A Smorgasbord of Railway Dynamic and Acoustic Phenomena

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ABSTRACT

Railway technology continues to provide challenges in the dynamical and acoustical analysis of engineering systems that typically involve understanding and controlling undesirable behaviour. This seminar will outline some of the new (and old!) research being undertaken in this area by the Rail Mechanics Group at The University of Queensland, Australia. In particular, results for prediction, measurement and control of rail corrugation, prediction and field investigation of railway noise and wheel squeal will be presented. The paper will provide an overview of major results in these areas and then focus on the prediction of vibration amplitude and sound pressure level of wheel squeal noise. The predicted squeal sound level trend is shown to compare well with testrig results at various crabbing velocities (or angles of attack) and rolling speeds. The results concur with experimental and field observations but also provide important theoretical insight into the useful mechanisms of controlling railway dynamic and acoustic phenomena based on underlying mechanics modelling.

1. INTRODUCTION

As railway technology continues to advance and provide a more efficient and faster means of transportation we will inevitably encounter a higher occurrence of dynamic and acoustic phenomena. Typically this is undesirable. Dynamic phenomena cause vibrations in vehicle/track structures which in turn causes fatigue damage and propagates pressure fluctuations through the air as undesirable sound; namely noise. The rail mechanics group at the University of Queensland has been studying a wide range of phenomena over the past 15 years. These projects include the modelling, prediction and control of rail corrugations, roughness generated rolling noise and wheel squeal. The approach has been to first obtain a detailed understanding of the phenomena via modelling of the important physical mechanisms. These are then validated with field and or laboratory measurements to obtain confidence. The validated predictive models are then used to obtain predictive insight into the critical conditions involved and then subsequently used as a testbed to develop and test mitigation/control techniques. This paper first provides a brief survey of these phenomena highlighting the importance of mechanics based modelling. Finally some recent predictive modelling results for amplitude of wheel squeal noise is provided using a concise mathematical model which has been validated with results from a rolling contact two disk test rig.

2. A SURVEY OF MODELLING RAILWAY DYNAMIC AND ACOUSTIC PHENOMENA

2.1 Rail Corrugations

Rail corrugation is a significant problem for the railway industry worldwide. It is a periodic irregularity that is observed to develop on the running surface of the rail and is characterised by long (100-400mm) and short (25-80mm) wavelengths (Grassie and Sato *et al.* [1,9]). The formation and growth process of rail corrugations is analogous to corrugations developing on unsealed roads. This irregularity grows in amplitude as a function of the number of wheelset passes, until typically removal by grinding is required to ameliorate the excessive noise and vibration caused by the corrugated rail. The most prolific type of corrugation is characterised by wear as the damage mechanism and is a particular concern to industry as no reliable alternative cure to regrinding affected surfaces exists. Because the problem has become so entrenched, a periodic preventative rail grinding schedule is often enforced costing the Australian industry alone in the order of AU$10 million per annum. These grinding costs will continue to grow with increases in rail traffic volume and speed unless an alternative reliable cure is developed.

The best solution for acoustic and vibrations problems is to control the source of the phenomena. Hence it is important to obtain a complete theoretical understanding of the mechanisms of corrugation growth. Over the past few decades, much international research progress has been achieved towards this. For example, the research of Hemplemann and Knothe [2], Igeland and Ilias [3] and Matsumoto *et al.* [4] amongst many others has resulted in the development of numerical models for simulating the characteristic behaviour of corrugation formation. Powerful insight has also been gained from simpler analytic models which overcome excessive computational expense with closed form solutions. These have been used to identify fundamental growth mechanisms and to investigate parametric trends that may reduce corrugation formation. Recent examples of this research include Muller [5], Nielson [6], Meehan *et al.* [7] and Bellette *et al.* [8]. In [5] and [6] analytical predictions showed that certain wavelength ranges of corrugation were promoted due to a wavelength based contact filtering effect, however the effects of dynamic wheel/rail contact forces were ignored. In [7], system stability of the interaction between structural dynamics and contact mechanics over multiple wheelset passages revealed the characteristic exponential growth of corrugations and a closed form analytic expression for this growth rate was obtained.

Grinding costs have motivated research on methods of corrugation mitigation, unfortunately with unreliable success(see reviews of [9,10]). The most renowned methods include railhead hardening or hardfacing, lubrication of the rail with friction modifiers and vibration damping of the vehicle and or track. Hardening of the rail contact surface has long been proposed as a cure as it provides resistance to wear and therefore supposedly also variations in wear. Field tests of HSH rails[11] versus conventional rails support this method although it's validity for all conditions and types of corrugations is debatable. Arguably the most promising solution for rail corrugation at present is friction modifiers. These are specialized lubricants applied to the wheel-rail contact that effectively control the coefficient of friction to approximately 0.35 [12] and lower traction and wear. Friction modifiers also produce a positive sloped traction-creep curve under full sliding conditions to eliminate stick-slip instability and squeal. In practice the lubricant has been shown to reduce the occurrence of some types of corrugations, however its robustness and practicalities of controlled use in the field are still under investigation. Tuned vibration dampers/absorbers on railway tracks and wheels have been proposed and tested to reduce wayside noise [13]. however the cost can be excessive and the effects on corrugation growth may only be minimal.

