Analysis of squeal noise produced in train wheels.

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Abstract

In this report the dynamic response of a train wheel is analysed. This study is related to the presence of squeal: a highly audible noise between 2 and 8 kHz generated by the train wheel during its rotation. A simplified train wheel geometry is analysed within computer aided Finite Element Analysis software (ABAQUS). Dynamic analysis with linear perturbation is performed in order to obtain the natural frequencies that the wheel owns itself due to its geometry. The first obtained eigenfrequencies of the wheel are discussed to determine, or not, some possible sources of this squeal noise which are obtained through the existing literature.

1. Introduction

Squeal noise has always been noticed as one of the most disturbing noise sources of railway systems, and especially when this phenomenon occurs in railway systems inside the cities, which with narrow streets and more sharp curves make it resonate louder.

Maximum noise levels have been measured at well over 100 decibels 7.5 meters from the track. This can be physically painful. In addition, production of the squeal consumes extra fuel, and wears wheels and rails.

The propensity for a wheel to resonate may relate to wheel geometry: its profile, or the extent to which it is out of shape, are main characteristic parameters.

In this document they are described some of the main possible sources of squeal generation, and then a dynamic numerical analysis of a simplified wheel model is performed in order to conclude if the described sources of squeal will, or not, make the analysed model produce squeal noise in typical railway conditions.

2. Sources of squeal

In this section they are studied the possible sources of squeal. After some literature research, some simplified expressions are obtained in order to compute the frecuencies that might be sources of squeal in a train wheel.

We assume that the wheel we are analysing pertains to a train which maximum speed is 350 $km.h^{-1}$.

I. Frequency of contact due to wheel rotation

Frequency of the contact between a random spatial point from the exterior part of the wheel and the rail is derived from the velocity of the train and the radius of the wheel, with the following relationship:

$$f_R = \frac{w}{2\pi} = \frac{V}{2\pi R} \tag{1}$$

where f_R is the frequency of contact by the wheel rotation, V is the train velocity $[m.s^{-1}]$, R is the wheel radius $[m]$, and w is the angular velocity of the wheel $[rad.s^{-1}]$.

*This report is based upon the *Simulation Project* of C.M.T. module.

II. Frecuency of the sleepers

It is going to be assumed that the rail is supported in sleepers located every 60cm, which is a common value, for instance, in Spanish railways.

The rail, in contact with the sleepers, could be modelled as a damped system (figure 1), like it is done in [2], by Silvsgaard, E.C., and it may present little dynamic bending deformations between them caused by the weight of the convoy as it passes through.

The rail itself may also be a possible source of resonance and squeal noise, but the dynamic analysis of the rail sleepers modelled as a discretely supported beam is not included in the present document, as this work is focused on squeal noise produced by the wheel. Moreover, geometrical data and materials properties data from the rail and the sleepers, which are not available for us at this point, would be needed.

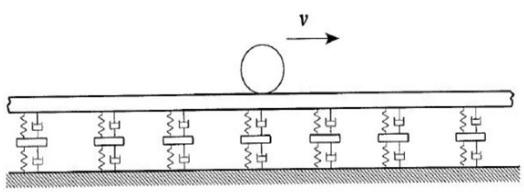

Figure 1: *A rail modelled as a discretely supported beam.*

Nevertheless, the effect of the rail bending is treated as linear perturbation that the rail produces on the wheel rotation. The frequency of this perturbation is related to the velocity of the train and the distance between the sleepers

$$f_S = \frac{V}{d} = \frac{V}{0.6} \tag{2}$$

where f_S is the frequency of linear perturbations due to the sleepers, and d is the distance between them $[m]$.

III. Stick-slip transitions of the wheel with respect to the rail

"Stick-slip mechanism" occurs when the sliding force caused by a curve becomes too large to be followed by the static friction force. Then, the sticking on the rail by creep changes to slipping with a decreasing friction force. Then, with decreasing sliding speed, the static friction can be built up and the wheel sticks again. Thus, this phenomenon occurs in rapid succession.

The frequency of the stick-slip transitions is analytically computed using the expression derived by Ruiten, C.J.M., in [3]:

$$f_{SS} = \frac{F_N}{V} \cdot \frac{\bar{v}}{2\pi m \eta} \tag{3}$$

where f_{SS} is the stick-slip frequency, N is the wheel load, V is the convoy velocity, $\bar{v}$ is the average slope friction creep-curve, m is the modal mass, and η is the critical damping of a standard wheel.

