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CURVE SQUEAL OF RAILBOUND VEHICLES (PART 2): SET-UP FOR MEASUREMENT OF CREEPAGE DEPENDENT FRICTION COEFFICIENT

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ABSTRACT
A laboratory test rig (scale 1:3) is designed for railbound vehicle squeal noise research. Using the test rig squeal noise can be generated and measurements can be performed in a well-controlled environment. The facility has been used for model validation and will be used to evaluate and optimise particular measures against squeal noise. This paper presents measurements of the friction coefficient of the rolling contact as a function of rolling angle, for cases with and without squeal (but with the similar contact surface conditions). For the non-squealing case good agreement was found between experiments and calculations. The calculated cut-on rolling angle for squeal noise for the current situation was also confirmed by the experiments. The effect of the occurrence of squeal onto the friction coefficient was found much stronger in the experiments than expected from theory.

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 creep can show unstable stick-slip behaviour, causing the wheel to oscillate and radiate loud annoying noise.

This is part 2 of a series of three papers on this topic, in which a description is given of the laboratory test facility and measured results of the friction coefficient. Part 1 [1] describes a theoretical model for the generation of squeal noise. This model analyses the stability of the contact creep in the frequency domain and, in case of instability, the resulting dynamic amplitude in the time domain. Part 3 [2] describes laboratory measurement methods and results.

For the validation measurements it is necessary to perform measurements in an environment in which all important parameters in respect to the squeal noise are known and/or can be controlled. Such measurements are almost impossible to perform in practice on a running train or tram in a curve. In such a situation important parameters such as wheel and rail profiles, contact position of wheel and rail, friction coefficients, at the instant of squeal, are difficult to determine and are usually only known as statistical averages. This is insufficient for the purpose of validation of the model. Therefore a laboratory test facility is designed. Once the parameters can be controlled, the statistical distributions, which occur in practice, can be reproduced as well.

2 - TEST RIG
The laboratory test rig is a scale model (1:3) of a wheel running on a second wheel which represents the rail ('rail-wheel'), see Figures 1 and 2. Several conceptual solutions have been considered but this concept has been chosen because it facilitated a test rig which is compact and therefore easy to control. The disadvantage of this concept is that the 'rail-wheel' has a finite radius of curvature in tangential direction, whereas for an actual rail this radius is infinite. The effect on the contact patch is compensated for by adjusting the lateral radius of the rail. In this way the various important contact parameters can be adjusted within approximately 10% from their value for an actual wheel/rail contact.

Ideally, the rail-wheel dynamics should resemble the rail dynamics. From theory however, it appears that this is not very critical as long as no sharp peaks or dips are present in the frequency response

functions. This is realised by increasing the damping of the rail-wheel by means of a pack of thin steel plates on both sides of the web. This damping method proved to be very effective ($\eta > 0.1$). Receptance measurements showed virtually no resonance peaks, which is also the case for actual tracks.
Squeal noise in the wheel can be generated by an adjustable misalignment of the axes of both wheels, which results in a certain rolling angle. Other parameters that can be controlled and/or monitored in the test rig are:

- wheel dynamics (impedance) by interchanging wheels,
- wheel damping by gradually adding damping plates (as used for the rail-wheel),
- lateral position of the contact patch on wheel,
- rolling speed,
- rail inclination angle,
- axle load,
- transverse radii of wheel and rail (by interchanging wheels),
- the surface conditions of wheel and rail-wheel,
- temperature and humidity.

These parameters are relevant for squeal-noise and cover all parameters that occur in practice.

