Comparison of Models for Predicting Curve Squeal Noise

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Abstract. Curve squeal is a highly unpredictable phenomenon due to its random nature. Efforts are made in both the academic and industrial community to analyse curve squeal and identify measures to reduce it. This paper aims to compare two different modelling approaches used in the QuieterRail project to predict curve squeal. The first approach focuses on efficient predictions in the frequency domain, while the second approach evaluates the full system response through a representation of the wheel/rail interaction in the time domain. Models are run with identical input data, corresponding to a resilient tramway wheel circulating on a slab track in a tight curve. Several output variables are compared to investigate the differences in each calculation step of the models: mobilities, friction law, instabilities, wheel vibrations. Despite the differences in formulations, the results are comparable.

Keywords: Curve squeal, friction-induced vibration, numerical methods.

1 Introduction

Curve squeal is a highly unpredictable phenomenon due to its random nature. Efforts are made in both the academic and industrial community to analyse curve squeal and identify measures to reduce it.

The QuieterRail project (2024-2027) aims to introduce a step change in the prediction and mapping of railway noise and vibration, in the acceptance testing of new rolling stock and in the promotion of cost-effective noise mitigation. One topic concerns curve noise of urban railways: new models are developed to predict curve squeal and flange noise.

This paper aims to compare two different modelling approaches. The first approach focuses on efficient predictions in the frequency domain. The second approach evaluates the full system response using a representation of the wheel/rail interaction in the time domain.

2 Methods

Frequency domain models for predicting curve squeal were developed since the early days of curve squeal research [1]. Their primary advantage lies in their computational efficiency, as they typically require small calculation time.

2.1 VibraTec model

The frequency domain model [2] implemented in the tool SONIA (Squeal Occurrence NoIse Analyzer), describes the wheel and the rail by their mobilities. The position of the wheel-rail contact point is an input to the model: it can be obtained from multi-body simulations or measurements. The contact is modelled by a linearised Hertzian normal stiffness and Mindlin's tangential stiffness, while the non-linear creep-force relation of Shen-Hedrick-Elkins, introduced in [3], is combined with a heuristic decreasing friction coefficient. The situation of constant friction is treated as a particular case of this heuristic law. Displacements are considered in normal and tangential directions.

Under the hypothesis of small harmonic oscillations around the quasi-static equilibrium, the dynamic equilibrium at the contact can be expressed through a closed loop transfer function in the frequency domain. By applying the Nyquist criterion, unstable frequencies at which squeal can occur are determined. However, this stability analysis does not provide the amplitude of the self-sustained vibrations. To address this, a simplified method is employed, allowing for the direct computation of the vibration amplitude in the stationary regime, known as the ‘limit cycle’. The method is based on a balance of the injected and dissipated power in the wheel-rail system and assumes a mono-harmonic periodic response of the structure at each potential unstable frequency [4].

The sound power radiated by the wheel, which is the dominant source of curve squeal noise, is estimated through the TWINS wheel radiation model [5].

2.2 ISVR model

The ISVR model follows a more complete approach that builds upon previous publications [6-7]. The starting point is the result of a quasi-static calculation of the railway wheelset behaviour in a curve using multibody techniques. This provides the locations of the wheel-rail contact points, as well as the mean values of creepages and forces transmitted between the wheels and the rails. These parameters serve as input data for a vibration model that incorporates a modal model of the wheels based on a finite element formulation and a simplified track model.

The formulation of the contact force in the normal and tangential directions adopts the same approach as the VibraTec model. The non-linear Hertzian contact theory is used for the normal direction while either the Shen-Hedrick-Elkins model or FASTSIM are employed for the tangential direction. The calculations presented in this paper will be limited to using the Shen-Hedrick-Elkins model.

The identification of unstable frequencies is based on the application of the Nyquist criterion, as in the case of the VibraTec model.

Self-sustained vibrations are obtained through numerical integration of the equations of motion in the presence of the contact forces, using a state-space approach. The modal approach allows for the inclusion of all wheelset modes, or a reduced number (down to two or even a single mode) which can be used to provide insight into the instability mechanisms.

3 Benchmark scenario

The models were run with identical input data, corresponding to a resilient tramway wheel circulating on a slab track in a tight curve, see Table 1. Quasi-static kinematic conditions, i.e. angle of attack and contact positions, are obtained from VibraTec multi-body simulations.

