# Session B4 : Systèmes pour la réduction du bruit ferroviaire – System for railway noise control

## SQUEAL - Noise reduction in urban transport by rail treatment

## Squeal : réduction du bruit d'un système de transport urbain sur rail

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*This paper summarises the results of the research carried out under the EC contract BRPR-CT97-0477, "Squeal noise reduction in urban transport by rail treatment".*

*Almost any rapid transit system has curved track sections in which squeal is generated. Squeal is normally most predominant in tight curves. Curved track sections where squeal is a problem normally cover a total track length of a couple of hundred metres in a normal rapid transit system of many kilometres of track. The most cost effective way of solving squeal noise problems is thus to treat only the track section causing the squeal rather than to treat all wheels on the entire fleet of vehicles. The aim of this project is therefore to find measures applied onto the rail, which can reduce the squeal noise generation. The considered solutions are based on the assumption that squeal is due to a lateral stick-slip of the wheel onto the rail that produces a vibration of the wheel which acts as a loudspeaker. Based on this assumption, it was decided to :*
*- develop a software to calculate the squeal noise level as a function of all track and rolling stock parameters ;*
*- determine the efficiency of the following rail treatments for squeal noise reduction :*

*- change of the friction-creep curve between the wheel and the rail by hardsurfacing of the railhead ;*
*- change the impedance of the rail by an add-on profile ;*
*- increase the rail damping by specific rail fasteners.*

*As planned, after intensive numerical and lab tests, the 3 different solutions were tested on test tracks. The main conclusions of the project are that :*

*- it is not always the loudspeaker mode of the wheel that is responsible for squeal, but squeal also occurs at a series of axial bending resonance frequencies of the wheel surface and tread.*
*- the theoretical model of a lateral stick-slip between the wheel and the rail cannot explain all squeal occurrences : there is also the longitudinal slip leading to torsion of the wheel axle.*
*- rail hardsurfacing is found satisfactory for squeal noise reduction, but needs further development (manufacturing process).*
*- the add-on saddle profile rail is a very promising solution that needs further optimisation.*
*- the damped rail fasteners lead to a significant reduction of the rail vibrations but no reduction of the squeal noise was obtained. The structure borne noise and vibrations are reduced very significantly.*
*- only solutions that eliminate simultaneously lateral and longitudinal slip seem to be effective.*

### Objectives of the project

The main objectives of the project were :

- the development of a calculation model and software in order to be able to predict the squeal frequency and noise level, as a function of the track and rolling stock parameters, with an accuracy of 3 % on the squeal frequency and 5 dB (A) on the squeal noise level.
- the selection, design and testing of different rail treatment measures in order to reduce or suppress squeal noise.

### Scientific and technical description of the project

#### Development of a squeal noise model including a suitable model for the roll-slip excitation of the wheel and rail

***Model based on Rudd's equation***

According to Rudd [1], there are three possible mechanisms for wheel squeal :

