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The Influence of Wheel/Rail Contact Conditions on Curve Squeal Noise: Experimental and Numerical Investigation

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Abstract

Curve squeal is still one of the most relevant railway noise problems in urban areas, disturbing thousands of inhabitants every day. In this paper, the variability of curve squeal generated by a modern low floor tramcar is investigated through noise and vibration measurements. Various wheel vibration modes are intermittently found to dominate the wheel vibration during a single curve negotiation. Curve squeal on the outer wheel is also highlighted. Dissimilar squealing patterns are observed in tramcars of the same type that negotiate the same curve. These findings reveal that curve squeal exhibits significant variability due to small changes in wheel/rail contact conditions, rendering the phenomenon, in some instances, highly unpredictable. Numerical simulations are carried out to give an explanation of these experimental findings. Curve squeal occurrence is predicted on both inner and outer wheels at frequencies close to the measured ones. Furthermore, it has been observed that a minor change in the track gauge can consistently influence the contact conditions in extremely tight curves. The contact between the flange back of the inner wheel and the check rail is found to alter the squealing frequencies predicted for a single contact point. It is shown that reducing the friction coefficient can mitigate the occurrence of curve squeal.

Keywords: curve squeal, noise measurements, vehicle dynamics, wheel/rail contact, stability analysis, tramways.

1 Introduction

Curve squeal is a loud and very annoying tonal noise, which often occurs when a rail vehicle negotiates a tight curve. This phenomenon is a serious problem in densely populated areas, affecting thousands of inhabitants every day.

In the last few decades, different mechanisms have been proposed to describe the root causes of the physical phenomenon. Curve squeal is usually attributed to the wheel/rail self-excitation caused by the falling behaviour of the friction curve in fully sliding conditions [1] or to coupling phenomena between two modes of the wheel [2] or between the wheel and the rail dynamics [3], induced by the contact forces that couple the system dynamics in normal and tangential directions. A comprehensive literature review of the several experimental and numerical investigations concerning curve squeal is provided by Thompson et al. in [4]. Curve squeal predictions can be carried out using either frequency- or time-domain formulations. In the former, the system is linearised for small fluctuations in friction force about the steady-state curving condition and the stability of the linearised system is studied to determine potential unstable frequencies [1], [5]. In the latter, the nonlinear equations are solved directly in the time domain [6], [7]. The role of the track dynamics and its conditions has been investigated theoretically in [3] and experimentally in [8], [9]. Experimental evidence and numerical predictions usually identify curve squeal in proximity to the natural frequencies of the wheel axial modes. Although the leading inner wheel of the vehicle is usually found to be the most critical concerning curve squeal occurrence [1], [4], also the outer wheel has been found to squeal in [2], [9]. Moreover, the presence of an additional contact between the back of the wheel flange and the groove rail on the inner leading wheel is found to alter the squealing frequencies and the wheel modes involved in the phenomenon [10]–[12]. All the experimental and the numerical analyses confirm the chaotic and intermittent nature of curve squeal, which is significantly influenced by the wheel/rail contact conditions, including the friction coefficient and the wheel/rail contact angle. These factors may vary significantly depending on the wear of the wheel/rail profiles. During the curve negotiation, contact conditions may also experience slight variations due to the differences in the track alignment caused by position tolerances in the track gauge. Thus, curve squeal predictions performed adopting a specific friction coefficient or fixed wheel/rail contact conditions may differ with respect to what highlighted in measurements. A comprehensive curve squeal prediction should consider all the possible uncertainty in the wheel/rail contact conditions. To include contact variability into curve squeal predictions, a statistical approach has been adopted in [10]. Hundreds of curve squeal simulations are carried out by systematically varying wheel/rail contact conditions with respect to the ones identified in a vehicle dynamics simulation adopting nominal wheel and rail profiles.

The objective of this research is to analyse some of the experimental findings previously presented in [8]–[11], which revealed intriguing variability in curve squeal occurrence, using numerical simulations. For this reason, the same numerical approach adopted in [11] is used to extend the analysis performed on the inner leading wheel of the vehicle also on the outer wheels. Simulations varying the track gauge are carried out to give explanations of some of the experimental evidence.

The paper is organized as follows. Noise and vibration measurements are reported in Section 2 to highlight the variability of squeal events in similar situations. The modelling approach adopted to predict the squeal occurrence is introduced in Section 3. The results of the simulations are presented in Section 4 and final comments are provided in the concluding section.

2 Noise and vibration measurements

Modelling curve squeal remains one of the most challenging tasks in addressing railway noise phenomena, given its intricate nature and inherent unpredictability. The root causes of the physical mechanisms are still unclear, despite several experimental and numerical investigations. Undoubtedly, curve squeal is strongly related to the friction conditions between the wheel and the rail. Understanding the behaviour of the friction coefficient during wheel/rail interaction for high values of relative sliding velocity between the two bodies is crucial to comprehend the reasons behind squeal occurrence and to develop accurate predictive models. Several tests have been conducted adopting reduced scale and full-scale roller rigs to capture the behaviour of friction coefficient under varying longitudinal and lateral creepages. Most of the tests highlight a decrease in the value of friction coefficient for increasing value of creepages. However, curve squeal was also found in presence of constant friction coefficients. The overview of all these findings is reported in [4].

