ENSTA ParisTech

DIRECTION DE L'INNOVATION ET DE LA RECHERCHE SNCF

Time Domain modelling of wheel/rail interaction, in the context of the railway rolling noise

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CIFRE PhD Thesis SNCF / ENSTA ParisTech

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Soutenue : **16th of March 2011**

Wheel/rail interaction noise modelling : Industrial issues

- The rolling noise is the **predominant source** in railway noise for the largest train speed range.

➢ **The rolling noise has to be controlled in order to respect the noise laws and directives.**

- The rolling noise is directly **linked to the surface condition** of the structures in contact.

➢ **The railway operator** needs to maintain the **acoustic performances** of its **rolling stock.**

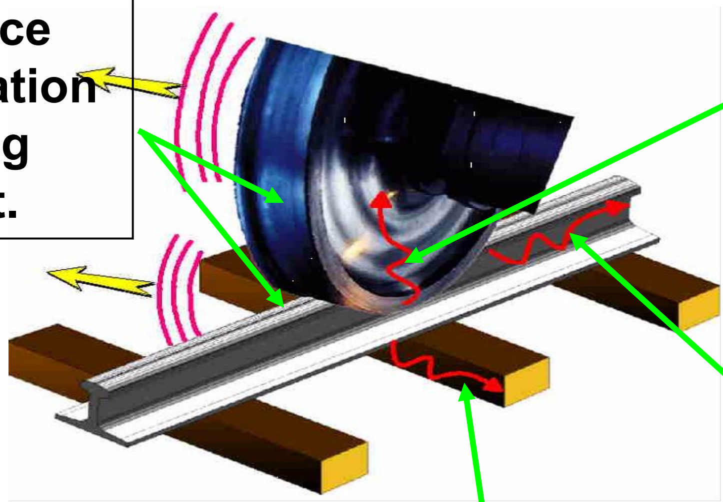
➢ **The infrastructure manager** needs to maintain **the acoustic performances** of the **railway tracks.**

New issue : the surface default detection on wheels and rail.

The rolling noise

The roughness (microscopic surface defaults) is the excitation mechanism, during wheel/rail contact.