Table 1. Benchmark input data

Input data	Value	Input data	Value
Rail type	35GP	Wheelbase	1.6 m
Rail pad vertical stiffness	350 MN/m	Wheel static load	50 kN
Rail pad lateral stiffness	70 MN/m	Curve radius	25 m
Rail pad damping	0.22	Angle of attack	40 mrad
Sleeper spacing	0.75 m	Train rolling speed	20 km/h
Wheel radius	0.3 m	Friction static coefficient	0.3

Several output quantities are compared for each calculation step of the models: wheel, rail and contact mobilities, friction vs. creep curve, instabilities, wheel self-sustained vibrations.

4 Results

4.1 Mobilities

Fig. 1 shows the total mobility at the wheel/rail contact point, obtained as a sum of rail, wheel and contact mobilities in the frequency range [0-6500 Hz].

Results from the two models are very similar and small differences in mobilities exist at specific frequency ranges. These are a consequence of different implementations of the track model and specifically the modelling approach for the foundation.

4.2 Friction law

Fig. 2 shows the friction coefficient as a function of creepage. Both constant and falling friction laws are simulated, but for the sake of conciseness only the curves corresponding to falling friction are shown. The friction curves are very similar.

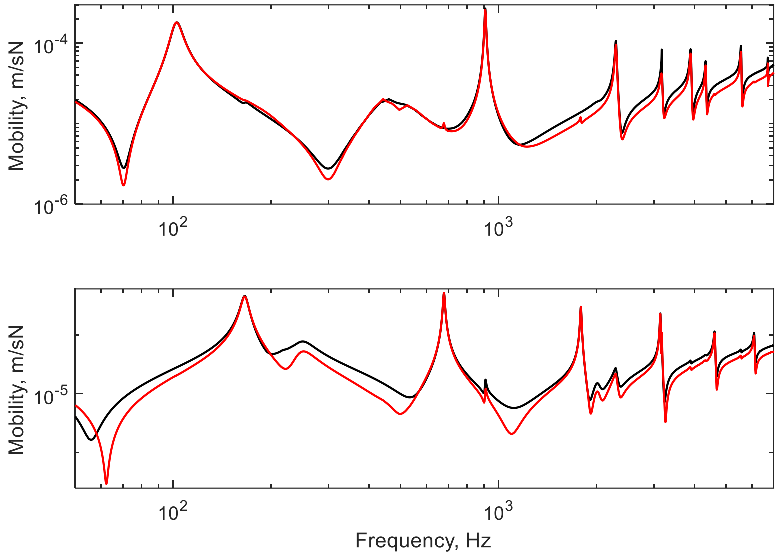

Fig. 1. Total mobility at contact point. Black: ISVR model, red: VibraTec model. Top: normal direction, bottom: tangential direction.

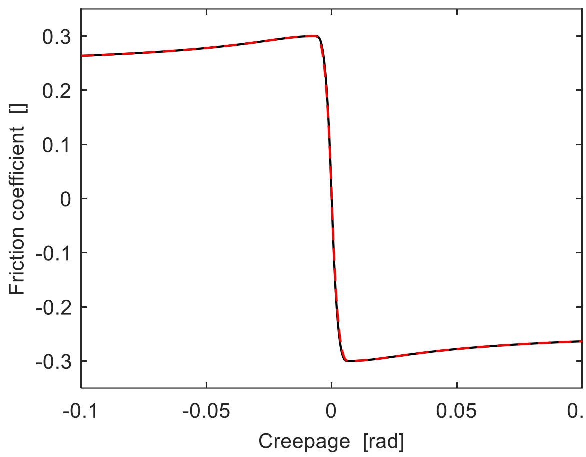

Fig. 2. Friction vs. creepage curve. Black: ISVR model, red dashed: VibraTec model.

4.3 Instabilities

The unstable frequencies identified for cases of both constant and falling friction are summarized in **Table 1**Table 2. The corresponding wheel mode type is also indicated.