Numerous experimental campaigns were performed by the authors in the last decade, mainly analysing the squeal noise behaviour of similar tramcars in different sites and under different environmental conditions. The results obtained through on-track measurements highlight strong variability also in quite similar conditions. All the measurements revealed that curve squeal was favoured in the presence of dry rails, but high squeal noise levels were also very frequently found suddenly after rain fall, while the track was drying. Friction modifiers such as lubricant and water have been proved to be very effective in completely suppressing the development of curve squeal [9]. Nevertheless, the variability of curve squeal events is observed to be linked not solely to the friction coefficient or, more broadly, environmental conditions. One may expect to find always similar squealing frequencies while tramcars of the same model (with identical vehicle architecture and equipped with the same type of wheels) run into the same curve in comparable environmental conditions at the same speed. This was not the case in results highlighted in [11]. This recent research emphasized that the squealing frequencies associated with the phenomenon can be significantly influenced by additional factors that define the actual wheel/rail contact conditions. The wear of the wheel and the rail profiles and the track construction tolerance (i.e. local variation in the rail gauge) result in a local modification of the wheel/rail contact position, contact angle or in the development of multiple wheel/rail contact points. These effects can cause intermittent behaviour of the phenomenon during curve negotiation but also frequency shift or changes in the wheel modes involved in the unstable mechanism. Some of these findings are presented in the following section.

Curve squeal noise and vibration measurements in two different sites are reported to highlight its variability and its dependence on the wheel/rail contact conditions. Different squealing frequencies are observed not only across different tests but also within the negotiation of a single curve. This analysis aims to show that also minor changes in the wheel/rail contact conditions can result into different squealing phenomena. All the measurements reported here are outcomes from curve negotiation at constant speed (10 km/h) of modern low floor articulated tramcars with identical architectures. These units consist of seven carbodies, with four mounted on individual bogies, while the remaining three are suspended between the bogied ones. The vehicles were running on typical tramway tracks with grooved rails. The only difference between the two sites is the curve radius, which is 17.5 m at site 1 and 24 m at site 2.

Noise and vibration measurements at site 1 (curve radius R=17.5 m)

The experimental campaign performed at site 1 involved both noise and vibration measurements. While track-side microphones were adopted to measure noise emitted by the whole tramcar, accelerometers were installed on the leading inner and outer wheels of the second bogie of the vehicle, in proximity to the wheel rim. With this sensor configuration, it is possible to point out the individual contribution of a single wheel to the overall generated squeal. In fact, the track-side microphones measure the noise emission of all the wheels so it is not straightforward to know whether different squealing frequencies should be attributed to a single wheel or to multiple wheels that are squealing together. Further details about the experimental setup are reported in [9]. First, the spectrogram of the noise emitted by the tramcar during the negotiation of the curve is shown in Figure 1. Sound pressure is measured by a microphone placed on the inner side of the curve at 2.5 m from track centre. Strong noise emission can be seen in proximity to 1500 Hz. A tonal but more intermittent contribution around 2500 Hz and 3800 Hz is also observed. These two squealing frequencies appear to be less persistent compared to the one at 1500 Hz.

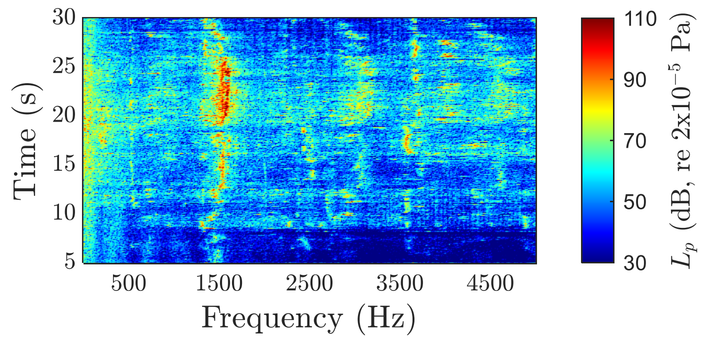


Figure 1: Spectrogram of SPL measured at site 1 (microphone placed on the inner side of the curve at 2.5m from track centre).

The analysis is further refined evaluating the vibration of the wheel rim in axial and radial direction (Figure 2). The spectrogram of the wheel acceleration (Figure 2a) highlights the presence of different peaks in the vibration levels in axial and radial directions close to 1500 Hz, 2500 Hz and 3800 Hz. In this case, the presence of multiple tonal contributions is more evident compared to the analysis performed with the microphone. Tonal wheel vibration at 550 Hz is also observed for a short time window at curve entrance and curve exit.