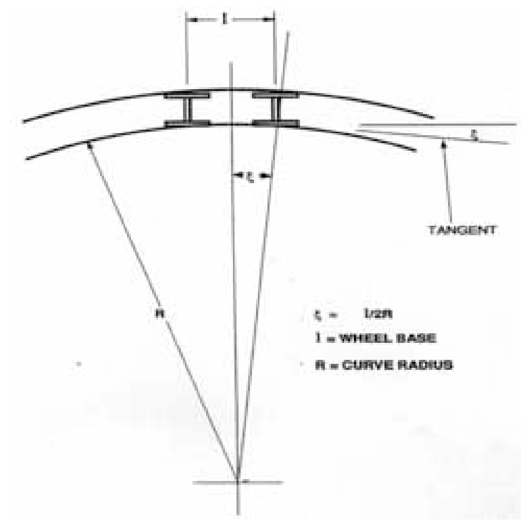



Fig. 3.1.1

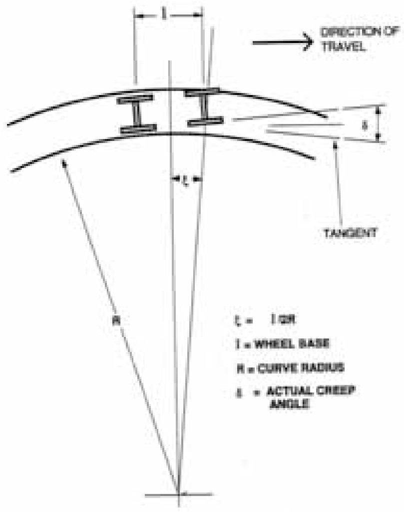



Fig. 3.1.2 : Truck crabbing under actual conditions

- longitudinal stick-slip due to the different translation velocities between two wheels on a rigid axle : Rudd thinks it is probably not the source of squeal, because trucks with independently driven wheels also squeal ;

- flange rubbing due to contact between the wheel flange and the outside rail head : Rudd thinks it is not a sufficient source mechanism if alone ;

- crabbing of the wheel across the top of the rail : according to Rudd, the most probable cause.

Due to the finite length of the 2-axles truck and the radius of curvature of the rail, both axles cannot lie upon curve radius, as shown on figure 3.1.1. This figure shows the geometrical relation between the creep angle ξ, the wheel base l and the curve radius R. Under actual conditions, however, the leading axle of the truck rides toward the outside of the curve, while the trailing axle travels between the two rails, i.e. a reduction of the creep angle at the trailing axle, but an increase of the creep angle at the leading axle, as shown on figure 3.1.2.

The radiated sound power can be calculated as :

$$L_W = 10\log\frac{\phi\rho c A u_0^2}{10^{-12}} \qquad (1)$$

where $\phi$ is the radiation efficiency, $\rho c$ is the air impedance, A is the wheel radiating area, $u_0$ is the wheel vibration velocity.

As a first approximation, and for a track in open air without major screen effect, the sound pressure level can be calculated by :

SPL = $L_W - 10\log(2\pi d^2)$ for reflecting ground
SPL = $L_W - 10\log(4\pi d^2)$ for absorbing ground

## Lumped parameter model with non-linear friction element

### *Why a lumped parameter model ?*

The squeal noise generation is a non-linear process. It is therefore necessary to establish a mathematical model that will incorporate the non-linearity of the process to be able to perform a parametric study. Since the non-linearity alone introduces severe difficulties in the modelling work, the wheel and rail structures need to be represented as simple as possible. Therefore a lumped parameter model has been developed, where the wheel and rail are represented as simple mass/spring-systems.

### *Short description of the model*

The lumped parameter model includes two damped single-degree-of-freedom systems on each side of a non-linear friction element. The system is driven to self-sustained vibrations by pulling the wheel end of the system with a constant velocity similar to the constant crabbing velocity occurring when a two-axle bogie with fixed axles is passing through a curved track.

The model, shown in summarised form in figure 3.1.3, includes one mode of the wheel (shown to the left of the friction element) and one mode of the rail (shown to the right of the friction element).


Fig. 3.1.3. : Conceptual sketch of the lumped parameter model for prediction of wheel/rail squeal noise

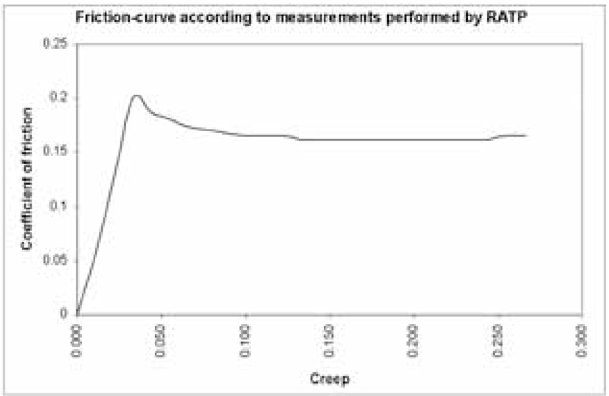



**Fig. : 3.1.4 Friction curve measured by RATP and used in numerical form in the computer model**

The non-linear friction characteristics of the contact between the wheel and the rail is stored as a (numerical) function in the ANSYS computer model and is based on measurements of actual wheel/rail friction, see figure 3.1.4. Dynamic data for the wheel and the rail have been taken from measurements performed by RATP.

