Final Project

Dynamic analysis of train wheel related to the presence of squeal

In this project you will analyze the dynamic response of a train wheel and attempt to predict the presence of squeal: highly audible noise between 2 and 8 kHZ generated by the train wheel during its rotation.

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Summary

1- Introduction

The purpose of this project is to analyze the dynamic response of the train wheel, trying to relate its eigenvalues to the presence of the squeal. The squeal is a highly audible noise between 2 and 8 kHz generated by the train during its rotation.

Probably, the main cause of the squeal is due to the coupling between the eigenfrequencies of the wheel and the frequency of its rotation. For this reason, the problem has been treated using the commercial software Abaqus, creating the finite element model, and solving the frequency analysis with the proper boundary condition.

Furthermore, the results have been analysed deeply, associating them to the possible cause of the squeal. In particular, in the problem the following causes are developed:

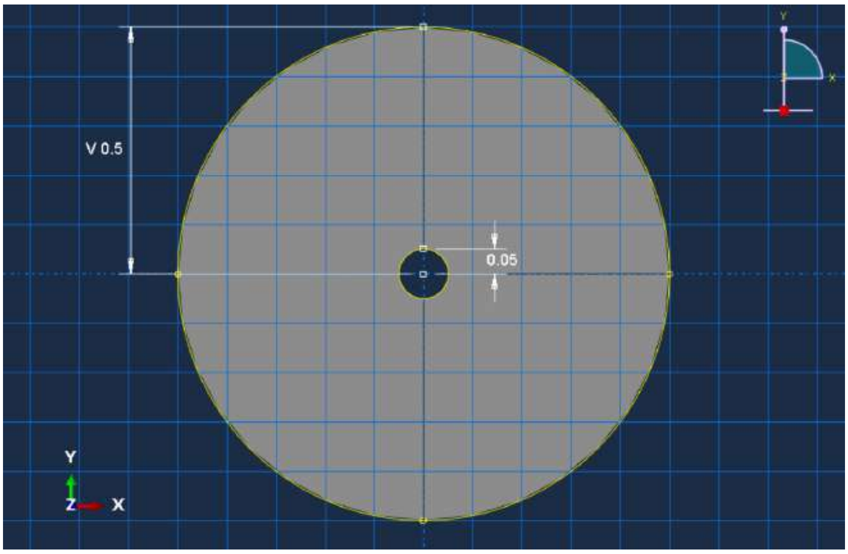
1) Coupling between the wheel eigenfrequencies and the frequency of the wheel due to its rotation.
2) Frequency of the sleepers (transversal beams that support the rails)
3) Frequency of the rail-junctions

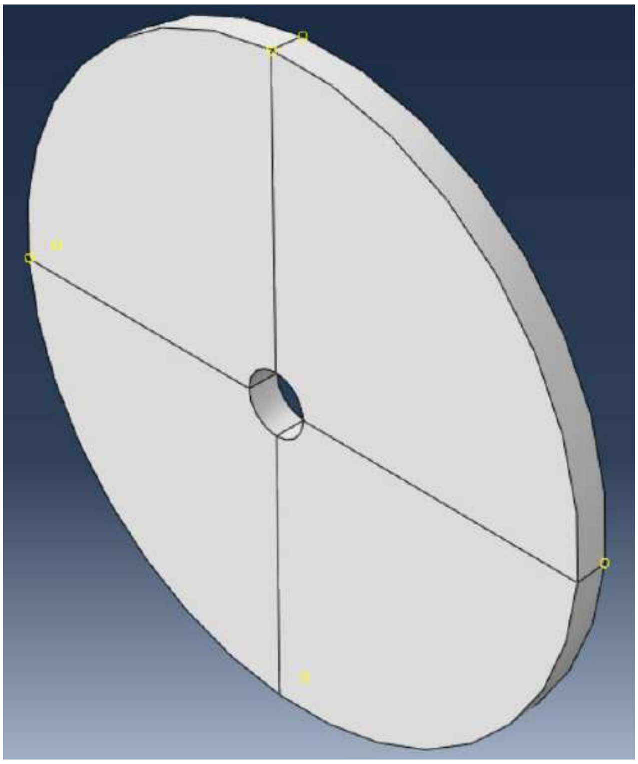
Since the squeal phenomenon happens with high speed, considering the maximum speed of the modern high-speed trains, it has been checked if this extreme velocity might be related to this issue. In particular, it has been considered the maximum speed of 350 km/h.

2- Problem statement

First of all, the geometric model with the following dimensions, has been creating on Abaqus.

Width	t	[m]	0.05
Internal diameter	d	[m]	0.10
External diameter	D	[m]	1.00





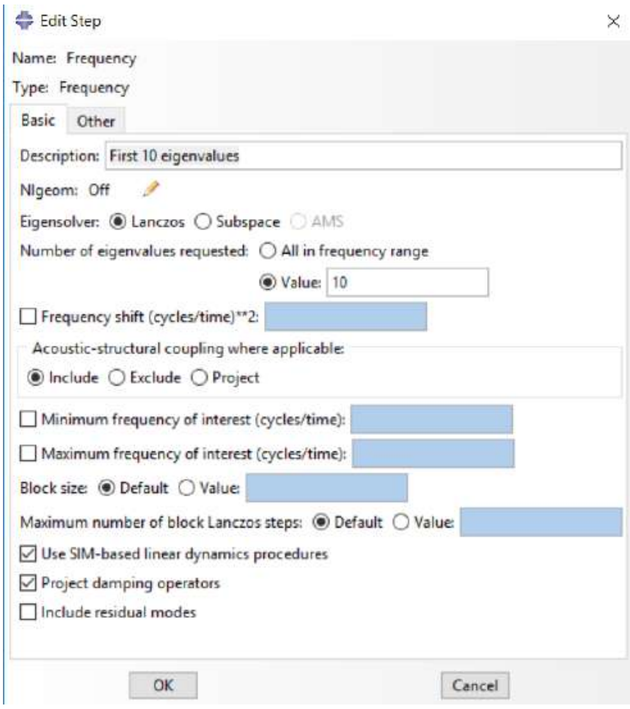
Using the extrusion it has been obtain the tridimensional geometry.

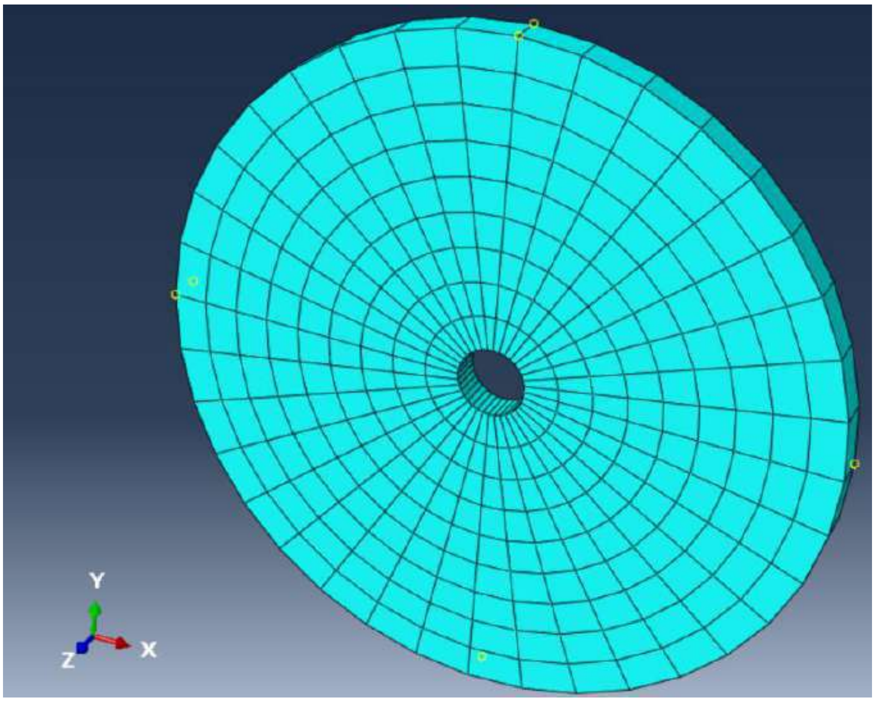
In order to obtain a structured mesh and to insert properly the boundary conditions, the model has been divided in 4 parts, using 6 datum points. It is possible to do this operation, thanks to the symmetry of the geometry.

