UPC - BARCELONA TECH

Master on Numerical Methods in Engineering

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Computational Mechanics Tools

COURSE SIMULATION PROJECT WITH ABAQUS

DYNAMIC ANALYSIS OF A TRAIN WHEEL

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Contents

1 Introduction

One of the most common issues with rail transit systems in inhabited areas is the acoustic pollution they genetare. One specific and relevant source of acoustic pollution is a phenomenon called wheel squeal. The wheel squeal is a high frequency, high pressure level sound caused by the vibration of the wheels that results in unacceptably noisy environments around the rail systems [3].

The characteristics of the wheel squeal are difficult to define and vary from vehicle to vehicle and from day to day, as it depends on geometric factors (wheel size and shape, curvature of rail), environmental factors (temperature, humidity of air), dynamic factors (linear velocity, angular velocity), etc. Figure 1.1 shows a summary of several possible causes of the squealing problem. Therefore, it is a complicated field of study without a unique and simple solution.

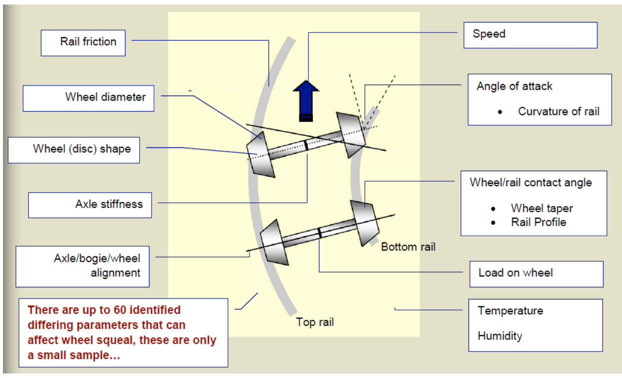

Figure 1.1: Possible causes for wheel squealing [4]

The goal of this project is to carry a dynamic analysis of a train wheel in order to study several hypothesis for the cause of the squeal and decide whether if they are plausible or not.

2 Problem Statement

As introduced before, the goal of this project is to analyze the dynamic response of a train wheel and try to predict the presence of squeal under different circumstances. The squeal is defined as a highly audible noise between 2 and 8 kHz generated when the wheel is rotating. The potential sources of squeal suggested in this study are:

- Coupling between the wheel natural frequencies and its frequency of rotation, considering that the train travels at a maximum speed of 350 km/h.
- Frequency at which the wheel hits the sleepers (transverse beams that hold the rails),that are located every 60 cm.
- The stick-slip transitions of the wheel with respect to the rail, specially when the rails have curvature.

The objective of the project is to conduct a dynamic analysis in order to decide if these hypotheses of sources of the wheel squeal should be discarded or accepted. A brief bibliographical research might be done to find other possible sources of squeal.

3 Theoretical framework - Free oscillation

A short introduction to the theory of dynamic analysis is presented next, to facilitate the comprehension of the following sections of the document.

The term vibration is usually related to a repetitive variation of a system's configuration around its equilibrium state. In general, dynamic forces that change with time are the cause of the vibrations. The presence of vibration means a conversion between kinetic and potential energy in the body, with dissipation mechanisms that produce the dumping of the vibration with time.

If a system is first displaced from its equilibrium position and then released, it will vibrate without any external excitation in what is called free oscillation or free vibration. The frequencies at which the system vibrates under these conditions are called natural frequencies and each natural frequency has associated a form of vibration called vibration mode.

The natural frequencies are an important feature to be studied for any system subjected to vibrations due to a phenomena called resonance. When the system is subjected to an external force of frequency equal to any of the system's natural frequencies, a condition called resonance is reached and it causes the amplitude of the vibration to rapidly increase. The amplitude increase tends to infinity if the system is not dumped and it can reach values that are too high for the safe functioning of the system and can lead to its failure. Thus, conducting dynamic analysis of systems subjected to any periodic excitation is a key point during its design.

In vibrations, as in any other physics field, it is necessary to obtain a model of the reality so it can be analyzed. This document will not go into the details of obtaining the model of a dynamic system and will assume the reader has a certain previous knowledge of vibrations theory.

