Force Density Measurements at Sound Transit

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ABSTRACT

The environmental assessment of ground vibration caused by rail transportation systems is becoming more detailed as new transit and commuter rail systems approach or encroach on sensitive manufacturing and laboratory research activities. Methods for measuring low frequency vehicle/track forces were developed to improve predictions of vibration in Seattle and identify appropriate vibration control strategies. These methods expand on the methodology of ground vibration prediction used by the U.S. Federal Transit Administration, based on the concept of force density level and line source response. Vehicle/track force density measurements were conducted in tunnels using sinusoidal excitation and statistical tests of significance to improved estimates. The methods were used to overcome the low vibration response of the tunnel structure at low frequencies in over-consolidated soils. The vibration data indicate the effect of track curvature, axle spacing, and profile grinding.

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1. INTRODUCTION

Sound Transit is planning a twin bored tunnel through the University of Washington campus in Seattle, Washington, USA. Ground vibration interference with sensitive university research is of great concern to both the University and Sound Transit. The project is currently in final design, and prototype testing of vibration control provisions will be underway in 2014 and 2015.

The prediction of ground vibration was based on methods outlined in the U.S. Federal Transit Administration guide[1] (3), relying on concepts of force density level (FDL) and line source response (LSR). The force density levels were measured for the Sound Transit light rail vehicle on direct fixation track in tunnel, using LSR's obtained with both a pneumatic hammer and an electro-dynamic shaker, and tunnel structure vibration during train passage. Ground vibration at frequencies below 10Hz were difficult to detect due to ambient vibration, and Student's T tests of significance were employed to subtract out ambient vibration energy from the vibration energy measured during train passage. Rail undulation measurements were undertaken to characterize the condition of the rail and potential to generate low frequency vibration at frequencies below 10Hz. These efforts and representative results are discussed below.

2. TEST CONDITIONS

The measurements were conducted in the Beacon Hill tunnel consisting of twin bored tunnels with a grouted pre-cast concrete segmental liner. The tunnel is similar to that planned for the University of Washington campus, including a similar curve and grade. The alignment, showing the curved sections of track, is shown in Figure 1.

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The track section consists of RE115 rail of lineal density 57kg/m supported on direct fixation fasteners of vertical stiffness approximately 25KN/mm at 750mm pitch, as is typical in the North America. The track gauge is standard at 1435mm.

The geological materials surrounding the tunnel consist of over-consolidated glacial tills and sands, with shear wave velocity of approximately 450m/s, similar to that found on the University of Washington campus.

The vehicle used by Sound Transit is a 70% low floor vehicle with a motored bogie at either end and an idler bogie with independently rotating wheels at the center of a three-section car body. The center section of the vehicle is essentially fixed to the center bogie. Resilient wheels of diameter 660mm are employed throughout. The test trains included three and four-car consists.

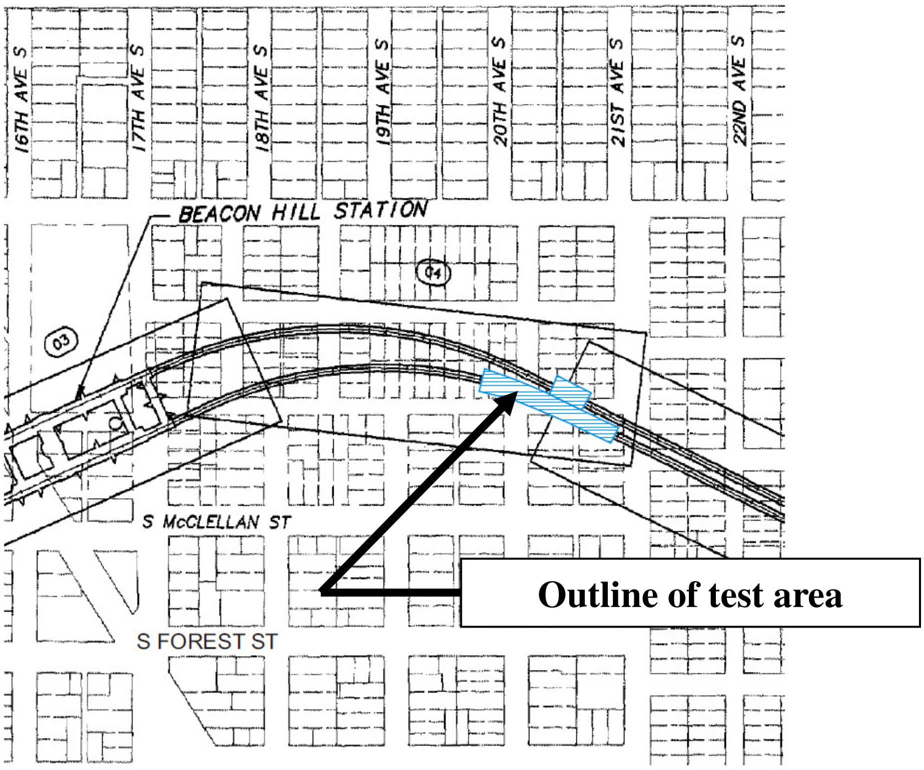

Figure 1. Tunnel Alignment At Beacon Hill

3. LSR MEASUREMENT

The LSR between the tunnel invert and tunnel bench was measured with two procedures. The first procedure included and electro-dynamic shaker, and the second included a pneumatic hammer. Figure 2 is a schematic of the tunnel testing arrangement.

