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The Application of Dither for Suppressing Curve Squeal

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Abstract
Curve squeal is a highly disturbing tonal sound generated by vehicles like railways, metros or trams, when negotiating a sharp curve. The probability that squeal occurs increases with reduced curve radius of the track. Curve squeal noise is attributed to self-excited vibrations caused by stick/slip behaviour due to lateral creepage of the wheel tyre on the top of the rail. With respect to the enormous number of the rolling stock units and the long lifetime of waggons there is an urgent need for a cheap and simple retrofitting measure to reduce curve squeal. The main objective of the paper is therefore to investigate the potential to reduce curve squeal by means of active control in the form of dither in an efficient and robust way. Dither control has been applied in the field of mechanical engineering for systems including non-linear components. There it has been shown to suppress self-excited oscillations very efficiently. The control is an open loop control. It consists in adding a forced vibration to the vibrational system. The demand on this additional signal is that it is higher in frequency than the friction-induced response. From a physical point of view, dither control modifies the effective friction characteristic.
Keywords: Curve squeal, dither, self-excited vibrations, noise control

1 BACKGROUND

2 Background

Curve squeal is a highly disturbing tonal sound generated by vehicles such as railways, metros or trams, when negotiating a sharp curve. The probability that squeal occurs increases with reduced curve radius of the track. For curves with a radius of 200 m and below, curve squeal is common. At the same time such tight curves occur mainly in urban areas where many people live close to the tracks. Therefore curve squeal can contribute to the negative health impact of traffic noise. Curve squeal is also a comfort issue for passengers. In [1] it is stated that about 7 percent of railway customers are highly disturbed by curve squeal noise. Curve squeal noise is attributed to self-excited vibrations caused by stick/slip behaviour due to lateral creepage of the wheel tyre on the top of the rail [2]. However, the actual mechanism of the instability is still a controversial topic. The falling friction curve has been identified by as series of reports (see e.g. [3] -[5] as well as the coupling between normal and tangential dynamics [6]-[8]. For the reduction of curve squeal noise mainly three measures are available

- There have been several attempts made to optimise the design of wheel and rail towards less curve squeal. Most successful seems to be the asymmetric rail profile [9] moving the contact position on the inner wheel outside the so-called “squeal zone”. There are, however, no implementations of improved design on the market.
- Lubrication of the rail with grease, water or specially developed friction modifiers has been shown to be efficient in some cases [10], in other cases it failed [11]. By reducing the friction coefficient curve squeal might be eliminated. Problems could be insufficient traction forces or the contamination of the soil close to the track or when using pure water the need to avoid freezing.
- Wheel dampers showed in some situations good results. By adding constrained damping layers [12] or so-called ring dampers [13] substantial reduction could be achieved if the damped modes were involved in

the curve squeal. In this context, the amount of reduction is not proportional to the increase in damping. Already little additional damping can suppress the self-excitation of stick-slip motions and therefore give high reduction.

None of the measures is really offering a sustainable and reliable solution to the problem of curve squeal. While new design does not solve the problems with existing infrastructure, wheel damping and lubrication have shown to work in certain situations, but to fail in others. With respect to the enormous number of the rolling stock units and the long lifetime of waggons there is obviously an urgent need for a cheap and simple retrofitting measure to reduce curve squeal.

