inter.noise 2000
The 29th International Congress and Exhibition on Noise Control Engineering
27-30 August 2000, Nice, FRANCE

I-INCE Classification: 4.1

EFFECTS OF RAIL FASTENING ON RAILWAY TRACK NOISE

S. Cox, A. Wang

Pandrol Rail Fastenings Ltd, 63 Station Road, KT15 2AR, Addlestone, United Kingdom

Tel.: +44 1932 834500 / Fax: +44 1932 850858 / Email: s.cox@pandrol.com

Keywords:
RAILWAY, RAILPAD, STIFFNESS, ROLLING

ABSTRACT
This paper describes measurements of deflection, vibration, and noise on three different track forms. The measurements were carried out as part of the Silent Track European project on railway noise at the Velim test centre in the Czech Republic in May and June 1999. Three track sections were tested – a reference track fitted with relatively soft railpads, a section with stiffer railpads, and a section of a new track design incorporating twin block sleepers, rail with a narrow foot, and a new fastening system. The tests showed the new fastening system to be performing satisfactorily from a mechanical and an acoustic point of view. The wayside noise level at 7.5m from the track was about 4 dB(A) lower than for the reference track. That for the track with stiff railpads was rather less than 1 dB(A) below that for the reference track.

1 - BACKGROUND AND SELECTION OF TRACK TEST COMPONENTS
The potential for reducing the component of rolling noise emitted by railway track by tuning its dynamic characteristics has received considerable attention over the past few years. The widely known TWINS software has shown that stiff railpads can reduce wayside noise under some circumstances, and this was confirmed in measurements made with relatively low speed trains as part of the OF-WHAT project [1]. On the other hand measurements on two parallel tracks fitted with pads of widely differing stiffness on the Belgian high speed line found no difference in wayside noise levels [2]. In these circumstances rolling noise from the wheels appears to predominate.

However the effect of changing track parameters on noise emission cannot be considered in isolation. Changes in pad stiffness, for example, may also have an effect on ride quality, track geometry deterioration, and track component wear. They may also influence the rate of growth of roughness on wheels and on the rail, which provides the excitation for rolling noise in the first place. Because of their beneficial influence in these areas, there has been a general trend in Europe towards the use of thicker, lower stiffness railpads in Europe. This was confirmed by a survey carried out at the outset of the Silent Track project. For this reason, a rubber pad with a relatively low stiffness was chosen for the Silent Track reference track.

The dynamic stiffness of this reference railpad at 20°C and 100Hz with small amplitudes of vibration was measured in laboratory tests [3], and was found to be 56 MN/m. A second, stiffer EVA pad was selected for another section of the test track, which had a dynamic stiffness measured in the same way [3] of 430 MN/m. In the event, the track tests took place during very hot weather and it was found that the EVA pad was more sensitive to temperature variation than the reference pad. This is discussed further below.

The concept behind the third track section arose from predictions showing that noise levels could be substantially reduced if the rail foot width could be reduced [4]. Practical considerations determined that a rail foot width of 100mm was the minimum considered for the trial. UIC60 rail had 25mm machined off each side of the foot to produce the required section. Because of its narrow base width, a conventional rail fastener could not be used without the danger of excessive roll under traffic. A modified version of the Pandrol Vanguard rail fastening system was adopted. Here, the rail is supported on rubber wedges acting under its head and on its web. These in turn are supported on side plates, which are braced against shoulders cast into the sleeper. The arrangement allows the rail to be fixed satisfactorily despite its narrow foot. A view of the rail fastening is shown below in Figure 1. The assembly was designed to have a dynamic stiffness close to that of the Silent Track reference track. This test section was built with twin block sleepers, which a separate design exercise had shown to radiate less noise at low frequencies than the monoblock type.

