Finite Element Analysis of Wheel/Rail Squeal Noise

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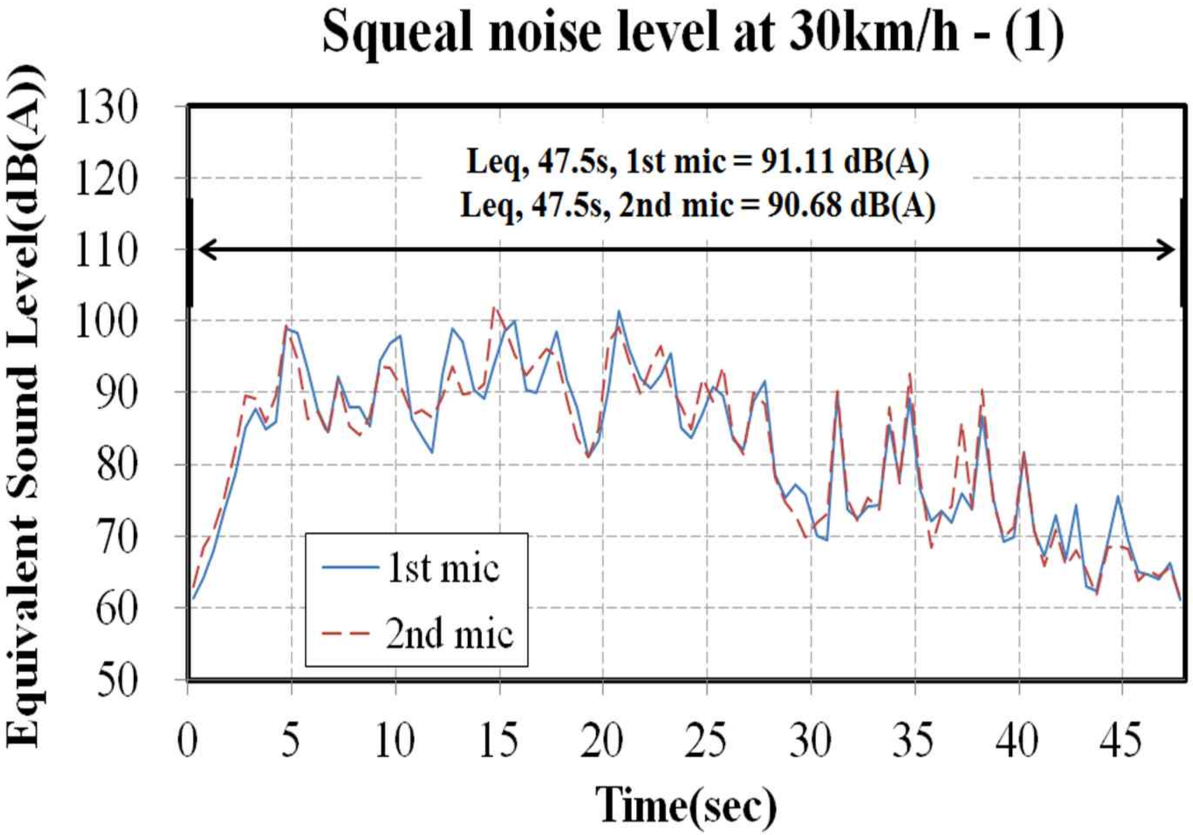
Contents

Introduction

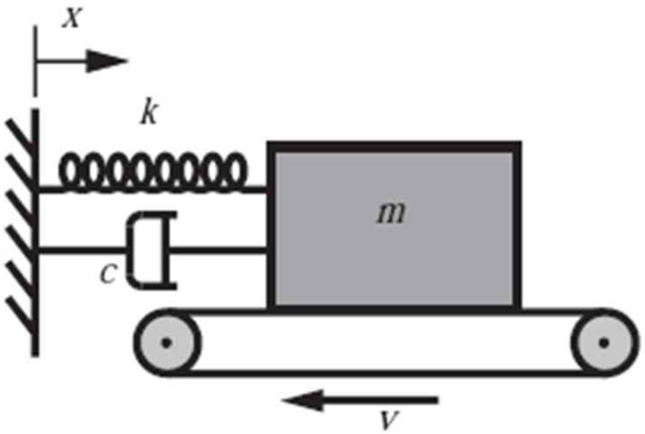
- **squeal noise is very uncomfortable :** ~100 dB(A)
- **develop analysis method for complex eigenvalues**
- **identification of parameters influencing on the squeal noise**



Introduction

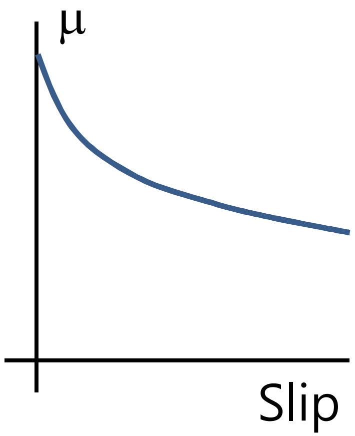
Models for squeal noise

Negative damping
(Mills, 1938)