In this analysis we use some typical values for this equation, described in the following table:

Table 1: *Adopted values to characterize the stick-slip phenomenon*

Parameter	Value	Magnitude
F_N	24000	$[N]$
V	0 - 350	$[km.h^{-1}]$
$\bar{v}$	1 - 10	–
m	303	$[kg]$
η	0.02	%

The value of $\bar{v}$ depends on the radius of the rail curve, and two extreme cases are taken, as it is done in [3]:

- $\bar{v} = 1$ for curve radius of $25m$.
- $\bar{v} = 10$ for curve radius of $80m$.

3. Numerical analysis

In this section it is described the numerical analysis which is performed in a simplified wheel geometry. The dynamic performance of the wheel which is to be related to squeal generation frequencies, described in the previous section, is computed through a linear perturbation analysis in ABAQUS.

I. Model and boundary conditions

The wheel model analysed in this document consists in a simple disc of diameter $D = 1.00m$ and thickness $t = 0.05m$, with a centred hole of diameter $d = 0.10m$. The CAD model is built by extrusion of the two proper circumferences.

The material for the model is assumed to be linearly elastic with the following material properties:

- Young Modulus $E = 2.10^9 kg.m^{-1}.s^{-2}$
- Poisson Ratio $\nu = 0.25$
- Density $\rho = 7800kg.m^{-3}$

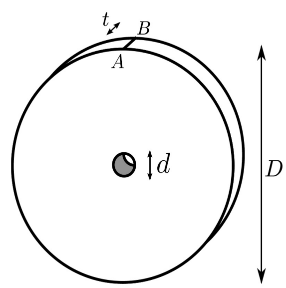

Figure 2: *Model description and parameters.*

Once the material is assigned to the geometry, they are imposed the boundary conditions. For linear perturbation analysis it is sufficient if they are prescribed all the displacements in both the interior edge surface and on an axial line which represents the contact of the wheel with the rail.

Other boundary conditions are tested, and it is realized that the prescription of the rotations XX, YY, and ZZ in the boundary conditions surface and line are not necessary and do not add any differences in the final results. This can be explained by the symmetry of the displacements in relation to the prescribed zones, and due to the restriction of symmetry of the stresses in a continuum body.

II. Discretization and results

The initial discretization of the wheel geometry is a structured hexahedral mesh with "Approximate element size" equal to 0.05.

Note that with this discretization, the element size coincides with the thickness of the wheel, so the mesh has only one layer of elements within the wheel deep.

The first ten eigenmodes are computed and the results are stated in the following table.

Table 2: *Obtained frequencies for the first ten eigenmodes with coarse Hexahedral mesh*

Mode	f_n	Mode	f_n
1	24.68	6	89.04
2	25.86	7	118.88
3	33.31	8	133.74
4	48.80	9	149.32
5	69.65	10	159.37

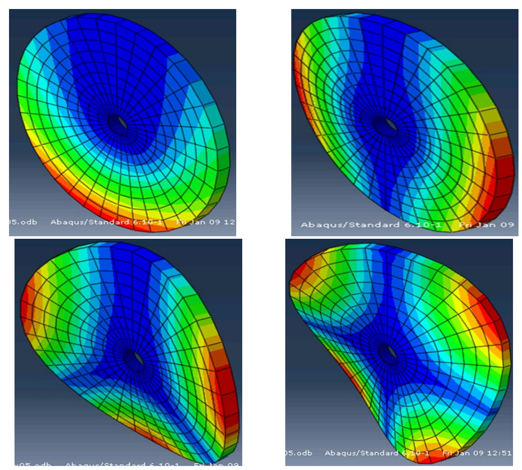

Figure 3: *Plot of the deformation for the first four natural modes -ordered from left to right and upside down-.*

Afterwards, some simulations are runned with refined meshes, and it is observed that the eigenmodes and the eigenfrequencies results have a huge dependence on the mesh in use. This drives us to study if the results stabilize with iterative mesh refinement.

Furthermore, they are tested different types of finite elements, including Wedge and Tetrahedral elements, always within structured mesh. Other works on this field corroborate that the more efficient elements to be used in this analysis rely on a structured mesh which divides the wheel radially.

In order to be sure that our computation is good enough, a comparison is made trying to reproduce with our analysis software an example from the article [1], by Heckland, M.A. et al.

In their work, the model geometry is analogous to ours, but the values of the parameters D, t, and d are slightly different.

Regarding the first eigenmodes, which are the most relevant for the squeal noise production, errors under 10% in comparison with Heckland, M.A. et al work are obtained for Hexahedral, Tetrahedral, and Wedge meshes with 0.001 element size. However, none of these element types can reproduce correctly the results from [1] with bigger element sizes.