The wheel radiates at high frequencies

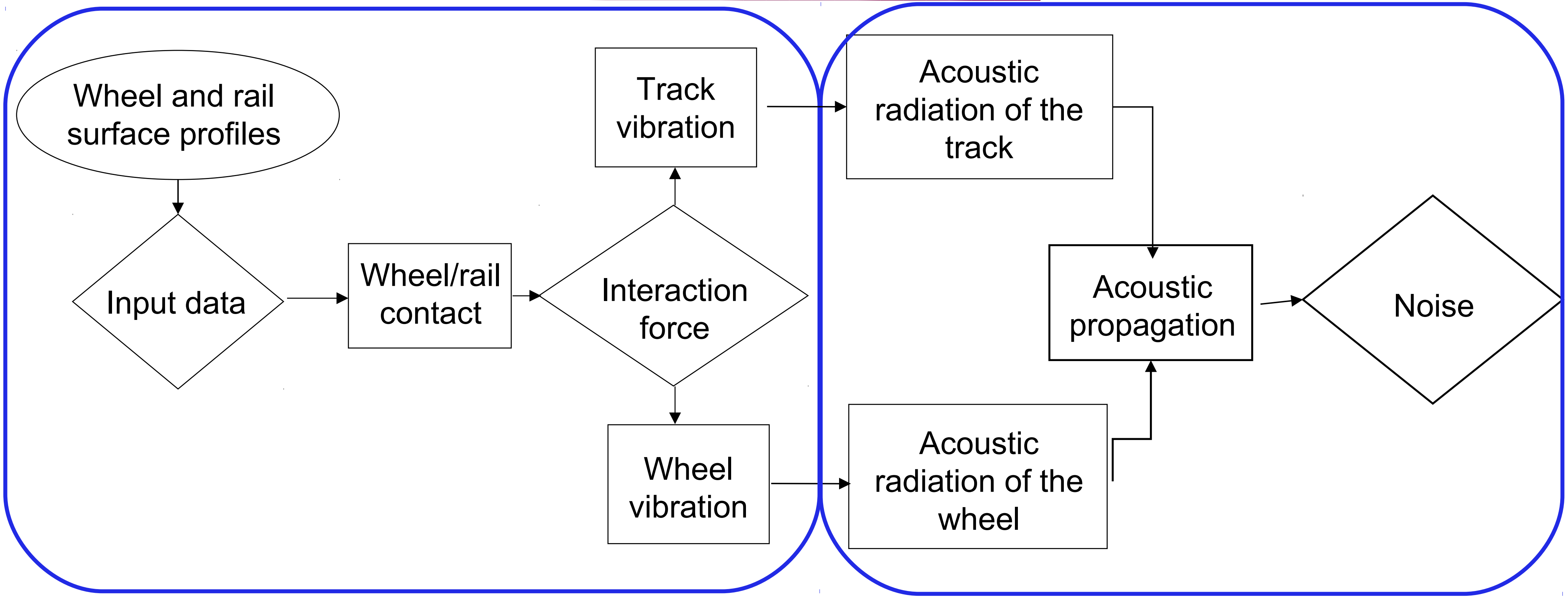


The sleepers radiate at low frequencies

The rail radiates at middle frequencies

The rolling noise is due to the wheel/rail interaction

modelling diagram of rolling noise



Simulation of the wheel/rail interaction

Simulation of the acoustic radiation and propagation

Example based on the frequency domain : TWINS (Track Wheel Interaction Noise Software)

- **Input data :** combined roughness + Wheel and track dynamic parameters.
- **Output data :** Noise levels of the sleeper, the rail and the wheel + Total noise level.

Frequency vs. time domain for rolling noise simulation

Frequency domain :

- **Linear approach**.
- Well adapted for cases of **microscopic roughness amplitudes.**

Time domain : extension to Impact noise

- **Discrete surface irregularities** lead to **non linearities and discontinuities** in the wheel/rail contact .
- A **time domain approach** is required to simulate the wheel/rail interaction in the Impact noise context.

PhD Work Aim

- **Develop a time domain model of wheel/rail interaction which could take into account the real wheel and rail surface profiles.**
  - Some modelling assumptions are given by previous research results on rolling noise.
  - Other requirements have to be checked by an experimental characterization of the wheel/rail interaction in real situation.

- **Carry out an experimental characterization of the wheel/rail interaction during pass-by.**
  - Influence of the a wheel default on the track behaviour.
  - Virginie Delavaud, Franck Poisson, Christophe Gérault, *Caractérisation expérimentale du bruit de roulement et du bruit d'impact,* Actes du 10e Congrès Français d'Acoustique, Lyon, 2010.

    Acoustique et Techniques, vol. 62, pp 43-49, décembre 2010.

Modelling assumptions

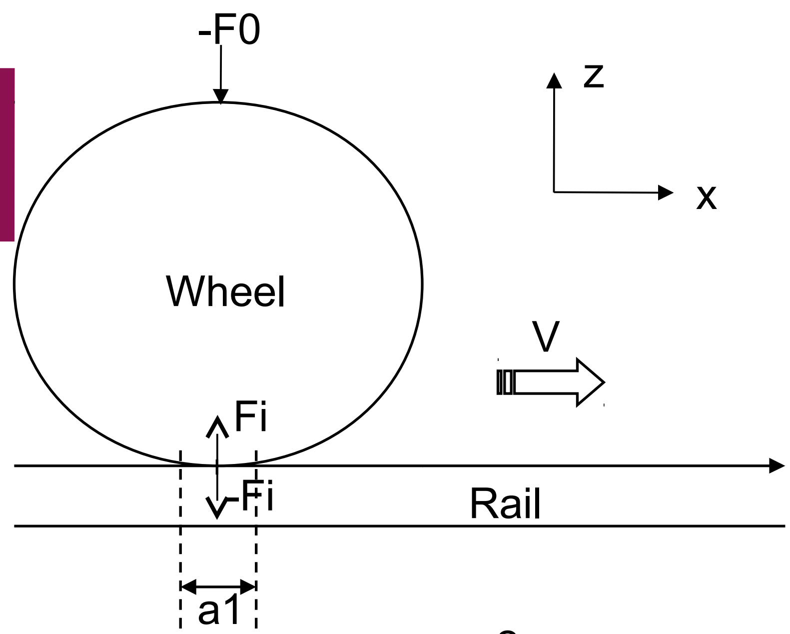
- **Straight track :**
  - Vertical contact model based on Hertz model ;
  - The tangential frictions are ignored.

