inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 1.3

CURVE SQUEAL OF RAILBOUND VEHICLES (PART 3): MEASUREMENT TECHNIQUES FOR WHEEL/RAIL CONTACT VELOCITIES AND FORCES AT SQUEAL NOISE FREQUENCIES

M. Janssens, W.-J. Van Vliet, P.-P. Kooijman, F. De Beer

TNO-Institute of Applied Physics, P.O. Box 155, 2600 AD, Delft, Netherlands

Tel.: +31-15-2692419 / Fax: +31-15-2692111 / Email: janssens@tpd.tno.nl

Keywords:
CURVE SQUEAL, RAILWAY, FRICTION, STICK-SLIP

ABSTRACT
A laboratory test rig (scale 1:3) is used to validate a frequency domain model for squeal noise of rail-bound vehicles. In order to check the different parts of the model various quantities are determined during the experiments: dynamic lateral wheel response at the contact point, radial and lateral dynamic contact forces at squeal noise frequencies and harmonics. The response of the contact point of wheel and rail-wheel during running tests is observed continuously in the test facility using magnetic vibration transducers. The dynamic interaction forces in the contact point are determined by combining measured results of the response near the contact positions at the rail-wheel and a measured transfer matrix of the (rotating) rail-wheel. Comparisons of measured and calculated results confirm the validity of the model.

1 - INTRODUCTION
Curve squeal is the intense tonal noise that can occur when a rail-bound vehicle traverses a curve or switch. During the curve passage, the necessary lateral creepage can show unstable stick-slip behaviour, causing the wheel to oscillate and radiate loud annoying noise.

Part 1 of this series [1] describes a calculation model which is developed to evaluate measures against squeal noise. An interesting feature of this model is that it evaluates the stability of the wheel/rail contact in the *frequency domain*. At the start of the unstable phenomenon the amplitudes are small and linearisation is still possible, which justifies the frequency domain approach. This enables the incorporation of existing frequency domain models for the dynamics of wheels and tracks. The resulting model can give fast assessments of possible unstable situations. In order to calculate the resulting amplitudes a time-domain approach is adapted.

Part 2 of the series describes a laboratory test facility which was built in order to be able to perform validation measurements in a well controlled environment, see [2].

The current Part 3 of the paper describes the measurement techniques used to determine various dynamic quantities in the vicinity of the wheel/rail contact during squeal. The results of the experiments are compared with calculations.

2 - MEASUREMENT OF DYNAMIC RESPONSES DURING SQUEAL
For the validation of the calculation model it is desired to determine various dynamic properties in the 'sound generation and transfer chain', see Fig. 1. Measuring only radiated sound would give little clues for model improvements, if inconsistencies are found. Futhermore, the transfer blocks at the right-hand side are the same as for rolling noise. Calculation models for wheel/rail rolling noise such as TWINS [3], are sufficiently validated for the time being. Therefore the validation exercise is concentrated on the left-hand side of the diagram in Fig. 1. The dynamic quantities that are of interest here are: dynamic slip velocity, the resulting dynamic forces in vertical and lateral direction.

Since measuring in the contact point of the two rotating wheels is virtually impossible it is chosen here to measure wheel and rail responses close to the contact point. The measurements are performed at

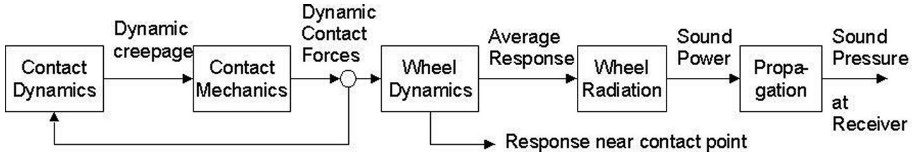

Figure 1: Diagram of transfer stages and dynamic properties in the squeal noise generation transfer chain.

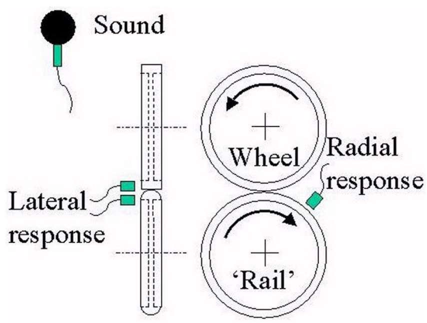

Figure 2(a): Response measurement during running test; wheel and rail-wheel dynamic responses are measured close to the contact position.