An Experimental Modal Analysis (EMA) has been carried out on the resilient wheel of these tramcars. Results are reported in [11]. The vibration peaks observed in Figure 2a are very close to the natural frequencies of the wheel axial mode with 2 Nodal Diameters (ND) at 535 Hz, the wheel axial and radial modes with 3 ND (1273 Hz and 1423 Hz), the wheel axial and radial modes with 4 ND (2230 Hz and 2479 Hz) and the wheel axial mode with 5 ND (3736 Hz). This is coherent with the experimental and numerical evidence also reported in literature, where curve squeal is usually found in proximity to the natural frequencies of wheel axial modes and sometimes close to the ones of the wheel radial modes. It may be noted that unlike in monobloc wheels, the mode shapes of resilient wheels are strongly coupled in the two directions especially at high frequencies, due to the presence of the rubber elements between the wheel rim and the wheel web (see [11]).

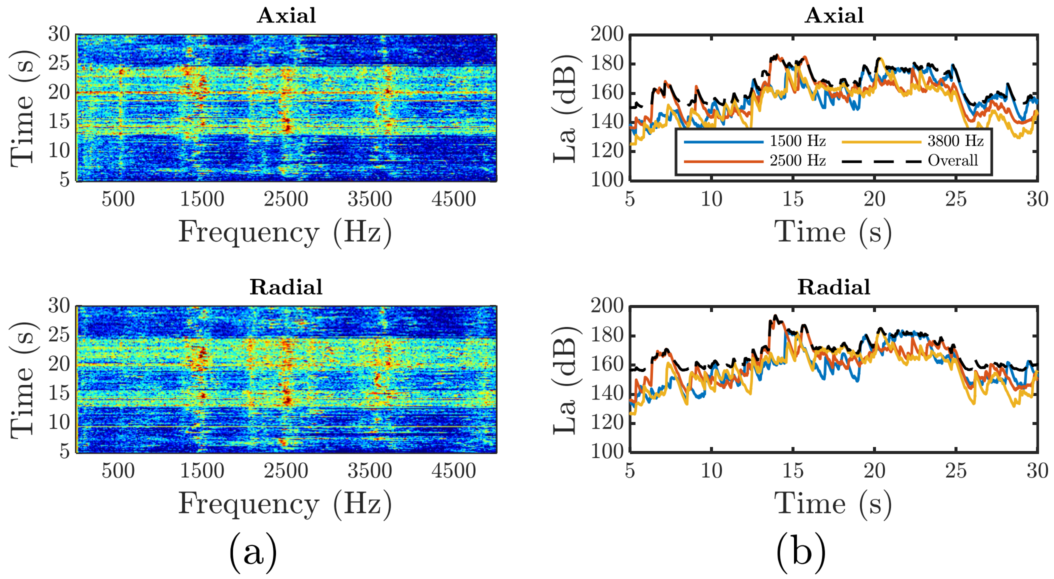


Figure 2: Vibration measurements at site 1: (a) spectrograms and (b) fast time weighting of the inner wheel acceleration ($a_{ref} = 10^{-6}\ m/s^2$) in axial and radial directions.

The signals are also processed adopting the fast time weighting (results are reported in Figure 2b). In this case, the overall vibration level is compared with the level in specific frequency bands that are defined to isolate the main tonal contributions highlighted in the spectrograms. While it might initially seem that multiple modes are simultaneously engaged in the squealing phenomenon when examining spectrograms, Figure 2 underscores a distinct intermittent behaviour. This consistently reveals a single dominant frequency band in the overall wheel vibration, which varies at different time instances. This is the evidence that multiple wheel modes can be involved in squeal during the negotiation of a specific curve, yet only a single mode dominates the limit cycle at a given time. This is typically also found in time domain simulations [5]–[7]. The vibration measured by the accelerometers on the outer wheel is shown in Figure 3. Tonal vibration is found close to 1500 Hz during most of the curve negotiation. A tonal contribution at 2500 Hz is also observed for a short time window at curve entrance and curve exit.

