Nonlinear dynamics of curved railway tracks in three-dimensional space

Liu, Xin ; Ngamkhanong, Chayut; Kaewunruen, Sakdirat

DOI:
10.1088/1757-899X/280/1/012036

Nonlinear dynamic of curved railway tracks in three-dimensional space

X Liu[1], C Ngamkhanong[1,2] and S Kaewunruen[1,2]

[1]Department of Civil Engineering, School of Engineering, The University of Birmingham, Birmingham, B15 2TT, United Kingdom
[2]Birmingham Centre for Railway Research and Education, The University of Birmingham, Birmingham, B15 2TT, United Kingdom

E-mail: CXN649@student.bham.ac.uk

Abstract. On curved tracks, high-pitch noise pollution can often be a considerable concern of rail asset owners, commuters, and people living or working along the rail corridor. Inevitably, wheel/rail interface can cause a traveling source of sound and vibration, which spread over a long distance of rail network. The sound and vibration can be in various forms and spectra. The undesirable sound and vibration on curves is often called 'noise,' includes flanging and squealing noises. This paper focuses on the squeal noise phenomena on curved tracks located in urban environments. It highlights the effect of curve radii on lateral track dynamics. It is important to note that rail freight curve noises, especially for curve squeals, can be observed almost everywhere and every type of track structures. The most pressing noise appears at sharper curved tracks where excessive lateral wheel/rail dynamics resonate with falling friction states, generating a tonal noise problem, so-call 'squeal'. Many researchers have carried out measurements and simulations to understand the actual root causes of the squeal noise. Most researchers believe that wheel resonance over falling friction is the main cause, whilst a few others think that dynamic mode coupling of wheel and rail may also cause the squeal. Therefore, this paper is devoted to systems thinking the approach and dynamic assessment in resolving railway curve noise problems. The simulations of railway tracks with different curve radii will be carried out to develop state-of-the-art understanding into lateral track dynamics, including rail dynamics, cant dynamics, gauge dynamics and overall track responses.

1. Introduction

Curve squeal is a strongly tonal noise emitted from wheel/rail contact caused by the passage of the train in tight curve tracks with a low speed [1]. The occurrence of squeal induces significant environmental impacts immensely annoying people living nearby due to its high frequencies characteristics. Based on previous studies, unsteady lateral creepage of the wheel is thought to be the prime reason of squeal, while other mechanisms such as longitudinal creepage and flange contact, do not necessarily eliminate squeal noise thereby are determined to be of secondary importance [2-4]. A number of researches have been carried out to demonstrate the mechanism of squeal noise by means of site investigation and numerical modeling. Negative damping mechanism has been primarily proposed by Rudd [5]. The friction coefficient, which is relevant to the sliding velocity, falls after creepage exceeds the saturation point. This falling characteristic of creep force at high creepage is believed to be the main reason of squeal. However, the mode coupling instability as a new mechanism between normal and tangential dynamic is getting growing support while the negative friction mechanism is no longer emphasized in research field [6]. It is still questionable at which mechanism results in wheel/rail squeal.

The variation of squeal noise has been observed on-site in conjunction with many critical parameters including curve radii, rail dynamics, cant dynamics and overall track response during operation. Previous work showed that squeal only occurs when the curve radius is smaller than 100b, where b is the wheelbase of the bogie [5]. The results of on-site measurements also presented that there is no substantial reduction in wheel squeal associated with limiting operation speed. The investigation implemented by J.Jiang [7] conducted the track dynamic in terms of fluctuation of curve noise after upgrading from timber sleepers to concrete sleepers. But the question remains as for how other parameters change the lateral track dynamic. In fact, data collected in the field suggest a diverse range of curving behavior, which is largely relevant to curve radii [8,9]. However, it is still uncertain to what extent the track lateral response is affected by rail radius. The occurrence of cant is believed to have positive impacts on reducing curve squeal due to its counteraction with creep force, an analytic approach has to be implemented for the determination of the effective range of cant. In reality, stable and unstable regimes of track deflection are predicted in operation condition due to high lateral load. The track dynamic response characteristics need to be taken into account to justify the track dynamic under different lateral load.