## Squeal model verification and tuning on existing curved test tracks which produce squeal

***Squeal model validation***

*Rudd's model*

*Noise and vibration measurements on three test tracks*

Measurements were made on three existing curved test tracks in tunnel, on RATP's metro network in Paris. The selection of the test sites was governed by following rules:
- repeatability of the squeal phenomenon;
- absence of wheel treatment on the rolling stock;
- constant train speed of ± 40 km/h.

The selected sites were:
- on line 2, between the station Avron and the terminus Nation (curve radius 100 m);
- on line 5, between the stations Gare du Nore and Gare de l'Est (curve radius 100 m);
- on line 12, between the stations Marx Dormoy and Marcadet-Poissonniers (curve radius 60 m).

The measurements include:

- measurement of the impedances of three different wheels, in order to determine the influence of the wheel type on the squeal frequency;
- measurement of the impedances of the rail, in order to determine the ratio of the wheel and rail impedances;
- vibration measurements on the rail during squeal;
- noise measurements 2 m from the track during squeal.

Following conclusions were obtained:

- wheel and rail lateral transfer function amplitudes are of the same order of magnitude at the wheel resonance frequencies;
- the main peaks in the rail vibration spectra occur at frequencies of 500 Hz and 1 200 Hz and their harmonics for the three sites (these frequencies do not vary for different curve radii and wheel types), with an equivalent modal damping of 1.25 % for the main peaks during squeal;
- the global rail vibration level is governed by the contribution of frequencies up to 2 kHz; the average global vibration levels measured on the rail and between 10 and 20 mm/s.
- the main peaks in the noise spectra occur at the same frequencies as in the rail vibration spectra (figure 3.2.1); table 3.2.1 summarizes the maximum global noise levels: there is no significant noise increase for the test site on line 12, where the curve radius is lower (60 m) than on the test sites on lines 2 and 5 (100 m).

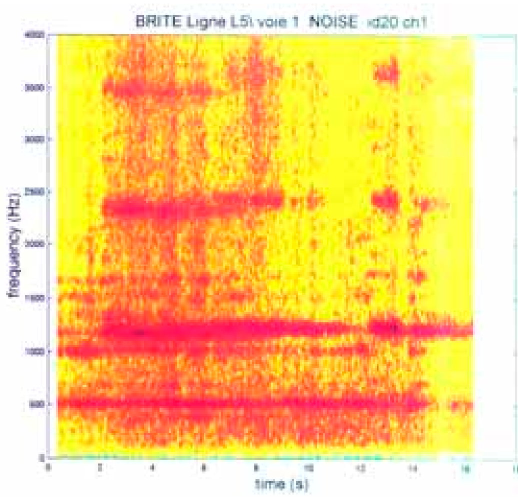



**Fig. 3.2.1.**

| Test site | SPL dB (A) |
|---|---|
| 2 | 110 |
| 5 | 112 |
| 12 | 110 |

**Tabl. 3.2.1 : Measured maximum global noise level**

*Tuning of the predicted squeal frequencies*

Finite element models of the different wheels used, were created, using 3D elements, with varying boundary conditions and different wear of the tread.
The eigenmodes were calculated, in each case, up to 4 kHz. This led to the following considerations:
the "umbrella" eigenmode, initially considered as responsible for squeal noise, occurs at frequencies ranging from 412 to 850 Hz depending on the considered wheel type, wearing condition and boundary conditions;
we found a series of axial bending eigenmodes of the wheel surface and tread whose corresponding eigenfrequencies were more independent of the considered wheel type, wearing condition and boundary conditions. The first two eigenmodes of this series occurred at frequencies around 500 and 1 200 Hz. They are shown in projection and in a yz-plane view on figures 3.2.2 to 3.2.5.
The models were further validated, using transfer functions measured on a static system consisting of complete wheel set standing on rails.