3 ACTIVE CONTROL OF SELF-EXCITED VIBRATIONS AND DITHER

Typical examples of self-excited vibrations are flutter, Rijke tube, combustion instabilities, or friction oscillators. In a self-excited system, non-oscillatory energy is transformed to oscillatory energy due to an often non-linear feedback process between an oscillator of the system and the non-oscillatory energy source. Active control has been shown to be very suitable to disturb this feedback process and suppress the build-up process of the self-excited vibrations. Maria Heckl was probably the first who successfully has used this approach for the control of the noise generation from the Rijke tube [14]. Other examples concern instable combustion [15], machine chatter using active bearings [16] or torsional vibrations in a drillstring [17]. Maria Heckl also used an active control approach for the control of curve squeal noise [18]-[20]. With a pre-calculated control law she fed back wheel vibrations to force transducers either on wheel or rail. She also demonstrated the functioning of the approach at a very simplified experimental set-up. In "standard" active control a negative copy of the original (vibration) field is created by active sources and added to the field to be reduced. This approach demands high accuracy in amplitude and phase for the added field, while for destroying the feedback process by active means the control laws are more relaxed. Both amplitude and phase have just to be in a certain range. This is the strength of the application of active control in the case of self-excited vibrations. In this aspect, dither goes even a step further. It has been widely used in different areas such as audio recording techniques, optics, image processing, and control technology. The expression refers to the application of signals/information to a system in order to modify its characteristics. A specific example is the use of dither to compensate for the negative consequence of quantisation errors. By adding shaped noise to the audio signals before the analog/digital converting process the audio quality is improved substantially. Dither control also has been applied in the field of mechanical engineering. Especially for systems including non-linear components dither can be used to suppress self-excited oscillations. Morgül [21] showed that in the case of friction oscillators, dither control could be used to prevent the build-up of stick/slip motion. He investigated the control of chaotic systems with dither and showed that there are no restrictions with respect to frequency content or time behavior of the signals others than that the frequency content has to be higher than the friction-induced response. In addition, dither control is an open-loop control. This means the signal is just superposed to the existing field without any feedback. Therefore its implementation is simple and does not require an as complex control environment as traditional active control approaches do. Up to now dither control has not been used for the control of curve squeal as far as the authors can conclude from literature. Dither control of self-excited vibrations has mainly been investigated for generic cases such as simple oscillators with friction elements (see e.g. [22] and [23]). There it was shown that dither control could be used to modify the effective friction characteristics. It also has been shown that the underlying friction characteristic is essential for the functioning of dither [23]. Dither control does not always suppress the self-excited oscillation, but can also lead to instabilities. Besides for simple oscilators, dither control has also been used for the control of brake disk squeal in automotive applications. Cunefare [25] and later Bardetscher et al. [26] showed for disc brake rotor squeal that "normal dither" could be used to reduce and even suppress the generation of squeal. The performance of the brakes was only marginally reduced. "Normal dither" in this context means that the dither control changed the normal force between brake pad and brake disk. An alternative would have been "tangential dither". In [23] a harmonic signal is used as dither with a frequency above the eigenfrequencies of the squeal modes under consideration. Even frequencies outside the audible range (i.e. above about 20 kHz) were shown to work in this application. In this case, the dither signal is not audible. Therefore the control is not adding a disturbing tonal signal. Dither seems to offer substantial potential for squeal noise reduction, although the validity of this statement is limited to cases investigated (see also [27]). In the case of friction, one could interpret dither as a controlled modification of the friction curve due to additional external forces. However, there is no general rule for selecting the signal form and amplitude, although e.g. in [28] it was shown that periodic square signals are much more effective than sinusoidal signals.

4 THE MODELLING OF CURVE SQUEAL AND DITHER

To model the intrinsically non-linear and transient phenomenon of curve squeal is an enormous challenge, just considering the complexity of the involved structures, the need to include non-linear transient rolling contact and the fact that the required frequency range easily exceeds 10-15 kHz. While frequency models can be used to investigate which modes are prone to squeal, amplitudes of squeal can only predicted by time domain models, a fact, which increases the computational effort substantially. Consequently many of the existing time domain models rely on simplifications of wheel, rail or contact formulation. Typically rail dynamic is neglected. The consequences of this simplification are not clear. The few who include rail dynamics (e.g. [29]-[31]) do not come to an agreement on this issue. However, we know from practice that changes in the track design can increase or decrease the problems with curve squeal [32]. Most models describe the relation between creep force and creep in an analytical form, which only partly can represent the non-linear process in the contact zone. However, even more advanced models can be questioned. Périard [29] for instance included a modified version of Kalkers steady-state contact model FASTSIM [33] in his squeal model. The consequences of using such simplified contact models are not clear.

4.1 The curve squeal model at Chalmers, Applied Acoustics

In the following an approach developed by Pieringer [34] is used. She combined pre-calculated impulse response functions for rail and wheel with the model by Kalker for transient rolling contact [35]. Rail responses are calculated with Wave Guide Finite Elements while for the wheel a standard Finite Element Model is used. In this way, the complete dynamics of wheel and rail in the required frequency range is considered in combination with a fully three-dimensional transient and non-linear contact model. The computational costs are acceptable in contrast to the computational costs of an equivalent full Finite Element Model for curve squeal. Zenzerovic then developed an "engineering" version of the model by Pieringer. The procedure for calculating the contact forces by time-stepping is described in [36, 37]. Three components are pre-calculated and act as inputs to the calculation.