The critical review reveals that lateral track dynamic characteristics have not been fully understood, especially when it comes to various curve radii, cants and lateral loads. The aim of this study is to investigate the lateral track behavior into the effects of their properties. The present analyses are based on numerical modeling, using Timoshenko beams in a general-purpose finite element package STRAND7 to build a three-dimensional track model. Both static and nonlinear transient analysis are considered. Dynamic displacement and acceleration have been evaluated at rail over sleeper, rail at midspan and sleeper at midspan under different curve radii and load cases. Dynamic characteristics are then highlighted in this paper. The findings will provide some key parametric insights into fundamental dynamics of track in the lateral direction and establish the development of the dynamic design of curve track.

2. Track model

The track model consists of a two-dimensional Timoshenko beam, which has been validated to be one of the most suitable options for concrete sleepers modeling [10,11]. In this investigation, the finite element models of railway tracks have been previously developed and calibrated against the numerical and experimental modal parameters [12,13]. A general-purpose finite element package STRAND7 is used for finite element simulation. It is noted that beam elements, which take into account shear and flexural deformations, are considered to simulate sleeper and rails in terms of lateral response. Each sleeper consists of 60 beam elements and each rail consists of 200 beam elements. The overall model was represented by 1348 beam elements. The 60kg rail cross section was introduced in track model. The trapezoidal cross section was assigned to the sleeper elements. The rail pads at rail seats were simulated using a series of spring-dashpot elements. The distance offset between rail and sleepers was set to 100mm which does not have impacts on numerical results [13]. In this study, stiffness and damping values of high-density polyethylene pads were assigned to these spring-dashpot elements both in vertical and lateral direction. Ballast was placed under sleeper, supporting the rails and sleepers, which provides compression only. Tensionless beam support was used to achieve the support condition with an allowance of hover over the support while the tensile supporting stiffness is omitted. It was validated in previous studies, effects of length and boundary of track in this investigation (18 bays or 10.8m) on the computation and frequencies of interest are negligible [10-13].

In total, seven types of curve radii have been evaluated for dynamic investigations using curves ranging from 100m to 600m with tangent one. Also, four types of cants were determined with the range from 0mm to 300mm. Engineering properties have been validated by a parametric analysis which is implemented by consideration of different rail, sleeper and ballast properties. In order to adopt these, the partial support condition is believed to conform with real condition of standard gauge tracks vastly. The nodes at both ends of track were fixed in every direction to represent the track boundary. Four separated forces with various magnitudes have been used to simulate the load condition of a passenger train bogie (2 per each rail, 2 m apart). This load magnitude has been used for benchmarking purpose [14,15]. The nondimensional analyses have been carried out to investigate the dynamic amplification over curve radii and frequency domain.

3. Results and discussion

The Linear Static Solver of STRAND7 has been used to evaluate the static response of railway tracks. The Nonlinear Transient Solver is then used to get dynamic behaviors of railway tracks after being verified by the quasi-static condition. Two moving load envelope with a speed of 10m/s have been established to evaluate the dynamic response of the rail based on an impulse excitation of a period of 0.0001s starting at time t =0.005s. The calculation time step is set to be 5×10^{-5} so that it can include high-frequency modes for responses of curving. With this FE model, the eigenfrequencies and corresponding eigenmodes are calculated up to 10kHz, which is believed to cover modes of squeal noises vastly. The dynamic analyses are taken into account for various curve radii, cants and lateral loads. To appropriately take into consideration of dynamic effects of moving load, calculation time has been set to be 5s thus the entire process of dynamic responses can be adaptively reflected in simulation associated with the length of track and vehicle velocity. Typical deformation and stress distribution of a railway track under the lateral moving load are shown in figure 1. The rails present positive bending due to the lateral load applied perpendicular to load moving direction.

