tolerance set by the user. Initially the tolerances were equal. This does not cause any problems with the direct method since exactly the same calculation is repeated until the two tests are verified. However, the two tolerances each play a different role in the indirect method.

The tolerance $tol'$ of the plasticity criterion determines how accurate a passage is. With low tolerance, the calculation of a passage requires more time but is more accurate and, all things being equal, the algorithm needs to simulate fewer passages to converge.

The tolerance $tol$ of the stationary aspect tests enables the calculation to be stopped when the plastic deformations from one passage to another are close enough. With small tolerance, the algorithm needs to simulate more passages.

Finally, we must choose a small $tol'$ and an $tol$ that is not too small to avoid a long calculation time, but still small enough to be sure we have reached a stationary state. We estimate that $tol = 1\%$ is an acceptable compromise. From this point, we have tested several values for $tol'$. In the end we have retained $tol' = 0.5\%$. We illustrate this by tracing the maximum of the difference in plastic deformations in $-\infty$ between successive passages, using several tolerance pairs (in the legend, the first tolerance is the one for the plasticity criterion, the second is the one for the stationary status test).

<table>
<thead>
<tr>
<th>H (GPa)</th>
<th>SigmaY tensile strength(MPa)</th>
<th>Pmax (Mpa)</th>
<th>f</th>
<th>a (mm)</th>
<th>b (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>500</td>
<td>1170</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>
Table 5.1.11: Parameters

Figure 5.1.25: Maximum in inter-passage plastic deformation difference according to the passage for several tolerance pairs.

We see that with the pair (tol’ = 0.5% ; tol = 1%) we converge in few passages and very quickly approach a stationary solutions (the blue curve is below the green one).
5.1.3.4. Study results

(a) **ELASTIC SHAKEDOWN EXAMPLE**

We present the plastic deformation results for an elastic shakedown case. Remember that we use the meshing in figure 5. The passage-by-passage method was used.

<table>
<thead>
<tr>
<th>H (GPa)</th>
<th>SigmaY tensile strength (MPa)</th>
<th>Pmax (Mpa)</th>
<th>f</th>
<th>a (mm)</th>
<th>b (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>500</td>
<td>1000</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 5.1.12: Parameters

The plastic deformations do not change over a cycle; they are constant along the horizontal line.
(b) PLASTIC SHAKE DOWN EXAMPLE

The plastic shakedown case is richer in terms of interpretation. We only represent certain components of the fields that we feel are the most relevant. The graphs for all the components for this elastic shakedown case are provided in the appendix. They were obtained by the passage-by-passage method.

<table>
<thead>
<tr>
<th>H (GPa)</th>
<th>SigmaY tensile strength(MPa)</th>
<th>Pmax (Mpa)</th>
<th>f</th>
<th>a (mm)</th>
<th>b (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>500</td>
<td>1250</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 5.1.13: Parameters

The algorithm converges after 8 passages. We can clearly see under the wheel-rail contact area that the plastic deformations change.

To better see how the algorithm works, we trace the plastic deformations for the passages 1, 2 and 8. We trace them in $x = -\infty$ according to the depth and also at fixed depth (0.7b approx) according to $x$.
We see that on the first passage the plastic deformations are zero upstream of the wheel. Passage 2 starts in $x = +\infty$ with the plastic deformations found on passage 1 in $x = -\infty$. For the stationary response, we find the same plastic deformations upstream and downstream of the wheel. The algorithm behaves as planned.

We note that the plastic deformations $XY$ change symbol according to depth.

Finally, we look at the residual stresses in the rail after passage of the train in stationary state. To do so, we trace the residual stresses in $x = -\infty$ according to the depth. They are at a maximum around $y = -b$. 
We may also trace the stress-strain curve to observe the load cycles. We see that the first cycles are open, and then converge on a closed cycle, showing a stationary state.
(c) **PARAMETRIC STUDY**

This involves determining the influence of different parameters on the nature of the asymptotic regime. Let us start by recalling the parameters selected for this study.

- Material parameters
  - Elasticity limit in simple traction $\sigma_Y$
  - Hardening modulus $H$

- Load parameters
  - Half-length of the wheel-rail contact $b$
  - Normal maximum pressure $P_{\text{max}}$
  - Coefficient of friction $f$

- Parameter related to modelling
  - Meshing parameter $a$

We give these parameters default values, which are used in the calculations, except where stated to the contrary. These are the values for the elastic shakedown example 5.1.(a).

<table>
<thead>
<tr>
<th>$H$ (GPa)</th>
<th>$\sigma_Y$ (tensile strength, MPa)</th>
<th>$P_{\text{max}}$ (MPa)</th>
<th>$f$</th>
<th>$a$ (mm)</th>
<th>$b$ (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>500</td>
<td>1000</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 5.1.14: Parameters for elastic shakedown
(i) Load factor

We define the load factor as the ratio $P_{\text{max}}/\sigma_Y$. Let us show the importance of this factor by tracing the elasticity and elastic shakedown limits in a diagram $P_{\text{max}} - \sigma_Y$.

![Diagram Pmax - \sigma_Y](image)

Figure 5.144: Diagram $P_{\text{max}} - \sigma_Y$

We obtain straight lines, which shows the usefulness of the $P_{\text{max}}/\sigma_Y$ ration to define the limits of the different asymptotic behaviours.

This might let us think that there are two values for the load factor, which define the elasticity and elastic shakedown limits (here 1.7 and 2.3). However, the other parameters may modify these threshold values.

Note that the two calculation methods provide the same limits. This result was never proved wrong throughout the study.
(ii) **Hardening modulus**

Let us study the influence of the hardening modulus $H$. To do so we will set $\Sigma Y$ at 500MPa and vary $H$.

![Diagram showing the influence of hardening modulus on load factor]

**Figure 5.1.45: Load factor – hardening module diagram**

The hardening modulus has no notable influence on the elasticity and elastic shakedown limits.
(iii) Coefficient of friction

Here we study the effects of the tangential stresses.

Figure 5.1.468: Load factor – rubbing coefficient diagram

We can see that the curve is not exactly symmetrical in relation to the y-axis. This can be explained simply by the fact that our model is not completely symmetrical since we chose to block movement according to \( x \) for the straight line P1-P2 located to the right of the mesh.

Concerning our study, the coefficient of friction lowers the elasticity and elastic shakedown limits. The phenomenon is low for around \( |f| < 0.25 \) environ. It is very high beyond this.

This means that for a rail located in a braking or acceleration area, we may easily obtain plastic shakedown, which will accelerate fatigue on the rail.
(iv) Size of the wheel-rail contact

First of all, we vary $b$ only by keeping the mesh presented in 5.1.3.3(a).

![Figure 5.1.47: Load factor – size of the wheel-rail contact (usual meshing) diagram](image1)

In principle, it is difficult to interpret these curves. In fact, reducing the contact surface seems to increase the elasticity limit, which is counter-intuitive. The explanation is related to the mesh used: at the contact level the mesh is denser, the meshes have a size of 0.3 mm over a length of 6 mm. But this is only adapted if $b$ is close to $a$. To show this, we have traced the same curves but by modifying each time to keep $a=b$.

![Figure 5.1.48: Load factor – size of the wheel-rail contact ($a=b$) diagram](image2)
The dominant effect is the fineness of the meshing. To study the influence of the contact surface properly, there must be meshing for which the densest part has meshes of characteristic size $b_{\text{min}}/10$ over a characteristic distance $2b_{\text{max}}$. This is what we have done, taking $b_{\text{min}}=2\text{mm}$ and $b_{\text{max}}=6\text{mm}$.

![Figure 5.1.49: Adapted meshing](image)

This mesh contains 140 columns of 100 elements each. The calculations are much longer and we come close to the limits of the computer's memory.

Here are the results obtained.

![Figure 5.1.50: Load factor – size of the wheel-rail contact (adapted meshing) diagram](image)

The size of the contact does not have a major influence on the nature of the asymptotic regime. This is normal as we model a rail that has an infinite length. Regardless of the size of the contact, there is always Hertz pressure applied to a semi-infinite solid object.
Things may be different in 3D since the ratio between the width of the rail and the width of the contact do not necessarily enable us to apply the semi-infinite solid object hypothesis.

If we imagine the contact on the side depths of the rail, the results will be modified.

Nevertheless, the size of the contact is the characteristic size of the problem. For example, it modifies the depth of the plastic deformations.

![Graph showing depth of the maximum plastic deformation according to the size of the wheel-rail contact](image)

Figure 5.1.51: Depth of the maximum plastic deformation according to the size of the wheel-rail contact

This curve has been traced with a load factor of 2.2 (elastic shakedown).

The maximum plastic deformation is found at a depth of around \(-0.64\) \(b\). The depth interval on which the rail plasticizes also depends on the size of the contact. It does not exceed a depth of \(-3b\).
**LIMITS**

We have used a simple hardening model that has the advantage that it can be programmed easily and only has a single parameter (H) but does have disadvantages. For a cinematic, linear hardening, we cannot simulate the plastic ratchetting phenomenon (Halphen, 1977); the algorithm always converges at worst on a plastic shakedown case. This type of model is only valid if the load is not too high. For a load factor of over 2.5, we obtain results that are not very convincing, which we illustrate with the plastic deformations graph for a load factor of 3.4.

![Figure 5.1.52: YY plastic deformations for a load factor of 3.4](image)

Concerning the behaviour of the program with the model, the calculation time and the number of passages for the indirect method on convergence do not follow a regular change. The algorithm seems to converge in particular with more difficulty just after the plastic shakedown limit.

<table>
<thead>
<tr>
<th>Material</th>
<th>E (GPa)</th>
<th>Nu</th>
<th>H (GPa)</th>
<th>SigmaY tensile strength (MPa)</th>
<th>f</th>
<th>a (mm)</th>
<th>b (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>210</td>
<td>0.3</td>
<td>70</td>
<td>500</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>210</td>
<td>0.3</td>
<td>90</td>
<td>300</td>
<td>0</td>
<td>3</td>
<td>4</td>
</tr>
</tbody>
</table>

Table 5.1.15: Parameters value

![Figure 5.1.53: Calculation time according to the load factor](image) ![Figure 5.1.54: Number of passages simulated by the indirect method according to the load factor](image)
(e) FUTURE DEVELOPMENTS

First of all, we want to set up a more realistic material model. We have done this for a cinematic linear and isotropic non-linear hardening combined, but the algorithm no longer converges as soon as we reach the elastic hardening limit. Nevertheless, this limit is obtained when the load factor is 2.3. This lets us think that the hardening law has no influence on the elastic shakedown limit.

Then we will move to a 3D model, which will enable us to modify the form of the wheel-rail contact (circular, elliptic, etc.) and to consider a contact on the depth of the rail. This will be done during July.

The longer-term objective is to use the stationary method with the ANSYS application and to study fatigue on the rail by using the Dang Van criterion in particular.
5.1.3.5. Conclusion

We have presented the stationary method and its implementation. Under certain hypotheses, this method has enabled us to carry out a parametric study. Let us summarise the main results.

For a cinematic linear hardening, in 2D, in the quasi-static context, the nature of the asymptotic regime is determined by:

- The load factor, equal to the ratio \( P_{\text{max}} / \sigma_Y \)
- The rubbing coefficient
- For a zero rubbing coefficient, the asymptotic regime is the following:
  - Elasticity for a load factor lower than 1.7
  - Elastic shakedown for a load factor between 1.7 and 2.3
  - Plastic shakedown for a load factor over 2.3
- The hardening modulus and the size of the wheel-rail contact has no significant influence.
- The mesh must be adapted to the size of the wheel-rail contact.

The rail fatigue study has two practical advantages: to be able to make a commitment on the lifetime of the rails installed and to optimise the rail replacement frequencies for the track maintenance.
5.1.3.6. Bibliographical references

(i) Stationary method and rail fatigue


(ii) Material mechanics


(iii) Finite elements and Castem documentation


[12] http://www-cast3m.cea.fr/cast3m/index.jsp notamment Utilisateurs\Notices et Utilisateurs\Documentation.

5.1.4 Measurements in Madrid

5.1.4.1. Introduction

Noise and vibration measurements have been performed between 19 and 20 May 2010 on the Metro Ligero Oeste – line Boadilla, in order to compare three CDM systems, which are:

- Classic + Manta (S1): CDM-QTrack-HP + FST on mat FSM-L13;
- Classic (S2): CDM-QTrack-HP;
- Comfort (S3): QTrack-XP.

The following measurements have been performed for all systems:

- Rail admittance measurements;
- Rail roughness measurements;
- Pass-by measurements: noise and vibrations.

The Technical Report attached in Appendix provides all the details of this measurement campaign.

5.1.4.2. Test Conditions

(a) TEST VEHICLE

The test vehicle is the ALSTOM CITADIS 302.

Figure 5.1.55: Citadis 302 in Madrid
(i) **Metro Ligero Oeste vehicles**

The Metro Ligero Oeste vehicles, model CITADIS 302, were manufactured by ALSTOM. They can carry nearly 200 passengers. They run on clean electricity, they are extremely quiet and have an extremely luminous interior design.

Metro Ligero Oeste S.A. now has 27 of the 70 CITADIS 302 vehicles, bought from ALSTOM in the first stage by MINTRA.

These are amongst the most modern vehicles in the world at present. They run in tracks on electric-driven steel wheels, they have an excellent braking system; they are easy to board (100% low floor), with four double doors and wide aisles to hold wheelchairs and bicycles.

Although light rail vehicles mostly run on their own rights-of-way, in the case of Metro Ligero Oeste, they are also designed to share their space with buses, cars, cyclists and pedestrians. They provide frequent services and have priority over the rest of the traffic.

(ii) **Technical details of light rail vehicles**

- Total length 32.34m
- Total width 2.40 m
- Total height 3.60 m
- Entrance height 320 mm
- Track gauge 1.439m
- Composite vehicle weight 39.9 tons
- Number of seated passengers 54
- Standing capacity 132
- Total capacity 186
- Power 4 x 120kW
- Power Supply: 750 V cc / 400 V ac / 24 V dc

(b) **Track System Types**

In three different sections, three different systems are measured:

- Classic + Manta (S1): CDM-QTrack-HP + FST on mat FSM-L13;
- Classic (S2): CDM-QTrack-HP;
- Comfort (S3): QTrack-XP.

The measurement sites are situated on the line from Puerta de Boadilla to Colonia Jardin. The location of the measurement sites is illustrated in the following figures, showing two maps and one photograph per section/system.
Figure 5.1.56: Map of system Classic + Manta

Figure 5.1.57: Map of system Classic

Figure 5.1.58: Map of system Comfort
5.1.4.3. Geometry Measurement

(a) Interpretation of Readings

(i) Measurement processing

The track irregularities measurement trolley (called Krab-Light trolley and illustrated hereunder) allows plotting 5 preliminary signals, which constitute the basis for post-processing. These 5 preliminary signals are:

- The preliminary gauge with a 1435mm nominal gauge basis
- The preliminary cant measured thanks to an inclinometer (positive at right, negative at left).
- The preliminary vertical alignment measured on the right of the trolley on an 1.9m asymmetric basis (positive at left, negative at right)
- The preliminary horizontal alignment on the right side and on the same basis.
- The preliminary twist with a twist-measuring pole.

Figure 5.1.59: Krab-Light trolley

(ii) Post-processed measurements

A post-processing is applied and allows bringing back the track irregularities to a workable basis. Characteristics of this post-processing is defined by EN 13848 standards.

These European standards require to process these data using a Butterworth type band-pass filter, in a wavelength range between 2 and 25m.

During post-processing, the software uses the 5 preliminary signals and the wavelength parameter defined by EN 13848 standards to obtain:

- A quasi-static signal (low-frequency or high-wavelength signal).
- A dynamic signal (high-frequency or low-wavelength signal).
- An overall signal (sum of both signals).

The post-processing allows obtaining the following data for each track parameter:
The post-processing calculates the following parameters for each filtered track irregularity:

- The quasi-static signal is a mean line, obtained by a least squares method. This signal is close to the designed signal.
- The dynamic signal is obtained calculating the gap between the quasi-static signal and the unprocessed signal.
- The overall signal is the sum of the quasi-static signal and the dynamic signal.
- The vertical alignment and the horizontal alignment are brought to a 10m asymmetric basis, according to European standards.
- The curvature is the track deflection measured on a 10m symmetric chord.
- The twist is calculated on 3 different basis, in accordance with the operator choice, with a maximum value of 20m (for Madrid measurement: 3m, 5m and 10m)

On the overall post-processed parameters, the following ones are studied:

- The gauge
- The dynamic signal of the cant
- The dynamic signal of the vertical alignment measured at the centre of the track on a 10m basis
- The horizontal alignment measured at the track centre axis on a 10m basis
- The twist on a 3m basis.
- The axis curvature (Curvature = 1000/R where R = arc radius)

Note: The design gauge for the ML-3 line is 1435 mm

These post-processed parameters are analysed as follows:

- A statistic study per 200m sections
- A study in curves.
(b) STATISTIC STUDY OF V1: [PUERTA DE BOADILLA-COLONIA JARDIN]

(i) Principle:

The statistic study is based on the previous numeric results to generate an analysis by 200m sections. Thus, for each section and for each geometric parameter, the mean value, the maximum value, the minimum value and the standard deviation are calculated.

The direction of travel for measurements on track V1 is shown hereafter:

![Direction of travel for measurement V1](image)

Figure 5.1.60: Direction of travel for measurements - V1

(ii) Gauge

The following figure presents the standard deviation for the gauge per 200m section:

![Gauge: Standard deviation each 200m](image)

Figure 5.1.61: Standard deviation for gauge per 200m section - V1

b.ii-1 General observations

We note that, on the overall length of the track V1, the mean standard deviation is 1.3mm. Thus, the track gauge is **1435mm +/- 2.6mm** for a 95% compliance and **1435mm +/- 3.9mm** for a 99% compliance.

However, on all sections analysed and considering a 95% compliance, 12 no’s of 200m sections are not compliant with the EN 13231-1 standard. In fact, the gauge in these sections is higher than +/-3mm. We will study these sections in order to define the reasons of this non-compliance.
b.ii-2 Non-compliance analysis

On non-compliant sections, we have selected 2 representative sections: the section [2,4: 2,6] and the section [9,6: 9,8].

b.ii-2.1 Section [2,4: 2,6]:

Figure 5.1.62: Gauge defects in section [2,4: 2,6]

We note that the track gauge exceeds the EN 13231-1 standard acceptance tolerances in a small radius curve (R=60m) as well as in a straight alignment. The **first defect is related to a wear phenomenon in small radius curves. The second one may be related to a track installation defect** (for information, this section is a descendant slope).

b.ii-2.2 Section [9,6: 9,8]:

Figure 5.1.63: Gauge defects in section [9,6: 9,8]

We note that the track gauge exceeds the acceptance tolerances in a small radius curve (R=40m). **This defect is related to a wear phenomenon in small radius curves.**
(iii) Vertical alignment

The following figure presents the standard deviation for the vertical alignment per 200m section:

![Figure 5.1.64: Standard deviation for vertical alignment per 200m section - V1](image)

**b.iii-1 General observations**

We note that, on the overall length of the track V1, the mean standard deviation for vertical alignment is 1.3mm. Thus, the vertical alignment range is between \(-2.6\text{mm}\) and \(+2.6\text{mm}\) for a 95\% compliance and between \(-3.9\text{mm}\) and \(+3.9\text{mm}\) for a 99\% compliance.

However, considering a 95\% compliance, we note in a 200m section a standard deviation higher than 4mm. So, this section is non-compliant with the EN 13231-1 standard acceptance tolerances for vertical alignment.

We will study this section in order to define the reasons of this non-compliance.
b.iii-2 Non-compliance analysis

b.iii-2.1 Section [2,8: 3]:

We note that the vertical alignment defects exceeds the acceptance tolerances in a small radius curve (R=50m). This defect is not related to a track irregularity but it is related to a post-processing fault.

In addition, we will study the section [9,6: 9,8], which is compliant for the vertical alignment but not for gauge, cant and twist

b.iii-2.2 Section [9,6: 9,8]:

We note that the vertical alignment oscillates a lot around its mean value. This is due to a cant defect in this section.
(iv) **Twist**

The following figure presents the standard deviation for the twist per 200m section:

![Twist: Standard deviation each 200m](image)

**General observations**

We note that, on the overall length of the track V1, the mean standard deviation for twist is 1.8mm. Thus, the twist range is between **–3.6mm and +3.6mm** for a 95% compliance and between **–5.4mm and +5.4mm** for a 99% compliance.

However, considering a 95% compliance, we note in a few sections a standard deviation higher than the EN 13231-1 standard acceptance tolerances.

We will study these sections in order to define the reasons of this non-compliance.
b.iv-2  Non-compliance analysis

On all non-compliant sections, we have selected 2 representative sections of the twist defects: the section [2,6: 2,8] and the section [9,4: 9,6]. In addition, we will analyse the section [9,6: 9,8], which combines several defects (gauge, cant and twist).

b.iv-2.1  Section [2,6: 2,8]:

Figure 5.1.68: Twist defects in section [2,6: 2,8]

We note that the twist defects are due to the variable applied cant in curve.

b.iv-2.2  Section [9,4: 9,6]:

Figure 5.1.69: Twist defects in section [9,4: 9,6]

We note that the twist defects are related to cant transition areas. Thus, in order to analyse deeper this kind of defects, it is necessary to know the design twist in these cant transition areas, which will allow us to highlight the real twist defect.
b.iv-2.3  Section [9,6: 9,8]:

Figure 5.1.70: Twist defects in section [9,6: 9,8]

We note that the twist defects are related to cant transition areas. Moreover, these defects are also related to a cant defect.
(v) **Cant**

The following figure presents the standard deviation for the cant per 200m section:

![Figure 5.1.71: Standard deviation for cant per 200m section - V1](image)

**b.v-1 General observations**

We note that, on the overall length of the track V1, the mean standard deviation is 1mm. Thus, the cant defect range is between \(-2\text{mm} \text{ et } +2\text{mm}\) for a 95% compliance and between \(-3\text{mm} \text{ et } +3\text{mm}\) for a 99% compliance.

However, considering a 95% compliance, we note that, in a 200m section, the standard deviation is around 3mm. This value is higher than the acceptance tolerance of the EN 13231-1.

We will study this section in order to define the reasons of this non-compliance.
b.v-2 Non-compliance analysis

b.v-2.1 Section [9,6: 9,8]:

Figure 5.1.9: Cant defects in section [9,6: 9,8]

We note that the quasi-static cant (or applied cant) is inconstant in the first curve. In addition, in this curve, the cant defect oscillates a lot, switching from a negative value to a positive value quickly. This can represent a risk for safety.
(vi) **Horizontal alignment**

The following figure presents the standard deviation for the horizontal alignment per 200m section:

![Horizontal: Standard deviation each 200m](image)

**General observations**

We note that, on the overall length of the track V1, the mean standard deviation is 1.8mm. Thus, the horizontal alignment defect range is between **-3.6mm et +3.6mm** for a 95% compliance and between **-5.4mm et +5.4mm** for a 99% compliance.

However, considering a 95% compliance, we note that, in several sections, the standard deviation is higher than the acceptance tolerance of the EN 13231-1 standard.

We will study this section in order to define the reasons of this non-compliance.
b.vi-1.1 Non-compliance analysis

On all non-compliant sections, we have selected a representative section for this defect: the section [0: 0,2).

Figure 5.1.74: Horizontal alignment defects in section [0: 0,2]

We note that the horizontal alignment exceeds the tolerances of the EN 13231-1 standard in small radius curves. This defect is not due to a track irregularity but to a post-processing defect.
(vii) **Compliance of the track V1 to the EN 13231-1 standard**

For the overall length of the track V1 and for each parameter, the percentage of non-compliant track is listed in the table below:

<table>
<thead>
<tr>
<th>Geometric parameter</th>
<th>Acceptance tolerances</th>
<th>% of non-compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gauge</td>
<td>± 3mm</td>
<td>4.93</td>
</tr>
<tr>
<td>Vertical alignment</td>
<td>± 6mm</td>
<td>0.39</td>
</tr>
<tr>
<td>Cant</td>
<td>± 3mm</td>
<td>1.64</td>
</tr>
<tr>
<td>Twist</td>
<td>± 4.5mm</td>
<td>5.17</td>
</tr>
<tr>
<td>Horizontal alignment</td>
<td>± 5mm</td>
<td>3.43</td>
</tr>
</tbody>
</table>

Table 5.1.17: Percentage of non-compliant track - V1

In order to highlight where these non-compliance are located, we will analyse these parameters in relation to the curve radius. The table presented hereafter shows the results:

<table>
<thead>
<tr>
<th>Geometric parameter</th>
<th>Tolerances</th>
<th>25 &lt; R ≤ 150</th>
<th>150 &lt; R ≤ 500</th>
<th>500 &lt; R ≤ 2000</th>
<th>R &gt;2000 and AD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gauge</td>
<td>± 3mm</td>
<td>6.58</td>
<td>3.54</td>
<td>4.49</td>
<td>3.27</td>
</tr>
<tr>
<td>Vertical alignment</td>
<td>± 6mm</td>
<td>1.10</td>
<td>0.23</td>
<td>0.17</td>
<td>0.09</td>
</tr>
<tr>
<td>Cant</td>
<td>± 3mm</td>
<td>3.36</td>
<td>1.34</td>
<td>1.15</td>
<td>0.81</td>
</tr>
<tr>
<td>Twist</td>
<td>± 4.5mm</td>
<td>10.51</td>
<td>10.43</td>
<td>1.33</td>
<td>0.94</td>
</tr>
<tr>
<td>Horizontal alignment</td>
<td>± 5mm</td>
<td>12.46</td>
<td>0.42</td>
<td>0.30</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Table 5.1.18: Percentage of non-compliant track per parameter and curve radius - V1

The gauge is the most sensitive parameter for concrete slab tracks (directly related to the wear phenomenon in small radius curves), we will compare the repartition graph in relation to the curve radius.