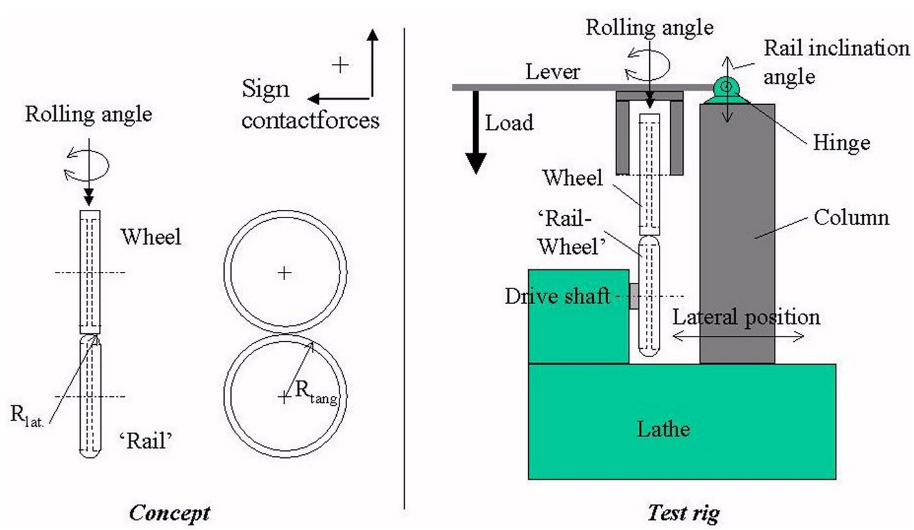

Figure 1: Diagram of laboratory test facility for squeal noise measurements; the facility is a 1:3 scale model; two wheels are built onto a lathe, which provides the drive at adjustable speeds; the various offsets and angles of the contact position can be adjusted.

3 - MEASUREMENT OF FRICTION COEFFICIENT

An key parameter for squeal noise generation is the friction coefficient of the rolling contact. This friction coefficient depends on various parameters e.g. the contact geometry, surface conditions, pre-load and, most important, on rolling angle. The friction coefficient can be estimated using the formulae given in Part 1, eqns (1) and (2), see ref. [1].
The resulting friction coefficient can also be determined in the test. Using the measured strain in the column (see Fig. 1) the quasi-static lateral (F_y) and vertical (F_x) contact forces can be derived. By definition, the friction coefficient is the ratio of these forces. Results of these measurements are given in Fig. 3 and Fig. 4 rig for a rolling speed of 4 km/h. Fig. 3 presents the contact forces as a function of rolling angle for cases with and without squeal noise (but with similar surface conditions). Due to the construction of the test rig the pre-load force F_x is not a constant but depends on the actual friction. Note: since the test rig is a scale model the contact forces are lower than for a full-scale wheel (1:3^2). The current results are equivalent to a full-scale load of about 1 tons, which is still rather small.

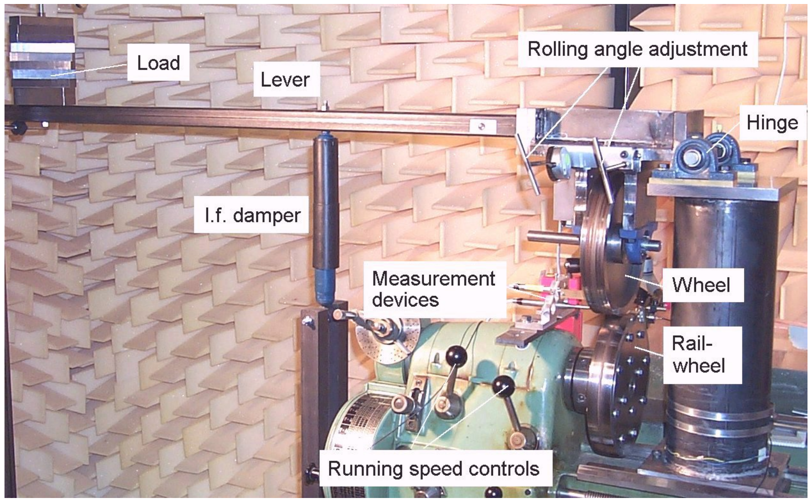

Figure 2: Photograph of the test rig; wheel and 'rail-wheel' are visible at the right hand side; a car damper is fitted to the lever for low frequency stability of the test rig; pre-load can be adjusted by adding metal blocks at the end of the lever (left).

Fig. 4 presents the friction coefficient based on the results in Fig. 3. For the squealing case the squeal cuts-on at a rolling angle of about 0.4 degrees. As soon as squeal occurs the friction coefficient drops relatively to the non-squealing case (similar to Remington [3]). Below this angle no squeal occurs and results of both measurements are the same.
In order to compare the results with theory the measured curves are adapted to represent constant pre-load of 1 kN using (Part 1, eqn 1) ref. [1] and standard Hertzian contact mechanics. This is presented in Fig. 5, which shows slight deviations from Fig. 4.
Fig. 6 presents the calculated results using (Part 1, eqn 1 and eqn 2), ref [1]. Given various uncertainties, e.g. in τ_W, τ_R and surface conditions, the agreement between calculations and measurements for the non-squealing case is very good. For the squealing case the agreement is clearly less. Although the cut-on angle for squeal for both calculations and measurements is about 0.4 degrees here, the drop of the friction coefficient observed in the measurements is hardly visible in the calculations (although present).

