TCRP
REPORT 67

TRANSIT
COOPERATIVE
RESEARCH
PROGRAM

Sponsored by
the Federal
Transit Administration

Wheel and Rail Vibration Absorber Testing and Demonstration

TRANSPORTATION RESEARCH BOARD

NATIONAL RESEARCH COUNCIL

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Wheel and Rail Vibration Absorber Testing and Demonstration

WILSON, IHRIG & ASSOCIATES, INC.
Oakland, CA

SUBJECT AREAS
Public Transit
Rail

Research Sponsored by the Federal Transit Administration in Cooperation with the Transit Development Corporation

TRANSPORTATION RESEARCH BOARD — NATIONAL RESEARCH COUNCIL

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TRANSIT COOPERATIVE RESEARCH PROGRAM

The nation's growth and the need to meet mobility, environmental, and energy objectives place demands on public transit systems. Current systems, some of which are old and in need of upgrading, must expand service area, increase service frequency, and improve efficiency to serve these demands. Research is necessary to solve operating problems, to adapt appropriate new technologies from other industries, and to introduce innovations into the transit industry. The Transit Cooperative Research Program (TCRP) serves as one of the principal means by which the transit industry can develop innovative near-term solutions to meet demands placed on it.

The need for TCRP was originally identified in *TRB Special Report 213—Research for Public Transit: New Directions,* published in 1987 and based on a study sponsored by the Urban Mass Transportation Administration—now the Federal Transit Administration (FTA). A report by the American Public Transportation Association (APTA), *Transportation 2000,* also recognized the need for local, problem-solving research. TCRP, modeled after the longstanding and successful National Cooperative Highway Research Program, undertakes research and other technical activities in response to the needs of transit service providers. The scope of TCRP includes a variety of transit research fields including planning, service configuration, equipment, facilities, operations, human resources, maintenance, policy, and administrative practices.

TCRP was established under FTA sponsorship in July 1992. Proposed by the U.S. Department of Transportation, TCRP was authorized as part of the Intermodal Surface Transportation Efficiency Act of 1991 (ISTEA). On May 13, 1992, a memorandum agreement outlining TCRP operating procedures was executed by the three cooperating organizations: FTA, the National Academies, acting through the Transportation Research Board (TRB); and the Transit Development Corporation, Inc. (TDC), a nonprofit educational and research organization established by APTA. TDC is responsible for forming the independent governing board, designated as the TCRP Oversight and Project Selection (TOPS) Committee.

Research problem statements for TCRP are solicited periodically but may be submitted to TRB by anyone at any time. It is the responsibility of the TOPS Committee to formulate the research program by identifying the highest priority projects. As part of the evaluation, the TOPS Committee defines funding levels and expected products.

Once selected, each project is assigned to an expert panel, appointed by the Transportation Research Board. The panels prepare project statements (requests for proposals), select contractors, and provide technical guidance and counsel throughout the life of the project. The process for developing research problem statements and selecting research agencies has been used by TRB in managing cooperative research programs since 1962. As in other TRB activities, TCRP project panels serve voluntarily without compensation.

Because research cannot have the desired impact if products fail to reach the intended audience, special emphasis is placed on disseminating TCRP results to the intended end users of the research: transit agencies, service providers, and suppliers. TRB provides a series of research reports, syntheses of transit practice, and other supporting material developed by TCRP research. APTA will arrange for workshops, training aids, field visits, and other activities to ensure that results are implemented by urban and rural transit industry practitioners.

The TCRP provides a forum where transit agencies can cooperatively address common operational problems. The TCRP results support and complement other ongoing transit research and training programs.

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Each report is reviewed and accepted for publication by the technical panel according to procedures established and monitored by the Transportation Research Board Executive Committee and the Governing Board of the National Research Council.

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FOREWORD

By Staff
Transportation Research Board

This report will be of interest to engineers responsible for wheel/rail noise control in the design, construction, and operation of rail transit systems. It provides the results of field tests performed at two light rail transit systems to demonstrate the effectiveness of wheel and rail vibration absorbers in reducing rolling noise on tangent track and wheel squeal on curved track. Testing of wheel and rail vibration absorbers was conducted at the Tri-County Metropolitan Transportation District of Oregon (Tri-Met). The wheel and rail vibration absorbers were tested individually, and in combination, on both tangent and curved track. Wheel vibration absorbers were also field tested on tangent and curved track at the New Jersey Transit Corporation's Newark subway system.

TCRP Report 23, Wheel/Rail Noise Control Manual, published in 1997, provides practical step-by-step procedures for identifying wheel/rail noise control technologies with demonstrated effectiveness. Procedures are included for identifying wheel/rail noise sources, developing mitigation designs, and estimating probable costs and effectiveness of control provisions. The manual covers noise generated on tangent track, curved track, and special trackwork.

Upon completion of this manual, recommendations were made to conduct field testing of new or emerging wheel/rail noise control technologies identified during the research leading to the preparation of the manual. Specifically, recommendations were made to test (1) several types of wheel and rail vibration absorbers, (2) nitinol wheel treads, and (3) piezo-ceramic vibration dampers. On the basis of these recommendations, additional TCRP funding was received to field test some of these technologies.

Under TCRP Project C-3A, "Field Testing of Wheel/Rail Noise Control Techniques," research was undertaken by Wilson, Ihrig & Associates, Inc. to field test several of the identified technologies. After reviewing the various technologies, the project panel selected wheel and rail vibration absorbers as the focus of the field testing effort. The objective of TCRP Project C-3A was to field test wheel and rail vibration absorbers on resilient wheels to determine the effectiveness of these technologies in reducing rolling noise on tangent track and wheel squeal on curved track.

To achieve the project objectives, the researchers first solicited transit systems interested and available to conduct the field tests. The researchers also identified available wheel and rail vibration absorber technologies and discussed possible field tests with manufacturers. As a result of these discussions, specific testing plans were developed for review and approval of the project panel. The selected wheel and rail vibration absorbers were laboratory tested to determine the resonance frequencies and damping properties and were field tested at Tri-Met and New Jersey Transit to determine noise reductions.

The selection of the wheel and rail vibration absorbers tested in this report was based on their availability for the project and the amount of or lack of previous testing performed in the United States. The purpose of this project was to provide information on the potential benefit of wheel and vibration absorbers generically—not to evaluate the capabilities of the products offered by specific manufacturers.

NOTICE TO READERS

The selection of wheel and rail vibration absorbers tested in this report was based on their availability for the project and the amount of previous testing in the United States. The purpose of this project was to provide information on the potential benefit of wheel and rail vibration absorbers generally and not to evaluate the capabilities of the products offered by specific manufactures. The specific manufacturers and products appear solely because they are considered essential to the clarity and completeness of the project reporting.

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AUTHOR ACKNOWLEDGMENTS

The research was performed under TCRP Project C-3A by Wilson, Ihrig & Associates, Inc., as prime contractor. The contract was monitored by Mr. Christopher Jenks and Ms. Monica Francois of the Transportation Research Board. Dr. James Tuman Nelson, Vice President of Wilson, Ihrig & Associates, Inc., was the Principal Investigator, supported by other Wilson, Ihrig & Associates, Inc. staff, including Mr. Thomas Bergen, Senior Consultant; Mr. James Phillips, Associate Consultant; and Mr. Gary Glickman, Assistant Consultant. Mr. Pedro Laconsay, Mr. George Smith, and Mr. Michael Amato performed the laboratory analyses of the very extensive data set that was collected. Ms. Julie Riffel and Ms. Beverly Milam assembled the various memoranda and reports.

The New Jersey Transit system and the Portland Tri-Met system provided test sites, vehicles, and labor for installation and testing of the treatments. The New Jersey Transit procured the wheel vibration absorbers and modified Bochum 84 wheels for testing. Mr. Stelian Canjea of the New Jersey Transit and Mr. Kenneth Kirse and Mr. Thomas Heilig of the Portland Tri-Met provided invaluable assistance, without which these tests could not have occurred. Mr. James Dwyer, of the Port Authority of Allegheny County, provided early assistance in evaluating the Pittsburgh system as a site for testing. Mr. Richard Trail and Mr. Jaideep Luthra of Penn Machine assisted in the procurement of the wheel vibration absorbers, and Mr. Gunther Veit of Schrey & Veit assisted with the procurement of the rail vibration absorbers.

CHAPTER 1

PROJECT SUMMARY

Wheel/rail noise measurements were performed to assess the effectiveness of wheel and rail vibration absorbers for reducing wayside noise. The project included procurement and installation of wheel vibration absorbers on Bochum 84 and 54 resilient wheels for testing at the systems of New Jersey Transit (NJT) and Tri-County Metropolitan Transportation District of Oregon (Portland Tri-Met), respectively, at both tangent and curved track. Rail vibration absorbers were also procured and installed at the Portland Tri-Met and tested at both tangent and curved track. The measurements included wayside noise, under-car noise, and rail vibration.

PROJECT RESULTS

The results of the study were as follows:

- The wayside rolling noise reduction obtained at tangent track with wheel vibration absorbers was less than 1 dB. The wheel vibration absorber was designed to control wheel squeal related to lateral bending oscillation of the tire. Rolling noise is believed to involve primarily radial vibration of the tire in response to wheel and rail roughness. The absorbers apparently had little affect on this component of tire vibration.
- Wayside rolling noise levels were slightly higher with the rail vibration absorbers relative to without by about 1 or 2 dB. This result is contradictory to expectations.
- There was no reduction of wayside rolling noise achieved by the combination of wheel and rail vibration absorbers.
- The rail vibration absorbers eliminated “singing rail” at the tangent track test section. The elimination of singing rail was very apparent and gave a very favorable impression of the wayside noise environment. Thus, the treated rail was considered to be quieter than the untreated rail, even though the maximum and single-event A-weighted noise levels were not reduced.
- Rail vibration levels at tangent track were significantly reduced with either wheel or rail vibration absorbers, the latter being most effective.
- Wayside rolling noise at tangent track appeared to be almost entirely produced by radiation from the wheel tire.
- The probability of occurrence and decibel level of wheel squeal noise at curves was less with wheel vibration absorbers than without, though wheel squeal was not eliminated.
- The wheel squeal produced by the Bochum resilient wheel alone was intermittent, and there were cases where the Bochum resilient wheel produced no wheel squeal without the wheel vibration absorbers attached. The Bochum resilient wheel is, by itself, effective in reducing the occurrence of wheel squeal.
- The high frequency stick-slip or flanging noise reductions at the curved track test section at the Portland Tri-Met East Portal were disappointing with either the wheel or rail vibration absorbers.
- There was a reduction of pre-passage curving noise radiation from the rail at the Portland Tri-Met East Portal with rail treated with rail vibration absorbers.
- Rail vibration levels caused by stick-slip forces at the curved track test section were significantly lower with the rail treated with vibration absorbers than with the untreated rail.
- The reduction of rail vibration with the wheel and rail vibration absorbers suggests that they might be beneficial in reducing rail corrugation. This last possibility is an avenue for further research that should be seriously considered as a means of reducing rail grinding costs.
- The modal tests indicate that the peak response of lateral bending modes at about 2,600, 3,800, 6,400, and 7,700 Hz were substantially attenuated. Lateral bending modes at about 500, 1,400, 5,100, and 9,000 Hz were for the most part unaffected.
- The modal data indicate that the radial response of the tire appeared to be attenuated between 2,000 and 2,600 Hz and between 3,000 and 3,300 Hz. However, the identification of radial modes is obfuscated by lateral bending of the tire, which produces rotation of the tire tread about its neutral axis, leading to a radial component of tire motion at its edges.
- The radial input mechanical impedance of the tire at anti-resonance frequencies appeared to be lessened relative to the untreated case. This may have lead to a reduction of rail vibration by the wheels treated with vibration absorbers. However, the mechanical impedance reduction was small below 1,000 Hz, while testing at tangent track indicated a significant reduction of rail vibration by the wheel vibration absorbers.

CHAPTER 2

INTRODUCTION

TCRP Project C-3A, “Wheel and Rail Vibration Absorber Testing and Demonstration,” was a demonstration of the noise reduction effectiveness of wheel and rail vibration absorbers at light rail transit systems. The project involved enlistment of selected transit systems, procurement of treatments, field and laboratory testing, and report preparation.

Prior work in the United States included testing of various constrained layer damping treatments and a Krupp vibration absorber on solid steel wheels in at the Metropolitan Transportation Authority-New York City Transit (MTA NYCT) (*1*), as well as testing of the ADtranz fin and block vibration absorbers on solid steel wheels at Washington Metropolitan Area Transit Authority (WMATA). The results of these tests were favorable in reducing wheel squeal. No measurements of rolling noise reduction were obtained for wheel vibration absorbers. No testing of rail vibration absorbers has been conducted within the United States, although a test of rail vibration constrained layer dampers was conducted at the Massachusetts Bay Transportation Authority (MBTA) under TCRP Project C-3, “Wheel/Rail Noise Mitigation.”

The present program was expanded to include combined testing of both rail and wheel vibration absorbers at both tangent and curved track. Testing of this type has not been conducted previously within the United States.

The test program included the following major phases:

1. Transit system selection,
2. Treatment selection,
3. Laboratory testing, and
4. Field testing.

This final report summarizes the test program and its major phases. The report begins with a description of wheel and rail vibration absorbers, followed by transit system selection. The discussion of field testing and results is broken into two parts: (1) rolling noise reduction and (2) wheel squeal. Modal analyses of wheel and rail vibration absorbers are also discussed.

CHAPTER 3

VIBRATION ABSORBERS

DESCRIPTIONS OF WHEEL AND RAIL VIBRATION ABSORBERS

Vibration absorbers are tuned spring-mass mechanical oscillators that produce a high mechanical impedance at the design resonance frequency at the point of attachment to a vibrating surface, thus reducing the response of the vibrating system at the absorber's resonance frequency. If the absorber is tuned to match the resonance frequency of the vibrating system, the combined resonance is split into two closely spaced resonances. Damping material is included in the absorber to absorb energy at the tuned resonance frequency of the absorber. The damping material broadens the frequency response at resonance while reducing the amplitude of the response. In this case, the vibration absorber is referred to as a "dynamic absorber."

The vibration absorbers may have multiple masses, thus producing several resonance frequencies and, combined with damping, a "broad-band" damping characteristic. For this demonstration, the selected wheel vibration absorbers are tuned dynamic absorbers, and the rail vibration absorbers are broad-band dynamic absorbers. However, these tuned wheel vibration absorbers were also designed to provide damping over a broad range of frequencies above their fundamental resonance frequency. Tuned wheel vibration absorbers are designed primarily for controlling wheel squeal at curves, where lateral oscillation of the tire in bending is the primary mode of vibration. The project was designed to assess the absorbers' effectiveness in reducing wheel squeal and to determine whether there were any ancillary benefits in reducing rolling noise. The broad-band rail vibration absorbers were intended to reduce both rolling noise and squeal. Manufacturers' data indicate that both of these products reduce both rolling noise at tangent track and wheel squeal at curved track.

To date, wheel vibration absorbers have received little or no application at U.S. transit systems, though they have been employed at some European transit systems. For example, the German Inter City Express (ICE) employs wheel vibration absorbers to control rolling noise, and several German tram systems employ tuned wheel vibration absorbers to control wheel squeal. Rail vibration absorbers have, apparently, received less attention than wheel vibration absorbers, either here or in Europe.

Wheel squeal is produced primarily by lateral slip of the steel tire across the rail head during curve negotiation, and it is generated by the negative slope of the friction force-versus-lateral-creep curve. Various methods have been proposed to control wheel squeal, including wheel vibration absorbers or tuned dampers. The Bochum resilient wheel is often used in favor of solid wheels for controlling wheel squeal. Other methods include lubrication with petroleum-based grease, PTFE (Teflon), vegetable-based oils and greases, and water sprays. Application of vibration absorbers is attractive as a low-maintenance noise control alternative with negligible environmental consequences.

Rolling noise at tangent track is produced by wheel and rail roughness and by variation of the modulus of elasticity and rail head curvature with distance along the rail. Rail noise radiation is controlled in part by resonances in the track and anti-resonances of the wheel, which produce peaks in wheel/rail dynamic contact force spectra. Recently, adverse community reaction to a "singing rail" phenomenon has been reported for lightly damped continuous welded rail on concrete tie track. Singing rail is related to pass- and stop-band vibration transmission characteristics of the discretely supported rail. Broad-band vibration absorbers are attractive for controlling singing rail and reducing wayside noise. Further, they may have a substantial effect in reducing rail corrugation rates where such are related to pinned-pinned resonances of the rail. Rail vibration absorbers may be most important in this latter regard.

CHAPTER **4**

TRANSIT SYSTEM SELECTION

Solid steel wheels are prone to squeal at curves with radii up to about 750 ft. Wheel vibration absorbers and dampers were tested in New York and shown to be effective in reducing wheel squeal. The resilient Bochum wheel is very popular at light rail transit systems and is usually effective in reducing wheel squeal. However, resilient wheels in general are not entirely effective in preventing squeal, as indicated by the Pittsburgh Port Authority of Allegheny County, the NJT, the MBTA (Boston) Green Line, the Sacramento RT, and the Portland Tri-Met. Squeal from resilient wheels has produced vigorous community reaction at some of these systems, though the squeal produced at these systems would probably have been more severe if solid wheels were used. A good example of this is the MBTA Blue Line, which used solid steel wheels. In view of the prior and successful testing of wheel vibration absorbers in New York, and the lack of testing with resilient wheels, the present program was confined to testing with resilient wheels. Thus, systems with resilient wheels were selected for testing. These included the NJT and the Portland Tri-Met systems.

NEW JERSEY TRANSIT

One of the systems selected for testing wheel vibration absorbers was the NJT system. The system is representative of those using Presidents Commerce Commission (PCC) cars. The NJT was experimenting with a set of Bochum 84 resilient wheels on one of the PCC cars. A new set of Bochum 84 wheels was purchased for testing, complete with threaded inserts in the tire for mounting the vibration absorbers. The vibration absorbers were provided by the manufacturer of the Bochum wheel, specifically tuned for the tire's response characteristics. The test locations included the main station loop and an at-grade turnaround. Additionally, measurements were conducted at a section of tangent ballast-and-tie.

No rail vibration absorbers were tested at the NJT.

PORTLAND TRI-MET

The Portland Tri-Met (Tri-Met) system was selected for testing both wheel and rail vibration absorbers. Tri-Met is representative of a modern light rail transit system with articulated vehicles. In this case, Car 115 of the Type 1 series of vehicles manufactured for Tri-Met by Bombardier was selected by Tri-Met for testing. This vehicle uses Bochum 54 resilient wheels on all three trucks.

All of the trucks of the Type 1 vehicle have solid axles. The Type 2 vehicle used at Tri-Met employs Bochum 54 wheels at each of the two driven trucks, and Bochum 84 wheels at the idling center truck. This latter truck includes independently rotating wheels on stub axles that accommodate the low-floor height design of the vehicle. Although the researchers were interested in testing the Type 2, physical interference precluded testing the wheel vibration absorbers.

Tri-Met was willing to test rail vibration absorbers at both tangent and curved track. The curved track was a section of 400-ft radius curve at which Portland was experiencing vigorous community reaction to noise and at which Portland had experimented with a number of noise mitigation measures. Thus, Tri-Met was also selected for testing rail vibration absorbers.