Figure 5.1.75: Comparison of gauge repartition graph in relation to the curve radius- V1
(viii) Conclusion

The percentage of non-compliant track for gauge, vertical alignment, and cant and horizontal alignment is lower than 5%. Thus, considering 95% compliance, we conclude that the track V1 of the ML-3 line is compliant to the EN 13231-1 standard acceptance tolerances for these track parameters. For the twist, the percentage of non-compliant track is a bit higher than 5% but this value may not be the real twist defect. In fact, we will be able to conclude for the compliance of this parameter when we will know the design twist in cant transition areas (areas where we note many twist defects).

In addition, for the gauge, a part of the percentage of non-compliance can be related to the wear phenomenon in small radius curves (for information, the ML-3 line of the Madrid tramway network was opened to traffic in 2007). The rest of this percentage may be related to track installation issues and must be highlighted. For the other parameters, the percentage of non-compliance is either due to a track installation issue or due to a post-processing defect. So, in order to refine the compliance analysis, the track design for each parameter must be an input data for post-processing.

According to the table 4, we can note that, in general, the percentage of non-compliant track increases when the radius curve is decreasing. In addition, we note that for the horizontal alignment, the percentage of non-compliant track is focused on the curve radius range between 25 and 150m and for the twist; it is focused on the curve radius range between 25m and 500m.

According to the comparison of the gauge repartition graphs in relation to the curve radius, we can note that, for straight alignment, the mean gauge is lower to the designed gauge (around 1434.5mm). We can also note that this mean gauge increases when the curve radius is decreasing up to 1436mm for the curve radius range between 25m and 150m.

Note that a particular attention must be paid to the section [9.6: 9.8], near Retamares station, which combine high track geometry defects for several parameters, and so, can represent of safety risk.
(c) **STATISTIC STUDY OF V2: [PUERTA DE BOADILLA-COLONIA JARDIN]**

(i) **Principle:**

The statistic study is based on the same principle than the statistic study of track V1. The non-compliance analysis highlighting the same reasons than these of track V1 analysis, we will only remind these reasons in this study.

The direction of travel for measurements on track V2 is shown hereafter:

![Direction of travel for measurement V2](image)

**Figure 5.1.76: Direction of travel for measurements - V2**

(ii) **Gauge**

The following figure presents the standard deviation for the gauge per section of 200m:

![Gauge: Standard Deviation per 200m](image)

**Figure 5.1.77: Standard deviation for gauge per section of 200m – V2**

c.ii-1 **General observations**

We note that, on the overall length of the track V1, the mean standard deviation is 1.3mm. Thus, the track gauge is **1435mm +/- 2.6mm** for a 95% compliance and **1435mm +/- 3.9mm** for a 99% compliance. However, on all sections analysed and considering a 95% compliance, several sections are not compliant with the EN 13231-1 standard. In fact, the gauge in these sections is higher than +/-3mm.
Remider of the reasons of gauge defects for track V1

- Defect related to a wear phenomenon in small radius curves.
- Defect may be related to a track installation defect

(iii) Vertical alignment

The following figure presents the standard deviation for the vertical alignment per 200m section:

![Vertical: Standard Deviation per 200m](Figure 5.1.78)

**c.iii-1 General observations**

We note that, on the overall length of the track V1, the mean standard deviation for vertical alignment is 1.3mm. Thus, the vertical alignment range is between **-2.6mm and +2.6mm** for a 95% compliance and between **-3.9mm and +3.9mm** for a 99% compliance.

However, considering a 95% compliance, we note that, in a 200m section, the standard deviation is around 3.5mm. So, this section is non-compliant with the EN 13231-1 standard acceptance tolerances for vertical alignment.

Remider of the reasons of vertical alignment defects for track V1

- Defect related to post-processing fault.
- Defect due to a cant defect (non applicable in track V2)
(iv) **Twist**

The following figure presents the standard deviation for the twist per 200m section:

![Twist: Standard Deviation per 200m](image)

**Figure 5.1.79: Standard deviation for twist per 200m section – V2**

**General observations**

We note that, on the overall length of the track V1, the mean standard deviation for twist is 1.7mm. Thus, the twist range is between **–3.4mm and +3.4mm** for a 95% compliance and between **–5.1mm and +5.1mm** for a 99% compliance.

However, considering a 95% compliance, we note in a few sections a standard deviation higher than the EN 13231-1 standard acceptance tolerances.

Reminder of the reasons of twist defects for track V1

- Defect due to a variable applied cant in curve.
- Defect related to cant transition areas.
(v) Cant

The following figure presents the standard deviation for the cant per section of 200m:

![Cant: Standard Deviation per 200m](image)

**General observations**

We note that, on the overall length of the track V1, the mean standard deviation is 1mm. Thus, the cant defect range is between **-2mm et +2mm** for a 95% compliance and between **-3mm et +3mm** for a 99% compliance.

However, considering a 95% compliance, we note that, in 2 no's of 200m section, the standard deviation exceeds the acceptance tolerance of the EN 13231-1 standard.

For the cant, the reason of the defect is different from the reason highlighted for the track V1. So, we will study one of these sections in order to define the reasons of this non-compliance.
Non-compliance analysis

Section [10,2: 10,4]:

Figure 5.1.81: Cant defects in section [10,2: 10,4]

We note that the applied cant is variable in the curve, which generates a cant defect.
(vi) **Horizontal alignment**

The following figure presents the standard deviation for the horizontal alignment per 200m section:

![Figure 5.1.82: Standard deviation for horizontal alignment per 200m section – V2](image)

**General observations**

We note that, on the overall length of the track V1, the mean standard deviation is 1.6mm. Thus, the horizontal alignment defect range is between **-3.2mm et +3.2mm** for a 95% compliance and between **-4.8mm et +4.8mm** for a 99% compliance.

However, considering a 95% compliance, we note that, in several sections, the standard deviation is higher than the acceptance tolerance of the EN 13231-1 standard.

Reminder of the reasons of horizontal alignment defects for track V1

- Defect related to post-processing fault.
(vii) Compliance of the track V2 to the EN 13231-1 standard

For the overall length of the track V2 and for each parameter, the percentage of non-compliant track is listed in the table below:

<table>
<thead>
<tr>
<th>Geometric parameter</th>
<th>Acceptance tolerances</th>
<th>% of non-compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gauge</td>
<td>± 3mm</td>
<td>5,10</td>
</tr>
<tr>
<td>Vertical alignment</td>
<td>± 6mm</td>
<td>0,29</td>
</tr>
<tr>
<td>Cant</td>
<td>± 3mm</td>
<td>1,45</td>
</tr>
<tr>
<td>Twist</td>
<td>± 4.5mm</td>
<td>4,69</td>
</tr>
<tr>
<td>Horizontal alignment</td>
<td>± 5mm</td>
<td>2,73</td>
</tr>
</tbody>
</table>

Table 5.1.19: Percentage of non-compliant track – V2

In order to highlight where these non-compliance are located, we will analyse these parameters in relation to the curve radius. The table presented hereafter shows the results:

<table>
<thead>
<tr>
<th>Geometric parameter</th>
<th>Tolerances</th>
<th>25 &lt; R ≤ 150</th>
<th>150 &lt; R ≤ 500</th>
<th>500 &lt; R ≤ 2000</th>
<th>R &gt;2000 and AD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gauge</td>
<td>± 3mm</td>
<td>4,71*</td>
<td>3,50</td>
<td>4,52</td>
<td>1,74</td>
</tr>
<tr>
<td>Vertical alignment</td>
<td>± 6mm</td>
<td>0,00*</td>
<td>0,14</td>
<td>0,16</td>
<td>0,08</td>
</tr>
<tr>
<td>Cant</td>
<td>± 3mm</td>
<td>4,68*</td>
<td>1,25</td>
<td>0,83</td>
<td>0,43</td>
</tr>
<tr>
<td>Twist</td>
<td>± 4.5mm</td>
<td>0,00*</td>
<td>10,05</td>
<td>0,94</td>
<td>0,73</td>
</tr>
<tr>
<td>Horizontal alignment</td>
<td>± 5mm</td>
<td>9,52*</td>
<td>0,36</td>
<td>1,26</td>
<td>0,11</td>
</tr>
</tbody>
</table>

* Due to a computer issue, this analysis has been done only on 0.9 of the 4km of track having a curve radius range between 25m and 150m.

Table 5.1.20: Percentage of non-compliant track per parameter and curve radius – V2

The gauge is the most sensitive parameter for concrete slab tracks (directly related to the wear phenomenon in small radius curves), we will compare the repartition graph in relation to the curve radius.
(viii) Conclusion

The percentage of non-compliant track for vertical alignment, cant, twist and horizontal alignment is lower than 5%. Thus, considering 95% compliance, we conclude that the track V2 of the ML-3 line is compliant to the EN 13231-1 standard acceptance tolerances for these track parameters.

For the gauge, the percentage of non-compliant track is a bit higher than 5%. A part of this percentage can be related to the wear phenomenon in small radius curves and the rest may be related to track installation issues.

For the other parameters, the percentage of non-compliance is either due to a track installation issue or due to a post-processing defect. So, in order to refine the compliance analysis, the track design for each parameter must be an input data for post-processing.

According to the table, we can note that, in general, the percentage of non-compliant track increases when the radius curve is decreasing, except for horizontal alignment and gauge in curve radius from 500m to 2000m. In addition, we note that for the horizontal alignment and cant, the percentage of non-compliant track is mainly focused on the curve radius range between 25 and 150m and for the twist, it is focused on the curve radius range between 150m and 500m.

According to the comparison of the gauge repartition graphs in relation to the curve radius, we can note that, for straight alignment and radius curves higher than 2000m, the mean gauge is around 1435mm. We can also note that this mean gauge increases when the curve radius is decreasing up to 1436mm for the curve radius range between 25m and 150m.
5.1.4.4. Roughness Measurement

(a) THE MEASUREMENT TROLLEY
The CAT trolley (Corrugation Analysis Trolley) is made up of a measuring head which is adjusted to run in the middle of one rail running band, a stabilising pole, a pushing pole and a laptop, which records the signal. A measurement is recorded every 1mm.

![CAT trolley](image)

Figure 5.1.84: CAT trolley

(b) THE MEASUREMENTS
The measurements were carried out between the 10th and the 12th of May 2010 on 10 sites of Madrid tramway line 3. These sites are usually 100m long and were selected as representative samples of the line. For each site, both rails were measured twice in the middle of the running band.

In this report, right and left rails are considered as in the Figure below:

```
<table>
<thead>
<tr>
<th>Travel direction</th>
</tr>
</thead>
<tbody>
<tr>
<td>right</td>
</tr>
<tr>
<td>left</td>
</tr>
<tr>
<td>left</td>
</tr>
<tr>
<td>right</td>
</tr>
</tbody>
</table>
```

![Right-left rail definition](image)

Figure 5.1.85: Right-left rail definition

The results show a good repeatability.
The following table gives an overview of the measured sites.

<table>
<thead>
<tr>
<th>Site</th>
<th>track</th>
<th>kp</th>
<th>Alignement</th>
<th>Type</th>
<th>Observation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>13+200-13+300</td>
<td>Straight</td>
<td>classic</td>
<td>Slope</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>11+200-11+300</td>
<td>Straight</td>
<td>comfort</td>
<td>Acceleration – slope</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>11+100-11+200</td>
<td>Straight</td>
<td>comfort</td>
<td>Slope</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>10+900-11+000</td>
<td>Straight</td>
<td>comfort</td>
<td>Slope</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>10+820-10-852</td>
<td>Curve</td>
<td>classic</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>3+300-3+400</td>
<td>Straight</td>
<td>classic</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>2</td>
<td>1+620-1+720</td>
<td>Straight</td>
<td>classic</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>2</td>
<td>1+552-1+592</td>
<td>S-curve</td>
<td>classic</td>
<td>Acceleration</td>
</tr>
<tr>
<td>9</td>
<td>2</td>
<td>1+215-1+290</td>
<td>Straight</td>
<td>Classic + manta</td>
<td>No lateral mat</td>
</tr>
<tr>
<td>10</td>
<td>2</td>
<td>0+780-0+880</td>
<td>Straight</td>
<td>Classic + manta</td>
<td>MLO reports corrugation in this area</td>
</tr>
</tbody>
</table>

Table 5.1.21: Measurement sites

Pictures of the 10 sites are shown below.

Sites 1 to 4 present an important slope.

Paving is generally deactivated concrete or impressed concrete, which are clean surfaced. But in a few sites, coating are paving stones, which are sometimes damaged by the expansion or dirt. These types of coating provokes dust to be on the rails and thus premature wear

Site 1

![Image of Site 1]
Site 2 & 3

Site 4
Situated station …

Site 5
Site 6

Site 7

Site 8

Site 9
RESULTS
The data analysis was carried out in line with both standards ISO 3095 and ISO 13231-3. In the present chapter, the different results obtained are presented.

ISO 3095

(i) Roughness levels
Roughness levels are calculated in 1/3-octave bands as defined in standard ISO 3095. The roughness level represented in the graphs below is the mean value of both rails roughness measured twice.

Target ISO 3095 level for reference tracks is indicated in red and the thick blue lines represents the range within which the roughness of Montpellier tramway network is situated. Great variations of roughness are usually met along one line.

For “classic” and “manta” track types, roughness variation according to measurement site is small. For wavelengths smaller than 200mm, the roughness is within 5dB of ISO 3095 limit for reference track.

Extra high performance jacket track type sections measured appear to have a much higher roughness. This can be explained by the coating, which is dirt on site 4. On sites 2 and 3, the pavement coating is worn. The graph below shows that site 4 rails roughness is much higher than the other two.

The sections measured here are on descending slopes, and the use of sand for breaking and / or a higher velocity can cause premature wear.

For long wavelengths, roughness is higher than on other tramway networks.

Moreover, corrugation with a wavelength 50-63mm appears on site 2.
Figure 5.1.87: Classic and floating slab track types roughness spectra

Figure 5.1.88: Comfort track type roughness spectra
(ii) Conclusions

The track roughness measured on Madrid network is particularly good in comparison with what could be found on some other networks.

The reason for this good track roughness is somewhat difficult to find even after analysis of measurement collected.

Is it related to:

− Madrid segregated track network?
− Grinding performed from March to November 2008?
− Continuously supported rails?
− Low traffic?

It is difficult to draw a conclusion at the moment.
5.1.4.5. Pass-by Measurement: Noise and Vibration

(a) SET-UP

The noise and vibration levels of vehicles passing by were measured on 19 May 2010. This was done for the three different systems/sections.

Accelerometers are placed on a straight line, perpendicular to the rail at distances 3 m (V1), 4 m (V2), 6 m (V3) and 8 m (V4) from the outer rail.

Microphones are placed at 7.5 m from centre track at 1.2 m (N1) and 3.5 m (N2) above top of rail. Vehicles drive at a constant speed of 30 km/h for all sections. The vehicle speed is verified using a speed radar.

Following paragraphs show pictures of the measurement setups for each site (system).

(b) SITE S1 CLASSIC + MANTA

Figure 5.1.89: Picture of Site S1 Classic + Manta
(i) Site S2 Classic

Figure 5.1.90: Picture of Site S3 Classic

(ii) Site S3 Comfort

Figure 5.1.91: Picture of Site S3 Comfort
(c) **RESULTS OF NOISE MEASUREMENTS**

All detailed measurement results can be found in appendix, showing the LMAX and LEQ spectra and the overall levels in function of the time for each pass-by measurement.

The measurements are summarized in tables Table 5.1.21. The A-weighted noise levels are about 3dB(A) higher for the system ‘Classic + Manta’ than for ‘Classic’ and ‘Comfort’. The noise levels for those last two are similar.

<table>
<thead>
<tr>
<th>Site</th>
<th>System</th>
<th>Direction</th>
<th>N1 [dB(A)]</th>
<th>N2 [dB(A)]</th>
<th>N1 [dB(A)]</th>
<th>N2 [dB(A)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>S1: 0+820</td>
<td>Classic + Manta</td>
<td>Colonia Jardin</td>
<td>73.4</td>
<td>72.5</td>
<td>70.3</td>
<td>69.1</td>
</tr>
<tr>
<td>S2: 1+760</td>
<td>Classic</td>
<td>Colonia Jardin</td>
<td>70.4</td>
<td>69.8</td>
<td>67.9</td>
<td>67.1</td>
</tr>
<tr>
<td>S3: 13+060</td>
<td>Comfort</td>
<td>Colonia Jardin</td>
<td>69.8</td>
<td>70.2</td>
<td>67.2</td>
<td>67.4</td>
</tr>
</tbody>
</table>

Table 5.1.21: Overview

(d) **RESULTS OF VIBRATION MEASUREMENTS**

All detailed measurement results can be found in appendix, showing the LMAX and LEQ spectra and the overall levels in function of the time for each pass-by measurement.

The measurements are summarized in table

<table>
<thead>
<tr>
<th>Site</th>
<th>System</th>
<th>Direction</th>
<th>LEQ V2 [dB(re.1e-9m/s)]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>V1</td>
</tr>
<tr>
<td>S1: 0+820</td>
<td>Classic + Manta</td>
<td>Colonia Jardin</td>
<td>92.9</td>
</tr>
<tr>
<td>S2: 1+760</td>
<td>Classic</td>
<td>Colonia Jardin</td>
<td>90.9</td>
</tr>
<tr>
<td>S3: 13+060</td>
<td>Comfort</td>
<td>Colonia Jardin</td>
<td>89.7</td>
</tr>
</tbody>
</table>

Table 5.1.22: Overview

He global LEQ vibration level for ‘Classic + Manta’ is 1.2 to 3.2 dB, higher than for ‘Classic’. The global LEQ vibration level for ‘Comfort’ is 0.5 to 3.0 dB, lower than for ‘Classic’.

<table>
<thead>
<tr>
<th>Site</th>
<th>System</th>
<th>Direction</th>
<th>LMAX [dB(re.1e-9m/s)]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>V1</td>
</tr>
<tr>
<td>S1: 0+820</td>
<td>Classic + Manta</td>
<td>Colonia Jardin</td>
<td>97.4</td>
</tr>
<tr>
<td>S2: 1+760</td>
<td>Classic</td>
<td>Colonia Jardin</td>
<td>96.0</td>
</tr>
<tr>
<td>S3: 13+060</td>
<td>Comfort</td>
<td>Colonia Jardin</td>
<td>92.5</td>
</tr>
</tbody>
</table>

Table 5.1.22: Overview
CONCLUSION

The approach pertaining to vibration mitigation has to consider the whole vibration transmission path and to be adapted to the vehicle that will be operated.

And the vibration performance of a transportation system must be defined in relation to its compliance with a level of vibration measured at building locations and not by empirical rules based on outdated data.

The measurements made on many other sites for several tram designs and manufacturers are correlating what could be observed in Madrid:

- A very high performance system (floating slab) is rarely required,
- A continuously supported rail system is sufficient in many cases.
5.1.5 Metro track form

5.1.5.1. The Mathematical Model

In the following paragraph the main characteristics of the numerical model of train-track interaction as well as of the wheel-rail contact model are briefly described.

(A) TRAIN-TRACK INTERACTION MODEL

An efficient multibody modelling approach for the study of the running behaviour of railway vehicles was developed in recent years by a research group established at the Department of Mechanical Engineering, Politecnico di Milano [1], [2], [3].

The mathematical model is based on a multi-body, large displacement schematisation of the trainset, allowing to analyse the non-stationary behaviour considering tangent track running and curve negotiation.

The system under study is subdivided into elementary units of the following types:

- Carbody, modelled as a single rigid body;
- Bogie assembly, modelled as a rigid bogie frame connected by primary suspensions to two flexible wheelsets;
- Other bodies, e.g. motors, converters, assumed to displace rigidly and being attached either to a carbody or to a bogie frame.

Elementary units are connected to each other by means of elastic and damping elements (linear and nonlinear) reproducing secondary suspensions, links between carbodies, and connections to other bodies (e.g. elastic motor suspension). By combining the above listed elementary units, any trainset architecture may be derived, such as rail vehicles formed by two bogies and one carbody, or more complex configurations formed by several articulated carbodies.

Each rigid body is assigned with 5 degrees of freedom, the forward speed of body centre of mass being set to a constant value \( V \), whereas for each flexible wheelset the movement with respect to the moving reference is defined as the linear combination of the unconstrained wheelset eigenvectors. For the study of vehicle stability and of ride safety, the model can be limited to the five modes corresponding to the rigid motions of the unconstrained wheelset (again considering a forward motion at constant speed), whereas in the study of high-frequency interaction of the train with the track (with applications e.g. to turnouts transit) bending and torsion modes as well as some modes corresponding to local deformation of the wheels are typically included in the analysis.

To represent kinematical effects associated with curve negotiation, the motion of each elementary unit is described with respect to a moving reference system travelling at constant speed along the track centreline, with \( Z \) axis tangent to the track centreline and \( X \) axis orthogonal to the rail level. By assuming small displacements relative to the moving reference of each module in the trainset, the equations of motion are linearised with respect to kinematical nonlinear effects only, and take the form:
\[
[M_\nu][\ddot{X}_\nu] + [C_\nu][\dot{X}_\nu] + [K_\nu]X_\nu = F_\nu(V, t) = F(V, t) + F_{nl}(X_\nu, \dot{X}_\nu) + F_C(X_\nu, \dot{X}_\nu, \ddot{X}_\nu, V, t)
\]

where \([M_\nu], [C_\nu] and [K_\nu]\) are the mass, damping and stiffness matrices of the trainset, \(X_\nu\) is the vector of trainset coordinates, \(F_\nu(V, t)\) is the vector of generalised forces produced in the secondary suspensions and carbody links by the different motions of the moving references associated with the modules connected by the suspension, \(F_{nl}(V, t)\) is the vector of inertial forces due to the non-inertial motion of the moving references, \(F_{nl}\) is the vector of nonlinear forces due to nonlinear elements in the suspensions and \(F_C\) is the vector of generalized forces due to wheel-rail contact, depending upon the motion of the trainset \(X_\nu, \dot{X}_\nu\) and upon additional coordinates \(X_\nu\) representing the track motion and their time derivatives \(\dot{X}_\nu\). The dynamic behaviour of the track being negotiated by the train is represented by the additional equation:

\[
[M_\nu][\ddot{X}_\nu] + [C_\nu][\dot{X}_\nu] + [K_\nu]X_\nu = F_C(X_\nu, \dot{X}_\nu, \ddot{X}_\nu, V, t)
\]

where \([M_\nu], [C_\nu] and [K_\nu]\) are the track mass, damping and stiffness matrices, defined by a linear finite element schematisation, and \(F_C\) is the vector of generalized nodal forces acting on the track due to wheel rail contact forces.

Equations (1) and (2) take the form of two coupled sets of nonlinear differential equations, with the coupling term being provided by wheel-rail contact forces.

**Figure 5.1.92: Train-track dynamic interaction model**
(B) **Wheel-Rail Contact Model**

Being the vehicle and track subsystems considered separately, the contact forces exchanged at wheel-rail interface are expressed as a function of both subsystems motion. This is why the independent coordinates $X_t$ and $X_v$ appear both in the track and in the vehicle generalized forces, as coupling terms between the two sets of equations.

At each integration step, the procedure adopted for contact forces computation follows the scheme reported below:

1. a first attempt solution for the independent variables $X_t$ and $X_v$ is predicted;
2. on the basis of the shape functions of the track finite element model, the track displacement at each wheel-rail contact point is calculated, together with its time derivative;
3. using modal superposition or rigid body kinematics relations, the vehicle displacement at each wheel-rail contact point is calculated, together with its time derivative;
4. starting from the displacements mentioned in items 2 and 3 and their time derivatives, the normal ($N_j$) and tangential ($F_{Tj}$, $F_{Lj}$) contact forces acting on the $j$-th wheel-rail couple are evaluated (Figure 2);
5. using again the shape functions of the track finite element model and the vehicle’s modal shapes, the contribution of the forces $N_j$, $F_{Tj}$, $F_{Lj}$ to the generalised forces $E_v$ and $E_t$ acting on the vehicle and on the track degrees of freedom is evaluated; steps from 2) to 5) are repeated for all wheels;
6. finally, once that the generalised forces $E_v$ and $E_t$ are determined, the equations (1) and (2) are solved, and a better approximation of the solution ($X_t$ and $X_v$) is achieved.

As far as point 4) is concerned, the computation of the normal contact forces $N_j$ is based on a multi-hertzian model (Bruni et al. 1999), while the tangential forces $F_{Tj}$ and $F_{Lj}$ are obtained according to the Shen, Hedrick & Elkins formulation (1983).

The wheel-rail contact model (Bruni et al. 1999) is based on a preliminary geometrical analysis, which can be carried out both on new and on measured wheel/rail worn profiles. Contact parameters such as the local radii of curvature and the contact angle are reported in table form, as functions of wheel-rail lateral displacement. The geometrical analysis also allows to determine the number of the potential contact points for a given wheel-rail relative position.