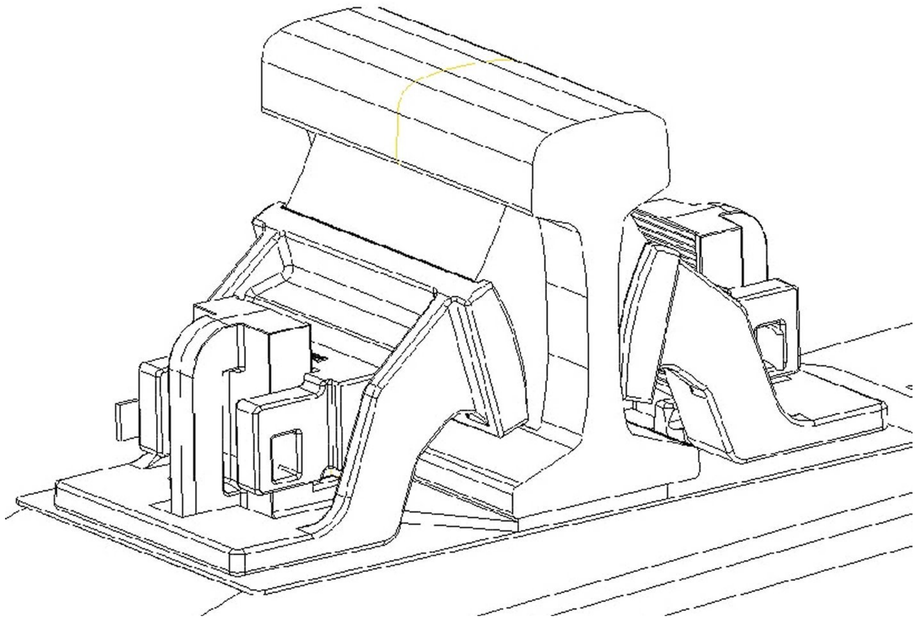

Figure 1: New design track fastener.

2 - TRACK SECTIONS AND MEASUREMENTS

The test track was straight, and on ballasted track, and consisted of:

- "Reference" test section (50 m): conventional track with new UIC60 rail, conventional monoblock concrete sleepers, and relatively soft reference studded rubber railpads.
- Buffer section (25 m): similar to retrofit section (see below).
- "Retrofit" test section (50 m): identical to the reference section but fitted with stiffer studded EVA railpads. So called because reference track could be converted relatively easily by adopting retrofit solutions.
- "New-Design" test section (50 m): consisted of 80 twin-block concrete sleepers to a new design, fitted with the new fasteners and UIC60 rail which had 25mm machined off each side of its foot to reduce the width to 100mm. So called because existing track would require substantial rebuild to accommodate the new design.
- Buffer section (25 m): similar to retrofit section.

The whole of the track except the reference section was initially fitted with tuned absorbers, which constituted another part of the trial. The measurements described below were carried out after these had been removed from the retrofit and new-design sections. Some of the damping material from the absorbers remained on the rail in these sections.

Deflections of the rail relative to the concrete sleeper, acceleration of the rails and sleeper, and noise pressure levels at 7.5 m from the track centre were measured. Track frequency response functions were also measured using an instrumented hammer.

Measurements were made under a test train which ran at an average train speed of around 100 km/hr. It was formed from one locomotive, a lab coach, and six flat wagons. Five different wheelset designs were tested on the otherwise identical flat wagons.

The roughness of the rail, which excites the vibrations which lead to rolling noise, was measured along a number of lines on all track sections. In most cases, the roughness was quite similar. On the new-design track section, the roughness was rather higher than this along one line of measurement, and rather lower along another. Observations of the contact path after train passage showed that wheels appeared to pass along the lines of both these measurements. No corrections have therefore been applied to the measurements reported here.

3 - RESULTS AND DISCUSSION

Deflection measurements of the rail relative to the sleeper were carried out on both rails on each test track section. Deflections were analysed separately for the three axle types – locomotive, lab coach and flat wagon. The vertical deflections of the rail foot on field side and gauge side have been averaged to estimate the vertical deflections of the centre of the rail. The lateral deflection of the rail head was estimated by first calculating rail roll from the difference between these same two measures, then multiplying by an appropriate factor derived from the geometry of the rail section, and finally adding the corresponding lateral deflection of the rail foot.