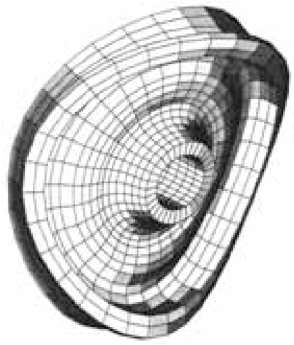


**Fig. 3.2.2. : Eigenmode at frequency ≅ 500 Hz**

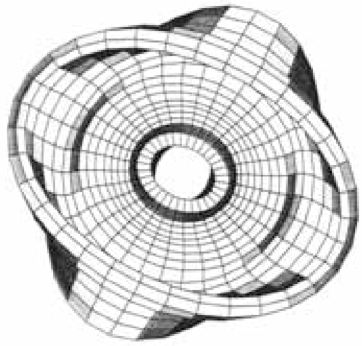


**Fig. 3.2.3. : Eigenmode at frequency ≅ 500 Hz**

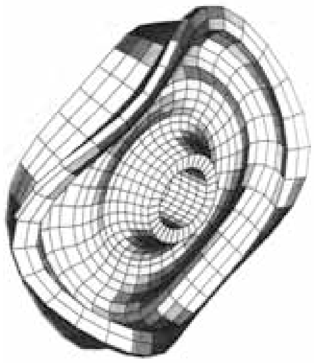


**Fig. 3.2.4. : Eigenmode at frequency ≅ 1 200 Hz**

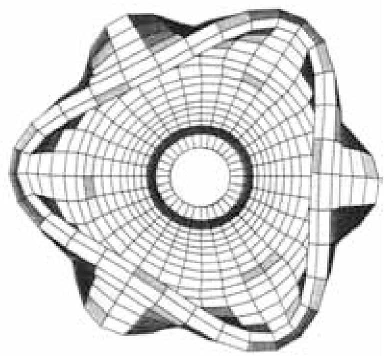


**Fig. 3.2.5. : Eigenmode at frequency ≅ 1 200 Hz**

*Tuning of the predicted sound pressure levels*

The sound pressure levels due to the wheel and the rail were calculated and compared to the measured values for each test site, leading to the following results :

- the contribution of the rail to the global SPL is negligible for the three test sites ;
- table 3.2.3 summarizes the calculated and measured
- global SPL for the three test sites, showing that a good calculated estimation of the squeal noise can be obtained.

| line | Calculated SPL | Measured SPL |
|---|---|---|
| 2 | 110 | 110 |
| 5 | 110 | 112 |
| 12 | 113 | 110 |

**Tabl. 3.2.3 : Comparison between calculated and measured SPL [dB (A)]**

it is very important to determine with a high accuracy the friction-creep curve.

<u>*Lumped parameter model with non-linear friction element*</u>

A standard wheel/rail configuration with data according to table 3.2.4 and predicted vibration velocity in the wheel according to figure 3.2.6 were used as a calibration case. The effects of parameter variations were evaluated by comparison to the results from the standard wheel/rail configuration.

| Parameter | Value |
|---|---|
| Loss factor, Wheel | 0.004 |
| Loss factor, Rail | 0.035 |
| Resonance frequency, Whee | 670 Hz |
| Resonance frequency, Rail | 400 Hz |
| Dynamic mass, Wheel | 80 kg |
| Dynamic mass, Rail | 125 kg |
| Vehicle velocity | 7 m/s |
| Axle distance | 2 m |
| Curve radius | 100 m |

**Tabl. 3.2.4. : Data for the standard parameters case**

As can be seen in figure 3.2.6, the system is squealing nicely at a frequency determined by the resonance (mode) in the wheel. This model was used to study the effect of parameter modifications.