Tangent Track Test Section

The tangent track test section consisted of RE 115 lb/yd rail with concrete ties and Pandrol clips and rail pads. The rail was ground approximately 1 year prior to testing and had an excellent running surface. However, the gauge corner was not ground to a rounded corner, leaving a sharp gauge corner. A number of factors suggest that gauge corner condition did not adversely affect the test: (1) there did not appear to be significant wear of the gauge corner, suggesting that the trucks were well centered in the track; (2) the noise levels recorded for the test vehicle with new wheels were lower than those observed for revenue service wheels, indicating that wheel condition was the major factor in wayside noise generation; and (3) wayside one third octave noise levels for each of the test vehicle passes were uniform and repeatable for each test condition. The experience of the author is that this location was excellent for testing.

Curved Track Test Section

The curve track section consisted of RE 115 lb/yd rail with mono-block rail supports. The rail was retained with Pandrol clips and supported by resilient rail pads. A restraining rail was originally installed at the low rail side, but was removed by Tri-Met some time before testing in an attempt to reduce curving noise. Absorptive sound barrier walls and water spray systems were introduced by Tri-Met to reduce and control wheel squeal. While sound barrier walls reduced wheel squeal, they did not eliminate it. The water spray system was effective in eliminating squeal. The water spray systems were turned off during testing.

The track gauge at the curve was 4′-9″, one-half inch wider than standard. The one-half-inch gauge widening in conjunction with the restraining rail is designed to prevent flange contact at the gauge corner, and thus reduce flange wear. While this is attractive to control wear, it was not effective in preventing squeal. The large gauge widening was probably a factor in continued wheel squeal and stick-slip flanging noise at this curve. The gauge corner and face were not excessively worn at the low rail, and both the low and high rails were in excellent condition, having received less than 1 year of revenue service.

CHAPTER 5

TREATMENT SELECTION AND INSTALLATION

The treatments tested under the project are described in the following sections.

WHEEL VIBRATION ABSORBERS

Two types of wheel vibration absorbers were considered for testing. One was the ADtranz fin absorber, described as a “broad-band” absorber. The manufacturer indicated that the fin shape of the absorbers makes the absorber act as a damped transmission line, broadening the frequency response of the absorbers. The manufacturer was unable to provide the absorbers within the time frame anticipated for testing. The second type of absorber was the VSG dynamic absorber manufactured by Bochumer Verein, the manufacture of the Bochum resilient wheel. Both the VSG absorber and the ADtranz fin absorber are cantilevered constrained layer plate designs. Both of these designs are indicated by their respective manufacturers to have broad-band characteristics. The VSG dynamic absorber was tested at both NJT and Tri-Met.

New Jersey Transit

A complete set of eight Bochum 84 resilient wheels with threaded inserts and VSG dynamic absorbers were purchased by NJT from Penn Machine Co. for application to a PCC car. These wheels were replacements for the PCC resilient wheels. The axles were shipped to Penn Machine for mounting the wheel centers, and later shipped back to NJT, where they were mounted on Car No. 9. The car originally had PCC Super Resilient wheels. Figure 1 illustrates the vibration absorbers mounted on the Bochum 84 wheel.

The cost of the vibration absorbers was $960 per wheel. Installation of the absorbers was performed by Penn Machine and NJT between tests. The installation required less than 15 min per wheel after the car was brought into the maintenance shop. The mounting bolts of the absorbers were tightened to specified torque and were secured with a locking compound. There were no difficulties with clearance.

Portland Tri-Met

A complete set of 12 Bochum 54 resilient wheel tires with threaded inserts and VSG dynamic absorbers was obtained for the project. The tires, less absorbers, were shipped to Portland, where they were mounted by Tri-Met on Car 115.

Figure 2 illustrates the vibration absorbers mounted on the Bochum 54 wheel. The mounting pattern differs slightly from that used at the NJT. Where the absorbers were mounted uniformly about the tire circumference on the NJT tire, the Tri-Met vibration absorbers were mounted in three groups of two absorbers per group. Thus, diametrical lines of symmetry were broken, which may have some advantage in destroying a mode of vibration.

The cost of the vibration absorbers was $800 per wheel. Installation was performed by Tri-Met during the course of testing, requiring approximately 15 min per wheel. The absorber mounting screws were tightened with a torque wrench. A locking compound was employed to fix the bolts.

RAIL VIBRATION ABSORBERS

Rail vibration absorbers were obtained from Schrey & Veit, which supplies the absorbers under license from Daimler Chrysler. The selected design consisted of a thick plate clipped to the rail foot, with absorbers clamped to the top of the rail foot and the side of the web. Each absorber thus contained four broad-band dynamic multidegree-of-freedom absorbers. Photographs of the absorbers are provided in Figure 3.

A design requirement was the ability to mount the absorbers on the rail without raising the rail. Earlier designs of rail vibration absorbers had the absorber body clamped to the underside of the rail foot, requiring considerable space between the bottom of the rail and concrete invert or ballast. The configuration employed here required less than 1 in. of clearance between the rail foot and concrete or ballast and thus was easily installed.

The cost of the rail vibration absorber was $240 per absorber assembly, including shipping. Installation was performed by five of Tri-Met’s maintenance-of-way personnel. The absorbers were partially assembled at the El Monica Yard, then loaded on a high-railer, and distributed along the test section at night. The following night, the absorbers were installed with an air impact wrench and tightened to recommend torques. Nonrevenue period begins at about 12 midnight and ends at about 5 a.m. The process required a total of three nights of installation.

Figure 1. VSG vibration absorber attached to NJT Bochum 84 resilient wheel.

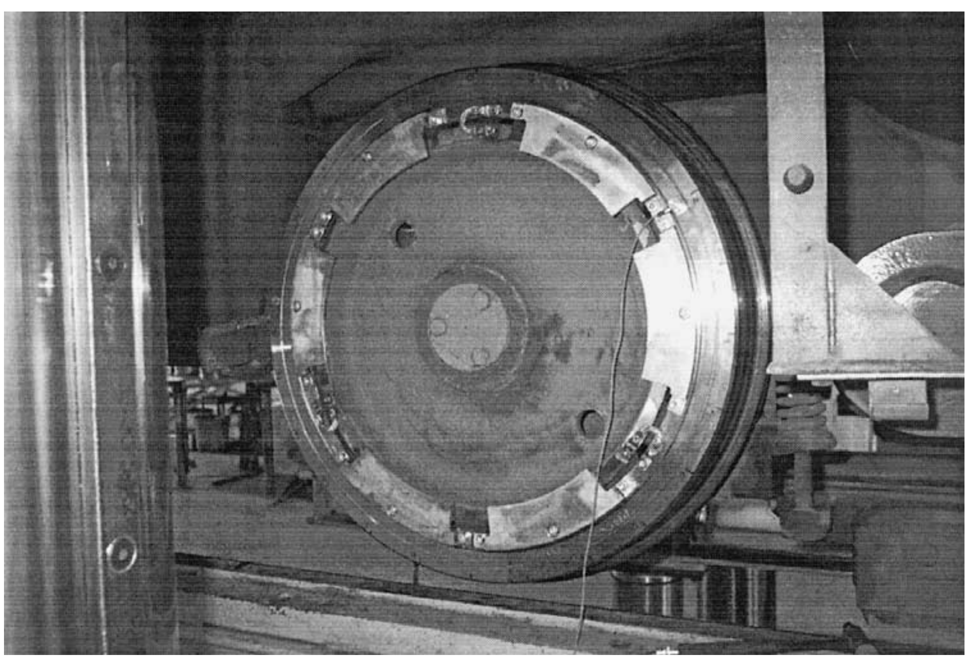

Figure 2. Tri-Met wheel vibration absorbers mounted on Bochum 54 wheels.

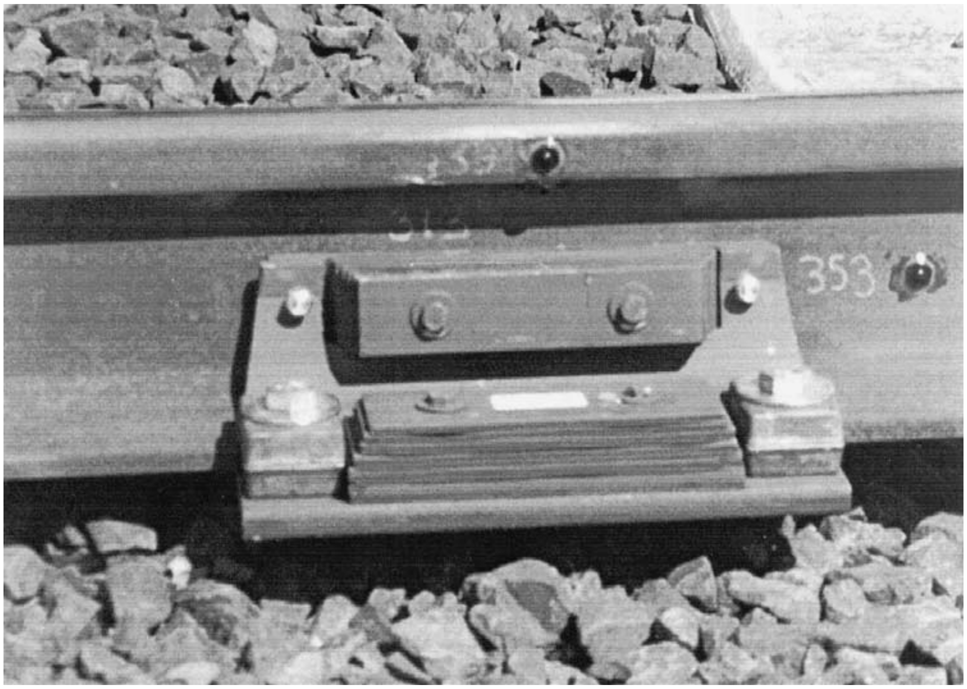

Figure 3.   Rail vibration absorber attached to 115 lb/yd RE rail at the Portland Tri-Met ballast-and-concrete-tie track.

CHAPTER **6**

LABORATORY TESTING

Modal analyses were performed on both the NJT Bochum 84 wheels and the Portland Tri-Met Bochum 54 wheels, with and without vibration absorbers fitted to the wheels. The purpose of the analyses was to determine the resonance frequencies and damping properties of the wheels with and without the vibration absorbers and determine mode shapes. The modal analyses were conducted at the shops of the respective transit systems. Originally, tests were planned to be conducted under laboratory conditions, but the shop conditions were adequate and desirable. Testing the wheels and vibration absorbers in situ is more representative than shipping a new tire to the laboratory and supporting the wheel on a special jig.

NEW JERSEY TRANSIT

The modal analyses of the Bochum 84 wheel were conducted over the audible frequency range up to 2,500 Hz. The lateral response of the tire over this range included two lightly damped modes at about 600 Hz and 1,600 Hz. The modal response of the tire at 600 Hz was unaffected by the wheel vibration absorbers. However, the transverse bending mode of the tire at 1,600 Hz was eliminated, leaving two highly damped split modes at either side of the primary undamped resonance frequency. This is a textbook case of the effectiveness of a dynamic absorber. The absorbers' effect on the 1,600 Hz lateral tire bending mode is reflected in the curving noise reductions discussed below.

The radial response of the Bochum 84 tire contained numerous resonances at about 600 Hz, 780 Hz, 1,250 Hz, 1,600 Hz, and 2,100 Hz. The responses at 600 Hz and 1,600 Hz appear to be cross-coupled responses of the lateral bending resonances of the tire and cause rotation of the tire about its neutral axis, thus producing a radial motion of the tire tread at its edges. Radial resonance frequencies were at 780, 1,250, and 2,100 Hz. Addition of the vibration absorbers modified the resonance frequencies slightly, but did not greatly alter the radial response, although the modal responses were in general about 3 to 5 dB less with the vibration absorbers attached than without.

PORTLAND TRI-MET

Experimental modal analyses were conducted at the Portland Tri-Met's Ruby Junction Yard and again at the El Monica Yard. The latter tests were repeat tests of those conducted at the Ruby Junction Yard to obtain greater spatial resolution and thus more accurate differentiation between radial and lateral modes of the tire. The tests conducted at the El Monica Yard on August 28, 1999, are reported here.

The modal tests were conducted on the center truck of Car 115. The center truck does not have traction motors, so the wheel sets were less damped than those of the powered trucks. The powered trucks have gearboxes and floating axles with flexible drive connectors between the gearbox and axle that add damping and nonlinearity to the system. The car was jacked up with the wheel sets suspended, free to rotate.

An instrumented hammer, accelerometer, and digital magnetic tape recorder were used for data acquisition. This procedure differed from that used at the NJT, where the modal response data were analyzed live in the field. The recorder facilitated data acquisition and allowed multiple analyses of the same data under laboratory conditions. The response data were collected by mounting an accelerometer at a specific location and by repeatedly impacting the wheel with the instrumented hammer at preselected points about the tire. Forty-eight equally spaced impact points were used on the lateral flat surface of the tire, separated by 7.5 degrees about the tire, for lateral response measurements. For the radial direction, two rows of 24 equally spaced (with 15-degree separations) impact points were selected, each on the cylindrical running surface. One row was adjacent to the flange, and the other was at the outer edge of the tire. This separation was employed to help distinguish between the lateral bending and purely radial modes of the tire. The measurements were repeated with the wheel vibration absorbers installed. For the lateral direction, three additional impact points were added to each absorber, thus including the response of the absorbers in the modal display. The analyses extended from 0 Hz to 10,000 Hz, four times the bandwidth used at the NJT.

The modal parameters obtained for the Tri-Met vehicle are summarized in Tables 1 and 2 for the lateral and radial bending modes, respectively. For the lateral bending modes of the tire, the damping ratios were increased in every case, as may be expected. The largest increase in damping ratio, 316 percent, was obtained at 2,588 Hz, which is one of the most significant components of wheel squeal. The next largest increases of 152 percent and 115 percent were obtained at 6,417 Hz and 3,823 Hz, respectively. The modal damping at 1,453 Hz was also improved significantly, but

TABLE 1 Lateral modal frequencies and damping factor estimates for treated and untreated Bochum 54 wheel

Mode	Untreated Frequency - Hz -	Untreated Damping Ratio	Treated Frequency - Hz -	Treated Damping Ratio	Damping Ratio Increase - %
1	102	6.33	100	7.28	15
2	530	1.47	518	1.76	20
3	1453	0.52	1453	0.68	31
5	2588	0.38	2554	1.58	316
6	3823	0.33	3808	0.71	115
7	5111	0.28	5129	0.40	43
8	6417	0.23	6448	0.58	152
9	7222	0.17	7827	0.26	53
10	9018	0.16	9057	0.18	13

not nearly as much as the above modes. This last mode is still a significant component of wheel squeal. Thus, although these wheel vibration absorbers are described as tuned spring-mass systems, they provide damping over a broad range of frequencies.

The lateral frequency response functions for the Bochum 54 tire are presented in Figure 4 for the undamped and treated wheel. These curves indicate that the lateral response of the treated tire is little different from those of the untreated tire below 2,000 Hz. Above 2,000 Hz, and especially at 2,600, 3,800, and 6,400 Hz, the resonant response of the treated tire was much reduced relative to that of the untreated tire. These data suggest that wheel squeal would be inhibited at these frequencies with the wheel vibration absorbers in place. However, the apparent lack of effectiveness at 1,400 Hz may prevent complete control of sustained wheel squeal at that frequency. However, the field data discussed later in the report indicate that squeal at this frequency was less with the vibration absorbers installed than without.

The spacing of the peak frequencies in the frequency response spectrum is uniform above 2,000 Hz. The response of the tire is not predicted by standard Bernoulli–Euler beam theory with dispersive bending wave propagation velocity. Timoshenko's theory of bending with rotary inertia and transverse shear better describes the response of the tire (*2*).

The radial modes of the tire are less easily identified, and many of the modes had frequencies close to those of the lateral modes. The peaks in the radial response that correspond to the frequencies of lateral modes are likely due to rotation of the tire about its neutral axis. This rotation produces a radial component of vibration at the edges of the tire. Excitation of the lateral bending modes is dependent on where the tire is struck relative to its neutral axis. The radial modes at frequencies of 2,489 and 2,495 Hz for the untreated and treated tires, respectively, are radial "ring" modes involving propagation of extensional waves around the tire, rather than bending. The wavelength of the extensional wave is equal to the circumference of the tire, measured at the centroid of the tire cross-section. For example, assuming a Young's modulus of 30,000,000 psi and density of steel of 450 lb/ft^3, the propagation velocity would be 211,000 in./sec. Assuming a centroidal diameter of 27 in., the computed ring frequency is 2,488 Hz, almost exactly as measured.

The relationship between the lateral bending modes at 2,490 Hz and a radial ring frequency at very similar frequency has not been explored, but is likely to have a pronounced effect on the tire's propensity to squeal. The radial motion of the tire will load and unload the contact zone, and this is likely to result in modulation of the friction force between lateral motion of the tire and rail. If this modulation frequency is close to a lateral bending mode frequency, the propensity for driving the lateral bending mode should be great. An interesting test would be to modify the lateral bending stiffness of the tire and drive the lateral bending mode away from the radial ring frequency to avoid any coupling and observe whether there is less wheel squeal.

Modulation of wheel/rail contact forces caused by rotational motion of the tire tread about its centroid, in turn caused by lateral bending of the tire, has been identified in the literature as affecting wheel squeal. If the contact zone was at the field side of the running surface, the interaction was predicted to exacerbate wheel squeal. On the other hand, if the contact zone was at the gauge side of the tread, wheel squeal was predicted to be inhibited or reduced. Thus, wheel profile and maintenance appear to be important in controlling wheel squeal related to various modes of vibration of the tire (*3*).

TABLE 2 Radial modal frequencies and damping factor estimates for treated and untreated Bochum 54 wheel

Mode	Untreated Frequency - Hz -	Untreated Damping Ratio	Treated Frequency - Hz -	Treated Damping Ratio	Damping Ratio Increase - %
1	531	1.70	519	1.53	-10
2	1128	1.03	1103	0.94	-9
3	1966	0.80	1950	1.03	29
5	2131	0.92	2089	1.34	46
6	2489	0.53	2495	1.97	272
7	3024	0.50	3072	0.39	-22
8	3115	0.19	3188	0.07	-63
9	3475	0.42	3459	0.68	62
10	3825	0.31	3808	0.42	35
11	4099	0.88	4061	0.19	-78
12	4209	0.55	4224	0.33	-40
13	5488	0.42	5471	0.40	-48
14	6412	0.17	6462	0.46	170

While the radial modes are of interest, rail vibration is probably most influenced by the radial anti-resonance of the tire, as contact forces would be expected to be greatest at these frequencies. The input or driving impedance of the tire in the radial direction is greatest at the anti-resonance frequencies. Figure 5 illustrates the radial frequency response function for the tire. The anti-resonances of the tire are identifiable as troughs in the frequency response function. At high frequencies, they appear to be less pronounced with the vibration absorbers than without. However, the absorbers appear to have little effect on the anti-resonances at 800 and 1,250 Hz, the frequencies at which wayside noise is greatest. This observation is *not* consistent with the rail vibration levels measured at tangent track with and without wheel vibration absorbers. The tangent track rail vibration was moderately less with the wheel vibration absorbers installed relative to the untreated wheel condition in the important frequency range of 500 to 2,000 Hz. More research in this area would appear to be desirable to define the roll of wheel damping on rail vibration.

The radial resonance frequencies are affected significantly in the 2,000- to 3,300-Hz range by the vibration absorber. At other frequencies, the frequencies are not changed significantly. At frequencies above 3,300 Hz, the responses are lower with the wheel vibration absorbers compared with those without. Below 2,000 Hz, the wheel vibration absorber had little effect on the radial resonance peaks of the tire. To the extent that the most significant range of wayside rolling noise is between 250 and 2,000 Hz, these results would suggest that the absorbers would have little effect on A-weighted noise radiated by the radial vibration modes of the tire, consistent with observations in the field.