3.1 Shaker

The shaker used for testing produced about 135N of force. The shaker was placed at positions along the safety walk and tuned to discrete 1/3 octave band center frequencies from 3.16 to 20Hz. Velocity transducers were positioned in the invert of the tunnel and spaced approximately 4.57m apart. The transfer mobilities at the discrete frequencies were constructed from the cross-spectral and auto-spectral components obtained by Fourier transformation of force and velocity with a custom multi-channel data analysis package (LXTRANS). The third octave point source response (PSR) between each source receiver pair was constructed from a continuous transfer mobility function that was obtained by cubic-spline interpolation of the discrete mobilities. The LSR was obtained for each shaker position by cubic-spline interpolation of the PSR over the invert positions and trapezoidal integration of the result. By the principal of reciprocity, the LSR thus obtained represents the LSR from the invert to the shaker position.

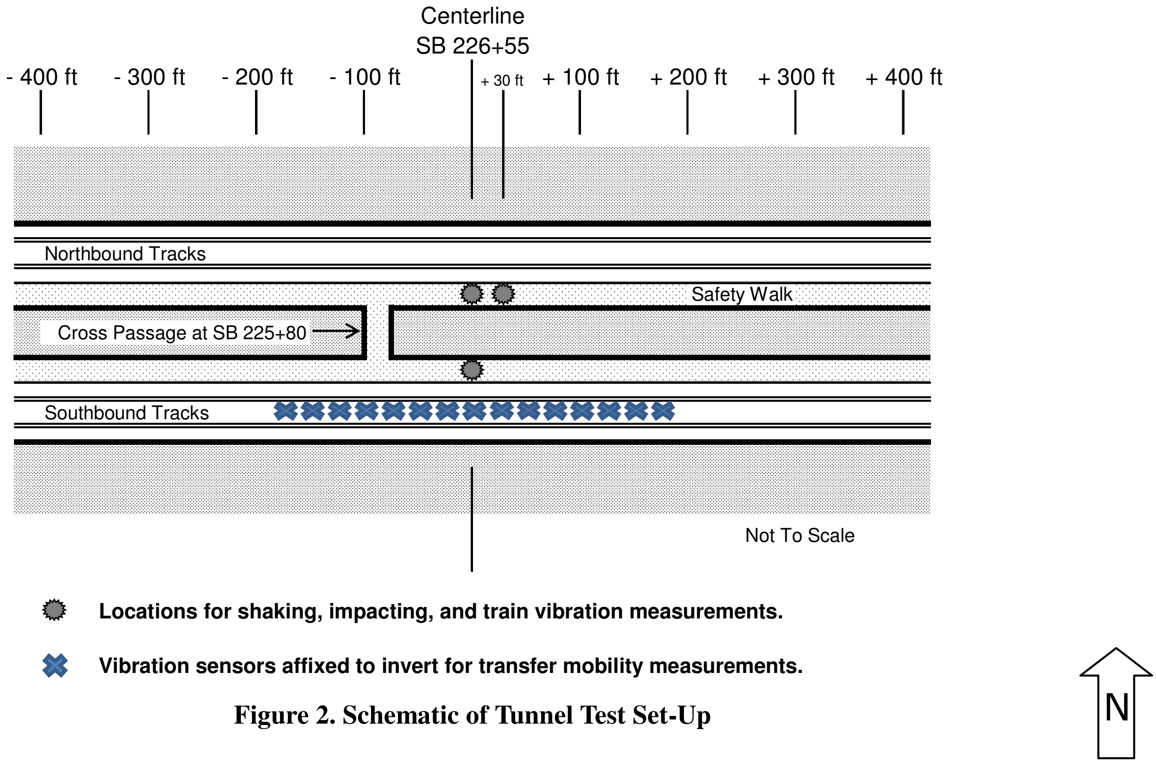

Figure 2. Schematic of Tunnel Test Set-Up

3.2 Pneumatic Hammer Tests

Transfer mobilities were measured with a pneumatic hammer to extend the frequency range up to 160Hz. The pneumatic hammer consists of a 12.2kg cylindrical mass guided by a 100mm diameter tube with pneumatic assist to both raise the hammer and drive the hammer downward onto a load cell. A 25mm thick rubber pad was positioned between the 300mm diameter aluminum load cell base and the safety walk. The transfer mobilities thus obtained were continuous in the frequency domain, and the point source responses were obtained by convolution of the mobility magnitude with the third octave band filter response and energy summing over the spectrum. Spline interpolation was applied to the third octave PSR's and the mobility magnitude squared was integrated over the length of the train to obtain the LSR.

3.3 Comparison of LSR's

The LSR's obtained with the shaker and pneumatic hammer at one of the measurement points are compared in Figure 3. The agreement between the two methods was quite good at all three locations at frequencies of 10Hz and above. Below 10Hz, the LSR included noise energy, and was unreliable. The LSR obtained with the shaker was thus used for frequencies below 10Hz, and the LSR obtained with the pneumatic hammer was used for frequencies above 20Hz. The LSR's were combined for frequencies of 10Hz to 20Hz.

The shaker could have been used for frequencies up to approximately 50Hz, the upper limit of the electro-dynamic shaker's range. However, this was not done due to limited time in the tunnel, as testing had to be completed during non-revenue-service hours. Secondly, the wavelength in the soil was about 9m, so that greater variation might be expected between third octave band frequencies, thus confusing the interpolation, although this may not be the case as indicated by the smoothness of the LSR at frequencies above 20Hz. One may consider swept sine testing at frequencies extending from 10Hz up to the upper limit as well. The main advantage of the sine-dwell approach is the enhancement of the signal-to-noise ratio with by cross-spectral analysis with very narrow spectral resolution. At the long wavelengths involved in propagation through the soil and structure, the real and imaginary parts were found to vary smoothly in the complex plane.