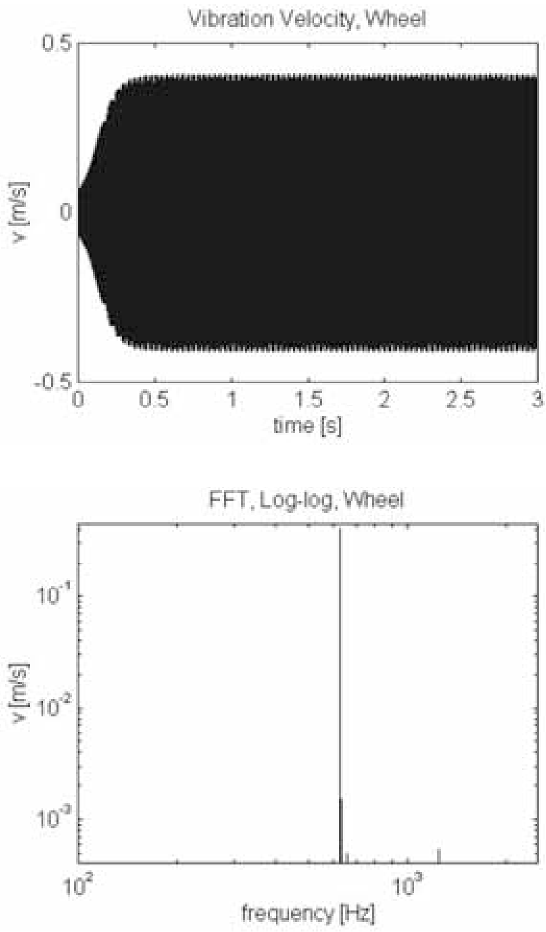



**Fig. 3.2.6 : Wheel disc vibration velocity during squeal for the standard wheel/rail configuration shown as time history and frequency spectrum**

***Parametric study to determine influential parameters***

A great number of different parameter combinations have been studied. The conclusions from the studies of the lumped parameter model are :

- In order to achieve reduction of wheel/rail squeal it is necessary to first reduce the mechanical impedance of the rail at squeal frequencies to a value below the wheel impedances.
- Increased damping of the rail will be effective against squeal as long as vibrational energy is fed into the energy absorbing layers on the rail. This will automatically happen for the wheel since, for normal systems, it shows the major vibration deflections[1]. Therefore an increase of the wheel damping will normally be effective against squeal. Damping applied onto the rail is normally not effective unless the rail exhibits high vibrations at squeal frequency during squeal.
- Damping will be an efficient measure against squeal if the rail impedances are decreased in order to allow an enough big part of the vibrational energy to be fed into the damping layer and absorbed.

*1 Because the wheel impedance is in most cases lower than the rail impedance.*

- If, however, the damping of the rail is increased to an extent that the rail impedance again will be comparable to or greater than the wheel impedance, then the wheel will take over and perform the major part of the squeal vibrations. For a particular design, which includes a reduction of rail impedances, there thus exists a maximum allowed loss factor for a maximum of squeal noise reduction. This turned out to be about 0.2 for the tested system according to the results from calculations with aid of the lumped parameter model.

## Design and prototype testing of squeal preventing systems

***Damped rail fastener = APT-BF fastener***

*Technical data RATP*

| | |
|---|---|
| Rolling stock | 5 trucks of ± 16 m length each |
| Loads per axle | empty : 4.5 to 6.7 tons<br>loaded : 7.3 to 9.5 tons |
| Commercial speed | 60 km/h max. |
| Distance between 2 axles of a bogie | ± 2,1 m |
| Distance between bogies | ± 10 m |
| Rail type | RATP 52 kg |
| Track | ballasted track with wooden sleepers |

*Description of the initial laterally damped fastener APT-BF*

The base plate (dimensions 380 x 230 mm) of the fastener is designed to fix the RATP 52 kg rail by means of NABLA-GRIFFON clips. The fastener is equipped with an anti-vibration pad in microcellular polyurethane whose dimensions and static and dynamic stiffnesses are to be determined in function of the desired performance. A tuning of the height is made possible by use of inserts with variable thickness. The preload system consists of two springs, the first of which gives the initial preload from 5 to 20 kN (function of the axle load), the second enabling the uncoupling of the stiff spring during the passages.

*Lab tests*

First lab tests were carried out on one fastener with the corresponding piece of rail. The maximum axle load fastening locations.

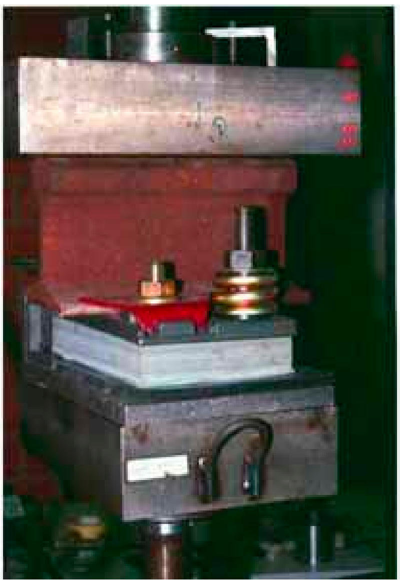
The static characteristics are :

- static deflection of about 4 mm at 25 kN load ;
- static stiffness of about 6 kN/mm.

The dynamic stiffnesses under different loads are :

- 5 kN : 9 MN/m
- 10 kN : 9 MN/m
- 20 kN : 5 MN/m
- 25 kN : 5 MN/m

i.e. the dynamic stiffness decreases for increasing load. It is to be noted that most resilient direct fixation systems existing today have generally a dynamic stiffness of more then 20 MN/m.



Further lab tests were carried out :

- static and dynamic behaviour of a 6 m long rail with 7 fasteners ;
- fatigue tests under an angle of 22°

*Anti-vibration performance on tangent track test site*

The APT-BF fastening system was installed on a test site on metro line 2 of RATP (Paris) between stations Ternes and Courcelles. The initial track consisted of wooden sleepers on ballast. Vibration measurements with both systems revealed that the APT-BF system led to a reduction of the vibration levels with about 14 dB on the tunnel wall, and even 20 dB in nearby buildings, i.e. nearly the performance of a floating slab.

The following modifications were introduced for use in curves :
- modification of the bolt type and height ;
- modification of the geometry of the base plate ;
- addition of an abutment system.

*Site testing for installing damped rail fasteners*

The test site is located on metro line 12 of RATP (Paris, see above). The installation of the laterally damped rail fasteners (APT-BF-C) on special (larger) wooden sleepers was carried out in the workshop. The installation on the test track took place in October. Noise and vibration measurements were carried out in a section located in the middle of the 60 m radius curve :

The results of these measurements are :

- A reduction of the tunnel wall vibrations with 12 dB (global reduction). A change in the vibration spectrum : the maximum vibration level is observed in the 1/3 octave frequency bands of 63 Hz – 80 Hz (before installation) and of 31,5 Hz (after installation). This results in a reduction of the structure borne noise of about 25 dB (A).

- A complete uncoupling between wheel and rail during squealing a gauge widening of 5 mm and a vertical rail displacement of 3 mm during vehicle passage. This was the expected performance of the fastener. The lateral rail vibrations at squeal frequency are reduced with 30 dB during squealing after installing the fasteners. There is no more stick-slip in the transverse direction.
- The squeal noise amplitude is not reduced. The wheel is still vibrating at about 500 Hz. The wheel is most probably excited by the torsion in the axle, generated by the relative rotating speed differences between both wheels during passage in small radius curves, generating a stick-slip phenomenon in the longitudinal direction. The longitudinal slip cannot be prevented by resilient fasteners, only by a rail surface treatment (which is effective in all directions), or by add-on rail profiles (see hereafter) which introduce flexibility in all directions.

***The saddle profile***

*What is a saddle profile rail ?*

A saddle profile rail is a rail concept where the top of the railhead is arranged in form of a U-shaped profile (called Saddle Profile, figure 3.3.1) where the wheel will run on the top of the saddle profile. The saddle profile is in turn uncoupled from the rest of the rail structure by a thin rubber mat.

