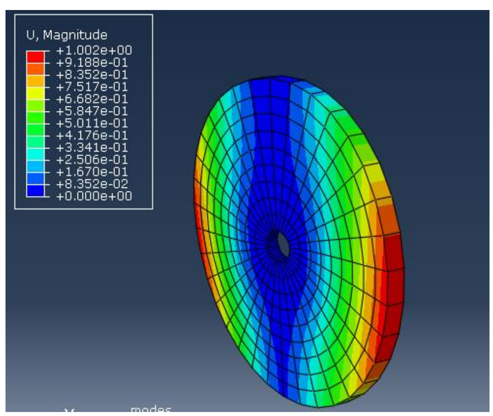
Master on Numerical Methods in Engineering

Computational Mechanical Tools

Simulation Project – Dynamics / Analysis of Train Wheel



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ABSTRACT

Interaction between train wheels and rails ignites a highly audible and unpleasant noise called squeal, which can easily elevate noise measurements up to 100 decibels in nearby areas. Environmental impact as well as nearby residential areas are in permanent contact with this noise, making it a problem to which the solution should be of pertinent matter.

Stick and slip, flanging, wheel frequency at different train velocities, and frequency of sleepers are few of the probable causes of this problem. A 3D model dynamic analysis is made in order to determine or rule out these and any more causes of the wheel squeal and furthermore propose effective solutions or mechanisms to diminish its consequences.

1 INDEX

2 INTRODUCTION

Squeal is an intense, loud, and somewhat disturbing noise. For our CMT Project we wrote the following article on the analysis of the dynamics of a train wheel, particularly focusing on the generation of a squeal.

Train wheel squeal is mainly caused by, but not limited to, a train's wheels rotation. A lot more sources can generate squeal, but since some are more highly audible than others it can make us ignore the ones with smaller frequencies. Some of the rail squeal causes can be: coupling between wheel eigenfrequency and wheel rotation frequency, frequency of sleepers, train wheel rotation, contact between wheel flange and rail, steep curves, and stick and slip amongst others.

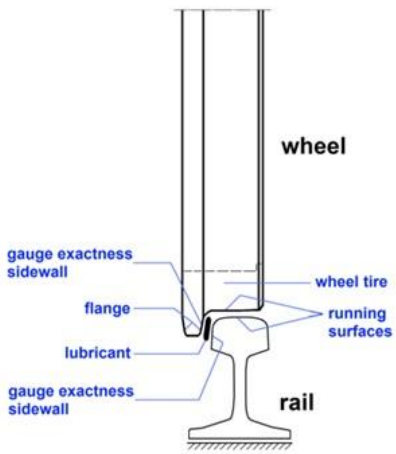


Ilustración 2-1. Components of a train wheel and rail.

Later on in the article we will analyse and discuss each of the aforementioned possible causes of squeal, which ones affect our problem and which ones can be considered negligible. First one is the coupling of the wheels eigenfrequencies and the frequency due to its rotation considering the train's velocity. If the frequency values match this can be considered our principal cause. Frequency of the sleepers, distance between the transversal beams providing support for the rail, can play a key role in our problem. If this distance is increased the rail could lose support and therefore increase vibration or deformation of the rails. Train wheel rotation could make the wheels and possibly other components of the train to vibrate at higher frequencies, so the max admissible speed of the train shall be analysed and discussed.

Other possible causes of squeal and other disturbing noises generated by the trains' components and rails could be the permanent contact between the flange and the rails, steep curves of the tracks (radii below 600m are known to cause high pitched noises according to the literature), and stick and slip regarding the static and kinetic friction between wheel and rail.

After successfully identifying the predominant causes of the train squeal we can then propose further actions to be taken, solutions, or alternatives to the problem as an initial approach. Therefore the main objective of the project will be that of recreating and analysing the dynamic response of the train wheel at certain speeds and its interaction with the rail in order to predict the presence of squeal.

3 PROBLEM STATEMENT

Interaction between train wheels and rail cause an audible squeal that can disturb bystanders and nearby inhabitants. All around the globe wherever a train is present and people living nearby there have been complaints about the wheel squeal from which case studies arise, making it a problem worth researching. Particular cases in Australia and Ontario, Canada; describe how the surroundings of train tracks have noise measurements of up to 100 dB, including measurements near residential areas, where they are most likely to cause discomfort. In both cases this noise exceeds the maximum permitted criteria by 10-15%.