RAIL VIBRATION ABSORBER TESTS

Tests were conducted on one of the rail vibration absorbers obtained for the project. The characterizing of a rail vibration absorber under laboratory conditions is difficult. The absorber is intended to be used on an infinite section of rail that has a continuous spectral response, while testing in a laboratory necessarily requires installation on a short piece of rail that has discrete modes. The approaches described here were intended to identify the damping capacity of the rail vibration absorber at various modal frequencies of the rail section. Unfortunately, the addition of the rail vibration absorber, including its mass, modified the modal frequencies of the rail, thus complicating the interpretation of the data. Nevertheless, the approaches are indicative of absorber damping.

The first test involved measuring the modal response of a 6-ft-long section of RE 115 rail with and without the rail vibration absorber mounted at its midpoint. The rail was placed on rubber blocks at the approximate location of the

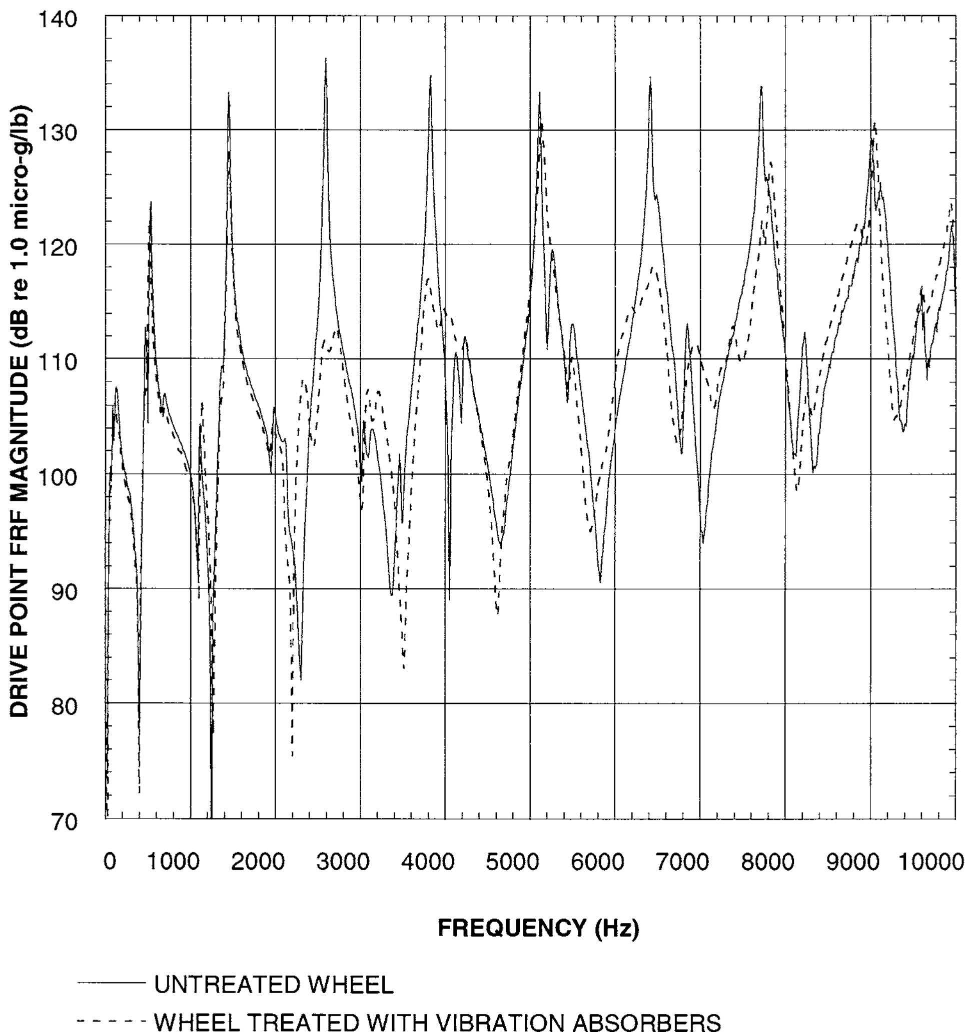

Figure 4. Lateral frequency response of the Tri-Met Bochum 54 tire.

first free-free bending mode nodal points. The test results are presented in Table 3. Without the rail vibration absorbers, the damping of the rail includes the material damping of the steel and the end-support conditions. The damping ratios were considerably higher with the vibration absorber attached than without. The damping thus obtained included the damping provided not only by the viscoelastic elements, but also by the friction between mating surfaces and bolted connections.

Assuming a modal mass of one-half the mass of the rail, or about 50 kg, the damping capacities of the absorber were estimated. The results are given in the extreme right-hand column of Table 3. The assumption of one-half the rail mass for the modal mass is appropriate for sinusoidal mode shapes in pure bending, but some of the modes involve torsion of the rail, for which the moment of inertia should be used instead. Also, rotary inertia of the rail was neglected. The researchers have not measured the modal masses for each of the vibration modes. However, the results indicate the order of magnitude of the damping capacity provided by the rail vibration absorber. The damping capacities obtained for the first few bending modes are probably most representative and are about 190 kg/s to 1000 kg/s.

The damping ratios were estimated with a modal analysis package, and there are significant uncertainties involved with this procedure for determining damping. In particular, with a lightly damped bare rail, the damping should be determined by measuring the log-decrement of the response to an impact, filtered with a one-third octave filter. With a short response period, such as was achieved with the vibration absorber attached, the response is obtained with the modal analysis program by fitting the frequency response at resonance to an idealized frequency response curve for a damped single-degree-of-freedom oscillator tuned to the modal frequency. The modal analysis package is probably accurate for the damped condition, but inaccurate for the undamped condition. However, it is the damped condition that is of principal interest.

During the tangent track tests, the wayside noise preceding and following train passage was attenuated or eliminated

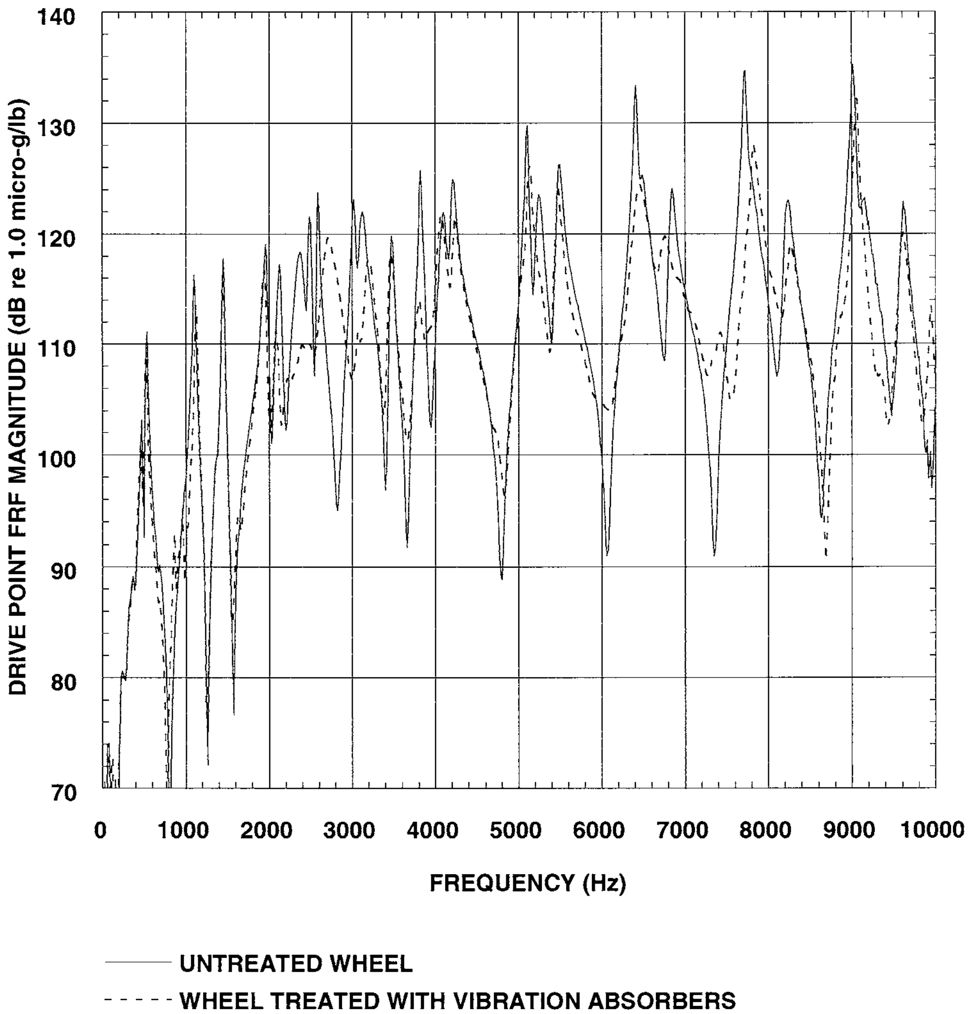

Figure 5. Radial frequency response of the Tri-Met Bochum 54 tire.

with the rail vibration absorbers. However, during passage, the wayside and under-car noise levels were higher with the rail vibration absorbers attached than without. The increase was concentrated primarily in the 1,250 Hz one-third octave band, with little change in other bands. This unexpected and contradictory result was believed to be due to bending of the mounting plate of the rail vibration absorber. A second test was employed to identify mounting plate bending on a short 25⅝–in. section of 115 lb/yd rail. The rail stub and absorber were placed on a 3-in.-thick rubber block, with the rail stub on its end. The fundamental resonance frequency of the short section of rail in bending about its x-x axis was of the order of 2,000 Hz. Introduction of the vibration absorbers greatly modifies the response of the rail, not only changing the damping of the rail, but also modifying the frequency and mode shape. The modal responses of this treated piece of rail were numerous and complex. Thus, it is difficult to identify a treated mode with an untreated mode. The results of the analyses are tabulated in Table 4.

The stiffness and mass of the rail flange have to be considered with those of the mounting plate when discussing mounting plate resonance. With no absorber attached, a transverse plane-strain mode was identified at a frequency of about 1,471 Hz, which involves opposed rotation of the rail flange and head with bending of the web about the longitudinal axis of the rail. With the absorber attached, a mode at a frequency of about 1,310 Hz looks particularly interesting and may be related to the above plane-strain distortion mode. This mode involves bending of the mounting plate in a direction along the rail, but with out-of-phase motion of one side of the mounting plate relative to the other. There is substantial bending of the bottom plate, which should be heavily damped by the absorber block clamped to the top of the rail flange. The damping factor at this mode was about 2.2 percent. Another mode was identified at 1,248 Hz for the short stub of rail and vibration absorber. While this mode is closer to the center frequency of the 1,250 Hz one-third octave band in which the noise

TABLE 3 Modal parameters for 6′-long section of rail with and without rail vibration absorber fitted

Mode	Untreated Rail Frequency - Hz -	Untreated Rail Damping Ratio - % -	Treated Rail Frequency - Hz -	Treated Rail Damping Ratio - % -	Damping Ratio Increase - %	Estimated Damping Capacity - 10^3 kg/s -
1	311	1.4	301	2.57	86	190
2	1,320	0.46	1,320	1.70	270	830
3	1,910	0.30	1,841	1.30	300	1,200
4	2,251	0.37	2,269	1.16	224	1,400
5	2,504	0.16	2,583	1.60	900	1,600

TABLE 4 Rail vibration absorber test result—rail vibration absorber mounted on 25⅝″ RE 115 lb/yd rail

Mode	Untreated Rail Frequency - Hz -	Untreated Rail Damping Ratio - % -	Treated Rail Frequency - Hz -	Treated Rail Damping Ratio - % -
1	598	0.20	493	3.0
2	936	0.28	877	5.6
3	1471	0.04	1478	2.2
4	1547	0.11	1544	1.4
5	1915	0.10	1859	1.7
6	1966	0.12	NE	NE
7	2081	0.12	2105	0.28

NE: Not Existent

increase was observed, it is very asymmetrical and may not be the most significant mode.

The principal conclusion of these tests is that there are a number of vibration modes, or resonances, of the rail and rail vibration absorber, which involve a combination of rail flange bending, mounting plate bending, and actual absorber resonances. Each of these resonances is capable of absorbing vibration energy, provided that there is significant participation by the absorber bodies. Unfortunately, one or more of the resonances associated with the mounting plate may actually increase noise radiation.

After reviewing the test data and mode shapes, the best approach to laboratory testing would appear to involve mounting the absorber on a very long section of rail, possibly with periodic rail supports as in actual track. This approach would give a realistic assessment of absorber vibration modes. Using a short section of rail, whether of 2 ft or 10 ft in length, artificially introduces various modes of vibration that are not present in an infinitely long rail. Nevertheless, the measurement of the changes in damping ratios brought about by the absorber for modal resonances of a short rail section of rail should give reasonable estimates of vibration energy absorption.

The pass-band vibration spectrum that extends from about 500 Hz to 2,000 Hz for untreated periodically supported rail appears to be effectively eliminated by the rail vibration absorbers at the tangent track test section at Tri-Met, where rail head vibration was actually reduced with the rail vibration absorbers relative to without. Thus, noise radiation from the rail was probably reduced, but the possibility exists that the resonating mounting plate of the absorber may have increased the wayside noise during train passage. Additional vibration measurements of rail flange and absorber mounting plate vibration would be necessary to confirm this.

CHAPTER 7

ROLLING NOISE REDUCTIONS

Rolling noise reductions were measured at both the NJT and at the Portland Tri-Met. Detailed one-third octave and narrow-band analyses were performed on the NJT test data, and one-third octave band analyses were performed on the noise and rail vibration data collected at the Portland Tri-Met. The NJT tests included measurements of wheel vibration absorber performance, while the Portland tests included measurements of both wheel and rail vibration absorbers in combination.

One-third octave band sound exposure levels (SELs), rather than maximum sound levels as normally used for rail transit noise analysis, are used extensively in this report. The sound exposure levels are directly related to the energy equivalent level (L_{eq}) occurring over some time period. That is, the L_{eq} may be obtained from the sound exposure level by adding ten times the logarithm of the ratio of the number of events divided by the time period in seconds. The sound exposure level is thus directly related to the current FTA guidelines for environmental impact analysis, which recommend L_{eq} and related Day-Night sound levels (L_{dn}) for environmental noise analysis. The SEL removes uncertainties or ambiguities related to integration times, as discussed below, and is thus desirable for research purposes, as well as for current environmental analysis procedures.

Maximum sound levels were also measured, although they are not reported extensively here. The maximum sound levels of noise occurring during vehicle passage can be obtained from the corresponding SEL by subtracting 10 times the logarithm of the duration of the passage. For a single car of length 100 ft traveling at 50 ft per second, the adjustment would be 3 dB. The maximum level should be defined with respect to the passby duration. Common practice is to analyze the sound data during the passby duration of, perhaps, 2 sec. The maximum level as measured with a fast or slow meter response of 0.125- or 1-sec averaging time, respectively, would be slightly higher than the 2-sec averaged data. The maximum level determined by subtracting 10 times the logarithm of the passby duration would be higher than the 2-sec analysis, but probably comparable with measurements obtained with a slow meter response. The data for maximum and SEL sound levels support this.

An analogous approach is taken for representing rail vibration data. Rail vibration data are presented as vibration exposure levels (VELs). The VEL is simply 10 times the logarithm of the integral with respect to time in seconds of the square of the vibration velocity over the entire duration of the event, and it is directly comparable with the wayside SEL. Thus, a direct measure of the wayside noise level as a function of rail vibration is obtained without concern over integration times.

NEW JERSEY TRANSIT

The rolling noise reductions for the wheel vibration absorbers were measured at a section of ballast-and-wood-tie tangent track paralleling Heller parkway. The section was about 200 ft in length and was located a few hundred feet south of the Heller Parkway Station. Noise was measured at two locations separated by 100 ft on the west side of the alignment at a distance of 26 ft from the nearest track center (southbound track). The microphones were positioned above an earthen berm, at about 10 ft above the top-of-rail. Vibration data were measured at the outside rail of the nearest track, opposite each microphone location. Data were taken for both revenue and test PCC car passbys.

Continuous recordings of car interior noise were made for the test car at the center of the car, 4 ft above the floor. The test car was run in revenue service.

Averaged one-third octave wayside SELs are compared in Figure 6 for Car No. 9 without absorbers, Car No. 9 with wheel vibration absorbers, and Car No. 11 with PCC super-resilient wheels. The results of the tests with and without vibration absorbers indicate virtually no difference in wayside noise. Greater difference was observed between the Bochum 84 wheel and the PCC Super Resilient wheel than between treated and untreated conditions of the Bochum 84 wheels. The PCC Super Resilient wheel produced more noise above 400 Hz than did the Bochum wheel. The higher levels obtained for the PCC Super Resilient wheel may have been due to greater roughness of the tire tread compared with that of the Bochum 84 tread, which was much newer. Between 50 and 160 Hz, the wayside noise from the PCC Super Resilient wheel was less than that produced by the Bochum 84 wheel. The reason for the difference has not been identified, but may be related to differences in stiffness of the wheels.

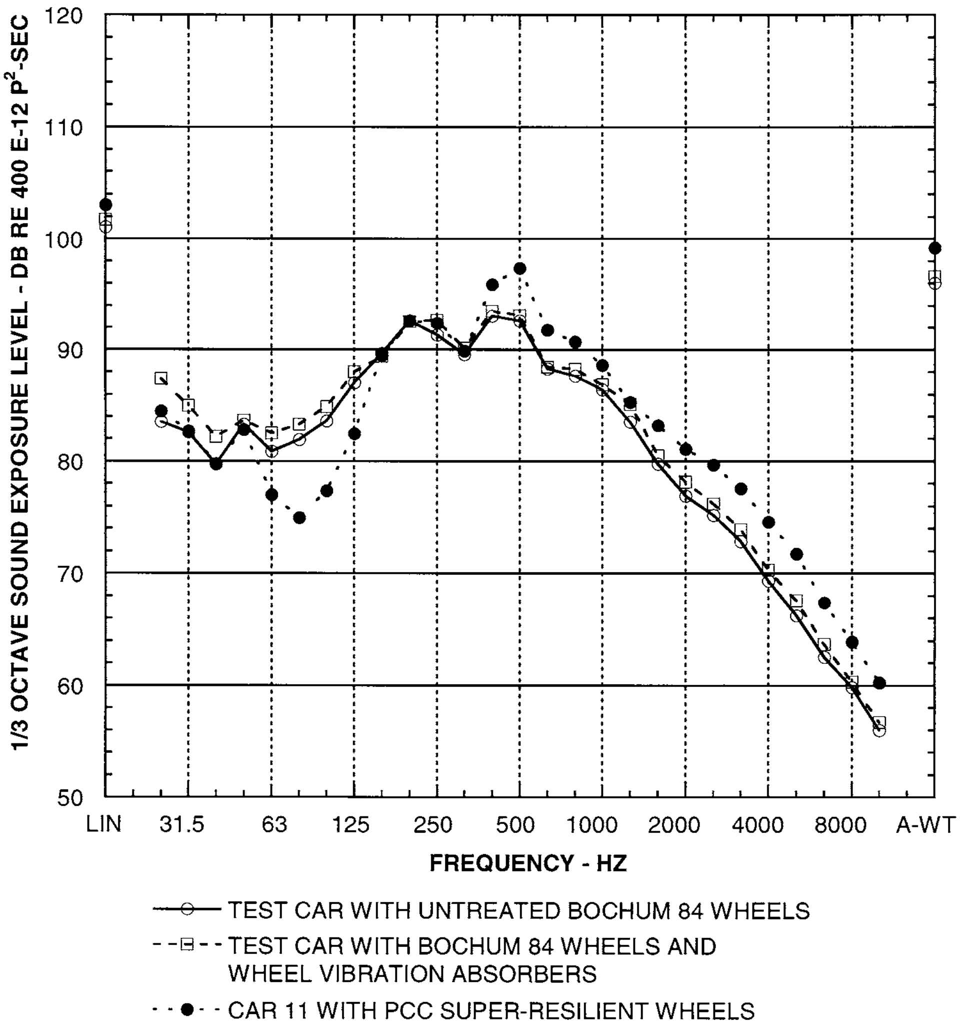

Figure 6. Wayside noise levels for treated and untreated wheels at the tangent track section at the NJT.