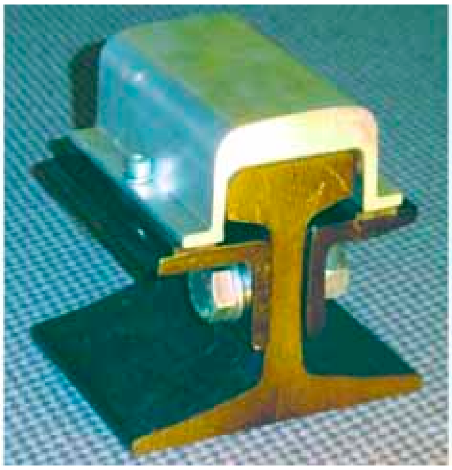


**Fig. 3.3.1**

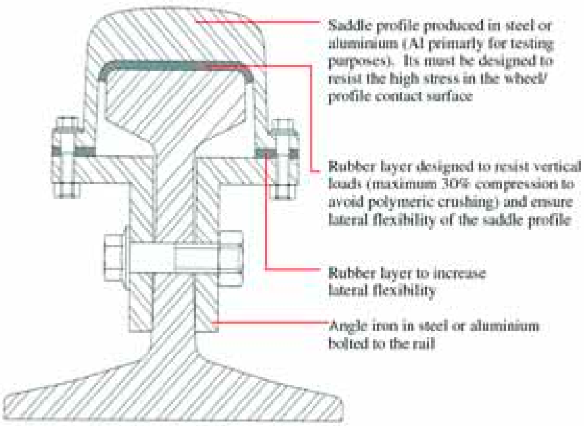



**Fig. 3.3.2. : Saddle profile for squeal noise reduction mounted on top of the railhead. Conceptual sketch - cross section**

*Reduced rail impedance – selecting the most feasible concept*

The study of the parameter variations revealed that for the normal wheel/rail configuration where the rail impedance is higher compared to the wheel then the wheel will just “slide around” on a resting, non-moving rail. We could also see that the non-linear nature of the process amplifies the importance of the differences in impedance between the wheel and the rail. This means that for any efficient treatment for squeal noise reduction aimed to be applied onto the rail, the rail impedance must first be decreased so that it is lower compared to the wheel impedance.

Decreasing the dynamic mass of the rail is one way of achieving this (decreasing the mechanical impedance of the rail) since the dominating part of the rail impedance at squeal frequencies would be the mass reactance. One way of achieving reducing the rail dynamic mass and thus also the rail impedance at squeal frequencies would be to uncouple part of the railhead from the rest of the rail structure. One way of achieving such lowering of the dynamic mass and impedance of the rail was found to be adding a U-shaped profile onto the railhead, which is elastically uncoupled from the rest of the rail. In the continuation this method will be referred to as the “Saddle Profile Rail. A sketch of the Saddle Profile Rail mounted on top of the railhead is presented in figure 3.3.2 as a cross section. There might be other ways of creating the lower rail impedance e.g. by in-lay structures mounted in grooves grinded into the railhead or by making the rail in a lighter material (such as aluminium or carbon-reinforced plastics). In this project, we though decided to select the Saddle Profile Rail to demonstrate the proper working of the concept.

*Squeal sound measurements after installation of the saddle profile rail*

The saddle profile was installed at a test site in Stockholm (SL network) in October 2000. Measurements of squeal sound levels were performed with aid of track side mounted microphones according to the sketch in figure 3.3.3. They were performed as soon as possible after completed mounting.
The measurements were performed with a stationary microphone at 1,3 m from the inner track edge and 0,5 m above rail upper surface.

Fig. 3.3.4 presents the squeal sound pressure levels during train passage. From the microphone placed beside the regular rail the typical tonal components associated with the squeal process can be found, but for the microphone position beside the rail with saddle profile, these components are completely eliminated.

The reduction in squeal sound pressure level is about 35 – 37 dB for the fundamental squeal frequency at 450 Hz and as much as 25 - 30 dB is achieved for a squeal frequency at 900 Hz. This means that the squeal sound is almost totally eliminated.