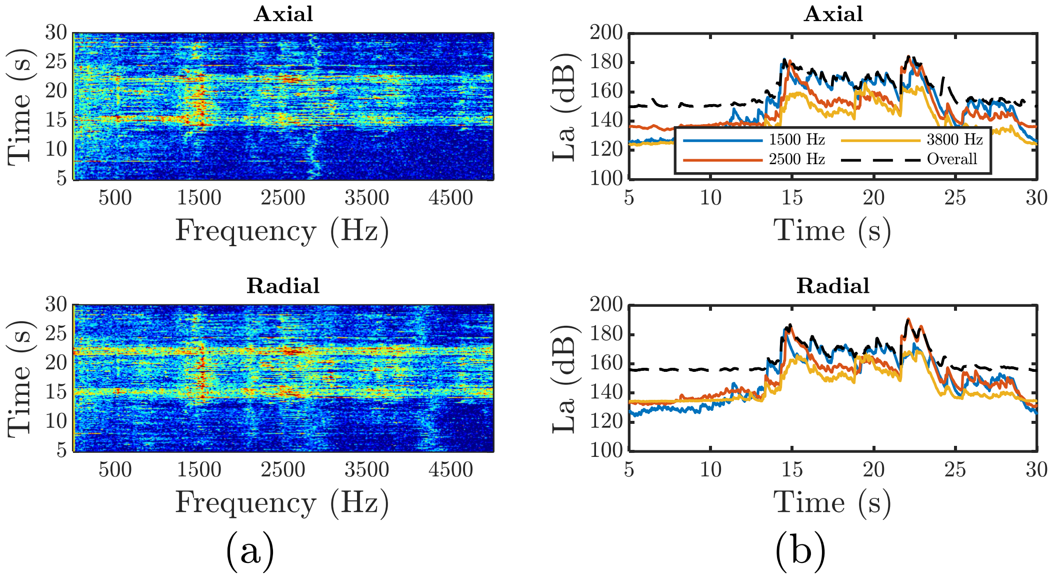


Figure 3: Vibration measurements at site 1: (a) spectrograms and (b) fast time weighting of the outer wheel acceleration ($a_{ref} = 10^{-6}\ m/s^2$) in axial and radial directions.

Noise measurements at site 2 (curve radius R=24 m)

Noise measurements conducted at site 2 allow comparison of the noise emitted by different tramcars with identical vehicle architecture in the same environmental conditions (see Figure 4). These results reveal quite dissimilar squealing patterns. The first tramcar (Figure 4a) generated squeal close to 550 Hz and 2500 Hz while the noise emitted by the second one (Figure 4b) is close to 1500 Hz and 2500 Hz. The differences between the spectrograms of Figure 4a and Figure 4b were attributed to the presence of single and multiple contact points between the leading inner wheel of each bogie and the rail [11]. All these experimental findings suggest that a comprehensive assessment of curve squeal occurrence on a generic rail vehicle should include an analysis of various combinations of wheel/rail contact conditions. Because it is difficult to reproduce a specific situation, numerical simulations can be useful to evaluate the impact of a change in wheel/rail contact conditions. In the following sections, the numerical methodology in the frequency domain adopted to predict squeal occurrence is briefly described. This approach is then used to analyse the potential squeal occurrence on the inner and outer leading wheels of the tramcar under varying wheel/rail contact conditions. The effectiveness of a potential mitigation solution, such as reducing the friction coefficient through the use of water or friction modifiers, is finally assessed.

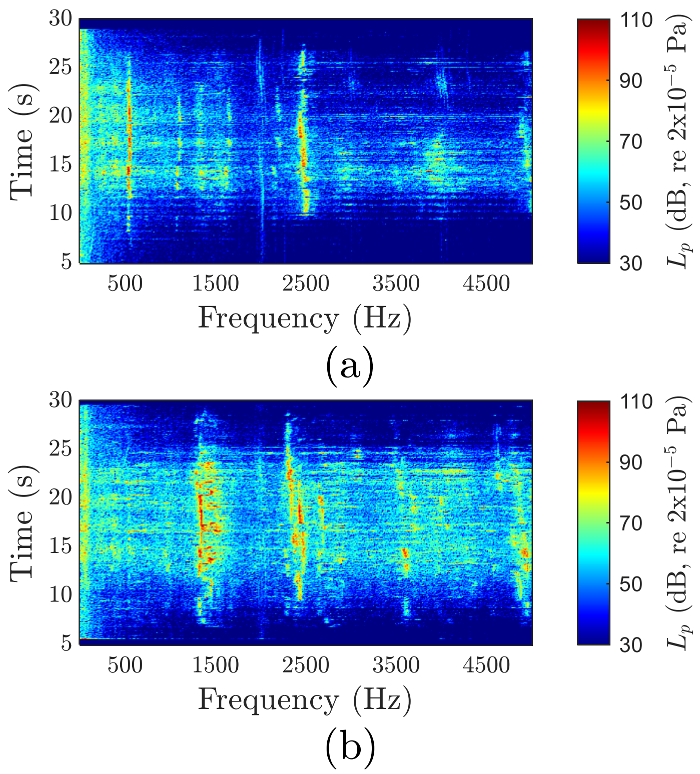
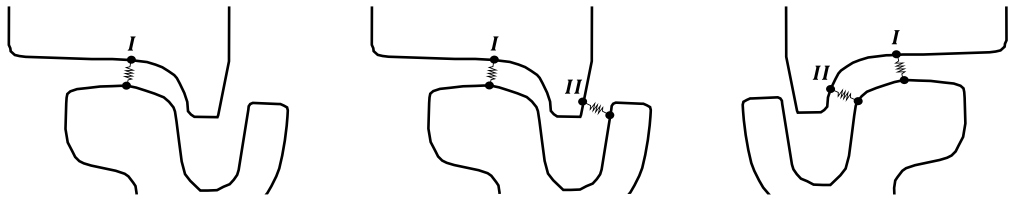


Figure 4: Spectrogram of SPL measured at site 2 in case of (a) single and (b) multiple contact points between the wheel and the rail (microphone placed on the inner side of the curve at 2.5m from track centre).