Alternatively, the present authors [8], have proposed and investigated controlling train pass speeds based on an efficient mechanics based predictive model for corrugation growth. A critical advantage of such validated models is their extremely fast computational speed, facilitating a complete investigation of the effect of parameter variations such as train pass speed on growth rate. To this end an analytical model for corrugation growth prediction developed by the authors [7,14] was utilised for insight and mechanics based analysis of the field data, taking into account the effects of environmental conditions, contact patch widening, deformation variations and vehicle pass speed variation/distribution. The conceptual block diagram for the corrugation formation model is depicted in Figure 1 (based on [2,7,14]), where a feedback process occurs over multiple wheelset passes due to the interaction of vehicle/track vibration dynamics I, contact mechanics II and a damage/wear mechanism III, initiated by an initial rail profile irregularity.

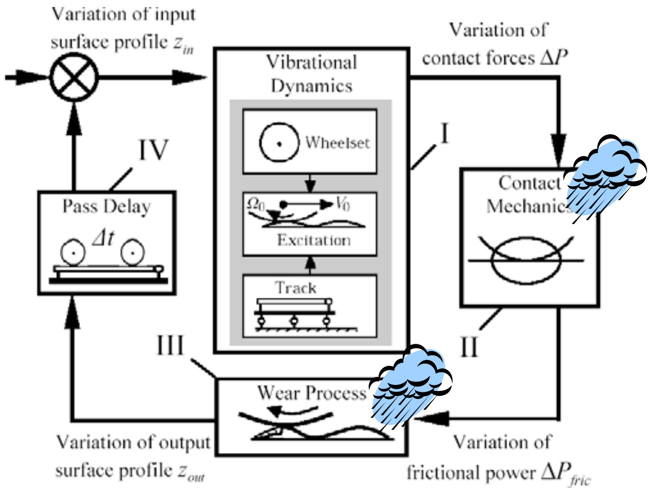


Figure 1. Feedback model for corrugation formation (based on [15]).

The model has been used for considering either vertical or lateral vibration modes, interacting with longitudinal or lateral contact forces. In such a way, both straight track [7] or cornering conditions [14] may be modelled. In Australia the model was used to simulate cornering corrugations which were shown to be dominated by vertical vibrations interacting with lateral traction, slip and wear variations based on evidence from previous modelling and field investigations [14]. The contact mechanics model is based on quasistatic microslip and considers small linear variations about nominal non-linear operating conditions using sensitivity coefficients derived from creep theory (ie

[15,16]). The wear model is based on assuming that the rate of wear is proportional to the frictional power dissipated. Due to computational efficiency, it allows for an investigation into growth rate changes due to different train pass speed distributions, for a range of initial track profiles and frictional/wear conditions over many vehicle passages. The field condition variations in contact patch geometry, rainfall, humidity primarily affects parameters associated with the wheel/rail rolling contact mechanics and wear on the rail surfaces (as indicated in figure 1). Details of the derivation of these results [14] are summarised in the following. In particular, under the assumptions that one mode of vibration of damping constant ζ_i dominates the response of the combined wheel/rail system, a very powerful algorithm for predicting the growth of corrugations can be shown to be given by,

$$\left|\frac{Z_{n+1}}{Z_n}\right| = 1 + G_{ri}, \quad G_{r_i} \approx K_b\left(1+\frac{K_{ci}}{4\zeta_i(1+\zeta_i)}\right) \tag{1}$$

This corrugation growth equation is in the form of compounding interest where G_r is the approximate exponential growth rate parameter, equivalent to an interest rate per vehicle passage. Z_n is the n^{th} profile, K_b is a grouped parameter represents the sensitivity of wear variations to lateral wheel/rail contact deflection variations K_{ci} is a grouped parameter, representing the modal sensitivity of the wheel/rail relative displacement to a change in input longitudinal profile. For realistic railway parameters, K_b, K_{ci} and ζ_i are always positive valued. It is noted that this exponential growth rate is expected only to occur when changes in environmental, vehicle passing speed, contact patch geometry and wear/deformation conditions are small. When a randomly distributed sequence of pass speeds is introduced of distribution p(x) the expected amplitude ratio of the transfer function (1) becomes,

$$\frac{Z_{n+1}}{Z_n} = exp\int_{-\infty}^{\infty} ln\left|K_b\frac{1}{1+k_c(R_r(V/\lambda)+R_w(V/\lambda))}+1\right|p(x)dx, \tag{2}$$

where multiple wheel passes at varying speeds are considered by varying the frequency as a ratio of speed V to fixed wavelength, λ. The model developed in [8] was first tuned and validated to field measurements of corrugations occurring on a suburban track in cornering taking into account non-constant train pass speeds over multiple passages. Predictions of the effects of altering the existing field measured pass speed distribution on corrugation growth rate and corresponding grinding cost predictions are provided in Figure 2.