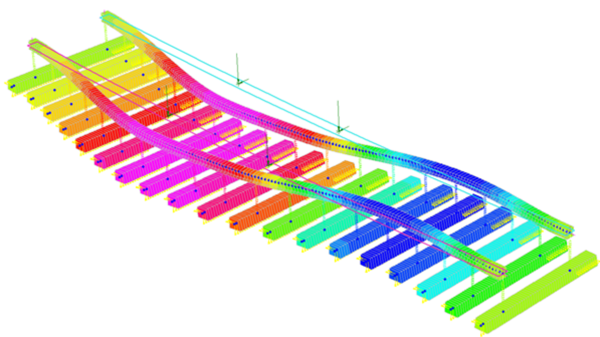


Figure 1. The example of the track's deformation under lateral moving load

3.1. Maximum displacement response

Figure 2 illustrates the dynamic lateral displacements of rail over sleeper at different curve radii. The L/V's are lateral to vertical load ratios where the vertical load was selected as 100KN as a benchmark for passenger train bogie. Varying load ratios were comparatively investigated, and similar trends to the results for the rail dynamic were exhibited. It is clear that the tracks with larger curve radii deform laterally more severe than tight curve tracks due to lower lateral resistance. The result can be effectively explained by arch mechanics that arches with larger curvature possess of higher compressive strength. Take load ratio of 0.2 as an example, displacement increases 75.8% for curve radius varying from 100m to 200m, while the increase rate is only 3.2% for radius varying from 500m to 600m. This implies that lateral displacement responses are more sensitive corresponding to low radii, which gave evidence on the appearance of squeal during train negotiating tight curves. It can also be observed from the graph that the lateral track displacement of the tangent track is similar to the value of track with a radius of 600m. For large curve radius, the lateral displacement of the track no longer changes significantly with increasing radius therefore the increase of radius plays a little role on the dynamic amplitude of track. The results above indicate that the increased track radius has positive effect on reducing curve squeal and squeal noise would disappear when the curve radius comes to a particular value.

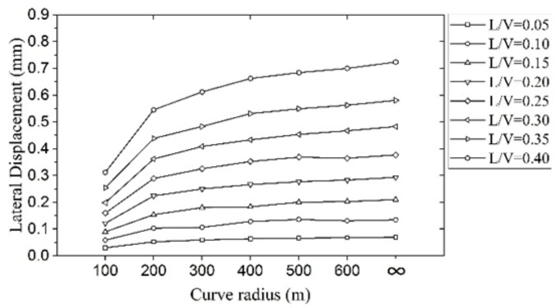
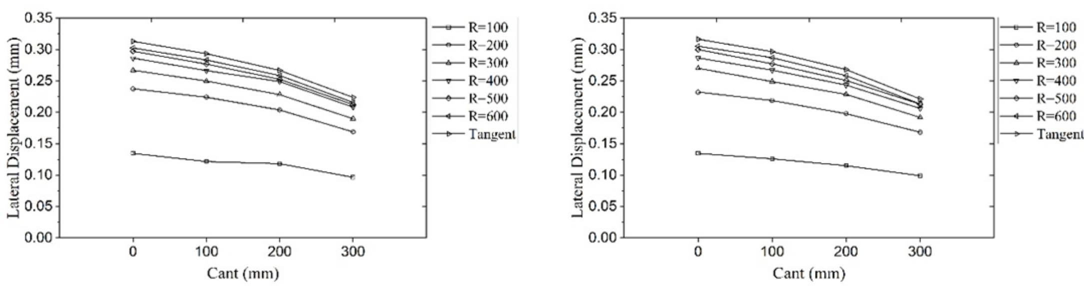


Figure 2. Dynamic lateral displacements of rail over sleeper at different curve radii, cant=100mm

The Rail cant is a crucial parameter in relation to rail noise vibration. To evaluate the effects of cants on dynamic responses, data from Figure 3 exhibits a comparison of dynamic lateral rail displacements based on the simulation performed with a vehicle velocity of V=10m/s for varying cants. The results apparently demonstrate the positive effect of including cant in curve track, which leads to a lower dynamic response when cant goes towards a higher value. The result can be inferred by the occurrence of centrifugal forces and compensative effect of cants. As expected, the railway track without a cant is more likely to cause severe damage to track components and even leads to derailment. Therefore, these negative effects, which could be avoided by adding suitable cants to the railway track, should anyhow not obstruct railway normal operation.