In the case of our model, results for several Hexahedral mesh sizes are saved and finally it can be seen -see table 3 and figure 4- that the results converge and stabilize when we reach element sizes of around $0.01 - 0.02$.

Recall that the geometry of the model that we try to reproduce, from Heckland, M.A. et al, is more than one order of magnitude smaller than our model, so the correct element size from this verification study can not be the same element size that is correct for the reproduced example, but the comparison with that model, and the feasibility to reproduce their results, can be thought as a verification procedure for the kind of analysis and the boundary conditions we are applying within ABAQUS.

Table 3: *Natural modes obtained within different element sizes.*

	Element size		
Mode	**0.05**	**0.02**	**0.01**
1	24.68	155	157.47
2	25.86	176	176.53
3	33.31	224	224.57
4	48.80	269	269.18
5	69.65	451	451.73
6	59.04	585	585.66
7	118.88	633	633.08
8	133.74	773	773.53
9	149.32	1002	1002.30
10	159.37	1061	1061.60

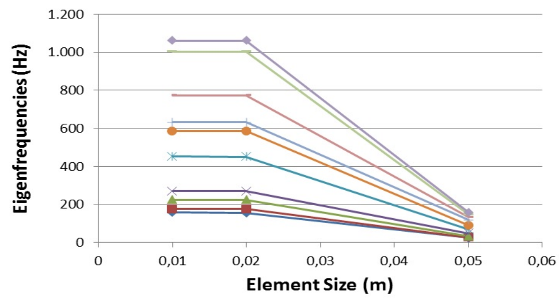

Figure 4: *Graphical representation of the first ten natural modes vs the element size of the mesh.*

4. Eigenmodes vs. sources of squeal

All the results obtained for the frequencies of linear perturbations caused by rotation, by the sleepers, and by the stick-slip transition, f_R, f_S, and f_{SS} respectively, which expressions are detailed in section 2, are computed in a range of velocities from 0 to 350 km/h and summarized in the following tables 4 and 5.

Table 4: *Table of the frequencies of the first two sources of squeal.*

$V[km.h^{-1}]$	**Rotation** (f_R)	**Sleepers** (f_S)
350	30.95	162.04
250	22.10	115.74
150	13.26	69.44
100	8.84	46.30
50	4.42	23.15
30	2.65	13.89
10	0.88	4.63

Table 5: *Frequencies of the stick-slip phenomenon at various velocities and curvature radius.*

	Stick-slip (f_{SS})	
$V[km.h^{-1}]$	$\bar{v} = 10$	$\bar{v} = 1$
350	64.78	6.48
250	90.69	9.07
150	151.16	15.12
100	226.73	22.67
50	453.47	45.35
30	755.78	75.58
10	2267.34	226.73

In the tables above (4 and 5), the cells marked in light grey correspond to frequencies that may couple with some natural modes (those gathered in table 3, from section 3). All the frequencies higher than the first eigenfrecuency -157,47Hz- belong to this group.

With dark grey, the frequencies correspond to the range 2-8kHz, which corresponds to the frequency of the annoying squeal noise.

5. Conclusions

The results obtained in the present study are similar to the results obtained in [3] and [5], and thus reliable, and they reflect that squeal-noise does not appear in the first eigenmodes, but in some higher modes.

The velocity of the train and the rotation of the wheel itself may not couple with the natural frequencies of the wheel, because they are smaller than the first mode.

The perturbations caused by the sleepers may couple with the natural frequencies at high velocities from 300-350 km/h on, but the modes that are achieved with these perturbations are under 2kHz. They may produce sound, but not annoying.

Stick-slip frequent transitions can produce squeal noise at lower velocities than 150 km/h, and are prone, contrary to what was expected, for curves with large radius.

Higher modes are not computed in this assignment because a precise result for those modes would require a high computational cost, as they should be found with thinner mesh.

6. Future work

Thinner mesh is necessary for precision in higher modes, which are expected to produce squeal noise between 2 and 10 kHz, because the deformation state of the high modes that we had computed becomes increasingly irregular in some regions of the wheel as the mode increases. For more reliable results, it should be needed a deeper particularization in the geometry definition of the model, and the rail model should be studied as a possible source of squeal by itself. Dynamic load analysis would not add any new information related to the values of the natural frequencies of the wheel, but may be useful to compute the exact intensity of the noise generated (dB).

7. Acknowledgements

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