The material has a linear elastic behavior with the following properties:

Density	ρ	$[Kg/m^3]$	7800
Yong Modulus	E	[Pa]	$210E9$
Poisson Ratio	ν	[-]	0.25

Creating a new step, with this options, the first 10 eigenvalues and eigenmodes are analysed and calculated.

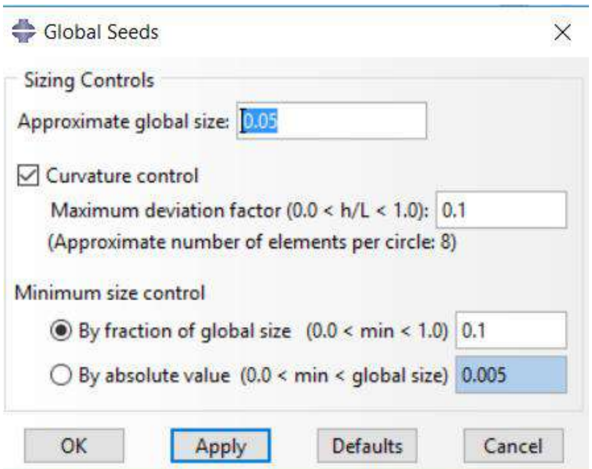




The key part to obtain good results is to create correctly mesh and boundary conditions. Indeed, this kind of problem is mesh dependent. For the shape of the wheel, the problem has been optimized, using a structured symmetric mesh.

Furthermore, more meshes are developed in the following chapter of the report.

Discretizing the wheel geometry with hexahedral finite elements and using "approximate global size" equal to 0.05, a 720 nodes mesh has been created, compatibly with the student version of the software.

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Total number of nodes: 720
Total number of elements: 324
      324 linear hexahedral elements of type C3D8R
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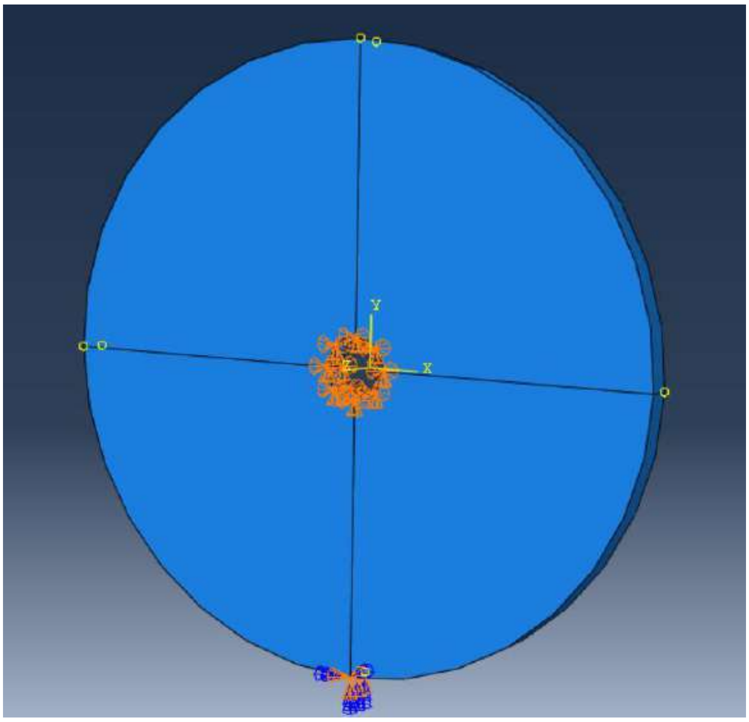
The following boundary conditions are imposed:



- Fixed displacement in the central hole of the model since in those points there is the linchpin that blocks the displacements.
- Encastred points in the lower line of the geometry, in order to simulate the instantaneous center of rotation in the motion on the rails. In this way the stick-slip is avoided.