The equation that represents the free vibration (with no dumping) of a system when its perturbed from its equilibrium state and then released is an homogeneous differential equation of the form:

$$M\ddot{\boldsymbol{u}} + K\boldsymbol{u} = 0 \tag{3.1}$$

where, in general for a multi-degree of freedom system

- M is the so-called mass matrix.
- K is the stiffness matrix
- $\boldsymbol{u}$ is the displacements vector

The solution of this equation can be assumed to be

$$\boldsymbol{u}(t) = \boldsymbol{a}\phi(t) \tag{3.2}$$

where $\boldsymbol{a}$ is a vector of constant parameters and $\phi(t)$ is just a function of time.

The configuration of the system, given by the vector

$$\boldsymbol{a} = \begin{Bmatrix} a_1 \\ a_2 \\ \vdots \\ a_n \end{Bmatrix}$$

is known as the *mode shapes*.

Replacing the solution in the original equation gives:

$$\boldsymbol{M}\boldsymbol{a}\ddot{\phi}(t) + \boldsymbol{K}\boldsymbol{a}\phi(t) = 0 \tag{3.3}$$

Dividing by $\phi(t)$, yields:

$$\boldsymbol{M}\boldsymbol{a}\frac{\ddot{\phi}(t)}{\phi(t)} + \boldsymbol{K}\boldsymbol{a} = 0 \tag{3.4}$$

Defining $\lambda = \frac{\ddot{\phi}(t)}{\phi(t)}$ and replacing in the previous equation gives:

$$-\lambda\boldsymbol{M}\boldsymbol{a} + \boldsymbol{K}\boldsymbol{a} = 0 \tag{3.5}$$

If $\boldsymbol{K}$ and $\boldsymbol{M}$ are positive definite, and $\lambda \in \mathbb{R}$. we can define now $\lambda = \omega^2$, and it yields

$$\boldsymbol{M}^{-1}\boldsymbol{K}\boldsymbol{a} = \omega^2\boldsymbol{a} \tag{3.6}$$

and this is an eigenvalue problem whose solution is obtained upon computing the determinant of the matrix,

$$\boldsymbol{D}\boldsymbol{a} = \omega^2\boldsymbol{a} \;\Leftrightarrow\; det(\boldsymbol{D} - \omega^2\boldsymbol{I}) = 0 \tag{3.7}$$

where $\boldsymbol{D} = \boldsymbol{M}^{-1}\boldsymbol{K}$ is the so-called dynamical matrix and $\boldsymbol{I}$ is the identity matrix of the required order.

From this problem, as many ω,$\boldsymbol{a}$ pairs as degrees of freedom of the system can be obtained. The natural frequencies are the eigenvalues ω and the eigenvectors $\boldsymbol{a}$ indicate how the body will deform when vibrating at that specific natural frequency (the mode shape). This information can be used to determine if it exists any external excitation whose frequency is close to a natural mode (coupling) that could cause a resonance phenomena.

4 Methodology

The procedure followed in this project requires completing a dynamic analysis in order to obtain the natural frequencies of the train wheel. This section explains the steps followed for carrying the dynamic analysis using *Abaqus*.

4.1 Wheel geometry

The first step of the analysis is modeling the geometry using the software's CAD tool. The real wheel is simplified as a three dimensional disc shown in Figure 4.1 and its assumed to have a linear elastic behavior. The dimensions and the properties of the material are given in table 4.1.

The geometry is obtained by drawing to concentric circles and extruding the surface defined by them. Then the material created using the given properties. Afterwards, a section is created and the previously defined material is assigned to it.

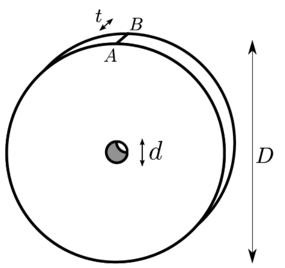

Figure 4.1: Wheel representation

Width	t	0.05	$[m]$
Internal diameter	d	0.10	$[m]$
External diameter	D	1.00	$[m]$
Density	ρ	7800	$[Kg/m^3]$
Young modulus	E	$210E9$	$[Pa]$
Poisson ratio	ν	0.25	$[-]$

Table 4.1: Wheel's geometry and material properties

4.2 Simulation setup

Next, the characteristics of the analysis need to be defined. An *step* is created selecting a 'linear perturbation' procedure and 'frequency' is marked as the quantity to be obtained, as we want to perform a dynamic analysis. For this case, it was considered sufficient to obtain the first 10 natural frequencies and modes of vibration of the wheel. With all of this parameters defined, the job can be submitted and the results visualized.