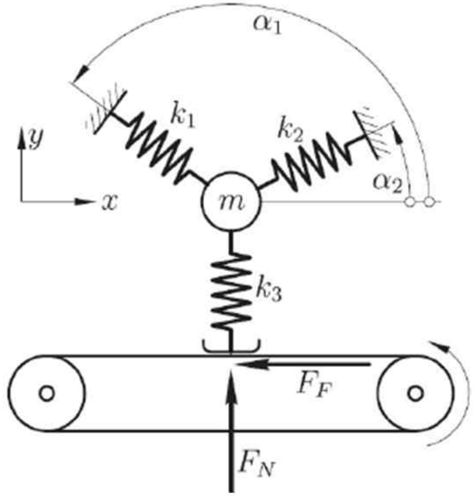


$$m\ddot{x} + (c - mg\mu_2)\dot{x} + kx = 0$$

$$\mu_k = \mu_k(v_s) = \mu_1 - \mu_2 v_s$$

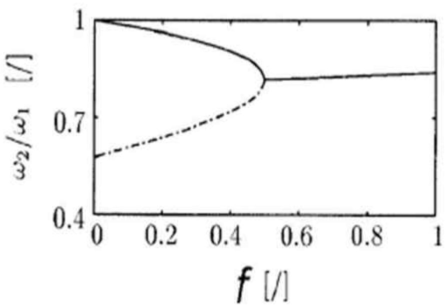


Mode coupling
(Hoffmann et al. 2006)

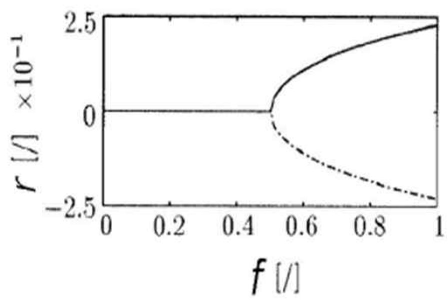


$$\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix} + \begin{bmatrix} k_{11} & k_{12} - fk_3 \\ k_{21} & k_{22} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

$y = c_1 e^{rt} \sin(wt + \varphi)$



Imaginary



Real

Finite element modeling

Models for squeal noise

Structure

Friction

$$(\lambda^2[M]+\lambda[C]+[K-K_f])\{y\}=\{0\}$$

$[K-K_f]$: asymmetric due to friction

Complex eigenvalue μ : $\lambda=\alpha \pm i\omega$

Effective damping : $-\alpha / |\pi\omega|$

The generalized displacement of the disc unit, y

$$y=\sum_{k=1} e^{\alpha_k t}(A\sin\omega_k t + B\cos\omega_k t)$$

When the real part of the eigenvalue is positive → The system may be unstable

↓

Potential squeal noise

Finite element modeling

Analysis procedure by ABAQUS

Step 1: a vertical load was applied on the pad holders in a static analysis. Tangential friction was assumed to be zero.

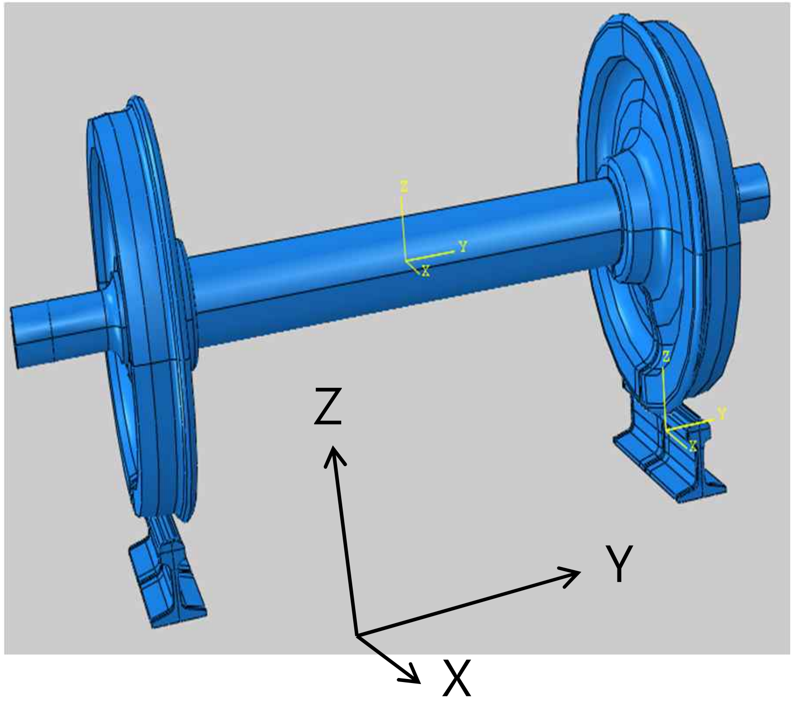
Step 2: a slip condition with friction was imposed on the wheel as a predefined variable ➔ asymmetric [K] matrix

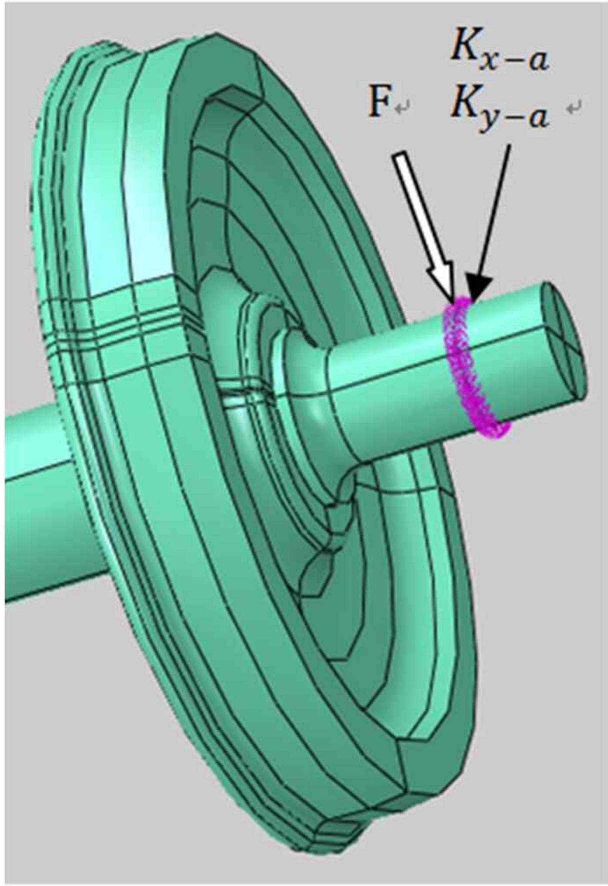
Step 3: the real eigenvalues and mode shapes of the model were extracted. The obtained set of eigenmodes provides the subspace for computing complex eigenvalues in the next step.