As an interdisciplinary group of 2 professional engineers, mechanical and civil, we can consider several perspectives for the stated problem. As a civil engineer you must consider all variables, so it can be said the closeness of residential areas to the public transportation medium was going to present problems in the future like sound and terrain vibration. From a mechanical point of view the permanent friction between wheel and rail is going to present sometime a wear out of the components, wheel and rail, in addition of the obvious squeal.

The stated problem requires a pertinent solution for which we must before account for all the causes of the train wheel. Most studies point to predominantly a stick and slip cause, but the problem is not reduced to only this. In this project we will develop a realistic 3D model accounting for the interaction of the train wheel and the rail in order to determine if train speed, frequency, or others involved factors also trigger the squeal noise. Once the problem is perceived, causes can be identified, then approaches to a possible solution can be delivered.

4 METHODOLOGY

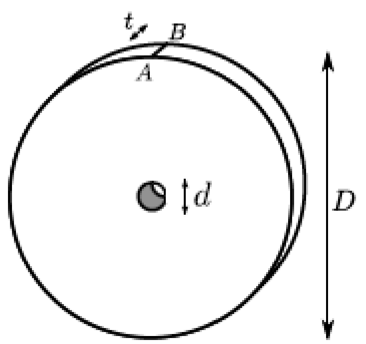
This part of the report covers the used tool, ABAQUS, and the procedures which were followed in the simulation project.

4.1 ABAQUS

Abaqus is a software suite for finite element analysis and CAE (computer aided engineering). The analysis of this project was completely executed in Abaqus, starting from the building of the CAD model to solving the problem and visualization of results.

4.2 MODELING THE TRAIN WHEEL

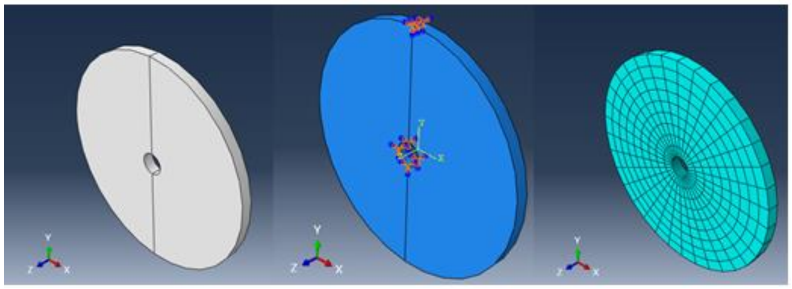
In order to study the dynamic response of the train wheel, modal analysis had to be performed. Basically the task was to create a simplified wheel model and with the use of Abaqus we will be able to specify the natural frequencies along with the vibration modes. As shown in Figure 4-1 the wheel was considered as a three dimensional disc with linear elastic properties.



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 4-1: Geometry and material properties of the wheel

In order to implement such a description in Abaqus, an extruded solid disc was created by drawing the two circles, outer and inner diameters, this way creating a surface, and then extruding them by the width of the disc. After that the material properties were assigned with same values given to us, detailed in Figure 4-1. Then, the boundary conditions were defined. Our desire was to model a train wheel while it was rotating, so there were 2 regions with a fixed boundary condition (encastre), as we can see in Figure 4-2-B. The first one was the inner surface of the wheel where it is attached to the shaft. And the second was the straight line which represents the contact between the wheel and rail. In order to define such a boundary a partition had to be made to the model as it is shown in Figure 4-2B. After that, a structured mesh with hexaedra finite elements was generated exactly as instructed, with approximated global element size of 0.05.



. *(A)* *(B)* *(C)*

Figure 4-2: Modeling stages of the wheel. (A) The extruded 3D disc, (B) the boundary conditions on the wheel, (c) The mesh.

By this point we were able to run a linear perturbation step, to determine the modes' shapes and natural frequencies. Before the job was submitted and ran, the number of the desired frequencies was set to 10.

4.3 MODELING THE RAIL

In order to investigate if the position of the sleepers is coupled with the squeal problem, a modal analysis was also performed on the part of a rail which is between two consecutive sleepers. Since it was proposed by the instructors that the sleepers are to be placed at a 60 cms distance, the rail was modeled as a beam element with 60 cms length, and (I) section as shown in figure 4-3. In order to design an accurate and realistic model we needed a technical guideline, one that was being actively used in actual railway transportation systems. Some research was done to find a technical guideline and the dimensions of the section were taken from this technical data sheet [(reference) Rail Technical Guide-TATA STEEL ®].