Table 2. Squeal instabilities

Constant friction		Falling friction		Wheel mode
ISVR model	VibraTec model	ISVR model	VibraTec model	
		166 Hz	166 Hz	0L1
	293 Hz		295 Hz	---
678 Hz	678 Hz	678 Hz	679 Hz	0L2
		1787 Hz	1788 Hz	0L3
		3139 Hz	3139 Hz	0L4

The models give similar results in both cases, except for the frequency 293/295 Hz, which is only output in the VibraTec model. The differences arise from the deviations in the total mobility and track model, that are mentioned in Section 2.2.
Additional simulations were carried out using a rigid track assumption. As this assumption removes the influence of track mobility, it makes the two modelling approaches more consistent. Table 3 shows the unstable frequencies identified under the rigid track assumption, where both models give the same results. It should be noted that no instability was found in the case of constant friction.

Table 3. Squeal instabilities, rigid track hypothesis.

Constant friction		Falling friction		Wheel mode
ISVR model	VibraTec model	ISVR model	VibraTec model	
---	---	165 Hz	165 Hz	0L1
---	---	676 Hz	677 Hz	0L2
---	---	1786 Hz	1787 Hz	0L3
---	---	3138 Hz	3139 Hz	0L4

4.4 Self-sustained vibrations

Different approaches are used to predict the amplitude of the self-sustained vibrations. The ISVR model involves a time integration of the equation of motion of the wheel modes, leading to a periodic or quasi-periodic limit cycle. The time-domain integration allows solution of the so called ‘mode competition’ between unstable frequencies found through the Nyquist criterion.

Conversely, the hypothesis in the VibraTec model is that the wheel vibrates periodically at one unstable frequency. The amplitude of this harmonic periodic motion is the limit cycle amplitude as derived by the method described in [4]. This hypothesis is applied to each of the unstable frequencies leading to several estimations of periodic vibrations. The maximum value among the limit cycle amplitudes is a conservative estimation of the wheel vibration velocity. An enhanced method is presented in [8], where multiple harmonics are considered. The comparison between self-sustained vibrations must be interpreted considering these differences.

Fig. 3 shows the time evolution of the wheel vibration obtained by the ISVR model, which shows how it grows until it reaches a constant amplitude after 0.3 s.

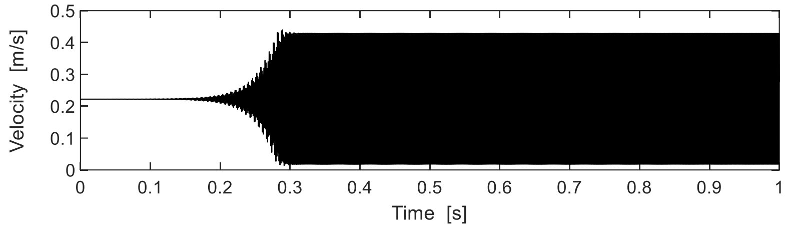

Fig. 3. Wheel self-sustained vibration.