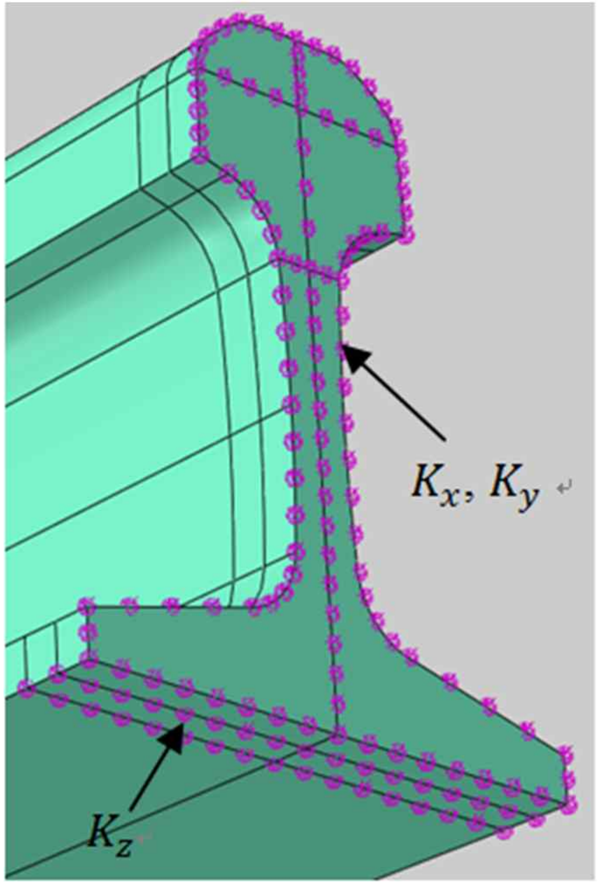
Step 4: the complex eigenvalues and mode shapes were obtained.

Finite element modeling

Boundary conditions





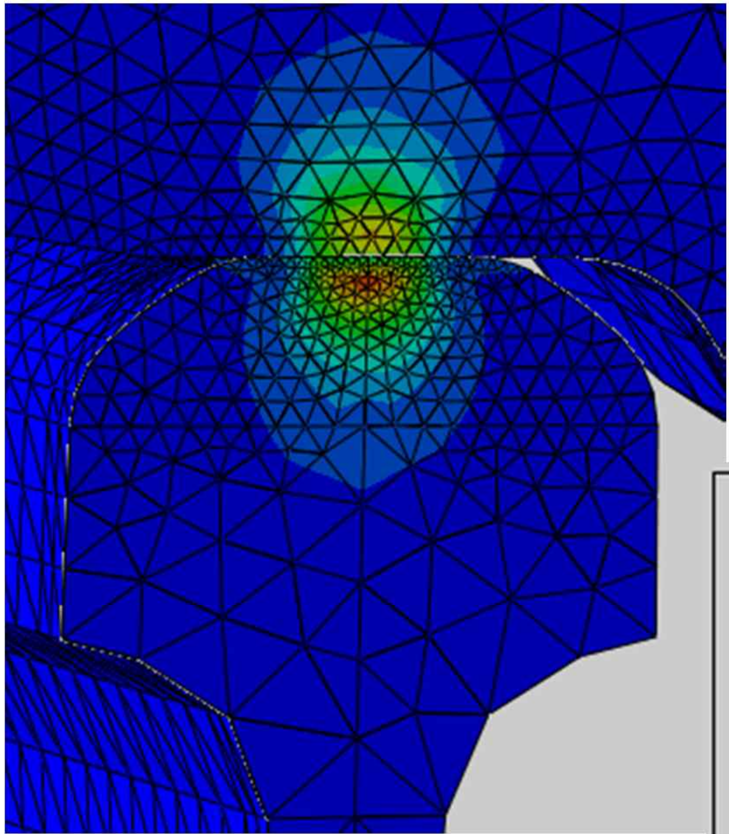


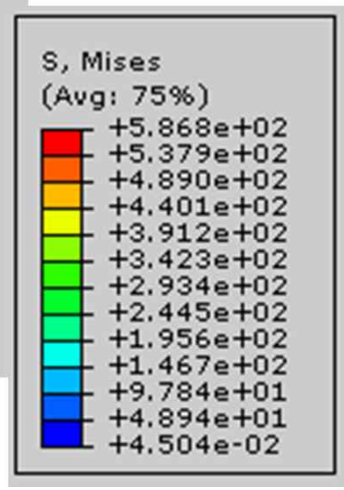
Wheel: 860mm, 1:40 slope

Rail : UIC60, 500 mm (between sleepers)

