UNIVERSITAT POLITÈCNICA DE CATALUNYA
BARCELONATECH

Escola Tècnica Superior d'Enginyers de Camins, Canals i Ports de Barcelona

MSc Computational Mechanics

Dynamic Analysis of Train Wheel

Computational Mechanics Tools

Author:
Raúl Bravo
Shushu Qin

Supervisor:
Dr.Amir Abdollahi

Contents

1 Introduction

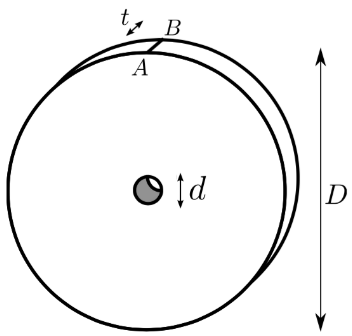
When a train is travelling, it often emits highly audible noise between 2 and 8 kHZ, known as squeal, which is generated by the train wheel during its rotation. The noise can be quite annoying to nearby residents. There is a range of sources for the squeal. It can be generated when a rail car negotiates a curve of short radius, which is called "Curve squeal". During a curve passage, some wheels rub with the flange against the rail, causing the intense tonal noise. Other wheels, (for example, leading inner wheel of a bogie) perform lateral creepage because the wheel movement does no align with the rolling direction[1]. The creepage can show unstable stick-slip behavior, causing the wheel to oscillate and radiate loud annoying noise. Apart from it, there is also longitudinal stick-slip which is caused by the different translation velocities between the inner and outer rails[2]. When a bogie traverses a curve the outer wheel must travel a greater distance than the inner one. For a solid axle this differential distance implies the possibility of one wheel sliding or creeping longitudinally on the rail. Squeal noise generation during braking is also a complicated problem for automobile manufacturers. This brake noise is perceived by customers as both annoying and an indication of a problem with the brake system[3]. The resonance phenomena is another source of squeal. It might happen when the wheel eigenfrequencies and the frequency of the wheel due to its rotation couple with each other or the frequency of the sleepers (transversal beams that support the rails) is around the natural frequencies of the wheel.

In this project, the author takes into consideration last two mechanisms of the wheel squeal: (i) the coupling between wheel eigenfrequencies and the frequency of wheel rotation, (ii) the coupling between wheel eigenfrequencies and the frequency of sleepers. It is aimed at checking whether the squeal noise will be generated when the train is traveling at the speed of 350 km/h and with the sleepers located every 60 cm. The dynamic analysis is done with a simple model of the steel wheel. Abaqus was firstly tried in order to compute the natural frequencies of the wheel. However, due to the low performance with the limited number of elements in Abaqus, other engineering softwares are applied in the project such as Kratos and Ansys. The author also did simulation with a more realistic model of the train wheel to see whether there is potential squeal problem for this wheel.

2 Problem Statement

In this project, two problems are dealt with:
(i)If the train travels at a maximum speed of 350 km/h, will the rotation of the wheel is coupled with the squeal problem?
(ii) If the sleepers are located every 60 cm, may the positions of the sleepers be coupled with the squeal phenomena?

The eigenfrequency analysis is done with both, a simple geometry of the wheel and a more realistic one. The geometry is generated as a three dimensional disc with the dimensions indicated in Figure 1. The material is assumed linearly elastic, with the material properties given also in Figure 1. A more realistic model is shown in Figure 2.