Examples of the rail deflections are shown in Figure 2. The largest deflections recorded – those under the locomotive are shown. The vertical deflections on the reference track and the track with new fastenings are similar. They are about twice those on the track with stiffer pads. Note that this does not imply a difference by a factor of two in low frequency fastener stiffness, because of the different distributions of load along the track. The track with new fastening system shows small rail head lateral deflections despite the narrow rail foot section and the relatively large vertical deflection. It can be concluded that the new fastener is performing satisfactorily from a mechanical point of view.

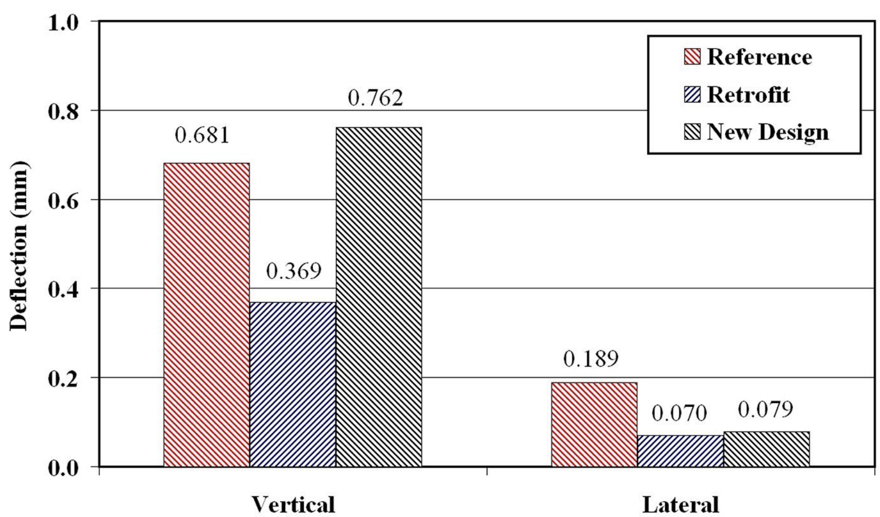

Figure 2: Deflections under locomotive.

Rail and rail seat velocity levels over a frequency range extending up to 2.5 kHz have been averaged across the whole train length and are shown in Figure 3. The rail seat vibration level for both the reference track and for the new design track section is 4.6 dB lower than on the track section with stiffer pads. This illustrates the benefits in terms of track deterioration of low stiffness track fasteners with a low stiffness. Rail vertical vibration level with new design is very similar to that on the track with a stiffer pad, and is about 2.6 dB lower than that with the reference pad. Rail web lateral vibration is about 3 dB lower than that with reference pad track and 1.8 dB lower than the stiffer pad track. Lateral velocity levels are, however, significantly lower than vertical velocities.

Wayside noise was measured at 7.5 m from the track centre and gives a good indication of noise levels likely to be perceived by those alongside the track. Analysis was been carried out separately for the different sections of the train. The averaged sound pressure levels are shown in Figure 4. The noise level for the locomotive and for the whole train are of interest because they are indicative of the differences which might be expected to be found on track where no special measures have been applied to wagons to reduce noise. The noise level on the stiffer pad track is between 0.6 and 0.8 dB lower than that with reference pad. The wayside noise level on the new design track was between 2.9 and 4.4 dB lower than the reference track. As expected, the difference was greatest for the quietest section of the train.