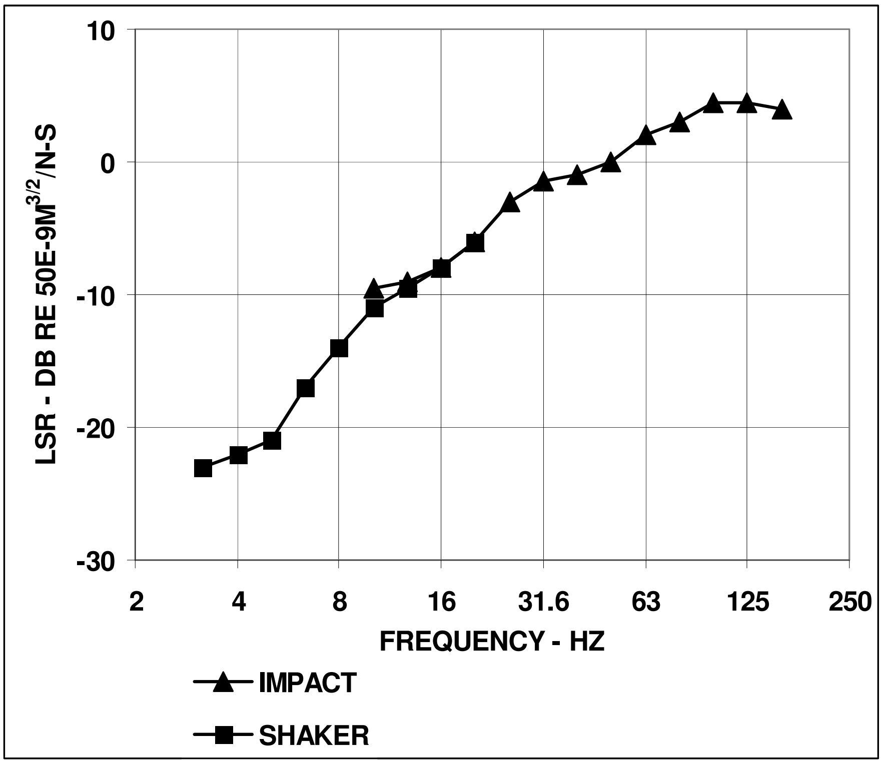

Figure 3. Comparison of Impact and Shaker LSR Test Result

4. TUNNEL STRUCTURE VIBRATION

Tunnel structure vibration during train passage was measured at the shaker test locations on the safety-walk for trains consisting of three and four cars. Train speeds ranged from 40km/h to 64km/h in 5km/h increments, with multiple runs at each speed. The vibration levels in decibels were averaged to determine the mean and standard deviation. The background vibration immediately preceding the train passage was measured, and similarly averaged. A Student's T-test was used to determine significance of differences between vibration with the train and vibration without the train. Where significant, the background vibration energy was subtracted from the train vibration energy to obtain the energy-mean vibration velocity at each location, speed. Where the significance test indicated no significant difference between ambient and train, 10 dB was subtracted from the ambient level to represent the train vibration velocity level. Representative spectra are plotted in Figure 4.

5. FORCE DENSITY ESTIMATION

The FDL at each measurement location was obtained by subtracting the measured LSR from the vibration velocity energy-mean levels for each speed. The resulting FDLs were energy-averaged over measurement locations to obtain the energy mean FDL at each train speed. The results are illustrated in Figure 5 for speeds of 40km/h up to 64km/h. Substantial variation occurred with train speed, as indicated by the emergence of strong peaks at about 8 to 10Hz and at 20 to 25Hz for speeds above about 40 to 48km/h. The peaks at 8 to 10Hz are in the neighborhood of the primary suspension resonance frequency, while the peaks at 20 to 25Hz appear to be produced by the center bogie with independently rotating wheels. Measurements of fleet vibration indicate that the peaks at 20 to 25Hz are not produced by all vehicles, and may be related to tread wear.