For the calculation model the friction coefficient curve for the non-squealing case is used as an *input*. The creepage is stationary for the non-squealing case. In case of squeal a non-stationary lateral creepage with large amplitude is superimposed onto the stationary creepage, and the friction coefficient is modified as a function of time according to the non-squealing curve. As a result of this the time-averaged friction coefficient for non-stationary creepage, is *lower* than for stationary creepage.

4 - CONCLUSIONS

A laboratory test rig is designed for railbound vehicle squeal noise research. This test rig comprises a 1:3 scale model of a wheel. Using the test rig squeal noise can be generated and measurements can be performed in a well controlled environment. This allows the acquisition of data [3] that can be used for validation of the theoretical model [1]. Furthermore, the test facility can also be used to evaluate and optimise particular measures against squeal noise such as wheel damping, lubrication, wheel and/or rail transverse profile adjustment.
Results are presented of measurements of the friction coefficient of the rolling contact as a function of rolling angle. This was done for a situation in which squeal noise occurs and in a situation in which no squeal noise occurs (but with similar contact surface conditions: no lubricants or the like). For the non-squealing case agreement between experiment and calculations was good. For the squealing case, it is expected from theory that the friction coefficient diminishes beyond the cut-on rolling angle for squeal noise. This was confirmed in the measurement, although the effect of the friction coefficient was much greater than indicated by the calculations. This is topic of further research.

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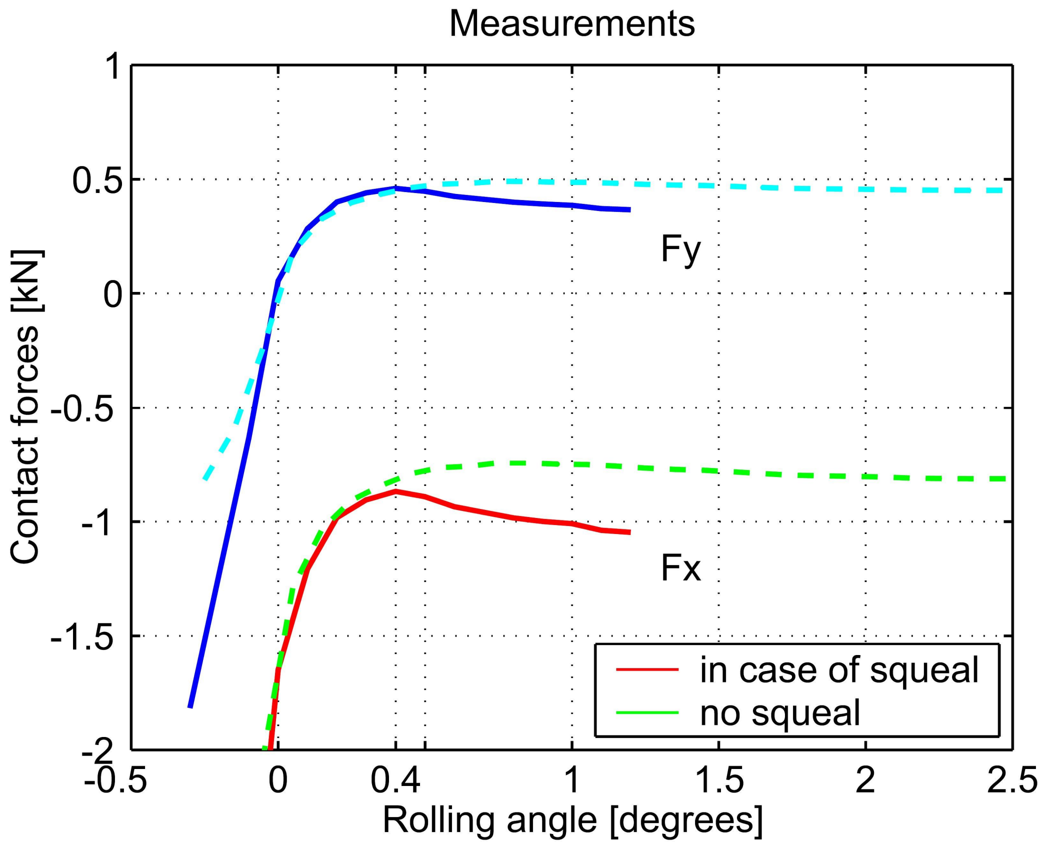

Figure 3: Measured contact forces in 1:3 test rig for a squealing and non-squealing case; F_x is the vertical force; F_y is the lateral force.