3 Modelling approach

The prediction of curve squeal occurrence involves two sequential calculation steps. The vehicle dynamic behaviour is first simulated in the time domain to analyse the tramcar behaviour during curve negotiation. Vehicle dynamics simulation is carried out through a multibody software which has been validated against experimental campaigns on different kind of tramcars [13]. In the second step, curve squeal occurrence is predicted through a wheel/rail coupled model formulated in the frequency domain. For this the vehicle dynamics parameters are linearized about a steady-state curving condition. This approach enables the stability analysis of the wheel/rail coupled system concerning that steady state condition, with system instability being linked to the potential development of squeal phenomena.

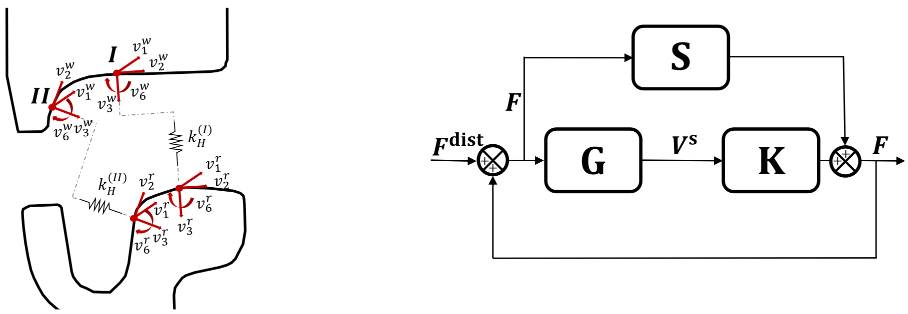
The model adopted in this research is based on the work done by Huang [5] and extended in [10], [11] to evaluate the impact of multiple contact points between the wheel and the rail. The wheel/rail interaction is described by means of a point-contact model. In this model, the wheel and the rail are coupled though a Hertzian spring and creep forces [1], [11]. A single contact point formulation is generally suitable to reproduce the contact conditions between the inner leading wheel and the rail (see Figure 5a). However, a multiple contact point formulation is necessary in presence of a second contact point between the wheel flange back and the check rail (Figure 5b) or when dealing with wheel/rail contact on the outer leading wheel (Figure 5c).



(a) (b) (c)

Figure 5: Overview of potential wheel/rail contact conditions on the tramcar leading axle during a left-hand curve: (a) inner wheel, single contact point on the wheel tread, (b) inner wheel, multiple contacts on the wheel tread and on the flange back and (c) outer wheel, contacts on the wheel tread and on the wheel flange.

The scheme of the wheel/rail interaction is shown in Figure 6a. According to [1], [5], the wheel/rail interaction can be schematized as a Multi-Input Multi-Output (MIMO) system (see Figure 6b).



(a) (b)

Figure 6: (a) Scheme of wheel/rail interaction and (b) wheel/rail self-excited loop.

The vector $\boldsymbol{V^s} = [v_1^s \quad v_2^s \quad v_6^s]^T$contains the longitudinal, transverse and spin sliding velocities and $\boldsymbol{F} = [f_1 \quad f_2 \quad f_6]^T$ is the vector containing the creep forces and the spin moment. The matrices $\boldsymbol{S}$ and $\boldsymbol{K}$ capture the influence on creep forces related to fluctuations in the creepages and in the normal load. A slip-dependent friction coefficient is included by adopting the heuristic formula also used in [5], [10] in the Shen Hedrick Elkins tangential contact model ($\mu(\gamma_{tot}) = \mu_0(1 - \lambda e^{-\tau/\gamma_{tot}})$) [14].

The matrix **G** contains the dynamics of the coupled system, including the wheel, the rail and the contact mobilities. The contact mobility is computed through the linearized Hertzian spring [1]. Wheel dynamics is incorporated by means of a modal superposition approach. The modeshapes of the resilient wheel of the tramcar are computed with a Finite Element (FE) model of the wheel. The model is calibrated with the natural frequencies and the damping ratios obtained through an impact test. The results of the modal analysis are reported in [11]. The amplitude of the wheel mobility in the lateral and vertical directions is presented in Figure 7.

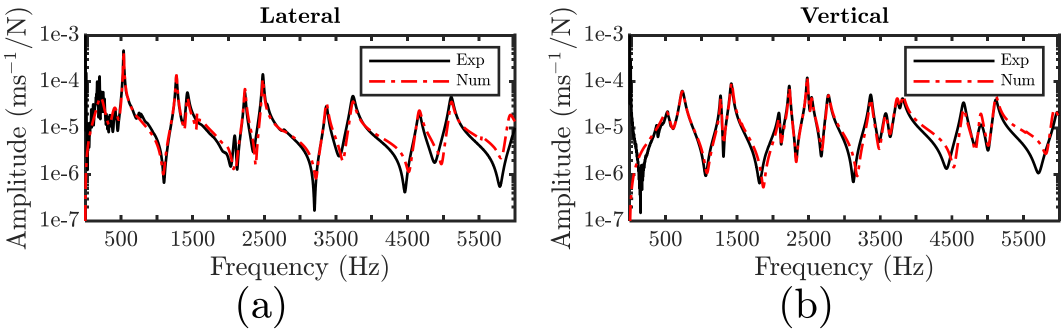


Figure 7: Wheel mobility in (a) lateral and (b) vertical direction.