***Development of a procedure for hardsurfacing the rail***

In order to prevent squeal, one of the proposed techniques is to modify the friction characteristics at the wheel-rail contact point. To achieve this, different hardsurfacing techniques and materials have been proposed by RATP. Treated rail samples have been tested with wheel samples in order to determine the corresponding friction characteristics. The conclusion of testing the friction

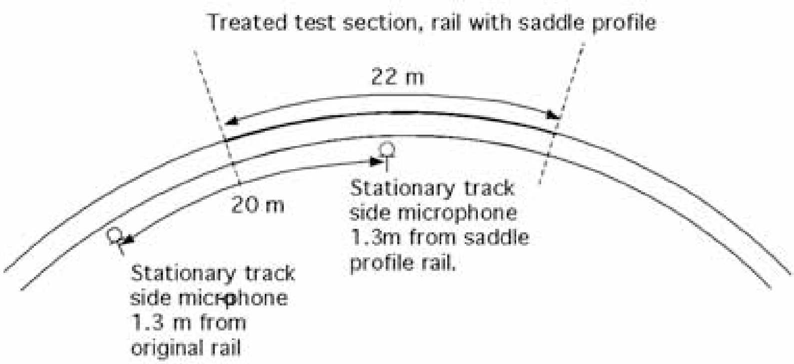



**Fig. 3.3.3. : Measurement set-up at the test site for comparative squeal measurements**

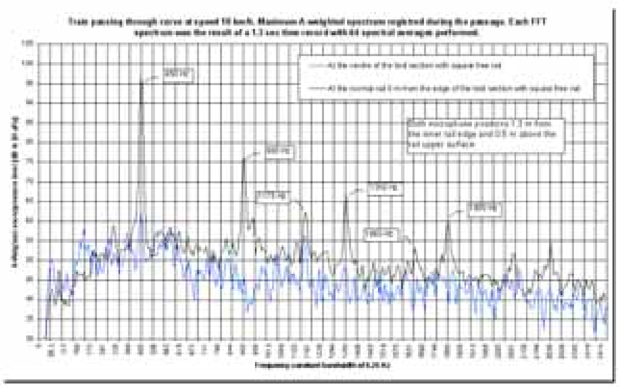


**Fig. 3.3.4. : The two microphone positions according to Fig. 3.3.3 are registered simultaneously during a train passage. FFT were performed on the signals at time intervals of 1.3 s, which produced 64 spectral averages. The frequency resolution is 6,25 Hz. The maximum spectrum produced during the whole passage for each microphone position is presented.**

characteristics is that the best solution is the Tribaloy treated rail. It leads to a lower and more stable friction curve than the other solutions, together with a minimal wear after fatigue testing. However, curving tests on a rail treated with Tribaloy led to the conclusion that the hardsurfaced rail could not be used on a real track, due to its too high brittleness. The research in this area is continued in the ongoing EC Research Project called Novastar.

## Results and conclusions

Almost all recent publications on squeal agree on the fact that the lateral stick-slip is predominant in generating squeal. The results obtained in this project have shown that the longitudinal stick-slip is also an important generator of squeal, and that it is not sufficient to work only on the lateral crabbing of the wheel across the top of the rail.

It is generally agreed that squeal is governed by the loudspeaker mode of the wheel. In this project this assumption was not confirmed. In the studied test cases, squeal frequencies were due to the first of a series of axial bending eigenmodes of the wheel surface and tread. Comparison between calculations and measurements of squeal frequencies and noise levels showed a good correlation, leading to the conclusion that the numerical models, after tuning, can be used with confidence.
A saddle profile put on the railhead was found to reduce the squeal sound pressure level by about 35-37 dB (A) for the fundamental squeal frequency and 25-30 dB (A) for its first harmonics. It is to be noted that this added profile on a rubber mat has an effect both in lateral and in longitudinal directions. The tested prototype was in aluminium. Its use on regular tracks would necessitate it to be in steel, i.e. further investigation is required. The tested damped rail fasteners have an excellent anti-vibration performance :
a reduction of the structure borne noise of about 25 dB (A)), and a vibration reduction of 20 dB in nearby buildings, i.e. nearly the performance of a floating slab. The fastener was not effective for squeal noise reduction since it only acts in transversal direction and it does not prevent the longitudinal slip. Rail hardsurfacing is a satisfactory solution but it needs further developments. The project goals were met. In the very near future, systems developed during this research project will be available on the market.

## Acknowledgements

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