High frequency vehicle-track interaction simulation

COUPLING OF AN ADVANCED TRACK MODEL WITH A MULTI-BODY SYSTEM

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Abstract

The forces caused by the high frequency vehicle-track interaction have a great impact on the track maintenance. They should be represented in vehicle-track models in order to predict their impact. As a result, a multi-body system (MBS) should be extended with an advanced track model. The MBS represents the vehicle and the wheel-rail contact with a great accuracy. The track will be modeled in two different ways: a moving track model and a continuous track model which is a finite element modeling (FEM). The first one will be a lumped-mass model. The most advanced system will be the second one, which is a MBS-FEM representation and offers a great precision to represent the high frequency dynamical properties. The system will be used to simulate pertinent phenomena such as a wheelflats, corrugations and rail joint. Based on the literature and the measurements, the model is validated in a wider frequency range than the one currently used (0-20Hz). The results given by both models are close to the literature and the measurement.
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1 Introduction

With the flourishing of new high-speed lines (HSL) and heavier freight traffic, the infrastructure is exposed to high charges. This is provoking higher maintenance costs due to more defaults on the track. In order to predict and prevent the creation of defects, new tools are required.

It’s one of the objectives of the vehicle-track interaction division which is part of the infrastructure branch of SNCF (French railway operator). The division works on many subjects such as the analysis of the cause of derailment or the identification of the hazardous vehicle-track geometry combinations.

Parts of the work are carried out with the use of simulation. This allows to obtain forces and accelerations without doing expensive measurements. The results will allow to predict the appearance of defect and the effect on the vehicle. However, some phenomena excite the system at frequencies higher than the one allowed with the current tool at SNCF. These irregularities cause high dynamical forces that have an influence on the vehicle behavior and the maintenance cost such as the deformation by an impact. These forces and accelerations cannot be modeled with the current tool.

The issue had lead to important research in the literature on pertinent phenomena such as wheelflats or corrugation. One of the limitations is the track used in the multi-body software (MBS) which is too simple. To solve the problem, different types of track models were developed. At the beginning, the models used were limited due to the massive computation time required. In recent years, the increase of computer power has lead to the development of more precise track models. Such improvements are not implemented at SNCF. Other ameliorations are necessary in order to model the specific phenomena with the MBS code, in the case of Vampire.

In order to enhance the modeling, studies can be carried out aiming to detect the limitation of the modeling of the vehicle-track interaction dynamical behavior. The track model will be the main part of work and implemented in a commercial MBS, Vampire. At the same time, the thesis will answer the following questions:

- What is the dynamical behavior of the railway track?
- How do the track and the vehicle respond to a high frequency excitation?

The outcome will permit to model the dynamic contact forces in different situations that will imply higher frequency excitation. It will result in a better understanding of the dynamical behavior of the track allowing to improve the comfort and the maintenance cost of the vehicle and the track.

The thesis will be decomposed in several chapters. In chapter 2, a description of the vehicle-track system is given. It will help to understand the complexity of the system. In chapter 3, the modeling of the system will be discussed. The model presented in the literature will be briefly described. In chapter 4, the methodology will be shown. The coupling and the
model will be explained. This will lead to chapter 5 with the real world application for the validation of the model. In the conclusion, a discussion around both models will be given.
2 The vehicle-track system description

The vehicle-track interaction is a complex system. It is commonly decomposed in 3 subsystems: the vehicle, the track and the wheel-rail contact that realizes the coupling between the two other subsystems as shown Figure 1.

The first subsystem is the vehicle. It is composed of 3 main parts as presented in figure 1a. The carbody carries the payload. The bogie is linked to the carbody through the secondary suspension. The wheelset is maintained by the bogie through the primary suspension. The wheelset is an unsprung mass, because the vibration induced by the rail aren’t filtered by a suspension, unlike the carbody and the bogie which are considered as the sprung masses.

![Figure 1: Description of the vehicle & the track](image)

The track is the second subsystem represented in figure 1b. The contact between the vehicle and the track is located on the rails which support the weight of the vehicle and give the direction. The rails are attached to the sleeper through the fastening and railpad. The railpad helps filter some vibration from the vehicle and keeps the rail attached to the sleeper. The sleeper is resting on the ballast which is composed of coarse stone that holds the sleeper in position. The subballast is formed by thinner particles. It is a transition layer. These both layers help transmitting the load from the sleeper to the bedrock.

These two subsystems are linked through the contact between the rail and the wheelset. This inter-connexion is the heart of the rail modeling. It’s where the excitation and the specific phenomena take place. They will be discussed at the end of this chapter. The behavior of the two subsystems will be studied in the beginning.

2.1 The vehicle behavior

The irregularity from the rail-wheel contact excites the vehicle in a wide spectrum from 0-500 Hz as see K.Popp et al [15]. In this article, the authors are doing a literature review on wear modelling in the mid-frequency range. The excitation causes dynamic reaction of the different elements due to the wider spectrum needed to model the phenomena. The suspension will be studied first and the flexible behavior of the element will be discussed in a second step.
2.1.1 Suspension characteristic & unsprung mass

The excitation is transmitted through the contact between the wheel and the rail. It will directly affect the wheelset. From the wheelset, the forces are transmitted to the other elements of the vehicle through the primary and secondary suspension.

These two suspensions levels are designed to satisfy safety and comfort criteria. Their behavior can be considered as a low-pass filter. The primary suspension has a cut-off frequency of 9-10 Hz. The secondary suspension has a cut-off frequency of 2-4 Hz.

The high frequency excitation is filtered. One of the reasons is the low eigenfrequencies of the carbody as shown in table 1. If an eigenmode of the carbody is reached the comfort will be reduced. For this reason, the flexible modes of the sprung mass (carbody and the bogie) are not studied.

Table 1: Carbody Eigenfrequency [6]

<table>
<thead>
<tr>
<th>Mode</th>
<th>Eigenfrequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vertical bending</td>
<td>10.9 Hz</td>
</tr>
<tr>
<td>Torsional bending</td>
<td>11.3 Hz</td>
</tr>
<tr>
<td>Lateral bending</td>
<td>11.9 Hz</td>
</tr>
</tbody>
</table>

However, the wheelset is an unsprung mass. The excitation is directly impacting the wheel. The wide spectrum might reach a wheelset eigenfrequency and should be studied as proposed by Knothe & Grassie [14] where the authors look at the literature for wear simulation until 1500 Hz.