The sets of equations (1) an (2) are numerically integrated using a modified Newmark method. At each integration step, convergence on the wheel-rail contact forces is reached iteratively. In this way, the dynamic response of each subsystem is automatically dependent on the motion of the other part of the system. Moreover, the forcing effect associated with the rail roughness is accounted for, in terms of rail displacements superimposed to the ones calculated, at the generic integration step, as a function of the track nodal coordinates. For given irregularity profile and train velocity, the former displacements are only a function of time.

As a result of the integration procedure, the time histories of the following quantities can be obtained:

- Track displacements/accelerations;
- Vehicle displacements/accelerations;
- Forces transmitted from the vehicle to the track.
rail and wheel profiles  geometrical analysis  contact geometrical parameters

elastic deformation in normal direction (penetration) normal forces (multi-hertzian model) generalized contact forces
tangential & longitudinal creepages tangential & longitudinal forces (Shen-Hedrick-Elkins model)

Figure 5.1.93: Wheel-rail contact model
5.1.5.2. Metro Madrid

(a) VEHICLE AND TRACK CHARACTERISTICS

Figure 5.1. shows a sketch of the implemented model of the Metro Madrid coach. The main vehicle characteristics are:

- Maximum axle load = 154kN;
- Tare axle load = 77kN;
- Wheelbase = 2.2 m;
- Pivot pitch = 11.1m.

Figure 5.1.94: Sketch of the implemented model of the Metro Madrid coach

In all the simulations presented hereafter, the vehicle has been assumed in full load condition (axle load = 154kN). The elastic and damping characteristics of both the primary and secondary suspensions are taken from a metro vehicle with similar characteristics.

Figure 5.1.93 shows the considered wheel irregularity (multiplied by a factor of 1000). Note that the shape of such irregularity changes form wheel to wheel, but has the same wavelengths and wave amplitudes.
Figure 5.1.95: Wheel irregularity (multiplied by 1000): right wheel (blue), left wheel (red).

Track irregularity input has been defined according to ORE B176 PSD function (small amplitude defects). Figure 5.1.94 shows the PSD function of the track vertical profile, while figure 5.1.95 shows the track vertical irregularity profile, reconstructed on the basis of a random phase generation. Note that, although not shown here, the ORE irregularity provided in input to the train-track interaction simulation includes also cross-level and alignment irregularities.

As far as the wheel and rail profiles are concerned, the ORE S1002 and the UIC54 rail were considered.

(i) **Track Model**

For the purpose of this research, a numerical model of the embedded-rail track system has been developed (Figure 5.1.) and implemented into PoliMi’s train-track dynamic simulation code. The linear visco-elastic characteristics of the REMS elastic bed take place between rails and tunnel (both schematised as equivalent Euler-Bernoulli beams). The bending stiffness and the mass per unit length of the rail can be easily determined form the rail producer catalogue, the properties of the equivalent beam representing the tunnel and those of the visco-elastic bed representing the ground were identified from impulsive tests carried out some years ago in Milano underground. The visco-elastic bed representing the ground has been introduced to determine the transmissibility (i.e. the capability of filtering out vibrations induced by train passage) of the considered REMS tracks.

Figure 5.1.96. Implemented model of the REMS track system
Stiffness characteristics of the embedded rail system were originally calculated by means of 2D FEM analyses. As an example Figure 5.1 shows the computed static response of the system (displacements and Von Mises stresses in the rubber jacket) for the comfort sample. This study was useful to design the three samples as a function of the desired train-track interaction. Once stiffness characteristics were determined, samples were manufactured by CDM and subsequently tested in PoliMi labs.

Figure 5.1.97. Embedded system static Von Mises stresses, Comfort sample.

For the three samples, in order to identify the real REMS visco-elastic bed characteristics, a series of tests were performed (Figure 5.1.95) in two different configurations (Figure 5.1.10 and Figure 5.1.11) useful to simulate typical loads for straight and curved track. Thanks to these tests it was possible to determine vertical / lateral equivalent stiffness and damping of the visco-elastic bed for each of the three REMS samples. Also fatigue and aging tests were performed in order to evaluate durability and possible variations of the identified parameters (see deliverables D1.1 and D3.1 for more details about the experimental test methodology and results).
For each sample both static and dynamic characterisation has been performed. As an example in Figure 5.1, the vertical stiffness of the Comfort sample is reported. During dynamic tests the sample has been forced with different actuation frequency and pre-load. The resulting stiffness is therefore reported as a function of these two parameters.

![Figure 5.1.10. 0-degrees test configuration, straight track.](image)

![Figure 5.1.11. 26-degrees test configuration, curve track.](image)

![Figure 5.1.98. Picture of the PoliMi test bench in the C4 laboratory, Milan](image)

![Figure 5.1.99. Vertical stiffness of the Comfort sample, function of the preload and of the actuation frequency.](image)
5.1.5.3. Numerical Results

The collected data (see experimental evaluation of the REMS visco-elastic bed etc.) both for train and track characteristics, allow PoliMi’s simulation code to provide the dynamic train-track response due to a known track geometry and train speed for the three types of REMS tracks considered in the present research, i.e. compact, classic and comfort tracks. Both wheel and rail irregularity input data were considered in all the simulations. While measured irregularity patterns were considered for the wheel, track irregularity input was generated according to the PSD functions defined in the ORE B176 Standard.

As an example, Figure 5.1. shows the numerically simulated time histories of the vertical rail and tunnel accelerations during tangent track running at 110km/h. Results are reported for the three REMS tracks and for the traditional direct fixation track. It can be clearly seen that the REMS track is able to reduce both rail and tunnel accelerations, in terms of amplitudes and of duration, since the longitudinal transmission of the vibration is better damped.

![Graphs showing numerical results for different track types](image)

Figure 5.1.100. Numerical simulations for the estimation of vibration mitigation performance (REMS and direct fixation). Metro Madrid train running in tangent track at 110km/h (full load). Simulations output: rail and tunnel acceleration time histories.
In the following, as an example, other typical results for assessing the dynamic behaviour of the track are shown: vertical rail displacement and gauge widening (Figure 5.1.12 and Figure 5.1.) during curved track running. All results are plotted when the first axle of Metro Madrid vehicle has covered a significant length in full curve.

As expected, the vertical displacement of the rail is higher for the REMS tracks than for the direct fixation track. However, even in the worst case (comfort track), the vertical displacement of the REMS track is smaller than 3 mm. The gauge widening is almost equal for both the REMS and the direct fixation track, due to similar lateral deformability. An absolute value of about 1.2 mm can be observed for the comfort track on a 300 m radius curve for the fully loaded vehicle.

Figure 5.1.121. Vertical rail displacement for the Comfort track (full load) and gauge widening.

Figure 5.1.102. Vertical rail displacement for the Direct Fixation track (full load) and gauge widening.
To assess track transmissibility, the power spectral densities of both the rail and the tunnel vertical accelerations were evaluated (Figure 5.1.). Thus, the transfer function of the tunnel vertical acceleration with respect to the rail vertical acceleration was computed showing the good filtering properties of all three REMS tracks (Figure 5.1.13). Note that the comfort track is able to completely filtering out the resonance peak at approx. 25 Hz.
Figure 5.1. shows the maximum value of wheel-rail contact force inclination angle for the compact track (a), for the classic track (b) and for the comfort track (c) with zero preload and both tare and full loaded vehicles. It can be seen that the highest inclination angle is equal to 21 deg and is obtained for the comfort track for the 300 m radius curve. Note that the results shown in figure 10 are obtained by using a one metre moving average to assess the mean behaviour during curve negotiation thus filtering out spikes associates to track and / or wheel irregularity. The maximum inclination angle is used for determining the fatigue behaviour of the tracks.

Another important parameter to assess the dynamic behaviour of REMS tracks during curve negotiation is the maximum value of the outer wheel lateral contact force component (Figure 5.1.). The same one metre moving average filter has been used. It can be seen that the highest lateral contact force component is obtained for the fully loaded vehicle running along the 300 m radius curve of both the compact or comfort tracks.

Figure 5.1.105. Maximum value of wheel-rail contact force inclination angle for the compact track (a), for the classic track (b) and for the comfort track (c) with zero preload and both tare and full loaded vehicles (1m moving average)

Figure 5.1.106. Maximum value of the outer wheel lateral contact force component for the compact track (a), for the classic track (b) and for the comfort track (c) with zero preload and both tare and full loaded vehicles (1m moving average)
5.1.5.4. References


5.1.6 General conclusion

5.1.6.1. Conclusions on Tramway Track form

A wide range of subjects was dealt with in the present document, all of them related to the forces applied to the track and the resulting stresses in rail, and leading to conclusions on track geometry, track roughness and vibration mitigation.

Complete models including track and vehicles were setup allowing for simulations taking into account the actual conditions of the track.

(a) TRACK GEOMETRY

The track geometry was studied in relation with its mechanical response and the resulting forces and stresses undergone. Following the measurements in Madrid of track for gauge, vertical alignment, and cant and horizontal alignment, an approach based on 95% compliance can be applied to conclude that the track V1 of the ML-3 line is compliant to the EN 13231-1 standard acceptance tolerances for these track parameters.

The results of the track geometry study can be proposed as a contribution for the development of urban track standards.

(b) RAIL FATIGUE

The rail fatigue was studied with two practical purposes: to be able to make a commitment on the lifetime of the rails installed and to optimise the rail replacement frequencies for the track maintenance.

The development of that study was the opportunity to implement the stationary method, which enables to undertake fast simulation of a high number of cycles.

(c) TRACK ROUGHNESS

The track roughness measured on Madrid network is particularly good in comparison with what could be found on some other networks.

The reason for this good track roughness is somewhat difficult to find even after analysis of measurement collected.

It is difficult to draw a conclusion at the moment.

(d) VIBRATION MITIGATION

The approach pertaining to vibration mitigation has to consider the whole vibration transmission path and to be adapted to the vehicle that will be operated.

And the vibration performance of a transportation system must be defined in relation to its compliance with a level of vibration measured at building locations and not by empirical rules based on outdated data.

The measurements made on many other sites for several tram designs and manufacturers are correlating what could be observed in Madrid:

– A very high performance system (floating slab) is rarely required,
– A continuously supported rail system is sufficient in many cases.
5.1.6.2. Conclusions on Metro Track form

Finally the study of the Madrid metro track form based on the REMS system was finalised following the calibration and adjustment phases carried out during laboratory testing.
5.2 FONCTIONAL SPECIFICATIONS FOR TRACK INFRASTRUCTURE

5.2.1 Sensitivity analysis on tramway track parameters by means of numerical simulation of train-track dynamic interaction

Within WP5.2, a numerical train-track interaction model was set-up to carry out a sensitivity analysis on tramway and metro track design parameters to evaluate their effects on track stability and vibration transmission. The following parameters were considered:

- different track design options (embedded rail, direct fixation, floating slab track) and rail support stiffness values;
- support stiffness variations corresponding to new/degraded track conditions;
- axle load variations (tare and full load);
- track irregularity corresponding to two different levels of degraded track geometry.

This work consisted of four different sections:

- sensitivity analysis for tramway tracks
- sensitivity analysis for metro tracks
- light rail parameters
- evaluation of existing standards
- overview of a new embedded track standard under development in the US by APTA/AREMA

5.2.1.1 Introduction

As a part of functional requirements for urban railway tracks, knowing the influence of several parameters on important factors such as track stability and vibrations. It is impossible to carry out measurements in repeatable conditions on different tracks with different vehicles at different speeds. Therefore, it has been decided to carry out this sensitivity analysis numerically with a software acquired by D2S and APT: Vi-Rail. It is a multi-body software dedicated to railway applications.

With the help of this software, along with a Finite Element Software (MSC.Nastran) and a programming tool (Matlab), simulations were carried out with following varying parameters:

(a) VEHICLE TYPE

- Tramway
  - Rail type: EB50T, NP4am, 35G Ri59N, Ri53N;
  - Track layout: 30 m radius curved track and straight track;
  - Vehicle velocity: 20 km/h in curve and 20, 40, 60 km/h on straight track.

- Light Rail
  - Rail type: UIC50, UIC54;
  - Track layout: 100 m radius curved track and straight track;
  - Vehicle velocity: 20, 40 and 60 km/h in curve and 20, 40, 60, 80, 100 km/h on straight track.
(b) **TRACK TYPE**
- Standard ballasted track with fixation every 60 and 75 cm;
- Track with rail directly fastened to a concrete slab: soft and hard railpad stiffness’s with fixation every 60 and 75 cm;
- Track with continuously supported rail: CDM-Classic and CDM-Comfort.

(c) **RAIL IRREGULARITIES**
- Measured rail irregularities
- Step function rail irregularity

The combination of these parameters leads to a large number of simulation. The first section of this report presents the methodology used to carry out all these simulation. The values used for all the parameters are presented in the second section while the last chapter discusses the results.

5.2.1.2. **Description of the methodology**

The aim of this sensitivity analysis is to study the influence of several parameters (type of rail, type of rail support, type of vehicle, speed ...) that one will call the input and two major quantities: the vibrations under the track and the stresses in the rail generated by a passage of a vehicle. The latter quantities can be considered as the outputs of the model.

The methodology is based on using several software to calculate the requested quantities. The figure below is schematic view of the methodology.

![Figure 5.2.1 Methodology used in the study](image-url)
CALCULATION OF THE FORCES AND DISPLACEMENTS IN THE TRACK

The forces applied by the vehicle on the rail and the subsequent displacement are calculated in Vi-Rail and post-processed in Matlab for presentation and figure generation.

(i) Calculations in Vi-Rail

Vi-Rail is a multi-body simulation software dedicated to railway applications. Vi-Rail is built upon MSC.ADAMS, widely recognized as the world’s leading mechanical system simulation tool. This software allows the simulation of the passage of a given vehicle on a given track at various speeds and to calculate the forces applied by this vehicle on the rails.

Moreover, rail irregularities can be modelled as well. Measured rail irregularities can be included in the model: the subsequent generated track displacement are thus varying over time, vibrations are generated. Using these irregularities therefore allows us to study vibrations generated by a vehicle passage, as one would measure vibration levels on a real track.

The figure below shows a schematic view of vehicle-track model in Vi-Rail.

(ii) Calculation in Matlab

The forces and displacements calculated with Vi-Rail are exported in text format and read in Matlab for post-processing.

Forces are plotted in function of time and the average value is saved in a spreadsheet for later use as an input in the FE rail model for stresses calculation.

The displacements over time are first transformed into spectra in the frequency domain through a Fast Fourier Transformation (fft). They are then derived to obtain the vibration velocities and plotted in third-octave band spectra. These spectra are the final results (output) regarding the vibrations. Conclusions can be directly drawn from there.
(b) **CALCULATION OF STRESSES IN THE RAIL IN MSC.NASTRAN**

The mean forces saved in the spreadsheet are used as input forces on Finite Element analysis on rail models in MSC.Nastran. MSC.Nastran is a well-known Finite Element structural analysis software allowing following types of calculation: linear static, modal analysis, thermal-transfer, acoustics, frequency-transient-spectral response, non-linear static and dynamic, optimization and aeroelasticity.

A Finite Element Model of a section of the rail on its support is model in Patran (pre and port-processing software for Nastran) and the forces coming from the spreadsheet, the wheel/rail contact forces, are applied on the rail head to calculate the stresses in the rail web and in the rail foot. These calculated stresses are the final results regarding the stress calculation and conclusions can be drawn from there.
5.2.1.3. Presentation of the models and their parameters

In this section, the vehicle and track models are presented as well as their parameters.

(a) PRESENTATION OF THE VEHICLE MODEL

For the purpose of this study, two types of urban vehicle have been modelled: a light rail vehicle and a tramway vehicle.

(i) Light rail vehicle

The light rail vehicle model is based on the metro from the RATP running on line 8. It is mainly made of a car body on two bogies.

a.i-1 Vehicle car body

Important parameters for the multi body of the light rail vehicle’s car body are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (with passengers)</td>
<td>24.5 *10³ kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>15*10³ kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>390*10³ kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>390*10³ kgm²</td>
</tr>
<tr>
<td>Vertical location of centre of gravity above railhead</td>
<td>1.8 m</td>
</tr>
<tr>
<td>Longitudinal distance between bogie centres</td>
<td>10.0 m</td>
</tr>
</tbody>
</table>

a.i-2 Vehicle bogies (without axles)

Important parameters for the multi body of the light rail vehicle’s bogies without axles are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>1800 kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>760 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>2.2*10³ kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>2.2*10³ kgm²</td>
</tr>
<tr>
<td>Vertical location of centre of gravity above railhead</td>
<td>0.5 m</td>
</tr>
<tr>
<td>Longitudinal distance between wheel sets</td>
<td>2.0 m</td>
</tr>
</tbody>
</table>

a.i-3 Vehicle axles

Important parameters for the multi body of the light rail vehicle’s axles are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>935 kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>530 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>85 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>530 kgm²</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>0.43 m</td>
</tr>
</tbody>
</table>
### a.i-4 Vehicle’s suspension

The vehicle’s suspension elements are mainly divided in the primary and secondary suspension: the primary suspension links the bogie to the axles while the secondary suspension links the bogie to the car body.

<table>
<thead>
<tr>
<th>Suspension Type</th>
<th>Longitudinal Stiffness</th>
<th>Lateral Stiffness</th>
<th>Vertical Stiffness</th>
<th>Longitudinal Damping</th>
<th>Lateral Damping</th>
<th>Vertical Damping</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Primary suspension</strong></td>
<td>1.4 MN/m</td>
<td>1.4 MN/m</td>
<td>1.2 MN/m</td>
<td>5.0 kNs/m</td>
<td>5.0 kNs/m</td>
<td>5.0 kNs/m</td>
</tr>
<tr>
<td><strong>Secondary suspension</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lateral stiffness</td>
<td>200 kN/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vertical stiffness</td>
<td>300 kN/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lateral damping</td>
<td>Viscous damping (see figure below)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vertical damping</td>
<td>in compression = 2.3 kNs/m</td>
<td></td>
<td>in traction = 5.7 kNs/m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Yaw damping</td>
<td>5.0 kNs/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Figure 5.2.3 Behaviour of the secondary lateral damper

Figure 5.2.4 Behaviour of the secondary vertical damper
(ii) **Tramway vehicle**

The tramway vehicle is the T3000 built by Bombardier for the STIB in Brussels. The figure below represents the vehicle.

![Diagram of T3000 Tramway](image)

**Figure 5.2.5 Drawings of the T3000 STIB Tramway**

### a.ii-1 Vehicle car body

Important parameters for the multi body of the tramway vehicle’s car body are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (fully loaded)</td>
<td>$47.9 \times 10^3$ kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>$15 \times 10^3$ kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>$390 \times 10^3$ kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>$390 \times 10^3$ kgm²</td>
</tr>
<tr>
<td>Vertical location of centre of gravity above railhead</td>
<td>1.38 m</td>
</tr>
<tr>
<td>Longitudinal distance between bogie centres</td>
<td>9.369 m</td>
</tr>
</tbody>
</table>

### a.ii-2 Vehicle bogies (without axles)

Important parameters for the multi body of the tramway vehicle’s bogies without axles are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>3116 kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>1722 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>1476 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>3067 kgm²</td>
</tr>
<tr>
<td>Vertical location of centre of gravity above railhead</td>
<td>0.3 m</td>
</tr>
<tr>
<td>Longitudinal distance between wheel sets</td>
<td>1.85 m</td>
</tr>
</tbody>
</table>
**a.ii-3  Vehicle axles**

Important parameters for the multi body of the light rail vehicle’s axles are listed below.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass</td>
<td>733.5 kg</td>
</tr>
<tr>
<td>Mass moment of inertia in roll</td>
<td>810 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in pitch</td>
<td>112 kgm²</td>
</tr>
<tr>
<td>Mass moment of inertia in yaw</td>
<td>810 kgm²</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>0.29 m</td>
</tr>
</tbody>
</table>

**a.ii-4  Vehicle’s suspension**

The vehicle’s suspension elements are mainly divided in the primary and secondary suspension the primary suspension links the bogie to the axles while the secondary suspension links the bogie to the car body.

<table>
<thead>
<tr>
<th>Suspension Type</th>
<th>Longitudinal stiffness</th>
<th>Lateral stiffness</th>
<th>Vertical stiffness</th>
<th>Longitudinal damping</th>
<th>Lateral damping</th>
<th>Vertical damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary suspension</td>
<td>1.84 MN/m</td>
<td>1.84 MN/m</td>
<td>1.58 MN/m</td>
<td>5.0 kNs/m</td>
<td>5.0 kNs/m</td>
<td>5.0 kNs/m</td>
</tr>
<tr>
<td>Secondary suspension</td>
<td>Lateral stiffness</td>
<td>200 kN/m</td>
<td>300 kN/m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Vertical stiffness</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Lateral damping</td>
<td>Viscous damping (see figure below)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Vertical damping</td>
<td>in compression = 2.3 kNs/m</td>
<td>in traction = 5.7 kNs/m</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Yaw damping</td>
<td>5.0 kNs/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

As no data was available on the damping of the suspension, the same values as for the light rail vehicle were used.
Figure 5.2.6 Behaviour of the secondary lateral damper

Figure 5.2.7 Behaviour of the secondary vertical damper
(b) **PRESENTATION OF THE TRACK MODELS**

The track is modelled twice: once in Vi-Rail for the multi-body simulation and once in MSC. Nastran for the calculation of stresses in the rail.

Track models vary in function of the support used: ballasted track, continuously supported rail on concrete slab or directly fastened rail on concrete slab.

Track models vary also in function of the type of rail used: NP4am, EB50T, 35G, Ri59N or Ri53N for the tramway vehicle and UIC50 or UIC54 for the light rail vehicle.

Finally, the track models vary also in function of their layout: curved track (30 m radius curve for the tramway vehicle and 100 m radius curve for the tramway vehicle) or straight track.

(i) **Rail support type**

The track model in Vi-Rail is a lumped parameters model as shown on the figure below.

![Schematic view of the flexible track model in Vi-Rail](image)

The parameters presented in this section are directly introduced in the model. Because the parts under the base bushing (see dummy parts here above) are fixed, it is not possible to extract there displacement. Therefore, displacements are measured above the sleeper and under the rail bushings. In the case of the ballasted track, the base bushing corresponds to the ballast. In the case of direct fixation and continuously supported track, there is no sleeper and no ballast. The base bushing stiffness is thus set to a very high value. For uniformity among the simulation, the base bushing for the ballasted track was also set to a very high value and instead, in the post-processing procedures, a filter representing the ballast is applied to the vibration levels measured under the rail bushings. This filter is represented on the figure below.
Figure 5.2.9 Frequency spectrum of the filter used to model the ballast

### b.i-1 Ballasted track

The ballasted track is rather standard the rail is fixed on a wooden sleeper through a rail pad and the sleepers are placed on the ballast. The parameters of the different elements constituting the model are listed below:

<table>
<thead>
<tr>
<th>Element</th>
<th>Young’s modulus E</th>
<th>Poisson’s ratio ν</th>
<th>Density ρ</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail</td>
<td>210 GPa</td>
<td>0.25</td>
<td>7850</td>
</tr>
<tr>
<td>Rail pad</td>
<td>100 kN/mm</td>
<td>2*10⁴ Ns/m</td>
<td></td>
</tr>
<tr>
<td>Ballast</td>
<td>34 kN/mm</td>
<td>2.5*10⁶ Ns/m</td>
<td>67 kN/mm</td>
</tr>
</tbody>
</table>

Vertical stiffness

<table>
<thead>
<tr>
<th></th>
<th>Vertical stiffness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail pad</td>
<td>200 kN/mm</td>
</tr>
<tr>
<td>Ballast</td>
<td>67 kN/mm</td>
</tr>
</tbody>
</table>

Vertical damping

<table>
<thead>
<tr>
<th></th>
<th>Vertical damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rail pad</td>
<td>2*10⁴ Ns/m</td>
</tr>
<tr>
<td>Ballast</td>
<td>1*10⁶ Ns/m</td>
</tr>
</tbody>
</table>

Two distances between sleepers were used 60 cm and 75 cm.
b.i-2 Continuously supported track

The continuously supported track is based on the CDM embedded rail system. Two systems were modelled: the CDM Classic system and the CDM Comfort system. The rail has the same material properties as for the ballasted track. The system is directly fastened in a concrete slab. The values of stiffness’s listed below have been measured by Polimi during their procedure for fatigue tests on CDM samples.

<table>
<thead>
<tr>
<th></th>
<th>Lateral stiffness</th>
<th>Vertical stiffness</th>
</tr>
</thead>
<tbody>
<tr>
<td>CDM Classic</td>
<td>50 kN/mm/Imrail</td>
<td>105 kN/mm/Imrail</td>
</tr>
<tr>
<td>Lateral damping</td>
<td>2*10^4 Ns/m</td>
<td>2*10^4 Ns/m</td>
</tr>
<tr>
<td>Vertical damping</td>
<td>50 kN/mm/Imrail</td>
<td>2*10^4 Ns/m</td>
</tr>
<tr>
<td>CDM Comfort</td>
<td>45 kN/mm/Imrail</td>
<td>50 kN/mm/Imrail</td>
</tr>
<tr>
<td>Lateral damping</td>
<td>2*10^4 Ns/m</td>
<td>2*10^4 Ns/m</td>
</tr>
<tr>
<td>Vertical damping</td>
<td>50 kN/mm/Imrail</td>
<td>2*10^4 Ns/m</td>
</tr>
</tbody>
</table>

b.i-3 Direct fixation track

The track with a rail directly fastened to a concrete slab has also been modelled with two different rail pad stiffness’s.

<table>
<thead>
<tr>
<th></th>
<th>Lateral stiffness</th>
<th>Vertical stiffness</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hard rail pad</td>
<td>60 kN/mm</td>
<td>185 kN/mm</td>
</tr>
<tr>
<td>Lateral damping</td>
<td>2*10^3 Ns/m</td>
<td>2*10^3 Ns/m</td>
</tr>
<tr>
<td>Vertical stiffness</td>
<td>40 kN/mm</td>
<td>50 kN/mm</td>
</tr>
<tr>
<td>Vertical damping</td>
<td>2*10^3 Ns/m</td>
<td>4*10^3 Ns/m</td>
</tr>
<tr>
<td>Soft fixation</td>
<td>40 kN/mm</td>
<td>50 kN/mm</td>
</tr>
<tr>
<td>Lateral damping</td>
<td>4*10^3 Ns/m</td>
<td>4*10^3 Ns/m</td>
</tr>
<tr>
<td>Vertical stiffness</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Vertical damping</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

As for the ballasted track, two distances between fixations were used 60 cm and 75 cm.
(ii) Rail types

Seven types of rails were used in the simulations: five for the tramway vehicle and two for the light rail vehicle.

b.ii-1 Rail types for the light rail vehicle

b.ii-1.1 UIC50

The figure below shows the parameters used to model the UIC50 rail as well as a drawing of the rail section.

![Drawing of the UIC50 rail section](image)

**Figure 5.2.10 Drawing of the UIC50 rail section**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass [kg/m&lt;sub&gt;rail&lt;/sub&gt;]</td>
<td>50.46</td>
</tr>
<tr>
<td>IXX</td>
<td>1.532*10&lt;sup&gt;-6&lt;/sup&gt;</td>
</tr>
<tr>
<td>IYY</td>
<td>3.055*10&lt;sup&gt;-6&lt;/sup&gt;</td>
</tr>
<tr>
<td>IZZ</td>
<td>1.785*10&lt;sup&gt;-5&lt;/sup&gt;</td>
</tr>
</tbody>
</table>

**Table 5.2.1 Parameters of the UIC50 rail**
b.ii-1.2 UIC54

Figure 5.2.11 Drawing of the UIC54 rail section

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass [kg/m(\text{rail})]</td>
<td>54.77</td>
</tr>
<tr>
<td>IX</td>
<td>1.915(\times10^{-6})</td>
</tr>
<tr>
<td>IY</td>
<td>4.208(\times10^{-6})</td>
</tr>
<tr>
<td>IZ</td>
<td>2.334(\times10^{-5})</td>
</tr>
</tbody>
</table>

Table 5.2.2 Parameters of the UIC54 rail
b.ii-2 Rail types for the tramway vehicle

b.ii-2.1 EB50T

Figure 5.2.12 Drawing of the EB50T rail section

<table>
<thead>
<tr>
<th>Mass [kg/m&lt;sub&gt;rail&lt;/sub&gt;]</th>
<th>50.1</th>
</tr>
</thead>
<tbody>
<tr>
<td>IXX</td>
<td>1.358*10&lt;sup&gt;-6&lt;/sup&gt;</td>
</tr>
<tr>
<td>IYY</td>
<td>4.117*10&lt;sup&gt;-6&lt;/sup&gt;</td>
</tr>
<tr>
<td>IZZ</td>
<td>1.985*10&lt;sup&gt;-5&lt;/sup&gt;</td>
</tr>
</tbody>
</table>

Table 5.2.3 Parameters of the EB50T rail
b.ii-2.2  NP4am

Figure 5.2.13 Drawing of the NP4am rail section

<table>
<thead>
<tr>
<th>Mass [kg/m]</th>
<th>61.45</th>
</tr>
</thead>
<tbody>
<tr>
<td>IXX</td>
<td>1.352*10^{-6}</td>
</tr>
<tr>
<td>IYY</td>
<td>1.042*10^{-5}</td>
</tr>
<tr>
<td>IZZ</td>
<td>3.466*10^{-5}</td>
</tr>
</tbody>
</table>

Table 5.2.4  Parameters of the NP4am rail
**Figure 5.2.14 Drawing of the NP4am rail section**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass [kg/m_rail]</td>
<td>61.45</td>
</tr>
<tr>
<td>IXX</td>
<td>1.26*10^{-6}</td>
</tr>
<tr>
<td>IYY</td>
<td>6.915*10^{-6}</td>
</tr>
<tr>
<td>IZZ</td>
<td>2.0715*10^{-5}</td>
</tr>
</tbody>
</table>

*Table 5.2.5 Parameters of the 35G rail*
b.ii-2.4 Ri59N

Figure 5.2.15 Drawing of the 35G rail section

<table>
<thead>
<tr>
<th>Mass [kg/m]</th>
<th>58.14</th>
</tr>
</thead>
<tbody>
<tr>
<td>IXX</td>
<td>1.288*10^{-6}</td>
</tr>
<tr>
<td>IYY</td>
<td>8.819*10^{-6}</td>
</tr>
<tr>
<td>IZZ</td>
<td>3.281*10^{-5}</td>
</tr>
</tbody>
</table>

Table 5.2.6 Parameters of the Ri59N rail
b.ii-2.5  Ri53N

Figure 5.2.16 Drawing of the Ri53N rail section

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass [kg/m\text{rail}]</td>
<td>52.97</td>
</tr>
<tr>
<td>IXX</td>
<td>1.278*10^{-6}</td>
</tr>
<tr>
<td>IYY</td>
<td>7.449*10^{-6}</td>
</tr>
<tr>
<td>IZZ</td>
<td>1.344*10^{-5}</td>
</tr>
</tbody>
</table>

Table 5.2.7  Parameters of the Ri53N rail
(c) **PRESENTATION OF THE TRACK LAYOUT**

For each vehicle type (light rail and tramway), there are two tracks a straight track and a curved track. Each track has a given length, curvature, rail inclination, gauge and rail irregularities.

- **Track length**
  The length of the track for both the light rail vehicle and the tramway vehicle is 500.0 m whether it is straight or curved.

- **Track curvature radius**
  When the track is curved, two curvature radii have been chosen in function of the vehicle type:
  - for the light rail vehicle, the curvature radius is 100.0 m;
  - for the tramway vehicle, the curvature radius is 30.0 m.

- **Rail inclination**
  The inclination of the rails is the same for every track 0.025 rad.

- **Track Gauge**
  The track gauge is the same for every track 1.435 m.

- **Rail irregularities**
  Two types of rail irregularities have been modelled in Vi-Rail.
(i) **Measured irregularities**

A rail roughness that has been measured on a track in Kingston (Canada) has been used. The rail roughness is displayed on the figure below.

![Measured rail roughness spectrum](image)

**Figure 5.2.17 Measured rail roughness spectrum**

In Vi-Rail, the irregularities in displacement over distance on the rail can be directly inserted in the model.
(ii) **Step function irregularity**

Another type of irregularities has been modelled in Vi-Rail: a step in the rail. The dimensions of this step are as follows:

- height of the step: 3 mm.
- length of the step: 10 mm.

The figure below shows a representation of the modelled step.

![Figure 5.2.18 Representation of the step irregularity](image-url)
5.2.1.4. Results

Following the methodology, results are obtained at different stages of the chain. In the following sub-sections, examples of results are presented. However, for size purposes, the major part of the results is presented in the appendices.

(a) RESULTS OUT OF VI-RAIL (TIME DOMAIN)

After the calculation in Vi-Rail, track displacement and wheel/rail contact forces are plotted in function of time.

(i) Displacements

On the figure below the displacement above the ballast and of the rails are displayed.

![Example of displacements computed in Vi-Rail](image)

The displacements under the track shown here above are post-processed in Matlab to obtain the frequency spectrum of the velocity of vibration. Indeed, as rail irregularities are inserted in the model (in the case of the figure above, measured irregularities); vibrations are generated by the vehicle passage.
(ii) Forces

The figure below shows the forces applied on the rail coming from the wheel/rail contact in function of time.

![Figure 5.2.20 Example of forces computed in Vi-Rail](image)

Four forces are displayed the vertical and lateral forces on both the inner rail and the outer rail. On the picture above, the forces presented come from the light rail vehicle running on the ballasted track with EB50T rail on the curved track. The evolution of the forces over time clearly shows the vehicle entering and getting out of the curve. One can also see the effect of the rail irregularities (here measured irregularities).

For the vertical force, in the beginning, the force is the same on both rails and when the vehicle enters the curve, the forces on the outer (inner) rail increases (decreases) to a certain constant level in the curve. When the vehicle gets out of the curve, the forces go back to the same level as before the curve corresponding to the axle load. The fact the vertical forces [increase](index) on the outer rail and [decreases](index) on the inner rail can be explained by the fact that there is no cant deficiency imposed to the track. If there was, one would expect the vertical forces to [increase](index) on the inner rail and [decrease](index) on the outer rail.

The force is calculated and stored in a spreadsheet (see table 4.2.3). This value of the force does not take into account variations due to the rail irregularities because it is an average value over time. For the cases where the vehicle runs in a curve the average of the force is calculated on the section of the time signal where the vehicle is in the curve and not on the whole signal. The values of the forces are further discussed below (see section 5.2.1.4.2.3).
(b) RESULTS OUT OF MATLAB

The displacement signals out of Vi-Rail are post-processed in Matlab to obtain frequency spectra of the vibrations under the track.

(i) Vibration velocity spectra

The time signals of the displacements under the rail pads coming from Vi-Rail are transformed into frequency spectra through a Fast Fourier Transform (FFT) algorithm and derived to obtain the velocity. The spectra are then plotted in third-octave band as shown on the figure below. The resulting spectra allow the comparison of vibration velocity spectra for the various combinations of parameters studied in this report. The results are presented twice as two types of rail irregularities were used in the simulation as explained in the above section: measured rail irregularities and a step function irregularity.

(ii) Measured rail irregularities

In this section the vibration spectra obtained by simulation in Vi-Rail with measured rail irregularities are discussed for some cases that cover the trends seen in all the results. The complete results are presented in appendix A5.2.1 of the report.

![Graph](image)

**Figure 5.2.21** Vibration velocity spectra for the light rail vehicle in a straight line on ballast with UIC50 rail and measured rail irregularities

On the figure above, the influence of the vehicle speed is clearly visible. The faster the vehicle, the highest the vibrations are. However, this influence is much more visible at lower and higher frequencies: for the frequencies around 32 Hz, the speed almost has no influence on the vibration levels.
Figure 5.2.22  Vibration velocity spectra for the light rail vehicle in a curve on ballast with UIC50 rail and measured rail irregularities

The figure here above lead to another interesting remark: when the vehicle runs at 40 km/h and 60 km/h peaks appear in the spectra at lower frequencies (8 Hz at 40 km/h and 13 Hz at 60 km/h). The lateral movement of the bogies in the curve that occur when the vehicle runs faster can explain these peaks. These movement are alternative movements that cause vibrations at a certain frequency and this frequency increases when the vehicle runs faster because the bogie’s movement will occur more frequently. At 20 km/h, apparently, the vehicle does not run fast enough for these movement to occur but when it runs at 40 km/h, lateral movement of the bogie occur and they are even more frequent (increased frequency from 8 to 13 Hz) and larger (higher peak at 13 Hz in the curve and 60 km/h).
Figure 5.2.23  Vibration velocity spectra for the light rail vehicle in a straight line with a UIC50 rail at 20 km/h for various rail support types and measured rail irregularities

The figure here above allows comparing the performances of the different rail support types studied in this report with respect to vibration mitigation. The ballasted track gives clearly the lowest levels but it is important to remember that an attenuation filter was used to represent the ballast that had not been taken into account in the flexible track in Vi-Rail. The chosen filter thus influences the spectrum for the ballasted track. The direct fixation system with a hard railpad gives the highest vibration levels, especially at higher frequencies. The CDM-Classic and the direct fixation with a soft railpad give very similar results and the CDM-Comfort gives vibration levels that are slightly lower.
(iii) **Step function rail irregularity**

In this section, the vibration spectra obtained by simulation in Vi-Rail with a step function irregularity are discussed for some cases that cover the trends seen in all the results. The complete results are presented in appendix A5.2.2 of the report.

![Image](Straight line on CDM-Confort (UIC50).)

**Figure 5.2.24** Vibration velocity spectra for the light rail vehicle in a straight line on a CDM-Confort track with a UIC50 rail and a step function irregularity

Unlike in the case of measured irregularities, the influence of the speed in the vibration spectrum due to a step function irregularity is much more visible in all frequency bands. The vibration levels almost vary with a factor of $20 \times \log_{10}(V_2/V_1)$: between 20 and 40 km/h, the difference is approximately 6 dB while between 20 and 80 km/h the difference is approximately 15 dB (not far from the 12 dB foreseen by the above formula).
Figure 5.2.25  Vibration velocity spectra for the light rail vehicle in a straight line on different types of track with a UIC50 rail at 60 km/h and a step function irregularity

The figure above shows a comparison between the vibration spectra under the different types of track. The same conclusion as for the results with a measured irregularity can be drawn to the exception that the differences between the track systems are more visible.
(c) **RESULTS OUT OF NASTRAN**

The contact forces calculated in Vi-Rail are used as input forces in Nastran to calculate the stresses in the rail web and in the rail foot.

The figure below shows the model of the EB50T rail on the ballasted track. The wheel/rail contact force calculated in Vi-Rail is applied on the railhead.

![Figure 5.2.26 Ballasted track Finite Element Model](image)

The whole model is composed of 3D Hex elements. The rail pad (for ballast and discrete systems) or the rubber materials (for the CDM system) are characterized by their Young moduli, which have been numerically calibrated to obtain the same stiffness as those measured by Polimi and presented in section 2.2. The table below shows the stiffness's Young moduli used in the FE models.

<table>
<thead>
<tr>
<th>Type of track</th>
<th>Support vertical stiffness [kN/mm]</th>
<th>Support horizontal stiffness [kN/mm]</th>
<th>Young’s modulus [MPa]</th>
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<tr>
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<td>CDM Comfort</td>
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Table 5.2.8 Parameters of the track Finite Element Models

The Von Mises stresses are calculated in every node of the mesh. The maximum stress in the rail web and in the rail foot are saved in an excel file.
(i) **Stresses in the rail web**

Among the stresses in all the nodes of the rail, the maximum is taken among the nodes located between two heights with reference to the bottom of the rail. The table below indicates the limits between which the nodes are considered for calculation of the maximum Von Mises stress.

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<tr>
<th>Type of rail</th>
<th>Minimum node height [mm]</th>
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Table 5.2.9  Location of the nodes where the maximum stress is computed in the rail web

The figure below shows an example of the Von Mises stresses in the EB50T rail on the ballasted track under the load of the tramway vehicle in a curve at 20 km/h.
(ii) **Stresses in the rail foot**

Unlike for the stresses in the rail web, the maximum stress in the rail foot is calculated among the nodes located on the bottom of the rail.

The figure below shows an example of the Von Mises stresses in the EB50T rail on the ballasted track under the load of the tramway vehicle in a curve at 20 km/h.

![Figure 5.2.28](image)

(iii) **Conclusions on the stresses in the rail**

Finally, the two tables below summarize the forces applied on the rails and the stresses in the rail for each simulation case.
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<th>Vehicle</th>
<th>Rail profile</th>
<th>Curve radius</th>
<th>Fixation system</th>
<th>Vehicle velocity</th>
<th>Outer rail Force</th>
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<th>60 cm Max stress</th>
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<td>40547.80</td>
<td>28.15</td>
<td>27.85</td>
</tr>
<tr>
<td>Light_Rail</td>
<td>UIC54</td>
<td>Infinite</td>
<td>Direct_Soft</td>
<td>80</td>
<td>40547.85</td>
<td>40547.79</td>
<td>28.15</td>
<td>27.85</td>
</tr>
<tr>
<td>Light_Rail</td>
<td>UIC54</td>
<td>Infinite</td>
<td>Direct_Soft</td>
<td>100</td>
<td>40547.93</td>
<td>40547.87</td>
<td>28.15</td>
<td>27.85</td>
</tr>
</tbody>
</table>

Table 5.2.10  Results for all simulation cases (forces on the rail and stresses in the rail)
Looking at the results presented in the tables above, sever conclusions can be drawn.

First, the wheel load applied on the rail is higher for the tramway than for the light rail. This is logical because the tramway is heavier. Another important point is that the vertical load increases on the outer rail in curve and decreases on the inner rail. This is also straightforward as there is no canting on the track.

In straight line, the load almost does not vary with the speed, the support type or the rail type. It is important to remember here that the load is the mean load so the influence of the irregularities on the rail is not taken into account. Therefore, the load does not vary with the speed.

The stresses presented in the table are computed in Nastran as explained in the section presenting the methodology. The stresses are classified in function of the distance between the fixations 60 cm, 75 cm and continuously supported rail. Of course, for the ballasted track and the direct fixation track do not have results for the continuously supported rail, as these are tracks where the rail is discretely supported. On the contrary, the CDM_Classic and CDM_Comfort tracks only have results for a continuously supported rail.

In curve, the stresses are a little higher in the rail on the ballasted track than in the direct fixation tracks. The stresses in the rail on the hard direct fixation are a little lower than on the soft direct fixation. Another interesting fact is that the stresses are higher when the distance between fixations is higher. This is logical since the bending deformation of the rail will be higher when the distance increases. The stresses in the rail on the continuous support (CDM_Classic and CDM_Comfort) are much lower than in the rail on discrete fixations.

The stresses are also lower in the rail foot than in the rail web. When it is a discrete fixations track (ballast, direct_soft, direct_hard), the maximum stress in the web is located in the section halfway between the fixations and the maximum stress in the rail foot is located on the edge of the fixation where the shear stress is the highest. When it is a continuously supported rail (CDM_Classic, CDM_Comfort), the maximum stress in the rail web and the maximum stress in the rail foot are both located in the section where the load is applied.

The stresses in the rail vary in function of the type of rail. The rails can be classified in function of their maximum stress (from highest to lowest):

- R59N -> NP4am -> 35G -> R53N -> EB50T for the tramway;
- UIC50 -> UIC54 for the light rail.
5.2.1.5. Conclusion

The sensitivity analysis carried and presented in this report leads to several interesting conclusions: conclusions on the stresses in the rail and conclusions on the vibrations levels under the track.

The wheel/rail contact forces calculated in Vi-Rail for different combination of vehicle, rail type, support, type and velocity were applied to track Finite Element track models in Nastran. The resulting stresses in the rail are discussed in this report.

From the results, it can be seen that, the more the railpad is stiff, the highest the stress. The stress is much lower when the rail is continuously supported. The stress also increases with the distance between the rail fixation. The stresses in the rail web in curve are much higher than on straight track. This cannot be said for the stresses in the rail foot. In curve, the stresses increases with the vehicle speed while it is not the case in straight line.

The displacements of the rail and of the base under the railpads are computed in Vi-Rail. The rail irregularities included in the track model (measured rail irregularities and step function rail irregularity) causes these displacements to vary around an average value (the static deformation): vibrations occur. These displacement in function of time are transformed into displacement in function of the frequency with a Fast Fourier Transform algorithm. The derivation of these displacement spectra lead to vibration velocity spectra. Several conclusions can be drawn by looking at these spectra.

The vibration levels increase with the speed. For measured rail irregularities, this increase is mainly visible at lower and higher frequencies. For a step function irregularity, the increase is of a factor $20 \log_{10}(V_2/V_1)$ in all frequency bands.

The vibrations under the ballasted track are lower than under the CDM-Classic, CDM-Comfort and Direct-Soft where they are similar. The Direct-Hard system gives the highest vibration levels.

An interesting behaviour has been discovered in the spectra: the lateral movement of the bogies in the curves also create peaks at low frequencies in the vibration spectra. These peaks appear above a certain vehicle velocity and increase in amplitude and frequency as velocity increases.

Finally, this study allows many conclusions to be drawn from its results. These conclusions have been discussed here. All these conclusions help establish functional requirements for urban railway tracks.
5.2.2 Sensitivity analysis on metro track parameters by means of numerical simulation of train-track dynamic interaction

5.2.2.1. Overview

In order to investigate functional requirements for metro track systems, PoliMi has provided numerical tools and simulations for performing parametric studies on track parameters. An extensive sensitivity analysis has been performed, by means of numerical simulation of train-track dynamic interaction.

As agreed among SP5 partners, the Metro Madrid case study was chosen for PoliMi’s numerical simulations. Since the focus is the metro track, vehicle characteristics were fixed, and variable track parameters were considered.

The final objective of the sensitivity analysis was to evaluate the effects on track stability and vibration mitigation of the following parameters:

- different track design options (embedded rail, direct fixation, floating slab track) and rail support stiffness values;
- support stiffness variations corresponding to new/degraded track conditions;
- axle load variations (tare and full load);
- track irregularity corresponding to different levels of degraded track geometry;
- rail profile and gauge.

5.2.2.2. Numerical simulation of train-track interaction

The numerical simulation of the dynamic interaction between train and track requires combined modelling of the two subsystems involved. To this end, a numerical procedure has been developed at Politecnico di Milano - Department of Mechanical Engineering, which is based on a finite element schematisation of the track/structure and on a multi-body schematisation of the train. The two subsystems are described through two separate sets of differential equations which are simultaneously integrated in the time domain. The co-simulation procedure is based on the coupling of the train and the track dynamics, as the result of the contact forces exchanged at wheel-rail interface: as it is shown in Figure 2.2.1, these contact forces are a function of both the vehicle and the infrastructure state variables. Both vertical and lateral dynamics of the entire system are taken into account.

Figure 5.2.29
The numerical model for the simulation of train-structure interaction: vehicle and structure as separate but interacting subsystems.

The following equations hold for the structure:

\[
\begin{bmatrix}
M_s \\
C_s \\
K_s
\end{bmatrix}
\ddot{x}_s + \begin{bmatrix}
M_s \\
C_s \\
K_s
\end{bmatrix}
\dot{x}_s + \begin{bmatrix}
M_s \\
C_s \\
K_s
\end{bmatrix}
x_s = \mathbf{F}_{sv}(\mathbf{x}_v, \dot{\mathbf{x}}_v, \ddot{\mathbf{x}}_v, \mathbf{x}_w, \dot{\mathbf{x}}_w, \ddot{\mathbf{x}}_w, t) + \mathbf{F}_{ss}(\mathbf{x}_v, \dot{\mathbf{x}}_v, t)
\]

where \([M_s],[C_s],[K_s]\) are the mass, damping and stiffness matrices of the structure, \(\mathbf{F}_{sv}\) is the column matrix of the generalised wheel-rail contact forces (which are a function of both the train and the structure state coordinates), \(\mathbf{F}_{ss}\) represents the generalised terms due to the non linear internal forces due to the particular formulation of some track components.

In the train multi-body model, each railcar or locomotive is composed of rigid bodies, connected by means of linear/nonlinear elastic and damping elements that reproduce the primary and secondary suspensions. The vehicle equations of motion are written with respect to an auxiliary moving frame of reference, travelling along the ideal track centreline and following the carbody centre of gravity. The vehicle relative motion with respect to this moving reference is considered.

For the generic \(i\)-th vehicle in the train assembly (\(n\) being the total number of cars), the following equations hold:

\[
\begin{bmatrix}
M_{vi} \\
C_{vi} \\
K_{vi}
\end{bmatrix}
\ddot{x}_{vi} + \begin{bmatrix}
M_{vi} \\
C_{vi} \\
K_{vi}
\end{bmatrix}
\dot{x}_{vi} + \begin{bmatrix}
M_{vi} \\
C_{vi} \\
K_{vi}
\end{bmatrix}
x_{vi} = \mathbf{F}_{vi}(\mathbf{x}_v, \dot{\mathbf{x}}_v, \ddot{\mathbf{x}}_v, \mathbf{x}_w, \dot{\mathbf{x}}_w, \ddot{\mathbf{x}}_w, t) + \mathbf{F}_{si}(\mathbf{x}_v, \dot{\mathbf{x}}_v, \ddot{\mathbf{x}}_v, t)
\]

where \([M_{vi}],[C_{vi}],[K_{vi}]\) are the mass, damping and stiffness matrices of the \(i\)-th vehicle, \(\mathbf{F}_{vi}\) is the column matrix containing the generalised wheel-rail contact forces and \(\mathbf{F}_{si}\) accounts for nonlinear suspension components of the rail vehicle.

The dependence on time \(t\) of the generalised contact forces is due to track and wheel irregularity which, for a given train speed, are assigned functions of time.

A key element in the simulation code is the non-linear wheel-rail contact model, which allows to accurately reproduce the normal and tangential contact forces exchanged at wheel rail interface.

The normal forces are evaluated through a multi-Hertzian model, which allows either to take into account the presence of multiple contact points, e.g. when contacts occur both on the tread and on the flange, or to approximate complex non-hertzian contact patches by one or more elliptic patches.

The lateral and longitudinal contact forces are then determined through the heuristic Shen–Hedrick–Elkins contact model, starting from the longitudinal and lateral creepages on each contact area, which are derived from the wheelset kinematics.
(a) **VEHICLE MODEL**

As agreed among SP5 partners, the Metro Madrid case study was chosen for PoliMi’s numerical simulations. In order to investigate the functional requirements for metro track infrastructure, vehicle characteristics were fixed, and a sensitivity analysis on track parameters was performed.

Figure 5.2.30 shows the vehicle considered in the simulations. The main vehicle characteristics are reported in table 5.2.11.

![The Metro Madrid train](image)

**Table 5.2.11 Main characteristics of the Metro Madrid vehicle**

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>maximum axle load</td>
<td>154 kN</td>
</tr>
<tr>
<td>tare axle load</td>
<td>96 kN</td>
</tr>
<tr>
<td>wheelbase</td>
<td>2.2 m</td>
</tr>
<tr>
<td>pivot pitch</td>
<td>11.1 m</td>
</tr>
</tbody>
</table>

The multibody model of one Metro Madrid coach was implemented into PoliMi’s simulator (Fig. 5.2.32):

- carbody and bogie frames are assumed to be rigid;
- wheelsets are modelled as flexible bodies, according to modal superposition approach;
- elastic and damping connection elements reproduce the primary and secondary suspensions.
Figure 5.2.32 Vehicle multi-body model

(b) TRACK MODEL

For the purpose of this research, a finite element model of the embedded-rail track system was developed (figure 5.2.33) and implemented into PoliMi’s train-track dynamic simulation code.

A linear visco-elastic bed reacting in vertical, lateral and torsional direction reproduces the rubber jacket enclosing the rail. Its visco-elastic characteristics were identified during the lab tests performed by PoliMi in SP1, on small-scale REMS samples.

Both the rails and the tunnel are modelled as equivalent Euler-Bernoulli beams: while the bending stiffness and the mass per unit length of the rail can be easily determined from the rail producer catalogue, the properties of the equivalent beam representing the tunnel and those of the visco-elastic bed representing the ground were identified from impulsive tests carried out by PoliMi in Milano underground some years ago. The visco-elastic bed representing the ground was introduced to allow investigating the transmissibility (i.e. the capability of filtering out vibrations induced by train passage) of the considered REMS tracks with respect to more traditional track systems.

Figure 5.2.33 Cross-section of the implemented finite element model of the embedded-rail track system
As anticipated before, the visco-elastic characteristics of the elastic bed enclosing the rail were identified during the lab tests performed by PoliMi in SP1, on small-scale REMS samples.

The test bench (figure 5.2.34) consists of a restraining frame, which carries the hydraulic actuator used to apply the test loads to the samples. All the tests were performed in force control.

The force applied on top of the rail is measured by a load cell, while the motion of the rail with respect to the concrete block (both being assumed to be rigid bodies) is measured by means of laser transducers.

Static and dynamic characterization tests were performed on all the three REMS samples (COMPACT, CLASSIC and COMFORT), to assess their stiffness and damping characteristics as a function of frequency, load amplitude and preload. The samples were tested by applying both a purely vertical force and a 26deg inclined load (see figure 5.2.35).

Table 5.2.12 summarizes the measured stiffness parameters, for the three REMS samples. These data were used in the train-track interaction simulations described in the next paragraphs.

Figure 5.2.34 Test bench for REMS samples characterization tests
Figure 5.2.35 Test configurations: vertical and inclined load

<table>
<thead>
<tr>
<th></th>
<th>vertical stiffness static (0 deg)</th>
<th>dynamic (0.9Q±0.1Q@10 Hz)</th>
<th>lateral stiffness static (26 deg)</th>
<th>dynamic (0.9Q±0.1Q@10 Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>compact</td>
<td>78</td>
<td>1175</td>
<td>28</td>
<td>55</td>
</tr>
<tr>
<td>classic</td>
<td>44</td>
<td>105</td>
<td>22</td>
<td>50</td>
</tr>
<tr>
<td>comfort</td>
<td>14</td>
<td>50</td>
<td>20</td>
<td>45</td>
</tr>
</tbody>
</table>

Table 5.2.12 REMS stiffness parameters measured during PoliMi’s lab tests
5.2.2.3. Simulations in tangent track

Tangent track numerical simulations of train-track dynamic interaction were first carried out to assess the performance of metro track systems in terms of vibration mitigation.

A sensitivity analysis on the principal parameters involved was performed, considering:

- different track design options (embedded rail, direct fixation, floating slab track) and rail support stiffness values;
- support stiffness variations corresponding to new/degraded track conditions;
- axle load variations (tare and full load);
- track irregularity corresponding to two different levels of degraded track geometry.

The sensitivity analysis concentrated on the embedded rail track system and the starting point were the support stiffness values of the three classes of REMS track (table 5.2.2). The performance of the embedded rail system was compared to that of direct fixation and floating slab solutions.

Both wheel and rail irregularity input data were considered in all the simulations. While measured irregularity patterns were considered for the wheel, track irregularity input was generated according to the PSD functions defined in the ORE B176 Standard.

(a) SIMULATIONS PERFORMED

A complete list of all the performed tangent track simulations is reported in table 5.2.13.

<table>
<thead>
<tr>
<th></th>
<th>tare/full load</th>
<th>track irregularity</th>
<th>degraded</th>
<th>#</th>
</tr>
</thead>
<tbody>
<tr>
<td>ER comfort</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>8</td>
</tr>
<tr>
<td>ER classic</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>8</td>
</tr>
<tr>
<td>ER compact</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>8</td>
</tr>
<tr>
<td>DF</td>
<td>both</td>
<td>2</td>
<td>N</td>
<td>4</td>
</tr>
<tr>
<td>FS</td>
<td>both</td>
<td>2</td>
<td>N</td>
<td>4</td>
</tr>
<tr>
<td>total number of simulations</td>
<td></td>
<td></td>
<td></td>
<td>32</td>
</tr>
</tbody>
</table>

Table 5.2.13 List of the simulations for tangent track running (V = 110 km/h)

As an example, figure 2.3.1 shows the time histories of the calculated vertical rail and tunnel accelerations during tangent track running at 110 km/h for the three REMS stiffness classes. These simulations refer to nominal (new) track parameters and to vehicle in full-load condition.
Figure 5.2.36  Numerical simulations for the estimation of vibration mitigation performance (REMS). Metro Madrid train running in tangent track at 110 km/h (full load). Simulations output: rail and tunnel acceleration time histories.

It can be clearly seen that, when increasing the track stiffness, the maximum rail vibration reduces, while higher vibrations are transmitted to the tunnel. A significant change in terms of duration can also be observed.
(b) **TRACK DYNAMIC PERFORMANCE ESTIMATION: VIBRATION MITIGATION**

The time-domain numerical simulations of train-track interaction in tangent track (see table 5.2.3) were post-processed, so as to obtain summarizing information concerning the track dynamic performance in the different conditions. As an example, figures 5.2.37 and 5.2.38 show the rail and tunnel one-third octave acceleration spectra (reference acceleration $1.0\times 10^{-6}$ m/s$^2$) and the corresponding vibration attenuation, for the three stiffness classes of the embedded rail track and the direct fixation track. In all the simulations, the Metro Madrid train is running in tangent track at 110 km/h, in full load condition.

As expected, among the three stiffness classes of the embedded rail track, the best performance in terms of vibration mitigation is that of the COMFORT solution, which proves to be better than the direct fixation track, at any frequency above 40 Hz.

![Figure 5.2.37](image1)

**Figure 5.2.37** Metro Madrid train running in tangent track at 110 km/h (full load): rail and tunnel one-third octave acceleration spectra (3 stiffness classes of embedded rail vs. direct fixation)

![Figure 5.2.38](image2)

**Figure 5.2.38** Metro Madrid train running in tangent track at 110 km/h (full load): vibration attenuation from rail to tunnel (3 stiffness classes of embedded rail vs. direct fixation)
(c) **EFFECT OF TRACK DEGRADATION**

Figures 2.3.4, 2.3.5 and 2.3.6 show how the performance of the three considered classes of the embedded-rail track varies, as a result of track degradation. The effect of track degradation due to repeated load cycles was considered by increasing the stiffness of each one of the three tracks by 20%.

Figures 2.3.4, 2.3.5 and 2.3.6 show the rail and tunnel one-third octave acceleration spectra (1.0e-6 m/s² reference acceleration), and the corresponding vibration attenuation, in case of new and degraded track, for the 3 stiffness classes of the embedded rail track. In all the simulations, the Metro Madrid train is running in tangent track at 110 km/h, in full load condition. For all the three stiffness classes, no significant variation in vibration mitigation performance can be observed.

![Graph showing rail and tunnel acceleration spectra](image)

**Figure 5.2.39** Metro Madrid train running in tangent track at 110 km/h (full load): rail and tunnel one-third octave acceleration spectra and corresponding vibration attenuation, in case of new (nominal stiffness) and degraded (+20% stiffness) REMS COMPACT track.
Figure 5.2.40 Metro Madrid train running in tangent track at 110 km/h (full load): rail and tunnel one-third octave acceleration spectra and corresponding vibration attenuation, in case of new (nominal stiffness) and degraded (+20% stiffness) REMS CLASSIC track.
Figure 5.2.41 Metro Madrid train running in tangent track at 110 km/h (full load): rail and tunnel one-third octave acceleration spectra and corresponding vibration attenuation, in case of new (nominal stiffness) and degraded (+20% stiffness) REMS COMFORT track.
5.2.2.4. Simulations in curve

As for tangent track running, curved track numerical simulations were carried out to assess the performances of metro track systems in terms of track stability. The following parameters were taken into account for the sensitivity analysis:

- different track design options (embedded rail, direct fixation, floating slab track) and rail support stiffness values;
- support stiffness variations corresponding to new/degraded track conditions;
- axle load variations (tare and full load);

The sensitivity analysis concentrated on the embedded rail track system and the starting point were the support stiffness values of the three classes of REMS track (table 5.2.2). The performance of the embedded rail system was compared to that of direct fixation and floating slab solutions. Both wheel and rail irregularity input data were considered in all the simulations.

(a) SIMULATIONS PERFORMED

Track stability was analysed in terms of rail vertical deflection and gauge widening. A reference curve of 300 m radius and 150 mm of cant was taken into account and the vehicle speed was imposed to be equal to 80 km/h. Simulations were performed for both tare and fully loaded vehicle and with new and worn wheel/rail profiles. For the comfort embedded track, the influence of degradation of track characteristics was also investigated. Degraded characteristics are taken from experimental tests carried out at Politecnico di Milano. Table 5.2.14 sums up the simulations carried out.

<table>
<thead>
<tr>
<th>Sl</th>
<th>tare/full load</th>
<th>W/R profiles</th>
<th>degraded</th>
<th>#</th>
</tr>
</thead>
<tbody>
<tr>
<td>ER comfort</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>8</td>
</tr>
<tr>
<td>ER classic</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>4</td>
</tr>
<tr>
<td>ER compact</td>
<td>both</td>
<td>2</td>
<td>Y</td>
<td>4</td>
</tr>
<tr>
<td>DF</td>
<td>both</td>
<td>2</td>
<td>N</td>
<td>4</td>
</tr>
<tr>
<td>FS</td>
<td>both</td>
<td>2</td>
<td>N</td>
<td>4</td>
</tr>
</tbody>
</table>

| total number of simulations | 24 |

Table 5.2.14 List of the simulations for curved track running (R = 300 m, h = 150 mm, V = 80 km/h)

(b) TRACK DYNAMIC PERFORMANCE ESTIMATION: TRACK STABILITY

As an example, figure 2.4.1 shows the achieved results in terms of time histories of vertical rail deflection (of both inner and outer rail) and gauge widening at a measuring point 8 m away from the end of the transition curve of the considered reference curve for the various track types considered. Both tare and fully loaded vehicle conditions are shown.
Figure 5.2.42 Metro Madrid train running on a curved track (300 m radius, 150 mm cant, 80 km/h speed):
Vertical rail deflection (upper graphs) and gauge widening (lower graph) of the various tracks considered for both tare and full load.
The passages of the four wheelsets (two bogies) are clearly visible in the vertical rail deflection graph. The difference between the passage of the first bogie and the second one is due to the roll motion of the carbody that is still present at that position. It can be clearly seen that, increasing the vehicle load (form an axle load of 96kN in tare conditions to and axle load of 154kN in fully loaded conditions), the vertical rail deflection increases. Moreover, for what concerns embedded tracks, vertical rail deflection increases from the stiffest one (compact track) to the most compliant one (comfort track) although no big differences between compact and classic track can be observed. Direct fixation track, instead has a higher vertical stiffness than embedded track thus leading to a vertical rail deflection that is almost half of that of compact track.

The time history of gauge widening shows the passage of the four wheelsets (more evident for the fully loaded vehicle) with superimposed oscillations, especially for the direct fixation track that has discrete supports and lower damping, generated by the train passage that propagate along the rails. The highest peak is reached for the first wheelset that has the highest angle of attack and therefore generates the biggest lateral force. The gauge widening dies out less rapidly than the vertical rail deflection thus leading to a residual gauge widening between the passage of the two bogies. For what concerns embedded tracks, the increased deformability form compact to comfort track determines and increased gauge widening (from 0.45 mm for the compact track in tare load conditions to 0.83 mm for the comfort track in tare load conditions). Moreover, it should be observed that the direct fixation track, due to its smaller damping characteristics, "feels" the train passage earlier than embedded tracks and that it experiences a higher peak gauge widening due to the fact that discrete supports are less capable to redistribute lateral forces.

The influence of track degradation and wear of wheel/rail profiles is, as expected negligible on vertical rail deflection and gauge widening. Table 5.2.15 sums up the results. For what concerns the vertical rail displacement, reference to the outer rail has been made being this rail the most stressed one due to the load transfer. For what concerns the gauge widening, instead, reference to the passage of the first wheelset has been made being this the peak value as explained above.

<table>
<thead>
<tr>
<th>Track Type</th>
<th>Tare load</th>
<th>Full load</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Vertical rail deflection [mm]</td>
<td>Gauge widening [mm]</td>
</tr>
<tr>
<td>COMPACT</td>
<td>0.577</td>
<td>0.448</td>
</tr>
<tr>
<td>CLASSIC</td>
<td>0.701</td>
<td>0.603</td>
</tr>
<tr>
<td>COMFORT</td>
<td>1.773</td>
<td>0.625</td>
</tr>
<tr>
<td>Direct Fixation</td>
<td>0.324</td>
<td>0.832</td>
</tr>
</tbody>
</table>

Table 5.2.15 Maximum values for the different track considered at different loads  
(R = 300 m, h = 150 mm, V = 80 km/h)

It can be clearly seen that the direct fixation track behaves better than embedded rail systems for what concerns the vertical rail deflection while it behaves worse for what concerns the gauge widening. This is due to the fact that, for embedded rail systems, the rail is laterally supported by continuous rubber elements. Moreover, embedded track suffer less than direct fixation track from corrugation phenomena due to the presence of a continuous support.
5.2.2.5. **Numerical-experimental comparison**

A joint test campaign on both direct fixation and comfort embedded track was performed on Metro Madrid line. Results were used to validate the numerical simulations with particular reference to vertical rail displacements and gauge widening. The experimental set-up was composed by a series of displacement transducers to detect vertical and lateral movements of particular rail sections: for the direct fixation track, a straight track was instrumented while for the embedded track system a curve having 300 m radius was instrumented. As an example figure 5.2.43 shows a picture of the instrumented direct fixation track section while figure 5.2.44 shows a picture of the instrumented embedded track section.

![Instrumented test section of the direct fixation track](image1)

![Instrumented test section of the embedded track](image2)

Tests were performed using a standard Metro Madrid coach passing on the instrumented sections at different speeds in both directions. As an example, figures 5.2.45–5.2.46 show acquired channels for a vehicle passage at about 80 km/h: figures 5.2.45 and 5.2.47 show the time-histories of the vertical rail
deflection (the blue line is referred to the outer rail and the magenta line to the inner rail) while figures 5.2.46 and 5.2.48 show the time histories of the rail lateral deflections (the blue line is referred to the outer rail and the red line to the inner rail) and gauge widening (green line) for two different passages (to and forth). It can be observed that the maximum measured vertical rail deflection is about 2 mm for both passages while the maximum measured gauge widening is about 0.6 mm when running in one direction and 1.3 mm when running the opposite direction. This difference is due to the fact that test section was close to the transition curve. Thus, in one direction the value is associated to full curve running while in the other direction it is influenced by transient loads due to the transition from curve entrance to full curve.

Both the measured maximum vertical rail deflection and the gauge widening values for full curve running are very close to the corresponding values simulated through the numerical train-track interaction model (paragraph 4) that correctly accounts for the deformability of the rubber elements and precisely reproduces wheel – rail contact conditions.

Figure 5.2.45 Metro Madrid train running on a curved track (300 m radius, 150 mm cant, 80 km/h speed, full curve): time-history of the measured vertical deflections of both inner (magenta line) and outer (blue line) rails
Figure 5.2.46  Metro Madrid train running on a curved track (300 m radius, 150 mm cant, 80 km/h speed, full curve): time-history of the lateral deflections of both inner (blue line) and outer (red line) rails and gauge widening (green line)

Figure 5.2.47  Metro Madrid train running on a curved track (300 m radius, 150 mm cant, 80 km/h speed, end of transition curve): time-history of the measured vertical deflections of both inner (magenta line) and outer (blue line) rails
Figure 5.2.48  Metro Madrid train running on a curved track (300 m radius, 150 mm cant, 80 km/h speed, end of transition curve): time-history of the lateral deflections of both inner (blue line) and outer (red line) rails and gauge widening (green line)
5.2.2.6. Concluding remarks

A numerical train-track interaction model was set-up to carry out a sensitivity analysis on metro track design parameters to evaluate their effects on track stability and vibration transmission. The following parameters were considered:

- different track design options (embedded rail, direct fixation, floating slab track) and rail support stiffness values;
- support stiffness variations corresponding to new/degraded track conditions;
- axle load variations (tare and full load);
- track irregularity corresponding to two different levels of degraded track geometry.

For straight track running, it is shown that increasing the track stiffness the maximum rail vibration reduces, while higher vibrations are transmitted to the tunnel. A significant change in terms of duration of the vibration can also be observed. Among the three stiffness classes of the embedded rail track system, the best performance in terms of vibration mitigation is that of the comfort solution that proves to be better than the direct fixation track, at any frequency above 40 Hz. For what concerns track degradation, no significant variation in vibration mitigation performance can be observed for all the three stiffness classes considered.

For curved track running, it is shown that the direct fixation track behaves better than embedded rail systems for what concerns the vertical rail deflection while it behaves worse for what concerns the gauge widening.

The numerical-experimental comparison shows that both the measured maximum vertical rail deflection and the gauge widening values for full curve running are reproduced by the numerical train-track interaction model that correctly accounts for the deformability of the rubber elements and precisely reproduces wheel-rail contact conditions.
5.2.3 Light rail parameters based on national guidelines and recommendations

5.2.3.1. Introduction:

The BOStrab (=German federal Regulations on the Construction and Operation of Light rail Transit Systems) is a national guideline for light rail. It is divided in the following eight parts:

1 General
2 Operating Panel
3 Operating Staff
4: Operational Installations
5: Vehicles
6 Operations
7 Procedural Formalities
8 Non-compliances, conclusion and transitional arrangements

Because of the long experience (the first issue was published 1938) in light rail behind the BOStrab especially the fourth part (operational installations) and the BOStrab based recommendations often were adapted as guidelines e.g. for Poland, Finland, Austria, Norway and partially Spain.

The “Rules on the Alignment of Rail Systems” for light rail contain essential parameters for Light Rail Installations in particular. In consideration of these rules, maintenance costs could be reduced.

Below, an example for a possible guideline summarising the basic data of the recommendations.

5.2.3.2. Rules on the Alignment of Rail Systems

(a) **GENERAL OBJECTIVES**

The objective of the alignment is the lining-up and overlapping of the alignment elements in a way favourable to the dynamics of vehicle movements as well as optimisation of the alignment parameters in respect of

- Safety,
- Speed,
- Dynamics of vehicle movements and ride comfort,
- Economic efficiency of the structure and operation including the maintenance.

These objectives are reached by applying the standard values.

(b) **STANDARD AND LIMIT VALUES**

The alignment parameters are to be determined on the basis of the project speed \( v_p \) and the standard values of these Rules. If deviations are made from the standard values for compelling reasons, the margin between the standard values and the limit values of these Rules can be applied provided that the project speed is observed. The limit values (as minimum or maximum values, respectively) of the alignment parameters shall not be exceeded.
(c) **SPEEDS**
A uniform project speed $v_p$ is to be determined for the alignment of a route network or for parts of the route network. This speed is to be adjusted to the maximum speed of the present and future vehicles. The project speed shall not be less than $v_p = 50$ km/h for in-street and segregated track formations and not less than $v_p = 70$ km/h for independent track formations. Lower speeds can be assumed for those parts of the network on which there is no regular passenger transport. If it is necessary to deviate from the standard values of the alignment at constraints, the permissible speed per $v$ determined by way of the limit values of the alignment parameters shall correspond at least to the project speed $v_p$. If the limit values of the alignment parameters cannot be observed with the project speed $v_p$ in special circumstances, a lower permissible speed per $v$ is to be determined in deviation to the project speed.

(d) **STRAIGHT**
The track shall be as straight as at all possible. A straight between two curves shall have the length

$$l_v \geq \frac{v_p}{10}[m]$$  \hspace{1cm} (eq 1)

It shall not be shorter than the distance between the centres of the running gears of the vehicles mainly operated and not smaller than 6 m. If this length cannot be observed, the intermediate straight shall not be considered by the calculation of the permissible speed per $v$.

(e) **CURVE AND SUPERELEVATION**

(i) **Transverse Acceleration**
For the transverse acceleration in the curve

$$a_q = \frac{v^2}{3.62 \times r} - \frac{u}{153} \left[m \right]$$  \hspace{1cm} (eq 2)

applies.

As a rule, the alignment is based on a slight, positive transverse acceleration ($a_q \sim 0.2 \text{ m/s}^2$)). The transverse acceleration shall not be higher than $a_q = 0.65 \text{ m/s}$. In special circumstances the limit value $a_q = 0.98 \text{ m/s}^2$ is permitted; this value is also decisive for the proof of the permissible speed

$$perv = \sqrt{\left(\frac{r \times (u + 150)}{11.8}\right) \left[\text{km} \right]}$$  \hspace{1cm} (eq 3)

(ii) **Curve Radius**
For the curve radius

$$r = \frac{11.8 \times v^2}{u + 153 \times a_q}[m]$$  \hspace{1cm} (eq 4)
follows from (eq. 2).

Curve radii are to be as big as possible as a function of the project speed \( v_p \).

On independent track formations curve radii of route tracks are to be so big that the maximum line speed \( \max v \) is not restricted in case of a transverse acceleration of \( a_q = 0.65 \text{ m/s}^2 \). However, they shall not be smaller than \( r = 240 \text{ m} \). On segregated and in-street track formations, the relevant local conditions are to be considered in view of these Rules. The minimum radius \( \min r \) amounts to 25 m. If track sections are connected via an intermediate curve, this curve shall have the length of an intermediate straight in accordance with Section 5. If possible, the platforms shall be located at a straight track. If platforms have to be located in a curve, it is usually not possible to observe the requirements for handicapped-friendly boarding and alighting.

(iii) Superelevation

For the partial compensation of the transverse acceleration the outside running rail is laid higher in the curve than the inside running rail by the dimension of the superelevation \( u \).

\[
\frac{11.8* v^2}{r} - 153* a_q [m]
\]  
(eq 5)

The speed decisive for the superelevation to be realised is to be determined on the basis of the speed/distance chart in accordance with Section 3.5.

The compensating superelevation \( u_0 \) amounts to

\[
\frac{11.8* v^2}{r} [mm]
\]  
(eq 6)

However, as a rule the superelevation is to be so selected that there is a slight, positive transverse acceleration \( a_q \sim 0.2 \text{ m/s}^2 \) (normal super-elevation \( \text{nor u} \)).

\[
\frac{11.8* v^2}{r} - 30[mm]
\]  
(eq 7)

During a transverse acceleration of \( a_q = 0.65 \text{ m/s}^2 \) the superelevation shall not be smaller than

\[
\frac{11.8* v^2}{r} - 100[mm]
\]  
(eq 8)

In special circumstances the minimum superelevation during a transverse acceleration of \( a_q = 0.98 \text{ m/s}^2 \) amounts to

\[
\frac{11.8* v^2}{r} - 150[mm]
\]  
(eq 9)

The superelevation shall not be bigger than \( u = 150 \text{ mm} \); in special circumstances a value up to the maximum superelevation \( \max u = 165 \text{ mm} \) is permitted.

Platform tracks shall be designed without superelevation.
(f) **TRANSITION CURVE AND SUPERELEVATION RAMP**

(i) **Transition Curve**

As a rule, there has to be a transition curve between a straight and a curve as well as between curves of different curvatures to minimise the transverse jerk.

For the transverse jerk $C$ in the transition curve

$$C = \frac{v \Delta a_q}{3.6 l_u} \left[ \frac{m}{s^3} \right]$$

(eq 10)

applies.

$\Delta a_q$ is the difference in transverse acceleration between the beginning and the end of the transition curve.

The transverse jerk shall not be bigger than max $C = 0.67 \text{ m/s}^3$.