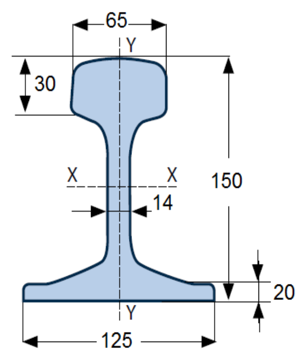


Figure 4-3: The section of the Rail

In Abaqus the part of the rail was defined as a wire of beam type, and the material properties were chosen to be the same as the wheel properties in Figure 4-1. Therefore, constraints for the displacement were set in both ends for all directions (encastre), and the mesh was generated with global element size of 0.02.

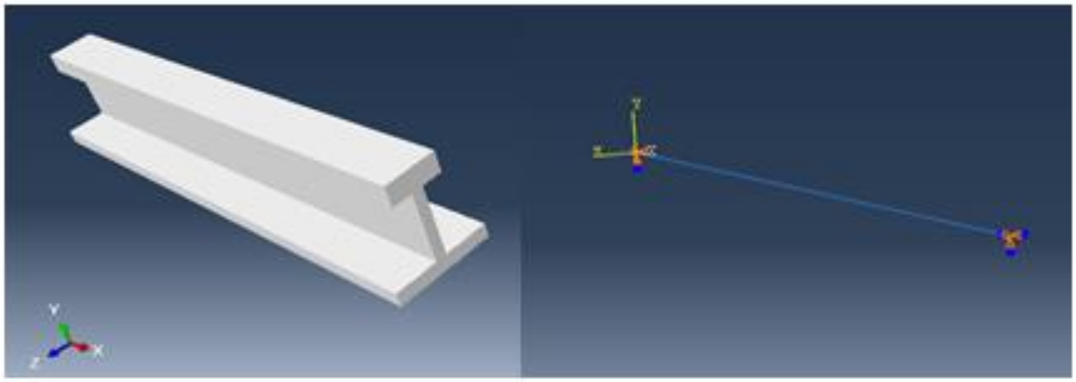


Figure 4-4: The rail model in Abaqus.

5 RESULTS AND DISCUSSION

5.1 WHEEL MODAL ANALYSIS

After calculating the first 10 eigenmodes, the first thing we noticed is that the value of the eigenfrequencies were much lower than the frequency range that was expected to generate the squeal problem. As it can be seen from Figure 5-1 the values were between 24.667 Hz (first mode) and 156.37 Hz (10^{th} mode). We even took it a step further, and in the case where the calculations were processed for a higher number of modes, 50, the frequency value was still out of the squeal range. Moreover, the boundary condition for the wheel-rail contact was manipulated by setting partial constraint on the displacement and there were still no significant changes in thefrequencies.

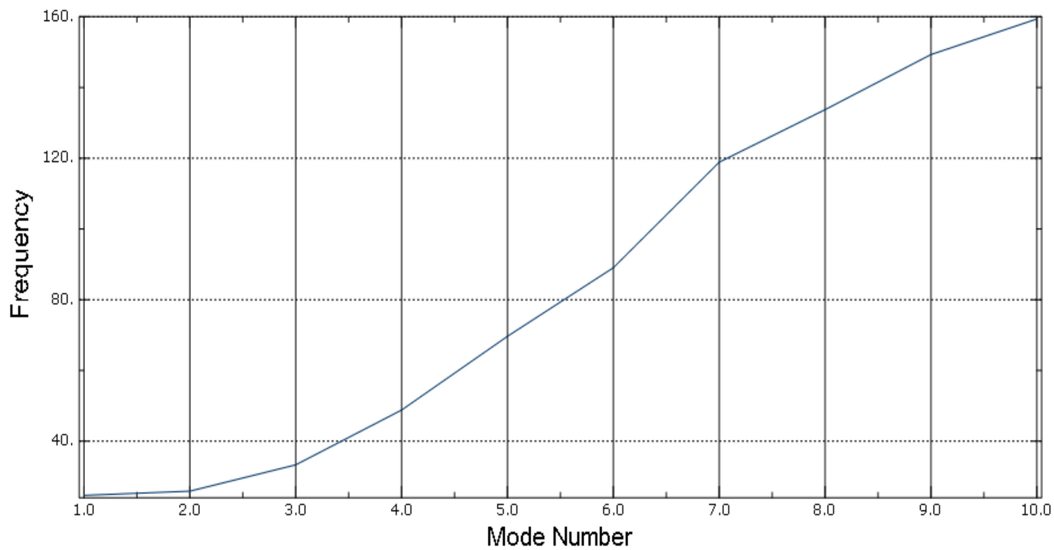


Figure 5-1: The first 10 eigenfrequencies of the wheel.