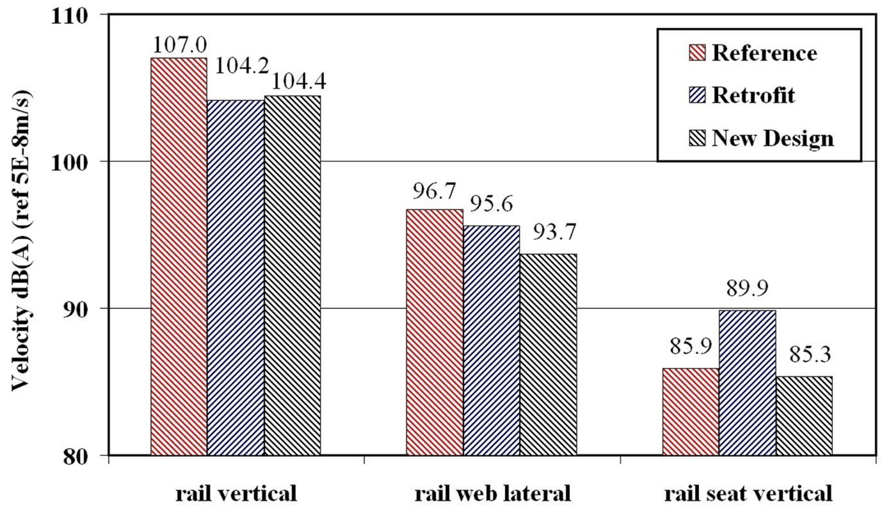

Figure 3: Vibration – rail and sleeper velocity levels.

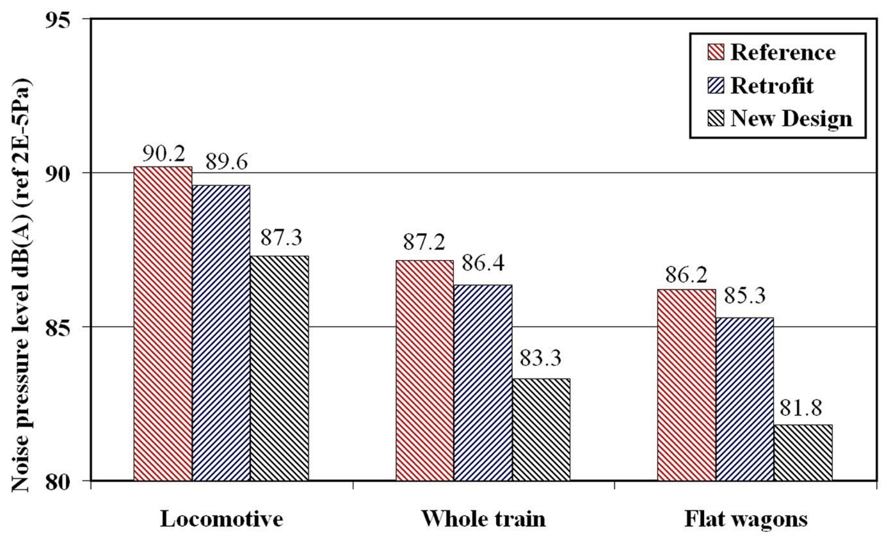

Figure 4: Noise at 7.5m from track centre.

The spectra of wayside noise (not shown) on each track section are all similar in shape and exhibit a broad peak over the frequency range between about 400 Hz and 2 kHz. As Figure 3 shows, vibration levels on the sleeper were much lower than on the rail – this is certainly true for the new-design track. These factors indicate that the reduction in noise level for the new-design track section is likely to be associated with a reduction in radiation from the narrow-foot rail, rather than from the new sleeper design.

The track frequency response functions are useful in estimating the dynamic stiffness of the different components in the track. The frequencies at which the rail and sleeper resonate on the flexibility of the pad were found to occur at about 270 Hz on the reference track, about 450 Hz on the track with stiffer pads, and about 300 Hz on the new design track. Given the lower stiffness values of the sleeper and rail in the new track design compared to the other two track sections, the corresponding stiffness of the fasteners are about 75 kN/mm, 250 kN/mm, and 80 kN/mm respectively. The factor between pad stiffness values on the reference track and on the retrofit track, expected to be about 8 times on the basis of laboratory tests, was therefore only about 3,5 times in practice. The difference can be explained by the different sensitivities of the two materials to temperature, which was retrospectively identified from the laboratory test results [3].

Finally, the responses of individual parts of the new design track system may be of particular interest since this is the first time that such a design has been investigated in track. Accelerations of the side plates, the shoulder, and the sleeper were measured under several passes of the test train, and compared with the accelerations of the rail. The results of these measurements are summarised in Figure 5. The measurements show the efficacy of the fastening system in reducing the level of rail vibration transmitted through into the cast-in shoulder and the railseat of the sleeper. Examination of the corresponding spectra (not shown) shows a peak in the side plate vertical response at a frequency of about 600 Hz. This may be due to bending of the side plate at this frequency.