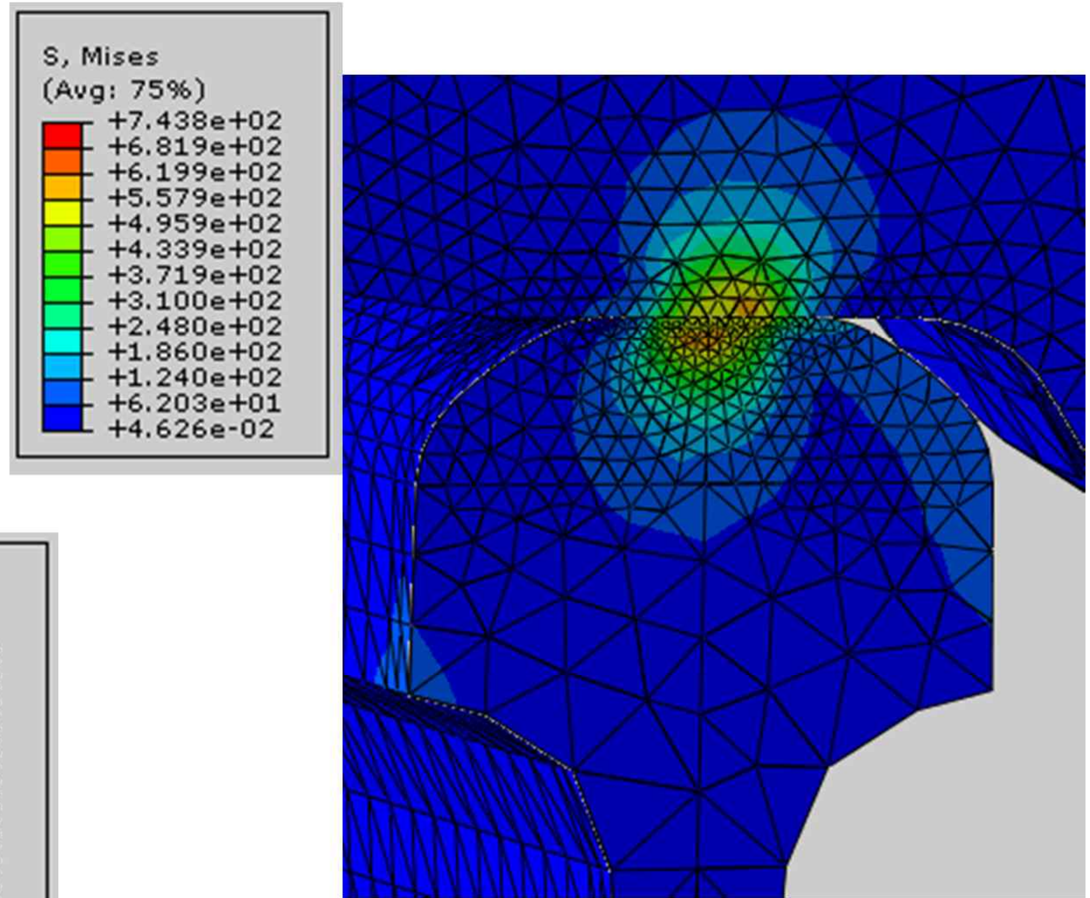
Results and discussion

Contact analysis- lateral creepage





(a) Vertical load

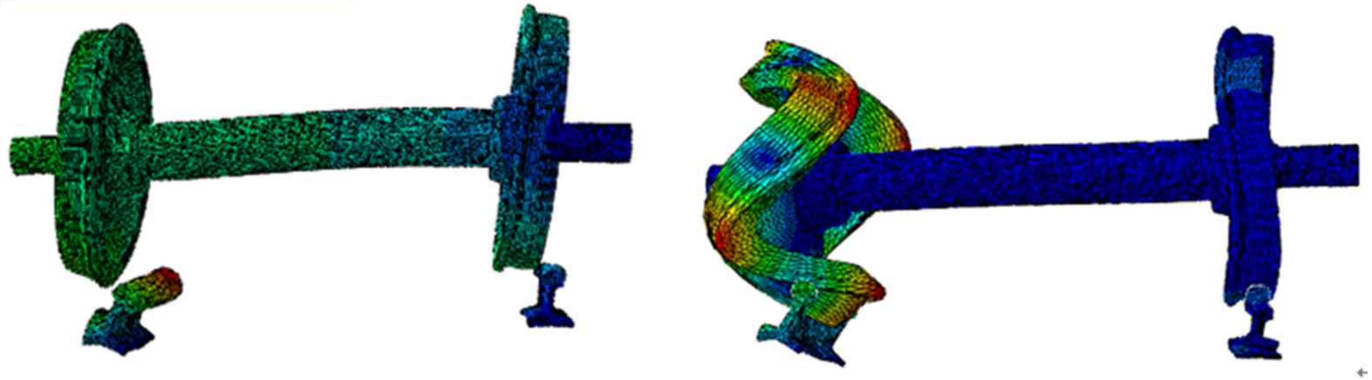


(b) Vertical and lateral slip

K_x=K_y=1000 N/mm, K_z = 10000 N/mm, μ=0.35.

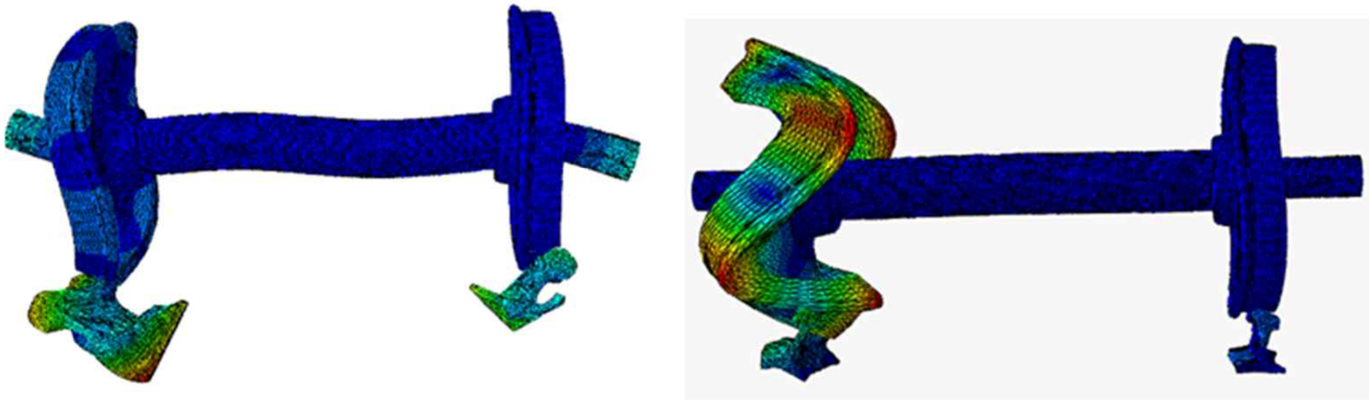
Results and discussion

Unstable modes



(a) 115.56 Hz (b) 1279.0 Hz

Figure 5: Flatter instability, K_x=K_y=1000 N/mm, K_z = 10000 N/mm, μ=0.35.