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

Figure 1: Simple geometry of the wheel

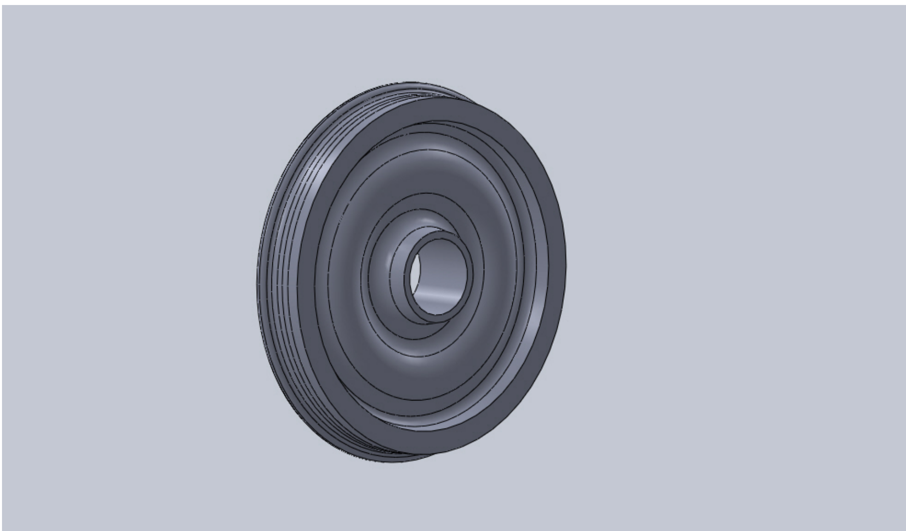


Figure 2: Realistic geometry of a wheel

3 Methodology

The process followed to obtain the dynamic analysis of the train wheel was in first instance to create a simplified model as described in the problem statement. The model was analysed and results were obtained for both, its eigen-frequencies and its corresponding critical velocities.

However, the simulation with Abaqus didn't give us reasonable results as the available number of elements is too small. When changing the element size, the results varied disproportionately. As a consequence, other engineering softwares such as Ansys and Kratos were employed to perform a mesh sensitivity test on the model. Using the mesh under the converged solution, the critical velocities were obtained again. In this project, we consider two parameters which contribute to to the squeal problem, the frequency of the wheel's rotation and the frequency of passing by the sleepers.

The frequency of the wheels own rotation is calculated by

$$f = v/(2 * pi * R)$$

where v is the velocity of the train and R is the radius of the wheel.

The second one is calculated by

$$f = v/D$$

where D is the distance of the sleepers.

The velocity that matches any of the natural frequencies of the wheel is defined as a critical velocity.

In the following sections, a discussion of the simulations is presented. Since the critical velocities found for the model of diameter 1m was close to the maximum velocity of 350 km/h, a series of simulations were performed in order to suggest improvements in the geometry that might lead to avoiding any resonance problem. For this, two new external diameters were tested to verify the correct direction of optimizing the model. An enlarged model with a diameter of 1.2 m and a model with a diameter of 0.8 m.

Finally, a series of simulations using a realistic model were undertaken. The realistic model has similar dimensions as the first proposed model in the problem statement, so it can offer a grasp of a realistic wheel of those dimensions. By comparing the results with the simplified model, we can see how efficient it is to analyse it with a simple circular plate.

The boundary conditions applied to the models are the same. These are applied on the inner area (where a shaft would be located), and on a line over the outer area (where the contact with the rail would occur). All the degrees of freedom are restricted on these parts of the model, as shown in Figure 3.

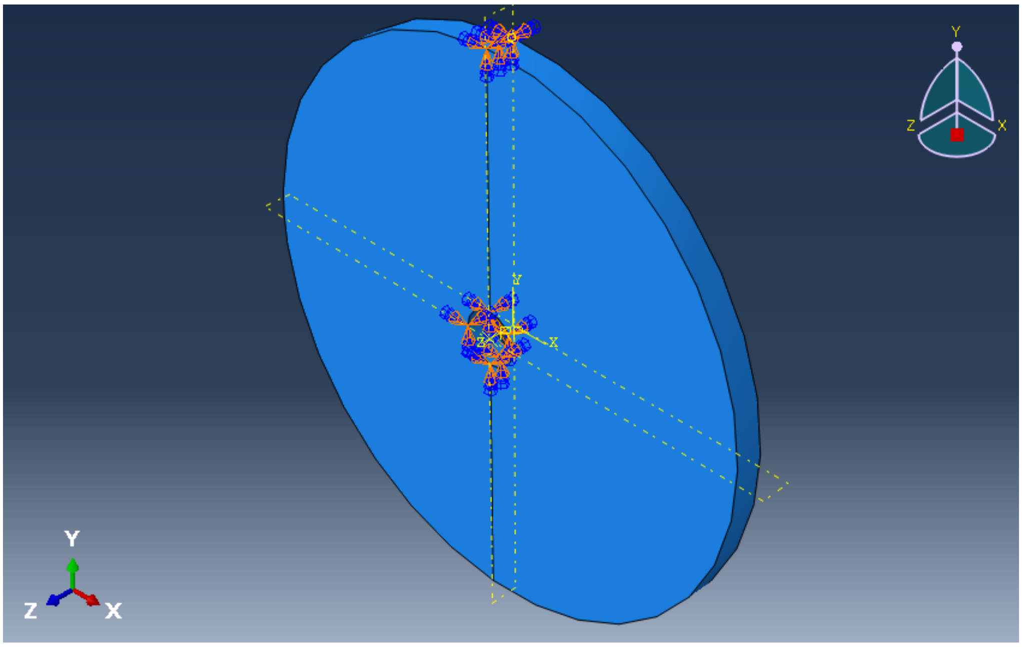


Figure 3: Boundary conditions applied to the models

4 Simulation with simplified geometry

4.1 Mesh Sensitivity

The first simulation is carried out with the simple geometry and recommended mesh in the assignment (Discretize the wheel geometry with hexahedra finite elements and using "Approximate global size" equal to 0.05). Here the type of element is set to be linear. The following plots were obtained for the first four deformation modes.