(a) Rail over sleeper (b) Rail at midspan

Figure 3. Maximum lateral displacements of track at a) Rail over sleeper b) Rail at midspan with varying cants

3.2. Lateral Acceleration

Figure 4 shows the corresponding Fourier spectrum for rail vibration at midspan with respect to curve radius of 200m, 500m and infinite. The most distinct resonance of 200m radius track occurs at 150Hz with the value of 131dB while the maximum response of 500m radius track happens at 251Hz with the value of 125.6dB. Therefore, the increase of track radius expands the resonance frequency and reduces the peak of vibration due to the time-variation of response spectrum. It can also be verified by the fact that the acceleration in the tangent track is much lower than that on curved tracks. The track responses are approximately equivalent in amplitude for various curve radius up to 1000Hz, although several discrete resonances and antiresonances appear in this region. On the contrary, the track acceleration in the frequency region of most interest for the squeal, 1000Hz-5000Hz, is much greater on 200m radius track than on 500m radius track and tangent track. This would lead to more unstable modes occur at squeal frequency region, which enhances the severity of squealing. When zoomed in, a comparison of acceleration levels between curve track and tangent track suggests that vibration in tangent track in high frequencies have little relation to curve squeal.

Figure 5 illustrates the effect of curve radius by showing the difference between the responses of rail with radius of 200m and 500m. Insertion gain as an indication of vibration mitigation has been used here to quantify the relative mitigation effects. In the overall frequency domain, the insertion gain distributes mainly in the range of -10dB-30dB. The region at frequencies below 1000Hz is much lower than that at frequencies between 1000Hz and 5000Hz where curve squeal is most likely to occur. Therefore, it is demonstrated that the responses of track in high frequencies are sensitive to track radius thus increasing of curve radius has significant positive effect on reducing squeal noise.

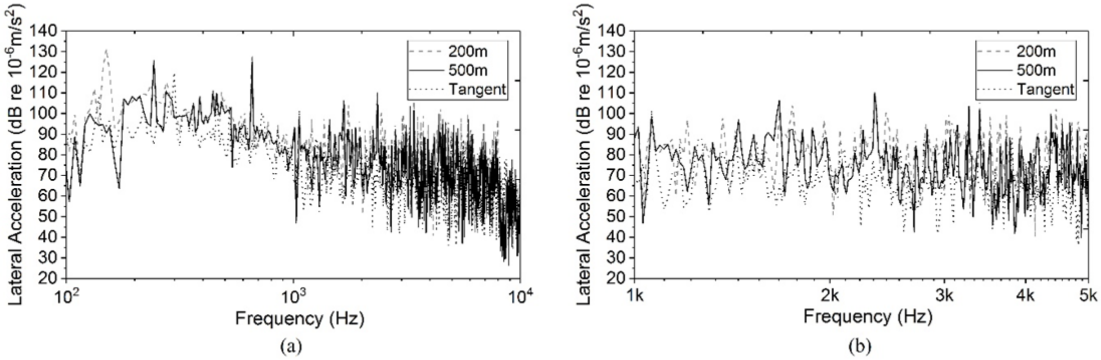


Figure 4. Spectra of the rail lateral acceleration at middle span of rail for varying types of track in the frequencies range of a) 100-10000Hz b) 1000-5000Hz

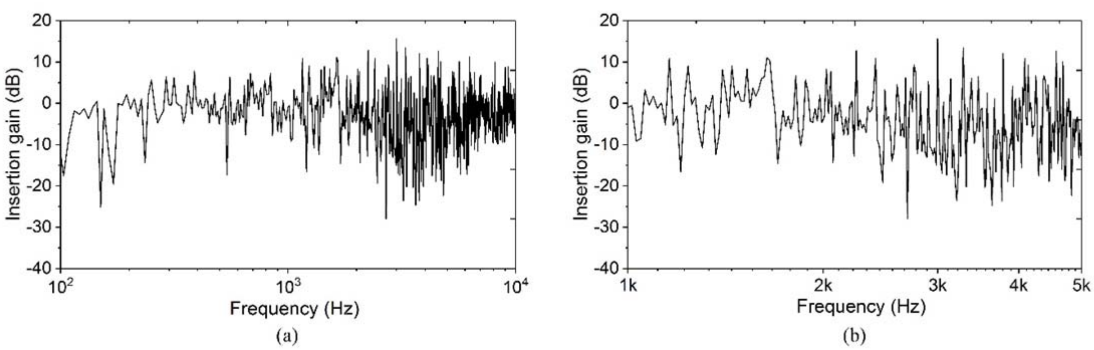


Figure 5. Insertion gain of track vibration due to change in track radius from 200m to 500m in the frequencies range of a) 100-10000Hz b) 1000-5000Hz