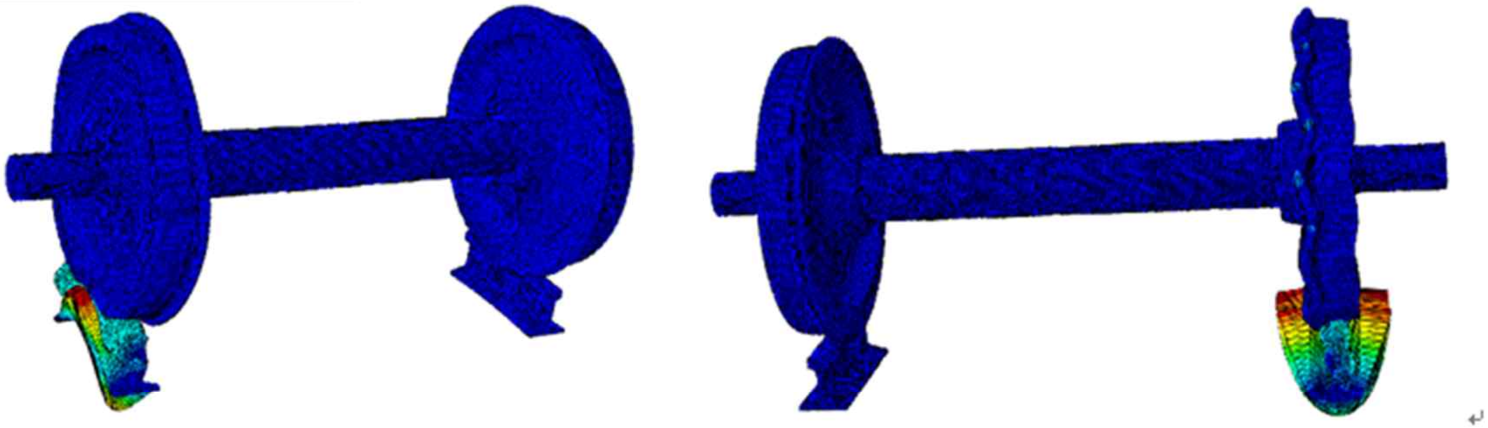


(a) 1196.3 Hz (b) 1278.3 Hz

Figure 6: Flatter instability, K_x=K_y=1000 N/mm, K_z = 10000 N/mm, μ=0.43.

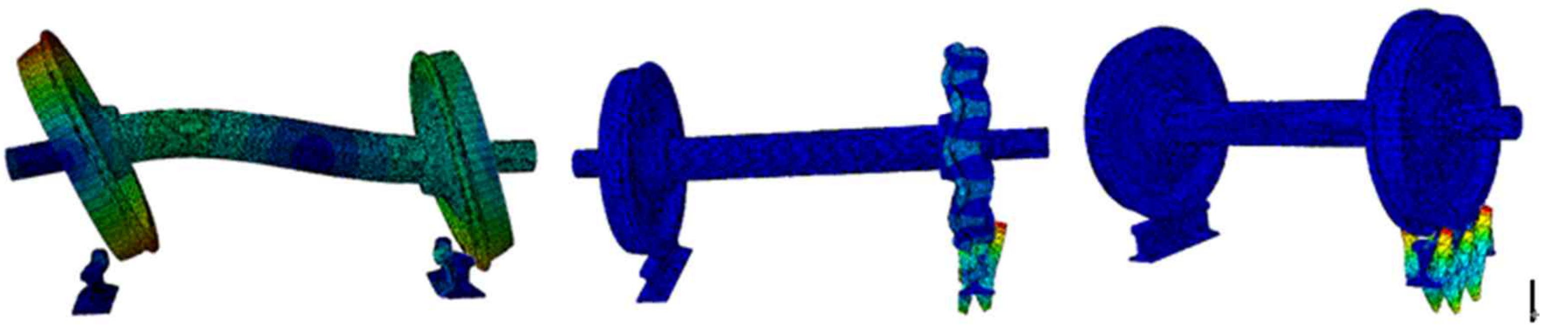
Results and discussion

Unstable modes



(a) 196.3 Hz (b) 1278.3 Hz

Figure 7: Flatter instability, $K_x = K_y = 10^4$ N/mm, $K_z = 10^8$ N/mm, μ=0.33.



(a) 217.68 Hz (b) 6916.0 Hz (c) 9396.9

Figure 8: Flatter instability, $K_x = K_y = K_z = 10^8$ N/mm, μ=0.35.