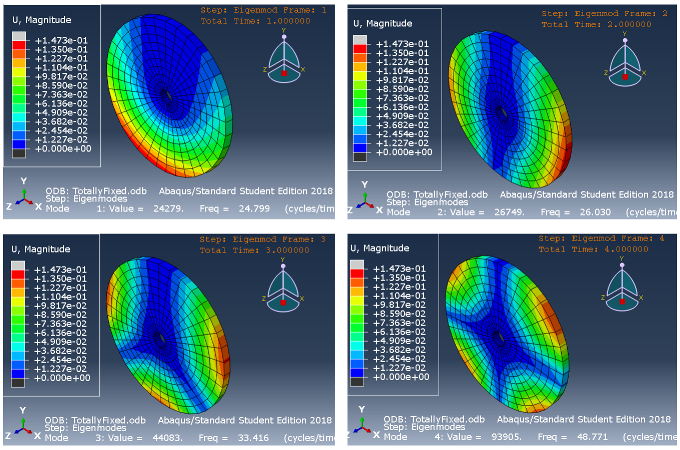


Figure 4: Modal deformation Abaqus

However, when reducing the size of the radius, it can be observed that the eigenfrecuencies increased disproportionally when the radius is around 0.3m. This is due to the fact that we reduce the element size when decreasing the size of diameter, so the first results were not accurate as the mesh is not fine enough. Notwithstanding, it was not possible to test the mesh sensitivity using Abaqus student version because of the limitation of 1000 nodes. A more accurate result can be obtained with p refinement. Therefore, another simulation with quadratic elements is done with Abaqus. Figure 6 shows that Abaqus Model using quadratic elements performed well even with a very coarse mesh while the Abaqus model using linear elements is totally out of range.

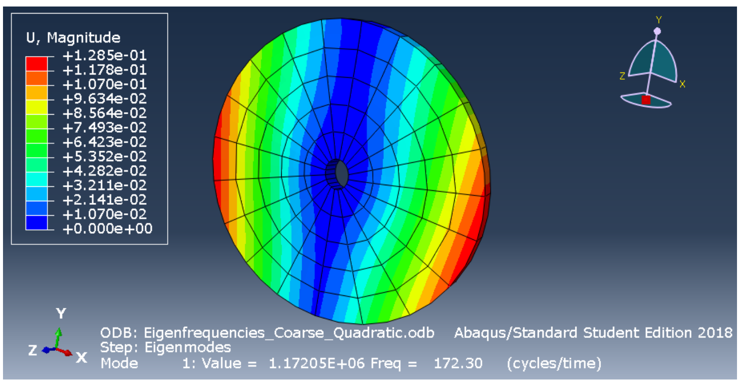


Figure 5: Mode 1 with a coarse quadratic mesh. Abaqus

In order to obtain accurate results from simulations, it is of great interest to do mesh sensitivity. The decision was taken then to use other softwares which allow a larger amount of nodes, in order to do the mesh sensitivity analysis and find the converged results. The softwares selected were Ansys, which in its academic version allows to use up to 32K nodes; and Kratos, which is open source, and therefore does not state a specific limit for the number of nodes. The same procedure explained in Methodology was followed, including the specification of the material properties, the geometry domain, and the boundary conditions. Besides the amount of nodes, the degree of the elements is a factor to consider. Ansys by default meshes using quadratic elements. Kratos also allows the usage of quadratic elements. The results are also compared with those from Abaqus with limited number of elements. Figure 7 shows that the first natural frequency of the model converges to a value of around 165 Hz for both Kratos and Ansys. One can also see from Figure 7 that Abaqus model with quadratic elements is off by around 7 Hz, which is remarkable taking into consideration the coarse mesh employed, and highlights the efficiency of using higher order elements.