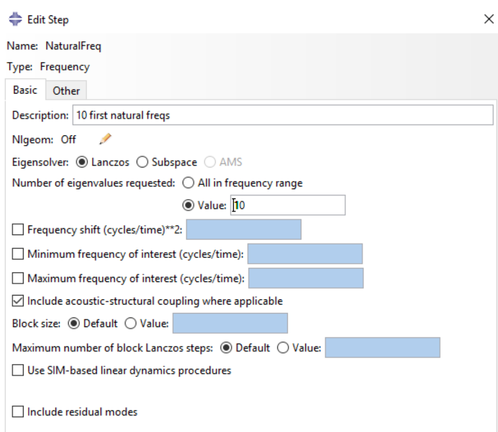

Figure 4.2: Set-up for the simulation

4.3 Boundary conditions and mesh

Once the geometry is defined, it is necessary to set boundary conditions that allow the problem to be solvable. For this case, the nodes close to the center (the inner surface in grey in Figure 4.1) are set to have zero displacement and rotation (encastre). Also, the nodes in line AB shown in the same figure have the condition of being symmetric with respect to the X axis. Figure 4.3 shows the model of the wheel in *Abaqus* after defining the mentioned boundary conditions.

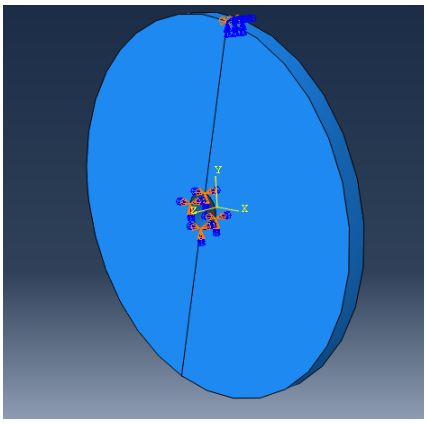

Figure 4.3: Geometry and boundary conditions

A partition of the geometry is created so that it becomes easier to assign the foregoing boundary conditions and the mesh can reproduce better the shape of the geometry near the hole. For the mesh, hexaedra finite elements were chosen. An approximate global size of 0.05 gives the mesh that can be seen in Figure 4.4.

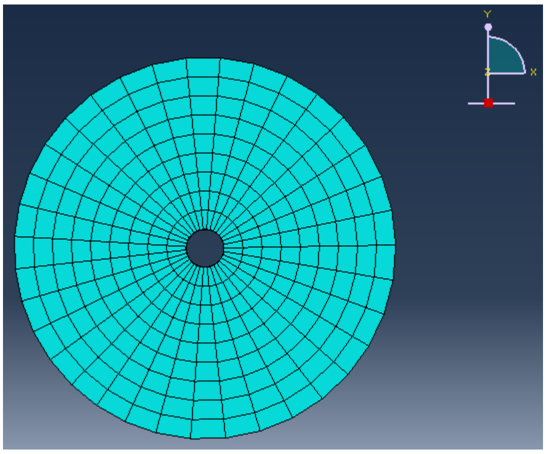

Figure 4.4: Meshed wheel

5 Results

This section is focused on studying the validity of the different hypothesis about the cause of the wheel squeal using the data obtained from the dynamic analysis. The hypothesis considered, recalling what was previously mentioned, are:

- Coupling with the frequency of rotation
- Coupling with the sleepers contact frequency
- Stick-slip transitions between the wheel and the rail

Each of the hypothesis will be studied separately.

5.1 Dynamic analysis results

The 10 first natural frequencies of the wheel are presented in Table 5.1 and the different modes of vibration can be seen in Figures 5.1 -5.5. The images show the initial shape and the deformed shape superimposed to facilitate the understanding of the wheel's deformation.