Results and discussion

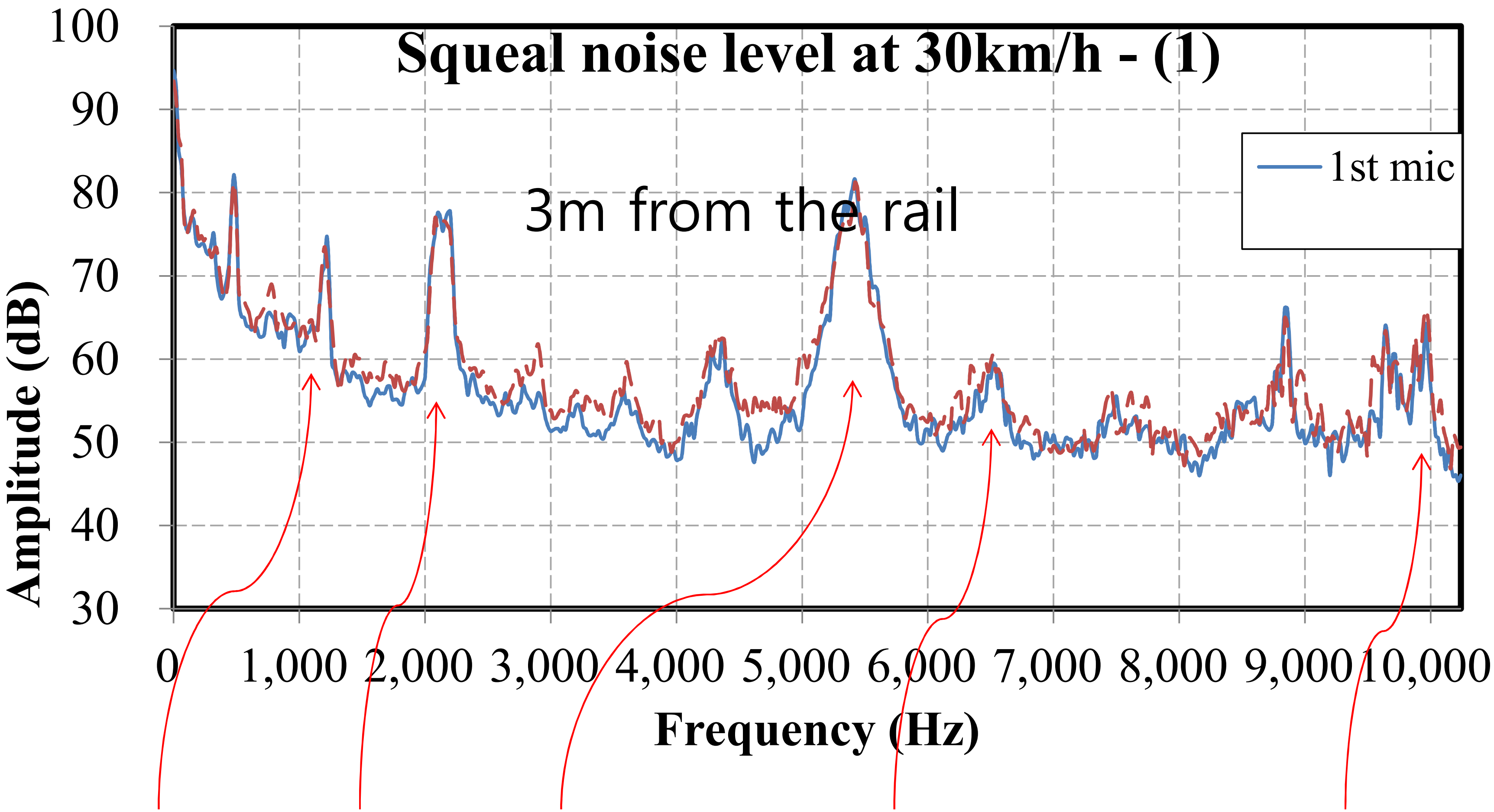
Unstable modes : parametric study

Rail supporting spring constant, K		Friction coefficient, μ	Axial load (Kg_f)	Instable modal frequency (Hz)
$K_x = K_y$ (N/mm)	K_z (N/mm)			
1000	10000	0.09	12000	None
1000	10000	0.10	12000	1280.2
1000	10000	0.20	12000	1277.5
1000	10000	0.31	12000	116.24, 1278.9
1000	10000	0.35	12000	115.56, 1279.0
1000	10000	0.37	12000	115.17, 1278.0
1000	10000	0.40	12000	114.78, 1278.8
1000	10000	0.43	12000	114.32, 1196.3, 1278.3
1000	10000	0.45	12000	114.15, 1197.1, 1279.9
1000	10000	0.47	12000	113.76, 1197.6, 1278.9, 5520.8(r)
1000	10000	0.35	10000	114.95, 1192.5, 1272.5, 6423.1(r)
1000	10000	0.35	14000	116.09, 1284.1
10000	10^8	0.35	12000	212.67, 2217.2(w,r), 10675(w,r)
50000	10^8	0.35	12000	214.23
10^5	10^7	0.35	12000	215.26, 9403.7(r)
10^8	10^8	0.35	12000	217.68, 6916.0(w,r), 9396.9(r)
10^8	10^8	0.12	12000	None

Friction coeff. μ <0.1 ➔ no unstable mode.