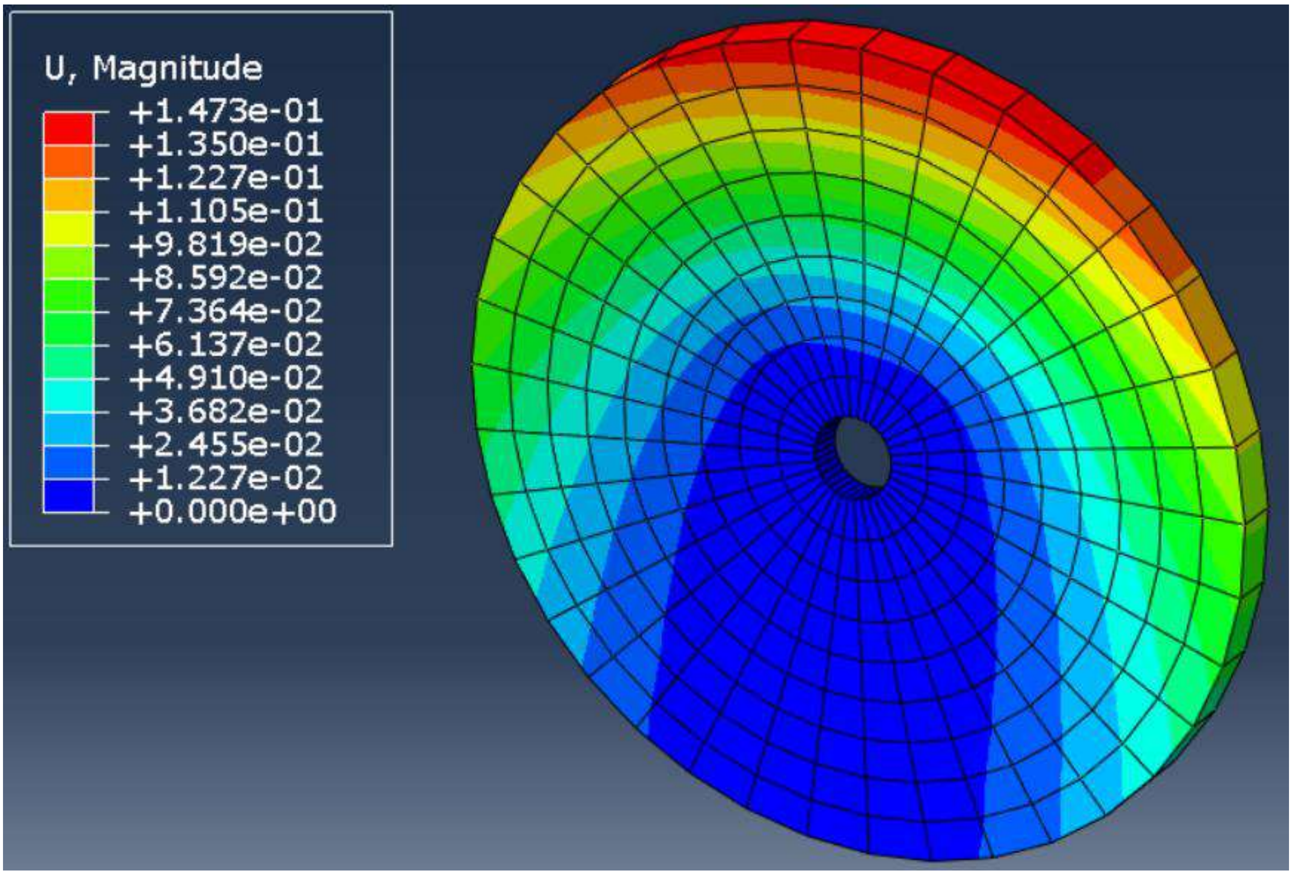
3- Results and discussion

3.1 – Eigenmodes analysis

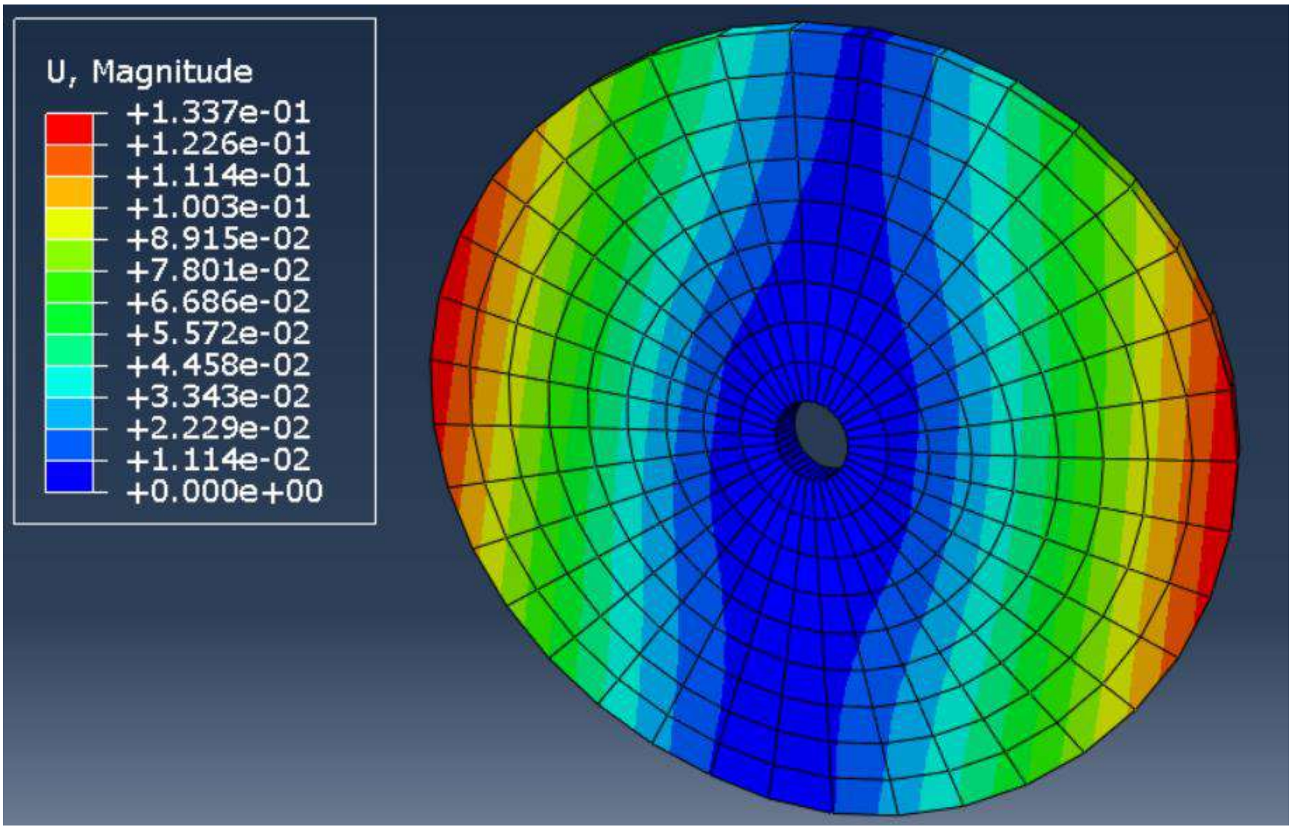
The results obtained are represented in the following table and images.

Index	Description
0	Increment 0: Base State
1	Mode 1: Value = 24559. Freq = 24.942 (cycles/time)
2	Mode 2: Value = 27138. Freq = 26.218 (cycles/time)
3	Mode 3: Value = 44174. Freq = 33.451 (cycles/time)
4	Mode 4: Value = 93651. Freq = 48.705 (cycles/time)
5	Mode 5: Value = 1.87203E+05 Freq = 68.862 (cycles/time)
6	Mode 6: Value = 3.02838E+05 Freq = 87.584 (cycles/time)
7	Mode 7: Value = 5.37297E+05 Freq = 116.66 (cycles/time)
8	Mode 8: Value = 7.23372E+05 Freq = 135.36 (cycles/time)
9	Mode 9: Value = 8.41635E+05 Freq = 146.01 (cycles/time)
10	Mode 10: Value = 1.00779E+06 Freq = 159.77 (cycles/time)

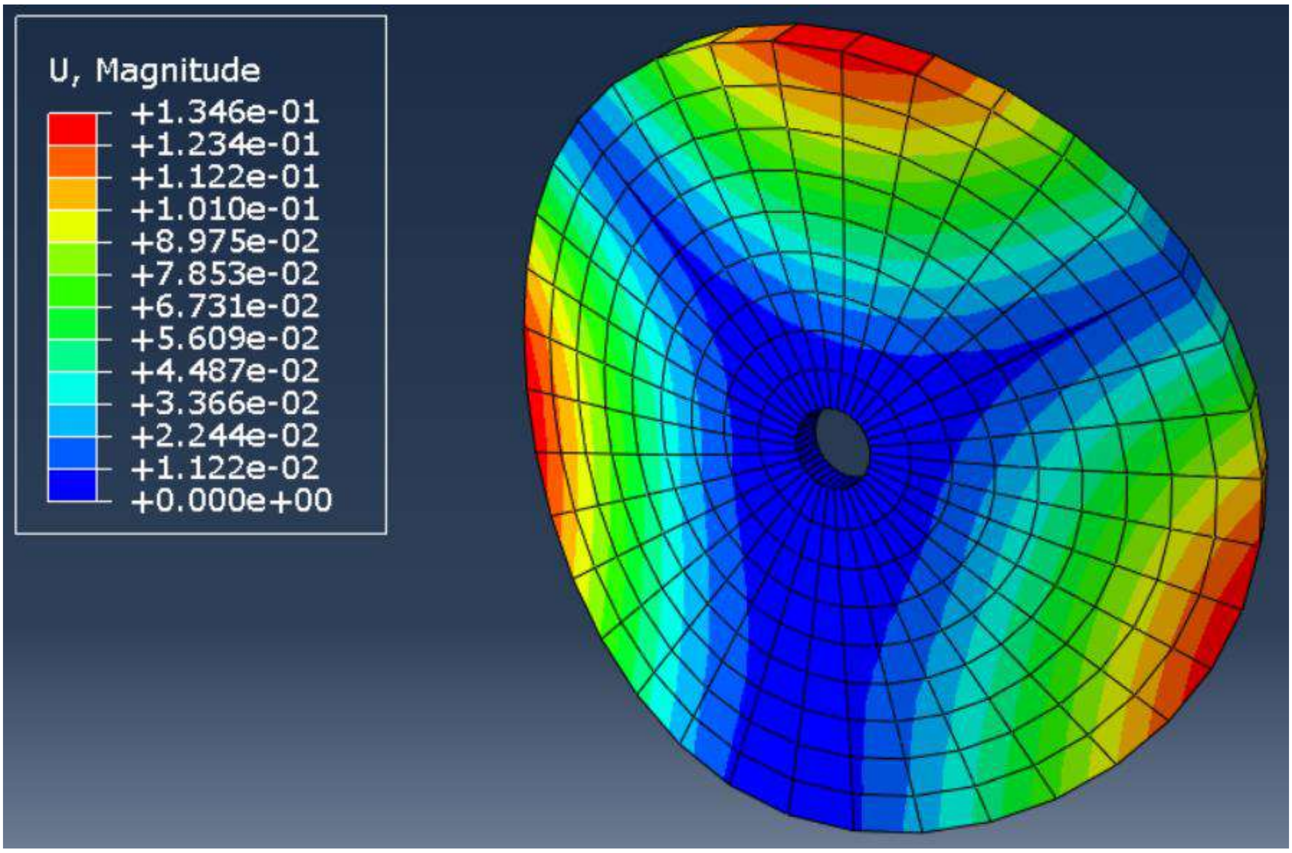
In particular the deformations of the first four eigenmodes are represented in the following images.



