THE EFFECT OF DAMPING THE WHEELS AND VARYING WHEEL/RAIL FRICTION COEFFICIENT ON RAILWAY NOISE

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ABSTRACT
The main noise source when a rail vehicle is circulating under approximately 280 km/h is the interaction between wheel and rail. This paper presents a summary of different solutions that could be envisaged to reduce the level of this source. Being the wheel the most critical component for noise emission, particular attention is being paid to existing solutions to add damping to the wheel. A wheel including a friction damping solution—a modified ring damper—is analysed in detail. Ring dampers are steel rings fitted inside grooves cut under the wheel rim, usually one at each side of the tread; in this case it is a heavier ring at one side of the wheel. The paper includes some experimental results obtained in the laboratory aiming at verifying the modal damping for the different modes with this “damped wheel” and some on track test for a global evaluation of the solution. Additionally, an analysis of the effectiveness of modifying the friction coefficient between the rail and the wheel is also reported. A “friction modifier” is added at the top of the rail and noise and vibration measurements have been carried out on the track for an experimental evaluation of its effectiveness.

RAILWAY NOISE
Introduction
The main railway noise sources are traction noise, rolling noise and aerodynamic noise. For a train running over 40 km/h and under 280 km/h approximately, rolling noise is the main noise source, i.e. wheel/rail contact phenomena generates a vibration of the wheel, the track and the sleeper. Figure 1, taken from reference [1] illustrates by an example the relative contribution of each of these elements to the total noise. This example of relative contribution should not be generalised as it depends strongly on the dynamic and geometrical properties (mainly roughness) of the different elements but gives a general idea that the wheel is the most contributor at high frequencies, whereas the sleeper is responsible for the low frequency noise.

![Figure 1. Spectrum of the overall sound emitted by a particular case typical of freight traffic broken down into the contributions from the various components, taken from ref [1]](image-url)
The excitation levels generated at the wheel/rail interface during rolling result from a complex combination of mechanisms associated to existing contact forces, roughness of the rail and wheel, local geometrical defects and vehicle speed. Railway noise continues to be a particular interesting research subject to assure the greening of surface transport. Being so several research projects have been financed in EU Research Framework Programs: as an example SILENT FREIGHT, SILENT TRACK. More recently the METARAIL project proposed different techniques to evaluate the partial contribution of the elements involved in the rolling noise source. STAIRRS developed a decision support tool to assess cost effectiveness of different noise mitigation options applied to railways. Results of these projects have been partially included in ISO 3095 (pr EN ISO 3095).

**Roughness effects**
The combined existing roughness of the wheel and the rail is the critical parameter for the rolling noise source. It is clear that grinding the rail and turning the wheel ensures a minimal noise generation. This implies for the operator and track managers an appropriate maintenance of both wheels and tracks and consequently an adequate strategy to optimize wheel and track life. Wheels can be machined in specific workshop lathes, whilst the rail is ground using grinding trains to deal with rail faults on the open track or tunnels, adjusted to varying conditions thus ensuring that, whatever the conditions, the necessary material is removed from the rail efficiently.

Table I summarises some noise measurements which were carried out before and after grinding a curve (average rail corrugation was 0.089 mm before grinding). The comparison is made with the same train, at the same speed and measurements were taken with a difference of only a few days, thus roughness growth should be unnoticeable. It is also interesting to note the differences between noise levels of train 1 and train 2, which show that roughness of the wheel has to be taken also in account.

<table>
<thead>
<tr>
<th></th>
<th>Before grinding</th>
<th>After grinding</th>
<th>Reduction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Train 1</td>
<td>$L_{eq}$ dB(A)</td>
<td>92.5</td>
<td>81.4</td>
</tr>
<tr>
<td></td>
<td>$L_{Amax}$ dB(A)</td>
<td>99.2</td>
<td>87.5</td>
</tr>
<tr>
<td>Train 2</td>
<td>$L_{eq}$ dB(A)</td>
<td>96.8</td>
<td>83.4</td>
</tr>
<tr>
<td></td>
<td>$L_{Amax}$ dB(A)</td>
<td>102.2</td>
<td>90.5</td>
</tr>
</tbody>
</table>

**NOISE OPTIMISED WHEELS**
An attempt to classify existing solutions
Smoothing the wheels and rails is the most efficient choice for reducing rolling noise. Additional benefits can be obtained with an optimised design of the rail and the wheel and specific solutions have been developed for wheels and rails. If we concentrate on wheels, the low noise wheels developed to date could be classified in four groups: resilient wheels, damped wheels, optimised wheels, and specially treaded wheels.