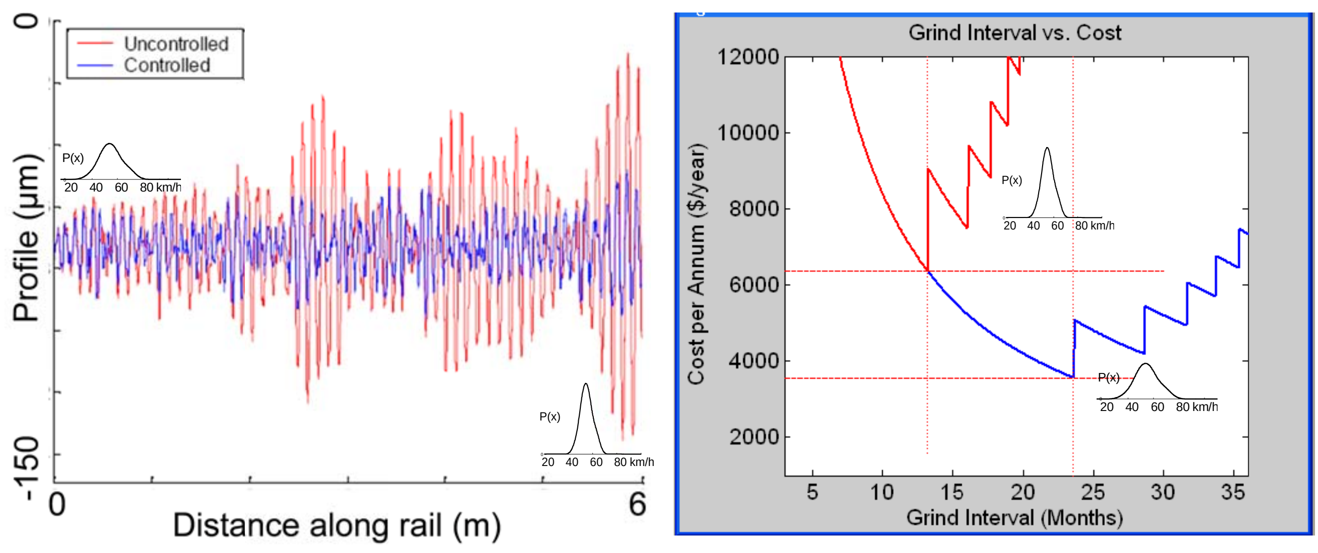


Figure 2. Results of predictive modelling for corrugation growth showing the effects of controlling successive train pass speed distribution on a) corrugation profile growth, b) grinding cost.

The results highlight a substantial reduction in corrugation growth, purely by altering the spread of train speeds at the affected site resulting in a reduction in grinding costs of up to 45%. Note the average speed has been kept the same and the lowest speeds would only affect timetables by less than 20s. In such a way, efficient, validated mechanics based modelling has been used to gain insight and test a proposed new control method for corrugation growth before laboratory and field trials presented in [17].

2.2 Railway Rolling Noise and its Growth

Railway rolling noise results from the rolling contact between the wheel and rail over surface roughness that induces vibration of both the vehicle and the track [18, 19]. Reducing railway noise is necessary to enable railway transportation increases in usage and speed. Noise barriers are widely used along railway lines, however, these are expensive and visually intrusive. Therefore, it is preferable to control the noise at its source where possible. The development of theoretical models for the prediction of dominant noise sources and how the noise is generated is essential for insight into better control methods. Although various models are available for predicting railway rolling noise eg [20,21], they are mostly empirical and therefore limited. However the use of predictive models based on the theory of the underlying mechanism of generation has been obtained. Remington [22] produced the first theoretical model of rolling noise, which formed the basis for further development and extension by Thompson [23]. Subsequent research funded by the European Rail Research Institute (ERRI) resulted in the implementation of the prediction model in a computer program, Track-Wheel Interaction Noise Software (TWINS) [24]. Full-scale validation experiments have shown that this software is capable of predicting noise emission accurately enough from a variety of wheel and rail combinations under European, Japanese and recently Australian conditions. Over the past few decades, research has been undertaken to understand the physics behind railway noise induced by roughness growth. Kaper [25] discussed wheel corrugation, its origins and effects on noise emissions. Thompson [19] and several other researchers ie [26] have found that rail corrugation can clearly increase railway noise by 10–15 dB. Although much work has been performed on roughness growth prediction and railway noise prediction, prior to the authors research, no attempt had been made to integrate these models for noise growth prediction.

The present authors recently integrated mechanics based models based on TWINS [23,24] to enable noise growth predictions based on rail roughness growth mechanics. The Railway Rolling Noise Prediction Software (RRNPS), has been shown to provide reliable predictions for typical Australian and European conditions [27]. The framework of this modelling and some pertinent results are shown below.