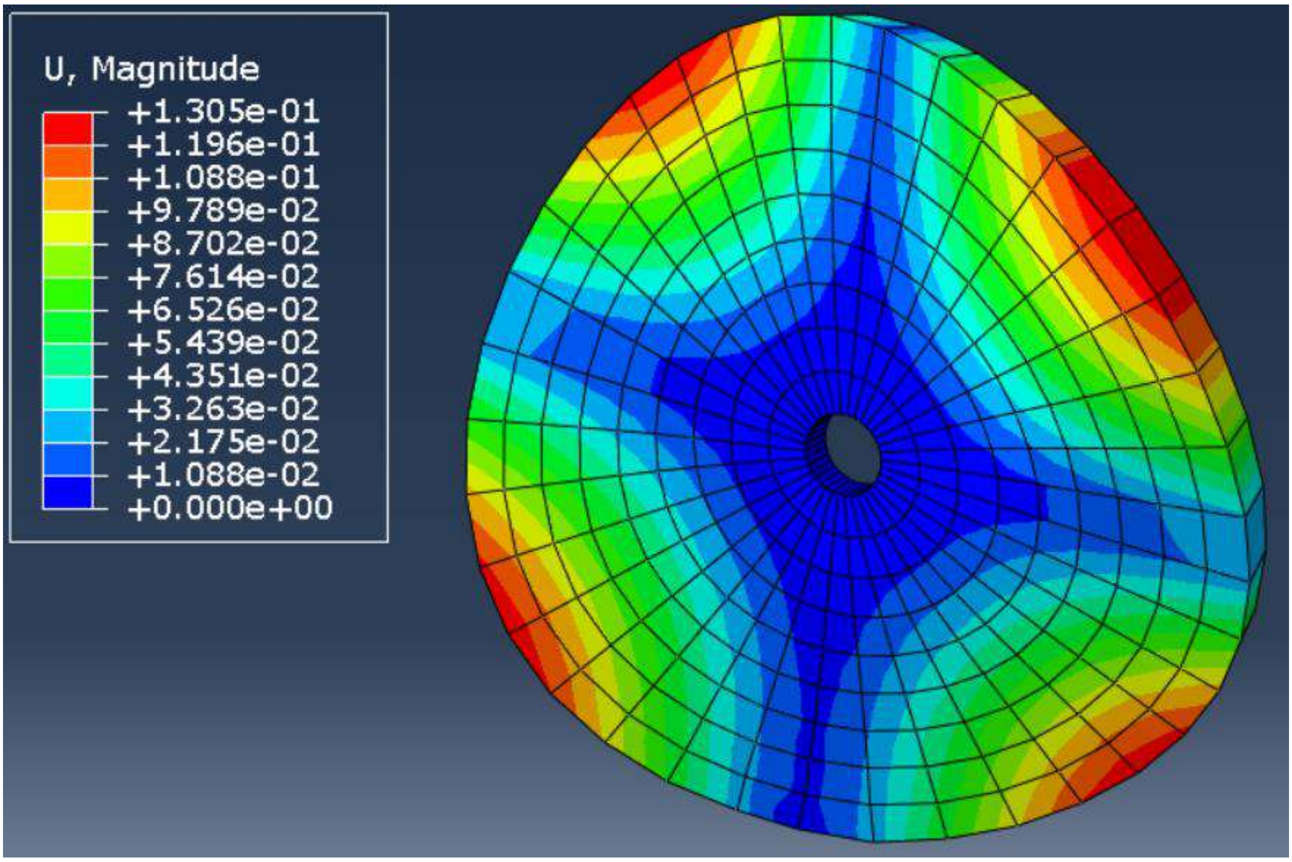
Deformation of first mode. Eigenvalue equal to 24559



Deformation of first mode. Eigenvalue equal to 27138



Deformation of first mode. Eigenvalue equal to 44174



Deformation of first mode. Eigenvalue equal to 93651

Using the eigenfrequencies of the wheel the corresponding critical velocity are calculated, first in meters per second and then in kilometers per hour. Considering only the eigenvalues it is possible to hypothesize the presence of a squeal for this critical velocity. In fact, when the train travels at this high speed (in this ideal model without friction and using a simplified shape geometry), the wheel is subject to strong stress and strain that can cause the phenomenon of resonance. The formula used to find the result is:

$$v^{crit} = f^{crit} \cdot 2\pi R$$

Frequencies [1/s]	Corrispondig velocity [m/s]	Corrisponding velocity [km/h]
24,942	78,31788	281,944368
26,218	82,32452	296,368272
33,451	105,03614	378,130104
48,705	152,9337	550,56132
68,862	216,22668	778,416048
87,584	275,01376	990,049536
116,66	366,3124	1318,72464
135,36	425,0304	1530,10944
146,01	458,4714	1650,49704
159,77	501,6778	1806,04008

From the table it is easy to see that only the first three values are reachable in the reality, the others are only theoretical. This is the reason why, traveling at these velocities, it is necessary to follow most restricting laws. Indeed, the squeal may be a minor problem, but also more dangerous issues might damage the mechanical part of the train until reaching the derailment.

3.2- Analysis for high-speed train

In the real case, high-speed trains can reach the maximum velocity of 350 km/h. It could be interesting applying the previous method in the opposite way, starting from a known velocity and finding the corresponding frequency.

Train velocity [km/h]	Train velocity [m/s]	Corrisponding frequency [1/s]
350	97,22222222	30,96249115

The found value does not correspond to any critical mode. On the other hand, to reach this velocity, the trail need to travel at two different critical velocity found before. To be more precise, when it travels at 281 and 296 km/h, the squeal may appear.

3.3 – Further analysis

3.3.1 – Sleepers analysis (fixed distance equal to 0.6 m)

Another cause of squeal might be related to the positioning of the sleepers. Indeed if the frequency of the wheel is coupled with the frequency of the sleepers, the phenomenon of resonance may appear. Assuming the distance between the sleepers equal to 60 centimeters, the corresponding critical velocity is calculated with the expressions:

$$f_{sleepers}^{crit} = v_{train}^{crit}/0.6$$

$$v^{crit} = f_{wheel}^{crit} \cdot 0.6$$

Sleepers critical frequency [1/s]	Critical velocity [m/s]	Critical velocity [km/h]
130,5298	14,9652	53,87472
137,2075333	15,7308	56,63088
175,0602333	20,0706	72,25416
254,8895	29,223	105,2028
360,3778	41,3172	148,74192
458,3562667	52,5504	189,18144
610,5206667	69,996	251,9856
708,384	81,216	292,3776
764,119	87,606	315,3816
836,1296667	95,862	345,1032

This critical velocity are easier to reach rather than the one found before. This means that the presence of the squeal is probable using this spacing between the sleepers.

In this case the wheel touches the sleeper in a different position of the perimeter. For the symmetry of the geometry it is not import to touch each time the same point of the wheel to reach the resonance. The vibration inside the material does not depend on the direction of the load.