- **Wheel model :**
  - Vehicle : static load above the wheel,
  - Wheel : mass-spring-damper system which moves along the rail at constant speed.

- **Rail Lateral vibration** is not considered.

- **Validity domain of the model :**
  - Frequency range considered below 2000Hz.
  - Train speeds lower than 100 km/h.

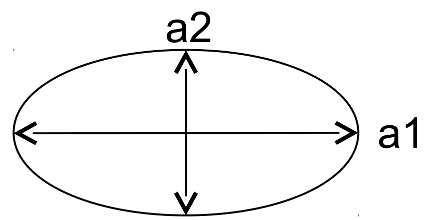
Excitation model : input data and contact model



- **input data** = relative displacement between wheel and rail called « **Relative roughness** »

$$w_r(x) = r_R(x) - r_W(x)$$

- **The contact ellipse dimensions** are calculated from the wheel static load F0.
- **Elastic compression :**

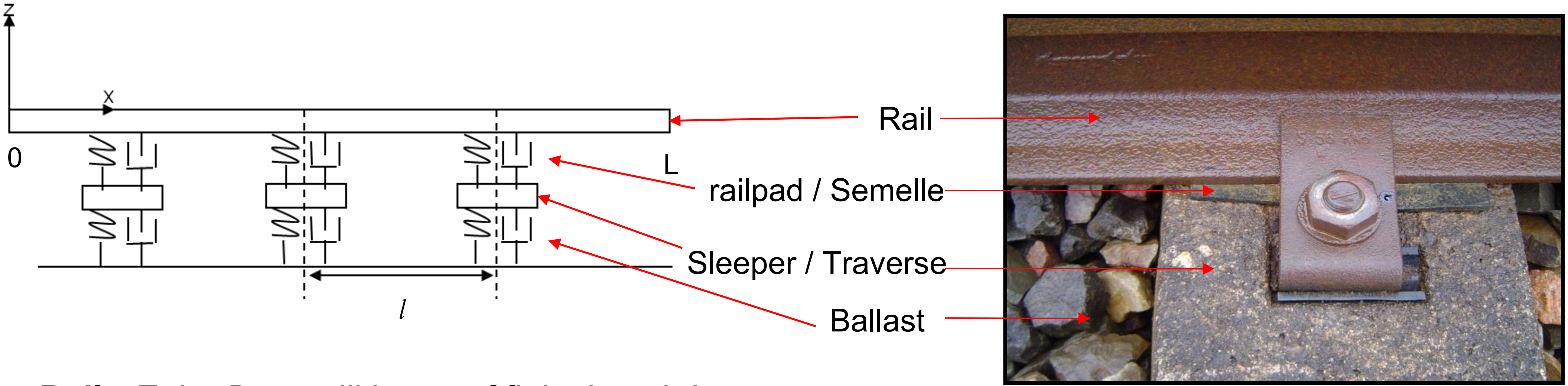


$$\delta(x, y, t) = w_R(x, t) - w_W(t) + w_r(x, y).$$

- **The normal force** at each sample point into the contact patch is calculated from the elastic compression by using a **non-linear Hertzian model modified to consider loss of contact.**

- The interaction force **Fi(t)** between wheel and rail is **an average of the normal force over the contact patch, at each time step.**

Track model



- **Rail :** *Euler-Bernoulli beam of finite length L.*
- **Periodic supports *:***
  - *Mass-spring-damper systems,*
  - *Period :* l .
- **The beam is assumed to have free ends**.

- **Absorbing boundary conditions :** *Method proposed by Svensson et al. [JSV, 2009].*

Absorbing boundary conditions in frequency domain Svensson et al. [JSV, 2009]

- **Impedance matching technique** to cancel boundary reflections of an Euler Bernoulli beam in frequency domain.