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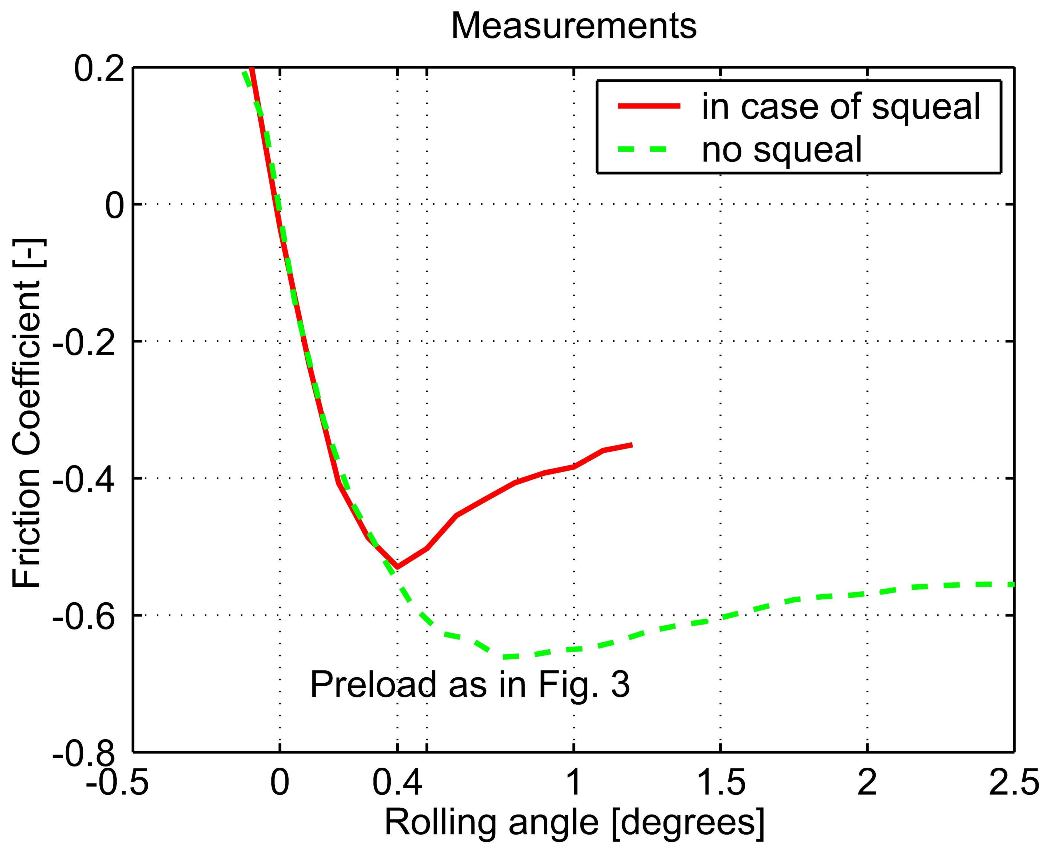

Figure 4: Measured friction coefficient in test rig; squeal cuts on at a rolling angle of 0.4 degrees.