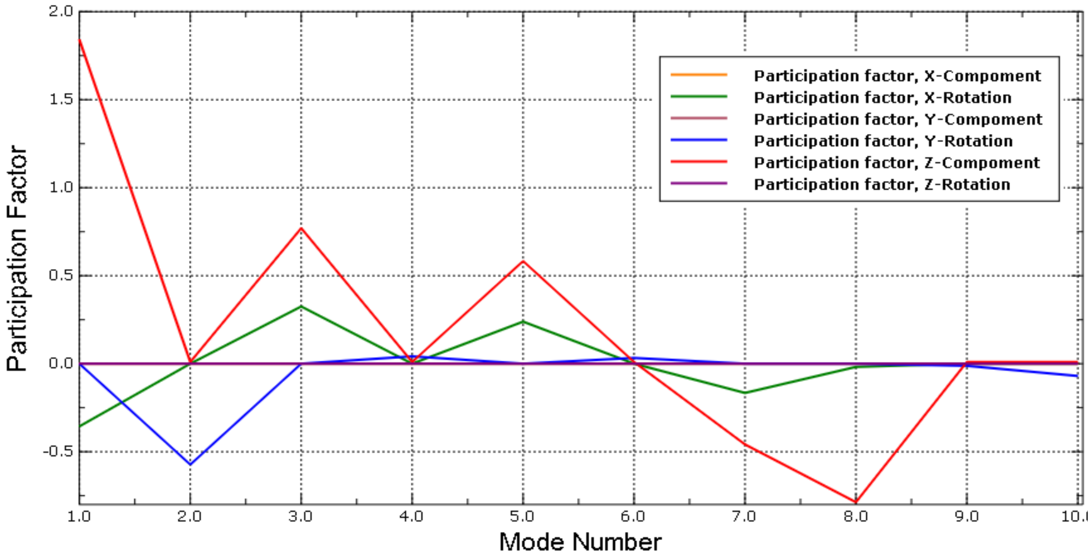
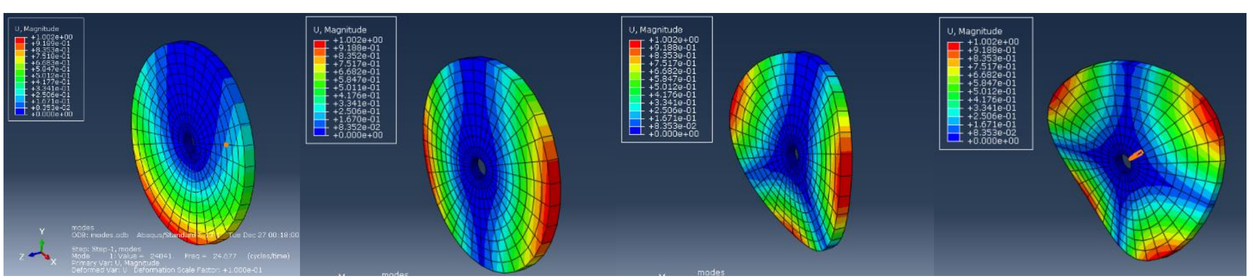


Figure 5-2: The participation Factors of each mode

The first 10 modes provided us with enough information to have a realistic visualization of the dynamic response of the wheel. For example, by taking a simple view on the participation factors of the modes, it can be deduced that the predominant degree of freedom is the Z-component (Z-axis is that around which the wheel rotates). In other words, this meant that the oscillation of the wheel is mainly witnessed in the Z-component. This is clearly demonstrated in the deformation spectrums shown below in Figure 5-3. In addition to that, in most of the eigenmodes the maximum deformation tend always took place in the outer diameter of the wheel. Thus, these observation boosts the probability of causing squeals in a different frequency range other than the expected 2-8Khz.



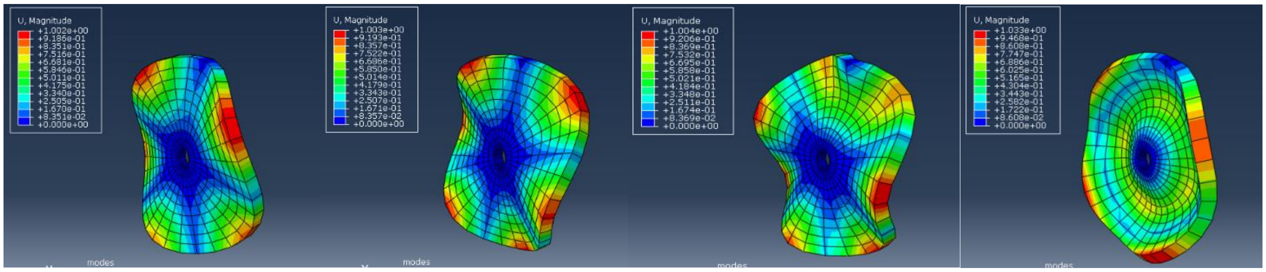


Figure 5-3: The Deformation in the first 8 eigenmodes, respectively.

It is also very likely to consider the squeal sources as friction between the wheel-rail contact surfaces. This is because the directions with a high participation factors are the same as the acting directions of possible friction forces. By that we do not mean only the Z-component but also Y-rotation component which was present in the second mode. So if this mode somehow was excited, from its shapes we can deduce the oscillation will definitely generate friction.

Correspondingly, from the eigenmodes spectrums and their participation factors, we can predict the direction of the force which tends to excite such modes. Since the Z-component is the most dominant, the possibility of resonance will increase when load acts in the Z-direction. So once the frequency of Z-directional load is closer to that particular eigenmode it will be rapidly excited. Of course this holds only in the modes which have high participation factor in the Z direction, namely 1, 3, 5, 7 and 8. As for the second mode it will probably tend to be excited more rapidly if the load was acting in a twisting form about the Y direction.

Therefore, when we consider the forces which might be expected to act in the Z direction, we discover that they are very likely to occur. Most importantly the contact load between the wheel flange and the rail, despite it takes place in many situations, it also can have a periodic behavior, for example if the rotation axis of the wheel is not aligned perfectly or there is uneven wheel tracking or the train is going through a curved path, in such cases the contact force might adopt the frequency of the rotation.

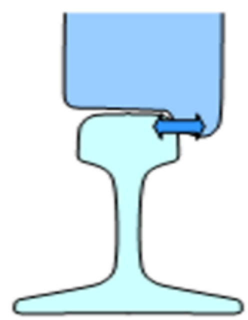


Figure 5-4: The contact between the flange and the rail.

Furthermore, if we consider the effective masses in Figure 5-5, it can be seen that not all of the modes contribute significantly on the dynamic response. The highest EFM value was for the first Mode which represents 50% of the total mass, and for mode 2, 3, 5, 7 and 8 their masses were less than 15%. Therefore, our consideration can be limited only to these modes, therefore it can be said that modes 4, 6, 9 & 10 are not of great importance as the first group. This can be said because even if these modes were excited the reaction force would be very small.

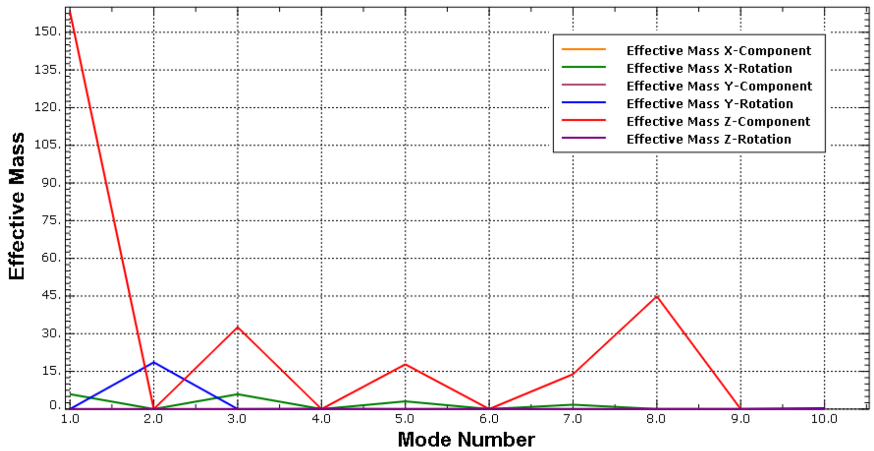


Figure 5-5: The value of the Effective mass for each mode.