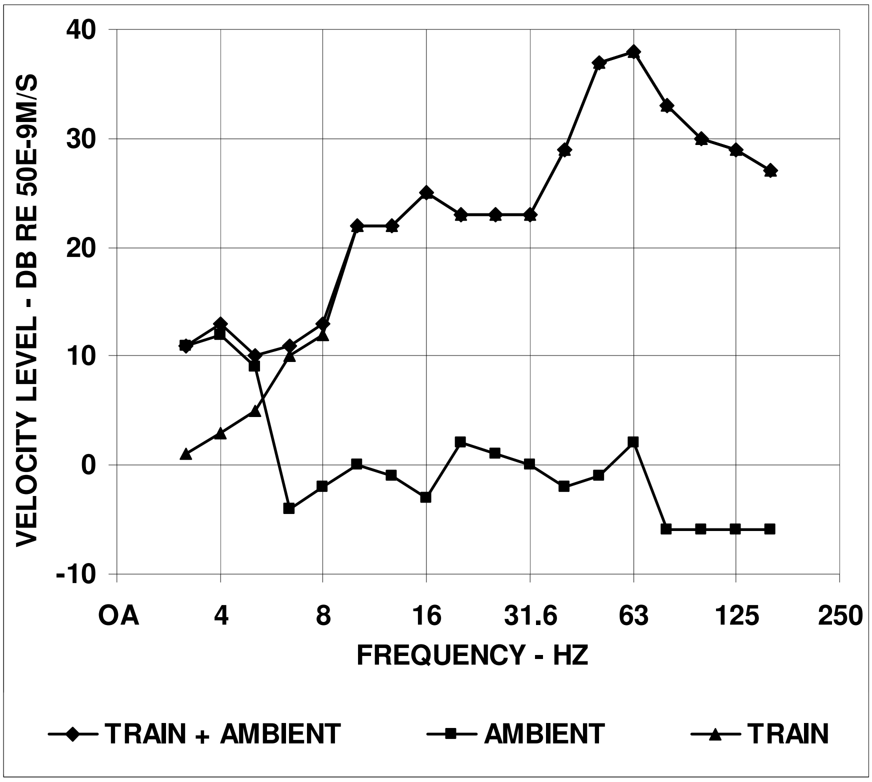

Figure 4. Tunnel Bench Vibration Velocity at 40km/h

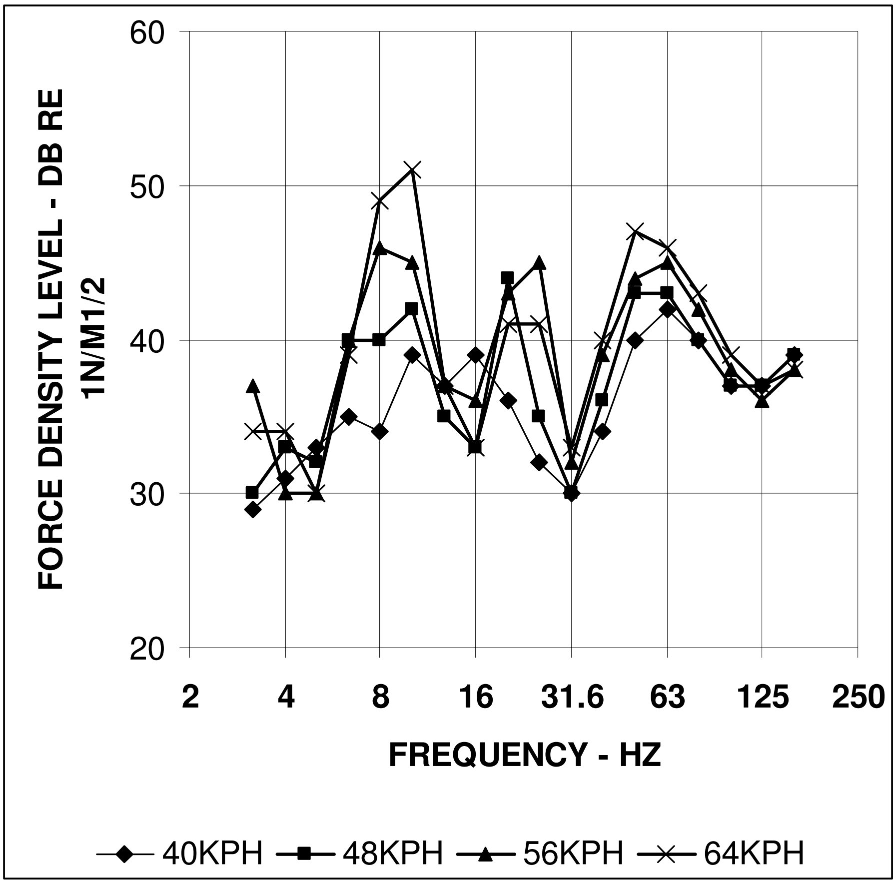

Figure 5. Beacon Hill 2012 FDL Estimates

6. WHEEL ROTATION

The low-frequency force density levels vary considerably with train speed. At 40km/h the FDL was about 30 to 40 dB re $1N/m^{1/2}$ with no pronounced peak below 31.6Hz. At train speeds of 56 and 64km/h, a pronounced peak occurred at 8Hz and 10Hz, respectively. The diameter of a new wheel is 660mm. Thus, the wheel rotation frequencies at these speeds are 7.5Hz and 8.6Hz, respectively. Assuming tread wear of as much as 12.5mm, the wheel diameter would be reduced to 635mm, for which the wheel rotation frequencies would be increased to 7.8 and 8.9 Hz. The frequency peaks are roughly consistent with these frequencies, strongly suggesting that wheel set rotation is involved.

Whether or not the peak is caused by wheel run-out or imbalance is not known. For example, a flexible coupling is employed between the gear unit and drive axle, and flexible couplings are often sources of vibration on rotating machinery. For example, measurements of bogie gear unit vibration indicate that the gear unit vertical vibration exhibits a strong peak at a frequency comparable with wheel rotation frequency, while no such peak is apparent at the bearing housing.

7. RAIL UNDULATION

Rail undulation due to rolling and roller-straightening of the rail may also cause low frequency vibration. The roller diameter of the roller straightening machine is of the same order as the wheel diameter, and may vary as the rolls are trued by the mill.

Samples of rail undulation over lengths of 30m to 60m were obtained with the laser metrology system, shown in Figure 6. Rail displacement as a function of position was analyzed with an “all-poles” method to estimate displacement spectra as a function of inverse wavelength. An example of the third octave undulation as a function of inverse wavelength is given in Figure 7. The spectral peak at about $0.2m^{-1}$ corresponds to a wavelength of 5m. Train speeds of 48 and 64km/h correspond to 13.3 and 17.8m/s. An undulation in the rail at 5m wavelength would produce peaks at about 2.7 and 3.6Hz, respectively, too low to produce the peaks at 8 to 10Hz. The next interesting peak is at $0.39m^{-1}$, corresponding to a wavelength of 2.56m. Speeds of 48 and 64km/h would produce peaks at 5.2 and 6.9Hz, again too low to produce the peaks at 8 to 10 Hz. The peak in the FDL at 48 and 64km/h occur between 8 and 10Hz third octave bands. The wavelength required to produce these peaks is roughly about 1.7m, corresponding to an inverse wavelength of $0.6m^{-1}$. There is no apparent peak in the displacement spectrum at this inverse wavelength in the example shown in Figure 7.

An equivalent third octave band displacement spectrum for 48km/h train speed is shown in Figure 8. The prominent peak at about 2.5Hz shown in this example is not apparent in the wayside vibration spectrum. While the rail vertical displacement peak is highest at these very low frequencies, the reaction forces on the rail due to the un-sprung bogie mass vary roughly in proportion to the square of frequency. Secondly, the bandwidth of the third octave frequency increases in proportion to frequency. The result is that the third octave forces acting on the rail increase roughly at a rate of 15dB per octave, notwithstanding resonances of the vehicle and bogie. Thus, the more important frequencies are those in the neighborhood of the primary suspension resonance and above.

A number of rail undulation spectra were obtained at Beacon Hill, though none at the measurement location of the FDL. The undulation data do indicate a considerable variation of spectra from one section of rail to the next. Further, the measurements indicate that the phase of sinusoidal undulation is typically not continuous across the welds of continuous welded rail, contributing to some randomness of sinusoidal forces that might be produced by rail undulation. This is, of course, beneficial in controlling low frequency ground vibration.

8. EFFECTS OF AXLE SPACING

Assuming perfectly round wheels, a single rolling wheel set would produce a vibration signature that is entirely or almost entirely due to rail roughness and undulation. A second wheel-set would produce the same vibration signature, though displaced in time by the delay between wheel-set passes. The next wheel-set would produce the same signature again, though displaced later time. The result is a time-domain filtering phenomenon that modulates the spectrum of the single wheel-set. An example of this is shown in Figure 9. The peaks and nodes of the modulation spectrum roughly correspond to the peaks and nodes of the tunnel bench vibration, indicating that rail undulation does indeed contribute to measured tunnel vibration. Also apparent in the spectra are peaks at about 6.5 and 8.1Hz that are not modulated by wheel-set passage. The peak at 8.1 Hz may well correspond to wheel rotation. The highest peak at 9.5Hz corresponds to a wavelength of 1.8m in the rail. The wheel diameter would have to be 580mm to for the wheel-set rotation to produce this peak.

The primary suspension resonance frequency of the vehicle is at about 8 to 10Hz. The spectral peaks of tunnel vibration are amplified by the primary suspension resonance, especially if they are coincident with the resonance frequency. A spectral peak related to the modulation spectrum, or, for that matter, wheel rotation, may be easily confused with the primary suspension resonance frequency. Thus, measurement of force density levels for transit vehicles must be done at various train speeds to characterize the vehicle.