On the contrary, if we want to consider the effect of the sleepers that touch every time the same point of the perimeter of the wheel, we can hypnotize that the perimeter is approximately equal to 3.1 meters. In this way a point, to complete a complete round, it overcomes 6 sleepers (without touching them). For this reason, it is necessary to overcome 26 sleepers before having a new contact between the point considered and the sleeper. The expression used to calculate this iteration is:

$$f_{sleepers}^{crit} = v_{train}^{crit}/(0.6 \cdot 26)$$

$$v^{crit} = f_{wheel}^{crit} \cdot (0.6 \cdot 26)$$

Frequency interaction wheel/26 sleepers [1/s]	Corrisponding critical velocity [m/s]	Corrisponding critical velocity [km/h]
5,020376923	389,0952	1400,74272
5,277212821	409,0008	1472,40288
6,733085897	521,8356	1878,60816
9,803442308	759,798	2735,2728
13,86068462	1074,2472	3867,28992
17,62908718	1366,3104	4918,71744
23,4815641	1819,896	6551,6256
27,24553846	2111,616	7601,8176
29,38919231	2277,756	8199,9216
32,15883333	2492,412	8972,6832

The velocity values found are only theoretical and are really far from the reality. So this is not a cause for the production of squeal.

3.3.2 – Sleepers analysis (modified distance)

It is possible to improve the analysis, trying to avoid the squeal problem related to the sleepers, changing the distance between them. Since the critical velocity found before was very low, the sleeper distance has been increased, to couple the squeal issue with higher velocity, more difficult to reach for a normal train.

For a distance equal to 1 meter, the following results are obtained.

Sleepers critical frequency [1/s]	Critical velocity [m/s]	Critical velocity [km/h]
78,31788	24,942	89,7912
82,32452	26,218	94,3848
105,03614	33,451	120,4236
152,9337	48,705	175,338
216,22668	68,862	247,9032
275,01376	87,584	315,3024
366,3124	116,66	419,976
425,0304	135,36	487,296
458,4714	146,01	525,636
501,6778	159,77	575,172

In this case the squeal problem is still present even for velocity lower than 150 km/h.

For a distance equal to 2 meter, the following results are obtained.

Sleepers critical frequency [1/s]	Critical velocity [m/s]	Critical velocity [km/h]
39,15894	49,884	179,5824
41,16226	52,436	188,7696
52,51807	66,902	240,8472
76,46685	97,41	350,676
108,11334	137,724	495,8064
137,50688	175,168	630,6048
183,1562	233,32	839,952
212,5152	270,72	974,592
229,2357	292,02	1051,272
250,8389	319,54	1150,344

In this case the squeal problem appears only for quite high velocity and it is possible to avoid this issue for low-speed trains using this distance. Of course this can create other kinds of problem because the sleepers have structural importance both for the load entity and the thermal expansion.

3.3.3 – Rail Junction

Another important interaction between the structure of the railway and the wheel that can create some perturbation, is the presence of welding junction within the railway.

In this case it is used the same approach of the study of the sleepers, considering, this time, a distance of 12 meters instead of 60 centimeters (that is a standard length of a section bar for the railways).

Frequency rail junction (every 12m)	Critical velocity [m/s]	Critical velocity [km/h]
939,81456	299,304	1077,4944
987,89424	314,616	1132,6176
1260,43368	401,412	1445,0832
1835,2044	584,46	2104,056
2594,72016	826,344	2974,8384
3300,16512	1051,008	3783,6288
4395,7488	1399,92	5039,712
5100,3648	1624,32	5847,552
5501,6568	1752,12	6307,632
6020,1336	1917,24	6902,064

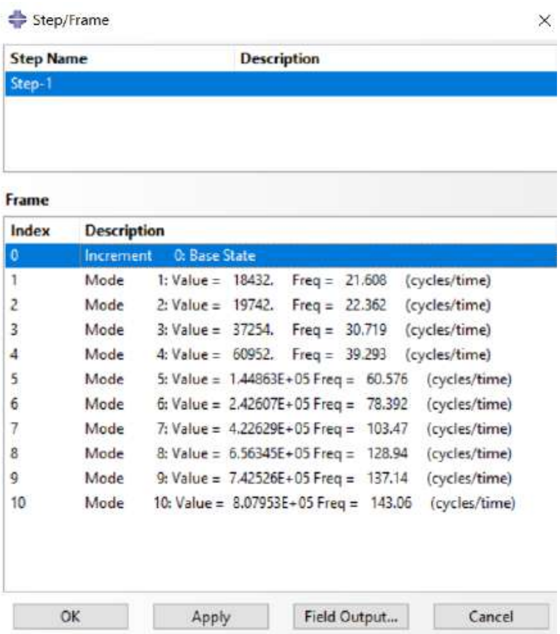
This results are meaningless in the real world, so the junctions are not a source for the squeal.

3.4 – Analysis of mesh dependence

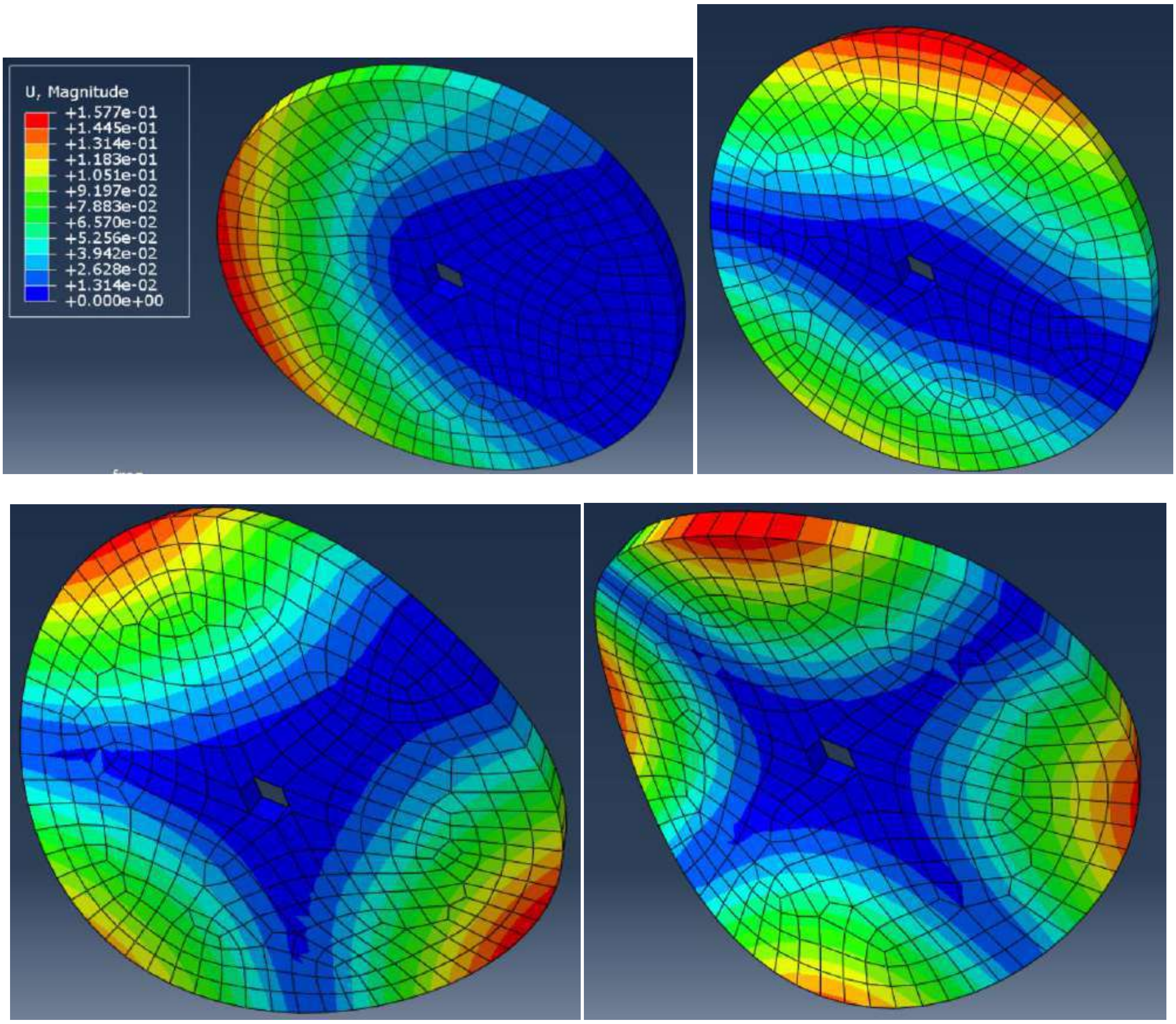
In order to prove the mesh depending of this problem, some different kinds of models have been creating. The method used to create the model is the same of before, changing the mesh option and creating

3.4.1- Unstructured mesh

As second attempt, the problem is solved with an unstructured mesh.