Results and discussion

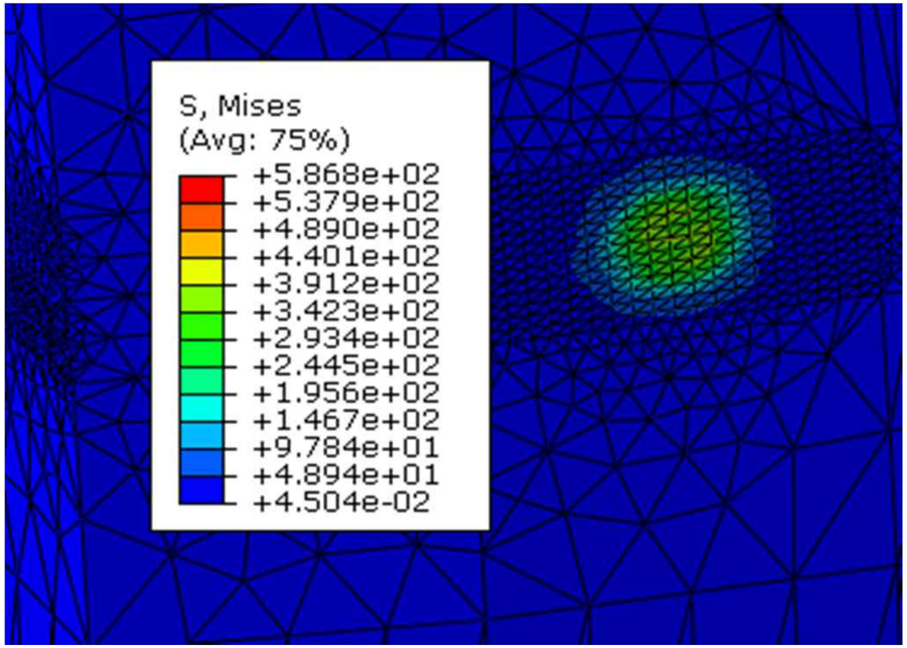
Squeal noise measured in track

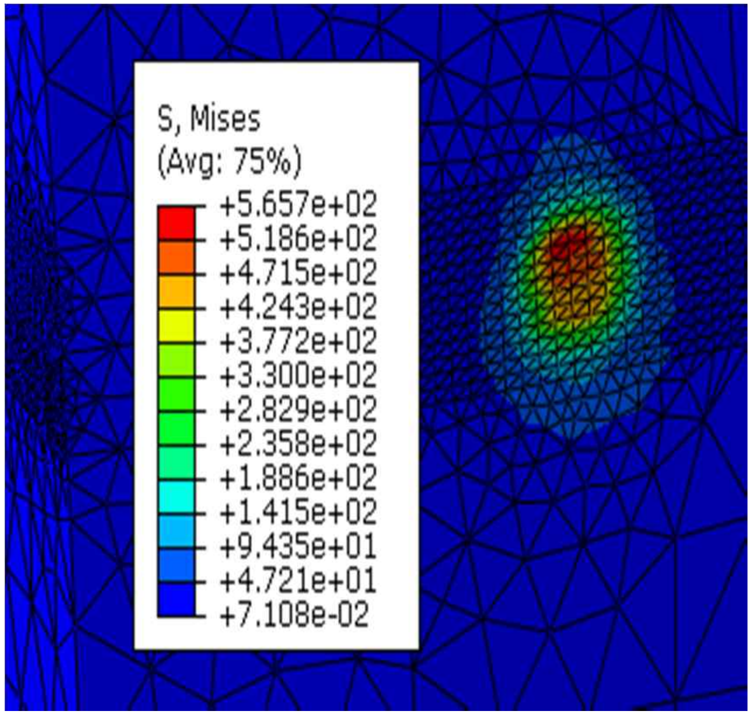


Numerical : 1278, 2217, 5520, 6423 10675 Hz

Results and discussion

Longitudinal creepage





K_x=K_y=1000 N/mm, $K_z = 10000$ N/mm, μ=0.35.