- Calculation of an absorbing impedance matrix which **cancels the reflection matrix**

$$\begin{pmatrix} a_p^- \\ a_N^- \end{pmatrix} = R \begin{pmatrix} a_p^+ \\ a_N^+ \end{pmatrix}$$

- **Reflection matrix defined by 3 impedance matrix**:
  - 2 to describe the Euler-Bernoulli beam,
  - 1 to define the boundary impedance,

➡ **Addition of an external active impedance load which is defined to cancel the reflection matrix.**

$$\hat{Z} = \hat{Z}^0 + \hat{Z}^{abs}$$

Application of the Svensson impedance matching method

- **Free ends :** $$\hat{Z}^0 = 0$$

- The **shear force** is only linked to the **normal velocity**
  The **bending moment** is only linked to the **rotational velocity**
  $$\hat{Z}^{abs} = \begin{bmatrix} z_{11}^{abs} & 0 \\ 0 & z_{22}^{abs} \end{bmatrix}$$

- **Incident near field is assumed to have lower influence** on the wave reflexion
  $$R = \begin{bmatrix} r_{11} & r_{12} \\ r_{21} & r_{22} \end{bmatrix} \qquad \begin{pmatrix} a_p^- \\ a_N^- \end{pmatrix} = R \begin{pmatrix} a_p^+ \\ a_N^+ \end{pmatrix}$$

➡ **Mathematical problem solved :** find z_{11}^{abs} and z_{22}^{abs} which cancel r_{11} and r_{21}.

➡ $$\begin{cases} \mathcal{T}_z(x_0,\omega) = z_{11}^{abs}(\omega)\dot{w}_R(x_0,\omega), \\ \mathcal{M}_y(x_0,\omega) = z_{22}^{abs}(\omega)\partial_x\dot{w}_R(x_0,\omega), \end{cases}$$

$$z_{11}^{abs}(\omega) = \pm\frac{EI}{a}\sqrt{\frac{\omega}{a}}$$

$$z_{22}^{abs}(\omega) = \pm\frac{EI}{\omega}\sqrt{\frac{\omega}{a}}$$

Absorbing boundary conditions in the time domain

- **Impedance in frequency domain** are defined by root square:
  - ➢ **Interpolation by rational fraction** to simplify the conversion in time domain.

- **Energy analysis in case of harmonic vibration of the complete absorbing boundary conditions in the time domain** :
  - ➢ Some terms in the time domain absorbing boundary conditions make the energy increase in local frequency range included in the global frequency range considered,

➡ Simplification of the absorbing boundary conditions in time domain :

$$\begin{cases} \partial_x^3 w_R(0,t) = -R_0 \partial_t w_R(0,t), \\ \partial_x^2 w_R(0,t) = \eta_0 \partial_t \partial_x w_R(0,t) \end{cases} \qquad \begin{cases} \partial_x^3 w_R(L,t) = R_0 \partial_t w_R(L,t), \\ \partial_x^2 w_R(L,t) = -\eta_0 \partial_t \partial_x w_R(L,t). \end{cases}$$

Numerical resolution

- **Resolution** of the numerical problem by using the **finite difference method**.

Adjustment of the model parameters

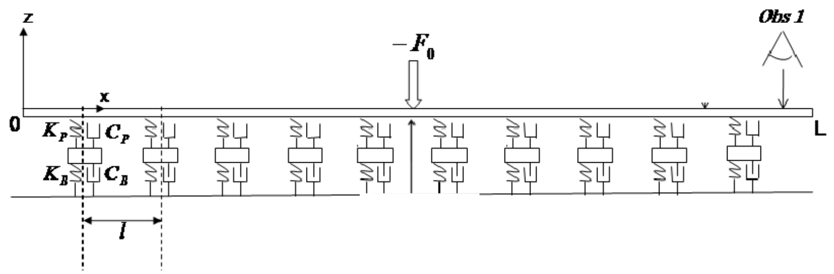
- **dx** is selected in view of the simulated situation.
- **dt** is imposed by the stability conditions of the finite difference scheme.

Simulated situations

1. Performances assessment of the absorbing boundary conditions : Simulation of a vibroacoustic measure with impact hammer.