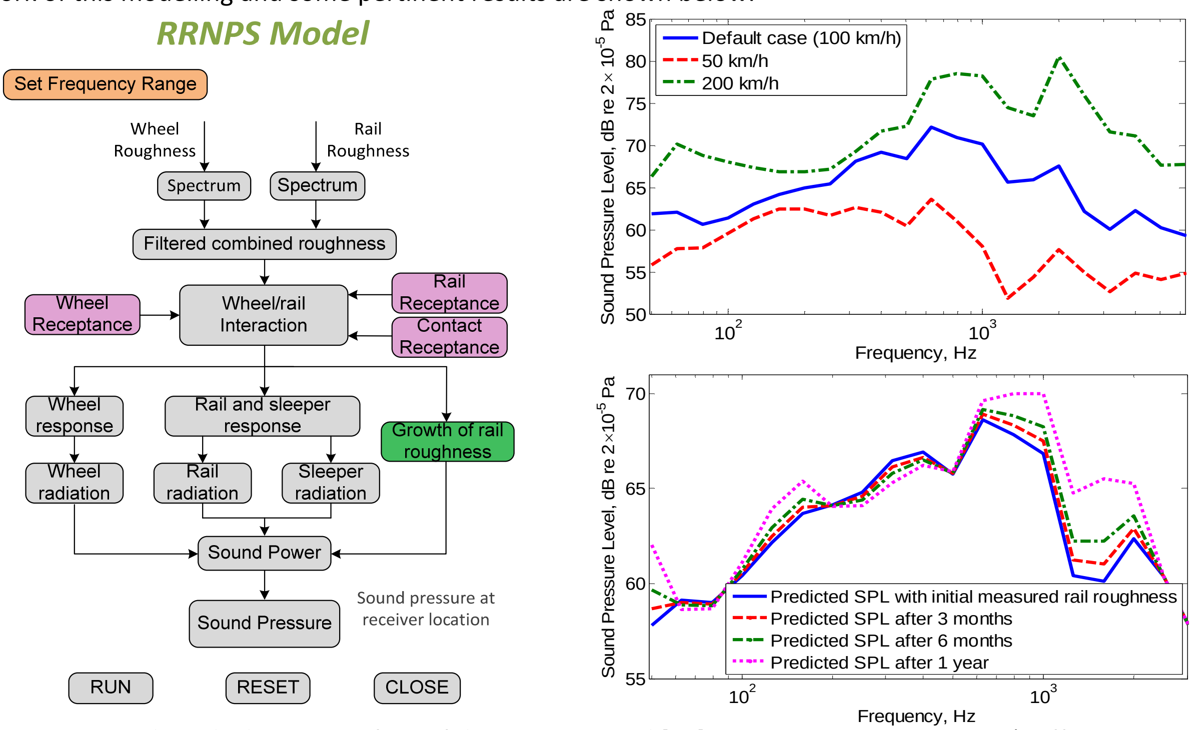


Figure 3. The calculation interface of the RRNPS model [24]. Predicted SPL spectra at a) different speeds. b) after roughness growth of various times

The results in Figure 3 highlight the effects of changes in traffic speed and roughness growth over time on sound pressure levels. It can be shown that the increased SPL is directly determined by the combined wheel/rail roughness levels growth. Due to the underlying mechanics modelling, sensitive areas in the frequency spectrum (ie 1 to 2 kHz) can be related to the underlying dynamics of the vehicle/structure to highlight possible areas for mitigation. In such a way, the RRNPS can be utilized as a predictive tool to investigate new strategies to control and mitigate railway rolling noise growth due to corrugation and rail roughness growth.

2.3 Wheel Squeal

Wheel squeal is an annoying high pitched tonal noise that can occur as a train negotiates a curve (corner) of a railway line. This phenomenon has plagued the railway industry for many years and continues to rise in importance as railway usage increases and subjective human noise tolerance decreases. Wheel squeal occurs due to top of rail lateral sliding as opposed to the broader spectrum “flanging noise” associated with wheel flange contact with the rail. Although much research insight has been obtained into the mechanisms of squeal over the past decade, the occurrences and amplitude of wheel squeal still appear unpredictable as it is sensitive to a wide range of parameters that naturally vary in the field. Much modelling of wheel squeal has been performed particularly following the renowned works of Rudd [28] and review by Remington [29] and Thomson et al [30] in which the fundamental mechanism due to lateral creepage was consolidated. Curve squeal is generally thought to originate from the unstable vibratory response of a railway wheel when subject to large creep forces whilst negotiating corners. The conventional mechanism from the literature is that the unstable excitation of the squealing wheel originates from a lateral 'stick-slip' mechanism in the contact region analogous to the bowing of a violin string (see figure 4 c). In particular, when a bogie negotiates a curve of a track, there is a misalignment between the rolling velocity and the wheel velocity, namely angle of attack, leading to a crabbing velocity, i.e., lateral sliding velocity, of the wheel across the top of rail. The squeal mechanism depends on the behavior of lateral traction ratio (lateral tangential force Q divided by the normal force N) and lateral creepage (approximately equal to the angle of attack) conditions during the excitation of a railway wheel. The friction coefficient and shape and slope of the traction/creepage curve (see Figure 4 c) is effected by the so-called third body of the contact; an interfacial layer consisting of any lubricants, contaminants and material generated as a result of the contact interaction [31]. If the crabbing velocity (or angle of attack) is large enough its oscillations will be centered in the full sliding region (marked negative damping in Figure 4c). The negative slope in the creep curve can be shown to be associated with negative damping of creep oscillations and hence squeal instability. This leads to self-excited "stick-slip" oscillations, which in turn excite wheel vibrations and radiated sound. It is noted that, conversely, some recent research contends that a modal coupling phenomena between the normal and tangential dynamics may cause the instability eg. [32]. Pure tone components of squeal, are generally related to wheel natural frequencies that correspond to the out-of-plane wheel bending (or axial) modes.