- The dynamic wheel and rail responses at the contact position are given as moving Green's functions.
- The geometrical information about the discretised contact patches of the contacting bodies is the second input. It includes elastic half-space influence coefficients, which describe the elastic half-space deformation field on the contact elements caused by the tangential tractions.
- Finally, a regularised friction curve for the global, one-point contact is pre-calculated for each set of contact parameters, assuming on Coulomb friction in each contact element. The contact parameters include the vertical force of interest, contact position, the Coulomb friction coefficient and the rolling speed.

For each time-step, the wheel and rail response are convolved to provide the displacements of both bodies as an input to the contact model. The normal contact problem is solved first, providing the normal load as an input to the tangential contact. The tangential problem is solved subsequently, and the vertical and lateral contact forces serve as inputs to the next convolution step of the dynamic wheel and rail response. To include dither is straight forward in this modelling approach as it means just adding an additional force which is varying over time in a prescribed way.

4.2 Test setup

For the test of the dither approach the model was adapted to a squeal rig recently built at Chalmers University of Technology by two of the authors [38]. The final goal is to validate the dither approach at this test rig experimentally. The rig consists of two wheels pressed with a defined preload against each other. The upper wheel represents the rail while the lower wheel represents the wheel. The lower wheel is driven by an electric motor. Different contact positions and lateral creepage can be chosen by altering the setup. For both wheels FE models where created to obtain the wheel receptances in vertical and lateral directions as well as for the coupled direction. The calculated receptances where compared with measured receptances for the freely suspended wheel as well as for wheels mounted in the rigg. For the simulations presented here the following data are used

- Rolling speed: 50 km/h
- Load: 20 kN
- Friction coefficient for the Coulomb friction: 0.55
- 3 degree angle between upper and lower wheel

For the realisation of the dither an additional lateral force is added at the contact point. This is of course not realistic for an experimental implementation, but it is the easiest way to implement dither forces in the model.

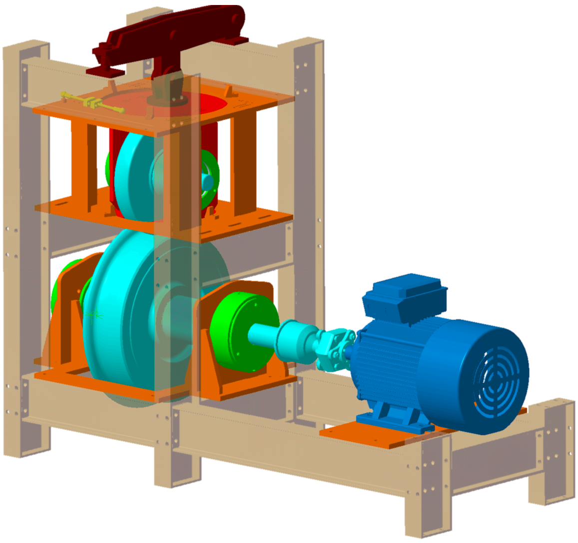

Figure 1. Squeal rig at Applied Acoustics designed for the validation of the simulation tool for curve squeal and the approach for dither. Black error indicates the case which is discussed in the following

5 RESULTS

As a very first step the simulations are run without dither as reference for a case where clear instability occurred. For the parameters described int the previous section the mode involved in the instability has its eigenfrequency slightly above 2000 Hz. In a second step the additional sinusoidal lateral force (the dither force) is varied in amplitude and frequency. The only condition for the frequency was that it had to be higher than the frequency involved in the instability, i.e. in this case above 2000 Hz. For each case the results with dither are compared with the results without dither with respect to the displacement of wheel and rail at the contact

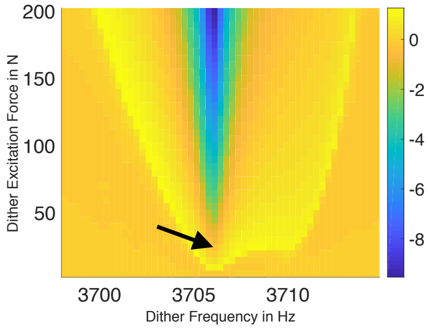

Figure 2. Mapping of the insertion loss in dB due to the application of a dither force

point. Figure 2 shows the results for frequencies around 3706 Hz where also an eigenfrequency of the wheel is situated.