The minimum length of the transition curve amounts to

$$\min l_u = \frac{v \Delta a_q}{2.4} \left[ m \right]$$

(eq 11)

due to the transverse jerk. The transition curve and the superelevation ramp shall have the same length as well as the same beginning and the same end (see Section 7.2).

As the shape of the transition curve the Clothoid

$$A^2 = r^* l_u \left[ m^2 \right]$$

is to be preferred.

If it is not possible to realise a transition curve, the permissible speed amounts to

$$perv = 3.6 \sqrt{\max C \cdot l_v \cdot \frac{1000}{\Delta k} \left[ \frac{km}{h} \right]}$$

(eq 13)

In this case the transverse jerk has an effect on a virtual length $l_v$. This length depends on the vehicle and is influenced by e.g. the distance between the centres of the running gears, the distance between the axles in the running gear and the rigidity of the transverse springs.

For $l_v$ the distance between the centres of the running gears of the vehicle type mainly operated is to be selected. Simplified, $l_v = 6 \text{ m}$ can be applied.

The difference in curvature amounts to

$$\Delta k = \frac{1000}{r_2} \pm \frac{1000}{r_1} \text{ with } r_1 > r_2.$$

The curvatures for opposite curves are to be added and the curvatures for curves of the same kind to be subtracted.
For the transition without a transition curve between two curves of different curvatures the permissible speed per \( v \) results to

\[
perv = 31.5 \times \sqrt{\frac{l_{v}}{\Delta k} \left[ \frac{km}{h} \right]} \tag{eq 13a}
\]

with \( C = 0.67 \text{ m/s}^3 \) according to eq. 13. With \( l_{v} = 6 \text{ m} \) it follows that

\[
perv = 5.7 \times \sqrt{\frac{1}{\Delta k} \left[ \frac{km}{h} \right]} \tag{eq 13b}
\]

For the transition without a transition curve between a straight and a curve the permissible speed per \( v \) results to

\[
perv = 3.15 \times \sqrt[3]{r} \frac{\left[ \frac{km}{h} \right]}{l_{v} \times v} \tag{eq 13c}
\]

with \( C = 0.67 \text{ m/s}^3 \) according to eq. 13. With \( l_{v} = 6 \text{ m} \) it follows that

\[
perv = 5.7 \times \sqrt{r} \frac{\left[ \frac{km}{h} \right]}{l_{v}} \tag{eq 13d}
\]

The permissible speed per \( v \) resulting from eq. 13 can be rounded up to full 5 km/h, except if the radii are smaller than 140 m.

\text{(ii) Superelevation Ramp}

The transition between a track section not superelevated and a superelevated section or between two track sections with different superelevation is realised by way of a superelevation ramp.

Usually, such a superelevation ramp is straight.

The standard gradient of the superelevation ramp amounts to

\[
\frac{1}{m} = \frac{1}{10 \times v \cdot p} \tag{eq 14}
\]

In special circumstances the biggest gradient of the superelevation ramp amounts to

\[
\frac{1}{m} = \frac{1}{6 \times perv}, \text{ however, maximum } \frac{1}{300} \tag{eq 15}
\]

On the basis of the gradient of the superelevation ramp the length of the ramp amounts to

\[
l_{R} = \frac{m \times u}{1000} [m] \tag{eq 16}
\]
If the superelevation ramp and the transition ramp have different lengths, the longer length of the two lengths $l_R$ and $l_U$ shall be used as the joint length with the same beginning and the same end.

If the superelevation ramp is located partially in a curve, there has to be superelevation according to eq. 9 in each point of the transition curve.

Between two superelevation ramps with different gradients there has to be a track section without superelevation or with constant superelevation. This section has to have the minimum length of an intermediate straight in accordance with Section 5. It is allowed to do without this intermediate length if the gradient of the two superelevation ramps against one another does not exceed the gradient according to eq. 15.

If opposite curves follow one after the other, the superelevation of the first curve shall continuously be transferred to the superelevation of the second curve (track scissors).

(g) GRADIENT AND CHANGE IN GRADIENT

(i) Gradient

The gradient of the route tracks is to be in agreement with the starting and braking power of the vehicles. As a rule, the gradient shall not exceed the value $I = 40 \%$.

In case of difficult topographical conditions, higher gradients can be realised if the vehicles are designed accordingly.

Tracks at stops/stations, terminal loops and depots shall not have a gradient. The gradient at stops of in-street rail systems shall only exceed $40 \%$ in special circumstances.

In special circumstances, there can be gradients in terminal loops and depots if it is prevented that the trains can roll away.

(ii) Transition from One Gradient to Another

(1) As a rule, changes in the gradient shall be rounded with a radius of

$$ r_a \geq 0.4 * v_p^2 $$

however at least $r_a = 1000 \text{ m}$. In special circumstances

$$ r_a \geq 0.25 * perv^2 $$

however at least $r_a = 625 \text{ m}$, is permitted. Additional sighting distance checks are not affected.

There shall not be a change in the gradient in superelevation ramps. If this cannot be avoided, the transition is to amount to

$$ r_a \geq \frac{6 * v_p^3}{m} $$

however at least $r_a = 2000 \text{ m}$. 

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(3) In case of in-street track formations the transition from one gradient to another is to be adapted to the local conditions; the minimum transitions that can be passed by the vehicles are to be observed.

(4) The standard transition according to eq. 17 also applies to switches in valleys, but only to switches on hilltops if it can be ensured that the switch blade or the movable crossing lies on the slide chair within the area of the point; otherwise the transition radius shall amount to at least $r_a = 5000$ m or this switch area shall be at least 5 m away from the vertical graduated transition curve.

(h) **DEVIATIONS FOR NON-STANDARD GAUGE RAIL SYSTEMS**

Besides the standard gauge (1435 mm) other gauges are included in the scope of these Rules. The gauges deviating slightly from the standard gauge are considered to be standard gauges. The below table shows the deviations from the formulas and numerical values mentioned in Sections 6 to 8 that apply to the gauges $s = 1100$ mm and $s = 1000$ mm (metre gauge).

<table>
<thead>
<tr>
<th>Table 5.2.16</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Gauge</strong></td>
</tr>
<tr>
<td><strong>eq 2</strong></td>
</tr>
<tr>
<td><strong>eq 3</strong></td>
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<tr>
<td><strong>eq 4</strong></td>
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<td><strong>eq 5</strong></td>
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<td><strong>eq 7</strong></td>
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<td><strong>eq 8</strong></td>
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<td><strong>eq 9</strong></td>
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<tr>
<td>Sec. 8.3(6)</td>
</tr>
<tr>
<td><strong>eq 14</strong></td>
</tr>
<tr>
<td><strong>eq 15</strong></td>
</tr>
<tr>
<td><strong>eq 19</strong></td>
</tr>
</tbody>
</table>
5.2.4 Applicability of rail standards to urban rail (tram & metro)

Appendix A5.2.3 (separate xls-file) contains an overview on the existing CEN and CENELEC standards evaluated for their applicability.

5.2.5 Overview of a new embedded track standard under development in the US by APTA/AREMA

5.2.5.1. Need for recommended practices

WP5.2 is tasked with the development of performance specifications for embedded track systems.

In early 2007, the APTA Track Noise & Vibrations subcommittee came to the conclusion that the “Track design handbook for light rail transit” (TCRP Report 57) was no longer up to date and the AREMA “Manual of Railway Engineering” was not up to date and not complete in the Chapters covering Rail Transit issues.

5.2.5.2. Five priorities

The APTA and AREMA committees defined five priorities:

- Non-ballasted track;
- Embedded (paved) track;
- Corrosion control;
- Wheel/Rail interface issues;
- Sharp curve design.

5.2.5.3. Work plan

(a) ISSUES TO BE ADDRESSED

The updated documents must incorporate the following issues:

- Use of Tee rail vs. Grooved/Girder rail;
- Slab design: reinforced or non-reinforced;
- Optimum track modulus;
- Method(s) of controlling stray currents;
- Optimum track gauge to wheel gauge clearance;
- Method(s) of fastening rails;
- Curve guarding methods;
- Controlling noise and vibrations.

(b) KNOWLEDGE INVENTORY

It was established that most items could be addressed with knowledge readily available within the group. For others research statements were developed for submission to TRB (Transportation Research Board) and TCRP (Transport Cooperative Research Program).
5.2.5.4. **Actual progress - Embedded track**

The embedded track standard is the most advanced and has been balloted.

The rewrite consists of the following chapters:

8.1 Introduction (Annex 1)

This section describes the differences between LRV (Light Rail Vehicle) and Streetcars, as they have different operating environments and thus different track systems (especially curves).

8.2 Track alignment (Annex 2)

Both horizontal and vertical alignment are addressed.

Specific attention was paid to curves and the transitions from tangent to curve and vice versa.

A separate note titled “Analysis of Lateral Acceleration and Jerk Rate for Establishing Superelevation and Spiral Length” (Annex 3) provides the background for the curve design. This note is important as allows and increase of the lateral acceleration from 0.1 g to 0.15 g, which in turn allows an increase in the allowable unbalanced superelevation (can deficiency) on curves and allows correspondingly higher speeds regardless of actual superelevation.

8.4 Rail (Annex 4)

As girder rail or grooved rail is no longer rolled in the US, this section discussion the use of standard tee rail as an alternative.

It also points out that the girder rails described in the old standard are completely obsolete and lists and describes to the various new rails that are currently available.

5.2.5.5. **Actual progress - non ballasted track**

The work on the non ballasted track has only just started and consists of a draft Research Needs Statement (Appendix A5.2.4).

The document discusses the points of overlap with the embedded track as well as some specific issues:

- Stray current protection: track to earth resistance (TTE);
- Existing norms and measurement protocols:
  - Effects of rebar and type of rebar (coating);
- Guarding for curves;
- Slab track design and construction:
  - Use of rebar or not?
  - Plinths and their attachment;
  - Top down or bottom up construction;
  - Uptimum track stiffness design;
  - Transition to and from ballasted track.
APPENDIX A5.2.1: VIBRATION VELOCITY SPECTRA WITH MEASURED RAIL IRREGULARITIES

VIBRATION SPECTRA FOR THE TRAMWAY VEHICLE

Vehicle in curve (30 m radius) and EB50T rail

![Graph showing vibration velocity spectra with measured rail irregularities for a tramway vehicle. The graph includes curves for Ballast, CDM-Classic, CDM-Confort, Direct-Hard, and Direct-Soft.]
Vehicle in curve (30 m radius) and NP4am rail

![Graph showing frequency vs velocity for different ballast types at 20 km/h (NP4am)]

Vehicle in curve (30 m radius) and 35G rail

![Graph showing frequency vs velocity for different ballast types at 20 km/h (35G)]
Vehicle in curve (30 m radius) and Ri59N rail

![Graph: Curve at 20 km/h (Ri59N)]

Vehicle in curve (30 m radius) and Ri53N rail

![Graph: Curve at 20 km/h (Ri53N)]
Vehicle in straight line and EB50T rail

**Straight line at 20 km/h (EB50T)**

**Straight line at 40 km/h (EB50T)**
**Straight line at 60 km/h (EB50T)**

- **Graph 1:**
  - Frequency [Hz]
  - Velocity [dB re 1e-9m/s]
  - Ballast
  - CDM-Classic
  - CDM-Confort
  - Direct-Hard
  - Direct-Soft

- **Graph 2:**
  - Frequency [Hz]
  - Velocity [dB re 1e-9m/s]
  - 20 km/h
  - 40 km/h
  - 60 km/h
Vehicle in straight line and NP4am rail

**Straight line at 20 km/h (NP4am)**

- Frequency [Hz]
- Velocity [dB ref 1e-9m/s]

**Straight line at 40 km/h (NP4am)**

- Frequency [Hz]
- Velocity [dB ref 1e-9m/s]
Straight line at 60 km/h (NP4am)

Straight line on Ballast (NP4am)
Straight line on Direct-Hard (NP4am)

Straight line on Direct-Soft (NP4am)
Vehicle in straight line and 35G rail

**Straight line at 20 km/h (35G)**

- Frequency [Hz]
- Velocity [dB re 1e-9 m/s]
- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft

**Straight line at 40 km/h (35G)**

- Frequency [Hz]
- Velocity [dB re 1e-9 m/s]
- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft
Vehicle in straight line and Ri59N rail

**Straight line at 20 km/h (Ri59N)**

**Straight line at 40 km/h (Ri59N)**
Straight line at 60 km/h (Ri59N)

Straight line on Ballast (Ri59N)
Vehicle in straight line and Ri53N rail

**Straight line at 20 km/h (Ri53N)**

**Straight line at 40 km/h (Ri53N)**
Straight line at 60 km/h (Ri53N)

Straight line on Ballast (Ri53N)
Straight line on CDM-Classic (Ri53N)

![Graph showing velocity vs. frequency for different speeds on CDM-Classic.]

Straight line on CDM-Confort (Ri53N)

![Graph showing velocity vs. frequency for different speeds on CDM-Confort.]

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Straight line on Direct-Hard (Ri53N)

Frequency [Hz]

Velocity [dB re 1e-9m/s]

-40
-30
-20
-10
0
10
20
30
40
50
60

20 km/h
40 km/h
60 km/h

Straight line on Direct-Soft (Ri53N)

Frequency [Hz]

Velocity [dB re 1e-9m/s]

-40
-30
-20
-10
0
10
20
30
40
50
60

20 km/h
40 km/h
60 km/h
VIBRATION SPECTRA FOR THE LIGHT RAIL VEHICLE

Vehicle in curve (100 m radius) and UIC50 rail

Curve at 20 km/h (UIC50)

Curve at 40 km/h (UIC50)
Curve on Direct-Hard (UIC50)

Curve on Direct-Soft (UIC50)
Vehicle in straight line and UIC50 rail

**Straight line at 20 km/h (UIC50)**

**Straight line at 40 km/h (UIC50)**
Straight line at 60 km/h (UIC50)

- Frequency [Hz]
- Velocity [dB ref 1e-9 m/s]

Straight line at 80 km/h (UIC50)

- Frequency [Hz]
- Velocity [dB ref 1e-9 m/s]
**Straight line at 100 km/h (UIC50)**

- **Frequency [Hz]**
- **Velocity [dB re 1e-9m/s]**

- **Ballast**
- **CDM-Classic**
- **CDM-Confort**
- **Direct-Hard**
- **Direct-Soft**

**Straight line on Ballast (UIC50)**

- **Frequency [Hz]**
- **Velocity [dB re 1e-9m/s]**

- **20 km/h**
- **40 km/h**
- **60 km/h**
- **80 km/h**
- **100 km/h**
Straight line on Direct-Hard (UIC50)

Straight line on Direct-Soft (UIC50)
Vehicle in curve (100 m radius) and UIC54 rail

Curve at 20 km/h (UIC54)

Curve at 40 km/h (UIC54)
Curve at 60 km/h (UIC54)

![Graph showing velocity vs frequency for different ballast types.

Curve on ballast (UIC54)

![Graph showing velocity vs frequency for different velocities.]
Curve on CDM-Classic (UIC54)

Frequency [Hz]

Velocity [dB ref 1e-9m/s]

-30
-20
-10
0
10
20
30
40
50

4 5 6 8 10 13 16 20 25 32 40 50 63 79 100 128 158 200

20 km/h
40 km/h
60 km/h

Curve on CDM-Confort (UIC54)

Frequency [Hz]

Velocity [dB ref 1e-9m/s]

-30
-20
-10
0
10
20
30
40
50

4 5 6 8 10 13 16 20 25 32 40 50 63 79 100 128 158 200

20 km/h
40 km/h
60 km/h

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Vehicle in straight line and UIC54 rail

Straight line at 20 km/h (UIC54)

Straight line at 40 km/h (UIC54)
Straight line at 100 km/h (UIC54)

Straight line on Ballast (UIC54)
Straight line on Direct-Hard (UIC54)

![Graph of velocity vs frequency for different speeds on a direct-hard track.]

Straight line on Direct-Soft (UIC54)

![Graph of velocity vs frequency for different speeds on a direct-soft track.]
APPENDIX A5.2.2: VIBRATION VELOCITY SPECTRA WITH STEP FUNCTION RAIL IRREGULARITY

VIBRATION SPECTRA FOR THE TRAMWAY VEHICLE

Vehicle in curve (30 m radius) and EB50T rail

![Vibration Velocity Spectra Graph](image)
Vehicle in curve (30 m radius) and NP4am rail

![Graph](image1)

Vehicle in curve (30 m radius) and 35G rail

![Graph](image2)
Vehicle in curve (30 m radius) and Ri59N rail

Curve at 20 km/h (Ri59N)

Vehicle in curve (30 m radius) and Ri53N rail

Curve at 20 km/h (Ri53N)
Vehicle in straight line and EB50T rail

**Straight line at 20 km/h (EB50T)**

- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft

**Straight line at 40 km/h (EB50T)**

- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft
Vehicle in straight line and NP4am rail

Straight line at 20 km/h (NP4am)

Straight line at 40 km/h (NP4am)
Straight line at 60 km/h (NP4am)

Straight line on Ballast (NP4am)
Vehicle in straight line and 35G rail

**Straight line at 20 km/h (35G)**

![Graph showing velocity in decibels (dB) vs. frequency in Hertz (Hz) for different types of ballast (Ballast, CDM-Classic, CDM-Confort, Direct-Hard, Direct-Soft) at 20 km/h.]

**Straight line at 40 km/h (35G)**

![Graph showing velocity in decibels (dB) vs. frequency in Hertz (Hz) for different types of ballast (Ballast, CDM-Classic, CDM-Confort, Direct-Hard, Direct-Soft) at 40 km/h.]

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Straight line at 60 km/h (35G)

Straight line on Ballast (35G)
Straight line on CDM-Classic (35G)

![Graph showing velocity vs. frequency for different speeds (20 km/h, 40 km/h, 60 km/h).]

Straight line on CDM-Confort (35G)

![Graph showing velocity vs. frequency for different speeds (20 km/h, 40 km/h, 60 km/h).]
Vehicle in straight line and Ri59N rail

**Straight line at 20 km/h (Ri59N)**

**Straight line at 40 km/h (Ri59N)**
Straight line at 60 km/h (Ri59N)

Frequency [Hz]

Straight line on Ballast (Ri59N)

Frequency [Hz]
Straight line on CDM-Confort (Ri59N)

![Graph showing frequency vs. velocity for 20 km/h, 40 km/h, and 60 km/h on a straight line on CDM-Confort (Ri59N).]

Straight line on Direct-Hard (Ri59N)

![Graph showing frequency vs. velocity for 20 km/h, 40 km/h, and 60 km/h on a straight line on Direct-Hard (Ri59N).]
Vehicle in straight line and Ri53N rail
Straight line at 40 km/h (Ri53N)

![Graph showing velocity (dB ref 1e-9m/s) vs. frequency (Hz) for different materials: Ballast, CDM-Classic, CDM-Confort, Direct-Hard, Direct-Soft.]

Straight line at 60 km/h (Ri53N)

![Graph showing velocity (dB ref 1e-9m/s) vs. frequency (Hz) for different materials: Ballast, CDM-Classic, CDM-Confort, Direct-Hard, Direct-Soft.]
Straight line on Ballast (Ri53N)

![Graph showing velocity in dB re 1e-9 m/s vs frequency for different speeds (20 km/h, 40 km/h, 60 km/h).]

Straight line on CDM-Classic (Ri53N)

![Graph showing velocity in dB re 1e-9 m/s vs frequency for different speeds (20 km/h, 40 km/h, 60 km/h).]
Straight line on CDM-Confort (Ri53N)

Straight line on Direct-Hard (Ri53N)
VIBRATION SPECTRA FOR THE LIGHT RAIL VEHICLE

Vehicle in curve (100 m radius) and UIC50 rail
Curve at 40 km/h (UIC50)

Curve at 60 km/h (UIC50)
Vehicle in straight line and UIC50 rail
### Straight line at 80 km/h (UIC50)

![Graph showing noise levels at 80 km/h](image1)

### Straight line at 100 km/h (UIC50)

![Graph showing noise levels at 100 km/h](image2)
Straight line on Ballast (UIC50)

Frequency [Hz] vs. Velocity [dB re 1e-9m/s]

- 20 km/h
- 40 km/h
- 60 km/h
- 80 km/h
- 100 km/h

Straight line on CDM-Classic (UIC50)

Frequency [Hz] vs. Velocity [dB re 1e-9m/s]

- 20 km/h
- 40 km/h
- 60 km/h
- 80 km/h
- 100 km/h
Vehicle in curve (100 m radius) and UIC54 rail
Curve on ballast (UIC54)

- Frequency [Hz]
- Velocity [dB ref 1e-9m/s]

- 20 km/h
- 40 km/h
- 60 km/h

Curve on CDM-Classic (UIC54)

- Frequency [Hz]
- Velocity [dB ref 1e-9m/s]

- 20 km/h
- 40 km/h
- 60 km/h
Vehicle in straight line and UIC54 rail

**Curve on Direct-Soft (UIC54)**

**Straight line at 20 km/h (UIC54)**
Straight line at 40 km/h (UIC54)

Frequency [Hz]

Velocity [dB ref 1e-9m/s]

- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft

Straight line at 60 km/h (UIC54)

Frequency [Hz]

Velocity [dB ref 1e-9m/s]

- Ballast
- CDM-Classic
- CDM-Confort
- Direct-Hard
- Direct-Soft
**Straight line on CDM-Confort (UIC54)**

![Graph showing velocity vs frequency for different speeds on CDM-Confort](image)

**Straight line on Direct-Hard (UIC54)**

![Graph showing velocity vs frequency for different speeds on Direct-Hard](image)
APPENDIX A5.2.3 – APPLICABILITY OF RAIL STANDARDS

Separate xls-file D0504_D2S_M48_Appendix C.
APPENDIX A5.2.4 – EXTRACTS FROM AREMA, CHAPTER 12 – PART 8

8.1. INTRODUCTION

This section of Chapter 12 deals specifically with the planning, design, construction and maintenance of facilities and tracks used for what is commonly called “street running,” where the tracks are embedded in pavement or other road surface, and generally the paving surface is even with the top of rail. Two types of street running track will be covered herein. Type 1 Embedded Track is founded on a concrete slab, similar to non-ballasted track (covered elsewhere), and the paving infill is usually concrete or asphalt, but can also be pavers, paving stones, grass, etc.. Type 2 is herein called “Paved Track” and is ballasted track of various types (concrete, wood, steel or plastic ties in crushed stone ballast, etc.) covered with either asphalt, concrete or other type of pavement. Both types of track may be used by a wide variety of steel wheeled vehicles including light rail vehicles, streetcars, trolleys or trams (the name depending on local preference), and sometimes shared use with freight trains, and the track structure must accommodate the types of traffic anticipated, including heavy-axle load rubber-tired traffic.

As the variety of vehicles that might use the tracks covered in this Section of Chapter 12 of the Manual are myriad, the following verbiage will be used to describe the typical vehicles, viz:

- Light Rail Vehicle (LRV): a vehicle of modern design, sometimes with four axles but frequently articulated and having six or more axles, used in street running but primarily intended for relatively high-speed travel between fairly widely spaced stations, often operated coupled in trains, top operating speed in the 55-65 mph range, and usually limited to minimum curve radii of 82-83 ft.
- Streetcar: a vehicle of either heritage or modern design, frequently having four axles, but sometimes articulated and having six or more axles, used primarily in mixed traffic, street running in downtown circulator operations, based on the tracking capabilities of the ERPCCar, top operating speed in the 30-45 mph range, and usually capable of negotiating curve radii down to 35-39 ft.

As all recommendations in this Section are related to hypothetical vehicles, not specific ones, it is absolutely essential that the designer and specifier be fully conversant with the operating and tracking capabilities of the vehicle(s) that will actually use the tracks, and to verify suitable track geometric and alignment criteria that will interact and work properly. It is equally essential that the track designer be constantly aware that there may be characteristics of the shared street civil or architectural design that may be detrimental to the design of good and safe track alignment, and that any conflicts should be resolved as early as possible in the planning..

Embedded track requires special planning and design approaches to integrate the rail facilities into the urban streetscape successfully and to have the rail vehicles interact efficiently and safely with the rubber-tired traffic in the shared roadway, while maintaining the appropriate balance between the needs of the rail transit system and other stakeholders in the busy urban environment. This starts with careful planning to be sure there are no glaring safety issues caused by the track alignment or facilities and that the rail vehicles will mesh well with the overall traffic plan and signaling. Further, that the installation in the streets will not significantly degrade the operation of the rail vehicles, such as excessive street surface drainage crossfall and curves without spirals. The planning should also include considerations of ancillary facilities such as locations and designs of overhead contact wire system poles, stations, stops, traction power substations, pedestrian crosswalks, safety zones, etc.
The design involves developing a comprehensive alignment plan and construction details that cover the unique requirements of street running, generally described as:

- Types of rail traffic; vehicle loadings and geometric requirements, wayside clearances, safety issues
- Locations and details for special trackwork, with particular attention to inspection and maintenance, as well as the interfaces and potential hazards associated with placement of special trackwork in areas shared with motor vehicles and/or pedestrians.
- Traction power, TP wayside facilities, stray current and corrosion control
- Integration of the track into the street design physically, operationally and esthetically
- Special considerations such as bridges, tunnels, viaducts, especially passenger/pedestrian safety issues
- Track maintenance inspection, access and repair considerations
- Traffic and rail signal integration; vehicle and pedestrian grade crossings, parking lanes and safety zones
- Station, stops and their amenities, including safe pedestrian access and protection from auto and rail traffic
- Maintenance and repair management considerations; life-cycle costs

The planning and the design items listed above are covered in detail in the sections following, and with references to other Chapters of the Manual and other authoritative sources, as appropriate. These recommendations are based not only on theory but also on documented experience from both successful and unsuccessful embedded and paved track and facilities projects and rail transit properties operating extensive embedded and paved track operations. Where criteria or plans are quoting a specific Agency’s standards, it will be noted, and the reader should be aware that such standards tend to be property-specific and should be thoroughly investigated as to their appropriateness for any other project. We quote below advice from the American Transit Engineering Association Engineering Manual, Way and Structures Division “W”, issue of 1923:

"This specification is intended to cover the construction of electric railway track in paved city streets. It is obvious that no general specification can be prepared for such work to cover all special types of track construction, or to meet special conditions. The scope of this specification has therefore been limited to an expression of the fundamental principle which should be followed out in constructing track in paved streets."

