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DEVELOPMENT AND TESTING OF WHEELS AND TRACK COMPONENTS FOR REDUCED ROLLING NOISE FROM FREIGHT TRAINS

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1. INTRODUCTION

Railway rolling noise comprises components radiated by the sleeper (at the lower end of the audible range), the rail and wheel (for frequencies from about 2000Hz to 5000Hz). The excitation of these components arises from the combined surface roughnesses of the wheel and rail at their contact during rolling. The finite length of the contact patch acts as a mechanical filter resulting in a progressive attenuation of the excitation at higher frequencies.

The OF WHAT project (Optimised Freight WHeel And Track) was set up by Committee C163 of ERRI with the objective of using calculations and prototype tests, to define the acoustic specifications of wheels and track components for freight traffic and thereby to establish a catalogue of optimised solutions for freight wheels on an optimised track.

The project team specified prototype wheels and track components on the basis of optimisation studies carried out using the TWINS model for rolling noise generation [1]. The specifications were then made the subject of a competitive invitation to tender. The need for the prototypes to be acceptable for implementation in the railway industry was considered in the specifications but was considered to be subordinate to the need to demonstrate, if possible, the noise reductions which could be achieved compared to a chosen reference wheel and track.

The components developed and manufactured for the project were:-

- rail pad with optimised stiffness
- rail pad with optimised stiffness and high material damping
- a dynamic absorber to be fixed to the rail (1 pair per sleeper bay) to increase damping of propagating rail vibration

- an optimised wheel shape of diameter 860mm
- a standard UIC freight wheel (920mm diameter) with dynamic absorbers fitted

In addition to these an existing small diameter wheel design (diameter 640mm) was also tested.

Three contiguous sections of test track were constructed at the CD test facility at Velim in the Czech Republic and running tests on this track were conducted in August 1995 using a train comprising 5 wagons fitted with the test wheels. The tests were controlled using a reference section of track and a wagon fitted with the UIC standard 920mm diameter freight wheels. The reference track comprised UIC 60 rail on bibloc sleepers with the standard rail pad supplied with the sleepers for the 'nabla' fastening system.

2.DEVELOPMENT OF SPECIFICATIONS

At the start of the project a study was commissioned which used the TWINS model to assess the possible benefits of the acoustically optimised components. In previous tests that were carried out to validate the TWINS model[2, 3], the *rail pad stiffness* had not been controlled. The OF WHAT project was therefore the first attempt to examine the effect of this parameter.

The specifications for the tests were focussed on the acoustic requirements. However, in the event of using any of the concepts in production components, the issues of their acceptability for industrial implementation would have to be resolved by individual railway authorities and manufacturers taking into account the full range of non-acoustic parameters.

Optimised rail pads and dynamic absorber

The theoretical work [4] has found that a way to reduce the sound power from the track is to use rail pads with a dynamic stiffness designed to balance the vibration of the rails and the sleepers such that the total sound power from these two components is minimised. The sound power from the rail may also be attenuated by reducing its effective radiating length, that is, by increasing the decay of vibration amplitude with distance. This is a function of the pad's stiffness and its damping. The damping of the rail vibration may also be increased using tuned absorbers.

Two configurations of the rail pad were specified, both with the same dynamic stiffness specification at 100Hz under loaded conditions. The required stiffness was in the range $5\times10^8 Nm^{-1}$ to $15\times10^8 Nm^{-1}$ and was compared to the $1.3\times10^8 Nm^{-1}$ measured value for the reference pads at 20°C. The 'Configuration 1' pads were to have the normal level of material damping (loss factor from 0.1 to 0.3). The 'Configuration 2' pads were to have a loss factor greater than 0.4.

A rail-mounted dynamic absorber was also specified to act over the frequency range 600Hz to 2800Hz.

Shape Optimised Wheels (860T wheel)

A number of wheel geometry parameters influence the wheel noise radiation. The basis for the design is to minimise the axial vibration of the web relative to the tread radial vibration. For the one nodal circle axial modes, increasing the symmetry of the wheel profile decreases the excitation of axial motion and reduces the dependence of this on the wheel/rail contact position. The optimum behaviour of the wheel design depends on the position of the contact point. Taking the expected contact position into account, a specific form of wheel design has emerged from the studies and this was chosen for testing. It comprises a thick web and a diameter of 860mm (centre of tread).

Wheels fitted with dynamic absorbers (920 ABS wheel)

The basis of the design specification for these wheels derived from theoretical studies that showed that most of the sound power radiated by the wheel is related to the response of the wheel modes which exhibit a significant radial vibration amplitude at the contact point. These modes are of two kinds:

(1) the radial modes (of differing numbers of nodal diameters) These have a high radial amplitude of the wheel tread coupled with a significant axial amplitude of the wheel web.

(2) the axial modes with one nodal circle which exhibit a rocking of the tread together with a bending of the web.

The theoretical investigations have shown that a significant sound power reduction of the wheel could be achieved by the implementation of two sets of dynamic absorbers acting at 1720Hz and 2330Hz.

Small Wheels (640 wheel)

As an addition to the prototype wheels built especially for the project, commercially available wagons fitted with very small diameter wheels were also included in the tests. The wheels have a 640mm diameter and a very thick, straight web. Theoretical studies have shown that such wheels can be expected to provide a substantial (> 10dB) reduction in radiated wheel noise.

3. TEST PROCEDURE

Measurements were commissioned by the project on four types of rail pad that were delivered for possible installation in the test track. These measurements were used in the final choice of two pad types for the two test track sections. Neither of the attempted Configuration 2 pads met the loss factor specification satisfactorily.