**Resilient wheels** are wheels in which the metal tyre is structurally isolated from the wheel hub, generally by an elastomeric material. Reference [2] provides measured levels for trams equipped with resilient wheels, for different tram and track types. **Damped wheels** are traditional Monobloc wheels equipped with an external damping mechanism. There are several methods to increase wheel damping: dynamic absorbers, ring dampers and viscoelastic constrained layers. In reference [3] the effect of dynamic absorbers and ring dampers on squeal noise is analysed and reference [4] describes the ring damped wheel and the constrained layer damped wheel as methods to reduce squeal noise.

Wheel noise emission can also be reduced by changes in the wheel geometry such as: reducing the wheel radius, making the web straight, positioning the centre of gravity of the wheel tyre in the “middle” plane of the wheel. In reference [5] a design optimisation strategy is described. Simulation results are given: a gain of 10 dB(A) is predicted, although measurements on real scale prototypes report only a 5 dB(A) gain. **Specially treaded wheels** reduce rolling noise through an increase in the contact area and a decrease in the contact stiffness between wheel and rail. Reference [6] describes these wheels and gives data on achievable noise
reductions and test results. In reference [7] the thin treaded wheel is said to be the best way to reduce rolling noise, together with wheel and rail surface smoothing.

**Ring Damped wheels**

Conventional ring dampers are steel rings fitted inside grooves cut under the wheel rim, usually one at each side of the tread, as shown in Figure 3-(a). As the wheel vibrates there is a relative displacement between the wheel and the ring. The friction forces that act at the interface introduce a damping effect. Wheel vibration modes can be classified as axial or bending modes and radial modes. The formers are classified according to the number of nodal diameters and the number of nodal circles, whereas the latter are classified according to the number of nodal diameters alone. Typically, the modes involved in squeal noise are the zero nodal circle axial modes (0Ln), while the modes responsible for rolling noise are the 1 nodal circle axial (1Ln) and radial (Rn) modes. The index n is the number of nodal diameters. Figure 2 provides a comparison of two wheels (a reference wheel and the same wheel fitted with two ring dampers), the damping effect of the ring can be clearly seen in the reduction of the peak amplitude of bending modes.

![Figure 2.-Comparison of the acceleration response to a lateral impact for two wheels (a reference wheel and the same wheel fitted with two ring dampers)](image)

Results from previous work [8] and from the theoretical analysis of the wheel/ring system [9] showed that if the ring mass could be significantly increased, sound pressure level (SPL) reductions greater than those achieved by conventional ring dampers would be obtained. Within the Brite-Euram project “Silent Freight” a new ring damper solution, which will be termed CAF damper [10], has been studied and is shown in Figure 3-(b).

![Figure 3.- Detail of ring dampers. (a) Conventional, (b) CAF](image)

**Track noise measurements**

Pass-by noise measurements were made during tests carried out at FGC (Ferrocarrils de la Generalitat de Catalunya). Two different units, manufactured by CAF, were used: one had undamped solid wheels and the other was equipped with CAF dampers. These units are used in a metro line which has a 1 m gage track. The aims of these tests were to assess the effect of the CAF damper on squeal noise reduction. The measurements were made in a tunnel section and the curve had a radius of 150 m and no corrugation. The microphone was located at the inner side of the curve, 2.5 m apart from the side of the inner rail and 0.2 m over the rail-head.
The maximum SPL octave band spectra measured in the non-corrugated track are shown in Figure 4. The SPL measured for the damped unit is smaller than that measured for the undamped unit above 1000 Hz the reduction being at least 10 dB(A) and as high as 17 dB(A) for the 4000 Hz octave band.

Figure 4: Octave band SPL on a non-corrugated curve. Speed=25 Km/h. 4 averages.

Although CAF dampers were mainly developed for squeal noise reduction, the effect on rolling noise was checked by making pass-by noise measurements on a straight track with the undamped and the damped units. The octave band curves of the maximum SPL measured at a speed of 50 Km/h are shown in Figure 5. The SPL measured on a straight track is somewhat reduced if CAF ring-dampers are used. The maximum levels are smaller for all but the 1000 Hz band, the maximum reduction being about 5 dB(A) at 8000 Hz.

Figure 5: Octave band SPL on a straight track. Speed=50Km/h. 4 averages.