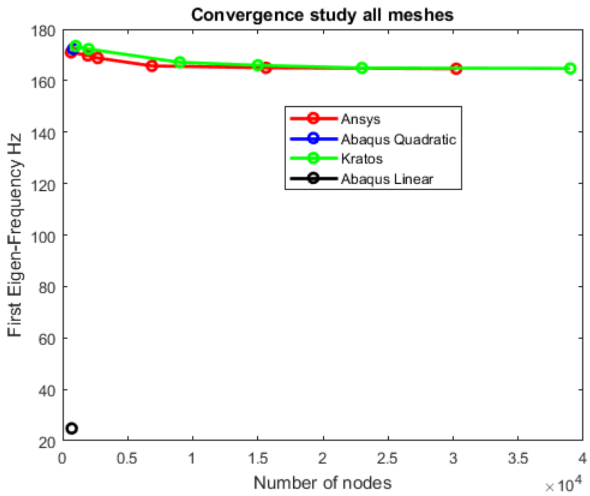


Figure 6: Convergence study for all element types

Figures 8 and 9 demonstrate the corresponding first four deformation modes in Ansys and Kratos with the finest meshes. It can be observed that the results differ greatly from the ones obtained with Abaqus using linear elements (As by the indications of the project statement). One detail that can be easily noticed is that the deformation for modes 1 and 2 get switched for a converged solution in contrast with the Abaqus solution.

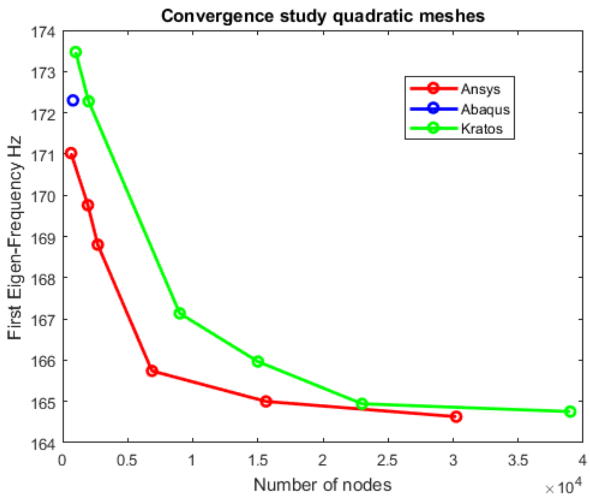


Figure 7: Convergence study with Kratos, Ansys and Abaqus(Quadratic)