Mode	Frequency $[Hz]$
1	24.759
2	25.116
3	26.256
4	38.749
5	48.386
6	82.899
7	87.851
8	133.11
9	141.63
10	146.99

Table 5.1: Wheel's natural frequencies

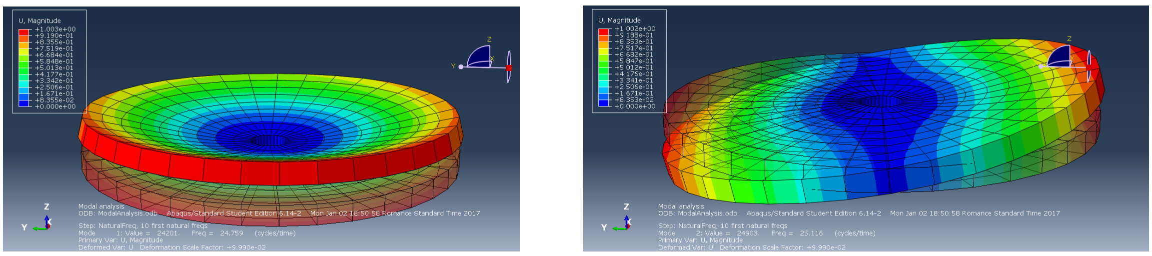

Figure 5.1: Modes 1 and 2 of vibration

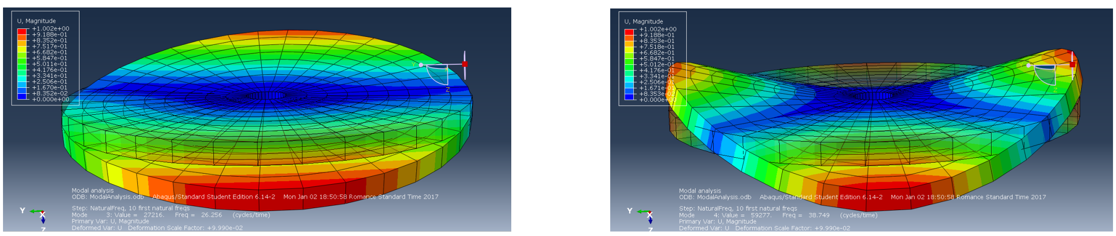

Figure 5.2: Modes 3 and 4 of vibration

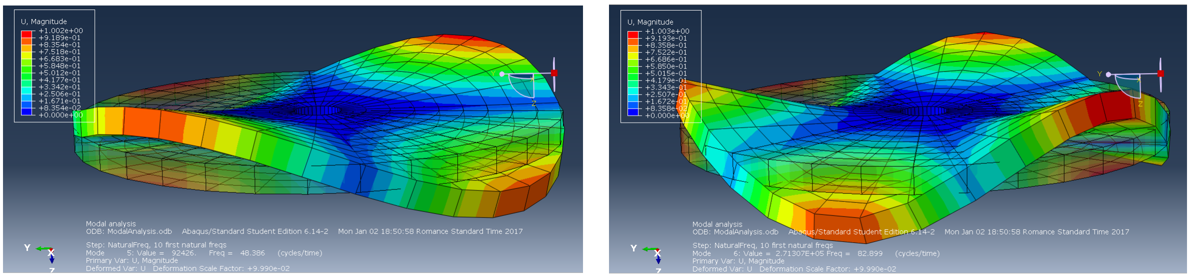

Figure 5.3: Modes 5 and 6 of vibration

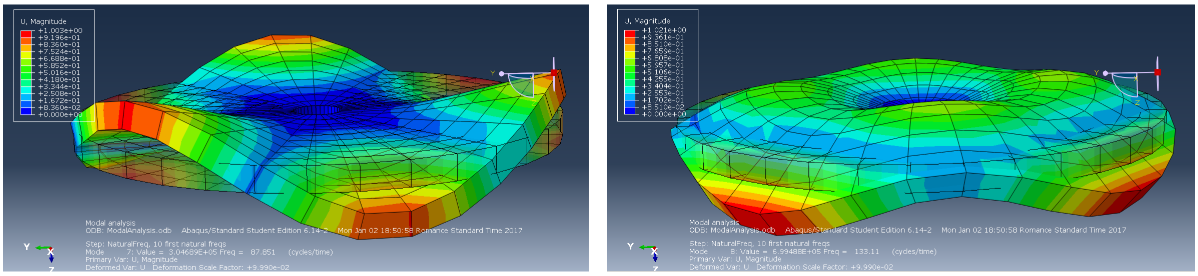

Figure 5.4: Modes 7 and 8 of vibration

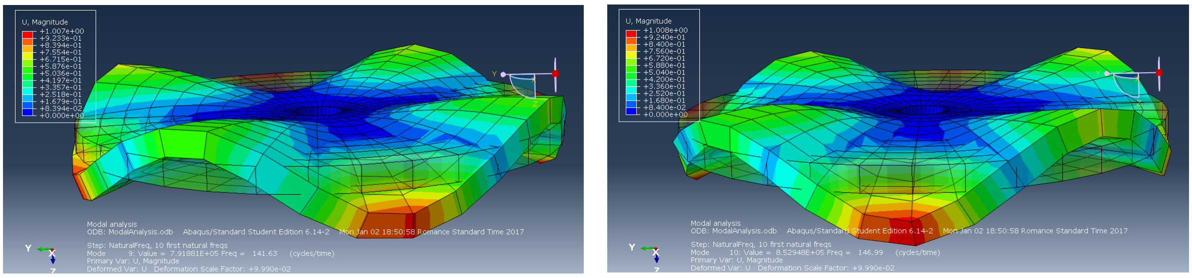

Figure 5.5: Modes 9 and 10 of vibration

5.2 Coupling with the frequency of rotation

The first hypothesis to be checked is if there exists coupling between any of the natural frequencies of the wheel and the frequency due to its rotation when it is traveling at a maximum speed of 350 km/h.

To convert the given velocity to the international system units:

$$\frac{350km}{h} \cdot \frac{1000m}{1km} \cdot \frac{1h}{3600s} = 97.22m/s \tag{5.1}$$

Then it is necessary to translate that linear velocity into an angular velocity that allows to compare with the natural frequencies obtained before.

$$\frac{97.22m}{s} \cdot \frac{1rev}{2\pi R} = 30.9497\frac{rev}{s}[Hz] \tag{5.2}$$

The value of the frequency of rotation of the wheel when it is traveling at its maximum speed is $f = 30.95Hz$. This value of frequency is very close to mode 3 of the free oscillation modes, that has a natural frequency $\omega = 26.25Hz$. That means that when the train is traveling at this speed, it is possible to have some kind of resonance phenomena that amplifies the amplitude of the motion of the wheel. In that case, the wheel would be suffering severe deformations that will change periodically with time. If the amplitude reaches values high enough, then according to [1], it would be possible for the wheel to enter a phase of stick-slip transitions with respect to the rail and thus produce squealing. This stick-slip transition phenomena will be presented and explained later on this section.

5.3 Coupling with the sleepers contact frequency

The second suggested cause of squeal noise is the coupling with the frequency at which the wheel enters in contact with the sleepers. The sleepers are the transverse beams that hold the rails together. They are regularly located every 60 cm.