(a) Vertical load

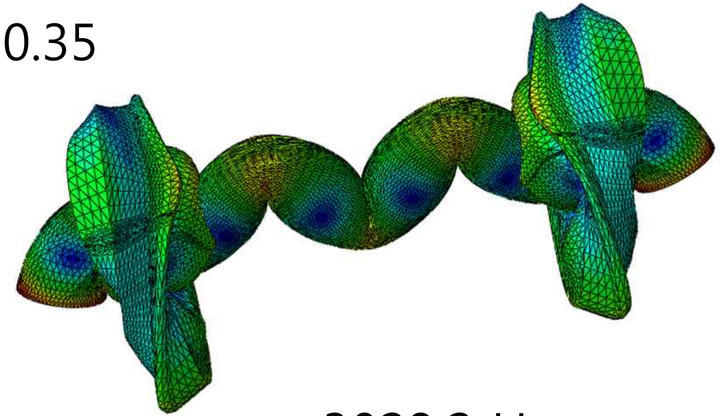
(b) Vertical load and longitudinal slip

Results and discussion

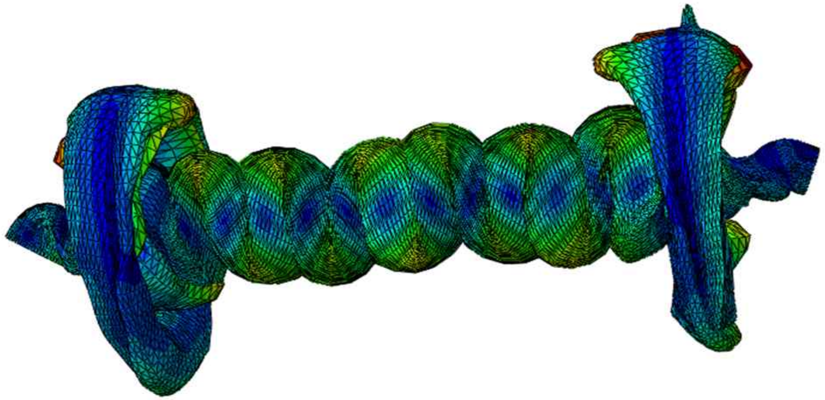
Unstable modes

kx=ky=10^5, kz=10^8

μ =0.35

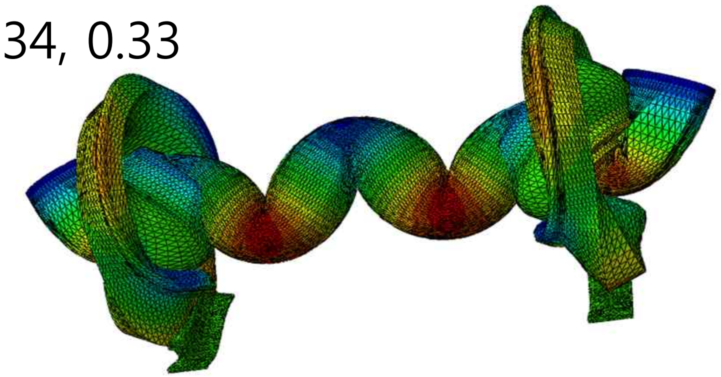


3028.2 Hz

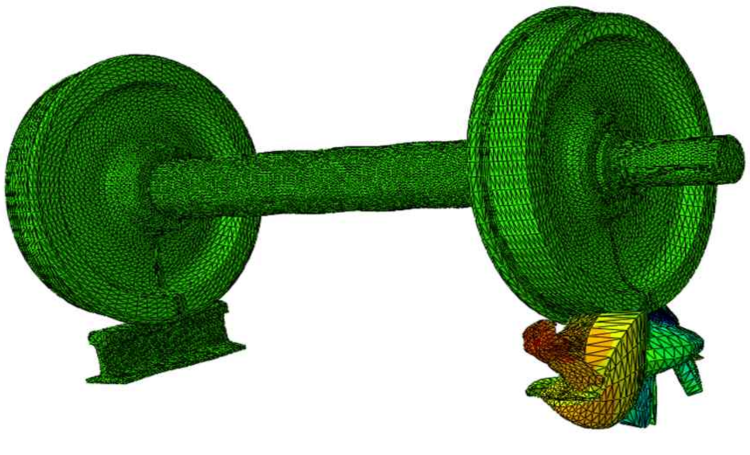


12402 Hz

μ =0.34, 0.33



3028.2 Hz

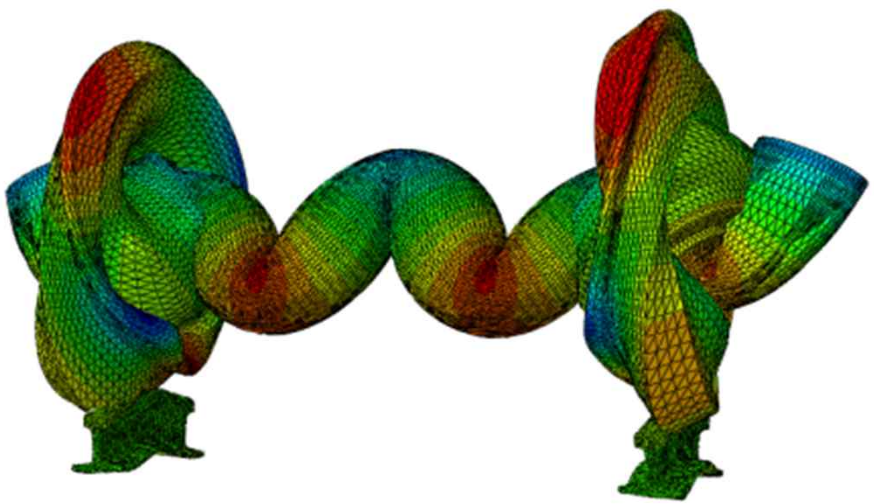


12402 Hz

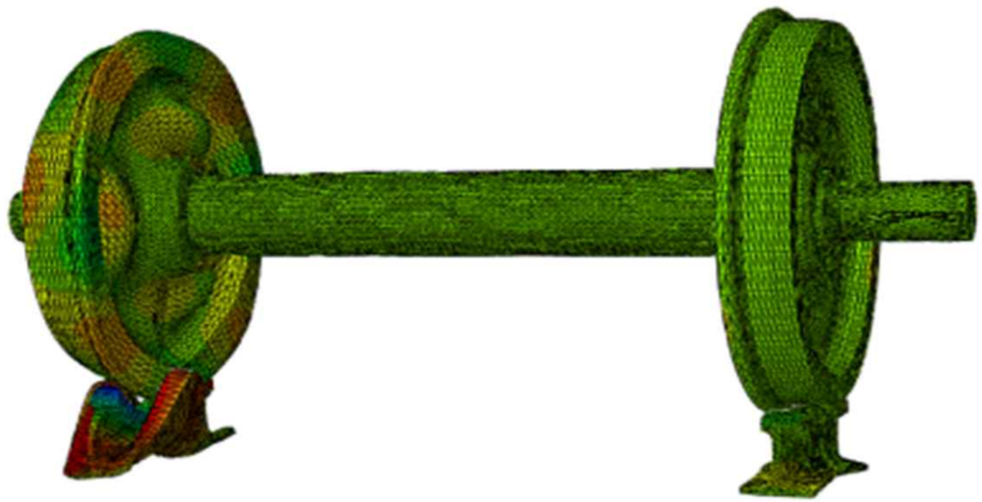
Results and discussion

Unstable modes

μ =0.35, Kx=ky=1000, kz=10000,



3027.8 Hz



6431.2 Hz

μ < 0.19 : no unstable mode

Conclusions

- the instable modes were dependent on the friction coefficient, vertical load and the boundary conditions applied to the rails.
- The numerical eigen frequencies are in a good agreement with the measured values.
- For lateral creepage, when $\mu < 0.1$, no squeal noise.
  wheel bending modes.
- For longitudinal creepage, when $\mu < 0.2$, no squeal noise.
  Wheel twisting modes, axle bending with multiple nodes.
- Track and Rail fastening type are also key parameters.
  In this study, they were considered as spring constants.

Thank you very much.