2.1.2 Wheelset structural flexibility

The behavior of the wheelset is important. In this section, the high frequency behavior of the wheelset will be discussed.

Figure 2: 3-D FE model of the wheelset [8]
As shown in figure 2, the eigenfrequencies are located in the studied spectrum. However, the result is for one type of wheelset. The shape of the wheelset (number of braking disc, the size of axle) will change the eigenfrequencies. But, they tend to be in the same interval as shown in table 2. The table reflects that the simulation and the experiments give quite close results. Differences are observed when the axles are made more rigid for example if there are brake disc like the ICE wheelset. In the frequency range chosen, the wheelset should be taken as a flexible body. But the bogie and the carbody can be considered as rigid body.

Table 2: Wheelset eigenfrequencies

<table>
<thead>
<tr>
<th>Wheelset type</th>
<th>First Torsion</th>
<th>First bending</th>
<th>Second bending</th>
<th>Ref</th>
</tr>
</thead>
<tbody>
<tr>
<td>Simulated by FE</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fingberg model</td>
<td>67 Hz</td>
<td>89 Hz</td>
<td>141 Hz</td>
<td>[8]</td>
</tr>
<tr>
<td>ICE, 4 axle</td>
<td>82 Hz</td>
<td>85 Hz</td>
<td>132 Hz</td>
<td>[18]</td>
</tr>
</tbody>
</table>

Measured in free boundary conditions

| ICE, 4 axle-mounted discs      | 96 Hz        | 82 Hz        | 162 Hz         | [2] |
| ETR500, no brake discs         | –            | 79 Hz        | 130 Hz         | [5] |

2.2 The track behavior

The rolling stock is moving on the track. The most common one in France is the ballasted track. It’s composed of elements with non-linear behaviour. This part will look at the behaviour of the track system.

2.2.1 Dynamical behavior of the railway track

At frequencies higher than 20 Hz, the track does not behave linearly and starts to be strongly non-linear due to the eigenmodes of the track elements. It is shown in figure 3 that represents the receptance which is the displacement of the track as a function of the force (K.Dahlberg [1]).

Figure 3 demonstrates the importance of considering the track behavior at high frequencies. The eigenmodes of the track should thus be taken into account. At the same time, the track can be considered linear for a limited spectrum.

In table 3, the track eigenfrequencies are listed. The first one is not represented in figure 3 because it appears in areas with soft subgrade such as clay. At the same time, the table 3 shows the importance of considering the ballast at frequencies lower than 300 Hz. The railpad plays an important role in the medium frequency range (50-500 Hz) as outlined in [15].

This shows that all track elements should be represented for a frequency range of 0-500 Hz. At the same time, the excitation coming from the studied spectrum might reach an eigenfrequency.
The sleeper should also be considered as a flexible body according to S.L.Grassie [9]. The author studied the eigenfrequencies of different sleepers in England, Germany, and Sweden. The natural frequency can have an impact on the maintenance cost, because the oscillation induced by the eigenmode will stress the element and decrease the life span. At the same time, the eigenmode will greatly reduce the lifespan of the element.
Furthermore, if we want to consider a frequency range up to 500 Hz, the different elements should be considered as flexible bodies.

### 2.3 The vehicle-track interaction phenomena

The two subsystems are linked through the rail-wheel contact. The connection transmits the vibration from one subsystem to another. The inter-connection is complex. Kalker has extensively studied this in [13]. This interface is the heart of the railway simulation and it’s where most of the excitation happens.

The interface is the location of most excitation. The origin can be from the vehicle or from the track. The dynamic interaction can be separated in three frequencies ranges. In these intervals, phenomena can be observed in each frequency interval which are given in the table 5 from Knothe & Ripke[7], Grassie & Knothe[14] & K.Popp[15]. Two families of phenomena can be observed: short term and long term. The long term phenomenon is mainly wear, which isn’t considered in the study. The short term are all the phenomena that appear quickly like a wheelflat.

#### Table 5: Frequency and Phenomena

<table>
<thead>
<tr>
<th>Freq (Hz)</th>
<th>0-20 Hz</th>
<th>0-1500 Hz</th>
<th>0-5000 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Phenomena</td>
<td>Derailment (Y/Q)</td>
<td>Out-of-Roundness</td>
<td>Noise</td>
</tr>
<tr>
<td></td>
<td>Comfort</td>
<td>Critical Speed</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Wear</td>
<td>Wear</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>Rail corrugation</td>
<td></td>
</tr>
</tbody>
</table>
As represented in table 5, the work will focus on the frequency range up to 1500 Hz. At the same time, frequencies up to 500 Hz are the most interesting as said by K.Popp[15]. This range contains 3 important phenomena which will be studied: the railhead corrugation, the rail joint and the wheelflat.

2.3.1 Railhead Corrugation

Corrugation is an irregularity that appears on the rail head with a short wavelength. This deflect is part of the short deflects that possess a classification depending on the defects wavelengths. One given by Alias [10] is based on the following division: corrugation, wavelengths 30-100 mm; short waves with wavelengths 100-300 mm and longer waves between 300 mm to 1 m.

It’s important to note that these irregularities cannot be measured by conventional measurement vehicles, because their lower wavelength measurement limit is usually 3 m. In order, to detect this phenomenon, a special tools needs to be used such as a trolley.

As shown in figure 5, the rail tread is damaged which causes noise and higher dynamic forces that can likely cause a sleeper resonance and damage the track. The load cycle might shorten the life span of the track elements too.

The origin of the corrugation is not clearly known. The book of T. Dahlberg [1] gives several suggestions and the origin might be a combination of these phenomena. Ripke and Knothe [3] stated that one cause can be the eigenfrequencies of the track. Alias [10] proposed that the cause can be the local formation of martensite due to slip of the wheel.

2.3.2 Joints

To link two rails with each other, a fish-plate is used. However, a gap exists between the rail. This is useful to allow for thermal dilatation during hot summer. Meanwhile, the gap
can have different shapes as summarized in table 6. They will impact the vertical and lateral behavior of the vehicle.