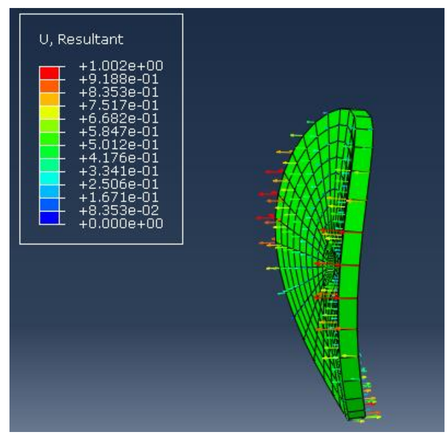


Figure 5-6: The shape & direction of Mode3

In case the train is traveling in a maximum speed of 350 Km/h, the rotation frequency of the wheel will be 30.94 Hz which is very close to the frequency of mode 3 which was 33.305 Hz. So, the frequency ratio (r) will be 0.93 which certainly represents a resonance, and with such ratio the output might be of some influence. But taking into account all possible errors and the fact that the mode can be excited only in certain direction the real behavior may not be that influential, but it will definitely generate squeal. Specially, if we consider the shape of the mode, as shown in Figure 5-6, the wheel is divided to three regions, two regions bend toward one direction and the central region in the opposite direction, while the maximum displacement is in the outer surface, clearly such oscillation will result in a rough contact between the wheel and rail which in return will generate squeals. So, the closer the value of the ratio is to 1 the more effective this oscillation becomes, and operating with a frequency ratio of about 0.93, we can say that it will be influential enough as to generate a squeal.

5.2 RAIL MODAL ANALYSIS

Regarding the rails the results were more interesting, after running the analysis for the first 30 natural frequencies, we found out that many of them lie on the frequency range we are interested in, the squeal frequency that is said to be generated. As shown in Figure 5-7, despite the first five frequencies which are lower than 2 KHz, all the remaining were between 2 KHz and 8 KHz. this tells us that the squeals might be a result from coupling with these frequencies.

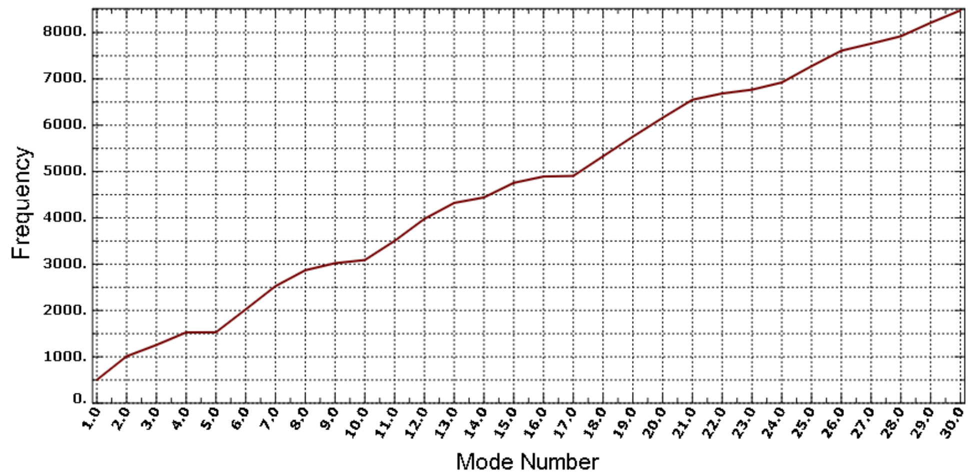


Figure 5-7: The first 30 eigenfrequencies of the rail.

In case the rail is considered as the source of the squeals, the coupled mode shape must be acting on the lateral Z direction, otherwise the wheel of the train will witness a harmonic motion instead of squeals. Once again, by taking a look on the plot on the participation factors Figure 5-8, it can be seen that there is a sufficient number of modes which fit such a description. Figure 5-9 shows some examples of these mode.