PORTLAND TRI-MET

Rolling noise reductions were measured at the Portland Tri-Met for the following test conditions: (1) no treatment, (2) wheel vibration absorbers, (3) rail vibration absorbers, and (4) rail and wheel vibration absorbers. The order of testing differed from the above, as the rail vibration absorbers were installed prior to any testing at the tangent track section. Data were taken for both the test vehicle and revenue service trains. Only the test vehicle data are reported here.

The rail vibration absorbers alone were tested on the night of July 27, 1999, between 11 p.m. and 4 a.m. This was the last test of the tangent track test series. The wheel vibration absorbers were attached to the test vehicle, and the combined treatment was tested on the night of July 30, 1999, between 11 p.m. and 4 a.m. The rail vibration absorbers were then moved to the curved track section during the month of August, and the wheel vibration absorbers were tested alone at the untreated tangent track on August 24, 1999, during the hours of 9 a.m. to 4 p.m. The wheel vibration absorbers were then removed from the vehicle, and the combined untreated rail and wheel condition was tested on August 26, 1999, between 9 a.m. and 4 p.m. during revenue service hours.

Wayside Noise Test Results

Wayside A-weighted SELs are for the test vehicle summarized in Table 5. Figure 7 includes one-third octave spectra for wayside noise measured at 32 ft from track center for 45 mph passes with the test vehicle. Data for the Tri-Met tangent track tests were taken at other train speeds and distances and for revenue service trains, but are not reported here.

TABLE 5 Wayside noise levels at tangent track test section

	SEL at 32 Feet			SEL at 57 Feet		
Treatment	**25 mph**	**35 mph**	**45 mph**	**25 mph**	**35 mph**	**45 mph**
None	73	77	80	68	72	75
Wheel	72	76	79	66	72	74
Rail	74	78	81	NA	NA	NA
Wheel & Rail	74	77	81	71	74	77

NA: Not Available

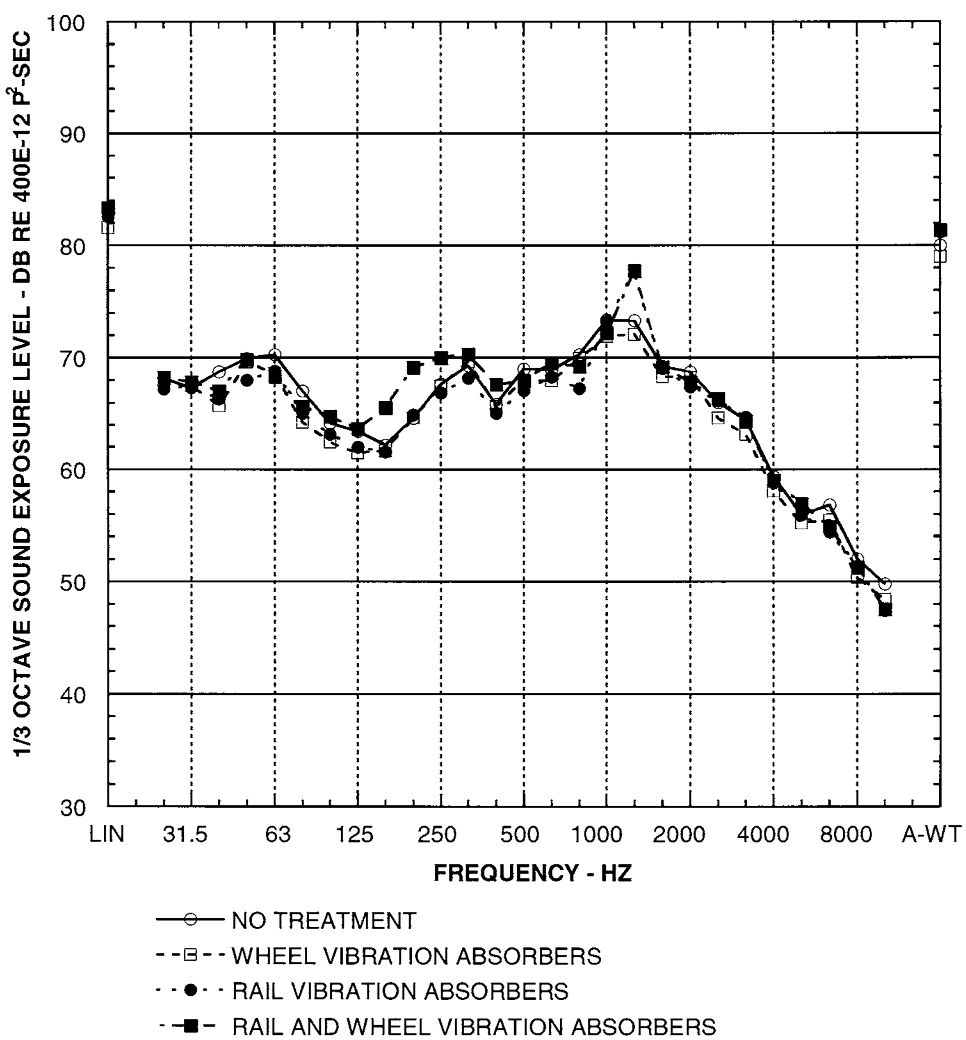

Figure 7. Wayside SELs for Tri-Met Car 115 vehicle on tangent ballast-and-tie track for various treatment conditions.

The data shown in Table 5 indicate at most 1 dB of noise reduction for the wheel vibration absorbers relative to the untreated case. The one-third octave band spectra in Figure 7 indicate very little difference in wayside noise levels for the treated wheel condition relative to the untreated condition, except for a very slight reduction at 1,000 Hz. Qualitatively, there was no difference in wayside noise.

The A-weighted noise levels presented in Table 5 indicate a modest increase of wayside noise levels for the treated rail condition relative to levels for the untreated condition and treated wheel condition. As mentioned above, this increase appears to be due to a resonance of the vibration absorber mounting plate. The wayside one-third octave noise spectra shown in Figure 7 indicate that the increase is due almost entirely to an increase in the 1,250 Hz one-third octave.

The noise levels for the combination of treated rails and treated wheels were also slightly higher than those for the untreated condition and almost identical to those for the treated rail condition. The one-third octave spectra provided in Figure 7 indicate very little difference between the treated rail and the combined treated rail and treated wheel conditions. Again, the 1,250 Hz one-third octave band level is higher for the combined treatment relative to that for the no-treatment and treated wheel conditions. The increase at 1,250 Hz appears to contribute the bulk of the noise increase.

Under-Car Noise Results

The relative noise reductions observed under the test vehicle were very similar to those observed at the wayside for the various test configurations. One-third octave spectra of under-car root-mean-square sound pressure levels are presented in Figure 8 for the 45 mph test runs. (The root-mean-square level

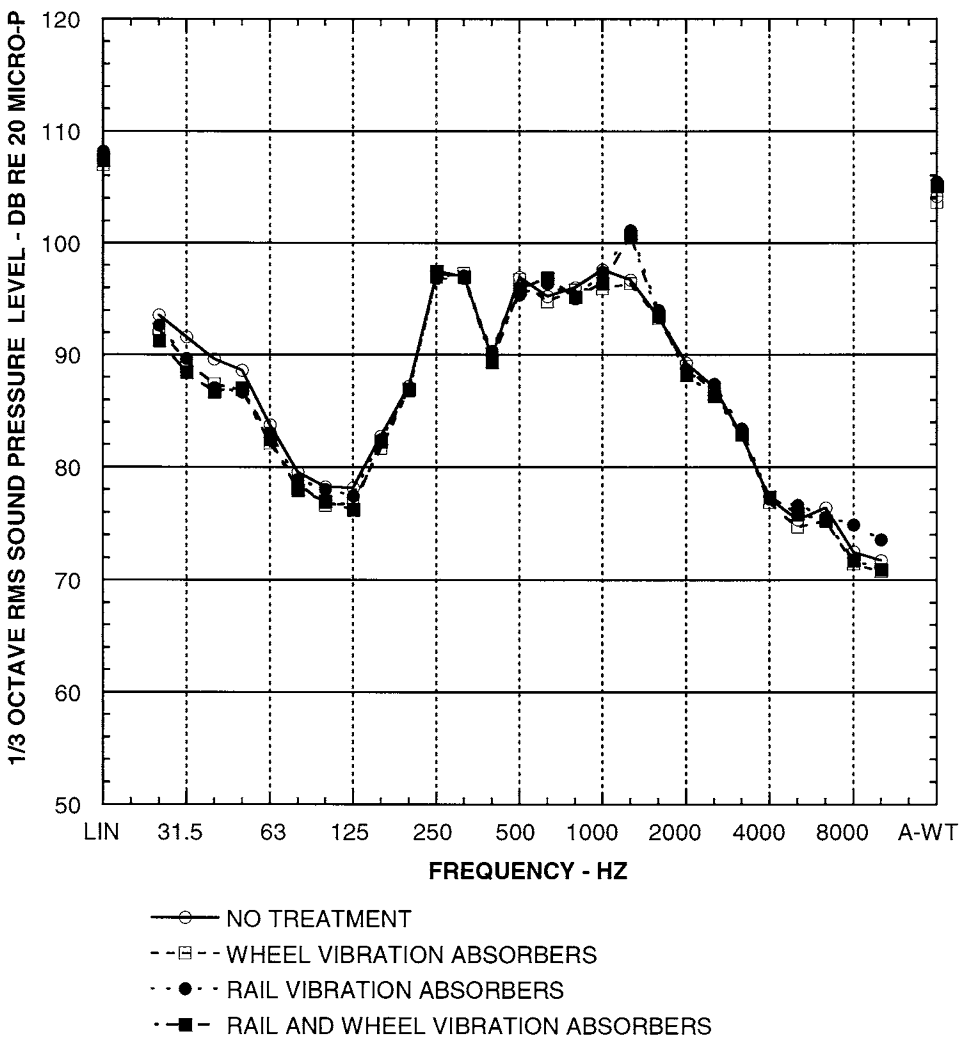

Figure 8. Under-car noise at 45 mph at tangent track test section for various treatment conditions.

differs from the single-event level, is the level in decibels of the sound pressure squared averaged over the measurement duration, and is representative of the continuous sound level.) These spectra are energy averages of the data obtained at both the A and B trucks. The under-car noise levels were noticeably higher in the 630 and 1,250 Hz one-third octaves with the rail vibration absorbers installed relative to the untreated condition and the treated wheel condition. This result is almost identical to that observed at the wayside. No such difference was observed at the control track.

Examples of under-car one-third octave noise levels measured at the tangent control track, adjacent to the treated track, are presented in Figure 9. The data shown for the rail vibration absorber test are for the untreated control section with untreated wheels. The data for the rail and wheel vibration absorber test are also for untreated rail, but with wheel vibration absorbers installed on the vehicle. These data indicate that little differences in sound levels were observed for the various tests. However, the under-car noise levels at the leading and trailing trucks on the control track were slightly lower by about 1 to 2 dB in the 1,000 Hz one-third octave for the treated wheel conditions relative to those for the untreated wheel conditions. While this difference is slight, it was observed for both the treated rail and untreated rail tests at this untreated control track and was also observed at speeds of 35 and 25 mph. The result is thus consistent and indicates a very slight reduction of noise in this one-third octave that might be attributed to the wheel vibration absorbers.

The control test data shown for the untreated rail and untreated wheel condition are virtually identical to the control test data shown for the treated rail test with untreated wheels, as it should be for the control track. Similarly, the

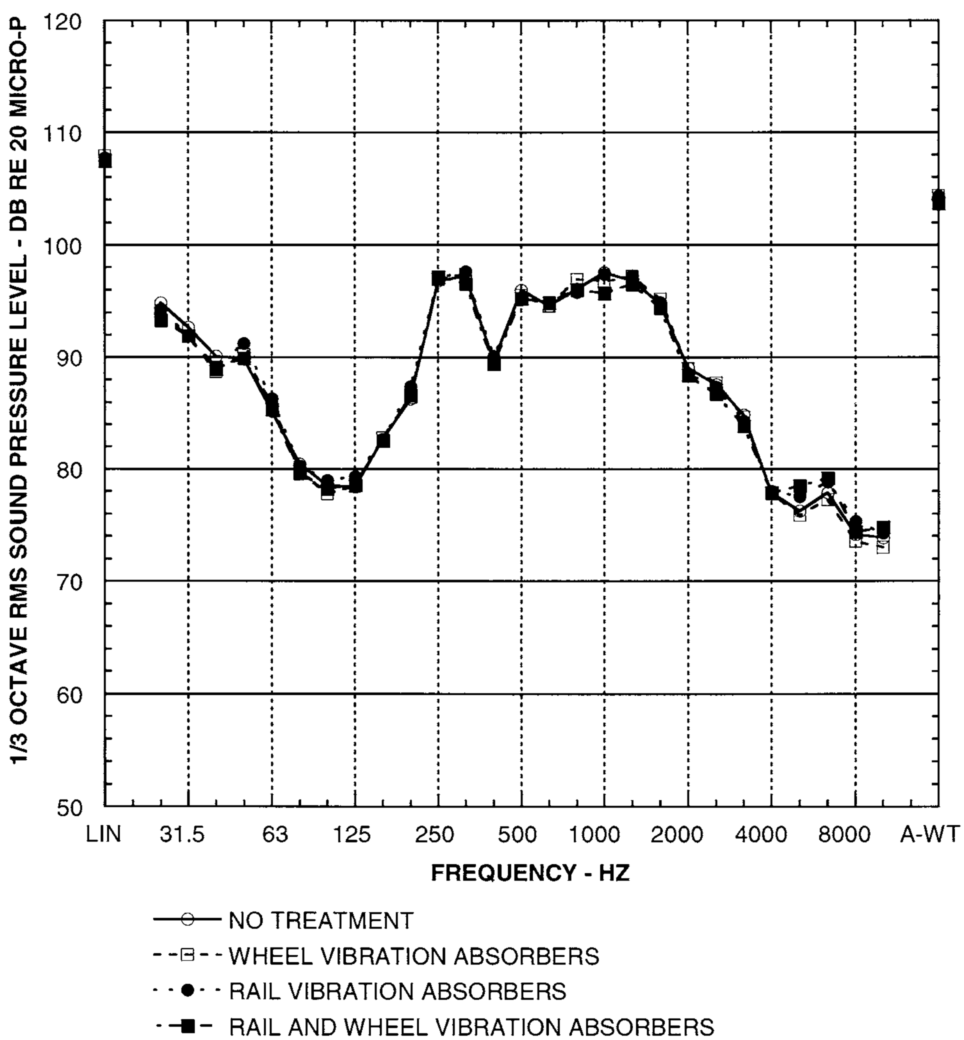

Figure 9. Under-car noise at tangent control track for various tests.

control track data shown for the treated wheel conditions are virtually identical to the treated rail and untreated rail tests. That is, the control track data indicate little or no variation in wheel condition between the treated rail and untreated rail tests in spite of approximately 1 month of operation of the vehicle in revenue service between the tests.

The data for the center truck were less reliable than the data obtained at the motored trucks because of a shift in microphone sensitivity, possibly related to wetting of the instrumentation at the water-sprayed curves. However, the center truck data, not shown here, are consistent with the motored truck data.

While the noise reductions for the wayside and under-car noise are disappointing, the results obtained for rail vibration reduction are quite dramatic and reveal basic information concerning the relative importance of the wheel and rail in the noise radiation process, as discussed below.

Rail Vibration Levels

Examples of one-third octave vertical and transverse vibration spectra measured at the rail head are presented in Figures 10 and 11, respectively, for a speed of 45 mph, the highest speed tested. The data are one-third octave velocity exposure levels in decibels relative to 10^{-12} in.^{2}/sec. These data are analogous to the wayside SEL and may be compared directly with the SEL. The velocity exposure level is independent of the integration time used during analysis. Rail vibration typically varies greatly with time and integrating over the entire passby signature provides a measure of the total vibration energy associated with the event, independent of the analysis window.

Rail head vertical and transverse vibration levels were significantly less with the wheel vibration absorbers installed than without. The difference was greatest for the untreated rail condition. The difference persisted with the treated rail,

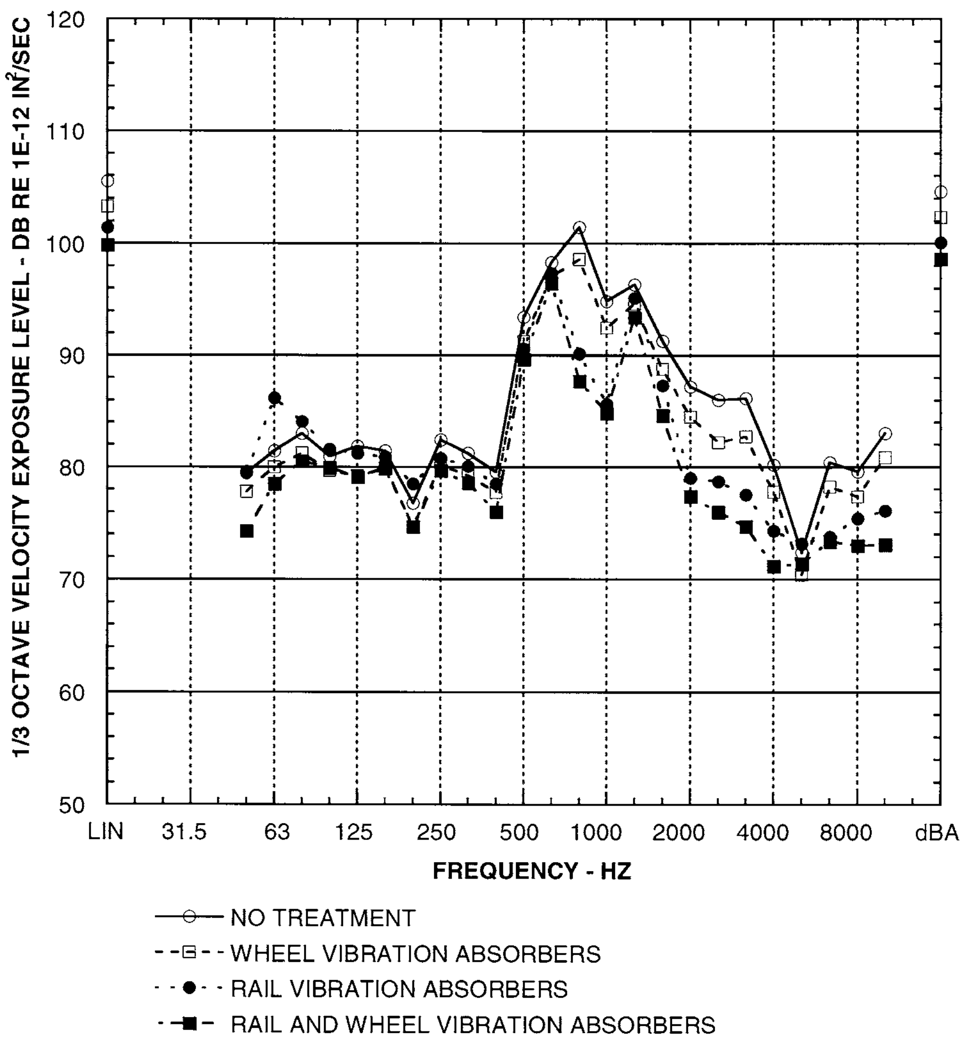

Figure 10. Rail head vertical vibration.