Table 6: Different type of Rail Joint

<table>
<thead>
<tr>
<th>Type of joint</th>
<th>Schema</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elevated Joint</td>
<td><img src="image" alt="Elevated Joint Schema" /></td>
</tr>
<tr>
<td>Shift Joint</td>
<td><img src="image" alt="Shift Joint Schema" /></td>
</tr>
<tr>
<td>Low Joint</td>
<td><img src="image" alt="Low Joint Schema" /></td>
</tr>
</tbody>
</table>

However, the gap causes an impact when a wheel passes. This shock will induce overload on the track. It will deteriorate the rail and the track. At the same time, the forces caused by the rail might modify the gap and increase the forces at the next train passage. As a result, the track requires more maintenance around this area.

Meanwhile, the new railway tends to use long welded rails. They do not use a rail joint. However, the joints are still present on old track and increase the maintenance on this area. They should be simulated to know the impact on the traffic comfort and the maintenance.

2.3.3 Wheelflat & Out of roundness

The wheel might start to slip during acceleration or stay blocked during braking. The slip will remove the wheel tread and causes an out-of-roundness or a flat as shown in figure 6. The phenomenon will introduce a high dynamic impact at each turn of the wheel.

The wheelflat tends to be replaced by an out of roundness because the sharp edge of the flat is removed by plastic deformation due to high stresses in the area. This causes rail deformation that can have significant impact on maintenance.

All these phenomena will cause high dynamic forces that can have an impact on the vehicle behavior and the maintenance cost such as the deformation by an impact. These forces and
accelerations must be modeled in order to predict and prevent new defaults. This is the reason for the development of a modeling tool.
3 Modeling of the dynamic coupling between the vehicle and the track

The modeling of the wheel-rail interaction has been the issue of a lot of research. Meanwhile, it’s the track model which represents a great limitation in today’s MBS simulations. The track model had been evolving since 1867 (creation of the winkler model [1]). At the beginning, the modeling was simple due to the limited computation power. Nowadays, the skyrocketing computer power has authorized the use of more sophisticated models. During the work, the vertical behaviour will be studied.

3.1 Type of railway track models

The literature offers lot of models. Two types of model can be discerned according to N.Chaar [4].

3.1.1 Moving track model

The model considers the track moving with the vehicle. There will be one model per wheelset. This type of model is the one used in most MBS software such as Vampire, because it allows fast computation.

![Moving Track example](image)

Figure 7: Moving Track example [12]

It can be at different levels of precision. Figure 7 represents the one used by Vampire. The software uses a linear spring with no mass. But, some models can be more sophisticated.

However, the model has great limitations in the term of representation, because only lateral and vertical behavior are considered. The frequency range is limited to 150 Hz [4].

3.1.2 Continuous track model

The limitation of the previous model can be eliminated with a continuous track model. The model is fixed and the vehicle moves on top of it. It allows the user to consider forces in all directions and wave propagation in the rail. It’s the most common type of track modeling in the literature.

However, the continuous model possesses a great drawback due the computational time. To limit the numerical requirements, two types of rail supports have been developed.
Continuous support model: Winkler model  The model is represented by a rail resting on a continuous support. Figure 8 illustrates the model developed by Winkler in 1867 [1]. It allows quick calculation. However, this model doesn’t allow high frequency studies due to the lack of elements such as the rail pad or the sleeper. Further, the Euler beam doesn’t consider shear deformation which is important at high frequency.

![Winkler model](image1)

Figure 8: Winkler model [1]

Nowadays, more advanced continuous track model have been developed. They can consider the sleeper or the shear deformation of the rail. But, they are limited due to the absence of the space between sleeper. Their advantage is the lower computation time compared with the discrete support model.

Discrete support model  To eliminate the drawback of the previous model, the recent years have seen another type of model emerged: the discrete support model. It considers the gap between the sleepers.

![Rail on discrete support](image2)

Figure 9: Rail on discrete support[1]

But, this model possesses a great drawback that is high computation time. The literature gives some good example in Bruni et al [7] where the authors observe a CPU time 33 times higher for the continuous track model than with a moving track model.

3.1.3 Comparison between moving track model and continuous model

As said in the previous section, both types of models have their advantages and inconveniences which are listed in table 7.

A great difference can be observed in table 7 like the wave propagation that causes frequency range limits for the moving track model. The possibilities of modeling a discrete defect
such as a loose sleeper are a great advantage. The validity range allowed by the continuous model permits to analyze the phenomena more accurately. However, all the possibilities are at the cost of an increased computation time. In order to compute long distance, it can be useful to be able to compute in a limited time. Based on this analysis, the thesis will discuss both types of model.

3.2 Implemented track model

Both models will be represented: the moving track model and the continuous track model with a discrete support.

3.2.1 Moving Track Model

The author has designed a moving track model as represented in figure 10. It allows to consider the railpad, ballast and subgrade dynamical behavior. The validity of the model is limited to 200 Hz.

Figure 10: Moving track model

The rail is considered as rigid, because the deformation of the element is not taken into account. At the same time, the sleeper should be considered as flexible, because the first
eigenmode can be in the validity range of the model. But, it would be an advantage to model the sleeper as rigid, in order to gain computation power.

![Figure 11: Sleeper displacement at 153Hz](image)

After a modal analysis of a sleeper resting on a continuous elastic support, the result shows that the sleeper has a vertical bending mode at 153 Hz. This eigenfrequency can have an influence on our model. However, the displacement of the point where the railpad is located, is negligible as shown in figure 11. So the sleeper can be modeled by a rigid body for frequencies up to 200Hz.

### 3.2.2 Continuous Track Model

The second model is a continuous track model. The main issue of the element is computation time. To reduce it, the model is not as complex as the moving track model and is the same as figure 9. The bedrock will be considered rigid. The support is discrete to allow for simulation up to 2000 Hz.

Each element is designed as followed:

- **Rail**: It’s represented as Timoshenko beam with the mechanical characteristic of the steel alloy used for the rail
- **Railpad**: It’s modeled as a linear spring with the same stiffness as previously
- **Sleeper**: It’s an Euler beam with the concrete mechanical characteristic
- **Ballast**: It’s a linear spring with the same stiffness as previously

The model obtained in ANSYS is shown in figure 12. The model is finite. In order to consider the system infinite, the boundary conditions (displacement) of each end of the same rail will be identical. This will eliminated side effect. The displacement will be identical at each end of the same rail. At the same time, the FE model will be twice as long as the distance between the wheelsets at each end of the vehicle. The length had been chosen to be 18
long enough not to have wave reflection. But, a study of the length should be carried to limit computation power.

3.3 Parameterization of the track model

The models proposed previously require a definition of the parameters. However, some assumptions will be done in order to simplify the model.

3.3.1 Railway Track variation

The implementation of the track will first be linked to the parameters of the different elements. The track behavior depends on the track components that can change along a line. For example, the sleeper can be made of wood, steel or concrete. The rail can be made of different alloys. The different types of sleeper will impact the static stiffness of the track as shown in figure 13. But, the figure shows that the displacements of the track with concrete sleeper are almost identical. At the same time, the new track has the same type of element for the sleeper and the rail which are well known after numerous tests done in laboratory.

Meanwhile, more complex parameters will affect the track behavior such as the age of the track. An old unmaintained track and a new track will not behave the same way. These parameters are more complex to quantify, because they depend on the traffic and the weather condition. The weather has a great impact too. As shown in figure 13, a wooden sleeper will be more affected by the weather than a concrete sleeper. The impact can fluctuate daily.

But, the new track uses only mono bloc concrete sleepers that are less affected by the weather. A sensitivity analysis of these parameter fluctuations on the dynamic behaviour of the track should be done.

Figure 12: ANSYS Track model
Track flexibility

Introduction, cont.

• The track flexibility affects the wheel-rail forces
• These forces cause the track and wheelset damage
• A stiffer track gives higher dynamic forces
• But a softer track still needs to be stable
• Special vehicles can measure the track flexibility

For dynamic simulations, various track models exist.

![Vertical displacement vs. force](image)

Figure 13: Vertical displacement of the track depending on the components and the weather [6]

All this variation should be considered if a more precise model is required. But due to a limited time, the thesis will use an arbitrary value obtained from the literature.

### 3.3.2 Linear modeling

The other aspect of the implementation is the element behavior under stress. The ballast and the railpad are viscoelastic elements. Their behaviors are very complex. The laboratory tests show a hysteresis effect in the railpad and the ballast and an non-linearity. This is shown in figure 14 from K. Popp [15]. In this figure, the behavior of the track is quasi-linear and can be considered linear.

The same analysis can be done on the ballast. From laboratory tests, the vertical behavior of the ballast is quasi-linear with a low hysteresis effect. This is not correct for the lateral and longitudinal forces. But, only the vertical effect is considered in this thesis.
Very complicated is the experimental and theoretical investigation of lateral, longitudinal and rotational pad stiffnesses. Since on the one hand the pad affects the train track dynamics strongly in a wide frequency range of about 200 Hz \% 700 Hz and on the other hand it is that component, which can be adjusted in an easy way, further research is needed.

The role of the sleeper in the short time behaviour is quite well understood. In the mid-frequency range the six rigid body motions are most important. At the eigenfrequencies, which are at about 150 Hz, 400 Hz and 800 Hz for a one-block concrete sleeper, ballast and subsoil already form a reasonably stiff support.

Experimental results are published in 18,39,80.

In contrast to the sleeper the ballast is the track component with the most unknown properties. The complicated boundary conditions at the top between sleeper and ballast and at the bottom between ballast and soil increase the difficulties to understand its influence and to investigate the physical effects experimentally. Nonlinearities are caused by gaps between sleeper and ballast and by the ballast properties themselves. The stiffness of such a granular material depends on the void ratio, on the loading velocity and on the stress state 43,44.

Inside the ballast strip this stress state is inhomogenous even on a small length-scale. The pressure is much higher for ballast stones below a sleeper than for stones filling the space between two sleepers. It is not clear what influence this variation in compression has on the wave propagation inside the ballast layer in longitudinal direction of the track. Additionally, the damping properties of the ballast, probably mostly due to dry friction, are unknown.

The behaviour of the subsoil is better understood. High damping exists due to the radiation of various kinds of waves, material damping can be neglected. Especially the surface waves lead to a sleeper coupling via the ground. For special layered configurations resonances occur in the stiffness frequency response function due to reflections of waves at layer boundaries. For a real track the characteristics of the subsoil below the moving train change. Therefore, transition radiation occurs 158,159.

All these physical effects cannot be considered by a single track model. Up to now, most models neglect all nonlinearities and assume a perfect periodicity of the track.

Figure 14: Pad dynamical behaviour[15]

4 Methodology & Coupling

The track modeling has been proposed in the previous chapter. In order to model the phenomena until 500 Hz, some improvements on the track and the vehicle modelling should be done. As said in chapter 2, the wheelset & the track should be made flexible.

The goal is to implement it in the MBS in Vampire. The replacement of the rigid wheelset by a flexible wheelset in Vampire will be discussed. Then, the coupling between the track and the vehicle will be studied.

4.1 Vehicle Modeling with Vampire

In chapter 2, the need for a flexible body in the vehicle-track interaction has been demonstrated. This part will be related to the MBS software used during this work. The software allows great precision in the contact and possess already an important library of elements. This advantage is combined with a quick numerical integration done with an Euler method.

The numerical integration is correct but requires a low time step in order to compute the correct value. This is important to notice because it will increase the computation time significantly.

However, the vehicle-rail contact possesses a great drawback. As said in chapter 2, the wheelset should be considered as flexible body. However, the wheelset cannot be modeled as a flexible body in Vampire. It’s modelling is imposed by the software. This implies a limitation in the frequency range. From chapter 2, it’s known that the torsional eigenmode at 65 Hz doesn’t affect vertical behavior. The second eigenmode at 90 Hz will significantly influence the vertical behavior. This eigenfrequency will be the upper limit of the frequency range.

4.2 Theoretical modeling

Even with this limitation, the coupling is of great interest. The main goal is to create a dynamical model that will react at each timestep depending on the contact forces.
In chapter 3, the track models were presented. But, the track and the vehicle aren’t coupled. So a link is required. The procedure is described in figure 15.

![Figure 15: Coupling between the MBS & Track model](image)

The multi-body simulation will provide the contact force between the wheel & the rail. This force will be extracted and sent to the track model. Then, the acceleration is calculated and a numerical integration is carried out in order to get the track displacement. This position will be transmitted to the Vampire model and the track position will be moved accordingly. This schema will be the same for both models.