Track mobilities are included by adopting a single Timoshenko beam on visco-elastic foundation [1]. The parameters of the model are calibrated by fitting experimental Frequency Response Functions (FRFs) of the Embedded Rail System (ERS) installed at site 2 (see Figure 8). Longitudinal and spin mobilities are neglected due to the high impedance of the track in these directions.

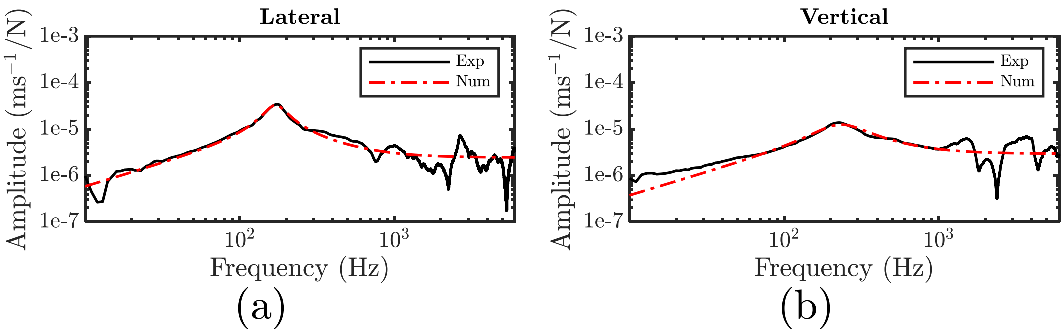


Figure 8: Track mobility in (a) lateral and (b) vertical direction.

Stability analysis is carried out through the Generalized Nyquist criterium for MIMO systems [1], [5]. Curve squeal prediction is carried out by generating 200 different cases, each one corresponding to a different combination of steady state curving parameters. These parameters are obtained by introducing a random variation through a uniform distribution centred in the steady state parameters identified in the vehicle dynamics simulation. Friction coefficient and falling friction parameters are the same as those adopted in [11].

4 Results

In this Section, the methodology described in Section 3 is adopted to analyse the effect of different wheel/rail contact conditions on curve squeal occurrence. A series of multibody simulations is carried out varying the track gauge to replicate various wheel/rail contact conditions that can arise due to track construction tolerances and wheel/rail profile wear. Simulations are carried out considering a curve radius of 17.5 m and a curve radius of 24 m to replicate the characteristics of the two sites (see Section 2). The tramcar speed is set to 10 km/h. For each situation, curve squeal predictions considering nominal track gauge (1445 mm) and an increased track gauge (1447 mm) are performed. In the nominal track gauge case, a single contact point on the wheel tread of the inner wheel is observed (as schematized in Figure 4a) for both sites. By increasing the rail gauge (1447 mm) a second contact point between the inner wheel and the check rail appears (see Figure 4b). This was also noted during the experimental campaigns conducted at both sites, revealing significant wear on the check rail. A similar variation in wheel/rail contact conditions can be expected due to the wear of the wheel and the rail profiles or due to the track construction tolerances. An overview of the normal contact forces acting on the inner and outer wheels is shown in Table 1.

		(a) Inner wheel		**(b) Outer wheel**	
Curve radius (m)	Track gauge (mm)	F_n^I (kN)	F_n^{II} (kN)	F_n^I (kN)	F_n^{II} (kN)
R=17.5	1445	32.8	-	16.7	24.1
	1447	28.4	5.8	20.7	18.4
R=24	1445	32.8	-	17.3	23.8
	1447	30.1	4.3	20.3	19.3

Table 1: Steady state normal load on tread, flange back and flange contact points varying the track gauge: (a) inner wheel and (b) outer wheel.

The results of the curve squeal prediction obtained for site 1 are reported in Figure 9. High curve squeal occurrence is observed close to 535 Hz and 2500 Hz in case of nominal track gauge (1445 mm). Lower curve squeal occurrence is also found at 1300 Hz. Instabilities are all close to the natural frequencies of the wheel modes with 2,3 and 4 ND. Curve squeal is also found on the outer wheel at 1500 Hz. The increase in the track gauge (1447 mm) leads to the presence of two contact points also on the inner wheel. It is observed that the occurrence of curve squeal in proximity to the wheel axial mode with 2 ND is completely suppressed. This is due to the effect of the flange back contact that prevents the excitation of this mode, which is almost purely axial. High curve squeal occurrence on the leading inner wheel is now found close to 1500 and 2500 Hz. It must be noted that these modes are characterized by a high axial and radial modal amplitude in proximity to the contact area (they are strongly coupled in the two directions). Despite the flange back providing a constraint to the wheel in the axial direction, it may promote the excitation of radial modes or, in this specific case, the modes with high modal component in the radial direction. The critical role of the check rail was also found in [4], revealing severe noise levels linked to the contact between the wheel and the check rail near the natural frequencies of the radial modes with 2 and 4 ND. Curve squeal on the outer wheel is again seen close to 1500

Hz. The result of this second set of simulations, adopting an increased track gauge, seems to be very close to the vibration pattern observed on the inner and outer leading wheels of the tramcar running in site 1 (see Section 2). The leading inner wheel is most of the time vibrating at 1500 Hz or at 2500 Hz (see Figure 2b) while the outer wheel is vibrating at 1500 Hz during the whole curve negotiation (see Figure 3b).