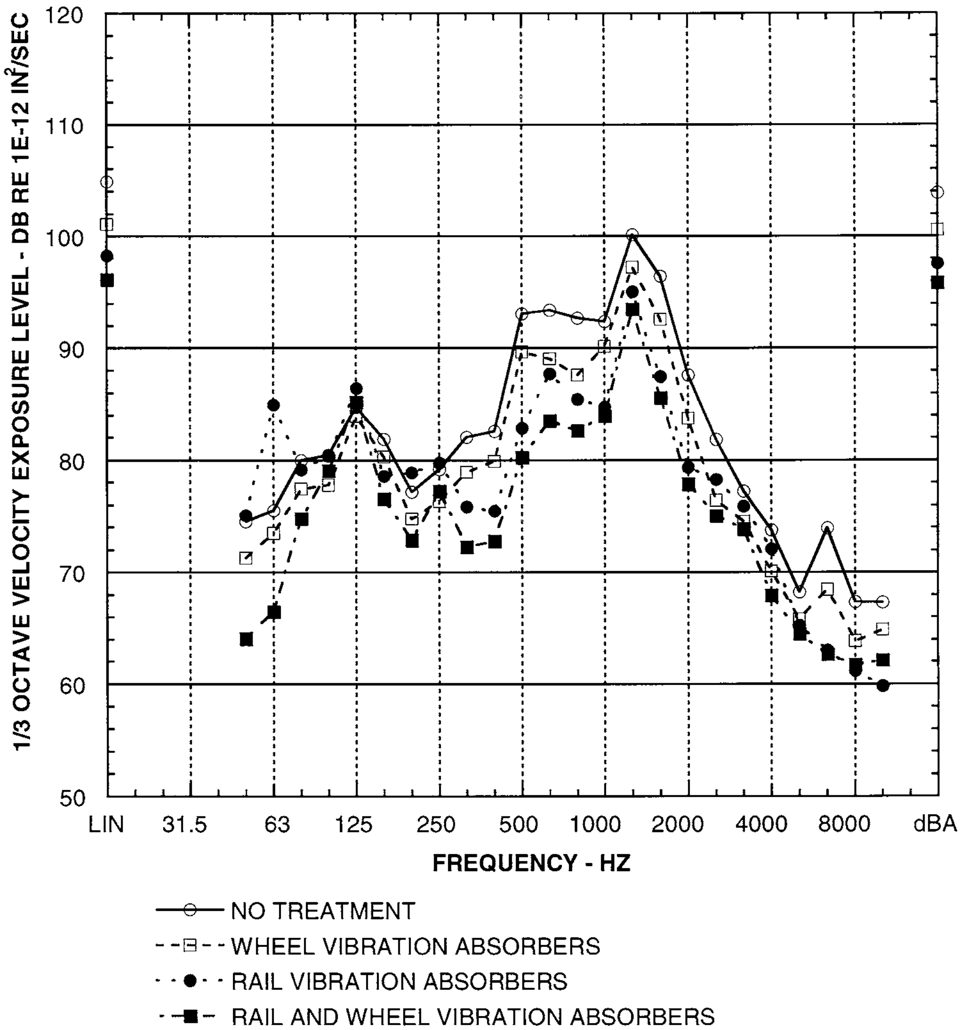

Figure 11. Rail head transverse vibration.

but the spectra of rail vibration were very similar for the untreated and treated wheel conditions. The peak frequency in the vertical and transverse rail vibration occurred at 800 and 1,250 Hz, respectively, for the untreated rail and wheel (no-treatment) and the treated wheel conditions.

Rail head vibration levels were significantly lower with the rail vibration absorbers installed relative to levels without. For the vertical direction, the peak at 800 Hz was eliminated, leaving two peaks, one at 630 Hz and a second at 1,250 Hz. The 1,250-Hz peak may be associated with the wayside noise level increase discussed above and is likely related to mounting plate vibration. Except for these two frequencies, the rail vibration was less with the rail vibration absorbers relative to without over the frequency range of 500 Hz up to the limit of the measurement at 10,000 Hz. Transverse rail head vibration was also much less for the treated rail condition relative to the untreated rail condition, with or without wheel vibration absorbers, over the frequency range of 316 Hz to the upper limit of the measurement at 10,000 Hz. A one-third octave spectral peak exists at 1,250 Hz for both the treated and untreated rail conditions. The peak was not shifted with introduction of the rail vibration absorber, but the level of the peak was reduced.

Figures 10 and 11 also indicate that the level of the transverse component of rail head vibration exceeds the level of the vertical component at the 1,250 Hz one-third octave. Evidently, the noise increase at this frequency with the rail vibration absorbers is related to lateral rail head vibration.

The rail vibration spectra vary significantly with train speed. At 45 mph, the peak in the one-third octave transverse vibration spectrum occurs at 1,250 Hz, while at 25 mph the peak occurs at 500 and 630 Hz. Grinding marks are evident in the rail head, and these grinding marks influence the relative significance of spectral peaks as a function of train speed. (These grinding marks are very common at transit systems. While they may produce an undesirable tonal component of noise, the noise is far more preferable than rail corrugation noise. Also, these grinding marks appear to wear away with time.)

The differences in rail vibration levels for the various treatment conditions are surprising in view of the minimal differences observed in the wayside and under-car noise data. These data suggest that the rail is not a significant radiator of wayside noise, disregarding the singing rail phenomenon that precedes and follows the passage of a train. Much of the literature assigns at least part of the wayside noise spectrum to radiation by the rail. These data would suggest that the wheel, or at least the Bochum wheel, is the only major source of wheel/rail rolling noise over the entire spectrum.

Singing Rail Vibration

At the end of testing in August, the rail vibration transducers were relocated to measure rail head vertical vibration at four points, each separated by 15 in. (one-half of the tie pitch). Thus, two transducers were located over two adjacent ties, and the other two were located in adjacent tie bays. The data were taken with the rail vibration absorbers removed. The data thus obtained indicate the presence of a pass-band characteristic for vibration propagation along the rail prior to and after passage of the train. Figure 12 presents two auto-spectra obtained between the ties at adjacent tie spaces. The pass-band begins at about 500 Hz and ends at about 2,000 Hz. A

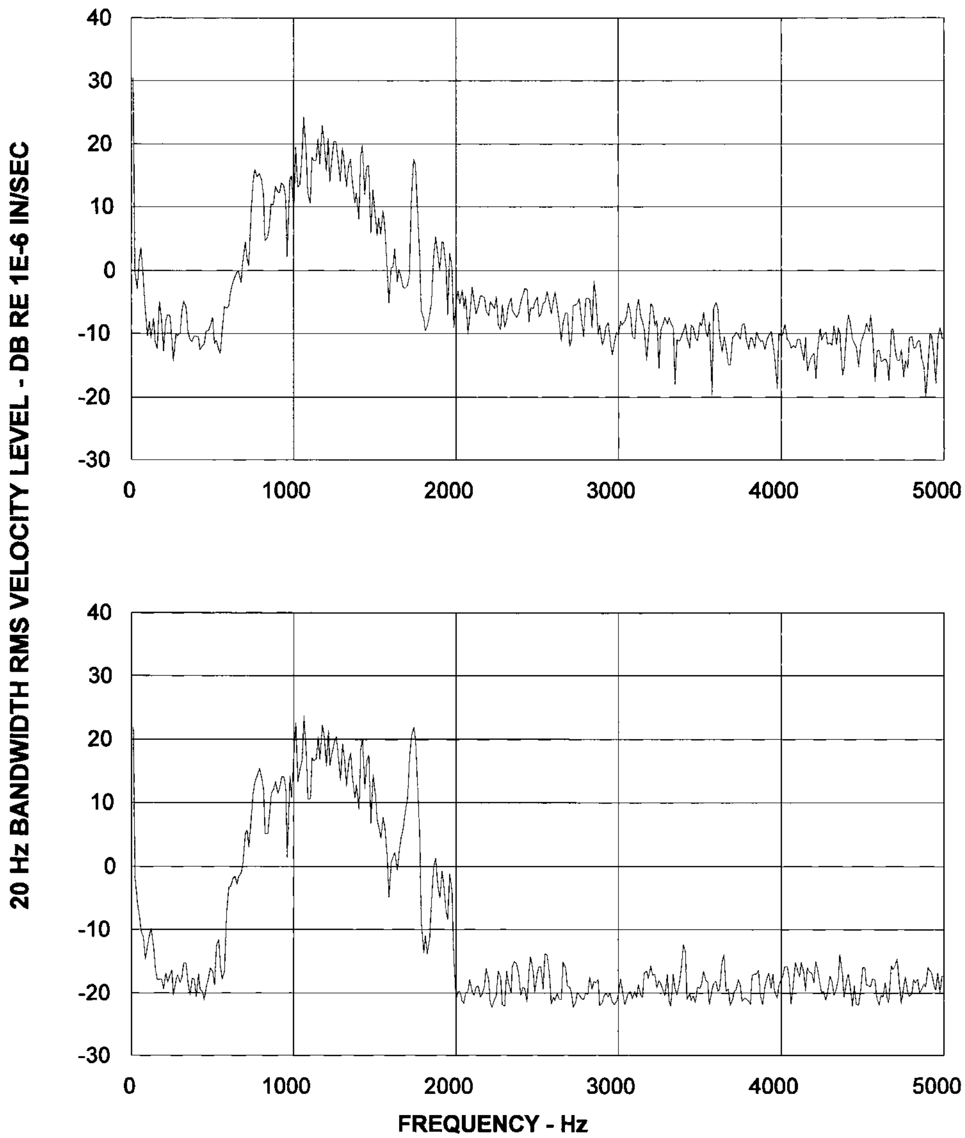

Figure 12. Auto-spectra of rail vertical vibration at two adjacent tie spaces prior to train passage.

peak occurs at about 800 Hz. A second peak occurs at about 1,750 Hz. The frequencies of both of these peaks are inside the pass-band of the vibration spectrum.

A transfer function between the two measurement points in adjacent tie spaces is presented in Figure 13. The phase angle between the two points decreases with increasing frequency, signifying propagation of vibration energy from one measurement point to the other. In this case, the energy was traveling eastward along the rail. Another transfer function (not shown here) obtained after passage of the vehicle indicates propagation in the reverse direction. The transfer function magnitude between approximately 600 Hz and 1,600 Hz is approximately 0 dB, although at the upper edge of the transmission band there is considerable variation of transfer function magnitude. The pinned-pinned mode frequency should be at the lower edge of the pass-band, which appears to be about 600 Hz. The theoretical pinned-pinned mode frequency is about 800 Hz. At 600 Hz, the phase angle is approximately 180 degrees out of phase, as expected for a pinned-pinned mode. Similarly, at 1,750 Hz, the phase angle is also 180 degrees out of phase. Outside of the pass-band, there is virtually no transmission of rail vibration.

These data confirm the existence of a pass-band and stop-band characteristic for rail vibration preceding and following passage of a train. The phenomenon is related to singing rail. Directly comparable data were not taken with the rail vibration absorbers installed. However, the absence of any audible singing rail with the absorbers installed indicates that the rail vibration absorbers effectively eliminated this mode of rail vibration transmission.

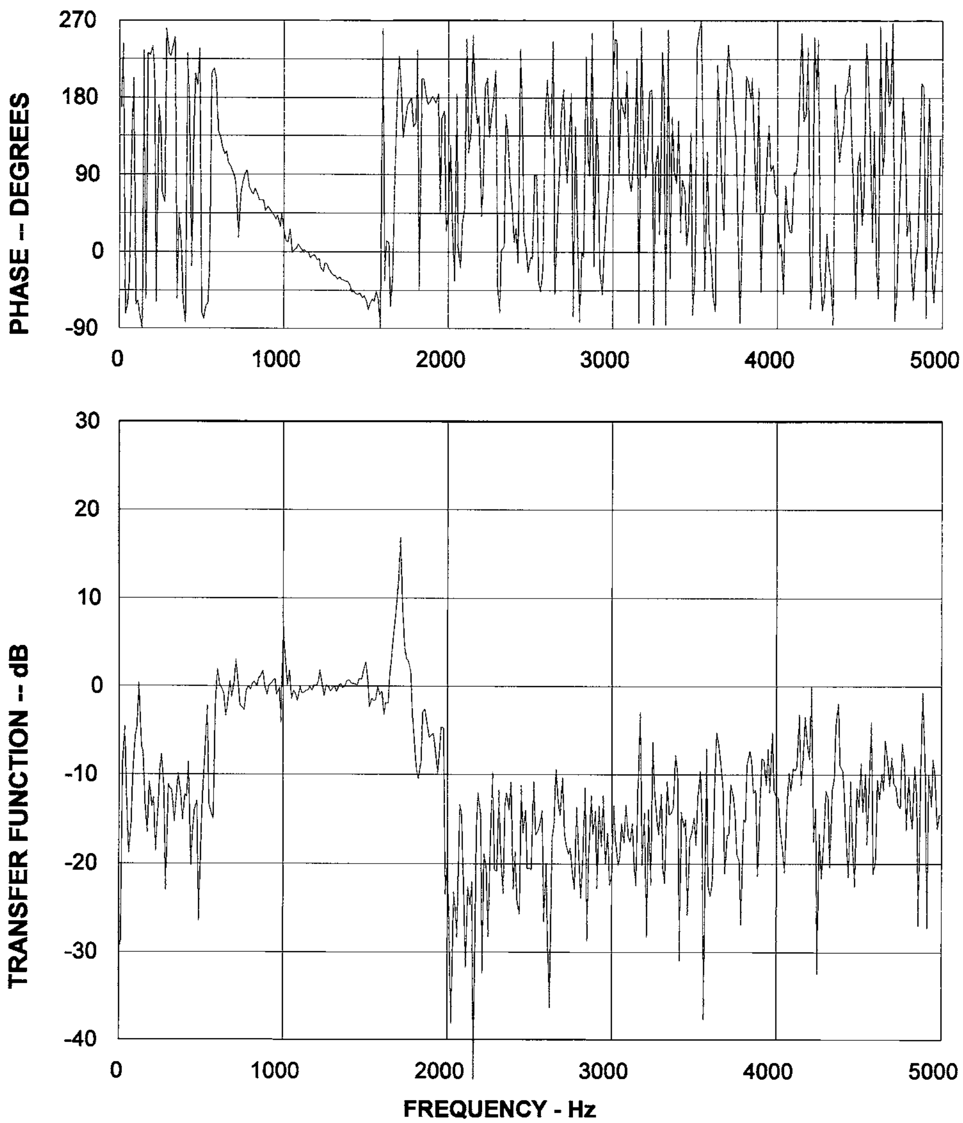

Figure 13. Transfer function between vertical rail head vibration at adjacent tie spaces without rail vibration absorbers.

Rail vertical vibration velocity exposure levels were simultaneously measured over the tie and over an adjacent bay with and without the rail vibration absorbers installed. Figure 14 includes one-third octave spectra obtained for the treated wheel condition, without rail vibration absorbers installed. Two separate passby samples are presented for each measurement point, and both are similar. However, there are significant differences between the two locations. The one-third octave velocity exposure level spectra contain peaks at both locations at 630 Hz and at 1,250 Hz. In both samples, the rail vibration levels at the mid-tie-space location were higher than those over the tie in the 630 Hz and 800 Hz one-third octaves, consistent with a pinned-pinned mode. At 1,250 Hz and higher frequencies, the rail vibration over the tie was higher.

A different vibration characteristic appears with both the rail vibration absorbers and wheel vibration absorbers installed. Figure 15 includes one-third octave band spectra measured for a single 45 mph passby with treated wheels and treated rails. The vibration energy is primarily concentrated in the 630 Hz one-third octave, but is still higher between the ties than over the tie. At 1,250 Hz, the vibration level is also slightly higher between the ties relative to that over the tie, in contrast to the untreated rail condition. Above 1,250 Hz the vibration levels are again higher over the tie relative to between the ties. Similar results are obtained at 35 and 25 mph,

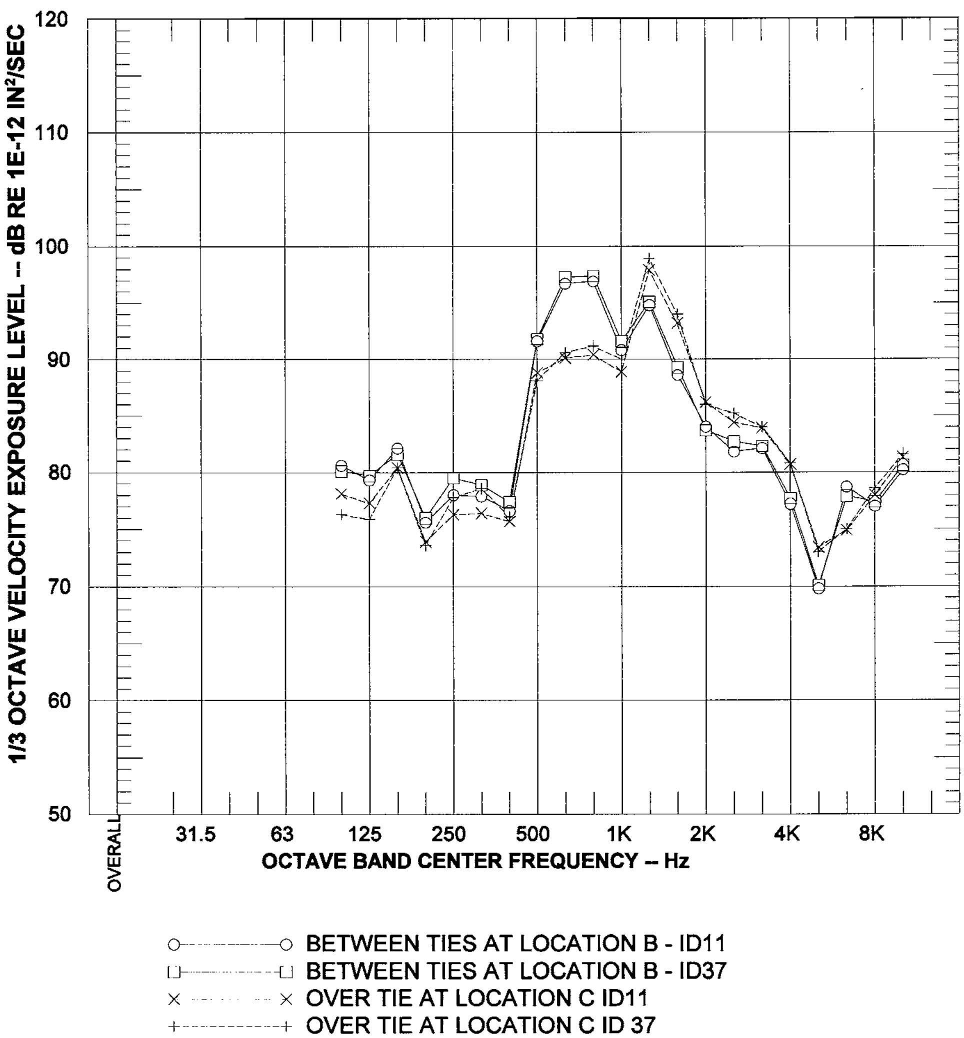

Figure 14. Rail head vertical vibration at mid-tie space and over the tie for 45 mph vehicle with wheel vibration absorbers.