As the table shows, the range of the solution is the same of before but the values are quite different, on one hand for the values of frequencies obtain and on the other hand, even the deformation is similar but not the same of before.



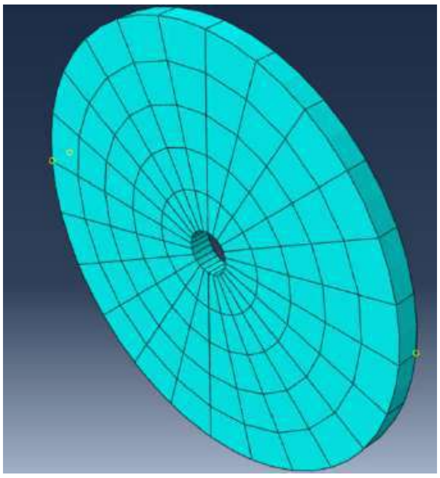
3.4.2 – Quadratic element type mesh

Trying to create a mesh with quadratic type element, with the prescribed element size of 0.05 with the student version of the software is impossible. The software created 2652 nodes, and the job cannot be submitted.

```
Total number of nodes: 2652
Total number of elements: 351
      351 quadratic hexahedral elements of type C3D20R
```

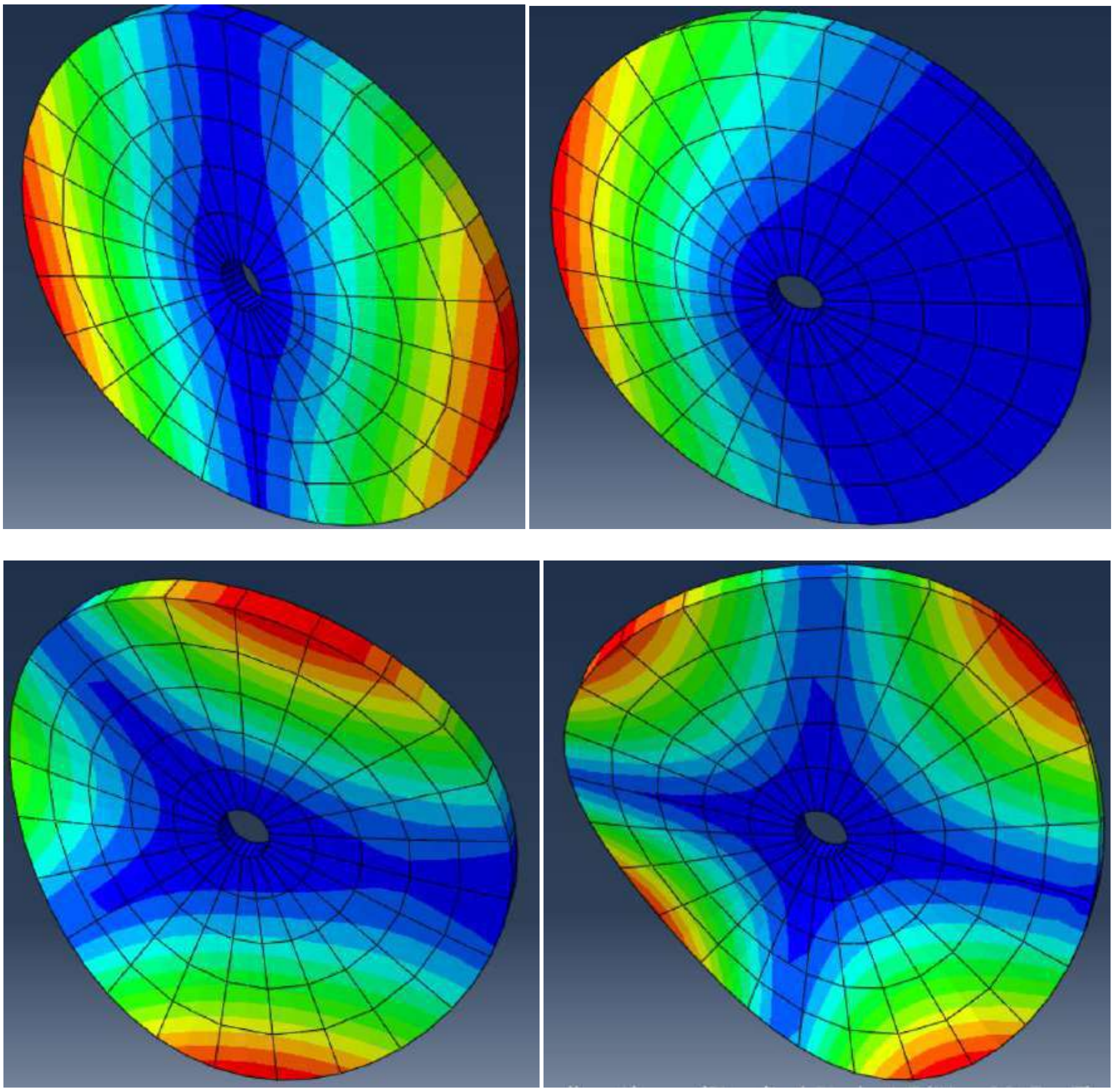
For this reason, a more coarse mesh is created. As it is possible to see in the image, the number of element is lower, but the number of nodes is higher than before.

```
Number of nodes: 960
Number of elements: 120
Element types: C3D20R
```



Index	Description				
0	Increment	0: Base State			
1	Mode	1: Value =	1.16992E+06 Freq =	172.15	(cycles/time)
2	Mode	2: Value =	1.42538E+06 Freq =	190.01	(cycles/time)
3	Mode	3: Value =	2.29518E+06 Freq =	241.12	(cycles/time)
4	Mode	4: Value =	3.57324E+06 Freq =	300.85	(cycles/time)
5	Mode	5: Value =	9.52718E+06 Freq =	491.25	(cycles/time)
6	Mode	6: Value =	1.62103E+07 Freq =	640.79	(cycles/time)
7	Mode	7: Value =	2.06815E+07 Freq =	723.79	(cycles/time)
8	Mode	8: Value =	2.85006E+07 Freq =	849.66	(cycles/time)
9	Mode	9: Value =	4.67936E+07 Freq =	1088.7	(cycles/time)
10	Mode	10: Value =	5.22837E+07 Freq =	1150.8	(cycles/time)

The results obtain are quite meaningless in terms of eigenfrequencies. There are not points in common with the previous solution. This is caused by the small number of element of the mesh. A different speech can be developed as far as concern the deformations: because of the boundary condition are always the same, the shape of the deformations for the modes are similar to that of the previous model.