The amplitude represented by different colours could be interpreted as the insertion loss with respect to the total displacement. However the interpretation is not so simple as it covers the whole frequency range inclusively the DC the part which dominates the resulting values. Closer investigation shows, that despite this short-coming the figure gives information when the dither will lead to a clear reduction. These are the values inside the light yellow contour. The black arrow indicates a very fortunate case as for a relatives small dither force a rather high reduction of the instability can be achieved. The resulting spectra for this case in comparison with the case without dither is shown in on the left side of Figure 3. The dynamic shown in the Figure is rather big. Therefore the two main frequency components are shown in an additional figure on the right. The displacement without dither are shown as dashed curves, the displacements with dither as solid curves. The

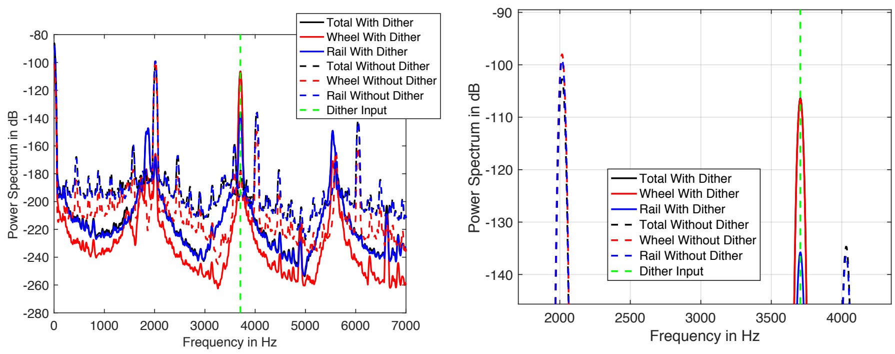

Figure 3. Spectrum of the displacements for the cases with and without dither (left). Zoom in for the two most prominent frequencies, the dither frequency at 3706 Hz and the frequency at which instability occurred at 2004 Hz (right)

dither suppresses completely the instability but of course a tonal response is created at the dither frequency. This tonal displacement however is about 10 dB below the displacement at the instability without dither. Increasing the dither force at about 3706 Hz will always stop the instability, however with increasing force the response of the structure to the dither increases as well. Therefore it might be important to keep the force as small as possible. In the example shown here the dither force was always acting from the very beginning of the simulations. However even for a delayed start of the dither, the dither force still manages to stop the instability as shown in Figure 4 on the right hand side.

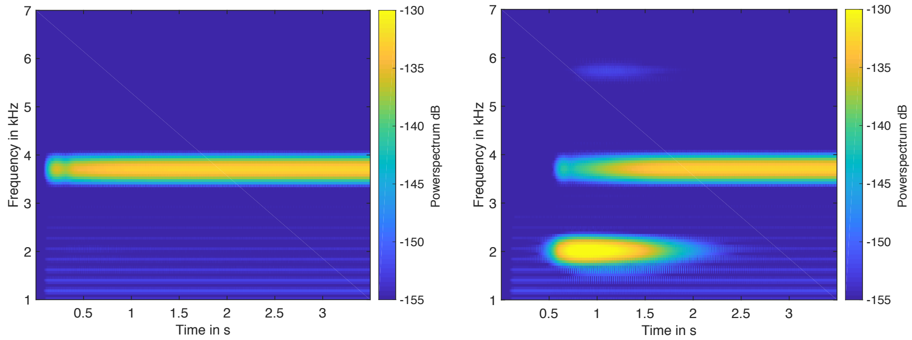

Figure 4. Spectogram of the total displacement when the dither force starts immediately (left) and after 0.5 seconds (right). The dither frequency is 3606 Hz, the instability occurs at 2008 Hz

6 CONCLUSIONS

The paper showed that the application of dither is a possible tool to abate curve squeal. The instability due to time varying friction between rail and wheel is suppressed when a dither force is applied with a frequency above the frequency of the occurring instability. However this demands a sufficient amplitude of the vibrations due to the dither force. As described in the previous text the particularity with self-exited vibration is that non-oscillatory energy is transformed into oscillatory energy due to an non-linear feedback process between an oscillator of the system and the non-oscillatory energy source. Is this feedback process disturbed or even destroyed, the flow of energy from the non-oscillatory source is not able to maintain the vibrations. When adding a sufficiently high dither force, the velocity at the dither frequency takes over the control over the friction between rail and wheel. The friction forces occurring in this case, however do not built up a instability but are only a reaction force due to the oscillatory movement of the wheel.

In order to create sufficiently high velocities on the wheel to take over the friction process it is of advantage to make use of eigenfrequencies of the wheel. By exciting at or close to resonance the "control effort" is kept to a minimum. Forces as used in the previous example were in the order of 10 N and it should not be a problem to implement those forces experimentally. It also has been shown that it is sufficient that dither is applied first when squeal occurs as an open loop response. As a final step an experimental validation is needed.

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