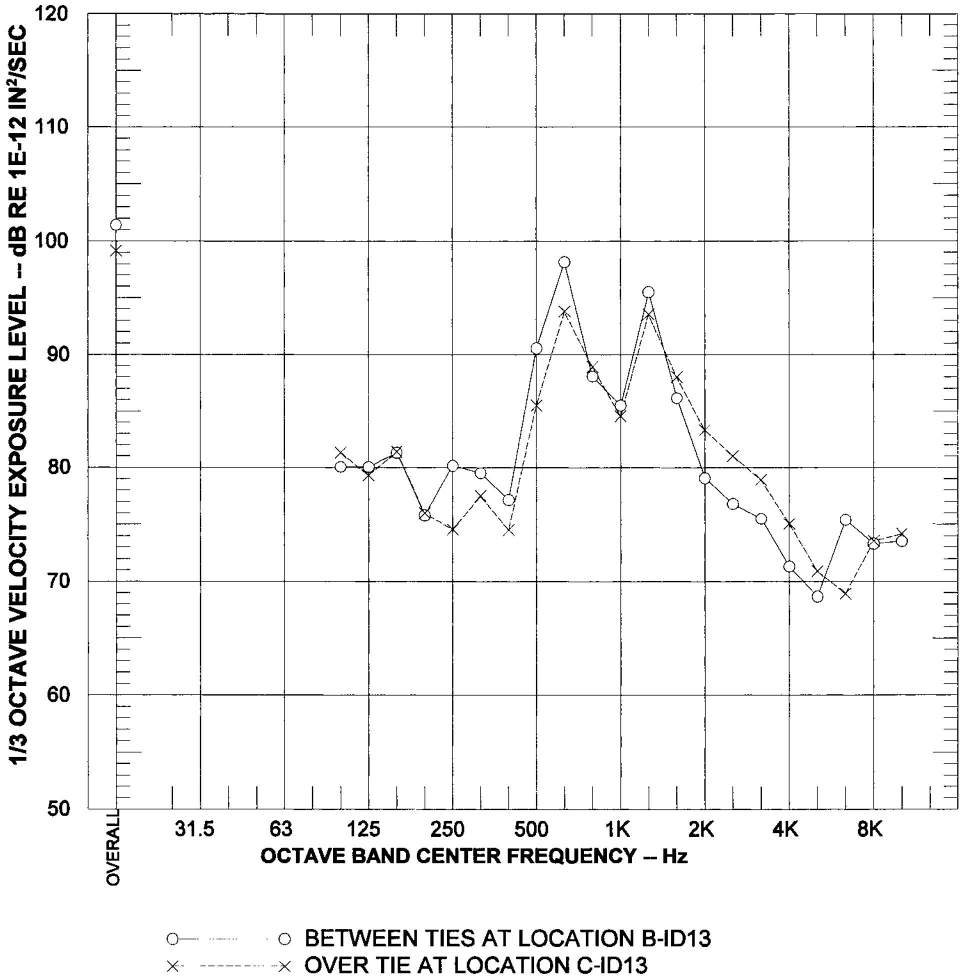

Figure 15. Vertical rail head vibration over tie and between ties for 45 mph vehicle with treated wheels on treated rail.

although the results at 35 mph were less pronounced. The rail grinding pattern in the rail head might be playing a part in the relative response of the rail to wheel rail forces over the tie and over an adjacent bay in the 1,250 Hz one-third octave band.

These data are time-integrated over the entire passage of the train, so that they include vibration energy during train passage as well as before and after. The results suggest that the pinned-pinned modes are still apparent at 630 Hz with the rail vibration absorbers installed, even though there is no transmission of vibration ahead of or behind the vehicle. These pinned-pinned modes are analogous to damped harmonic oscillators and would still respond to local excitation as the wheel passes.

CHAPTER 8

WHEEL SQUEAL NOISE REDUCTION

Wheel squeal noise and rail vibration data were collected for treated and untreated wheels at the NJT and the Portland Tri-Met systems. Additionally, noise and rail vibration data were obtained for the treated and untreated rail condition at the Portland Tri-Met's East Portal curve. These data are summarized below.

NEW JERSEY TRANSIT

Wheel squeal noise and rail vibration data were collected at the Franklin Avenue turnaround and at the Penn Central Station curve. Chronic wheel squeal produced by the vintage PCC cars with Super Resilient wheels occurs at these short radius curves. NJT had installed a set of Bochum 84 wheels on one PCC car (Car No. 6) prior to this project, and NJT purchased a second set with wheel vibration absorbers for testing under this project.

Measurement Locations

The Franklin Avenue turnaround is a 42-ft radius curve at the northern end of the system in Newark. Transit vehicles negotiate the turnaround prior to entering the Franklin Avenue Station. Microphones were positioned at both sides of the track at 12 ft from the track center. Both microphones were positioned at 5 ft above the top-of-rail.

Recordings of rail horizontal analog acceleration data at the rail head and rail web, and rail vertical acceleration data at the foot, at both rails were attempted. The rail vibration data contained ultra-sonic acceleration signals that saturated the accelerometer charge amplifiers. This development was entirely unexpected, but illustrates the complexity of rail vibration spectra under stick-slip excitation. The data were entirely unusable. However, the experience guided the successful instrumentation design for the Portland tests.

The curve at the Penn Station Loop is located in the subway surrounding the NJT maintenance shop and connects the inbound and outbound Penn Station platforms. Transit vehicle noise was measured at the walls of the subway at both sides of the track at a distance of 6.5 ft from the outside rail and 7 ft from the inside rail. Both microphones were at 5 ft above the top-of-rail.

Rail vibration was measured at both rails at the Penn Station Loop. As with the Franklin Avenue curve data, the acceleration data were clipped by ultra-sonic acceleration signals that saturated the accelerometer charge amplifiers. During the wheel vibration absorber tests, the charge amplifiers were modified to attenuate the acceleration signals, allowing a limited amount of acceptable vibration data to be recorded. However, the data were inconclusive and are not reported here.

Car interior analog noise data were also recorded on digital tape in Car No. 9. A microphone was positioned mid-way in the aisle of the car, at about 4.5 ft above the floor.

Procedures

Car No. 9 was operated in regular revenue service. Data were taken for up to five passes of the test vehicle and most of the revenue vehicles. A complete loop of the system required about 25 min, so that the data were collected over the course of 3 hours at each test location. Measurements were first made for Car No. 9 without wheel vibration absorbers attached and then with the absorbers attached.

The recorded noise data were reproduced in the laboratory and Fourier analyzed with a 20-sec averaging time. All of the Car No. 9 data were analyzed in this fashion, and some of the data were analyzed for revenue service PCC cars with Super Resilient wheels. One-third octave analyses were performed on the recorded noise data for both treated and untreated wheel conditions. The analysis included one-third octave SELs. An integration time of 20 sec was used.

Wheel Squeal Noise Data

Observations during the testing and analysis of the data indicated that the Franklin Avenue curve was most representative for this study. Significant wheel squeal was not generated by either the untreated or treated Bochum 84 wheels at the Penn Station curve, so that an adequate estimate of performance could not be obtained at this curve. The discussion below thus centers on the Franklin Avenue turnaround data.

Wheel squeal noise levels for the untreated revenue cars with PCC Super Resilient wheels, the untreated Bochum 84 wheels, and the treated Bochum 84 wheels are presented in

Figure 16. These data indicate a 10-dB reduction of wheel squeal at the 1,000 and 1,600 Hz one-third octaves and, apparently, an elimination of wheel squeal at these frequencies. The data also indicate that the PCC Super Resilient wheels produced the highest levels of wheel squeal, occurring at the 800 and 1,600 Hz one-third octaves.

The wheel squeal noise levels for the untreated Bochum 84 wheels alone were roughly 20 dB lower than those for the PCC Super Resilient wheels. This result for the Bochum resilient wheel is consistent with other studies at the Southeastern Pennsylvania Transportation Authority (SEPTA) (*4*) and San Francisco's Bay Area Rapid Transit (BART) (*5*), where wheel squeal levels with the resilient Bochum 54 wheel were much less than with solid steel or aluminum-centered wheels. What is interesting here is that the wheel squeal noise levels with the PCC Super Resilient wheel were significantly higher than those for the Bochum wheels.

Data collected at the low rail side of the curve are consistent with those collected at the high rail side. There does appear to be an additional component of wheel squeal at the 630 Hz one-third octave at the low rail side for the Bochum 84 wheel. However, the noise levels were generally lower on the low rail side, suggesting that the squeal was generated at the high rail.

The laboratory analyses indicate that the first and second bending modes of the Bochum 84 tire were at 630 Hz and 1,600 Hz. There did not appear to be a lateral bending mode at 1,000 Hz, where a wheel squeal component was observed. Neither did there appear to be a radial mode near 1,000 Hz. There is a possibility that the rail contributed a mode at 1,000 Hz, such as a pinned-pinned mode for either lateral bending or torsion. However, if it did, the mode was apparently damped by the wheel vibration absorbers.

Figure 17 illustrates a narrow-band spectrum of wayside noise obtained at the high rail side for the untreated Bochum 84 wheels of Car No. 9. The four samples shown indicate the variation of the frequencies of excitation. The frequency components corresponding to the 630 Hz and 1,600 Hz lateral tire bending modes are apparent. Additional discrete frequencies occur between 800 and 1,000 Hz, and at about 1,400 Hz. Figure 18 is a similar comparison, but for the

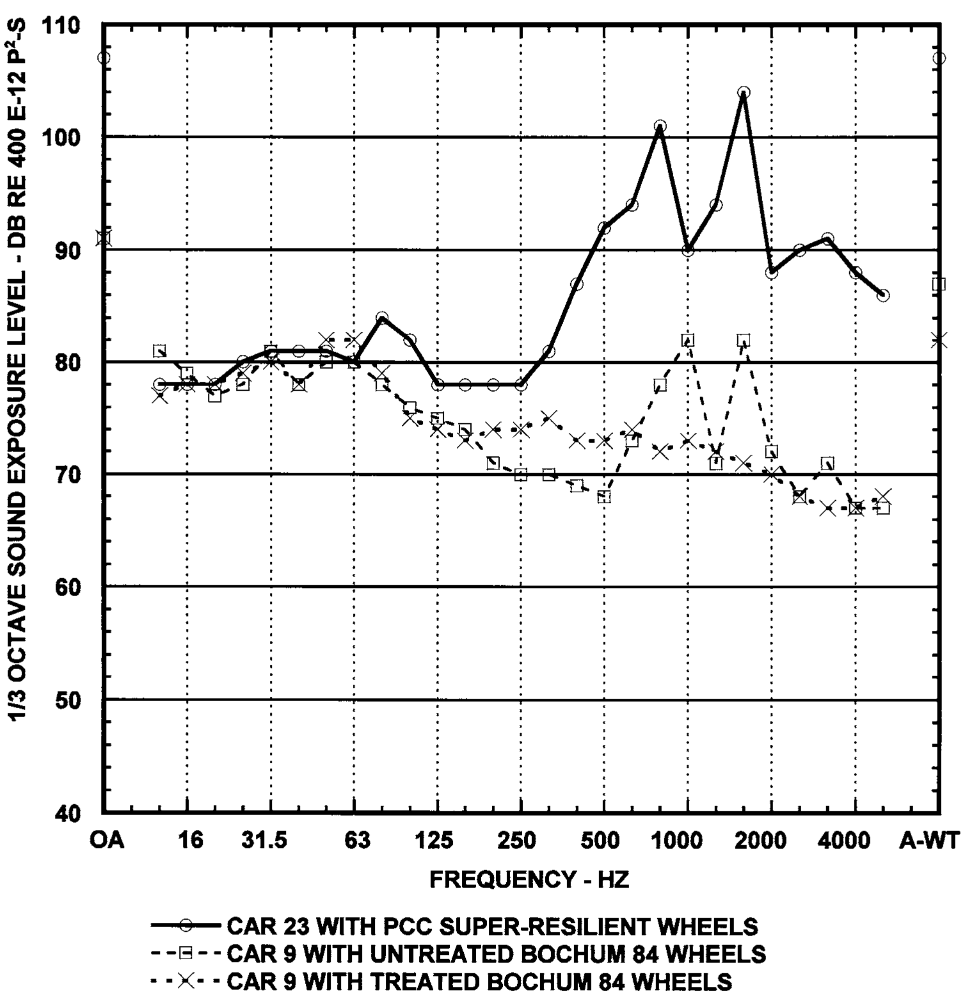

Figure 16. One-third octave wayside noise at high rail side of the Franklin Avenue curve.

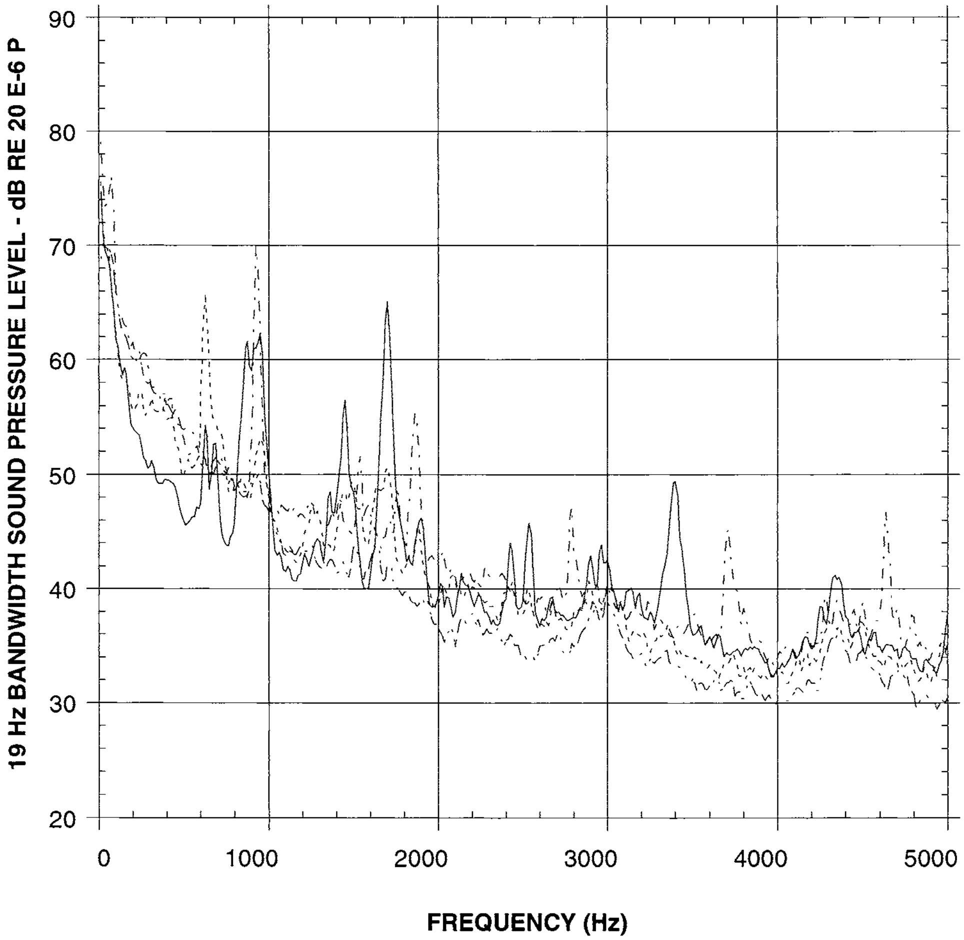

Figure 17. Fourier spectra for four samples of wayside noise at the high rail side for Car No. 9 with untreated Bochum 84 wheels.

treated wheel condition. Discrete frequencies are still apparent, but greatly reduced relative to the untreated case.

The samples of wayside wheel squeal noise for the Bochum 84 wheels at the Franklin curve were not consistent. In some cases, little wheel squeal was produced, while in another, the wheel squeal was substantial, though less than the wheel squeal produced by the PCC Super-Resilient wheel. With the vibration absorbers, the wheel squeal noise levels were consistently less than without.

PORTLAND TRI-MET

The Portland Tri-Met curved track testing was conducted at a section of 400-ft radius curve at the Eastern Portal of the Sylvan Hills Tunnel. This section of track was the subject of vigorous community reaction to wheel squeal noise, and the Portland Tri-Met expended considerable funds and effort in attempting to remedy the situation. Noise control measures implemented prior to the study include installation of sound barrier walls and a water spray system.

Tri-Met has long been concerned with wheel squeal at various curves and has experimented with on-board oil drop lubrication and dry-stick lubrication, and wayside hand lubrication. Tri-Met is currently experimenting with wayside lubrication with Teflon-based lubricants and vegetable oil–based lubricants. The current tests were conducted prior to installation of a Teflon-based wayside lubrication system at the East Portal curve, but after installation of the sound barriers and water spray system.

The tests were conducted with an instrumented test vehicle and revenue service trains. The test vehicle was a Type 1 vehicle manufactured by Bombardier and was one of Tri-Met's original vehicles. The vehicle is articulated and has two motored trucks at either end and an idling center truck. The center truck has a bolster with center plate and thus is

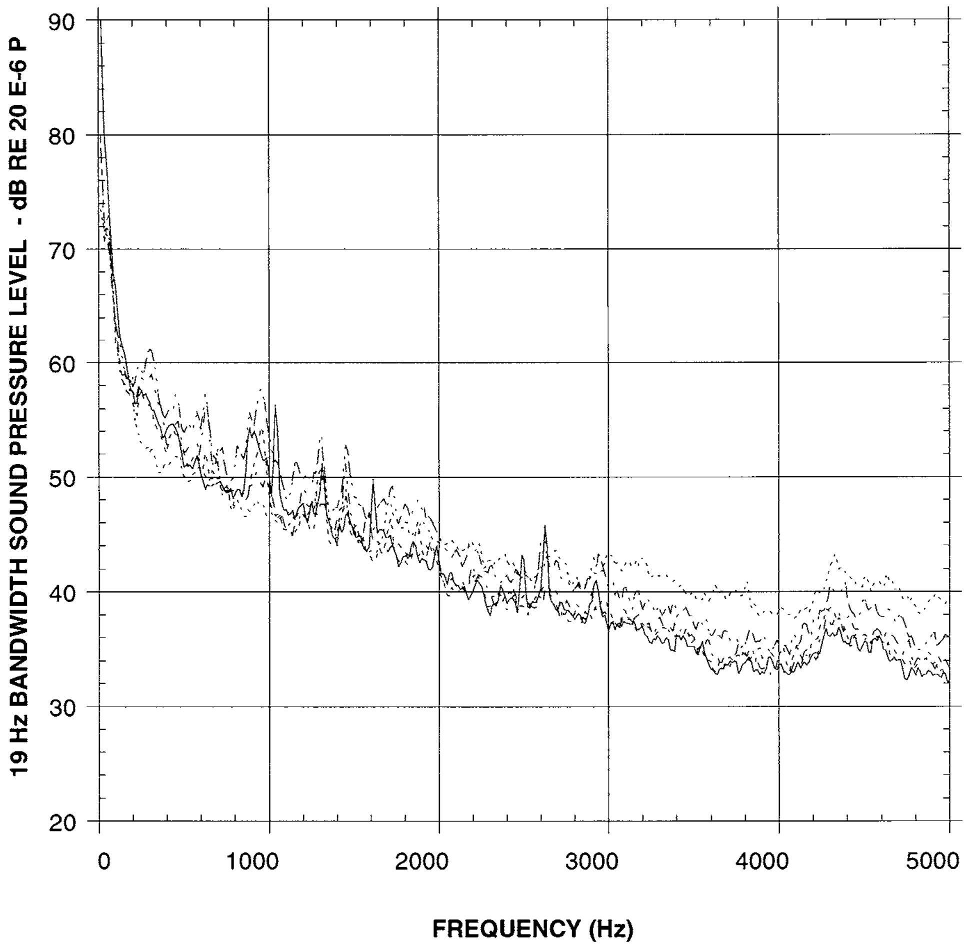

Figure 18. Fourier spectrum of wayside noise at the high rail side for four samples of Car No. 9 with treated Bochum 84 wheels.

allowed to steer through curves without lateral forces applied by the coupler.