4.3 Moving track model

The moving track model is discussed first. The coupling design will be analysed. This will be followed by the validation of the coupling.

4.3.1 Concept

The coupling between Vampire and the track model is implemented using the module called 'subroutine' which is built in the Vampire software. The language C is used.

The coupling is done as shown in figure 16. The contact forces are computed in the vehicle model with the displacement from the previous timestep. The vertical contact forces are extracted and used to calculate the track acceleration in the subroutine. Then, it's integrated...
in the subroutine by the Runke-Kutta method in order to obtain the track displacement. Finally, the subroutine informs Vampire about the position of the track at the next step.

At the same time, the Vampire track model already implemented in the model cannot be disabled. In order to reduce its influence, the stiffness of the track is set rigid to prevent unwanted track displacement. A sensitivity analysis of the influence of the Vampire track stiffness will be carried out to prevent numerical instability.

### 4.3.2 Validation

In order to use the coupling presented previously, a validation should be performed to insure the accuracy of the result. Several parameters of the coupling will be studied to see their impact as presented in table 8. Two track configurations will be analyzed:

- **Perfect Track**: a tangent track without irregularity
- **Impulsion track**: a tangent track with a peak of 10 mm at 100m as shown in figure 18.

For the study in this section, the model will be a reproduction of the track implemented in Vampire which is only a spring. The Vampire track is set to be rigid. The aim is to compare the vertical contact forces with and without the coupling. In order to validate the coupling, the result should be identical.

The vehicle used in the coupling simulation is a single wheelset with the properties resumed in table 9.
Table 8: Study of influence of parameter

<table>
<thead>
<tr>
<th></th>
<th>Prefect Track</th>
<th>Impulsion Track</th>
</tr>
</thead>
<tbody>
<tr>
<td>Timestep</td>
<td>×</td>
<td>×</td>
</tr>
<tr>
<td>Stiffness of the Vampire track</td>
<td></td>
<td>×</td>
</tr>
</tbody>
</table>

Table 9: Vehicle properties

<table>
<thead>
<tr>
<th></th>
<th>X (along rail)</th>
<th>Y (across rail)</th>
<th>Z (vertical)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bogie, mass 5000 kg</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bogie, moment of inertia (kgm²)</td>
<td>2000</td>
<td>3000</td>
<td>2000</td>
</tr>
<tr>
<td>Wheelset, mass 2000 kg</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wheelset, moment of inertia (kgm²)</td>
<td>1200</td>
<td>0</td>
<td>1200</td>
</tr>
<tr>
<td>Primary suspension, stiffness (MN/m)</td>
<td>12</td>
<td>20</td>
<td>1.2</td>
</tr>
<tr>
<td>Primary suspension, damping (kNs/m)</td>
<td>2</td>
<td>2</td>
<td>20</td>
</tr>
</tbody>
</table>

**Perfect Track**  In this part, the influence of the time step will be studied. The reference of the study will be the result without the coupling. The time steps $1e^{-3}$ cannot be used because it leads to instability. The time step $1e^{-4}$ and $1e^{-5}$ give some interesting results as shown in figure 17.

![Figure 17: Contact force depending on coupling and timestep at 5 m/s](image)

![Figure 18: Impulsion track irregularity](image)

As shown in figure 17, the mean values of the 3 curves are the same. A small sinusoidal variation in the contact force can be observed. This variation has an amplitude that increases with a smaller time step. However, the amplitude is very small and this variation can be neglected compared to a physical phenomenon.
Impulsion Track

**Time step Influence**  The impact of the time step on the simulation can be seen in figure 19a. A difference between with or without coupling can be observed. However, the timestep doesn’t influence the mean value. A remark can be made on the noise that increases as a function of the timestep as observed in figure 17. Therefore, the timestep chosen is the same as previously.

![Graph A](image1.png)  
(a) Influence of the timestep  
(b) Influence of the Vampire track stiffness

![Graph B](image2.png)

Figure 19: Influence of the coupling parameter

**Vampire Track stiffness Influence**  The Vampire track displacement should be minimized as explained before. This is done by using a stiff track. However, an infinite stiff track isn’t possible in Vampire and a very stiff track will lead to instability. So a work will be done to detect the correct value for the stiffness. The result is shown in figure 19b and the mean value and the standard deviation are resumed in table 10.

<table>
<thead>
<tr>
<th>Influence of Vampire track with coupling</th>
<th>Mean Value</th>
<th>Standard deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1000MN/m with coupling</td>
<td>6.7526 N</td>
<td>0.031842</td>
</tr>
<tr>
<td>3000MN/m with coupling</td>
<td>4.1413 N</td>
<td>0.020027</td>
</tr>
<tr>
<td>10000MN/m with coupling</td>
<td>4.037 N</td>
<td>0.018849</td>
</tr>
<tr>
<td>Vampire without coupling</td>
<td>4.625 N</td>
<td>0.021544</td>
</tr>
</tbody>
</table>

As predicted, a lower Vampire track stiffness will allow more displacement and a stiffer one will move less. The track with a Vampire track stiffness which gives the closer result to the one without the coupling is 3000$MN/m$. This value will be used.
4.4 Continuous track model

The use of a continuous track model is more complex than the moving track model coupling because of the utilization of a FE software which does not possess an easy implementation with Vampire.

4.4.1 Concept

As shown in figure 20, the procedure is closely related to the one used for the moving track model. The contact forces from the MBS will be used to compute the displacement with a FE model.

![Diagram of continuous track model coupling between Vampire & ANSYS](image)

Figure 20: Continuous track model coupling between Vampire & ANSYS

Then, the contact forces are sent to the ANSYS model through a text file, which is regenerated at each step. The forces are applied to the ANSYS model. Then the displacement computed by the Ansys model is written in a text file at each step. It will be read by Vampire and modify the Vampire track position.

At the same time, the Vampire track model already implemented in the vehicle model cannot be disabled. In order to reduce its influence, the stiffness of the track is set extremely rigid to prevent unwanted track displacement.