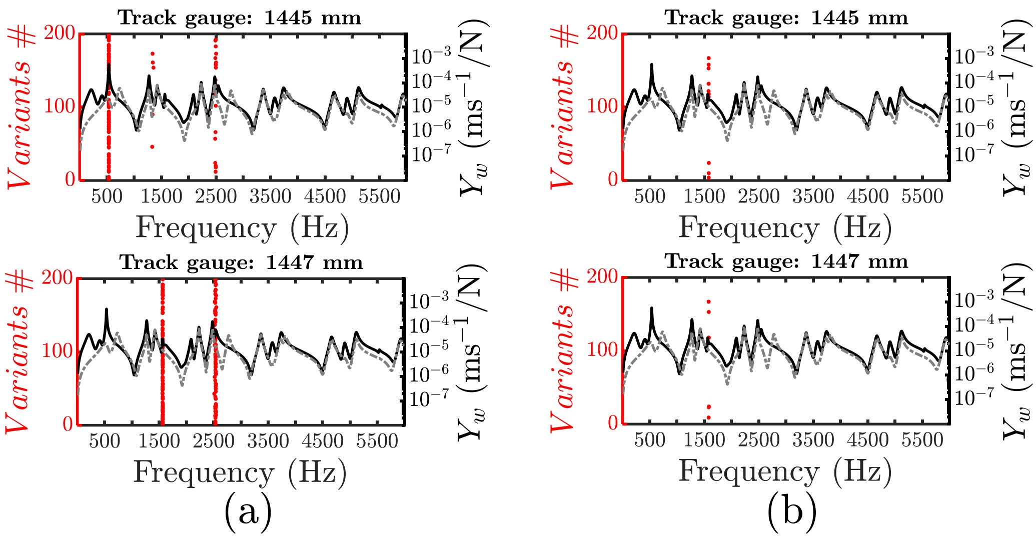


Figure 9: Curve squeal occurrence in scenario 1 simulation (R=17.5 m) varying the track gauge: (a) inner wheel and (b) outer wheel. Unstable frequencies are marked with red dots (•) against the wheel mobility in axial (--) and radial (--) directions.

The results of the curve squeal simulations based on site 2 (R=24 m) are shown in Figure 10. The results are similar to those from case 1. Higher curve squeal occurrence is observed close to 1300 Hz for the nominal track gauge. Some instabilities are also detected close to the natural frequencies of the wheel modes with 5 and 6 ND. When the track gauge is increased, curve squeal is found close to 1500 Hz and 2500 Hz, similarly to the simulation results for site 1. Predictions on the inner wheel are in agreement with the noise measurements from site 2. The instabilities identified in the simulations with a nominal track gauge are similar to the squealing frequencies found in case of single contact point between the wheel and the rail (Figure 4a) while the simulations obtained with increased track gauge are coherent with the squealing frequencies measured in case of multiple contact points between the inner leading wheel and the rail (Figure 4b). A more detailed analysis on this set of simulations can be found in [11]. Squeal at 1500 Hz is found on the outer wheel. The number of unstable cases in Figure 9b and Figure 10b is much lower than those observed in the inner wheel simulations. This confirms that the inner wheel is the most critical concerning squeal phenomena, although squeal can still occur on the outer wheel.