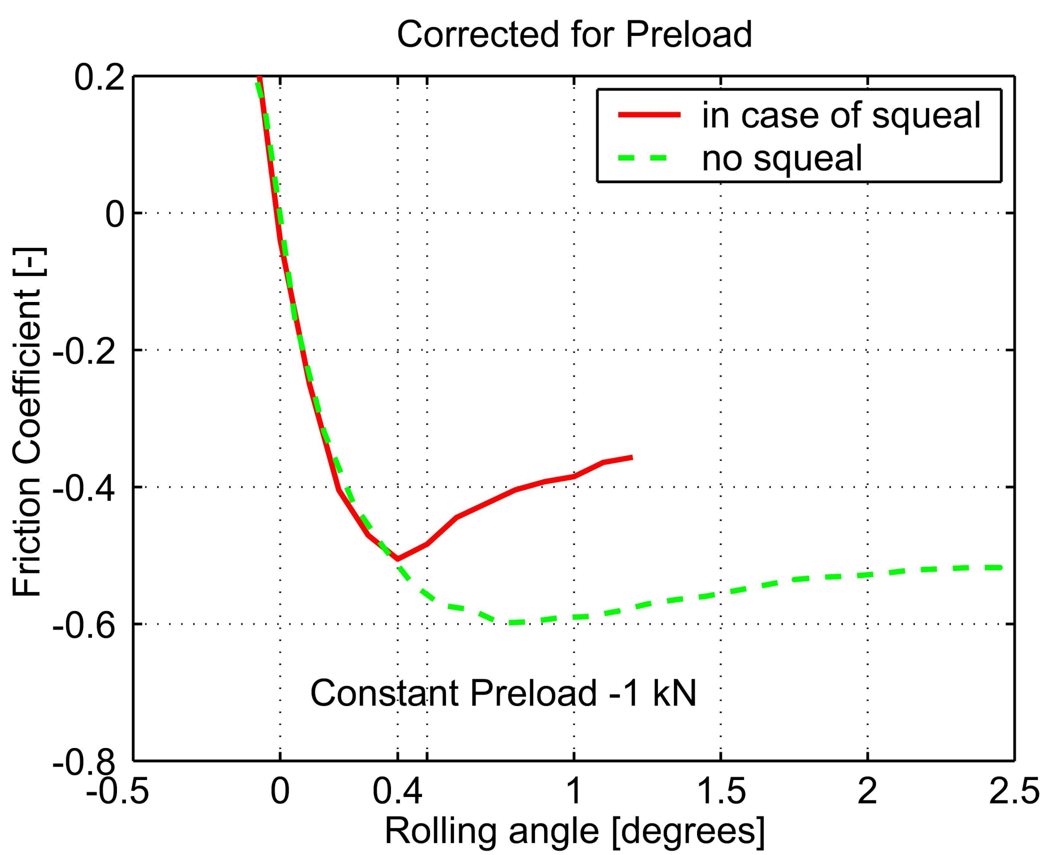

Figure 5: Measured friction coefficient corrected for variations in vertical pre-load, using (Part 1, eqn 1), ref [1].

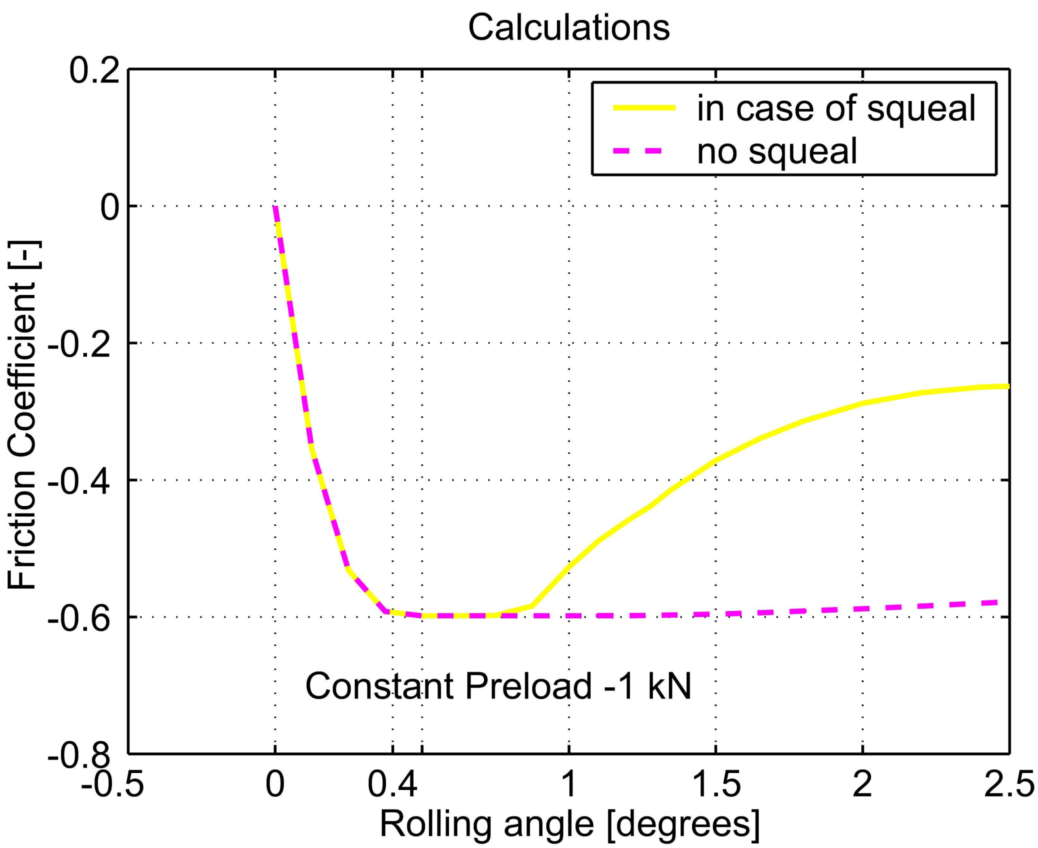

Figure 6: Calculated friction coefficient for the test rig, using (Part 1, eqn 1 and 2) ref [1]), τ_W 530 N/mm^2; τ_R 530 N/mm^2; ab.72 mm^2.