2. Simulations of wheel pass-by at 80 km/h : Case of the reference wheel and the wheelflat.

Influence of the boundary conditions at the beam end



Legend of the figures

—— 15 spans, Absorbing boundary conditions

······· 15 spans, free boundary conditions

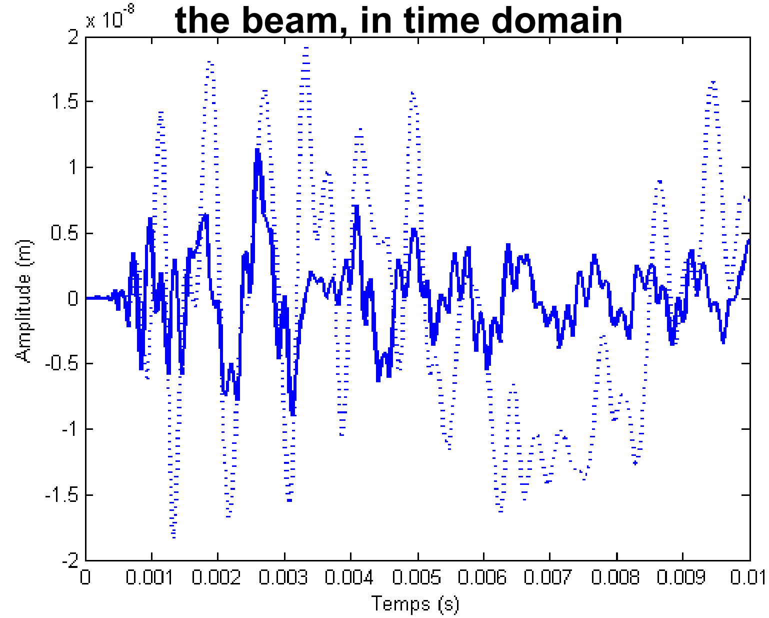
Time domain

- Lower vibration amplitudes in case of absorbing boundary conditions.

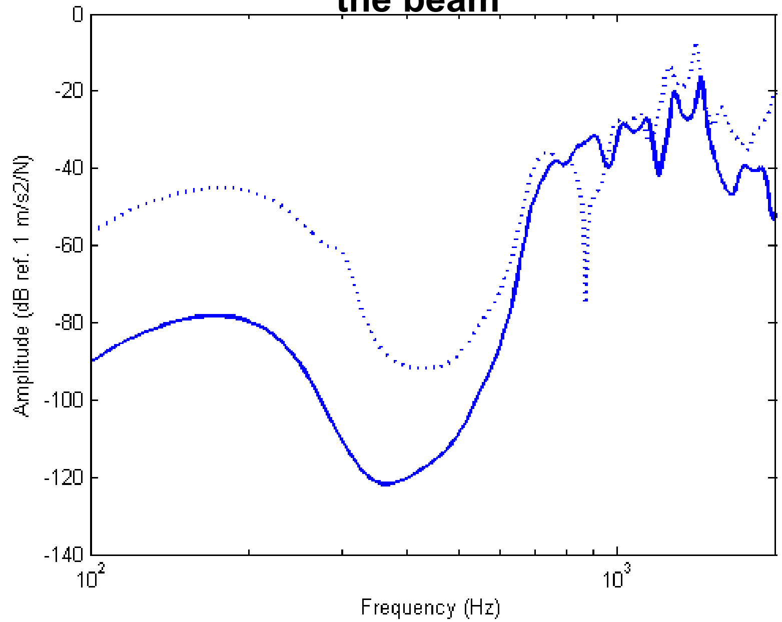
Frequency domain

- Energy absorption at low frequencies (<700 Hz) ;
- Correction of an anti-resonance (~900 Hz) ;
- Lower influence of the boundary conditions at higher frequencies ([900;1500] Hz).

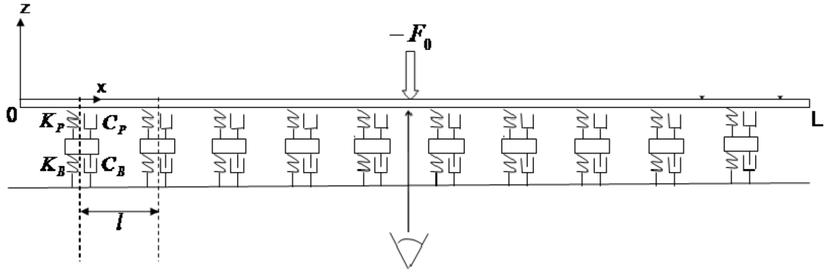
Rail vertical displacement at the end of the beam, in time domain