Much research on the modelling of squeal has been performed in the past with differences in modelling details of wheel/rail mechanical impedances, vertical dynamics [4, 14], contact forces and wheel sound radiation (eg [33], [34], [35]. Some have also included wheel/rail roughness or wheel rotation effects [33]. Recently, a transient analysis of the lateral creepage of the wheel was performed to account for nonlinearities of friction forces and resultant excited wheel modes appeared to match field observations better [36]. Notably, a time domain model was presented by Heckl and Abrahams [33]. This paper concluded that curve squeal is an unstable wheel oscillation that grows to a limit cycle oscillation, whose velocity amplitude is equal or very close to the crabbing velocity. Furthermore, the simulation results of Chiello et al. [37] also showed that the vibration velocity stabilises below the lateral sliding velocity. This was investigated further by the present authors in [38] in which a numerical power balance analysis was identified as important, however an analytical prediction and explanation was not achieved. Laboratory experimentation using twin disk and bogie testrigs have also been used for investigation under controlled conditions and validation of model predicted conditions [34],[39] and [40]. In Monk-Steel et al. [39] the inclusion of longitudinal creep was shown to increase in the threshold of lateral creepage necessary for squeal by changing the slope of the friction curve. In Koch et al [40], a relation between noise level, rolling speed and angle of attack was confirmed experimentally and the average friction coefficient as a function of lateral creep was measured/inferred in dry and wet conditions. In [12] friction modifiers were shown to cause substantial (~12 dB) noise reductions associated with squeal and flanging but had varying results in [41].

The model used for the present analytical investigation has already been described in [42,43] and is reintroduced here for convenience. Wheel squeal may be concisely modelled based on the conceptual diagram of Figure 3. An important parameter, the crabbing velocity arises due to the misalignment between the wheel velocity and rolling velocity as shown in Figure 4(a). Hence contact lateral force Q arises as well according to the rolling frictional behaviour. In the lateral direction, the dominant vibration of the wheel is analogous to the self-excited friction oscillation of mass connected to a spring and damping on a rolling belt as demonstrated in Figure 4(b).

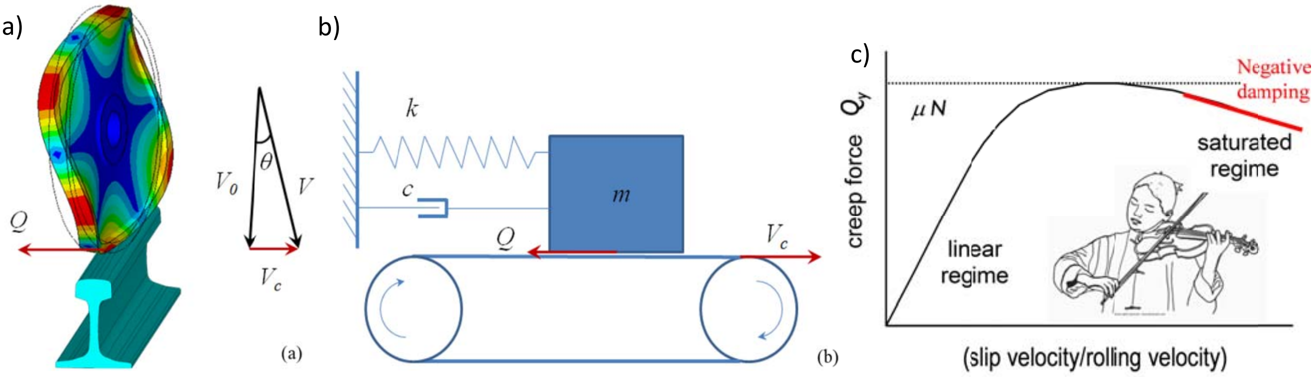


Figure 4 (a) 3D model of a squealing wheel rolling on top of rail, (b) analogous SDOF squeal mode self-excited by friction oscillations in lateral direction associated with c) the negative damping region of the lateral creep curve.