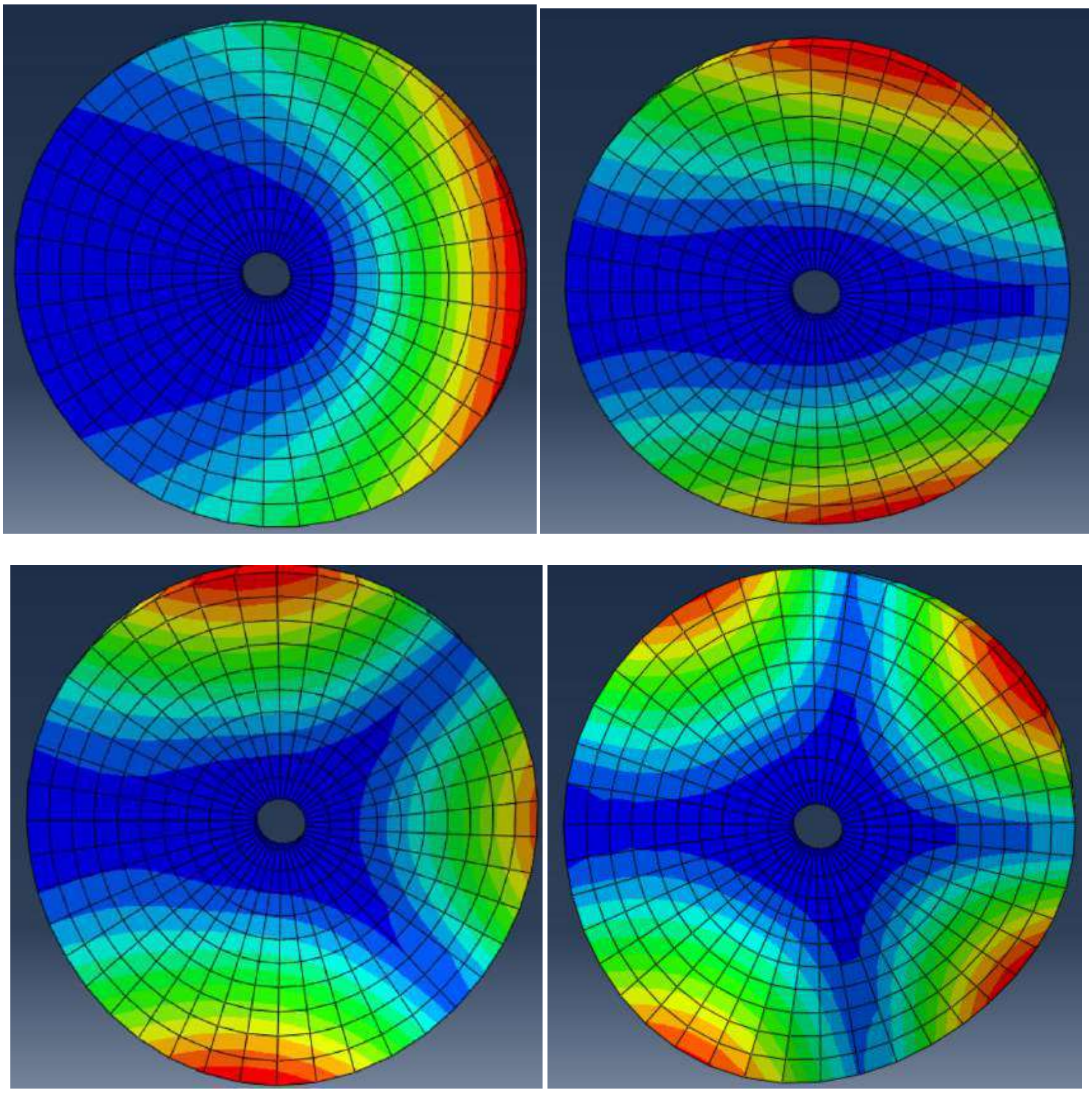


3.4.4 – More refined mesh

The convergence of the solution may be reach solving the problem with a more refined mesh. The purpose of this chapter is to compare the solution obtained before with a new solution, obtained for the maximum number of node insertable in the model. A model with 924 nodes and 420 linear type elements is created.

Index	Description
0	Increment 0: Base State
1	Mode 1: Value = 25618. Freq = 25.474 (cycles/time)
2	Mode 2: Value = 31919. Freq = 28.435 (cycles/time)
3	Mode 3: Value = 51992. Freq = 36.290 (cycles/time)
4	Mode 4: Value = 1.13060E+05 Freq = 53.515 (cycles/time)
5	Mode 5: Value = 2.24129E+05 Freq = 75.348 (cycles/time)
6	Mode 6: Value = 3.71333E+05 Freq = 96.984 (cycles/time)
7	Mode 7: Value = 6.22725E+05 Freq = 125.59 (cycles/time)
8	Mode 8: Value = 7.82726E+05 Freq = 140.81 (cycles/time)
9	Mode 9: Value = 9.99038E+05 Freq = 159.08 (cycles/time)
10	Mode 10: Value = 1.03879E+06 Freq = 162.21 (cycles/time)

The values obtained are almost perfectly comparable with the results obtained in the main chapter. This statement is valid both for the eigenvalues and eigenmodes.

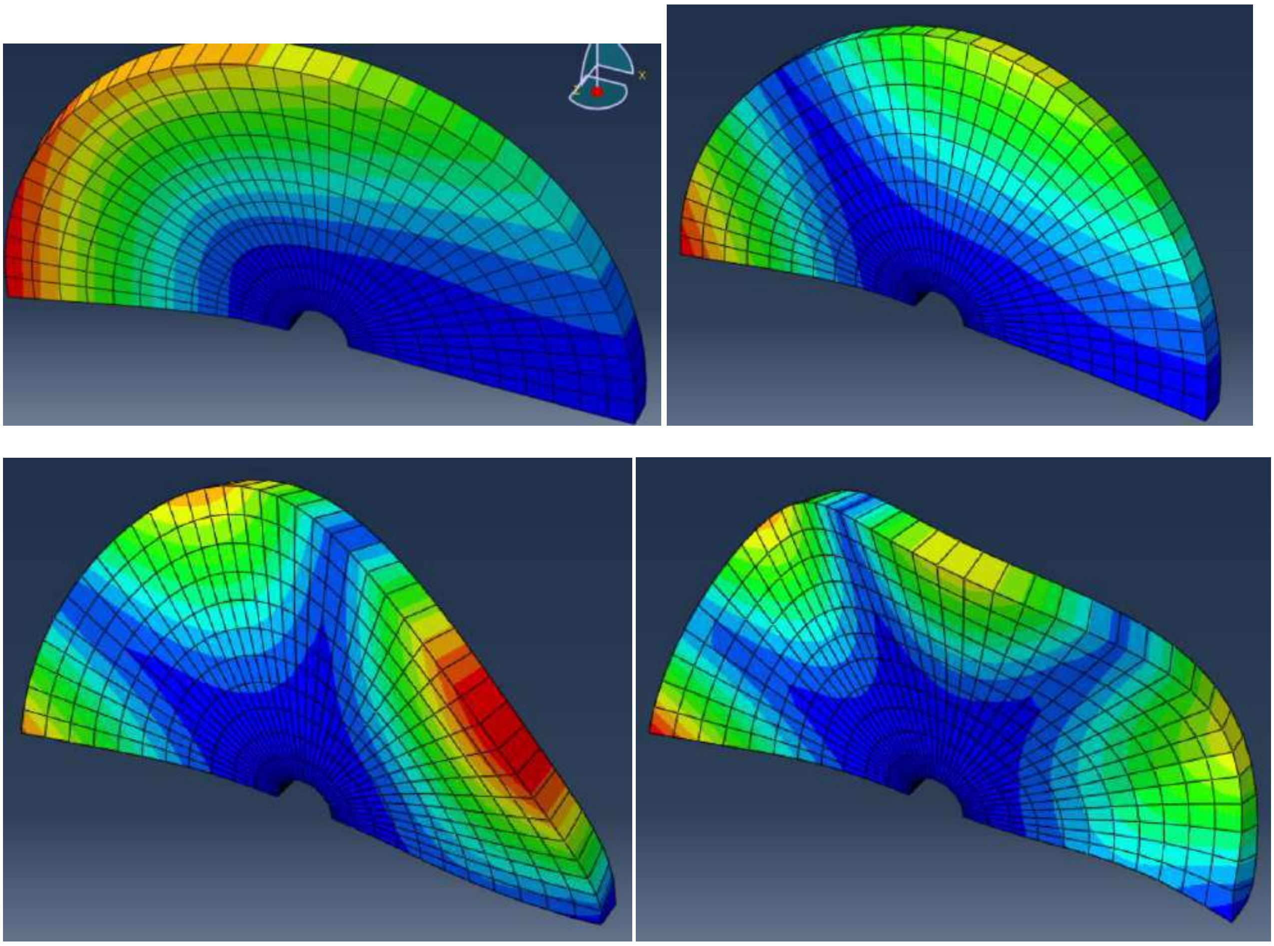


3.4.4 – Symmetry problem

Because of the symmetryc geometry of the wheel, an attempt to study just an half of wheel is compute. This is useful to reduce the computational cost of the analysis. In this case, the mesh can be even be refined, because the 960 nodes and 429 elements are concentrated only in one half of the wheel.