Rail vertical accelerance at the end of the beam

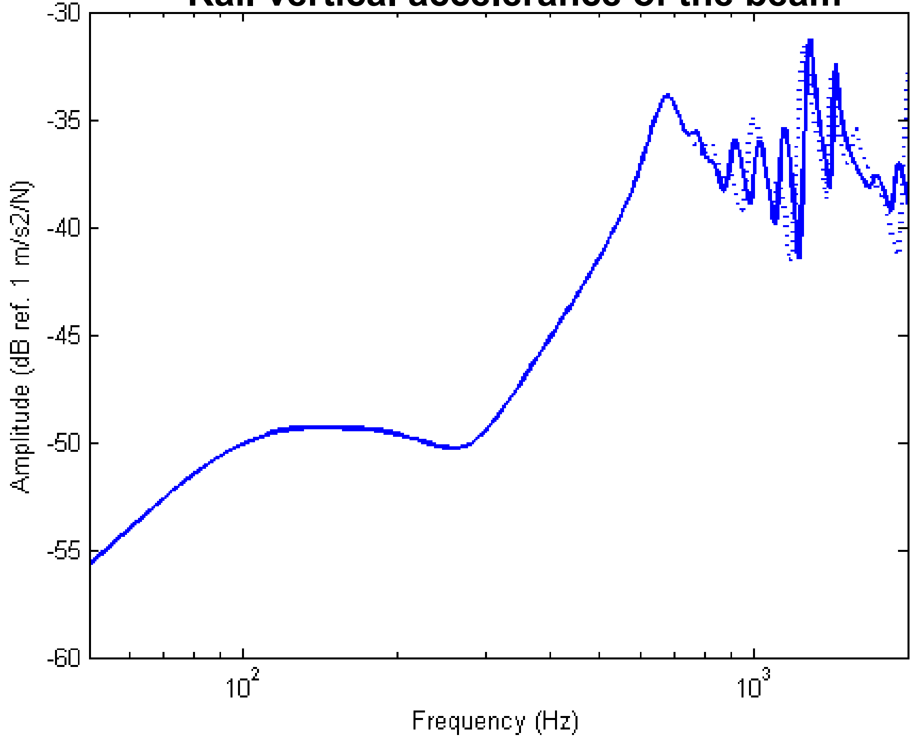


Comparison of the simulated accelerances at the middle of the beam



Legend of the figure

- 15 spans, Absorbing boundary conditions (solid line)
- 15 spans, Free boundary conditions (dotted line)

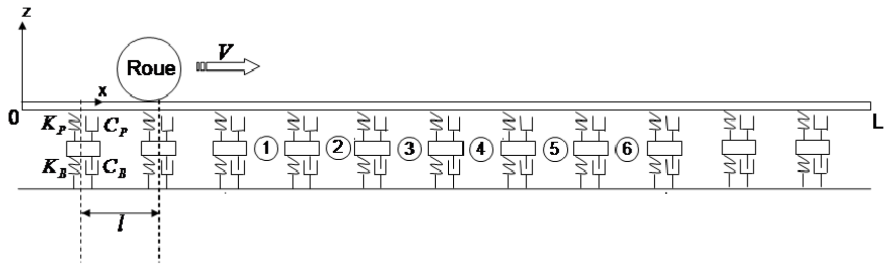


Behaviour at the middle of the beam

- **Below 700Hz:** behaviour controlled by rail supports ;
- **Above 700 Hz:** Free wave propagation onto the rail, Resonances present in both accelerances.

➡ **Limitation of the absorbing boundary conditions performances in the frequency range of free wave propagation onto the rail.**