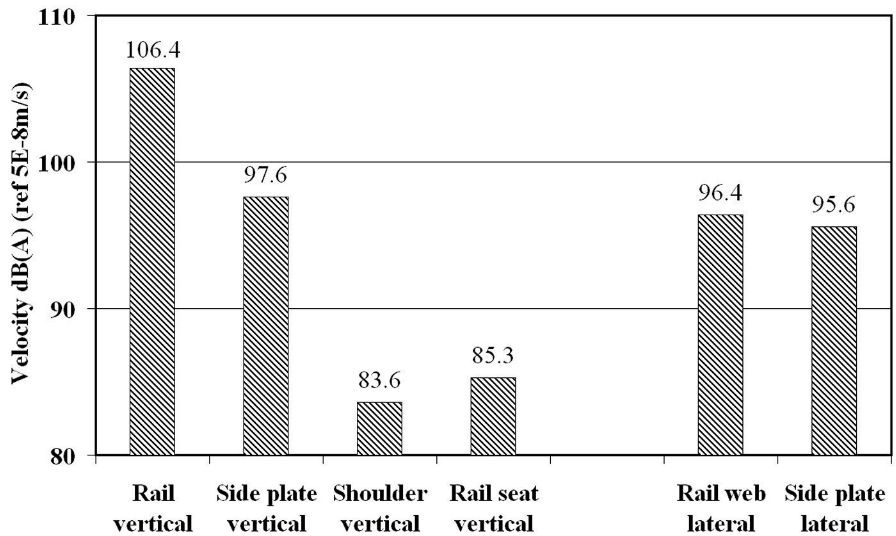

Figure 5: Vibration levels on new design track components.

These findings, from vibration measurements on the new design track under traffic, were supported by measurements of frequency response of the individual components made using an instrumented hammer.

4 - CONCLUSIONS

Measurements of dynamic deflections under traffic during tests indicate that a new design of track fastening is capable of fixing a rail section with a narrow foot. The vertical deflection of the fastener was comparable with that of the test reference track, which is itself representative of modern track forms used in Europe. The measurements showed that new fastening system allowed only small rail head lateral deflections, despite the narrow rail foot section and the relatively large vertical deflection which is required to protect the sleeper and reduce dynamic force transmission.

Vibration and wayside noise measurements for a track section with the new fasteners and narrow rail foot have been compared with those on a reference conventional track section with a relatively soft railpad, and on a similar conventional track section fitted with stiffer railpads. Laboratory measurements prior to the track tests had indicated a factor of about 8 times between the stiffness of the two pads, but high temperatures at the test site and differing sensitivities to temperature reduced this factor to approximately 3.5 times in track when the measurements were made.

The vibration measurements indicate that the rail vertical vibration level with new fastening system was similar to that on the track with a relatively stiff pad, and was about 2.6 dB lower than that on the reference track with a soft pad. Rail seat vibration was 4.6 dB lower than for the track with stiffer pads and similar to that with the reference pad track.

The wayside noise level at 7.5m for the track with stiffer pads was between 0.6 and 0.9 dB lower than for the track with the softer reference pads. The noise level with new design was between 2.9 and 4.4 dB lower than for the reference track.

Accelerations of component parts of the new fastening system were generally much lower than the rail vertical vibration.

REFERENCES

1. **N. Vincent et al**, Theoretical optimisation of track components to reduce rolling noise, *Journal of Sound and Vibration*, Vol. 193(1), pp. 161-171, 1996

2. **S.J. Cox & A. Wang**, Effects of pad stiffness on rail roughness growth and wayside noise levels for high speed track, 1999

3. **VIBRATEC**, *Laboratory characterization of rail pad dynamic properties*, Silent Track Report 81223, 1998

4. **SOGERAIL**, *Effect of rail foot width on the acoustic emission*, Silent Track Report 080198, 1998