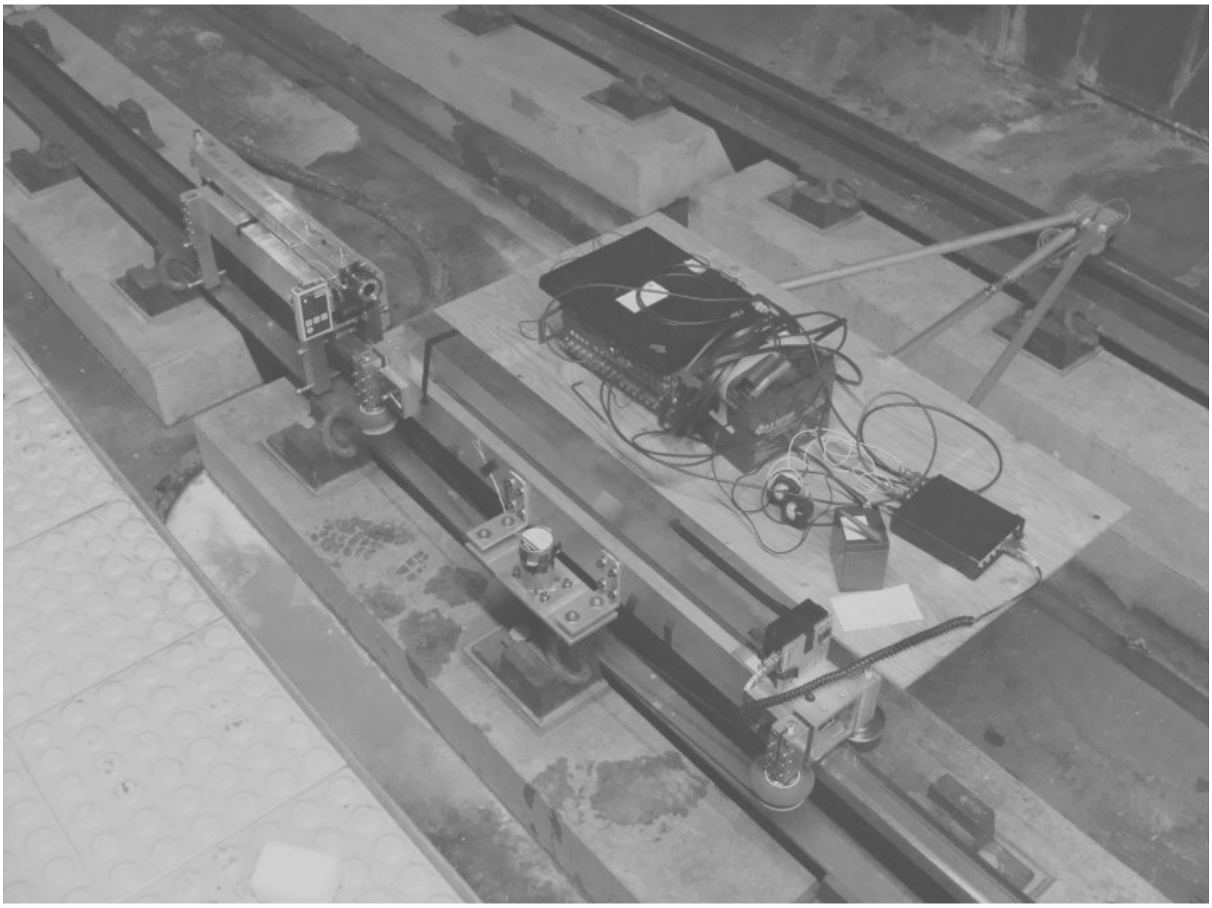

Figure 6. Rail Undulation Measurement

9. PEAK AT 20 to 25HZ

The third octave peaks in the FDL at 20 and 25Hz shown in Figure 5 are not easily identified with any geometric property of the vehicle or track. Further, test data conducted on the Sound Transit vehicle bogies suggest that the source is the center bogie with independently rotating wheels. While not significant in the context of impact on residences above the tunnel, this component of vibration may propagate efficiently over long distances and impact sensitive research instruments. The peak might be related to wheel tread profile and curving performance of the independently rotating wheels of the center bogie. The lateral stiffness of the resilient wheels is low compared to that of a solid wheel, and lateral stick-slip of the wheel tread may occur at frequencies in this range.

10. PROFILE GRINDING

The FDL obtained during testing conducted in 2012 was lower than the FDL measured previously in late 2009 at frequencies above about 40Hz. This was attributed to profile grinding that was conducted in late 2009, just after the 2009 FDL test. The differences between the FDLs measured before and after profile grinding, are plotted in Figure 10. Negative values indicate a reduction due to profile grinding. The differences substantial at 40Hz, suggesting that profile grinding is an effective tool for vibration control at ground-borne noise frequencies.

The time period between FDLs was a little over two years. Subsequent tunnel bench vibration data measured in 2013 suggest that wayside vibration in this frequency range has not increased appreciably, suggesting a low rate of wear and especially low rate or lack of corrugation formation.