Tri-Met also has Type 2 vehicles manufactured by Siemens. These vehicles are articulated, with two motored trucks and an idling center truck. The vehicle has a three-piece body, with the center truck supporting the center piece. Thus, forces exerted by the leading and trailing portions of the body may affect the steering of the center piece and center truck. Tri-Met's engineers were of the opinion that the center truck of the Type 2 vehicle produced more curving noise than the center truck of the Type 1 vehicle. This appeared to be the case during testing, but quantitative analyses were not conducted to confirm this.

Measurement Locations

Two sound level meters were positioned with microphones positioned against the absorptive sound barrier wall at the high rail side of the eastbound curve. The sound barrier wall was treated with 4-in.-thick glass fiber acoustical absorption, so that essentially free-field conditions were established for frequencies typical of wheel squeal. The distance from the track center to the microphones was limited to about 10 ft.

Microphones and pre-amplifiers were mounted on the under-side of Car 115 near each truck. At the driven trucks, the microphones were mounted between the truck and the nose of the vehicle, with the microphone at about 3 ft from the axle. Thus, data were taken for the leading axle of the vehicle, regardless of direction. The center truck noise was measured with a single microphone mounted under the car, about 3 ft from one of the axles. Due to the asymmetry of the location, data could not be taken for the leading axle in all cases at the center truck. The center truck microphone was subject to some sensitivity drift of about 3 dB, which is believed to have been caused by dampening of the microphone at the various curves with water spray systems. Fortunately, the data obtained at the leading and trailing trucks were of excellent quality, not subject to sensitivity drift.

Accelerometers were attached to both of the rail heads in both vertical and horizontal directions at a point midway between the microphones. Additional accelerometers were attached to the rail web in the horizontal direction. Attachment was made by adhering threaded mounting studs to the rail with a moderately rapid-curing epoxy. Rail vibration was measured at a single cross section of the curve.

Procedures

All of the curved track tests at the Portland Tri-Met were conducted during the hours of 10 a.m. and 3 p.m. The test car, No. 115, was operated during revenue service, but did not participate in revenue operations. The tests at the East Portal were hampered by operational restrictions that required the car to be switched at points well removed from the test curve. The result was that as much as 90 min passed between passage of the test car at the test curve. However, under-car noise data for a large number of curves were obtained during turnarounds. These latter data provide a good estimate of the noise reduction effectiveness of the wheel vibration absorbers and are discussed later below.

Test Results

The results of the tests are summarized below.

Wayside Noise with Test Vehicle

The wayside noise with the test vehicle include the following combinations: (1) no treatment, (2) wheels treated with vibration absorbers, (3) rails treated with vibration absorbers, and (4) both wheels and rails treated with vibration absorbers. Figures 19 and 20 include one-third octave

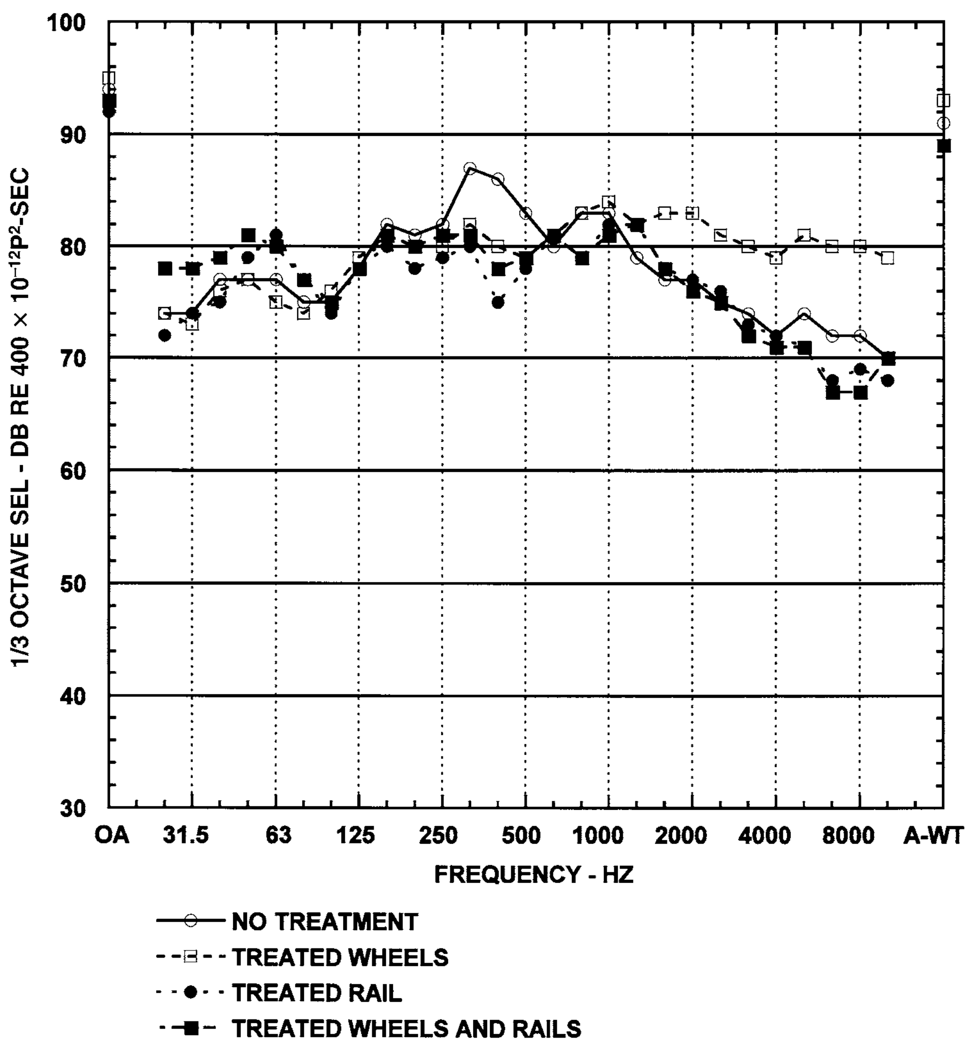

Figure 19. One-third octave SELs for eastbound 25-mph test vehicle at East Portal test section.

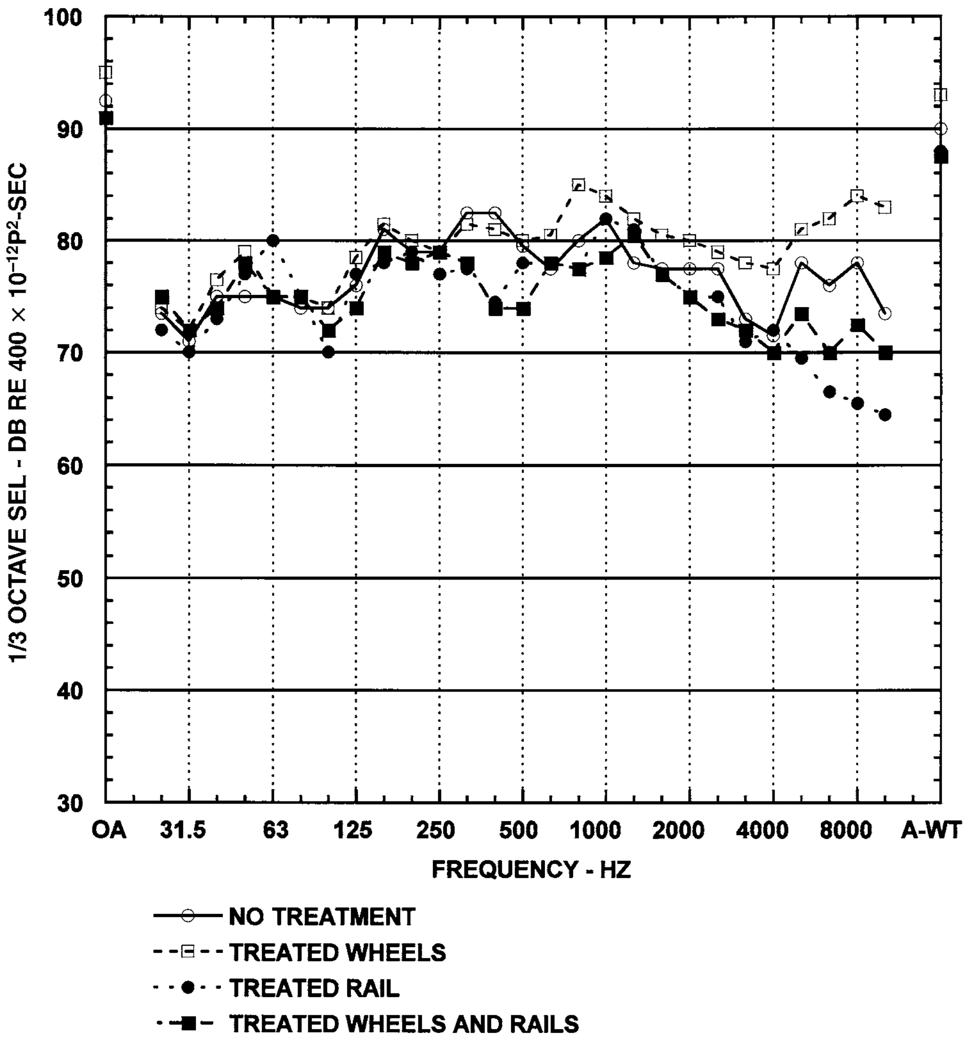

Figure 20. One-third octave SELs for eastbound 20 mph test vehicle at East Portal test section.

spectra for each of these conditions for train speeds of 25 and 20 mph, respectively. The spectra are energy averages of the one-third octave band SEL measured for each test vehicle passage by integrating over the noise signatures.

There is some evidence of stick-slip noise above 5,000 Hz for the no-treatment condition. With wheel vibration absorbers, the noise levels were higher at frequencies above 1,250 Hz than levels with no treatment. These elevated noise levels, or high-frequency stick-slip noise, are produced. The wheel vibration absorbers would not increase noise levels at these frequencies. However, they did not prevent stick-slip noise from occurring. There is little evidence of high-frequency stick-slip noise with the rail vibration absorbers installed, either with or without the wheel vibration absorbers, in the data shown for 25 mph in Figure 19. However, at 20 mph, some stick-slip noise increased the one-third octave levels at frequencies above 4,000 Hz.

These data indicate that mid-range noise levels between 315 and 500 Hz were highest without any treatment. The noise levels between roughly 200 and 500 Hz appear to be significantly lower with the rail vibration absorbers relative to those levels for the no-treatment and treated wheel cases. This is particularly noticeable in the data shown in Figure 20 for the 20-mph runs. The rail vibration absorbers would have attenuated or eliminated noise radiation ahead of and behind the vehicle in the mid-frequency range, by eliminating the pass-band of rail vibration transmission. Thus, the rail vibration absorbers would be expected to reduce the wayside noise exposure.

The A-weighted SELs with the rail vibration absorbers installed were 89 dBA, about 2 to 4 dB lower at 25 mph than those levels observed for the no-treatment and treated wheel conditions, respectively. The difference appears to be of the order of 2 to 5 dB for the 20-mph data shown in Figure 20.

Similar performance was observed for the 15-mph runs, though data for treated rail only were not obtained.

The data obtained for the test vehicle indicate a positive benefit for the wheel vibration absorbers between 315 and 500 Hz, but do not indicate that they reduce stick-slip noise at frequencies above 1,250 Hz. The rail vibration absorbers appear to provide a reduction of mid-range noise between 315 and 500 Hz and at frequencies above 1,250 Hz. This latter result is not supported by the revenue service train data, as discussed below.

Wayside Noise with Revenue Trains

Wayside noise data for revenue service trains were obtained between test vehicle runs. The data provide a measure of the noise reduction performance of the rail vibration absorbers under normal service operation with the Type 2 vehicle consists and combination Type 2 and Type 1 consists.

Energy averaged SELs are presented in Figures 21 and 22 for train speeds of 25 and 20 mph, respectively. These data indicate a consistent reduction of mid-range noise levels at frequencies between roughly 250 Hz and 2,000 Hz. The rail vibration absorbers are likely most effective at fre-

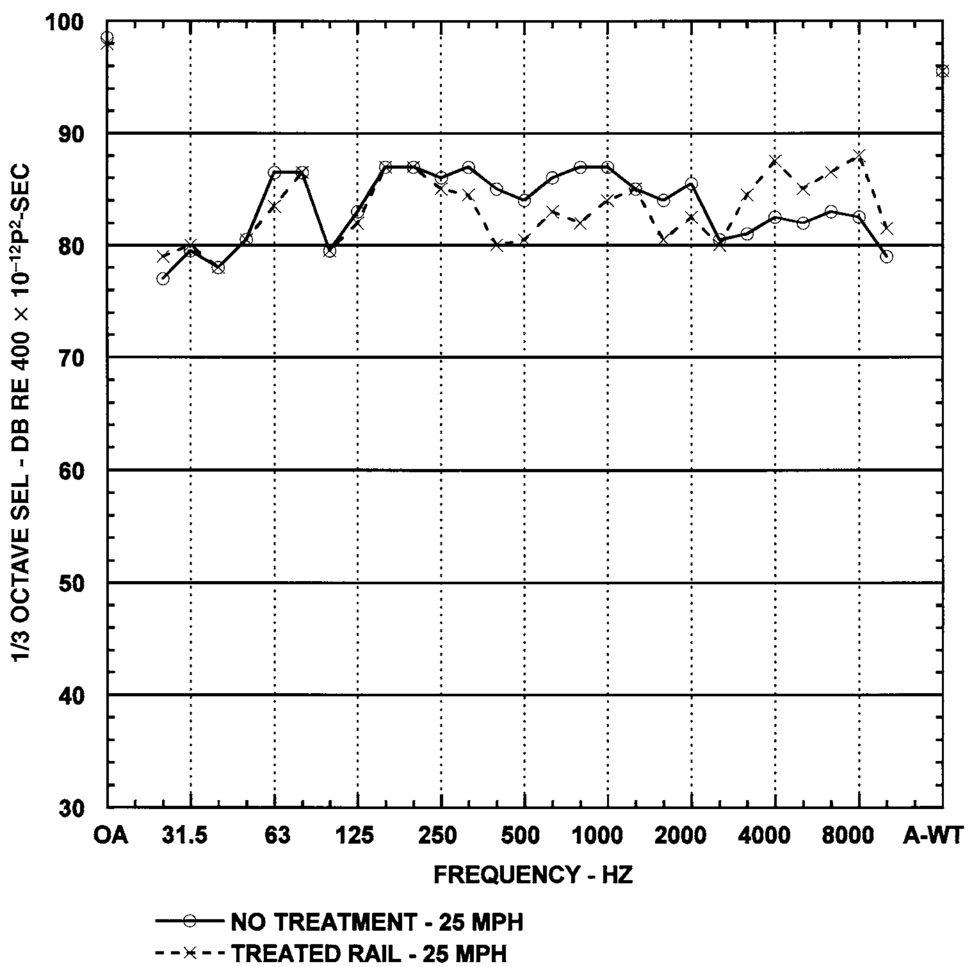

Figure 21. One-third octave SELs for eastbound revenue service trains at 25 mph at the East Portal curve.

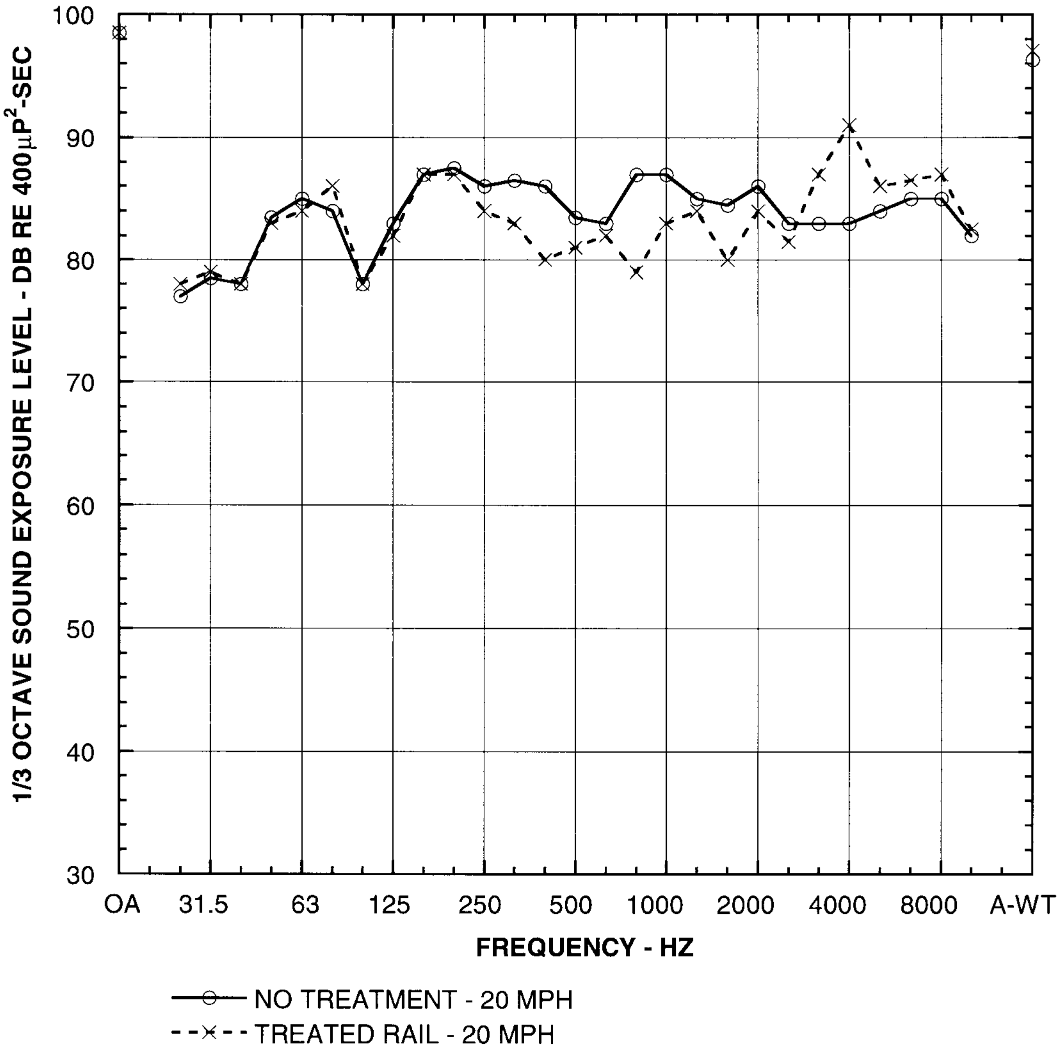

Figure 22. One-third octave SELs for eastbound 20-mph revenue service trains at the East Portal curve.

quencies above 500 Hz, and these data indicate that significant noise reductions might have been obtained with them.