The wheel squeal dominant vibration mode can be described by an effective spring-mass-damper system as,

$$m\ddot{y}(t)+c\dot{y}(t)+ky(t)=Q(\zeta)=-\mu(\zeta)W \tag{3}$$

where m is the modal mass, c is the modal damping coefficient, k is the modal stiffness which also account intrinsically for the modal participation from the lateral force Q. The modal parameters of the dominant mode were curve fitted from a receptance spectrum of a modal test [24] and are listed in Figure 4. In order to obtain an efficient solution, a simplification of the creep behaviour described in (3) is used. In particular, it is assumed that the full creep behaviour can be characterised by linear slopes, k_1 and k_2, for the slip and full sliding regions, respectively, as,

$$\mu(\zeta)=\begin{cases} k_1\zeta' & for\ \zeta'\le 1 \\ k_1+k_2\{\zeta'-1\} & for\ \zeta'>1 \end{cases} \quad \text{where} \quad \zeta'=\frac{\zeta GabC_{22}}{\mu_0 W}=\frac{\left(V_c+\dot{y}(t)\right)k_3}{V_0 W} \tag{4}$$

where ζ' is a normalised version of lateral creepage ζ that is the ratio of the relative velocity between the wheel and rail and the rolling speed V_0, $\dot{y}(t)$ is the vibration velocity of the wheel and V_c is crabbing velocity. The lateral crabbing velocity between two wheels can be calculated with angle of attack θ and rolling speed V_0, i.e., $V_c = V\sin\theta$. These simplified results may be used to capture the fundamental mechanism of squeal. In particular, solving equations (3,4) for small oscillations about a nominal crabbing velocity V_c yields,

$$m\ddot{y}(t)+\left(c+k_{1,2}k_3/V_0\right)\dot{y}(t)+ky(t)=0 \tag{5}$$

where $k_{1,2}$ represents the local slope of the creep curve under nominal conditions. Equation (5) is in the form of the well known second order vibrating system that has its stability determined by the sign of the effective system damping term $c + k_{1,2}k_3/V_0$. A negative system damping term indicates instability and system energy input to vibrations causing them to grow to a steady nonlinear squeal amplitude. A closed form analysis and prediction of the nonlinear squeal amplitudes is described subsequently.

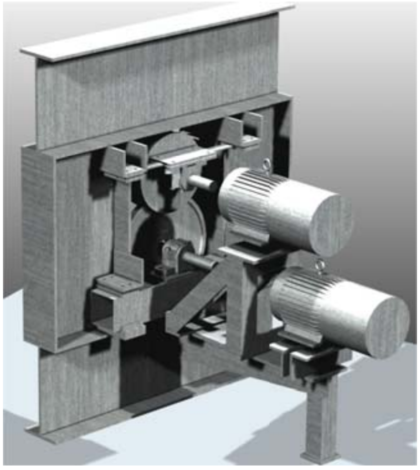
3. RECENT RESULTS FOR WHEEL SQUEAL NOISE PREDICTION

Despite considerable research effort, there has been uncertainty in fully understanding and theoretically predicting squeal vibration amplitude and trends of how squeal noise varies with important parameters such as crabbing velocity (angle of attack). In particular, models have in general involved too much complexity to be able to efficiently predict squeal amplitudes and perform detailed investigation of the effects of critical parameters on squeal sound levels. Recent results for the prediction of vibration amplitude and sound pressure level of wheel squeal noise using a concise mathematical model validated with results from a rolling contact two disk test rig are provided in the following.

3.1 Wheel squeal experimental methods

The experimental results have been obtained previously in [43] using a rolling contact two disc test rig developed for the investigation of squeal noise (see figure 5). The lateral force between the upper and lower wheel is measured with strain gauges. The angle of attack between the upper and lower wheel can be adjusted and measured as described in [38]. The test rig was run at various crabbing velocities (angles of attack) and the sound was recorded with a microphone placed 5 cm away from the lower wheel and 80 cm above the ground.

a)



b)

Description	Value
Radii of longitudinal and tangential curvature for the lower wheel (R_1, R_{1t})	0.213 m, 0.300 m
Radii of longitudinal and tangential curvature for the upper wheel (R_2, R_{2t})	0.085 m, 0.040 m
Young's modulus of upper and lower wheel (E)	175 GPa
Density (ρ)	7800 kg/m^3
Poisson's ratio (v)	0.28
Angle of attack range	0 ~ 26 mrad
Normal loading (W)	1000 N
Modal mass (m)	3.1 kg
Modal damping (c)	42 Ns/m
Modal stiffness (k)	1.6E8 N/m

Figure 5 Rolling contact two disc test rig used for the investigation of squeal noise (a) 3D view of the test rig, (b) Parameters of the test rig

The vibration characteristics of the test rig were determined using modal analysis of impact hammer tests and analysed using the finite element method. The vibration characteristics of the lower wheel acquired from finite element analysis and model tests correlate well with the results of sound recordings [42].

