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# Influence of spin creepage and contact angle on curve squeal: A numerical approach

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ABSTRACT

Curve squeal is a loud tonal sound that may arise when a railway vehicle negotiates a tight curve. Due to the nonlinear nature of squeal, time-domain models provide a higher degree of accuracy in comparison to frequency-domain models and also enable the determination of squeal amplitudes. In the present paper, a previously developed engineering time-domain model for curve squeal is extended to include the effects of the contact angle and spin creepage. The extensions enable the evaluation of more realistic squeal cases with the computationally efficient model. The model validation against Kalker's variational contact model shows good agreement between the models. Results of studies on the influence of spin creepage and contact angle show that the contact angle has a significant influence on the vertical-lateral dynamics coupling and, therefore, influences both squeal amplitude and frequency. Spin creepage mainly influences processes in the contact, therefore influencing the tangential contact force amplitude. In the combined spin-contact angle study the spin creepage value is kinematically related to the contact angle value. Results indicate that the influence of the contact angle is dominant over the influence of spin creepage. In general, results indicate that the most crucial factors in squeal are those that influence the dynamics coupling: the contact angle, wheel/rail contact positions and friction.



## 1. Introduction

When a railway vehicle negotiates a narrow curve, a noise disturbance known as curve squeal may arise. Curve squeal is usually characterised by high noise levels and is tonal, i.e. dominated by a single high-pitched tone in the range of 250 Hz to 10 kHz [1–3].

The curve radius at which squeal may occur is dependent on the length of the vehicle bogie wheelbase [4]. In a sufficiently tight curve, creepages will be high enough for the tangential contact force to be near or at saturation. Saturation, according to [5,6], creates conditions for the development of frictional instability. In general, such conditions and possibly squeal are expected in curves of a radius less than 200 m [2]. Unfortunately, the incidence of tight curves is highest in cities and urban areas, leading to a high number of people being exposed to squeal. The rise in awareness of noise impact on public health [7] led to an increase in the need to address the squeal problem. This, in turn, led to an increased need for better understanding and modelling of the phenomenon.

During curving the railway wheel slides laterally on the rail. This sliding motion, referred to as lateral creepage, provides energy input to the wheel and rail. In addition to lateral creepage, longitudinal creepage is also present between wheel and rail during curving. However, longitudinal creepage was early discredited as a possible source of energy in squeal [4,8]. During

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squeal, the wheel dominantly vibrates in the lateral direction [3,9,10]. Therefore, a lateral excitation force efficiently sustains vibrations. This leaves lateral creepage as the main energy input sustaining curve squeal.

In addition to lateral and longitudinal creepage, spin creepage may also be present in the contact. Spin creepage describes the relative rotational motion between the wheel and rail. In general, spin alone is not considered to lead to squeal, but it influences contact forces and processes in the contact area. Therefore, spin creepage may influence the energy input to the wheel/rail system.

Input energy itself is not enough for curve squeal to develop and a mechanism that turns energy into unstable vibrations is necessary. Only with a combination of input energy and an instability producing mechanism can self-excited vibrations occur. Two commonly discussed excitation mechanisms are the falling friction mechanism [4,11] and the dynamics-coupling mechanism [1].

The starting point of this paper is the computationally efficient engineering time-domain model for curve squeal presented in Ref. [12]. The engineering model is based on the dynamics-coupling mechanism, also known as geometric coupling or mode coupling in the context of brake squeal [1,9,13–17]. Squeal is considered to occur due to the vertical-lateral dynamics coupling of the wheel dynamics. This coupling is present because of the geometric and loading asymmetry of the wheel, which shows non-zero vertical and lateral displacements when loaded purely vertically (or laterally) [13].

In comparison to frequency-domain models, time-domain models are more accurate and enable the determination of squeal amplitudes. Curve squeal of low amplitude is probably not going to create a high disturbance, whereas high-amplitude squeal will. Time-domain models can also account for the nonlinearities present in the wheel/rail system, which gives higher accuracy. As will be presented in the paper, curve squeal shows a highly nonlinear behaviour, emphasising the need for a nonlinear model. Examples of time-domain models for squeal include, but are not limited to [1,13,18].

In this paper, the ability and accuracy of the engineering model [12] are improved by including contact angle and spin creepage effects. These are especially important in analyses of cases with significant contact angle values. An example is contact occurring close to or at the wheel flange and rail gauge corner. The computational efficiency of the engineering model [12] is fully retained.

The paper is organised as follows. Section 2 presents the extended engineering model for squeal. Wheel and rail models are briefly presented followed by the description of the contact model. Emphasis is put on extensions of the model: the contact angle and spin creepage. Section 3 presents the validation of the squeal model against Pieringer's model [13] based on Kalker's variational contact model [19]. Finally, Section 4 presents results from studies of the contact angle and spin creepage influence on curve squeal occurrence and amplitudes. Section 5 summarizes the findings of the paper.

## 2. The engineering model for curve squeal

The engineering model for squeal [12] was developed on the basis of the time-domain squeal model by Pieringer [13]. The engineering model, Fig. 1, consists of three main submodels: the wheel, the rail and the rolling-contact model. The model applies a Green's functions and convolution approach, as used by Pieringer [13], instead of a classic time-integration scheme. The sound radiation evaluation is a post-processing step and is not performed in the present paper.

### 2.1. Wheel and rail models

A single flexible C20-metro steel wheel with a nominal rolling diameter of 780 mm and a worn S1002 wheel profile was modelled. This kind of wheel is found on, for example, vehicles of the Stockholm metro system. A finite element (FE) model based on axisymmetric elements was used to determine the wheel response for frequencies up to 7 kHz [13].

The rail is a BV50-type rail, a common Swedish rail type, with a worn profile and 1:40 inclination. A three-dimensional waveguide finite element (WFE) model was used to obtain the response of the rail for frequencies up to 7 kHz [13]. Further details about the wheel and rail models can be found in Refs. [13,20].

Fig. 2 shows the wheel (a) and rail (b) cross sections with three different contact positions at which the wheel and the rail make contact. The contact positions were determined from the measured wheel and rail profiles, and are defined with the lateral

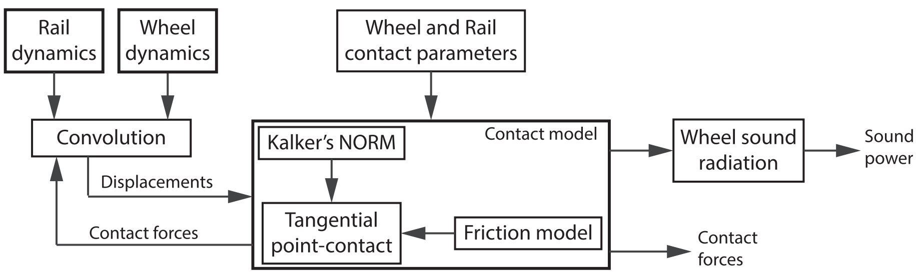


**Fig. 1.** Schematic of the engineering model for curve squeal [12].