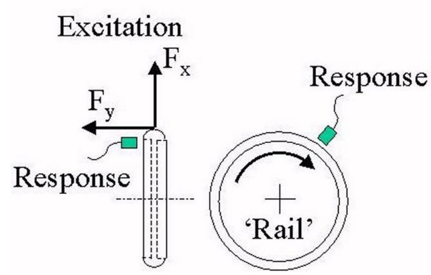

Figure 2(b): Rail-wheel transfer functions measurement; rail-wheel is excited at the contact point using hammer excitation.

a points fixed in the non-rotating frame, not at points fixed to the wheel and rail-wheel. This is done using (contact-free) magnetic transducers. The transducer set-up is illustrated in Fig. 2. The magnetic transducer signals were calibrated by comparison with accelerometer signals (for the non-rotating case).

3 - DETERMINATION OF DYNAMIC INTERACTION FORCES

In Part 2, ref [2] measured results for the contact forces were presented. Those results are the time-averaged stationary contact forces. In the current paper results for the dynamic contact forces will be presented (at squeal noise frequencies). These forces are determined by an indirect technique, using the response transducer set-up from the previous section. This consists of the following steps:

1. Determine the vertical and lateral responses of the rail-wheel during running tests. This is stored in a frequency dependent 2-element vector $\{v(f)\} = [v_x(f), v_y(f)]^{\mathrm{T}}$.

2. Determine the transfer function between a point force excitation at the *rotating* rail-wheel with the wheel removed, see Fig. 2b. This is done for vertical and lateral direction. The result is stored in a frequency dependent 2 ×2 transfer matrix $[H(f)]$.

3. Estimate the vertical and lateral interaction forces $\{F(f)\} = [F_x(f), F_y(f)]^{\mathrm{T}}$ using:

$$\{F(f)\} = [H(f)]^{-1}\{v(f)\} \tag{1}$$

Due to the high damping of the rail-wheel the transfer functions in matrix $[H(f)]$ appeared to be very smooth as a function of frequency. As a result of that the transfer function measurements could be performed with a non-rotating rail-wheel, although in principle the dynamics of a rotating wheel differ from the stationary case. Measuring in the stationary situation eases the application of the excitation force. Using the above approach it appeared feasible to determine dynamic interaction forces in the frequency range 500-6000 Hz.

4 - EXAMPLE RESULTS

The test facility will be used for a range of tests for both model validation and squeal noise abatement measures. Here, some measured results will be given and compared with calculated results. We expect to have more results available during the Internoise 2000 conference.

Fig. 3 presents (a zoom-in at the peak of) the sound pressure level measured near the wheel (30 cm distance; axle height). The first line present the sound pressure in case of a hammer blow at the pre-loaded stationary wheel. This results in sharp resonance of the two-nodal diameter axial mode at 1248 Hz. The second and third line present the base squeal noise peak for 0.5 degrees rolling angel and for 1.0 degrees rolling angle. It appears that these squeal noise frequencies do not coincide with the nearby wheel resonance frequency, but are lowered by 5-10 %. A frequency shift is also predicted by the calculation model, but only from 1248 to 1236 Hz (1 %). Further, the squeal noise peaks are much broader than the hammer blow resonance. This is due to the fact that the observed squeal noise frequency was not constant but varied about 50 Hz per wheel revolution (chirp-like). The level integrated over the peaks is: 95 dB for 0.5^o and 99 dB for 1.0^o.

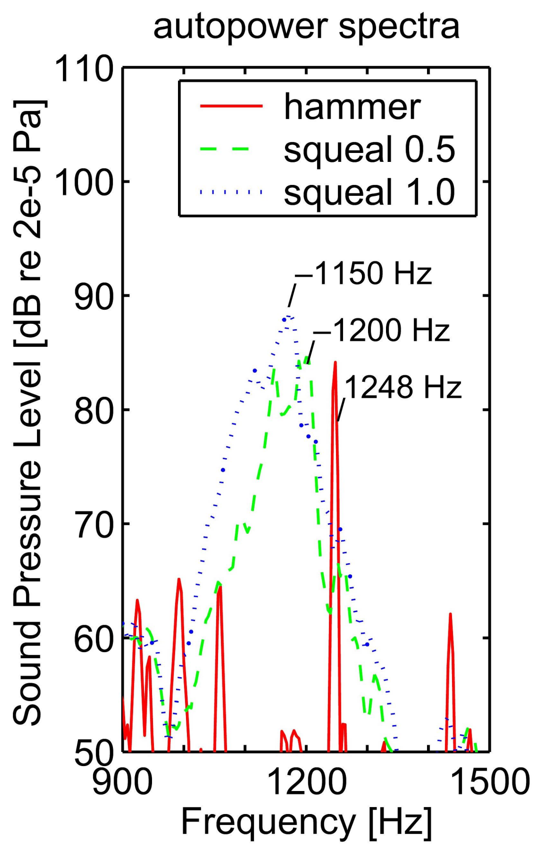

Figure 3: SPL around 1200 Hz for a hammer impact (non-rotating wheel) and for two squealing cases (0.5^o and 1.0^o rolling angle).