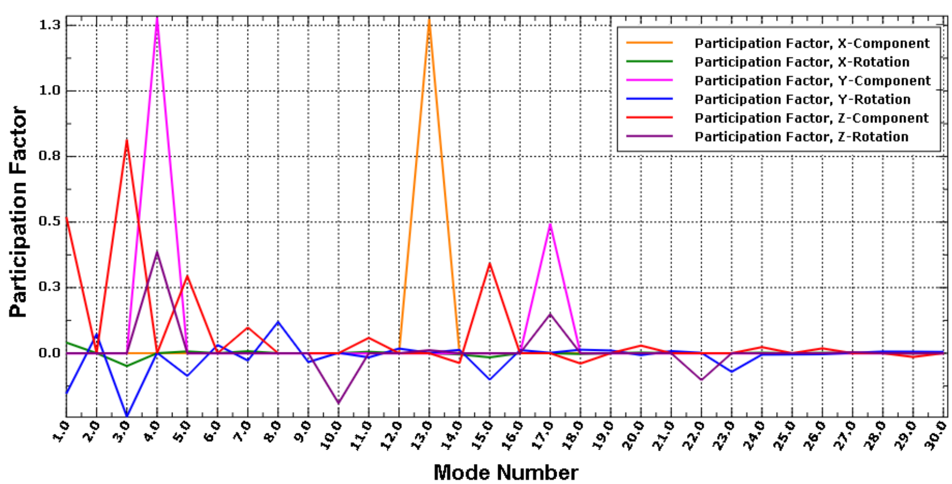


Figure 5-8: The participation Factors of each mode.

From a loading point of view, it can be noticed that there is huge possibility that these modes could be excited. Same as the wheel, this might be due to flange- wheel contact, curved paths, faulty bogies..etc.

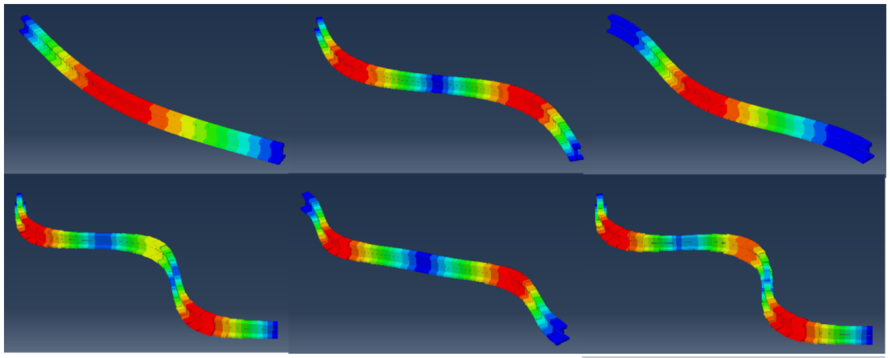


Figure 5-9: The Deformation in the eigenmodes number 1, 2, 3, 5, 8, &15 respectively.

6 CONCLUSIONS AND FUTURE WORK

6.1 CONCLUSIONS

After finishing the process and performing thorough analysis of each component of the model we can draw certain conclusions as are the following:

First of all, there is presence of a squeal as we already knew, but not caused by the reasons we previously thought. The squeal generated in a frequency range between 2-8 Khz is not caused by the rotation of the wheel per se. We found out thanks to the model in Abaqus that the dynamic response of the wheel at Mode 3 has a frequency value of 33.305 Hz, while the wheel's frequency of rotation while traveling at the train's maximum velocity is of 30.96 Hz. A frequency ratio of 0.93 leads us to think there will be a coupling between these 2 frequencies that will effectively ignite a squeal, but not in the range of 2-8 Khz as expected.

It can also be stated the due the reaction of the wheel's deformation, flanging is going to cause friction in the Z component with a force of bigger magnitude than expect, which will cause a more disturbing noise due to the wheel-rail interaction. The friction between the rail material and the wheels, all steel, the wheels' deformation, coupling up with curves in the tracks and rail irregularities can all sum up to a bigger problem than expected.

Furthermore, it can be stated that the rails also play a very important role in this problem. After watching the rails' dynamic response to the loads, sleepers separation, and wheel interaction we see the rails do produce eigenfrequencies in the values of 2 Khz and more. Therefore we can correctly state trains will generate various squeals, probably some more audible than others given the different frequency ranges at which they act, but we can successfully state all of the components in the train wheel-rail interaction will cause some sort of audible discomfort to nearby users or inhabitants.