3.2 Wheel squeal amplitude analysis

The equations of motion may be solved for limit cycle analysis to analytically determine the amplitude of wheel squeal under simplified assumptions. This provides an almost instantaneous solution that may be used to investigate a wide range of conditions and how the vibration amplitude compares with the crabbing velocity analytically. It is assumed wheel squeal is characterised by an initial growth in vibration until a steady state amplitude A is reached as,

$$y(t) = A\sin(\omega t) \tag{6}$$

The nonlinear limit cycle amplitude of the squeal associated with the noise is governed by an energy balance process based on the effective positive or negative damping characteristic of the slopes of the creep curve. In particular, for small vibrations, in the fully sliding region of the creep curve, the negative slope of k_2 causes a positive power input to the squeal vibration which causes growth if it is greater than the power output due to damping c. The vibration amplitude grows to larger creep oscillations that start to impinge on the positive slip side of the creep curve. This positive slope k_1 of the creep cycle causes a negative power input (or power output) to the vibration which acts like additional damping to c and causes them to decay. The vibration will continue to grow until a balance is achieved between the power input and output over the different slopes of the creep curve so that a steady state (limit cycle) amplitude is reached. This vibration amplitude $A\omega$ may be obtained by solving equations (3-6) to obtain

$$A\omega = \left(\zeta_c V_0 - V_c\right)/\cos(\omega t_c) \quad \text{where} \quad \sin(2\omega t_c) - 2\omega t_c = 2\pi\left(V_0 c_d - k_2 k_3\right)/\left(k_3\left(k_1 - k_2\right)\right) \tag{7}$$

where ζ_c is the critical creep (between the positive and negative sloped sides of the creep curve). Equation (7) can be easily solved for the critical value of ωt_c using any root finding function and hence the steady state squeal vibration velocity amplitude $A\omega$. Hence solution for the squeal amplitude via equations (7) is almost instantaneous and allows a wide range of parameter investigations to be carried out as shown in the following.

3.3 Wheel squeal amplitude results

The analytical results described by solution (7) to the numerical solution of the exact equations of motion is provided in Figure 5. The testrig parameters chosen are described in Figure 5 b) as well as crabbing velocity 0.39m/s and testrig rolling speed 800 rpm (17.8 m/s).

b)

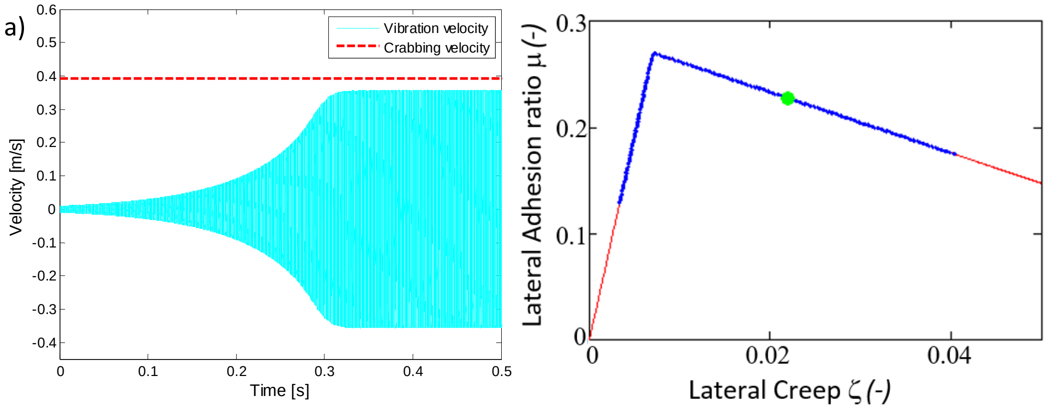


Figure 6 a) The simulated vibration velocity at a crabbing velocity of 0.39m/s b) the corresponding approximated creep curve (-), analytical solution for creep oscillations range (-) and quasistatic lateral creepage (•)

Figure 6 a) shows the simulated steady state vibration velocity settling at approximately 0.35m/s, a value close to but under the crabbing velocity (0.39 m/s). This compares well with the analytical solution of 0.33m/s (equation (7)) using the approximate creep curve described in Figure 4 with k_1 = 0.27 and k_2=-0.02. Figure 6b) shows the creep oscillation range and highlights that the steady state squeal amplitude is achieved via creep oscillations spanning both the positive and negative sloped regions of the curve to enable a balance of energy. It was of interest to investigate the effect of crabbing velocity on SPL. In this case the sound power is proportional to the vibration velocity squared [18]. The analytical predictions based on solution to equations (7) were plotted with real experimental recordings on the two disk testrig under the same squeal conditions.