Fig. 4 presents the measured (according to section 3) dynamic interaction forces in case of squeal (1.0^o) for the base frequency and 4 harmonics. Both vertical and lateral interaction forces show clear peaks at the squeal-noise frequencies. The horizontal bars in the graph present the total level for each harmonic for the lateral interaction force. A calculated result is given as well. The lateral force level is strongly underestimated at the base frequency. This is related to the fact that the squeal noise frequency was calculated to high; towards the wheel mode resonance the wheel mobility rises sharply and the force amplitude is partly determined by this mobility. For the higher (off-resonant) harmonics the agreement is much better.

Fig. 5 presents the measured lateral wheel vibration velocity near the contact point. Again the squeal noise base frequency and harmonics are clearly visible and rise over 40 dB above the background level. As for the forces the horizontal bars present the total level per harmonic. The solid bar presents the measured results and the dotted bar presents the calculated result. Now, the agreement for the base frequency is much better than for Fig. 4. The agreement for the harmonics is also very reasonable. The level at 5750 Hz is now strongly overestimated (whereas the lateral force was predicted correctly). This is due to deviations between measured and calculated vertical force and wheel mobility (not presented).

Summarizing, it can be stated that the results of the calculation model are promising. Further measure-

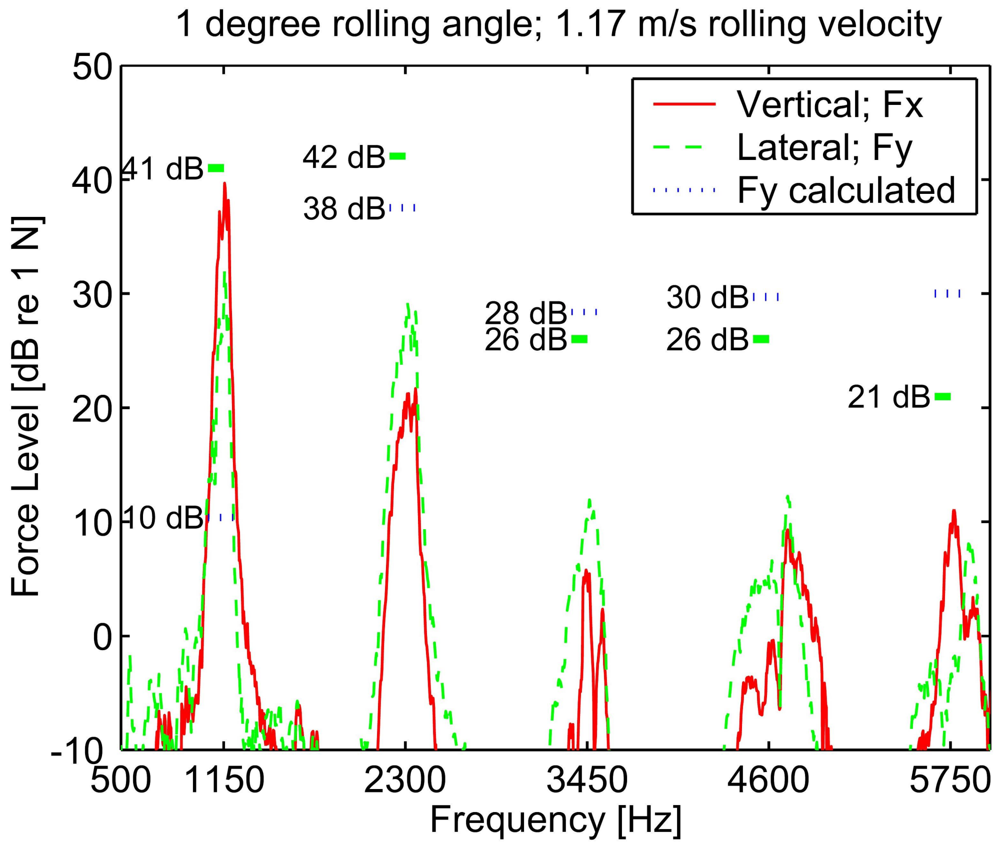

Figure 4: Dynamic interaction forces in vertical and lateral direction in case of squeal; base frequency (varies 2.5 % around 1150 Hz) and 4 harmonics are presented.

ments with different rolling angles, speeds contact points etc. are planned.