The surface roughness on the rail at each track section and on each of the test wheels was carefully measured in order to enable comparison of the different components on the basis of a constant level of roughness.

The noise measurements were made as the averaged level at three microphone positions at 2.5m radius from the rail head.

4. RESULTS

Tables 1 and 2 summarise the results of the analysis for the changes in the individual track and wheel noise contributions to the overall level (shown in shaded boxes). The effect of these changes on the overall noise level is shown in the unshaded boxes. The results have been normalised for the roughnesses measured on the respective wheels and rails but taking account of the contact filtering effect.

		Reference track	Track with Config. 1 pads	Track with Config. 2 pads	Track with Config. 2 pads + rail abs
Wheel		0	-5.4	-4.4	-4.4
920 ref.	0	0	-2.5	-2.3	-2.3
920 ABS	-3.9	-0.9	-4.2	-3.8	-3.8
860T	-1.5	-0.5	-3.1	-3.0	-3.0
640	c. -18	+0.2	-4.3	-3.8	-3.8

Table 1. Changes in noise levels at 60 km/h (dB(A))

		Reference track	Track with Config. 1 pads	Track with Config. 2 pads	Track with Config. 2 pads + rail abs
Wheel		0	-4.7	-5.6	-7.6
920 ref.	0	0	-2.5	-3.0	-3.7
920 ABS	-5.1	-1.0	-4.2	-5.0	-6.2
860T	-0.7	-0.3	-3.0	-3.3	-4.2
640	c. -18	-0.3	-4.3	-5.3	-6.9

Table 2. Changes in noise levels at 100 km/h (dB(A))

5. DISCUSSION

Optimum Stiffness Pad

Both configurations of rail pad achieved significant reductions in the track noise level of 4 to 5dB(A) compared to the reference pad. This reduction is due to the change in balance of the sound powers of the sleeper and the rail through the control, of the pad stiffness. The changes in the sleeper and rail vibration responses were shown in the vibration measurements as well as in the overall sound levels.

The results show that the Configuration 1 pads are closer to the optimum stiffness for the train speed of 60km/h and that the, slightly stiffer, Configuration 2 pads are closer to the optimum stiffness for the 100km/h train speed. The reason for this is the increase in rail radiation relative to the sleeper radiation as the speed changes from 60km/h to 100km/h.

Rail pad damping

The TWINS study predicted that a doubling of the pad loss factor from the, then assumed, value of 0.25 to 0.5 would lead to an additional reduction of the track sound power by approximately 2dB. Although two suppliers, both of whom proved competent to produce pads to a specified dynamic stiffness, were commissioned to produce a highly damped pad, neither manufacturer was able to meet the specified loss factor value of 0.4 or greater with a material that was acceptable in terms of its thermal variation of stiffness or creep deformation. The combination of the fact that the Configuration 2 pads did not exhibit the specified level of damping, together with the small change in noise level predicted, meant that the concept of a highly damped pad could not be tested.

Rail Absorber

A complete set of measurement results for this component was not obtained. However, it was shown that the absorbers reduce the vertical rail vibration significantly in the 1000Hz and 1250Hz one-third octave bands and the lateral rail vibration in the 1000Hz to 5000Hz bands. At the train speed of 60km/h the dominant frequency components of the rail noise are below this frequency and no reduction of the overall noise level can be expected. For the 100km/h train speed, the track component of noise is estimated to be reduced by 2dB, in addition to the reduction achieved by optimising the rail pad stiffness.

Where rail absorbers are used with optimised stiffness pads the track noise comprises equal contributions from the sleepers and the rail. The reduction of the rail component using rail damping, in this situation, can only therefore lead to, at best, a 3dB reduction in the track noise. Where the pad stiffness is below the optimum value a greater reduction in the track noise may be achievable.

Optimised Shape Wheel (860T wheel)

The 1dB(A) reduction in radiation achieved by the 860T wheel is much lower than the predicted value of 4dB(A). Whilst the validity of the experimental result may be questioned because of great uncertainties of the roughness levels on this wheel, it is considered that further investigations are required to account for this anomalous result.

Wheel Dynamic Absorbers (920 ABS wheel)

A 4dB(A) wheel noise reduction was obtained from the wheel fitted with the dynamic absorbers. This improvement is lower than was expected (between 6 - 9 dB(A)) but can be explained by the fact that the absorber did not conform to the performance specification. It is considered therefore that further optimisation would be achievable through development of the absorbers.

Small Diameter Wheels (640 wheel)

On the comparison based on the normalisation of noise level measurements the wheel showed a substantial reduction of the wheel noise component of approximately 18dB.

A fundamental consequence of reducing the wheel diameter is a reduction in the size of the contact patch. This leads directly to a reduced contact filtering effect on the combined wheel and rail roughness resulting in increased track noise. For the diameter of 640mm, this is a 2dB increase. This has led to a small actual increase in the overall noise emission at 60km/h, as the rail noise dominates the overall level. A productive compromise wheel diameter should be obtainable that results in the optimum overall noise reduction.

6. CONCLUSION

A number of options for the reduction of railway rolling noise have arisen out of detailed theoretical studies and have been tested in full scale running tests which involved the construction of a test track and manufacture of prototype wheels. The experimental data obtained clearly demonstrate the effect of the rail pad stiffness in controlling the track noise and also show the development required to obtain the full benefit of the wheel options.

7. REFERENCES

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8. ACKNOWLEDGEMENT

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