The operating system used is a 32-bits. This causes limitations for the number of digit that can be transmitted. For this reason and in order to reduce computation time, if a value is lower than 0.01 \( kN \), the ANSYS calculation will not be used. This will reduce computation time.
4.4.2 Validation

In order to use the coupling between the MBS and the FE model, a validation should be performed to insure the accuracy of the result. Several parameters of the coupling will be studied to see their impact as presented in table 11. Two track configurations will be analyzed:

- **Perfect Track**: a tangent track with non irregularity
- **Sinusoidal Track**: a tangent track with a sinusoidal defect

Table 11: Study of influence of parameters

<table>
<thead>
<tr>
<th></th>
<th>Perfect Track</th>
<th>Sinusoidal Track</th>
</tr>
</thead>
<tbody>
<tr>
<td>Timestep</td>
<td>×</td>
<td></td>
</tr>
<tr>
<td>Stiffness of the Vampire track</td>
<td>×</td>
<td></td>
</tr>
<tr>
<td>Speed</td>
<td>×</td>
<td>×</td>
</tr>
</tbody>
</table>

**Perfect Track**

**Timestep selection**  In figure 21, the difference of stability can be seen. The simulation with a timestep $1e^{-5}$ is equal to zero compared with the timestep $1e^{-4}$ which possesses a transient part with a high amplitude. This can lead to instability. The timestep $1e^{-5}$ should be used.

![Figure 21: Difference of response on a perfect track depending on the excitation at 10 m/s](image)

Figure 21: Difference of response on a perfect track depending on the excitation at 10 m/s
**Speed influence**  In figure 22a, the speed 5 \( m/s \), 10 \( m/s \) and 25 \( m/s \) was tested. The oscillation of the contact force is small for the timestep \( 1e^{-5} \). The variation is due to the vehicle model and can have various origins. As said previously, the variation is too small to activate FE computation. At the same time, the increase of speed reduces the oscillation. So the influence is negligible.

![Figure 22: Speed](image)

**Vampire track stiffness influence**  From figure 22b, the high track stiffness leads to higher oscillation amplitude. The results obtained with a Vampire track stiffness of 3000 \( MN/m \) and 1000 \( MN/m \) are almost identical. In order to reduce oscillation amplitude and to prevent unwanted track displacement, the value 3000 \( MN/m \) is chosen.

**Sinusoidal Track**  The simulation is done for a track model composed of a sinusoidal track with a wavelength of 5 m.

As shown in figure 23a, the contact force obtained is not usable due to noise. This confirms the issue brought by the coupling. In order to study the noise, a power spectral density analysis is done in figure 23b.

![Figure 22: Vampire track stiffness](image)
(a) Temporal analysis  

(b) Power spectral analysis

Figure 23: Comparison between a sinusoidal track at a speed of 10 m/s with and without the coupling

In figure 23b, the system doesn’t contain noise until 80 Hz. It is interesting to notice that the noise amplitude increases with the frequency. The noise origin could be the approximation done on the force and the track displacement in the output text file. In order to reduce noise, a low-pass filter is set with a cut-off frequency of 90 Hz in figure 24.

Figure 24: Result of the simulation on a sinusoidal track with a filtering at 90 Hz

In figure 24, the signal contains noise, but the amplitude and the phasing is correct. The coupling gives acceptable results if a cut-off frequency is set to 90 Hz.
5 Case study

In the previous chapter, the coupling and the model have been discussed. The coupling and
the two models created will permit to run simulation on different types of defect which have
an impact on the vehicle behaviour in the frequency range up to 500 Hz. It’s important to
remember that the model is limited to 90 Hz due to Vampire. Different simulations will be
done as shown in table 12 to validate the two models in a wider frequency range up to 90 Hz.

Table 12: Type of defect

<table>
<thead>
<tr>
<th>Theoretical defect</th>
<th>Moving Track model</th>
<th>Continuous Track model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Receptance</td>
<td>Sinusoidal track</td>
<td>Sinusoidal track</td>
</tr>
<tr>
<td></td>
<td>Random track</td>
<td>Random track</td>
</tr>
<tr>
<td>Real defect</td>
<td>Real Track</td>
<td>Corrugation</td>
</tr>
<tr>
<td></td>
<td>Dipped joint</td>
<td>Wheelflat</td>
</tr>
<tr>
<td></td>
<td>Corrugation</td>
<td>Wheelflat</td>
</tr>
<tr>
<td></td>
<td>Wheelflat</td>
<td></td>
</tr>
</tbody>
</table>

The theoretical defects will be used to validate the system in the frequency range 0-20 Hz
where Vampire is valid. The goal is to see the proper behavior of the coupling and the model
working together. The second type of irregularity used will be the real situation defects. They
will validate the model in the frequency range up to 90 Hz. The comparison will be done with
the literature and measurements.

5.1 Moving Track model

The validation will start with the moving track model. The vehicle used will be a four axle
vehicle proposed by Vampire. It has an air spring as a secondary suspension and a shear
spring as a primary suspension.

5.1.1 Theoretical defects

Receptance First, a validation of the correct behavior can be made through a frequency
domain analysis. The vehicle-track system will be excited stepwise at certain frequencies.
The force and the displacement will be measured. The result is given in figure 25. The result
is compared to the example given by T. Dahlberg [1].
Figure 25: Track transfer function

Figure 25 shows the presence of a peak corresponding to the eigenmode of the system. It is located at 110 Hz and corresponds to the ballast eigenmode. This behavior is similar to the one found in [1].

**Sinus Track**  The behavior of the vehicle on a track presenting a sinusoidal irregularity will be studied. The wavelength is 5 m at 5 m/s, 10 m/s & 25 m/s. This study will allow to analyze the track behavior. The result with the new track model will be compared with the result without the coupling. This will be done in spatial and frequency domain. The sampling rate is 100 Hz. The choice of the cut-off frequency is motivated by the model frequency limit.

Figure 26: Response of the system to a sinusoidal irregularity with a wavelength 5m at 10 m/s in time step and the power spectral analysis

Figure 26 shows the same amplitude for the Vampire model and the moving track model. This is confirmed by the power spectral density analysis. But a delay can be observed. This can be due to the coupling and the damping introduced in the track.
Now, two speeds can be tested: 5 m/s and 25 m/s. In figure 27, it can be seen that the dephasing increases with the speed. At the same time, the response at a speed of 25 m/s is greatly filtered.