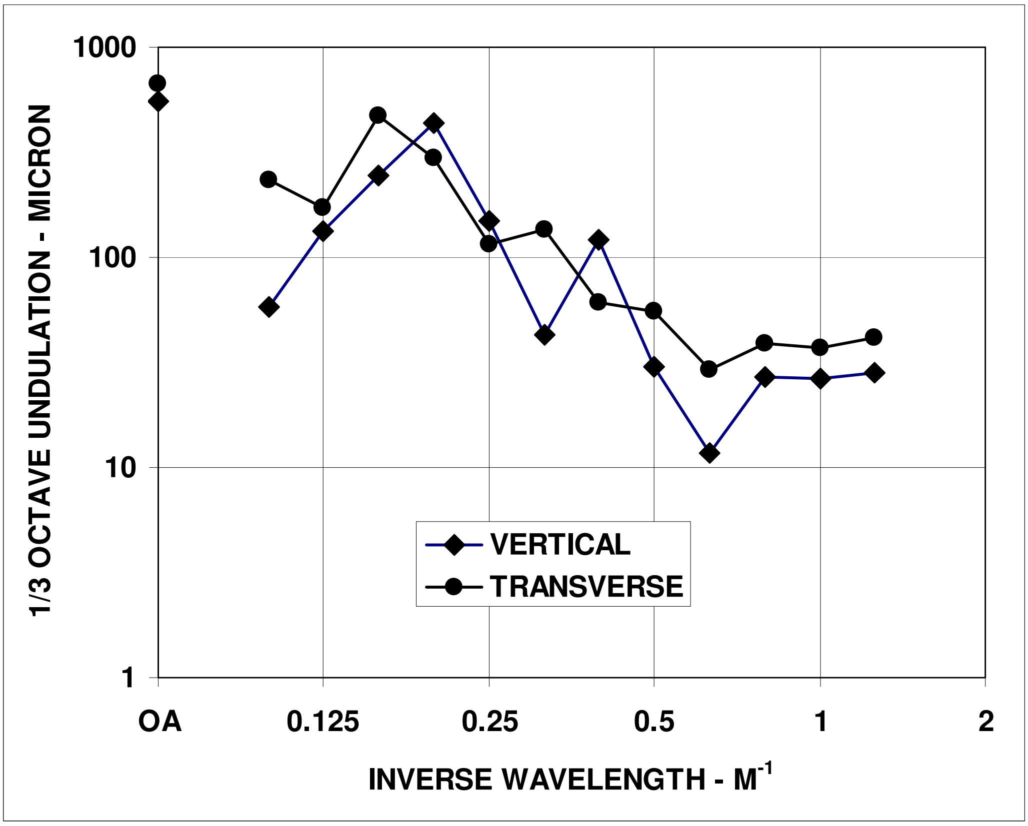

Figure 7. Rail Undulation versus Inverse Wavelength

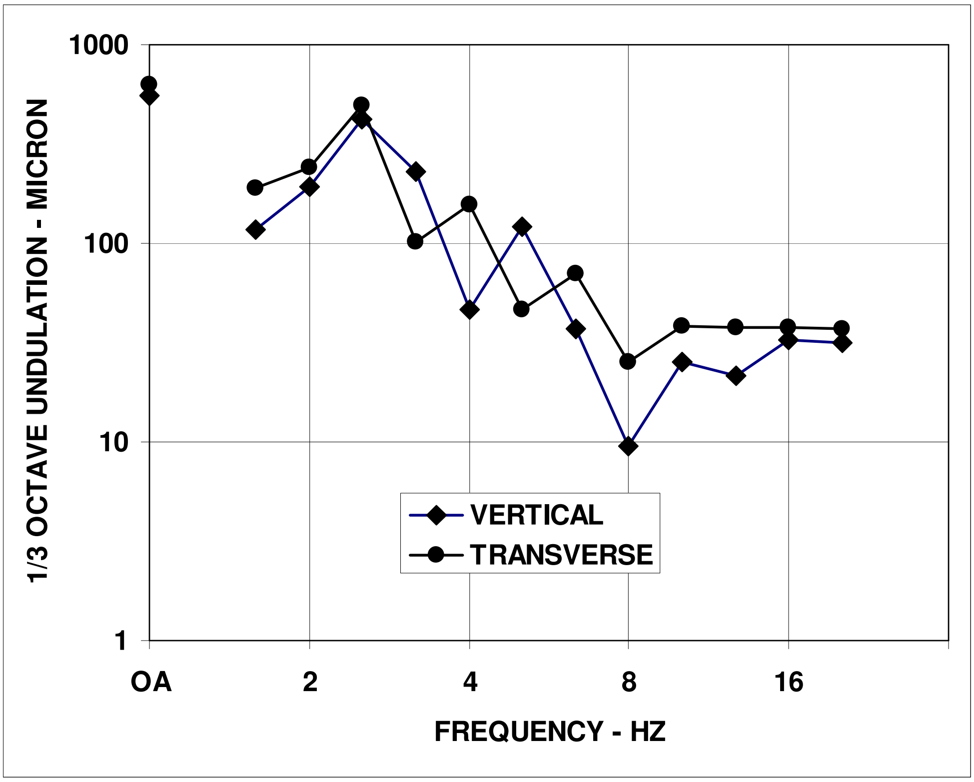

Figure 8. Equivalent Rail Vibration Displacement at 48km/h

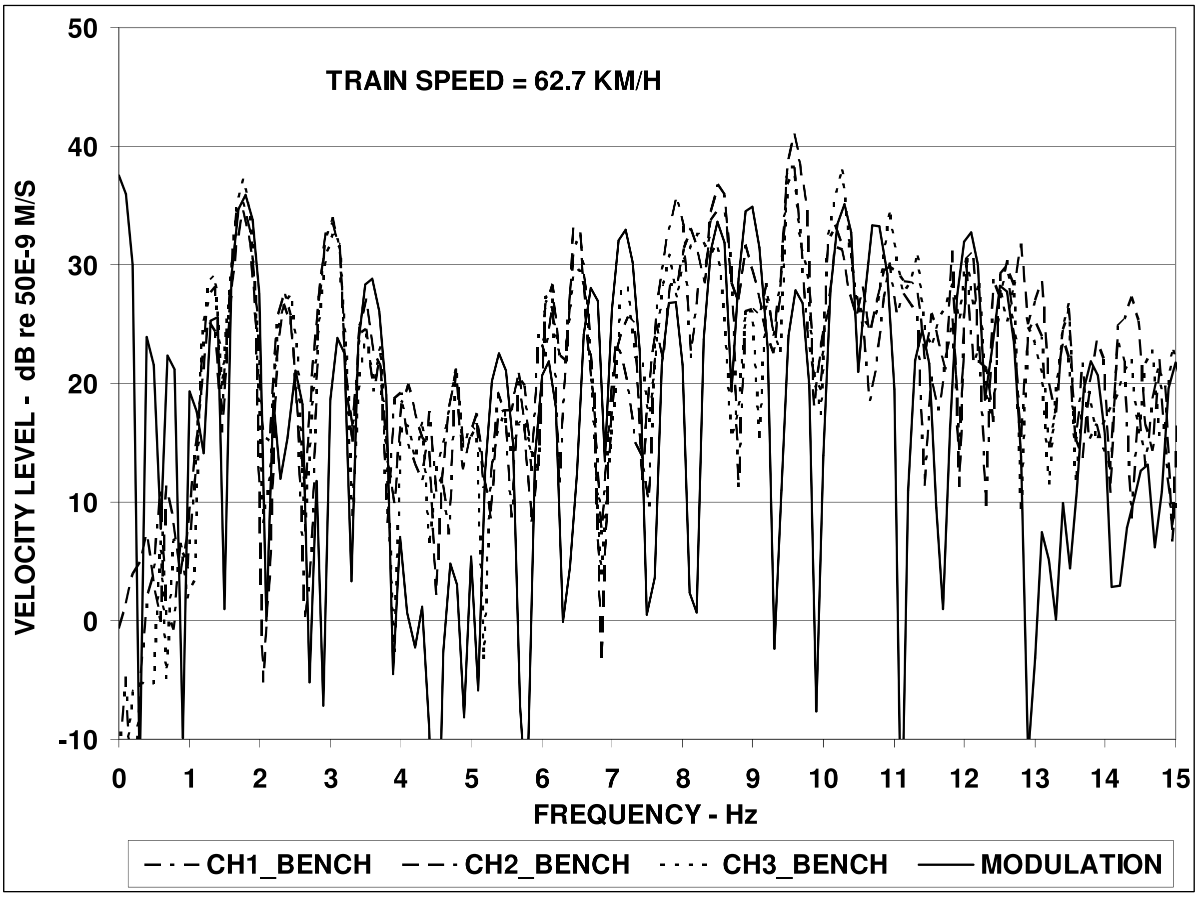

Figure 9 Comparison of tunnel bench vibration with modulation spectrum

11. TRACK CURVE

Track curvature may increase ground-borne vibration levels, though this may be due to increased roughness at curves as apposed to curving performance. The differences between force density levels measured at curved section of track and a tangent section of track are shown in Figure 11. These data were obtained in 2009, also in the Beacon Hill tunnel, on resilient direct fixation fasteners. The average difference was about 1 to 2 decibels at frequencies below 50Hz, but was as much as 6dB at 63Hz third octave.

12. CONCLUSIONS

The foregoing indicates the complexity involved measuring and interpreting low frequency vibration produced by rail transit vehicles. The low frequency peaks at 8 to 10 Hz may not be entirely due to wheel run-out, and may well be due to rail undulation, even though the rail undulation measurements at different sections of track do not support this. The relationship