AREMA Committee 12 believes that was good advice in 1923, and is still good advice at present, and therefore we are following it as faithfully as possible in our development of this Manual material.

8.2 TRACK ALIGNMENT

8.2.1 GENERAL

Alignments for embedded track in streets are frequently more constrained than for other LRT track types (ballasted and direct fixation.) Embedded tracks follow streets within traffic lanes and curb offsets, make tight turns within street intersections and follow pavement cross sections and profiles.

The primary objectives of any track alignment are cost effectiveness, operating efficiency and passenger safety and comfort. The alignment recommendations in this section include worst case criteria for application to embedded track alignment. Like all alignments, the absolute maximum/minimum
alignment criteria herein are to be avoided in favor of longer tangents, flatter curves, and longer spirals wherever possible. Where the costs of street modifications are minor, they should be incorporated if they will improve the alignment. Extensive use of absolute maximum/minimum values results in slower operations and higher maintenance costs.

It is recommended that these worst case criteria be combined with more conservative criteria into a single criteria document for any specific project. Alignment criteria may be found in Chapter 3 of TCRP Report No. 57, and in both Chapter 5 Part 3, and Part 3.5 of this chapter of the AREMA Manual. Developing a general criteria that includes worst case allowable criteria will reduce the time consuming effort required to grant variances from the general criteria that are often needed otherwise. As stated above, these worst case criteria should be applied only when general criteria will not produce a feasible design. Even with comprehensive criteria containing desired values, minimum/maximum values, and absolute minimum/maximum values, field conditions will occur requiring engineering analysis of alternatives, judgment and compromise to arrive at a safe, efficient solution.

The criteria in Part 8.2 are based on a typical light rail vehicle (LRV). If possible during preliminary design, the vehicle parameters affecting alignment criteria should be established. For final design, it is imperative that the vehicle parameters affecting alignment criteria be established and the project alignment criteria adjusted accordingly. Due to the greater variance amongst contemporary streetcars and trolley cars (either new or restored vintage cars) compared with contemporary LRVs, the advice in this paragraph is of even greater importance for these type operations.

Street running embedded track speeds are usually limited to the legal speed of the roadway traffic which is seldom over 35mph. For embedded track in open running territory where the typical LRT vehicle is capable of a sustained operating speed of 55mph or higher, more conservative (lower maximum and higher minimum) values should be considered for alignment criteria although the absolute maximum/minimum values given here are still applicable.

Combinations of any of maximum grade, maximum unbalanced superelevation, minimum horizontal curve radius and minimum vertical curve radius should be avoided. (Further guidance on this issue is to be developed.)

These criteria assume standard gauge track (56.5 inches.) plus or minus small adjustments for tight gauge and gauge widening.

Many of the criteria stated herein are excerpted from or derived from information in TCRP Report No. 57. TCRP Report No. 57, Track Design Handbook for LRT is available from the US Transportation Research Board (TRB).

8.2.2 VEHICLE INTERFACE

These embedded track alignment criteria reflect the operating limitations of typical modern LRT vehicles. Circulator system vehicles (street car and trolley car are used synonymously herein) are often capable of tighter radius horizontal and vertical curves than an LRV.

Individual vehicles may be significantly different in one or more operating characteristics than the typical values given here. This is known to be specifically true for vehicles with trucks having independently turning wheels.
These criteria, based on typical values for an LRV, may be considered useful for preliminary design but they should be adjusted as the actual operating characteristics are established. It is imperative for an efficient final design that the vehicle specification (or consultant or manufacturer) be consulted as to vehicle limiting operating characteristics. Alignment criteria for final design must be compatible with the selected vehicle. Vehicle characteristics should be based on worst case of new or deteriorated condition. For example, minimum clearance under the vehicle which affects allowable crest vertical curve radius may be reduced for worn or collapsed suspension compared with new conditions.

For an average, modern, bi-directional, coupled, fully loaded, articulated LRV, typical limiting operating characteristics are:

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum vehicle operating speed</td>
<td>55 mph</td>
</tr>
<tr>
<td>Maximum allowable grade</td>
<td>7%</td>
</tr>
<tr>
<td>Minimum horizontal curve radius</td>
<td>82 feet</td>
</tr>
<tr>
<td>Minimum vertical curve radius</td>
<td>crest: 820 ft, sag: 1150 ft</td>
</tr>
<tr>
<td>Maximum allowable rate of twist</td>
<td>1 inch in 25 ft</td>
</tr>
<tr>
<td>Maximum vehicle roll angle</td>
<td>&lt; 1.5 degrees (stabilized suspension)</td>
</tr>
<tr>
<td>Typical truck spacing</td>
<td>22 to 30 ft</td>
</tr>
</tbody>
</table>

For comparison, typical trolley car limitations are:

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum vehicle operating speed</td>
<td>35 mph</td>
</tr>
<tr>
<td>Maximum allowable grade</td>
<td>9%</td>
</tr>
<tr>
<td>Minimum horizontal curve radius</td>
<td>35 feet, centerline of track. (Some trolley minimum radius criteria are stated as inside rail radius, which is also sometimes used to designate turnout radius.)</td>
</tr>
<tr>
<td>Minimum vertical curve radius</td>
<td>crest: 310 ft; sag: 560 ft</td>
</tr>
<tr>
<td>Maximum allowable rate of twist</td>
<td>1 inch in 12.5 ft</td>
</tr>
<tr>
<td>Maximum vehicle roll angle</td>
<td>Varies</td>
</tr>
<tr>
<td>Typical truck spacing</td>
<td>22 ft</td>
</tr>
</tbody>
</table>

### 8.2.3 Horizontal Alignment

Horizontal alignment consists of tangents, circular curves and spirals in various combinations.

#### 8.2.3.1 Tangents

- The desirable minimum tangent between curves should be the truck spacing plus axle spacing of a truck (overall wheelbase) so that a vehicle will have adjacent trucks exit one curve before entering another. No portion of the tangent should be superelevated.
- The absolute minimum tangent between curves is zero so long as the resultant geometry does not exceed the vehicle coupler maximum angle, the speed does not exceed 20 mph and the adjoining curves are unsuperelevated.
If adjoining curves are superelevated, they must have spirals or intervening tangent of sufficient length to meet superelevation runoff requirements.

The foregoing criteria apply to reverse curves. For curves in the same direction, a smoother ride results from a compound curve rather than a short tangent between the two curves. Compound curves should have spirals connecting the different radius portions of the curve. The spiral shall begin with the radius of one curve and uniformly increase/decrease to the radius of the other adjacent curve and not be back to back spirals meeting at a common tangent.

8.2.3.2 Curves

- The desirable minimum curve radius is 1.5 times the absolute minimum radius.
- The absolute minimum radius is that radius at which a coupled vehicle will negotiate the curve.
- The desirable minimum length of circular curve (in feet) is three times the normal operating speed (in mph) of the curve. For spiraled curves this is the length of the circular curve plus one half the sum of the lengths of the spirals.
- For unsuperelevated spiraled curves the minimum circular curve length is zero, ie back-to-back spirals.

8.2.3.3 Superelevation

Street running track does not often allow for design of actual superelevation (E_a) based solely on operating speeds. While actual superelevation is not precluded on street running track, it is likely that the superelevation will have to accommodate the cross slope of the street as well as the desired superelevation. Negative superelevation can occur and speed should be adjusted accordingly.

Since street running requires frequent speed reductions and stops to accommodate street traffic, the maximum E_a should not exceed 3 inches. Exceptions to this maximum, such as roadway curves with larger than 3 inch of cross slope in the roadway and where frequent stopping of trains is unlikely offer opportunities to use higher actual superelevation. On tangents the maximum cross slope should not exceed one inch. On tangents and curves, the differential between the street cross slope and track cross slope/superelevation should not be greater than one inch.

8.2.3.4 Allowable Speed on Curves

Based on ride comfort for short trips and assuming well maintained alignment on LRT embedded track systems, the recommended lateral acceleration of 0.1g (6 inches of unbalance) may be increased in critical locations to a lateral acceleration of 0.15g (9 inches of unbalance superelevation).

Allowable speed on a curve is:

\[
V = \sqrt{\frac{E}{0.0007 D}}
\]

Where \( V \) = speed in mph

\( E \) = total superelevation in inches, the sum of \( E_r + E_u + E_a \)

Where \( E_r \) = equivalent car body roll allowance which for a stabilized vehicle is 1.5 inches and for unstabilized suspensions is 3 inches
\( E_u = \) design unbalance: up to 4.5 inches recommended with 7.5 inches maximum for stabilized suspension cars and up to 3 inches recommended with 6 inches maximum allowed for unstabilized suspensions

\( E_a = \) actual superelevation in inches.

\( D = \) degree of curvature \((5730/\text{radius in feet})\)

### 8.2.3.5 Spirals

Spirals should be used on all mainline (passenger carrying) embedded track curves. For zero actual superelevation on embedded track curves, spiral length is determined based on the rate at which lateral acceleration (unbalance) is introduced. The maximum rate of change of lateral acceleration (jerk rate) is 0.1\( g/s \). The absolute minimum length spiral \( L_s \) is therefore:

\[
L_s = 0.29 V E_u
\]

Where \( L_s \) = length of spiral in feet

\( V = \) velocity in miles per hour

\( E_u = \) unbalance from the curve computation in inches

When curves have superelevation in them, the rate of attainment should not exceed a vertical acceleration rate of change of 0.1\( g/s \). The equivalent formula is:

\[
L_s = 0.29 V E_a
\]

The ability of the vehicle to withstand twist must also be considered when \( E_a \) is used. For a typical LRV with an allowable rate of twist of 1 inch in 25 ft, the formula is:

\[
L_s = 25E_a
\]

The longest spiral computed using these three formulae determines the actual spiral length to be used. The more conservative formulae given in Part 3.5 of this Chapter should be used where they do not cause excessive cost to implement.

There are many different methods for computing spiral parameters. The notations and formulae in Chapter 5, Part 3 are recommended for spiral layout computations.

Many different philosophies have been used to proportion \( E_a \) and \( E_u \) on curves. See TCRP Report No. 57 for applicable formulae.

### 8.2.4 VERTICAL ALIGNMENT

Vertical alignment is comprised of tangential gradients joined together by parabolic vertical curves.

#### 8.2.4.1 Tangent Grades

- Maximum gradient must be based on vehicle braking and tractive effort. Typically for LRVs this requires that sustained grades over 2500 ft long not exceed 6\% and shorter sustained grade not exceed 7\%.
Minimum tangent length between vertical curves; desired 100 ft; minimum is truck spacing plus axle spacing on a truck (overall wheel base), usually about 40 ft. Absolute minimum is zero.

Desirable grade at stations is 0% to 0.35% and in the United States may not exceed 2% in order to comply with ADA provisions.

8.2.4.2 Vertical Curves

Vertical alignment must follow street grades including crown of cross streets unless the streets will be re-graded and re-crowned as part of the track construction. The critical vertical curve is the minimum radius the vehicle will accommodate for sag and crest curves. The minimum vertical curve must allow for clearance of the underside of the vehicle adjusted for wear and collapsed suspensions.

A typical LRV’s vertical curve radius limit is usually around 820 ft for crests and 1150 ft for sags. Using these values, the equivalent minimum curve length (LVC) can be determined from:

\[ LVC = 0.01AR \]

Where \( A \) = algebraic difference (using the percent grade as whole numbers, i.e. 2.0% = 2, -2.0% = -2 and 0.35% = 0.35) of gradients connected by the curve, and

\( R \) = Limiting radius in ft

For example, crossing a street with a 2% crown (1:50 cross slopes) the minimum LVC = 0.01 x (2 minus -2) x 820 = 32.8 ft. This length LVC would fit a 40 ft wide street.

The minimum crest LVC is

\[ LVC = AV^2/25 \]

Where \( V \) = design speed in mph

The minimum sag LVC is

\[ LVC = AV^2/45 \]

Using the above sample curve, the speed for the 32.8 ft long crest LVC should not exceed 32.8 = 4V^2/25. \( V = 14.3 \) mph.

Back to back reverse curves are acceptable as long as the above minimums are met by each curve.

Drafted by: Arthur Keffler, 4/7/08
Revised 7/30/08, 8/6/08, 9/17/08, 1/12/09, 1/16/09, and 2/12/09

8.4 RAIL
8.4.1 Rail Considerations

This section discusses rail sections and provides information and recommendations for their application in embedded track. Both tee rails and grooved rails are used in constructing embedded tracks. Grooved rails have the advantage of a built-in flangeway, but tee rails are equally satisfactory when properly applied and more economical, and are now used by most systems for rehab and new construction. In order of frequency of use, the four major types of embedded track and rail usage in North America are as follows:
a. Tee rail (nearly all 115RE) in concrete with formed flangeways, frequently installed with rubber/plastic coverings and sometimes rubber/plastic flangeways that are part of the rail electrical isolation system, also occasionally embedded in elastomeric grout

b. Grooved rails of various heights and weights, frequently installed with rubber/plastic coverings and sometimes embedded in elastomeric grout; all are usually part of the rail electrical isolation system

c. 115RE with a flangeway formed using “Strap Guard”, mountings similar to a. & b.

d. 115RE on the high side and a grooved guard rail on the low side of guarded curves, mountings similar to a. & b.

### 8.4.1.1 Rail Selection Criteria

When considering the specifications for a rail section or sections for use in embedded track the following six most important considerations should be used to evaluate tee and grooved rails sections:

1. Suitability for the application:
   a. beam strength
   b. head profile to match wheel profiles and have recommended gauge face angle
   c. projected wear life of plain and premium rails
   d. height of section which impinges on excavation and paving depth details
   e. applicability to the project paving and rail mounting details, and providing a suitable, ADA-compliant flangeway that is architecturally pleasing and maintainable
   f. requirement for guarding, either curves or fully guarding all tracks
   g. matching prior rail usage on the property
   h. adequate cross section area and conductivity for negative return without excessive voltage drop

2. Cost factors:
   a. first cost
   b. premium feature first cost
   c. projected life-cycle cost
   d. projected future cost for repair or extensions
   e. Added cost for guarding devices where needed

3. Availability:
   a. rolling frequency
   b. projected long-term availability
   c. multiple sources preferred
   d. availability of premium features and long lengths
   e. compliance with Buy America provisions, if applicable

4. Metallurgy & maintenance
   a. weldability, electric flash-butt and thermite
   b. requirements for special treatment of welds such as post-weld hardening
   c. ease of compromise welding to rails of different metallurgy
   d. running surface hardness achieved by alloying or heat-treating, or both
   e. subject to brittle fracture (especially in cold climates)
   f. grinding – are grinders available to be used for corrugation removal and re-profiling

5. Adaptability to special trackwork
   a. availability of matching cast and/or built-up components
   b. adaptability to machining and pre-curving
   c. section height suitable for use of asymmetric switch tongues/points
d. suitability for laying in plates or DF Fastenings

6. Quality Assurance
a. Availability of industry recognized quality standards & inspection techniques
b. QC requirements that lend themselves to normal field inspection methods
c. Availability of trained inspectors and suitable equipment to verify the QC requirements

8.4.1.2 Use of Tee or Grooved Running Rails

Based on the criteria above, many properties in North America have selected the 115RE tee rail section for use in embedded track. The selection was based on the following conclusions:

a. Suitable for most applications regarding strength, head profile, wear life, height, etc.
b. Interfaces well with the AAR 1B wheel profile; reasonably well with ATEA-type wheel profiles
c. Readily available from several producers; Buy America compliant
d. Initial cost; reasonable delivery times
e. Available head-hardened and in long lengths; some mills furnish CWR
f. Easy to weld, both flash-butt and thermite
g. Some matching special trackwork appliances available
h. Adequate current capacity for most operations
i. Dimensions, properties and quality are controlled by AREMA Chapter 4 specifications, which are well respected and understood in the industry

It is recommended that the designer or specifier give proper consideration to all the factors listed above, and apply proper weighting of those factors based on project-specific criteria, including the historical or aesthetic concerns. The 115RE rail section is normally more cost-effective than grooved rail, and can be used where practical. Alternatively, other tee rail sections can also be used, such as 85 ASCE, 90 ARA-A, or 100 ARA-B, if available, either new or Class I condition relay. However, there are situations where grooved rails are preferred, and may have attributes that offset some or all of the additional cost of the rails, such as:

a. The integral flange guard, in the old days called “the tram”, provides additional protection against heel-climb derailments, especially on sharp curves and in special trackwork
b. Having the infill paving, especially asphalt, flush with top of rail on each side reduces the potential for raveling or chipping and spalling of the pavement
c. The relatively small flangeway opening reduces the tripping hazard for pedestrians and bicycles vs a large, tooled flangeway
d. Grooved rail is much easier to lay in elastomeric grout embedment, as it doesn’t need a separate flangeway formed in the grout
e. Concrete placement/finishing with modified paving machinery is easier with grooved rail

It should be noted that using tee rail in embedded track requires a means to maintain a suitable flangeway opening in the infill paving, such as:

a. In Portland cement concrete, a blocked-out, troweled or screeded flangeway of appropriate dimensions and shape can be easily formed in the concrete
b. In less rigid paving infills, such as hot-mix asphalt, pavers, brick, crushed stone, a flangeway guarding device will be required such as shown in 8.4.1.3 or a rubber or plastic flangeway former
c. In rails mounted in polyurethane or similar resilient polymers, a flangeway must be formed in the polymer, by pouring the polymer low on the gauge side, by use of a flangeway forming blockout, or a flangeway forming device as shown in 8.4.1.3
8.4.1.3 Typical Flangeway Guarding Methods & Appliances

When tee rail is used, a flangeway can be tooled into the concrete; however, this is not always acceptable. Therefore, other methods of forming the flangeway are shown in the four figures below. These are only four of many possible methods, some proprietary, which will produce a satisfactory flange guard. Where curves are to be guarded, a restraining guard rail device must be added to the tee rail. Flange guard must not be confused with restraining guard rail; some of the designs shown are configured correctly and are robust enough to act as restraining guard rail, some are not. A careful choice must be made as to the appropriate design for service as a restraining guard rail and the specifications should cover in detail the proper mounting method and required hardware.

Note: There are other methods, as noted above, available to provide a flangeway, some proprietary. See additional details including electrical isolation techniques in Section 8.5. Installation Methods
8.4.1.4 Discussion of Girder Rail and Grooved Rail Usage

Historic Domestic Girder, Grooved Girder & Grooved Girder-Guard Rails

Prior to the 1940’s, domestic steel mills produced plain girder, grooved girder or grooved girder-guard rail sections in several heights and weights, as shown in the cut, above. After the 1980’s, when only two sections were available that were originally intended for freight service, there has not been any grooved girder rail rolled in North America. None of the Sections shown on AREA Plan 1003-40 are available new. Therefore, if tee rail with or without a flangeway guarding device/method is not a viable option, either for practical or architectural reasons, it will be necessary for the designers and builders of embedded tracks to use grooved rails of non-domestic manufacture. Because this situation has lasted a long time, the usage of European specialty rails purpose-designed for use as grooved running rails, restraining guard rails and switch points has become fairly common. Information on most of these rails is in the following section.

8.4.1.5 European Grooved Rails and Special Trackwork Rail Profiles

The grooved rails produced primarily by European rolling mills are not currently covered by AREMA specifications; they are covered by several European standards organizations, which control both the design and manufacture. Section 8.4.2 will provide information on the standards organizations and their respective specifications and recent changes in those organizations’ responsibilities. This is furnished as information only, not an AREMA specification. It is the responsibility of the designers and users to familiarize themselves with the appropriate, current specifications for rails and special rail sections contemplated for use in North American projects that will be rolled in non-domestic mills, and with the terminology used. This section will cover the topics listed below, viz:
a. Information on the standards organizations controlling the specifications applicable to grooved rails and special rail sections produced primarily in non-domestic mills and to which AREMA specifications do not presently apply, and limited details of those specifications and/or recommended practices

b. The changes in nomenclature applicable to certain rails and special sections produced to European or other standards that are frequently used in North America

c. Typical manufacturing specifications, tolerances and testing of the rails noted in a., and b., above

d. Drawings and physical characteristics of certain rails and special sections produced to European or other standards that are frequently used in North America

e. General recommendations for selection of appropriate rails and special sections

f. Special considerations related to handling, welding (both field and shop), laying and de-stressing of embedded rails

8.4.2 Standards Organizations and Relevant Standards or Recommended Practices

8.4.2.1 Standards Organizations

a. UIC - The International Union of Railways (UIC is a French acronym for, “L’Union Internationale des Chemins de fer”) is an international organization based in France whose purpose is to promote the interests of railway transport on a worldwide basis, including technical cooperation. Prior to the creation of the European Union, many rail standards were controlled by the UIC, and rail sections were named with UIC in the nomenclature, such as UIC-60 and UIC-33. That role is now filled by the CEN (see below). The UIC is similar to the AAR combined with the focus on passenger transport of APTA.

b. VDV – The Association of German Transport Undertakings (in German = “Verband Deutscher Verkehrsunternehmen”; formerly “VöV”) is an organization of German-speaking public transit and freight rail groups to provide cooperative technical guidance similar to AREMA; the specifications they publish are recommended practices, not standards. Grooved rails were, and in some cases still are, supplied per VDV specifications, and tramway special trackwork is still controlled by VDV.

c. CEN – The European Committee for Standardization (languages: English, German, French) is based in Brussels, Belgium, and publishes standards for a multiplicity of technical endeavors, including rails controlled by the steel committee. The CEN is like ASTM, ACI, ASME, IEEE, SAE, AAR, AREMA, etc. rolled into one standards organization. The signatory countries, now more than thirty, are required to accept the “European Norm” standards as their own without alteration. These standards have “EN” plus an identification number and date of approval in the name; in the case of grooved rails and special “construction” sections, the CEN standard is EN 14811:2006, which replaced both UIC and VDV specifications in most cases. If the standard has “pr” before the name, such as prEN 14811:2006, that indicates a “provisional” status; the provisional standard has been approved by the sponsoring committee, but has not been approved by all the signatory countries. However, it is generally considered to be in effect as approved standards drafted by the designated controlling committee(s) are seldom rejected by the signatories. The information following is based primarily on the CEN EN 14811:2006 standard with some additional information from CEN standard EN 13674:2005 which covers tee (Vignole, also called flat bottom) rails and special sections of interest such as restraining guard rails, STW construction rails, and asymmetric switch point sections.

And for domestically produced tee rails the relevant standards are controlled by:

a. American Railway Engineering and Maintenance-of-Way Association (AREMA) – domestic tee rails only
b. American Society of Civil Engineers (ASCE) – lighter tee rail sections, mainly 85 AS, primarily an industrial section, but rolled regularly

8.4.2.2 Applicable European CEN Standards EN 14811, EN 13674, and VDV

The nomenclature of grooved rails and certain special construction rail sections have been standardized and harmonized per Table 1, below. All drawings, plans, specifications and procurement documents should reflect the proper CEN Standard nomenclature, where applicable, to avoid confusion and errors. If the profiles are per VDV standards, the same information noted should appear in all documents.