4. Conclusion

A numerical simulation has been implemented in this paper to identify the lateral dynamic characteristics of the curved track with many influencing parameters including track radii, cants and lateral to vertical load ratios. Parametric studies performed with curved track model have indicated many important results.

The relative rail displacement can be reduced by the increasing of curve radius. However, for the tight curve, relative displacement responses are more sensitive to the change of radius. This provides evidence on the occurrence of squeal in tight curve. Cant is believed to have a positive effect on dynamic responses. However, a high cant is demonstrated to incur severe dampening performance, therefore, a reasonable range of cants need to be established. Moreover, track acceleration is greater on track with lower radius in higher frequencies. The insertion gain in squeal region is two times of that in lower frequencies. When the results for different track types are compared, it is determined that vibration in the tangent track would not incur squeal noise.

Acknowledgement
The first author would like to thank Beijing Jiaotong University for his sponsorship and also a collaboration between Beijing Jiaotong University and University of Birmingham. The second author gratefully appreciates the School of Engineering and Birmingham Centre for Railway Research and Education of his PhD scholarship. The last author gratefully acknowledges the Japan Society for the Promotion of Science (JSPS) for his JSPS Invitation Research Fellowship (Long-term), Grant No L15701, at Track Dynamics Laboratory, Railway Technical Research Institute and Concrete Laboratory, the University of Tokyo, Tokyo, Japan. The authors are sincerely grateful to European Commission for the financial sponsorship of the H2020-RISE Project No. 691135 "RISEN: Rail Infrastructure Systems Engineering Network," which enables a global research network that tackles the grand challenge in railway infrastructure resilience and advanced sensing in extreme environments (www.risen2rail.eu).

5. References

[1] Thompson D 2009 *Railway noise and vibration: mechanisms, modeling and means of control* 1st ed (United Kingdom: Elsevier Science Technology)
[2] Vincent N, Koch J R, Chollet H and Guerder J Y 2006 *Journal of Sound and Vibration* **293** 691
[3] Koch J R, Vincent N, Chollet H and Chiello O 2006 *Journal of Sound and Vibration* **293** 701
[4] Pieringer A 2014 *Journal of Sound and Vibration* **333-18** 4295
[5] Rudd M J 1976 *Journal of Sound and Vibration* **46-3** 381
[6] Jiang J, Anderson DC and Dwight R 2015 *Proc.Int.Conf. Noise and Vibration for Rail Transportation Systems* (Berlin: Springer) p 587
[7] Jiang J, Ying I, Hanson D and Anderson D C 2015 *Proc.Int.Conf. Noise and Vibration for Rail Transportation Systems* (Berlin: Springer) p 441
[8] Kaewunruen S and Remennikov A M 2010 *Insight-Non-Destructive Testing and Condition Monitoring* **52-1** 20
[9] Kaewunruen S and Remennikov A M 2007 *Ndt&E International* **40-7** 510
[10] Kaewunruen S and Remennikov A M 2008 *Engineering Failure Analysis* **15-8** 1065
[11] Grassie S L 1995 *Journal of Sound and Vibration* **187-5** 799
[12] Kaewunruen S and Remennikov A M 2008 *Experimental Techniques* **32-4** 29
[13] Kaewunruen S, Remennikov A M 2009 *Proc.Int.Conf. Mechanical Vibration* (New York: Nova Science) p 621
[14] Mirza O, Kaewunruen S and Galia D. Seismic 2016 *Structural Engineering and Mechanics* **57-3** 569
[15] Kaewunruen S and Chamniprasart K 2016 *Proc.Int.Conf. on Computational Mechanics* (Gothenburg: Chalmers University Press)