**FRICTION MODIFIERS**

**Description of the principle**

The idea of controlling friction is not new. For example, grease lubricant has been used to prevent wheel flange wear, and sand has been sprayed on the rail to increase friction coefficient when higher traction forces were needed. However, although varying friction conditions can reduce contact forces, this can also generate problems when required traction forces are not enough for climbing slopes, or for instance, for braking or accelerating near a station.

Friction modifiers have been used in the past to prevent or to reduce problems related to wheel-rail contact, such as squeal noise and wear. In [11] Eadie explains the behaviour of a friction modifier –HPF- by means of the traction-creepage relationship. HPF belongs to a family of products that provide a controlled coefficient of friction at the wheel-rail contact. They make possible to achieve a specific friction range, and this is their main difference when compared to ordinary grease. Apart from the positive friction characteristic, the HPF controls the friction level at the top of rail, reducing the friction to a controlled lower level without affecting traction and braking. This reduces tangential forces, which are directly involved in the wear damage mechanism. Therefore, the effect of HPF is double. First, it reduces the friction between wheel and rail and second, changes the friction characteristics from negative to positive.

Originally, HPF was not thought as a solution for rail corrugation. It was successfully used mainly to reduce noise, in particular squeal noise. It was also applied to diminish lateral forces and energy dissipation. But its efficiency in reducing the growth of rail corrugation has also been reported [12], [13].
Track noise and acceleration measurements

The effectiveness of friction modifiers are illustrated in this section. Firstly some track tests are reported in non-corrugated curve track. It was a tunnel section and the curve had a radius of 150 m. The microphone was located at the inner side of the curve, 2.5 m apart from the side of the inner rail and 0.2 m over the rail-head. The maximum SPL octave band spectra measured are shown in Figure 6. One can see that in this case, the 250 Hz octave band is not dominant, which means that no corrugation noise was present. The SPL measured for the damped unit is smaller than that measured for the undamped unit above 1000 Hz the reduction being at least 10 dB(A) and as high as 17 dB(A) for the 4000 Hz octave band. The HPF decreases noise levels in the whole frequency range, the reduction being 10 dB(A) at 500 Hz and at least 20 dB(A) above 1000 Hz. If both CAF dampers and HPF are used no additional reduction is achieved compared to the HPF alone.

![Figure 6: Octave band SPL on a non-corrugated curve. Speed=25 Km/h. 4 averages](image)

The measurement of rail accelerations gives a good reference to estimate the influence of the friction modifier on the wheel/rail interface; it is even a more direct indicator than noise. In what follows an extensive number of tests are shortly summarised. The results given next are based on two different tested curve sections having the same length and radius. In the following discussion, the inner rail of both sections are labelled track 1 and track 2.

The first test performed was based only on track 1 before grinding it (the mean corrugation amplitude was 0.16 mm). Its purpose was to compare accelerations produced in this track with and without HPF. For this test, the HPF was manually applied along the inner rail. The results are shown in Figure 7 were the reduction on the acceleration level when using HPF can be easily observed.

![Figure 7.- Rms vertical acceleration profile with (dark line) and without HPF on the same rail with same corrugation (track 1). Vehicle speed = 63 km/h](image)

In a second test, both tracks were ground. The HPF was applied on track 1 by a trackside applicator. Nothing was applied on track 2. Accelerations were measured and the results of track 1 and track 2 are compared. Figure 8 shows the results of the computed acceleration along the lateral and vertical directions, respectively. In this figure, the maximum value of the rms profile is displayed for each pass-by train versus the speed of the corresponding pass-by train. Notice that the effect that HPF has on the reduction of the acceleration values is substantial.
Figure 8.- Vertical (left) and lateral (right) rms maximum accelerations in the mid span; rail without corrugation

FINAL COMMENTS

The paper has illustrated with some examples the effectiveness of different commonly used solutions for the reduction of rolling noise. It is clear that smoothing the rails and the wheels by an appropriate maintenance program provides excellent results although establishing such a program is far from an easy task. The use of friction modifiers to avoid stick/slip phenomena is a good decision particularly for critical track sections having small curves or crossing particularly problematic areas. Furthermore the use of noise optimised wheels provides an additional gain. The particular wheel to be chosen may be one or another based on vehicle and exploitation characteristics. However friction damping solutions have shown a particular good performance with a relatively low cost, and could even be retrofitted to already existing wheels. The practical results described in the paper are far from a complete survey of noise solutions but at least could clarify the reasons of the effectiveness of some of them and why several aspects still remain under investigation.

References: