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NOISE REDUCTION EFFECTIVENESS OF WHEEL AND RAIL VIBRATION ABSORBERS

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ABSTRACT
The rolling noise reduction effectiveness of wheel and rail vibration absorbers was evaluated at a section of tangent ballasted track with concrete sleepers in Portland, Oregon, USA, as part of Transit Cooperative Research Program Project C3A. Preliminary results of the rolling noise tests are reported here for various combinations of treated and untreated rails and wheels. The wheel and especially the rail vibration absorbers significantly reduced rail vibration at audio frequencies, but had little effect on wayside rolling noise. While the rail vibration absorbers did not reduce the maximum pass-by noise significantly, the "singing rail" noise was eliminated entirely. The "singing rail" vertical vibration transmission spectrum had a pass-band characteristic, as expected for periodic supports. The vibration data indicate that the wheel was the most significant source of noise above 250 Hz.

1 - INTRODUCTION
Wheel and rail vibration absorbers were tested at the Tri-Met system in Portland, Oregon, USA, under the Transit Cooperative Research Program (TCRP). The tests were designed to measure the wheel squeal and rolling noise reduction effectiveness of wheel and rail vibration absorbers and demonstrate their practicality for application to United States light rail transit systems. Some of the preliminary results of these tests are described below with respect to tangent track rolling noise at a section of ballasted track with concrete sleepers. The data indicate the of vibration transmission characteristics of discretely supported rail and the relative significance of the rail and wheel in noise radiation.

Wheel and rail vibration absorbers are attractive noise control treatments for rail transit systems, because they are applied directly to the noise radiating components. A vibration absorber is a spring-mass system tuned to specific frequencies to reduce wheel squeal. The vibration absorbers described here were actually damped spring-mass systems, more accurately described as dynamic absorbers, the theory of which is described by Snowdon [1].

The wheel vibration absorbers tested at Portland consisted of cantilevered steel plates and a constrained layer of elastomer, and were tuned to control lateral tire bending modes at frequencies of 2,600 and 3,800 Hz. The absorbers were bolted to the exterior faces of the Bochum resilient wheel treads. Energy absorption is enhanced at the absorber's tuned resonance frequency or frequencies. The absorbers are tuned to control wheel squeal at tire modal frequencies. The wheel vibration absorber has also been claimed to be effective in reducing wayside rolling noise, which is the subject of this paper.

The rail vibration absorbers tested at Portland were multi-degree-of-freedom absorbers with multiple steel plates and elastomer damping layers. Each of the absorber assemblies included four of these multi-degree-of-freedom absorbers, one clamped to each side of the rail web, and one clamped to the top of the rail foot at each side. These dynamic absorbers were designed to be effective over a broad range of frequencies. Rail vibration absorbers have been claimed to reduce rail vibration, and thus control squeal at curves and rolling noise at tangent track, but have received little or no attention in the United States. The absorbers included a thick steel plate that passed beneath the rail foot and provided a means of clamping the absorber to the rail. The design was intended to allow installation without disturbing the rail, thus simplifying installation, and avoiding problems at systems with 3rd rails. Each rail vibration absorber weighed about 25 kg.

2 - PROCEDURE

The measurements were performed at a section of tangent ballast-and-concrete-sleeper track. The rail was RE115 (115 lb/yard), retained with Pandrol clips and resilient sleeper pads at nominally 760 mm spacing. Sleeper pitch appeared to vary by about plus or minus 25 mm. The rail was recently ground, and exhibited an excellent running surface. However, the gauge corner was not properly finished, leaving a sharp corner. While this might cause excessive noise under certain conditions, the noise produced by the test vehicle with the new Bochum wheels was not excessive. Further, inspection of the gauge corner indicated little wear, suggesting that the wheels were well centered on the rails of this tangent track section, and the noise data obtained from one run to the next were very consistent. Thus, the sharp gauge corner is not believed to be affecting the test results nor contributing to wayside noise. The rail vibration absorbers were mounted on the rail in each sleeper bay. Thus, 240 absorbers were installed on a 90 m section of track.

Two microphones were position at about 10 m feet from the near track center and 1.5 m above the top of the rail, and separated by 15 m. Rail head vertical and horizontal vibration were measured simultaneously at points opposite each microphone with piezo-electric accelerometers mounted on Endevco cementing studs that were glued to the rail with epoxy.

The measurements were conducted both with the rail vibration absorbers in place, and without. For each of these conditions, data were taken with a single test vehicle equipped with and without wheel vibration absorbers.

3 - TEST RESULTS

Wayside 1/3 octave band noise single event level (SEL) spectra measured for various combinations of wheel and rail vibration absorbers are presented in Figure 1. These data are averages over multiple runs and over both microphone locations. Consistency between data was very good. The data for the rail vibration absorbers were collected in July of 1999, and the data for the untreated track were collected in August of 1999. Data were also collected for the untreated westbound track adjacent to the treated eastbound track, but are not shown here. The data shown are time integrated sound energy exposure levels (SEL), with units of decibels relative to 400 $\times 10^{-12}$ Pascal^2-second. The data represent the total noise contribution from the passing train.

These data indicate that the rail vibration absorbers had virtually no effect on the wayside noise level spectrum, except, perhaps, at about 1,250 Hz. At this frequency, the wayside noise with the rail vibration absorbers was slightly higher than without. The spectral peak is believed to be due to a bending resonance of the base plate that was used to clamp the absorber assembly to the rail. The data also indicate that the wheel vibration absorbers had very little effect on wayside noise. The A-weighted SEL and maximum sound levels corresponding to the spectral of Figure 1 are listed in Table 1:

Treatment Configuration	**SEL**	**Maximum Level**
	dBA Re 400E-12 P^{2}s	**dBA Re 20 micro-P**
No Treatment	80	76
Wheel Vibration Absorbers	79	75
Rail Vibration Absorbers	81	78
Rail & Wheel Vibration Absorbers	81	78

Table 1: A-Weighted sound levels.

Corresponding rail 1/3 octave band velocity exposure levels are presented in Figures 2 and 3 for vertical and transverse horizontal rail vibration, respectively. These data are levels in decibels relative to 10^{-12}m^2/sec, and are thus directly related to the wayside SEL. The data are averages over both multiple runs and over both positions. In contrast to the data shown for the wayside noise, the rail vibration velocity spectra were substantially reduced with the rail vibration absorbers. There was a reduction of vertical vibration at frequencies above 630 Hz, and a reduction of horizontal vibration at frequencies above 250 Hz.

The data also indicate that the rail vibration velocity levels were reduced with the wheel vibration absorbers by roughly 3 to 5 dB over much of the spectrum above 630 Hz. This surprising result is not adequately explained, because the wheel vibration absorbers appear to have had little effect on the wayside noise. The difference was greatest without the rail vibration absorbers, but is still present with the rail vibration absorbers. These differences were also observed for train speeds of 40 and 56 kilometers per hour. There is no evidence of other noise sources, such as traction motor fans, contributing to the wayside noise.

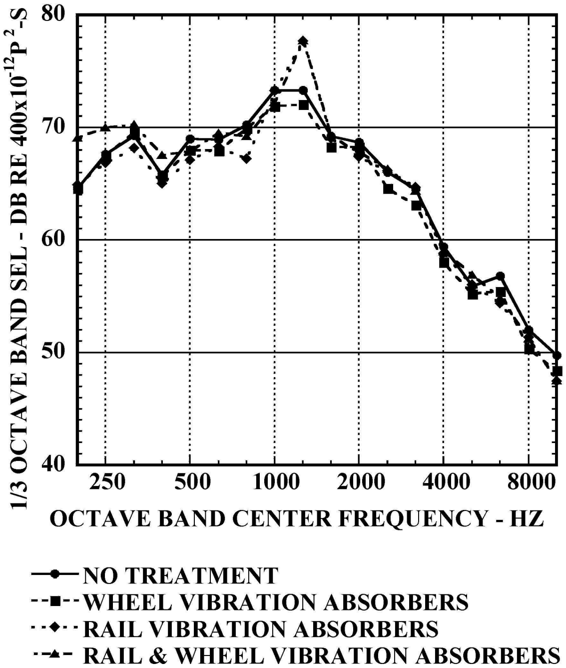

Figure 1: Wayside SEL at 10 m from track center for 72 kph test vehicle.

4 - SINGING RAIL

Noise radiation from untreated track prior to and after train passage can be a significant cause of community reaction, and has been referred to as "singing rail". A similar effect was observed at the tangent track test section in Portland, Oregon. While the rail vibration absorbers did little to reduce the wayside noise, they completely eliminated the singing rail noise prior to and after passage of the test vehicle. The result was dramatic, and resulted in a qualitative improvement of the wayside noise environment by reduction of the time duration of audible train noise.

To investigate this further, additional rail vibration measurements were conducted at the tangent track test section with four accelerometers separated by 380 mm, corresponding to one-half of the sleeper pitch of 760 mm. Two accelerometers were located over the sleepers, and two between the sleepers. The accelerometers were mounted on the underside of the rail head, and were oriented vertically. Data were recorded for several passes of revenue trains without wheel or rail vibration absorbers. The data were analyzed with a Fourier analyzer to obtain auto- and cross-spectral components over a frequency range of 0 to 5,000 Hz.