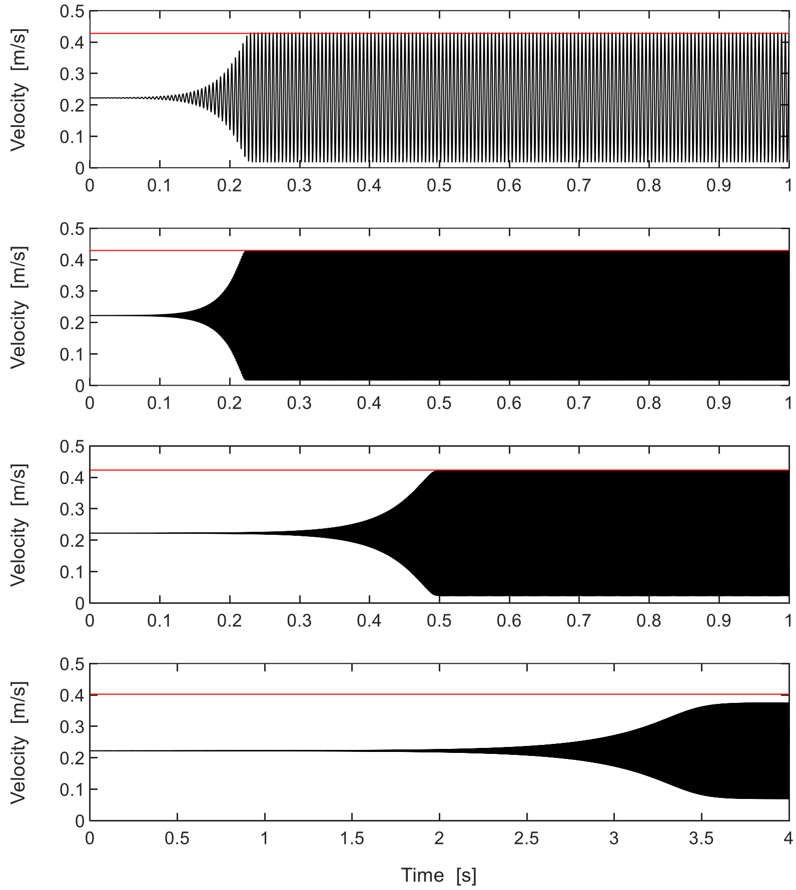

Fig. 4. Wheel self-sustained vibration, frequency contributions. Top to bottom: 165 Hz (0L1), 676 Hz (0L2), 1786 Hz (0L3), 3138 Hz (0L4). Black: ISVR model, red: VibraTec model.

The time evolution of single unstable frequency contributions was also computed, by considering a wheel modal base containing one mode at a time (see Fig. 4). The

VibraTec model results correspond to the amplitude of the limit cycle harmonic motion at the unstable frequency (Table 4). Despite the differences in the formulations, the amplitudes of stationary vibrations are very similar.

Table 4. Limit cycle amplitudes (in m/s) for each frequency contribution.

Unstable frequency	ISVR model	VibraTec model
165 Hz (0L1)	0.4295	0.4276
676 Hz (0L2)	0.4281	0.4293
1786 Hz (0L3)	0.4210	0.4230
3138 Hz (0L4)	0.3746	0.4023

Fig. 5 shows the spectra of the vibration velocity (Fig. 3) for different time intervals: from 0 to 0.25 s (transient regime) and from 0.5 to 0.75 s when the stationary regime known as limit cycle has developed. During the buildup phase, four frequencies coexist, but only the frequency 676 Hz (0L2 mode) and its harmonics persist in the final stationary regime. Previous unpublished measurements highlighted that for this configuration squeal occurs for frequency ranges close to the 0L2 mode. The numerical simulation results are thus very encouraging.

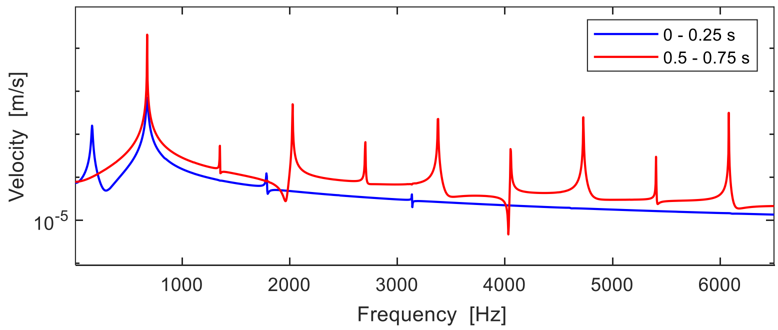

Fig. 6. Vibration velocity spectra at different time intervals. Blue: transient regime. Red: stationary regime (limit cycle).

5 Conclusions

A comparison between two different squeal models is presented. Calculations are performed with the same input data, corresponding to a tramway test case. Despite the differences in formulation and simplifying hypothesis used in the VibraTec model, results are comparable.

This work allowed to identify strengths and weaknesses of models developed in the framework of the QuieterRail project. Both the ISVR and VibraTec models predict potential unstable frequencies in a similar fashion and give similar results. The VibraTec model allows a fast conservative estimation of wheel vibrations. This approach is helpful in industrial applications, where the aim is to find mitigations measures capable of avoiding squeal instabilities. The ISVR model provides an accurate prediction of the time evolution of self-sustained vibrations from the transient regime to the limit cycle. Such a model is needed to study the accurate physics of squeal.

Future work comprises comparisons against field measurements on a tramway network as well as model enhancements such as consideration of flange contact and the complete wheelset, and the introduction of longitudinal creepage effects.

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