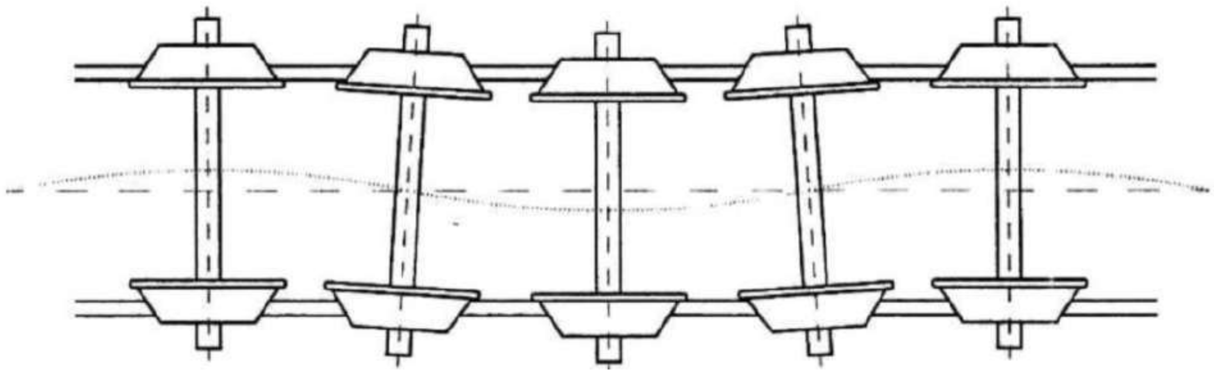
Index	Description
0	Increment 0: Base State
1	Mode 1: Value = 35817. Freq = 30.121 (cycles/time)
2	Mode 2: Value = 54620. Freq = 37.196 (cycles/time)
3	Mode 3: Value = 1.67202E+05 Freq = 65.079 (cycles/time)
4	Mode 4: Value = 3.78385E+05 Freq = 97.901 (cycles/time)
5	Mode 5: Value = 7.93520E+05 Freq = 141.77 (cycles/time)
6	Mode 6: Value = 1.09243E+06 Freq = 166.35 (cycles/time)
7	Mode 7: Value = 1.26827E+06 Freq = 179.24 (cycles/time)
8	Mode 8: Value = 1.55955E+06 Freq = 198.76 (cycles/time)
9	Mode 9: Value = 1.89825E+06 Freq = 219.28 (cycles/time)
10	Mode 10: Value = 2.87799E+06 Freq = 270.00 (cycles/time)

As the table displays, the value obtain are similar but not the same of before, especially for the highest modes. This is due to the fact that, even if the geometry is symmetric, the same is not valid for this dynamic problem. The frequency problem, indeed, depends strongly from the boundaries conditions and the shape of the model. For these reasons the results are not considered valid.



3.5 – Comments about other squeal sources

One another importance source for the squeal is created by the unstable lateral creepage between wheel and rail. Indeed the motion of railway vehicles on the track is not perfectly straight. The vehicles, mainly due to the taper of the rims and the investable small irregularities of the rolling surface and the laying of the track, proceed with a swinging motion which sends the edges of the wheels to alternatively strike against the rails.



This kind of motion depends on the velocity, on the weight and the friction coefficient. Obviously, the wheel striking against the rail, for some particular frequency, can cause the squeal.

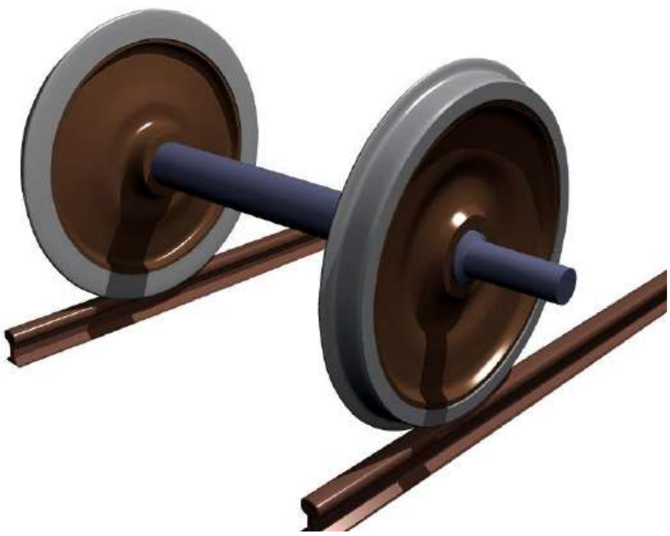
A further source of squeal is the stick-slip transitions of the wheel with respect to the rail. Stick-slip is a mechanical phenomenon that regards the sliding friction, caused by spontaneous motion and characterized by violent accelerations that occur between two surfaces in sliding contact. When there is the "abrupt" transition from the static friction (which prevents relative motion between the surfaces) to the dynamic friction (which works on the sliding of them) and then the static friction (which is greater than the dynamic friction) it lowers up to the value of the dynamic one, this phenomenon occurs.

4- Conclusions and future work

Studying the problem, we discovered that the squeal is not always related to high-velocity. This phenomenon may appear for many reason, even for structural issues as the distance between the sleepers. In particular we tried to solve this problem suggesting a new distance for the sleepers, but without ignoring the structural issues that will occur in this case.

Nevertheless, the main source of squeal is connected to high-velocity. We discovered that the eigenfrequencies of the wheel are coupling at least with velocity bigger than 280 km/h.

We analyze only theoretically other possible sources of squeal, so a future work might be an acoustic analysis of the problem (indeed we know that the high audible noises are between 2 and 8 kHz). Another possible improvement of the project, could be obtained analyzing a more realistic shape for the wheel (inserting the interaction between linchpin and wheel) and considering other sources of squeal as the lateral creepage.



Further, a more specific model, without geometry simplification, might deeply change the eigenmodes. The real shape of the wheel is not a cylinder but a truncate cone. Besides, even the presence of the lateral wings can have a non negligible important in this problem.

Another interesting consideration that can be done about this problem, is that the software version used is a limited student version. Indeed it is not possible to insert more than 1000 nodes in the model. Of course this is a big limitation for a finite element program. In this case the model in quite small, regular and simple so the results are considered correct, but in other situations it could be different. For example, using a limited number of node, it is impossible to mesh properly the wheel with quadratic type element. In theory this is the best kind of element to model circular geometries like the one we have. The quadratic shape function, indeed, are better adapted to the curve geometries. In this case the mesh is quite refined, so this difference may be neglected, and the results are still considered valid even with linear type elements.

Finally, to validate the results, the problem has been solved with a more refined mesh, obtaining similar results.

5 – References

Giunta, M. (2013). Lecture 3, Meccanica della locomozione. *University of Reggio Calabria.*

Beer de F G, Janssens M H A, Kooijman P P: Squeal noise of rail-bound vehicles inuenced by lateral contact position*. Journal of Sound and Vibration, 267, p.497-507, 2003.*

Abaqus tutorial for eigenfrequencies and eigenmodes.

Lectures slides about Dynamic.

6 – Appendix

The work has been entirely done in group, without subdivisions but always working together at the same argument.