The cross-spectrum and coherence of rail vertical vibration velocity recorded at two adjacent sleeper bays are presented in Figure 4. The cross-spectrum is in decibels relative to 1E-12 $(m/sec)^2$ with a 20 Hz effective noise bandwidth. The cross spectrum contains a broad peak between 500 and 2,000 Hz. A minor though well defined peak occurs at about 800 Hz, which is approximately the theoretical pinned-pinned mode frequency for the rail. The coherence function is close to unity between about 600 Hz and 1600 Hz, and is significant between about 500 and 2000 Hz.

The transfer function magnitude between vibration data recorded at the two adjacent sleeper bays is presented in Figure 5. The transfer function clearly shows the pass-band for vibration transmission between 500 Hz and 2,000 Hz. The theoretical lower limit of the pass-band is the pinned-pinned mode frequency of the rail in vertical bending, calculated to be 800 Hz. However, the passband extends below the theoretical pinned-pinned frequency of 800 Hz. The phase spectrum corresponding to the transfer function spectrum is plotted in Figure 6, and indicates that the phase angle between the two measurement points is roughly about 180 degrees at 500 Hz. This is opposite-phase condition is consistent with a pinned-pinned mode. There are many possible reasons for the discrepancy, such as variation of sleeper pitch, and contaminating torsional and lateral vibration modes with their own pass-band and stop-band characteristics.

5 - CONCLUSION

Both the rail and wheel vibration absorbers were effective in reducing rail vertical and horizontal vibration, but had little effect on the wayside SEL rolling maximum and SEL noise levels. This result suggests

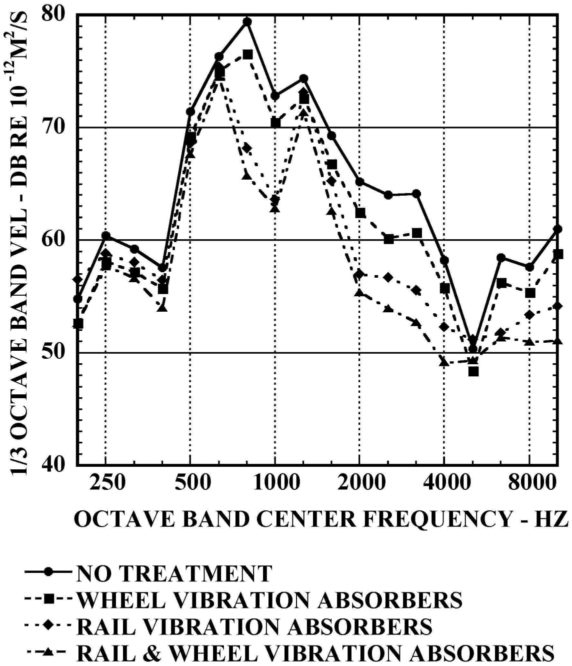

Figure 2: Rail vertical vibration exposure level for 72 kph test vehicle.

that the resilient wheel tire is the most significant noise radiator above 500 Hz during vehicle passage. The insignificant noise reduction obtained with the wheel vibration absorbers is consistent with the lack of significant rolling noise reductions observed by Saurenman, et al, [2] for damped wheels and untreated Bochum resilient wheels relative to solid steel wheels at other U.S. rail transit systems at speeds less than 100 kph. The additional damping provided by the wheel vibration absorbers in addition to the damping provided by the resilient Bochum wheel appears to be insufficient to reduce rolling noise at this section of tangent tack. The rail vibration absorbers did eliminate noise radiation by the rail (singing rail) prior to and after vehicle passage, thus greatly reducing the time duration of audibility of wheel rail noise. This qualitative improvement is not represented by the SEL or maximum level.

The reduction of rail vibration by the rail and wheel vibration absorbers suggests that rail dynamic contact forces are reduced at frequencies that are comparable with those for short-pitch rail corrugation. Further research would be needed to determine the affect of rail and wheel vibration absorbers on rail corrugation. The insignificant change in wayside noise is not consistent with a reduction of contact forces. The use of the energy sum of the rail vibration velocity may be a factor in the analysis, since the rail vibration attenuates rapidly with distance from the source with the rail vibration absorbers in place. Less clear is why the wheel vibration absorbers would reduce rail vibration.