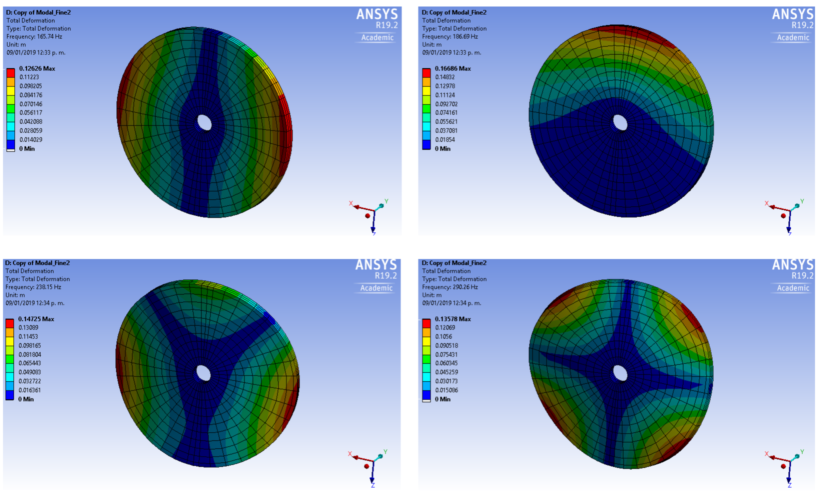


Figure 8: Modal deformation Ansys

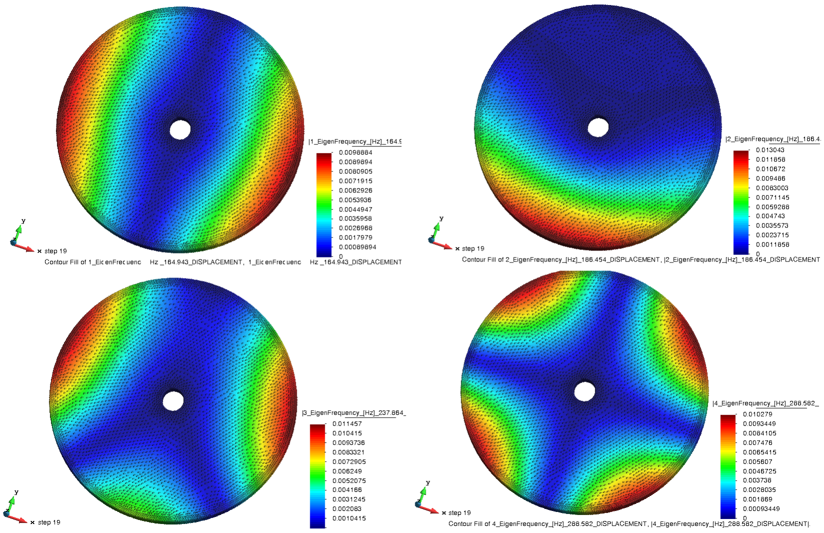


Figure 9: Modal deformation Kratos

4.2 Critical Velocities

The converged model in Kratos gave the natural frequencies shown in table 1. In Figure 10, the green lines are the first four natural frequencies. The critical velocities are found at the cross point between the natural frequency and the frequency of the wheel passing by the sleepers or the frequency of the rotation. The first critical velocity happens due to the distance of sleepers. It is around 350km/h. Since the maximum velocity of the train is 350 km/h, there might be a problem of resonance because it is too close to the first natural frequency. As for the other critical velocities, it is not of interest since they are much larger than the highest velocity. A study will be done in the following section to test whether a change of radius would have an effect on the simplified geometry, in order to allow for the velocity of 350 km/h to be reached without generating squeals.

Mode	f	v
1	164.75	356.40
2	186.41	401.76
3	237.79	511.92
4	288.12	622.08
5	478.85	-
6	624.31	-
7	669.17	-
8	819.54	-
9	1066.13	-
10	1122.04	-

Table 1: Eigen frequencies of the simple model

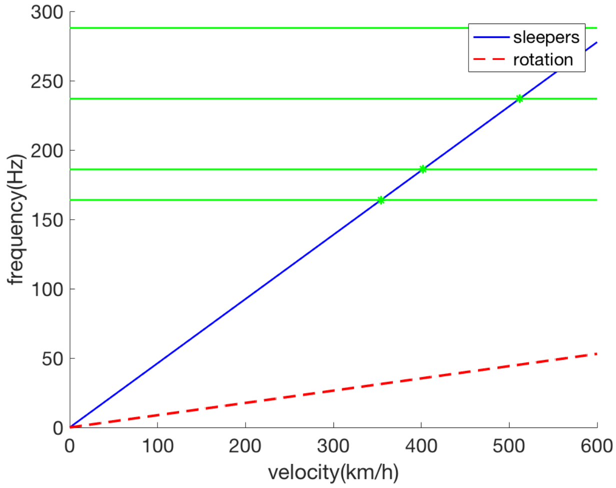


Figure 10: Critical velocity of simplified model

4.3 Optimization of the wheel

From the previous section, two options exist for the software to study the natural frequency of the wheel model. In this section, only Ansys will be used; as it provides a simpler way of modifying the geometries. The main objective in this section is to determine whether a wheel of different radius would allow for the maximum velocity of 350 km/h to be reached, without a problem of resonance.

Two new radius were tested. Both with a difference of 20% with respect to the original diameter of 1m, that is, the model with enlarged diameter is 1.2 m and the model with reduced diameter is 0.8 m. Both models keep the inner diameter of 0.1m. Figure 11 and Figure 12 show that the changes in the diameter do have influence on the critical velocities but the frequency of the train passing by the sleeper dominates. Therefore, in order to reach 350km/h, it is necessary to decrease the slope of the blue line in the figure i.e. increase the distance between sleepers so that the train can never reach the critical velocity.

Mode	f (R=0.4m)	v (R=0.4)	f (R=0.6)	v (R = 0.6m)
1	484.74	1047	196.85	425.32
2	553.65	1196	235.05	507.72
3	705.01	1523	312.77	675.62
4	854.43	1846	386.82	835.83
5	925.3	-	551.86	-
6	1294.3	-	621.49	-
7	1756.6	-	833.26	-
8	1835.8	-	1038.1	-
9	2069.8	-	1119.3	-
10	2100.5	-	1255.2	-