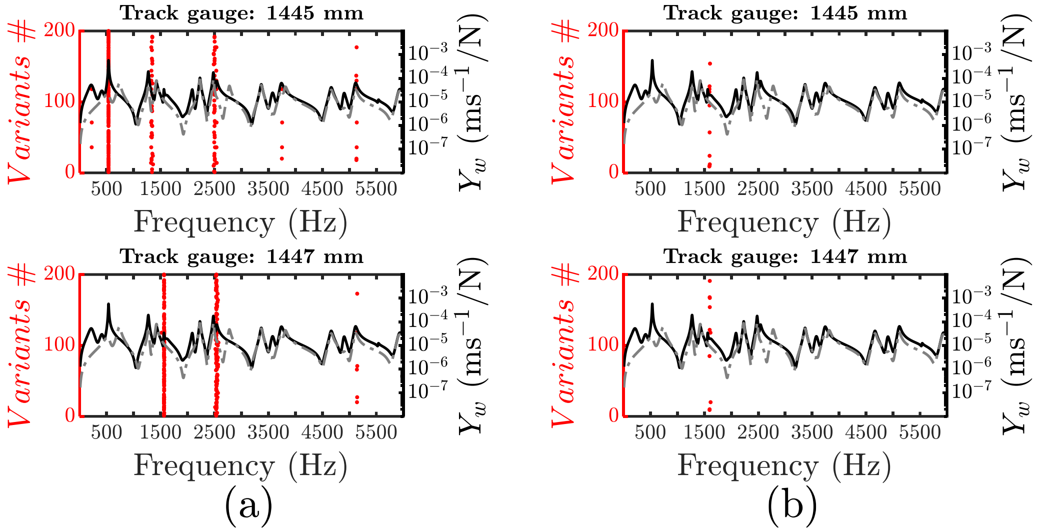


Figure 10: Curve squeal occurrence in scenario 2 simulation (R=24 m) varying the track gauge: (a) inner wheel and (b) outer wheel. Unstable frequencies are marked with red dots (•) against the wheel mobility in axial (--) and radial (--) directions.

The methodology is finally adopted to investigate the impact of the friction coefficient on curve squeal occurrence. The simulations of site 2 (R=24 m) with nominal track gauge are repeated by considering three different ranges for friction coefficients. The results are presented in Figure 11a. The instability occurrence (%) is reported, computed as the ratio between the number of unstable points at each frequency over the total number of simulated variants. The curve squeal occurrence decreases with the friction coefficient. No unstable cases are found for a friction coefficient lower than 0.2. This was also found in experiments at site 2, where the track was artificially wetted highlighting that water can completely mitigate the phenomenon. Two passages of the same tramcar over wet and dry rail are compared in Figure 11b. In contrast with the dry case, no tonal noise is found in the wet case and the noise levels are more than 15 dBA lower over the whole curve. This analysis highlights the potential effectiveness of adopting wheel/rail friction modifiers to mitigate curve squeal.

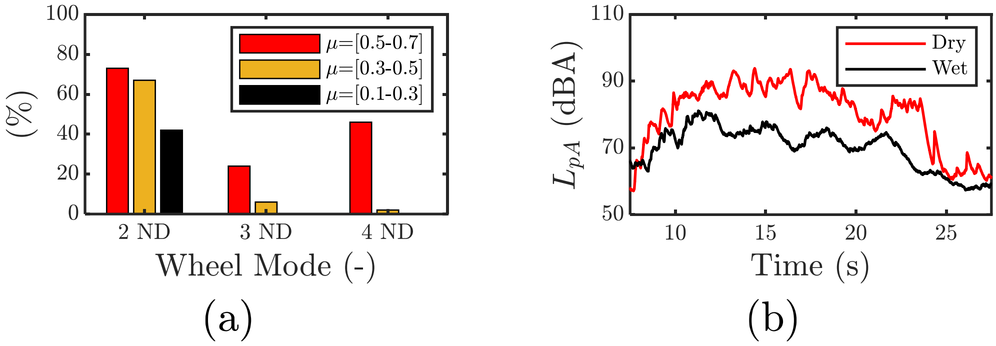


Figure 11: Curve squeal occurrence varying the friction coefficient: (a) numerical simulations and (b) pass-by noise measured at site 2 for dry and wet rail.

5 Conclusions

In this article, a set of noise and vibration measurements of curve squeal generated by modern low floor tramcars in two different sites is analysed. Measurements performed at site 1 allow the intermittent nature of curve squeal through the measurements of wheel vibration levels. It is demonstrated that squeal occurs in various wheel vibration modes during the curve negotiation. Nevertheless, it is also observed that the wheel's vibration is consistently dominated by a single mode at any given time. Vibration measurements also detect curve squeal on the leading outer wheel of the same bogie. The analysis of the noise measurements from site 2 reveal different squealing frequencies of two tramcars with identical architectures that negotiate the same curve a short time apart. This is attributed to different wheel/rail contact conditions due to a different wear of the wheel profiles of the two vehicles. All these findings confirm that realistic prediction of curve squeal requires consideration of some variability in the wheel/rail contact conditions. Numerical analyses to give explanation of some of these experimental findings are carried out through vehicle dynamics simulations in the time domain and curve squeal predictions in the frequency domain. The analysis is performed varying the wheel/rail contact conditions to highlight all the possible wheel modes involved in the phenomenon. Simulations adopting different track gauge are used to reproduce the effect of possible track misalignment and/or the effect of the wear of wheel/rail profiles. These simulations reveal that a small change in track gauge results in the presence of a second contact point between the wheel flange back and the check rail. This contact condition can alter the frequencies involved in the squealing events. The predicted squealing frequencies are close to the ones measured at the two sites. The numerical simulations also reveal curve squeal on the outer wheel, as observed at site 1. The occurrence of curve squeal on the outer wheel is however limited to a lower number of cases with respect to the inner wheel. This is coherent with many experimental investigations found in literature, where curve squeal is usually observed on the leading inner wheel of the vehicle. The effect of friction modifiers is finally simulated by reducing the value of the friction coefficient between the wheel and the rail. The decrease in the friction coefficient leads to a significant reduction in the instability occurrence, which confirms that curve squeal can be effectively mitigated through the application of water or other wheel/rail friction modifiers.

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