between rail undulation and wayside vibration may be obscured by the modulation spectrum of the axle passage. Other mechanisms might be involved, such as parametric excitation of flexible couplings used between the gear and axle, which may produce forces at some multiple of wheel rotation.

Lateral slip of the independently rotating wheels of center truck may be occurring at 20 to 25 Hz, depending on train speed. Not all vehicles produce this component of vibration, and the available data indicate that the peak may be related to wheels that have substantial wear. Hollow-worn wheels may be causing poor curving performance of the center truck. However, other data not reported here indicate that these peaks may be occurring on tangent track as well. Thus, the nature ground vibration produced by rail-vehicle systems is exceedingly complicated and deserving of further investigation.

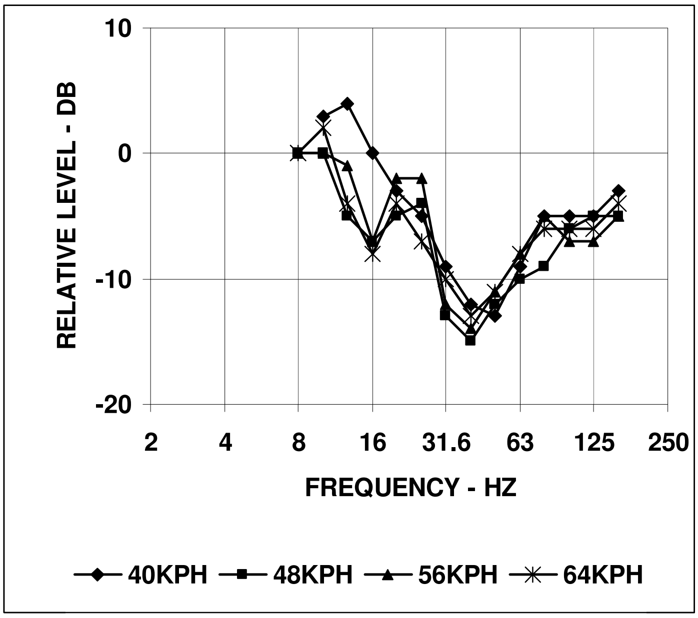

Figure 10 Profile Rail Grinding Effectivenss

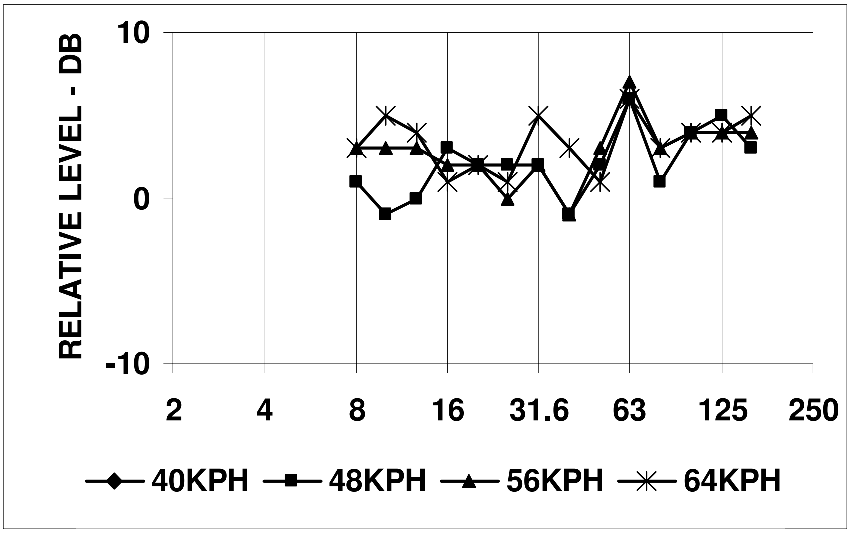

Figure 11 FDL At Curve Relative to FDL at Tangent

ACKNOWLEDGEMENTS

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1. U.S. Federal Transit Administration, Transit Noise and Vibration Impact Assessment, 1995.