These effects are linked to the raise of the frequency excitation according to the equation: $f = v/\lambda$ with $f$ the frequency, $v$ the speed and $\lambda$ the wavelength. Since, the wavelength is constant, the speed variation modifies the excitation frequency.

Therefore, the dephasing increases with the frequency of excitation. The damping, which is represented in our model, is adding dephasing. The Vampire track model doesn’t possess damper. The effect of damping is to increase the dephasing.

The noise is more noticeable at low frequency excitation than at high frequency. It is due to the filtering caused by the sampling rate. The high frequency noise is filtered out.

**Random Track** The second study will be carried out on a tangent track with a random irregularity with a wavelength between 0.5 m and 2 m.

As it has been done in the previous section, the study will focus on different speeds. In our case, 3 speeds will be analyzed: 5 m/s, 10 m/s, 25 m/s. The sampling rate is 100 Hz.
(a) Time step analysis

(b) Power spectral analysis

Figure 28: Response of the system to a random track at 5 m/s

(a) Time step analysis

(b) Power spectral analysis

Figure 29: Response of the system to a random track at 10 m/s
Figure 30: Response of the system to a random track of a wavelength at 25 m/s

As explained before, the wavelength is identical for the three results. The speed is the only variable. The amplitudes are quasi-identical between 5 m/s in figure 28 and 10 m/s in figure 29. The difference comes from the spectral analysis. The excitation frequency range is larger at 10 m/s than at 5 m/s in figure 29. The reason is the same as explained for the sinus track.

The main comparison is done with the contact forces at 25 m/s in figure 30. The signal shown in figure 30, doesn’t contain a lot of noise as compared with the contact forces at 5 m/s and 10 m/s. This difference is due to the fact that the excitation frequency is closer to the sampling frequency and some of the information will be filtered out.

5.1.2 Real Track defect

The model can now be tested based on specific phenomena for which the response is known thanks to measurements or the literature.

The first test will be done with a measured irregularity. The wheelflat will be studied based on the comparison with the literature and measurements. The second is a corrugation model with a comparison with the literature. The final analyses is done on a rail joint using the comparison with a measurement.

Real track irregularity A first analysis is done on the low frequency excitation. The comparison is performed between the Vampire simulation with and without the coupling and the measurement. The value is compared will be the contact force on the left wheel of the first wheelset. The sampling rate is 20 Hz. The result is given in figure 31. It shows that the Vampire simulation with or without the moving track model does not make great difference.

The main reason is the simulation for a frequency lower than 20 Hz doesn’t require an advanced track model as said in chapter 2. It is the reason for the similar values between the model. The difference between the results and the measurement is mainly due to the limitation brought by the vehicle model limited to 8 Hz.
Wheelflat  This type of simulation is quite present in the literature. To be able to validate this phenomenon, the result will be compared with the literature and then a measurement.

Literature  The literature comes from Baeza et al [17] where the authors validated a new continuous track model. The first remark that can be made is the proximity between the amplitudes found with our model and the literature in figure 32. However, some great difference can be remarked. The transient part is completely different and the number of peaks too.

This difference is due to the frequency range studied. The limitation due to the rigid wheelset might have an impact on the transient part. The eigenmodes of the wheelset might be excited which will change the response.
**Measurement**  The measurement is performed by SNCF on a commercial passenger car with the vertical acceleration of the wheelset at the axle box. The sampling rate is 90 Hz. The comparison is given in figure 33.

![Graph](image_url)

**Figure 33:** Response of the system to a wheelflat

Some difference between each impact is observed in figure 33 between the simulation and the measurement. The reason is the absence of knowledge on the wheel radius due to commercial use of the vehicle.

At the same time, the result is quite close. The amplitude is corresponding and the transient period is almost of the same length. Some dephasing can be observed, but it might be linked to the one remarked in the theoretical defect.

**Corrugation**  The next step is the study of corrugation. As said in chapter 2, corrugation is a short wavelength irregularity. During the study, only one wavelength will be studied: 111 mm. Background information can be found in Baeza et al [17]. It’s located around the ballast eigenfrequency. The validation will be done by comparing with the result in [17].

In figure 34, the comparison is done between the contact force with a coupling and the track irregularity. The amplitudes are identical with the literature. The period is slightly different. But a dephasing can be observed. It validates the model in terms of corrugation.

**Joint**  A joint will be now studied. It will be compared with the contact forces measured during a test campaign on joint track. The sampling frequency is 90 Hz.
considered to represent the track, and 90 modes are used for each rail. The selected load cases study the dynamic responses due to a rounded wheel flat or a corrugated rail. Results are illustrated in Fig. 13. In these figures, the wheel–rail contact force responses are illustrated for some combinations of irregularity and train speed. Good agreement between the results from the two models is observed.

6. Conclusions

This work presents a method for simulation of the dynamic interaction between train and track. The method is computationally efficient in the sense that a reduced number of coordinates is sufficient. The method proposes a modal substructuring approach of the system by modelling those elements with linear dynamic behaviour (rails and sleepers) with modal coordinates, and by introducing interconnection elements between these structures (wheel–rail contact, railpads and ballast) by means of their interaction forces. The obtained results are similar in the amplitude as observed in figure 35. However, the transient part is quite different. The lack of information on the short track defect (<3m) and the bad track quality lead to differences on the transient part.

5.2 Continuous Track model

For this section, the same vehicle as previously will be used.

5.2.1 Theoretical deflects

Sinusoidal Track In a first step, a track with a sinusoidal irregularity is studied. The irregularity wavelength is 5 m. The result is obtained for a sampling rate of 90 Hz as explained...
previously. The result is summarized for a speed of 10 m/s in figure 36a and 36b. In this part, only one speed will be studied due to the long computation time required.

![Figure 36: Comparison between a Sinus track at a speed of 10 m/s with and without the coupling](image)

In figure 36a, the amplitude and phase are identical. The reason is that the excitation frequency is lower than 20 Hz. So the result is similar to the one without coupling. The noise introduced by the coupling can be observed.

To look at the response in the frequency domain, a power spectral analysis is done in figure 36b. The values obtained are close to the results in the moving track model.

**Random Track** Now, the random track will be implemented with the same parameter as in the previous chapter. The speed is 10 m/s and the sampling rate is 90 Hz. The results are summarized in figure 37a and 37b. In this part, only one speed will be studied due to the great computation time required.