Table 2: Eigen Frequencies for wheel with different radius

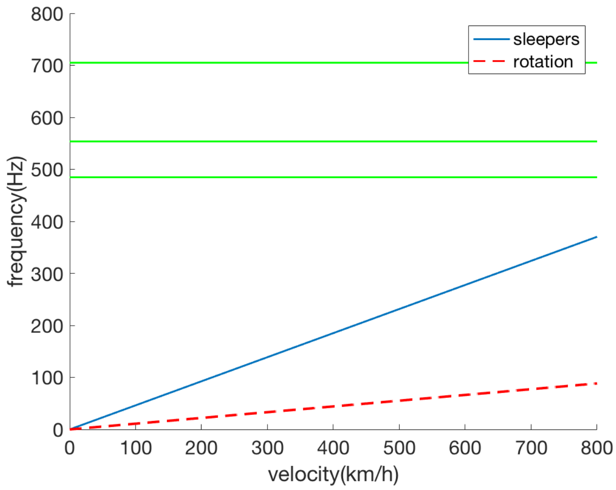


Figure 11: Critical velocities for wheel R=0.4

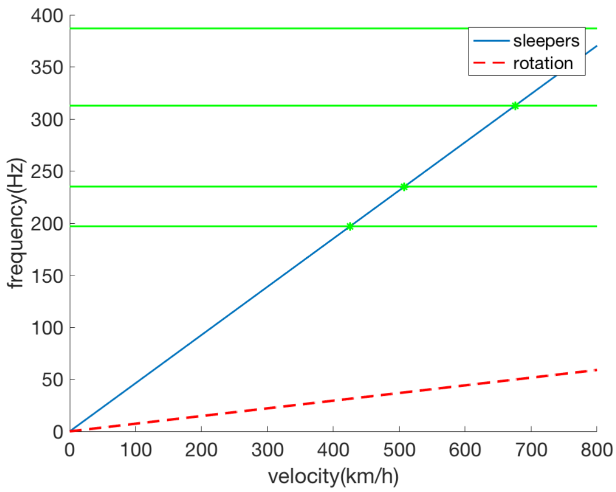


Figure 12: Critical velocities for wheel R=0.6

Both changes in the diameter of the wheel have the desired effect to raise the natural frequency. However, there is a limit on how much one can modify this simplified geometry to obtain reasonable results. For example, another geometry was attempted with a diameter of 1.6 m. The model can be seen in figure 13.

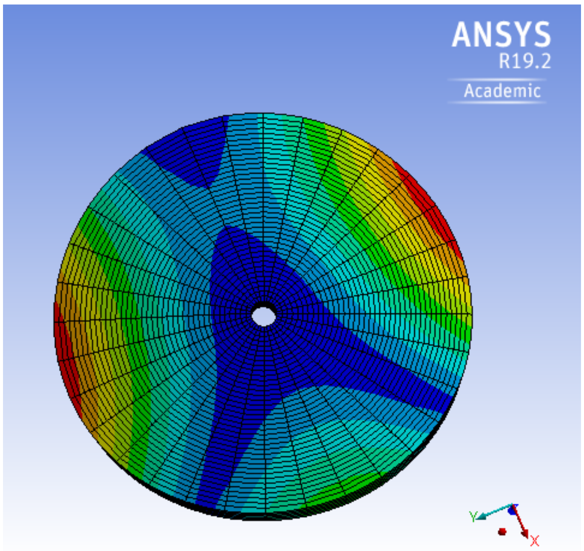


Figure 13: Super enlarged diameter model. Outer diameter 1.6m

For the super enlarged diameter model, the first natural frequency is 104 Hz, which is a frequency that can be reached during the operation of the train as shown by Figure 10, and Figure 11; no doubt this is a poor performance. But more importantly, the ratio of diameters is too large, and this does not reassemble a wheel one can encounter in operation.

Then it was clear the need to see how a more realistic model would perform. Next section studies exactly that.

5 Simulation with realistic model

The simulation of a real train wheel is considered in this section. The objective is to find the eigen-frequencies, not of a simplified model, but of a real model of similar dimensions as the one proposed in the problem statement. The software used will be Ansys, because of its large limit size for the models, and its tools to modify geometries.

This will be performed in the following way:

- Obtain a real model of similar dimensions
- Apply boundary conditions and mesh
- Undertake a mesh sensitivity test
- Obtain the eigen-frequencies of the real model
- Obtain the critical velocities

The CAD model was obtained from the internet, from the website GrabCAD using a workbench account (`https://grabcad.com/library/train-wheel-1`). This CAD model was transformed to the format Ansys supports, which is a .xt extension. The dimensions of this model is shown in Figure 14. As can be seen, the external radius is 1m. Which matches with the original geometry of the problem statement. Moreover, the inner radius is about double of the original simplified model. The thickness is also different.