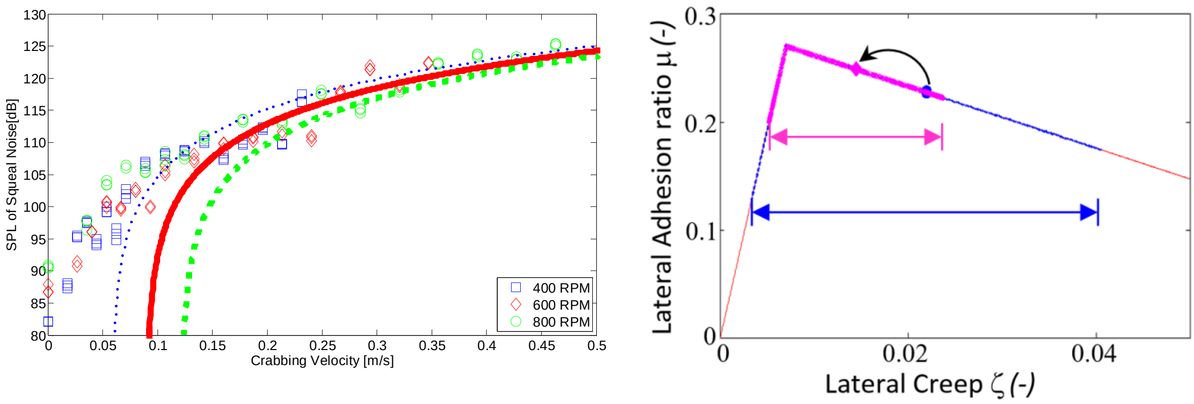


Figure 7 a) Squeal noise amplitude versus crabbing velocity; experimental and analytical prediction b) The approximated creep curve (-), analytical solution for creep oscillations range for nominal conditions (-) for crabbing velocity (▯) and quasistatic lateral creepage for nominal conditions (•) for crabbing velocity reduced by 34% (•)

The results in Figure 7 a) show that the experimental trend of sound pressure level of squeal noise increasing with crabbing velocity is well predicted by the analytical solution for crabbing velocities greater than 0.15 m/s. This is consistent for squeal conditions. Below 0.15 m/s the trend is not well matched because the experimental noise level is no longer dominated by squeal ie other sources such as rolling noise is largely contributing which is not modelled by the analytical solution. The squeal vibration amplitude is strongly dependent upon crabbing velocity (or angle of attack) as is illustrated on the creep curve of Figure 7 b). In this case only a 34% reduction in crabbing velocity is required to achieve a 50% reduction in squeal vibration amplitude. This is due to the crabbing velocity moving closer to the critical slip velocity and lowering the amplitude of creep oscillations required to obtain the positive damping effect of the positive sloped slip region to stabilize the vibrations. The results of Figure 7 highlights how a simplified but validated mechanics based model of squeal can be used to reveal new insight into how squeal noise amplitude varies with angle of attack and why this occurs.

4. CONCLUSIONS

Recent results in the modelling, prediction and control of rail corrugations, roughness generated rolling noise and wheel squeal has been surveyed. The general approach to investigating these railway dynamic and acoustic phenomena has been to obtain a detailed understanding via modelling of the important physical mechanisms. Validation with field and/or laboratory measurements enables the models to provide efficient predictive insight into the critical conditions involved. The models can then be used as a testbed to develop and test mitigation/control techniques. The importance of mechanics based modelling has been highlighted in these recent advances.

Mechanics based theoretical modelling for rail corrugation has been developed and validated to enable efficient prediction of its growth under a wide range of conditions. These have then been used to investigate a novel means of corrugation control via changing the distribution of pass speeds at an affected site. The results show that a substantial reduction in corrugation growth rate of over 50% may be achieved, such that the time interval between grinds is expected to increase by a factor of 1.7. Testrig and field testing has subsequently confirmed these results.

Similarly railway rolling noise predictions under a wider range of railway conditions have been achieved by development of mechanics based models for the radiation of sound based on vibrations excited from rail roughness. The RRNPS simulation has been developed based on TWINS and validated at a typical site in Australia to within 2 dB(A) accuracy. Railway rolling noise levels increase with increasing train speed, a doubling of speed leading to an increase in overall noise levels of up to 10 dB(A). The model has then been extended to include prediction of noise growth with roughness growth over time. The system provides a very powerful tool for investigating methods of railway rolling noise mitigation and has recently been used for the investigation of friction modifiers on railway noise.

Finally, a mechanics based model integrating the lateral rolling contact mechanics with the modal vibration of the wheel is used to theoretically predict the vibration velocity amplitude of wheel squeal. The limit cycle analysis results show that the vibration velocity amplitude reaches a steady state approaching the crabbing velocity under reasonable conditions in agreement with previous research. The efficient theoretical model is then used to predict the effect of crabbing velocity (or angle of attack) on wheel squeal noise amplitude and the results compare very favourably with the trend found experimentally. The results highlight the primary importance of crabbing velocity (or angle of attack) as well as the creep curve parameters that may be controlled using third body control (ie friction modifiers). The results concur with experimental and field observations but also provide important theoretical insight into the useful mechanisms of controlling wheel squeal noise and quantifies their relative merits.

These results highlight the power of mechanics based modelling to achieve effective source based control in railway dynamic and acoustic phenomena.

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