6.2 FUTURE WORKS

As for future works, as it has already been proven the existence of wheel squeal and its causes, ideally the next research focus should be primarily the decrease or total elimination of this problem. Mechanisms for wheel squeal noise control treatment can be divided in categories and subcategories. One category could be composed of all the already existing trains and tracks, to which certain noise control treatments should be used: those that can be applied to existing trains, rails, and stations. On another category we can have those non existing (still in a design phase) trains, tracks, and stations. Even more, these 2 main categories could be subdivided into 3 more subcategories: depending on the area or component to which they will be applied. These subcategories are represented by on-board mechanisms, applied directly to the trains wheels and suspension; on the track work, meaning on the rails or lubrication systems; and finally those mechanisms applied to the train stations, acting as sound barriers, noise cancellation or vibration absorbing walls.

The next table provides several useful information about several noise controlling mechanisms or processes, an approximate cost, location, and effectiveness.

TABLE 5-5 WHEEL SQUEAL NOISE CONTROL TREATMENTS

LOCA-TION	TYPE	NOISE RED. - dB	COST - $	COMMENTS
Onboard	Resilient Wheels	10 to 20	$2,000 to 2,300/wheel	Well demonstrated to be effective.
	Constrained Layer Damped Wheels	5 to 15	$500 to $1200/wheel	Effective
	Ring Damped wheels	5 to 10	30-50 per wheel	Used by Chicago CTA and MTA NYCT. May be most economical treatment.
	Wheel Vibration Absorbers	5 to 15	$500 to 700/wheel	Demonstrated effectiveness in trials.
	Conical Wheel Taper	Elimination at large radius curves	$0	Must be in combination with gauge widening or asymmetrical rail profile and flexible truck.
	Flexible Primary Suspension	Elimination at large radius curves	Unknown	Conical treads required to induce axle alignment with radius
	Steerable Trucks	Elimination at large radius curves	Unknown	Conical treads required to induce axle alignment with radius
	Onboard friction modifier	Possible elimination.	$1,400 /vehicle/year	Limited effectiveness below 90 foot radii
Track-work	Asymmetrical rail profile	Elimination at large radius curves	Nil	Must be in combination with conical wheels and flexible truck.
	Petroleum lubrication	Partially eliminates squeal	$10,000 to $40,000	Can lubricate flange only, so that effectiveness is limited.
	Water spray lubrication	Eliminates squeal	$10,000 to $40,000 per track curve	Not practical in freezing weather, though antifreeze can be used with water.
	Maintain constant gauge or gauge narrowing in curves	Reduces truck crabbing and potential for squeal	Nil	Gauge narrowing has been correlated with squeal elimination in conjunction with resilient wheels, HPF friction modifier, and elastomer rail embedments. (See Text)
	Restraining rails	Has inconsistent effectiveness	$100 to $200/tr-ft	Reduces truck crabbing and lateral slip, and, thus, squeal. Flange face must be lubricated.
	Rail vibration dampers	Unknown	$20 to $50/ft	Reported to be effective in Europe
	Rail Inlay - (anti-squeal)	Reduces squeal		Reduced or eliminated squeal at WMATA for several months
Wayside	Sound Barrier walls	7 to 10	$20/sf	Does not eliminate squeal
	Absorptive Barriers	9 to 12	$25/sf	Does not eliminate squeal
	Berms	10 to 13	NA	Does not eliminate squeal
	Subway wall treatment	5 to 7	$7-$10/ft^2	Does not eliminate squeal
	Receiver Treatments	NA	$5,000 to $10,000 per receiver	Does not eliminate squeal. Noise reduction dependent on construction

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8 APPENDIX

SECTION ASSIGNATION:

Both members of the team played an active role in every section of the process, analysis and written report. However, each of the members carried out a direct focus on each section as follows:

ACOSTA, ARTURO

Analysis of results to draw conclusions and plant future works and possible solutions.

Main role writing Abstract, Introduction, Problem Statement, Conclusions and Future Works.

DAWI, MALIK

Carried out the design of the 3D model and analysis in Abaqus. Analysis of job submission to state clear facts and explanations of the Abaqus results.

Main role in writing Methodology, Results & discussion.