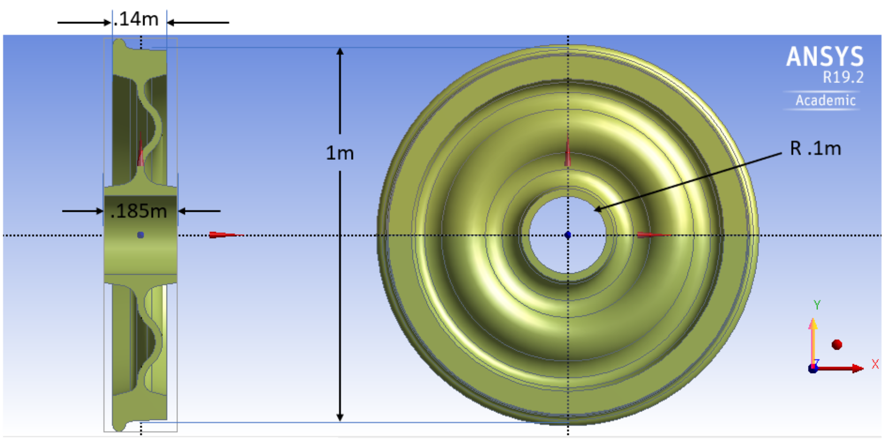


Figure 14: Real geometry dimensions

Boundary conditions are once again imposed as restricted displacement over the inner area of the wheel and also over the line where the contact zone would exist.

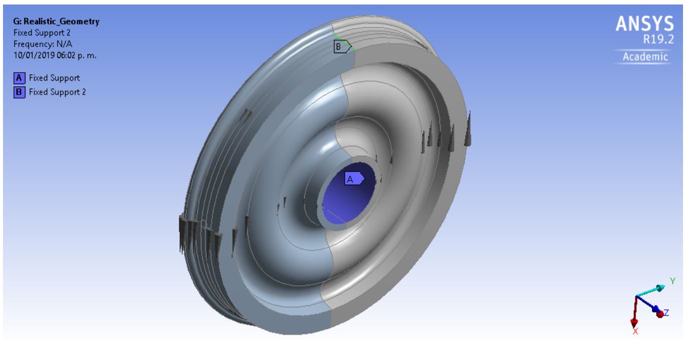


Figure 15: Fixed Supports for the Realistic Geometry

Again mesh sensitivity is done for the real model as is shown in Figure 16.

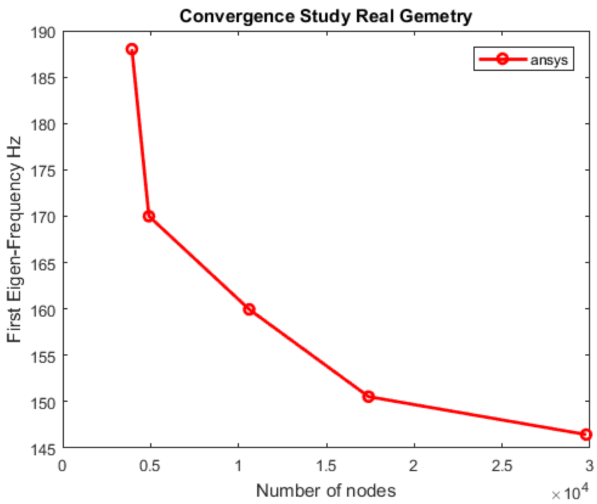


Figure 16: Mesh sensitivity study in Real Wheel

The modes calculated by the finest mesh are illustrated in Figure 17 and more details can be found in Table 17. The deformations of the first four mode are similar to those of the simplified model. The natural frequencies found are not so large, and that means that the critical velocities will be incurred during the operation of the train. From Table 17 we can see that the first critical velocity is 310km/h and the second around 380 km/h. There is high possibility that the wheel squeals when travelling.