ACKNOWLEDGEMENTS

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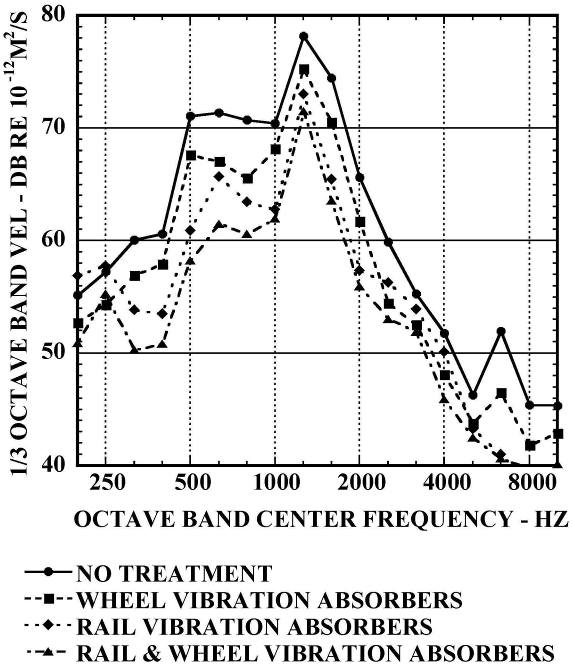

Figure 3: Rail transverse horizontal vibration exposure level for 72 kph test vehicle.

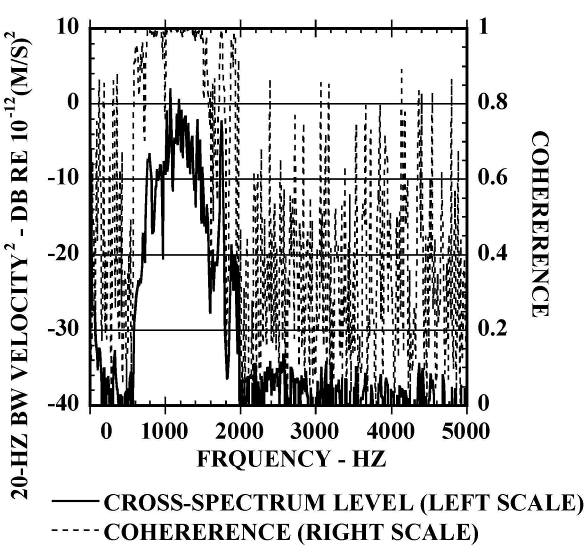

Figure 4: Cross-spectral analysis between rail vibration at adjacent sleeper bays (no treatment).

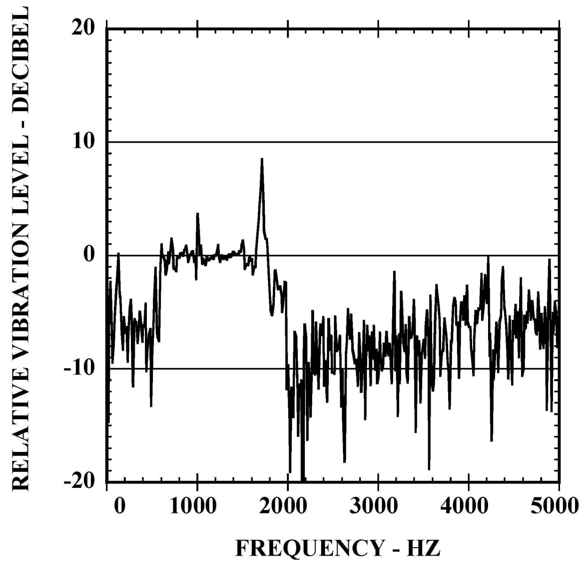

Figure 5: Transfer function between rail vertical vibration at adjacent sleeper bays prior to train passage (no treatment).

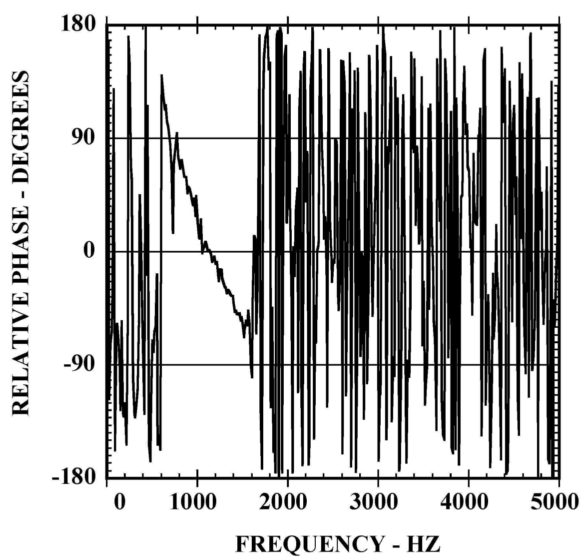

Figure 6: Phase transfer function between rail vertical vibration at adjacent sleeper bays prior to train passage (no treatment).