Assuming that the train is traveling at a given velocity V m/s, then the frequency of contact f_c between the wheel and the sleepers would be obtained as:

$$V\frac{m}{s} \cdot \frac{1cycle}{0.6m} = f_c\frac{cycle}{s}[Hz] \tag{5.3}$$

If this frequency of contact f_c is equal to any of the natural frequencies of the wheel then resonance will happen. Under resonance conditions, the vibration of the wheel will be amplified and the stick-slip transition phenomenon will appear. Thus, the sleepers can be a cause of the squeal noise of the wheel depending on the traveling speed of the train.

Table 5.3 summarizes the forward traveling velocities that will generate frequencies of contact that are exactly the natural frequencies of the wheel. The coupling seems to be specially relevant for low speeds, so it will happen for instance when the train is entering or leaving a station. Areas around train stations are usually highly inhabited so this cause of squeal is relevant and should be taken into account.

Mode	Frequency $[Hz]$	Velocity of coupling $[m/s]$
1	24.759	14.85
2	25.116	15.06
3	26.256	15.75
4	38.749	23.24
5	48.386	29.02
6	82.899	49.73
7	87.851	52.71
8	133.11	79.86
9	141.63	84.97
10	146.99	88.19

Table 5.2: Traveling speeds that cause coupling

5.4 Stick-slip transitions between wheel and rail

The stick-slip phenomenon is a type of spontaneous motion that can occur while two objects are sliding over each other. As the name indicates, this phenomenon involves two surfaces alternating between two states: sticking to each other and sliding over each other. Normally, the static friction coefficient between two surfaces is larger than the kinetic friction coefficient. If a force large enough is applied to one of the surfaces, it will start sliding and the friction coefficient will decrease from its static to its dynamic value. The sliding motion will be kept until the force applied is not sufficient to overcome the dynamic friction. Then the surface will stop and the cycle will start again. When this phenomenon occurs between the wheel and the rail, it can cause the wheel to oscillate and radiate the squeal noise.