5 - CONCLUSIONS

A laboratory test facility is used to validate a calculation model for squeal noise. In the test facility the dynamic responses of wheel and rail-wheel can be measured during operational conditions. This enables the determination of dynamic contact forces in vertical and lateral direction at squeal frequencies. The measured results are compared with calculations and good agreement was found. The current results are very promising and it is expected that both the calculation and the test rig can serve adequately for the assessment of squeal noise abatement measures.
An interesting feature found in the measurements is that the base squeal noise frequencies is not equal to the wheel resonance frequency, but slightly lower (by 5 to 10%, depending on rolling angle). The calculation model also predicts this feature, although the shift is less (1 %).

ACKNOWLEDGEMENTS

This research was funded by the Dutch Railways Rail Infrastructure Management NS-RIB, the Technology Centre for Transport and Infrastructure CROW, the Dutch Ministry of Housing, Spatial Planning and the Environment VROM and the three Dutch tramway companies of Amsterdam GVBA, Rotterdam RET and the Hague HTM. The Institute for Sound and Vibration Research, NS Technisch Onderzoek and NedTrain Consulting were research partners.

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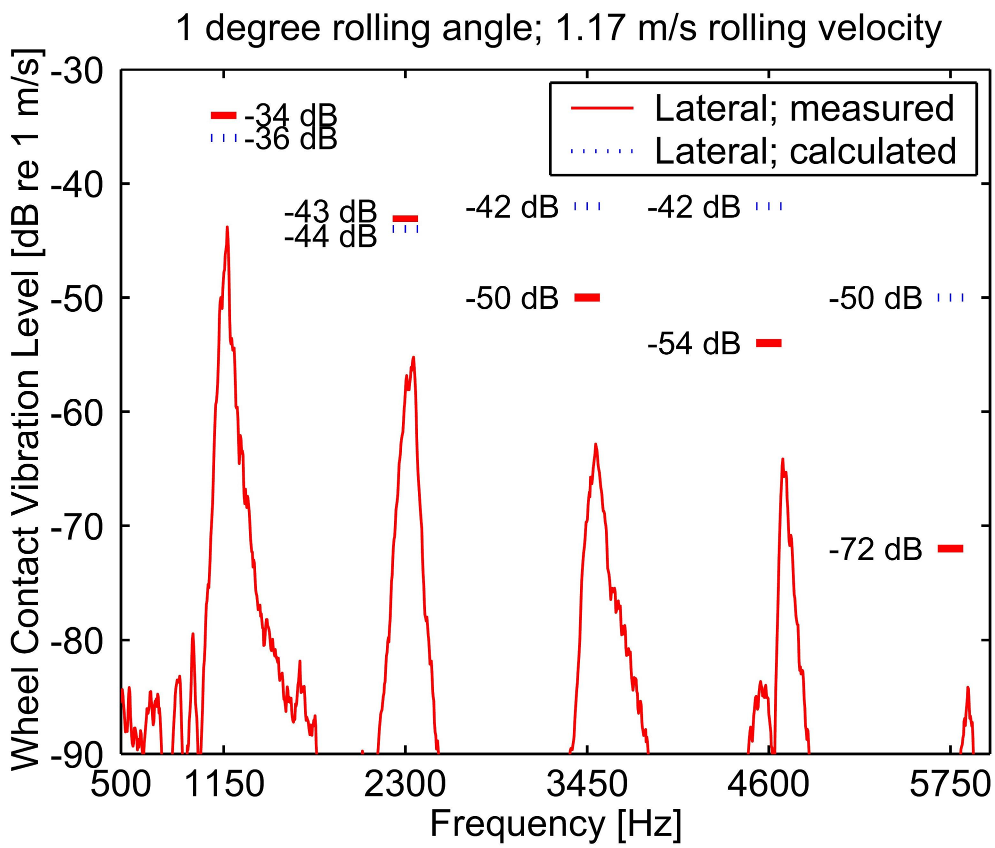

Figure 5: Measured lateral wheel velocity near the contact at squeal noise frequencies; horizontal bars give total levels per harmonic (measured and calculated).