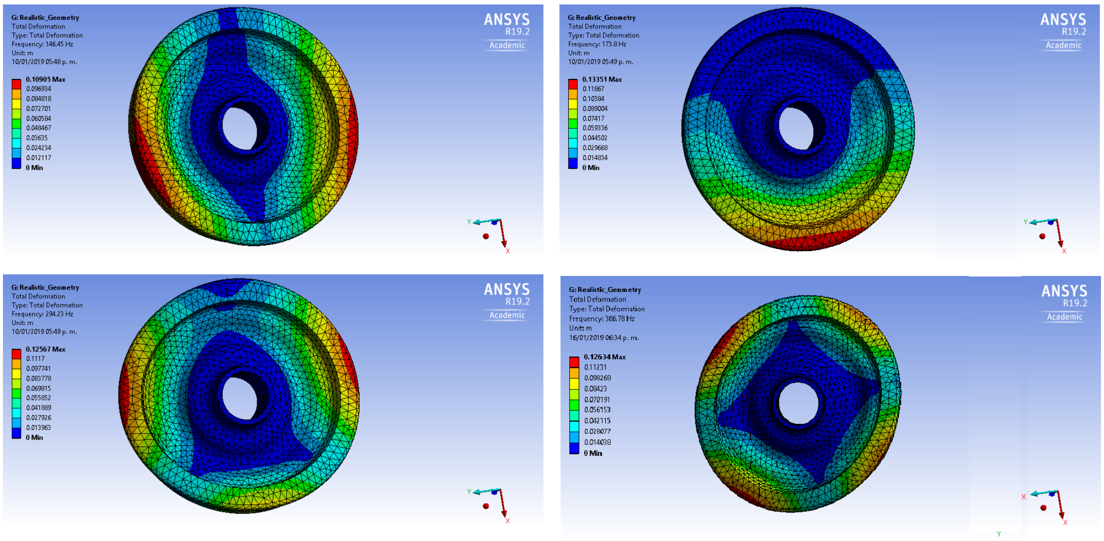


Figure 17: Modes of the real model

Mode	Frequency	Critical Velocity
1	143.45	309.85
2	173.8	375.40
3	294.23	635.53
4	386.23	834.25
5	522.34	-
6	725.84	-
7	774.88	-
8	968.58	-
9	998.04	-
10	1268.1	-

Table 3: Eigen-frequencies for the real geometry

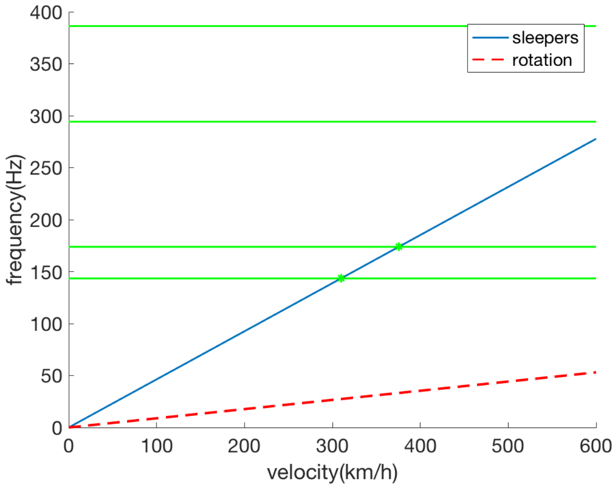


Figure 18: Critical velocities for the real model

6 Conclusion and future work

After the present study, the following conclusions were achieved:

- Both the simple and real models of the wheel would generate squeal, as they both have natural frequencies that would be reached during the operation of the train.
- Studies were performed on modified versions of the simplified geometry to check whether changing the diameter can have a positive effect on its natural frequency. The results show that slight variations on the outer radius can potentially be beneficial to avoid squeal. However, more work is to be done on the optimization of the design, using realistic modified geometries with refined enough meshes.
- Other sources of squeal can be studied and simulated, which would require much more computational power and a more sophisticated modelation to be able to capture contact, lateral creepage, among other phenomena.
- From the analysis of critical velocities, it is clear that the distance between sleepers is dominating the squeal problem, not the rotation of the wheel. Another possible improvement is to change the distance between the sleepers for new railways, and whenever possible.

References

[1] FG De Beer, MHA Janssens, and PP Kooijman. Squeal noise of rail-bound vehicles influenced by lateral contact position. *Journal of Sound and Vibration*, 267(3):497–507, 2003.

[2] CE Tickell, P Downing, and CJ Jacobsen. Rail wheel squeal–some causes and a case study of freight-car wheel squeal reduction. *Proceedings of ACOUSTICS 2004*, 2004.

[3] M Triches Jr, SNY Gerges, and R Jordan. Reduction of squeal noise from disc brake systems using constrained layer damping. *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, 26(3):340–348, 2004.

Division of the tasks

The tasks are divided as follows:

- The simulation under ANSYS and ABAQUS was done by Raul and that under KRATOS was done by Shushu.
- The report was written cooperatively. Shushu did the part of critical velocities and introduction. Bravo did the part of simulation and conclusion.
- The presentation was also done cooperatively. Shushu created the framework and insert the pictures and Bravo helped to put relevant information and comment inside.