As mentioned before, the coupling between the natural frequencies and the frequency of rotation or frequency of contact with sleepers may result in resonance and thus in a violent vibration of the wheel. This situation can lead to the stick-slip phenomenon and thus generate the squeal noise.

The stick-slip phenomenon can also appear when a train traverses a curve passage or a switch between rails. During the curve passage, some wheels can suffer from lateral creepage. Lateral creepage is produced when the movement of the wheel is not perfectly aligned with the direction of the curve, and it is defined as the tangent of the rolling angle α. The rolling angle is the angle between the rolling direction and the direction of the movement, as shown in Figure 5.4. The creepage can show unstable stick-slip behavior in the contact area between wheel and rail [1].

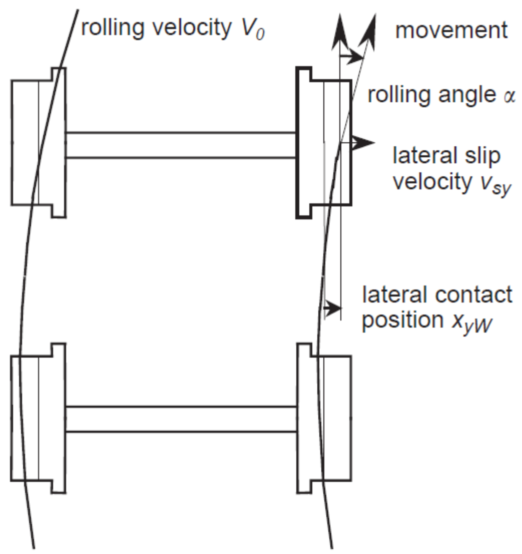

Figure 5.6: Lateral contact position and creepage [1]

This phenomenon is not completely understood, as too many variables are involved and the relationships between them are complex. Some experimental studies as [7] have tried to develop measuring techniques to identify the wheels causing squeal during railway operations and some others as [1] and [8] have tried to model the dynamics of the system in order to predict the squeal appearance. A large amount of searching on the phenomenon is still to be done.

6 Conclusions. Future work

As it has been mentioned several times along this document, the squeal noise is a very complicated issue for rail systems all across the world. A brief literature research is presented in [8] and points out that several and varied causes for the squeal noise have been proposed in the literature:

- Imperfections in rail joints and wheel geometry
- Cone form of wheels that results in different linear speeds of the left and right wheel
- Periodical structure of rails with sleepers and the instability of motion over periodically placed supports
- Contact problem between wheels and rails, with stick and slip zones that generate waves which deform both contact surfaces
- Non-linear friction in the stick zone
- Influence of material hardening
- Deformation of wheel/axle system as a result of impacts during rolling motion
- Strong hits of perfectly round wheels on imperfect rails
- etc.

Not all these hypothesis have been proofed yet and many others have been proposed and then discarded after some study.

At the same time that the research moves forward, the industry has been working on trying to solve the problem in-situ. Several companies have discovered that improving the lubrication system in critical points such as tight curves or switches reduces the squeal noise significantly so they have installed special lubrication stations at these locations. This would reduce the squeal noise produced by causes related with friction and the relative motion between the wheel and the rail.

Another possible solution to eliminate the sleepers as a probable cause of squealing could be to place them randomly instead of regularly. Then the frequency of contact with the wheel will not be constant and there will be no chance of coupling with any of the natural frequencies of the wheel.

The squeal noise is a problem that will increase as transportation systems keep increasing along with the world's population. It is expected that the research of this phenomenon and all its possible causes will also increase until some explanation and feasible and reliable solutions are found.

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A Work distribution

The work distribution of this project was as follows.

- Mainly, Samuel dealt with the *Abaqus* simulation, i.e creating the wheel geometry, setting up the simulation requirements (material, steps, etc..), assigning the boundary conditions and obtaining the results. Similarly, Samuel also focused on the presentation of the project, preparing all the slides.

- On the other hand, Magdalena focused on the writing of the report, proposing the different points that the project should include and also the information organization, i.e. introduction, mathematical background, methodology, etc.

Even though it might seem that we have worked quite independently, the truth is that we mainly discussed together the basic points of the project such as boundary conditions for the simulation and information distribution of both report and presentation.