Table 1
Revised Standard Nomenclature of Grooved Rails and Construction

<table>
<thead>
<tr>
<th>CEN Standard Profile Designation</th>
<th>Prior Profile Designations (VDV, UIC, etc.)</th>
<th>Generally Applicable To</th>
<th>Fig.</th>
</tr>
</thead>
<tbody>
<tr>
<td>51R1</td>
<td>Ri 52-R13, Ri 52</td>
<td>Running rails, H = 130 mm</td>
<td>1</td>
</tr>
<tr>
<td>53R1</td>
<td>R1 53-R13, Ri 53</td>
<td>Running rails, H = 130 mm</td>
<td>2</td>
</tr>
<tr>
<td>55G1</td>
<td>35 GP</td>
<td>Running rails, H = 152.5 mm</td>
<td>3</td>
</tr>
<tr>
<td>56R1</td>
<td>Ri Ic</td>
<td>Running guard rails, H = 160 mm</td>
<td>4</td>
</tr>
<tr>
<td>59R1</td>
<td>Ri 59-R10, Ri 59</td>
<td>Running rails, small g.c. radius, H = 180 mm</td>
<td>na</td>
</tr>
<tr>
<td>59R2</td>
<td>Ri 59-R13, Ri 59N</td>
<td>Running rails, large g.c. radius, H = 180 mm</td>
<td>5</td>
</tr>
<tr>
<td>60R1</td>
<td>Ri 60-R10, Ri 60</td>
<td>Running rails, small g.c. radius, H = 180 mm</td>
<td>na</td>
</tr>
<tr>
<td>60R2</td>
<td>Ri 60-R13, Ri 60 N</td>
<td>Running rails, large g.c. radius, H = 180 mm</td>
<td>6</td>
</tr>
<tr>
<td>62R1</td>
<td>NP4aMod</td>
<td>Running guard rails, H = 180 mm</td>
<td>7</td>
</tr>
<tr>
<td>67R1</td>
<td>Ph 37a</td>
<td>Running rails, large flangeway, H = 180 mm</td>
<td>8</td>
</tr>
<tr>
<td>49E1A1</td>
<td>Zu2-49</td>
<td>Switch tongue profile, H = 116 mm</td>
<td>9</td>
</tr>
<tr>
<td>61C1</td>
<td>Ri ii</td>
<td>STW const. grooved rail, flange-bearing, H = 160 mm</td>
<td>10</td>
</tr>
<tr>
<td>75C1</td>
<td>BA 75</td>
<td>STW const. grooved stock rail, H = 180 mm</td>
<td>11</td>
</tr>
<tr>
<td>76C1</td>
<td>VK Ri 60</td>
<td>STW const. blind groove guard rail, H = 180 mm</td>
<td>12</td>
</tr>
<tr>
<td>33C1</td>
<td>U69, UIC33, Ri 1-60</td>
<td>Frog guard &amp; restraining rails, H = 93 mm</td>
<td>13</td>
</tr>
<tr>
<td>Fz 36¹</td>
<td>Fz 36, Zu 36</td>
<td>Switch tongue profile, H = 75 mm</td>
<td>14</td>
</tr>
<tr>
<td>GGR-118²</td>
<td>GGR-118</td>
<td>Running grooved guard rails, H = 168,3mm (6.625-in)</td>
<td>na</td>
</tr>
</tbody>
</table>

Footnotes:
1) Section is not controlled by standards; produced per producing mills’ and/or users’ designs & specs
2) Section is no longer rolled but is in track on several NA properties, as info only; not a CEN standard
3) g.c. is gauge corner

General notes:
1. The rail sections listed above either are being or have been used in North America with some regularity.
2. Not all sections listed in Table I are illustrated on the following pages. In addition, many more rail sections (profiles) not listed here are available from some manufacturers. Those have not been included here because they have either not been adopted as CEN standards or have seen
little or no use in North America. For other sections available, refer to mill catalogues and to the referenced CEN standards, or other standards if not covered by CEN.

3. Sections including the letter “R” in their designation are grooved rails; in German, “Rillenschiene”. Grooved rails, commonly known in North America as “girder rails”, are rolled with an integral flangeway in the head of the rail, and used in the construction of ordinary embedded track. Sections including the letter “C” in their designation are known as “construction rails” and used in the fabrication of special trackwork. When fabricating STW using construction rails, the flangeways and head contours are machined (see Fig. 12). Additional details of embedded special trackwork construction are in 8.7, presently under development, which also covers designs outside the scope of CEN and VDV standards.

4. For a more detailed discussion of the application of grooved rails in LRT construction, especially as relates to wheel profile/rail groove matching, please see Transit Cooperative Research Program (TCRP) Report #57 “Light Rail Track Design Handbook”.

8.4.2.3 Rail Profile Drawings with Properties of the Sections/Profiles

Figures 1 through 14 shown below have the principal dimensions called out, along with basic section properties. Some sections are not shown where they are almost identical to another section, with the key differences noted. For complete dimensions and properties, please refer to the appropriate CEN or VDV Standard, or the producing mill’s drawings or catalog.

*Fig 1 is typical of the drawing style & dimensioning to be used for all rail sections shown; these figures are provisional only until artwork is completed*
Note: Profiles 59R1 and 60R1 are similar to 59R2 and 60R2, respectively, except that the gauge corner radius is 10mm (0.394-in), rather than 13mm, and the flangeway is approximately 3-5mm narrower.
Fig 11 - Profile 75C1

Fig 12 - Profile 76C1

Fig 13 - Profile 33C1

Fig 14 - Fz36 Tongue Rail
8.4.2.4 Properties of Grooved and Construction Rail Profiles/Sections

Table 2

Table of Properties - Grooved Rails and Construction Rails per CEN Standards EN-14811 and EN-13674, & Some Proprietary Items (Partial List)

<table>
<thead>
<tr>
<th>CEN Profile</th>
<th>Linear Mass</th>
<th>Area</th>
<th>I xx (Note 2)</th>
<th>I yy</th>
<th>$S_{head}$</th>
<th>$S_{base}$</th>
<th>Rail Lg/Unit Wt</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>kg/m</td>
<td>lb/yd</td>
<td>cm²</td>
<td>in²</td>
<td>cm⁴</td>
<td>in⁴</td>
<td>m/tonne</td>
</tr>
<tr>
<td>51R1</td>
<td>51.37</td>
<td>103.6</td>
<td>65.44</td>
<td>10.14</td>
<td>1289</td>
<td>30.93</td>
<td>695.6</td>
</tr>
<tr>
<td>53R1</td>
<td>52.98</td>
<td>106.8</td>
<td>67.49</td>
<td>10.46</td>
<td>1326</td>
<td>31.82</td>
<td>738.4</td>
</tr>
<tr>
<td>55G1</td>
<td>54.77</td>
<td>110.4</td>
<td>69.78</td>
<td>10.82</td>
<td>2076</td>
<td>49.82</td>
<td>681.5</td>
</tr>
<tr>
<td>56R1</td>
<td>55.98</td>
<td>112.9</td>
<td>71.31</td>
<td>11.05</td>
<td>2477</td>
<td>59.45</td>
<td>802.0</td>
</tr>
<tr>
<td>59R1</td>
<td>58.97</td>
<td>118.9</td>
<td>75.12</td>
<td>11.64</td>
<td>3267</td>
<td>78.41</td>
<td>886.2</td>
</tr>
<tr>
<td>59R2</td>
<td>58.14</td>
<td>117.2</td>
<td>74.07</td>
<td>11.48</td>
<td>3211</td>
<td>77.06</td>
<td>853.0</td>
</tr>
<tr>
<td>60R1</td>
<td>60.59</td>
<td>122.2</td>
<td>77.19</td>
<td>11.96</td>
<td>3353</td>
<td>80.47</td>
<td>928.6</td>
</tr>
<tr>
<td>60R2</td>
<td>59.75</td>
<td>120.5</td>
<td>76.11</td>
<td>11.80</td>
<td>3298</td>
<td>79.15</td>
<td>920.1</td>
</tr>
<tr>
<td>62R1</td>
<td>62.37</td>
<td>125.8</td>
<td>79.45</td>
<td>12.31</td>
<td>3535</td>
<td>84.84</td>
<td>1042.0</td>
</tr>
<tr>
<td>67R1</td>
<td>66.76</td>
<td>134.6</td>
<td>85.04</td>
<td>13.18</td>
<td>3554</td>
<td>85.30</td>
<td>1250.0</td>
</tr>
<tr>
<td>49E1A</td>
<td>63.14</td>
<td>127.3</td>
<td>80.43</td>
<td>12.47</td>
<td>1098</td>
<td>26.35</td>
<td>681.9</td>
</tr>
<tr>
<td>61C1</td>
<td>60.79</td>
<td>122.6</td>
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<td>12.00</td>
<td>2631</td>
<td>63.14</td>
<td>834.0</td>
</tr>
<tr>
<td>75C1</td>
<td>75.23</td>
<td>151.7</td>
<td>95.84</td>
<td>14.86</td>
<td>3596</td>
<td>86.30</td>
<td>967.5</td>
</tr>
<tr>
<td>76C1</td>
<td>72.73</td>
<td>146.6</td>
<td>92.65</td>
<td>14.36</td>
<td>3949</td>
<td>94.78</td>
<td>1049.0</td>
</tr>
<tr>
<td>33C1</td>
<td>32.99</td>
<td>66.52</td>
<td>42.02</td>
<td>6.51</td>
<td>297</td>
<td>7.13</td>
<td>218.8</td>
</tr>
<tr>
<td>Fz 36</td>
<td>33.99</td>
<td>68.53</td>
<td>46.8</td>
<td>7.25</td>
<td>933.7</td>
<td>22.41</td>
<td>1190.0</td>
</tr>
<tr>
<td>GGR-118</td>
<td>58.3</td>
<td>117.6</td>
<td>74.3</td>
<td>11.52</td>
<td>2640</td>
<td>63.43</td>
<td>777.0</td>
</tr>
</tbody>
</table>

Note 1: values are valid to only three significant figures at this writing; they should be verified prior to performing stress calculations and writing firm procurement or construction specifications

Note 2: Some sections show the Moment of Inertia to the Iₓₓ axis (the base), not the Iₙ Neutral Axis; see appropriate producer’s drawing to verify the geometric properties of the section/profile of interest

8.4.2.5 Manufacturing Methods, Tolerances and Testing

All European specifications for grooved rails and construction rails require the use of steel produced by the continuous casting process, with vacuum-degassed steel specified for rails to be head-hardened; however, there are some substantial differences in the philosophy behind the specifications, viz

1. Manufacture:
   a. EN 14811 is performance-based, rather than prescriptive, wherever possible
   b. The six grades of non-alloyed rail steels are classified by hardness, not tensile strength; three grades are as-rolled, three grades are heat-treated
   c. The hardnesses specified range from 200-240 HBN to 340-390 HB
   d. The allowable mass of included hydrogen is specified for each grade in PPM, and is controlled by testing the blooms
   e. Alloyed rails are covered by agreement between customer and producer
   f. EN 14811 references other CEN standards to specify steel grade nomenclature, and tensile and hardness testing
   g. Quality management is based on the producer adhering to the requirements of EN ISO 9001
2. Tolerances:
   a. Rails are produced to two different tolerance levels, analogous to railroad vs industrial quality
   b. Many more measurement points on the profile are required in EN 14811 than prior standards
   c. The profile and straightness tolerances are generally greater in EN 14811 than in AREMA Chapter 4, Table 4-2-2 (ie: in EN 14811, height of rail ±0.059-in [±1.5mm] vs Chap. 4 + 0.030-in. [0.76mm]- 0.015-in [0.38mm] based on the premise that the traffic is relatively low speed
   d. Construction rails used in making special trackwork have tighter tolerances than running rails
   e. Both minor upsweep and downsweep are acceptable
   f. Rail length tolerance is much tighter than Chap. 4

3. Testing:
   a. Testing procedures are generally similar to AREMA practice
   b. For the as-rolled profiles, hardness testing is required on the running surface only; for heat-treated, both running surface and internal hardness testing is required
   c. Purpose-designed gauges are used for profile checking
   d. No tests are specified to determine residual stresses

8.4.2.6 Additional Considerations for Grooved Rail Selection
   a. The selection criteria listed in Section 8.4.1.1 are equally applicable to grooved rails of non-domestic manufacture,
   b. Investigate the popularity of a candidate profile/section regarding how often it is rolled, by how many producers, etc., as this has important implications regarding long-term availability and cost
   c. Determine the chemical composition and hardness of a candidate section to make sure that welding will not be difficult or require special procedures, such as post-hardening; if special procedures are required, make sure they are covered in the construction specifications
   d. Obtain proper handling information from the producer regarding slinging long rails with spreaders and put this information in the specs
   e. Determine compatibility of candidate section with the wheel profile(s) to be used; note, for instance, that 59R1 and 59R2 have different gauge corner radii and slightly different groove widths, important considerations in sharp curves
   f. Determine that shipment of a candidate section will be done so as to protect the rails from salt-spray corrosion during transit, and that an appropriate spreader is available to unload the rails without damage

8.4.2.7 Special Considerations Regarding Handling, Welding, Laying and De-stressing Rails
   a. The recommendations in Chapters 4 and 5 should be followed faithfully, plus some special considerations listed below
   b. Handling: special care should be taken when lifting or moving grooved rails, as the thin web and base flanges make it easy to cripple the base or web if the rails are overbent in handling (see 8.4.2.6.c and 8.4.2.7.d), or to twist it beyond the yield point
   c. Welding: care should be exercised in both flash-butt and thermite welding to make sure the web and base are not overheated, or base droop and/or web curling may occur
   d. Laying: welded strings should not be dragged around sharp corners or otherwise mishandled as noted in 8.4.2.7.b to prevent kinking or twisting the rails
   e. De-stressing: there is no common agreement at this time whether embedded rails need de-stressing in the conventional sense specified for open track, as sun-kinks are not likely; however, it is prudent to lay the rails at something near the average ambient temperature to reduce any tendency to have pull-aparts. This practice is also recommended for all running rails in embedded tracks.
APPENDIX A

Commentary on Analysis of Lateral Acceleration and Jerk Rate for Establishing Superelevation and Spiral Length

Introduction

The US rail industry standard for lateral acceleration and jerk respectively for a long time has been 0.1g and 0.03g/s. The standard used by railroads and transit properties in the US is based on research conducted 50 years ago and was applicable to all types of cars including dining cars where a smooth ride was essential. Today, several European countries allow higher rates. SNCF (French National Railways) uses 0.15g for lateral acceleration and 0.1g/s for jerk for its railroads including the TGV system. Some higher values for jerk rate have been suggested by research on high speed rides but do not seem to have been put into practice. Subjective experiments of ride comfort on curves were judged as "noticeable lateral acceleration" at 0.1g and "strongly noticeable but not uncomfortable" at 0.15g. For short LRT rides, strongly noticeable lateral acceleration now and then would seem to be an acceptable ride condition. While the data is less conclusive for jerk, several studies support a higher rate with some research suggesting it is not a factor in ride comfort at all. It therefore seems reasonable to consider a somewhat higher jerk rate as well.

Increasing maximum allowable lateral acceleration equals use of a higher limit for unbalanced superelevation (cant deficiency) on curves and correspondingly higher speeds regardless of actual superelevation.

Jerk rate is one of three parameters (jerk, twist, and rate of twist) used to establish minimum spiral length. Allowing a maximum higher jerk rate will allow shorter spirals. In unsuperelevated curves common to embedded track, jerk is the only parameter used to determine spiral length.

Various researchers from Hirshfeld (1932) and Code (1955) to more recent studies for high speed rail travel in the US, France, Germany and Japan (1989 to 2004) have examined ride comfort versus unbalanced superelevation on curves and jerk rates for spirals. The results of those studies produced recommended rates that range from less than 0.1 g to 0.16g for lateral acceleration. For jerk rate, the studies recommendations range from 0.03 g/s to 0.25g/s with additional other limitations for the higher jerk rates. Analyses of ride comfort relative automobile and airplane performance under situations somewhat analogous to railroad curving have been made. Analyses of ride comfort versus vibration levels and uneven ride conditions (lateral jolt due to track irregularities) have also been made and comparisons made to ride comfort on railroad curves. The overall conclusion of these studies is that severe jolts and long term vibrations have more to do with rider comfort than reasonable lateral acceleration levels and spiral jerk rates. Safety (rather than comfort) limits were examined in one report which suggest that as jolt rates (and spiral jerk rate) increase, the lateral acceleration must be decreased so that the two in combination do not produce an unsafe ride. Unsafe meaning some standing riders would lose their footing.

Ride comfort is a subjective parameter, and while for the sake of analysis, it is equated with precise values of acceleration (g) it is not really a precise parameter. Ride comfort is affected by the vehicle characteristics as well as the track design. Vehicle characteristics vary significantly from one design to another. Code used a wide variety of passenger cars in his ride comfort studies and in the end, simplified the varying performance of the cars into just two classes, those with loose suspensions and
those with stabilized suspensions. These two factors – the subjective nature of ride comfort evaluation and the variability of the cars to affect ride comfort - make research to establish values for all systems problematic. A better approach is to evaluate ride comfort for a given system by operating its vehicles at varying speeds around a number of curves to decide, for the specific system what constitutes a comfortable ride.

**Lateral Acceleration Discussion**

Ride comfort on the body of a curve is determined from a combination of vehicle roll and unbalance of the curve. A 0.1g value is equivalent to 6 inches total unbalance. For a loosely sprung vehicle, up to 3 of those 6 inches is consumed by vehicle roll leaving 3 inches as a maximum design value for alignment criteria. For more stable cars (ie those with suspensions that limit roll to 1.5 degrees or less per AREMA Chapter 5 test procedure), the E\(_u\) max for design rises to 4.5 inches since the vehicle roll uses 1.5 inches or less of the combined total of 6 inches allowable unbalance.

SNCF uses 0.15g for lateral acceleration. This has been a suggested acceptable level by others in the US but does not appear to have been implemented elsewhere. SNCF also commits to maintaining track alignment to limit lateral jolts due to misalignment to less than 0.025g/s. An FRA sponsored 1991 ride safety (not comfort) study indicates that it is safe to operate at speeds equal to 0.15g lateral acceleration if track alignment is well maintained so as not to introduce excessive jolts due to misalignment into the ride. It concluded up to 0.183 g/s jolt with 0.15g lateral acceleration as safe. The safety study was based on analysis of ride quality on many curves at various speeds. Recent TTCI (2008) research demonstrated that wheel climb derailment potential is virtually unaffected by unbalance whereas lower rail angle and track perturbations are the principle causes of wheel climb.

Using the higher 0.15g value for lateral acceleration allows increasing the total unbalance from 6 to 9 inches. This allows a 3 inch E\(_u\) design limit for loosely sprung vehicles to rise to 6 inches E\(_u\) and a 4.5 inch E\(_u\) design limit for stabilized suspension vehicles to rise to 7.5 inches E\(_u\). For an 82 ft radius unsuperelevated curve (typical embedded track street corner turn and loop situations) the 3 to 6 inch E\(_u\) increase, when using the standard formula E = 0.0007V\(^2\)D to compute velocity (V), improves the speed from 8 to 11 mph (approx) and a 4.5 to 7.5 inch E\(_u\) increase improves the speed from 10 to 12mph.

Use of E\(_u\) for the E value on unsuperelevated curves without adding the unbalance component due to car roll results in a conservative (lower) speed and lower g value being computed when using the standard formula E = 0.0007V\(^2\)D to compute velocity (V). See note at end of this section on D vs R. This is regardless of combination of E\(_a\) plus E\(_r\) generally considered to be the total E value. The design formula should include an E\(_r\) (for roll) value when computing V. Since less than the full E value (E\(_a\) + E\(_u\) + E\(_r\)) is used in the current commonly used design formula, the lateral acceleration for the design speed is below 0.1g. Another way to look at this is that it is safe to increase the running speed of a curve without exceeding the ride comfort for which it was designed although the faster one goes the more uncomfortable the ride becomes. By adding the E\(_r\) value, the design formula becomes E\(_a\)+E\(_u\)+E\(_r\) = 0.0007V\(^2\)D. The inclusion of 1.5 inches for a stabilized vehicle for E\(_r\) in the formula further improves the design allowable speed for an 82ft unsuperelevated curve to 14 mph from 12 mph.

The matter of whether or not increasing allowable lateral acceleration increases the risk of wheel climb derailment or overturning should be considered. Even with shallow wheel flanges, wheel climb is caused by wheel/rail angle, angle of attack, and suspension stiffness. Lateral acceleration due to
speed (unbalance) if increased indefinitely leads to vehicle overturning not wheel climb. This is so because the higher lateral force on the wheel due to higher lateral acceleration is offset by more of the vehicle weight transferring to the vertical component on the wheel. TCRP Report No. 57 has formulae for analyzing overturning which may be used for comparison with the proposed lateral acceleration/Eu values. Using the TCRP formula, the overturning speed for an 82ft radius, unsuperelevated curve is 26 mph or about twice the proposed operating speed of 14mph. The safe speed is defined in TCRP Report No. 57 as the speed at which the vehicle becomes unstable and in danger of derailment upon introduction of any anomaly in the track. The maximum safe Eu value, using the TCRP formula for safe speed is 9.6 inches Eu which is greater than the proposed maximum of 9 inches. Embedded track, once built to accurate alignment, should retain the accuracy of that alignment indefinitely.

In summary, increasing the allowable lateral acceleration from 0.1g to 0.15g and adding 1.5 inches for well suspended vehicles for the Eu value to the formula for computing speed will result in an allowable safe increase in speed from 10 to 14 mph for an 82 ft radius unsuperelevated curve. Based on observations by trackwork engineers riding on various LRT systems, this modest adjustment to the design criteria will do no more than reflect actual operating conditions on systems where operators frequently increase speed before a train has cleared a curve.

(Note: The standard formula \( E = 0.0007 V^2 D \) uses D based on \( D = 5730/R \). This formula was derived when curves were surveyed with transit and tape methods and defining a curve by “Degree of Curve” was useful in the field for staking curves. As noted in surveying texts this method of staking a curve becomes progressively more inaccurate as radius of curve decreases. It is accurate, however for converting \( R \) to \( D \) for use in the above formula for computing speed (V) or total superelevation (E) even at the small radii anticipated for LRVs and trolley cars. In other words, \( D \) should not be used to “define” the radius of a curve of less than 300 feet but may be used to convert \( R \) to \( D \) in the above formula.)

**Jerk Rate Discussion**

The jerk rate establishes the time needed to introduce the lateral acceleration or unbalance of a curve at the beginning and end of a circular curve. A constantly increasing amount of lateral acceleration beginning at zero and ending at the desired lateral acceleration value for a curve is achieved through the passage, at a constant speed, of a vehicle traveling along a constantly increasing degree of curvature, ie a spiral.

The length of the spiral determines the time required to go from zero lateral acceleration to the level of lateral acceleration of the circular curve. It has been demonstrated that the amount of lateral acceleration (\( E_u \)) is more important to ride comfort than the rate at which it is introduced (spiral length). Never the less, an unreasonably high rate of introduction of lateral acceleration (jerk rate) is undesirable, especially for high levels of \( E_u \). If no spirals are used, the jerk rate is theoretically infinite. In reality, the play between the wheels and track gage along with dynamic response of the vehicle reduces this infinite rate to a jerk rate that is measurable though high.

The current US standard of 0.1g lateral acceleration, coupled with an 0.03g/s jerk rate dictates the introduction of \( E_u \) over 3.33 seconds. This, for a typical maximum \( E_u \) of 4.5 inches transforms into the familiar formula for determining spiral length: \( L_s = 1.09VE_u \).
In the US, just as with lateral acceleration, a conservative low jerk rate of 0.03 g/s was adopted as standard. However, numerous studies, beginning with Hirshfeld, concluded that higher jerk rates were acceptable with respect to ride comfort. The FRA Ride Safety Study of 1989 concluded that jerk rate was not significant to ride comfort and that rates (either jolt or jerk) as high as 0.183 g/s were safe for lateral acceleration values up to 0.15g. The 1978 NEC study of ride comfort endorsed the SNCF’s values of 0.15g and 0.10 g/s with a limit on jolt of 0.025 g/s. The 2004 FRA Study for high speed rail between Richmond and Charlotte endorsed the same SNCF values and noted that a jerk rate as high as 0.25 g/s would be acceptable so long as no track irregularities were to occur that would momentarily raise the jerk rate to a higher level.

The conclusion from these studies is that a jerk rate of 0.10 g/s would not produce an unacceptable ride on embedded track which, once properly constructed to a smooth alignment, would preclude any unusual jolt values from occurring. In fact, a jerk rate of 0.10 g/s is conservative compared with some recommendations.

For a lateral acceleration maximum of 0.15g, a jerk rate of 0.1g/s means the spirals will be long enough to introduce the Eu over 1.5 seconds. The spiral formula for a 0.15 g lateral acceleration (7.5 inches Eu) and 0.10 g/s jerk rate becomes \( L_s = 0.29VE_u \).

To put this in perspective an 82 ft radius unsuperelevated curve used by a stabilized suspension vehicle could have the following designs:

a. Existing standard of 0.1g and 0.03g/s: \( E = 4.5 \) inches with roll ignored.
   \[ E = 0.0007 V^2 D \quad V = 9.6 \text{ mph} \quad L_s = 1.09VE_u \quad L_s = 47.1 \text{ ft} \]

b. Proposed rates of 0.15g and 0.10g/s Total E, including roll, of 9 inches, design Eu of 7.5 inches.
   \[ E = 0.0007 V^2 D \quad V = 13.6 \text{ mph} \quad L_s = 0.29VE_u \quad L_s = 29.6 \text{ ft}. \]

It should also be noted that the lateral offset of the unspiraled versus spiraled curve is 1.12 ft for a. and 0.46 ft for b.

**Conclusions**

Based on the foregoing analysis of research conducted by others plus observations of actual operating conditions, the following is recommended.

1. That the allowable lateral acceleration of 0.1g be increased to 0.15g which corresponds to an increase of 3 inches of allowable unbalance for a total Eu of 9 inches (7.5 inches for design plus 1.5 inches for roll).
2. That the E value used in the formula \( E = 0.0007 V^2 D \) include the allowance for vehicle roll, which for modern LRVs would be 1.5 inches along with the new design value for Eu of 7.5 inches. These first two recommendations will result in higher allowable speeds on curves.
3. That the allowable jerk rate for spiral design be increased from 0.03 g/s to 0.10 g/s. The formula for computing spirals will be \( L_s = 0.29 V E_u \) for a maximum of 7.5 inches Eu. This will result in shorter spirals with correspondingly smaller offsets.
4. Application of these higher values for lateral acceleration and jerk imply a commitment to high quality construction and maintenance of track alignment. They should be used only where allowed by field conditions.
5. These recommendations apply only to standard gauge embedded track.