Figure 37a shows that the amplitude obtained with coupling is close to the one without it. The phase obtained is identical for the two simulations.

In the power spectral analysis in figure 37b, the difference is more noticeable at high frequency.

### 5.2.2 Real situation defects

The model can now be tested through specific phenomena where the response is known from the measurements and literature.

The wheelflat will be studied using a comparison with the literature and a measurement. The second application case will be a corrugation model with a comparison with the literature. The final case will be a rail joint with a comparison with the literature.

**Wheelflat** The same procedure will be done as previously
Simulation with the continuous track model at v=10m/s
Simulation without the Vampire track model at v=10m/s

(a) Time step analysis

Simulation with the continuous track model at v=13 m/s

(b) Spectral analysis

Figure 37: Result of the simulation on a random track with and without coupling

**Literature**  The literature result comes from Baeza et al [17] which validates the model results. The first remark that can be drawn is the proximity between the amplitude found with our model and the literature in figure 38. However, the amplitudes between the moving track model and the continuous track model are different. This could be due to the consideration of the rail deformation. Significant differences can be recognized. The transient part is completely different and the number of peaks too. This might be due to the difference of modeling between the work of Baeza and the model used here such as the track stiffness.

![Results](image)

(a) Result found in literature [17] 
(b) Simulation

Figure 38: Response of the system to a wheelflat

**Measurement**  The measurement is performed by SNCF on a commercial passenger car with the vertical acceleration of the wheelset at the axle box. The sampling rate is 90 Hz. The comparison is given in figure 33.
Figure 39: Response of the system to a wheelflat

The simulation and the measurement give the same peak amplitude. The transient parts are almost of the same length. However, the transient parts are not similar as for the moving track model. This difference can be due to the difference in modelling.

**Corrugation** The corrugation will be studied with a wavelength of 111 mm by comparing with the study from Baeza et al [17].

![Graph](image1)

(a) Result found in literature [17]  
(b) Simulation

Figure 40: Response of the system to a wheelflat

The results obtained are close in terms of amplitude. A small dephasing can be observed. The signal is not a perfect sinus because part of the information is filtered by the cut-off frequency.
5.3 Comparing results from both models

The last part will deal with the comparison between the two models. The defects studied will be:

- the sinusoidal irregularity with a wavelength of 5m
- the corrugation with a wavelength of 111mm
- the wheelflat

As shown in figure 41a, the amplitude and the period are the same. The main difference comes from the noise in the continuous track model. But, as explained previously, it’s due to the coupling.

![Comparison results from both models](image)

(a) Sinusoidal defect

(b) Corrugation defect

Figure 41: Comparison result from both models

In figure 41b, the period is identical. But differences can be observed on the amplitude. The continuous track model contact forces are higher than the moving track model contact forces. The difference might come from the additional deformation of the rail which is not considered in the moving track model.
To conclude, figure 42 shows some differences between the two models. The phase is almost identical, because the vehicle runs at the same speed on the same irregularity. However, a difference of amplitude and the transient part can be observed. The main difference comes from the fact that the response to a wheelflat changes every time due to non-identical initial condition. But, the results are still similar. The transient part is longer in the continuous track model due to the wave propagation consideration.
6 Conclusion & Future work

Within the thesis, the railway track has been studied. The start of the work was the modelling of the different components of the track in a MBS software. This study was followed by the implementation and the validation of two types of track models. Both have pros & cons which have lead to the development of both models.

The differences between moving track model and the continuous track model are given in table 13.

<table>
<thead>
<tr>
<th></th>
<th>Moving Track Model</th>
<th>Continuous Track Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Computation time</td>
<td>30 s</td>
<td>5 h 35 min</td>
</tr>
<tr>
<td>Theoretical frequency limit</td>
<td>200 Hz</td>
<td>2000Hz</td>
</tr>
<tr>
<td>Real frequency limit</td>
<td>90 Hz</td>
<td>90 Hz</td>
</tr>
<tr>
<td>Wave propagation</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Modeling High frequency</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Modeling punctual defect</td>
<td>No</td>
<td>Yes</td>
</tr>
<tr>
<td>Example: Floating Sleeper</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The significant difference is the computation time. The moving track model is 580 times faster than the continuous track model. At the same time, the simulations carried out are more precise with the continuous track model. The possibilities with the continuous track model are greater. The possibilities for example to easily model the punctual defects or the wave propagation, allow to simulate the phenomena with a greater accuracy. It’s the model of choice.

As it has been shown in the previous chapter, both models produced correct results compared with the literature and the measurement. The theoretical defects allowed to validate the system in the frequency spectrum used by Vampire. The real irregularity such as the wheelflat, corrugation and the rail joint permitted to validate both models in a wider spectrum.

However, some differences were observed due to the modeling. The greatest divergence comes form the coupling that introduced noise and might alter the result.

From this result, some improvements are proposed which would improve the current model and the implementation.

During the thesis, the corrugation, the wheelflat and a real track had been the reference in the validation. This choice allowed to present the limitations and the validity of the model.

However, the validations were carried out for a short and precise frequency excitation such as corrugation or an wide frequency excitation like for the wheelflat. It will be interesting to obtained the track irregularity at known frequency. This will allow a better validation. The track measurement will require tools to measure short irregularity wavelength. The vehicle will need to be equipped with accelerometer.
This setting will allow for a better validation of the current model for all frequencies. In order to improve the modeling, several points should be dealt with in depth. They are outlined below.

The main thing to be improved is the modeling of the wheelset eigenmode. It is the main drawback of this work. A lumped mass model can not be accomplished without losing the stability of the system. Another multi-body software should be tested considering the wheelset as a flexible body as well as the railway track.

At the same time, a new contact model could be implemented. It will improve the precision by not using an Hertzian contact, but a non-elliptic contact area. Such contact models exist, but they are not implemented in Vampire.

In the same process, the model could be extended to the lateral direction. The precision of the model will be improved.

At the beginning of the thesis, an hypothesis had been made on the parameters. First, more studies could be done to precisely determine the track parameters. The imprecision and the unknown parameters could be reduced significantly.

The second was the linearity of the components. This isn’t true at very high frequency for the track and the vehicle elements. It will be important to take into account nonlinearities for modelling very high frequencies.
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