Pass-by simulation results :
Influence of a wheel default vs. spatial step



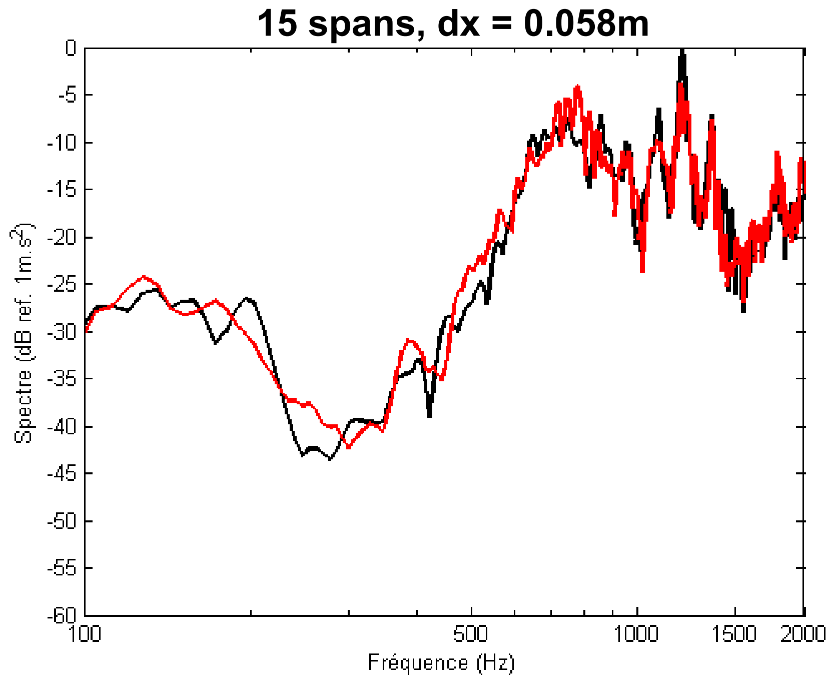
Mean Vertical acceleration spectrum of the rail

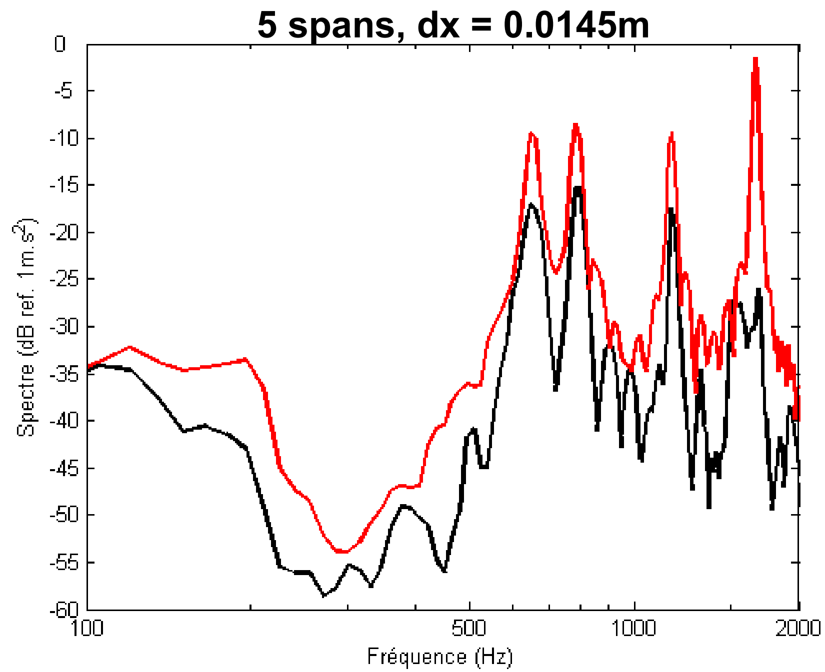
— Reference wheel case

— Wheelflat case

- **Flat length : **0.025 m.
- **First case :** 15 spans, dx = 0.058m **> flat length**.
- **Second case :** 5 spans, dx = 0.0145m **< flat length**.

The **rail acceleration spectrum during the wheelflat pass-by** presents a higher energy than during the reference wheel pass-by for the second case.





Main conclusions

- **Excitation model:**
  - *New input data processing* adapted to a time domain approach : the relative roughness.
  - *Modified nonlinear Hertzian contact model* which allows loss of contact.
  - *Wheel model :* simple mass-spring-damper model.

- **Track model:**
  - Rail : Euler-Bernoulli beam with *absorbing boundary conditions* based on Svensson et al. Method.
  - *Supports* : mass-spring-damper systems periodically distributed under the rail.

- **Numerical resolution using a *finite difference scheme*.**

- **Limitations of the approach used :**
  - Euler-Bernoulli beam model.
  - Absorbing boundary conditions.

Perspectives

- **Scientific perspectives :**
  - Timoshenko beam.
  - The impedance technique can be applied with less simplifications and for a Timoshenko beam model.
  - Acoustic radiation : require models adapted to the time domain approach.

- **Industrial perspectives : Wheel default identification**
  - Develop a method combining measurement and simulation.
  - The measurements allow the detection of a default on wheels .
  - The simulation would allow to get back the wheel roughness profile of the wheel and thus the default nature.

Thank you for your attention !