The noise levels above 2,500 Hz were higher with the rail vibration absorbers than without. Again, one would not expect the rail vibration absorbers to increase noise at these frequencies, and the apparent increase is likely due to changes in surface conditions. Note that the data for the treated rail were obtained about one month after the data for the untreated rail. The fact that no such increase was observed for the test vehicle further indicates the difficulty in obtaining consistent wheel squeal noise reduction data. Also, the high-frequency noise levels shown for the rail vibration absorbers above 2,500 Hz might have been higher without the absorbers.

Under-Car Noise Levels at East Portal

Under-car noise data recorded over the course of passage over the test section at the East Portal were analyzed with a one-third octave analyzer. Each sample was at most about 8 to 12 s in length, the time required for the vehicle to traverse the 300-ft test section. Representative energy averaged one-third octave sound levels for all three trucks combined for a speed of 25 mph are compared in Figure 23. The results indicate that there was little noise reduction attributable to the wheel vibration absorbers. (As will be discussed below, test results at other curves indicate more favorable performance of the wheel vibration absorbers than at this test section.) With respect to the rail vibration absorbers, the 630 Hz and 1,250 Hz one-third octaves were higher with the rail

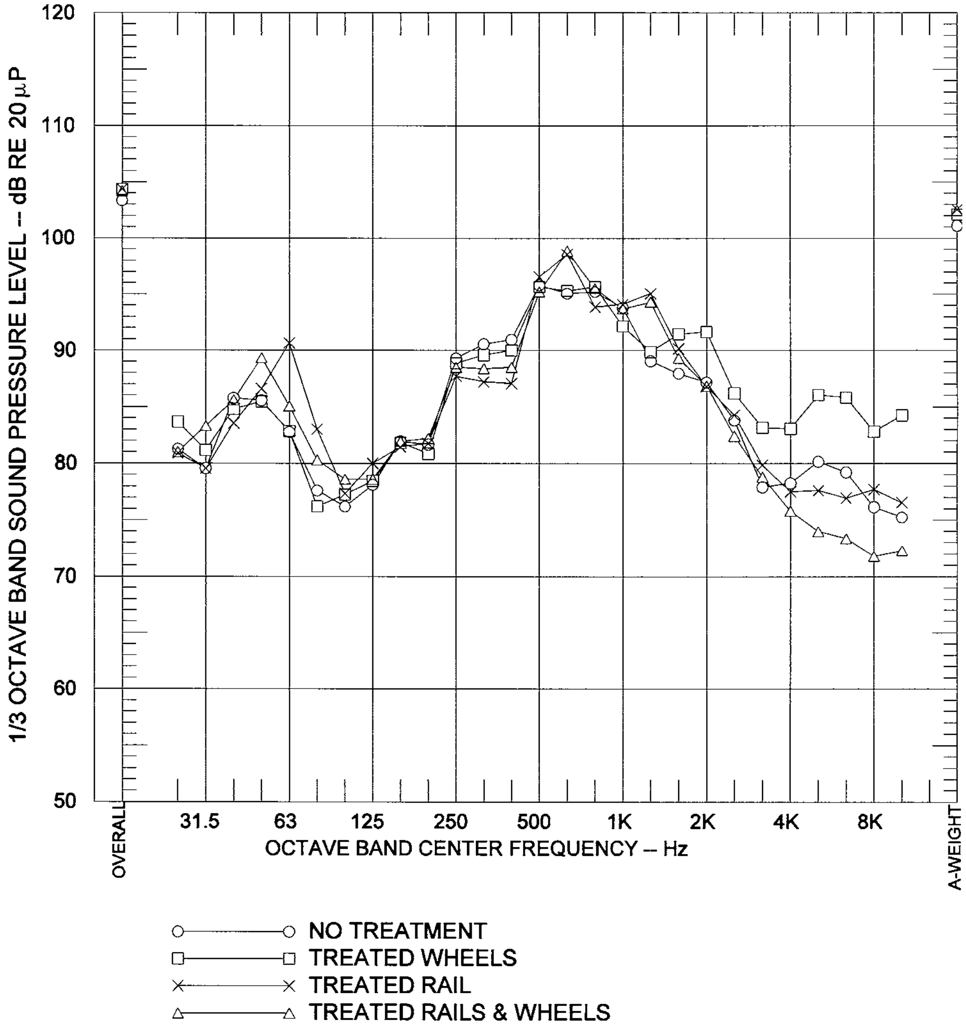

Figure 23. Under-car noise during passage over test section at East Portal curve at 25 mph—energy average of all trucks.

vibration absorbers than without by about 3 to 5 dB, as was observed at the tangent track test section. However, this increase did not increase the A-weighted sound levels by more than a decibel. The increase may have been due to resonances of the rail vibration absorber mounting plate. The stick-slip noise above 4,000 Hz was least with the rail vibration absorber installed, indicating a positive result.

Rail Vibration for Test Vehicle

The rail vibration data concerning the curved track tests at the Portland Tri-Met indicate that rail vibration levels varied considerably between test conditions. The results are surprising in view of the lack of significant differences in wayside noise at the test curve, but are consistent with data obtained at the tangent track.

Figures 24 and 25 illustrate the one-third octave vibration velocity exposure levels in decibels relative to 10^{-12} in.^{2}/sec for the transverse and vertical directions, respectively. The highest vibration levels were obtained with the wheel vibration absorbers installed. However, this is likely due to a failure of the absorbers to control stick-slip noise at frequencies above 5,000 Hz. There is little difference between the data for the treated and untreated wheels with the rail vibration absorbers installed.

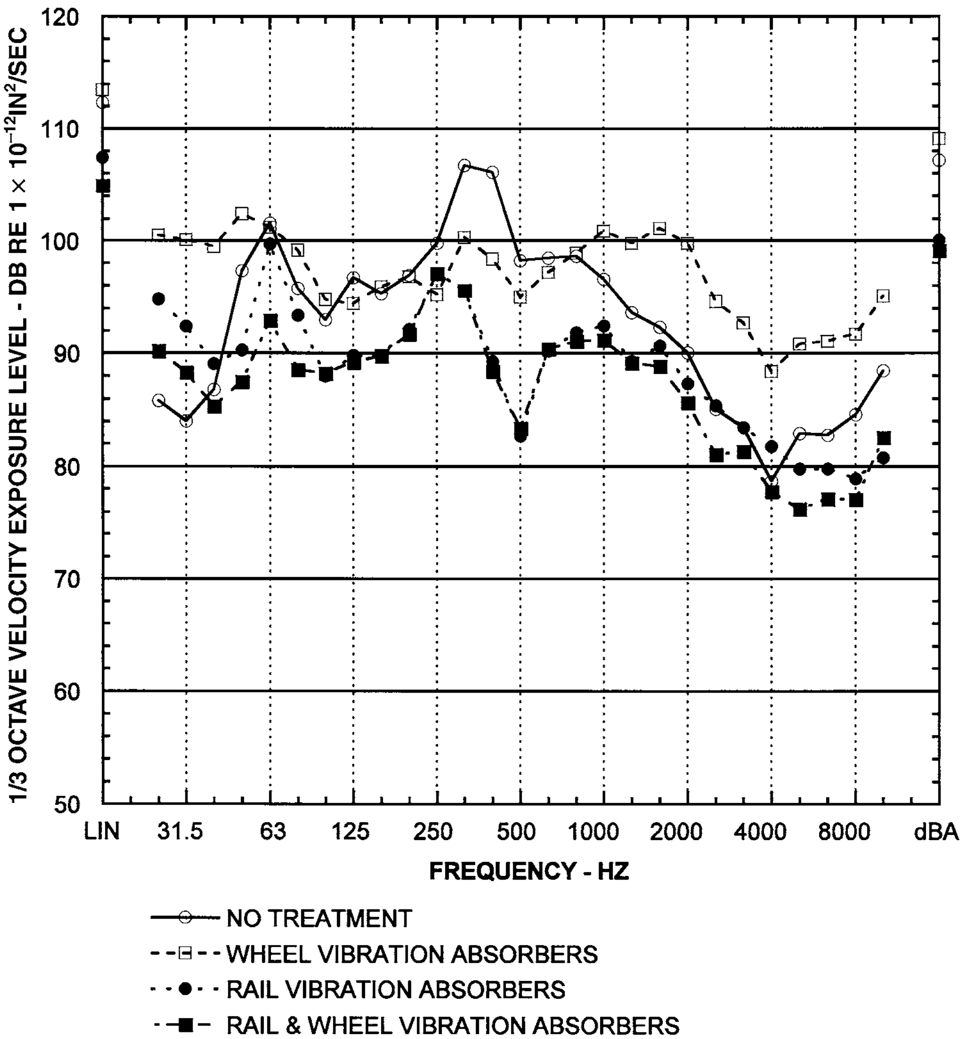

Figure 24. High rail transverse vibration velocity exposure for 25 mph test vehicle at East Portal curve.

The transverse rail vibration data were very much lower with the rail vibration absorbers installed than without, with or without the wheel vibration absorbers. The rail vibration absorbers attenuate rail vibration transmission propagated ahead of and behind the train, as well as ahead of and behind each wheel set, thus reducing the rail vibration energy per unit length. The data suggest that the rail vibration energy is reduced by a factor of 10, or 10 dB. This is a very positive result that is not reflected in the wayside noise levels, suggesting again that the wheels are the primary radiators of wheel squeal noise, similar to the tangent track test results.

Figures 26 and 27 include one-third octave spectra of vibration velocity exposures at the low rail of the East Portal curve with the test vehicle operating at 25 mph. Unfortunately, data for the treated rail and combined treated rail and treated wheel conditions are not available for the vertical direction. The transverse vibration data are available and are perhaps most relevant. As at the high rail, greater rail vibration levels were observed at the low rail for the treated wheel relative to the untreated wheel condition. High-frequency stick-slip noise is the primary cause. The rail vibration was lowest for the treated rail condition, at least for the transverse direction.

Rail Vibration Data for Revenue Service Trains

Rail vibration data were also collected for revenue service trains. The high-frequency stick-slip vibration of the rails was

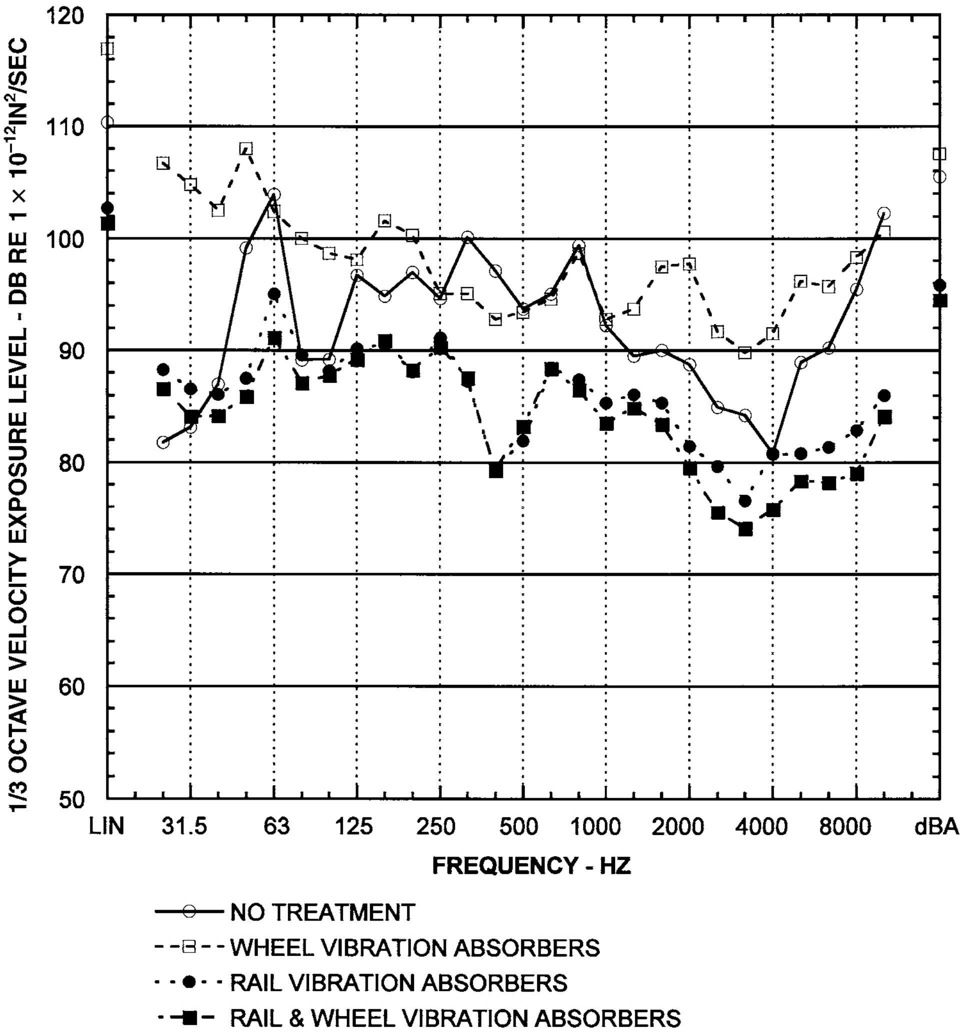

Figure 25. High rail vertical vibration velocity exposure levels for 25 mph test train at East Portal curve.

greater during the treated wheel test than during the untreated wheel test for the untreated rail, as was observed with the test train data. None of the revenue service trains had wheel vibration absorbers, suggesting that the high-frequency stick-slip noise increase with the test vehicle was due to factors other than the wheel vibration absorbers.

The high rail vibration exposures were generally lower with the rail vibration absorbers installed than without in the frequency range of 250 Hz to 2,500 Hz. At the low rail, the vibration exposures were less with the rail vibration absorbers than without—over a broader frequency range, extending from 250 Hz to the upper limit of the measurement, 10,000 Hz. These results are not shown here.

Under-Car Noise at Other Curves

While the results for the wheel vibration absorbers were rather disappointing at the East Portal curve, the under-car data collected at other curves were very encouraging. Wheel squeal was not entirely eliminated at curves, but there were reasonably consistent reductions of the level and duration of wheel squeal. Detailed one-third octave band spectra for each of these curves were obtained, though they are not reported here.

The curves at which under-car noise data were obtained include: (1) Burnside, (2) Gateway, (3) Sunset, (4) Beaverton, (5) Hillsboro Main Street Curve, and (6) Ruby Junction Yard.

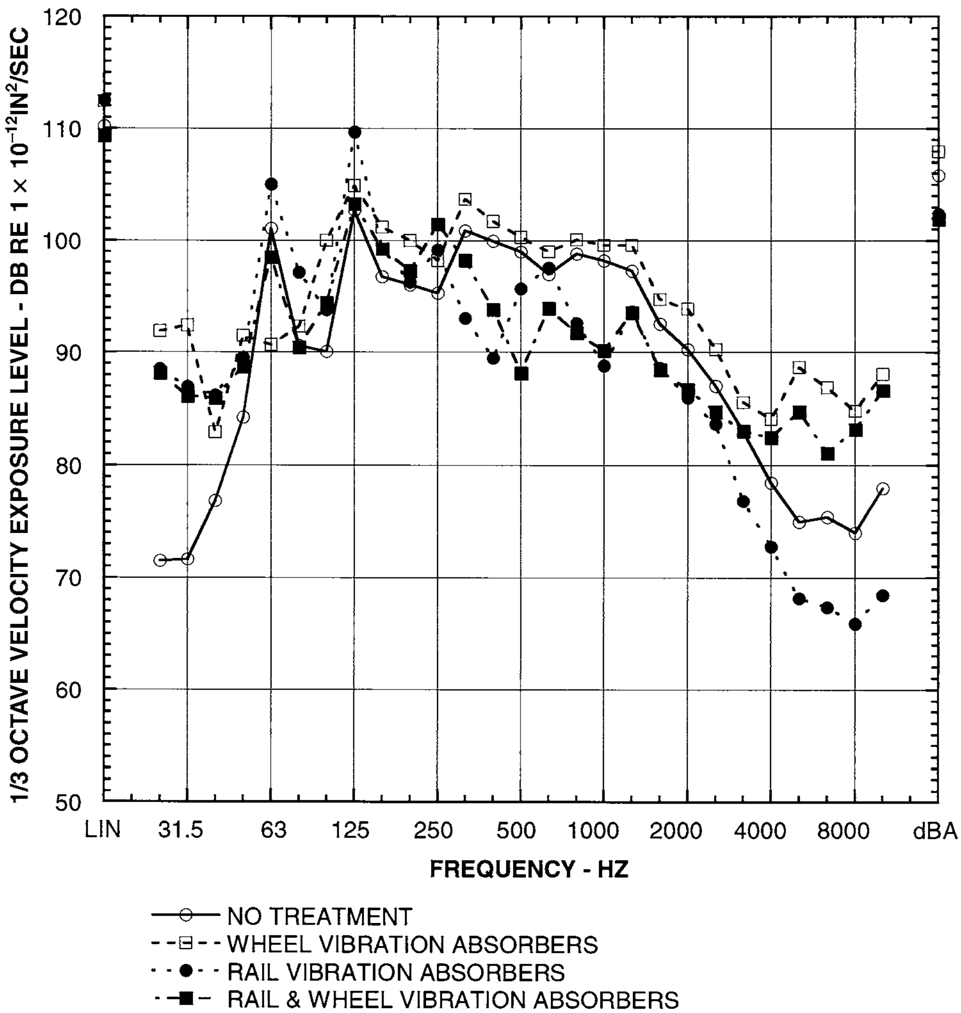

Figure 26. Low rail transverse vibration velocity exposures for 25 mph test vehicle at East Portal curve.

The curve radii ranged from 83 ft to 400 ft. Of these, the Burnside curve has the shortest radius, 83 ft. The Ruby Junction Yard curves also have short radii and represent extreme examples of wheel squeal noise. Most of these curves are ballast-and-concrete-tie track with RE 115 lb/yd rail.

Wheel squeal is difficult to quantify due to its intermittent characteristics, especially for a wheel such as the Bochum resilient wheel, which is already damped to some extent. While eliminating wheel squeal is the desired, but elusive, goal, there does appear to be a net improvement by addition of the wheel vibration absorbers. This is most apparent in certain one-third octaves, namely those encompassing the squeal frequencies associated with resonances of the tire.

The data collected at these curves were further evaluated by selecting the sound levels of the 1,250 and 1,600 Hz one-third octaves. These data better represent the squeal noise levels than simply the A-weighted noise levels, which include noise over the entire audible spectrum. The results are summarized in Table 6. The typical noise reduction was 5 to 6 dB. At the Ruby Junction Yard, the noise reduction was 17 dB, although this was based on a single pass through the yard. The lower noise reductions for the other curves are obtained for multiple passes, which include cases of no squeal for the untreated wheel and cases of squeal with the treated wheel.

Figures 28 and 29 illustrate the narrowband spectra of wheel squeal noise recorded at leading Truck A and trailing Truck B, respectively, during negotiation of a curve at the Ruby Junction Yard. Data are shown for both treated and untreated wheel conditions. Without the vibration

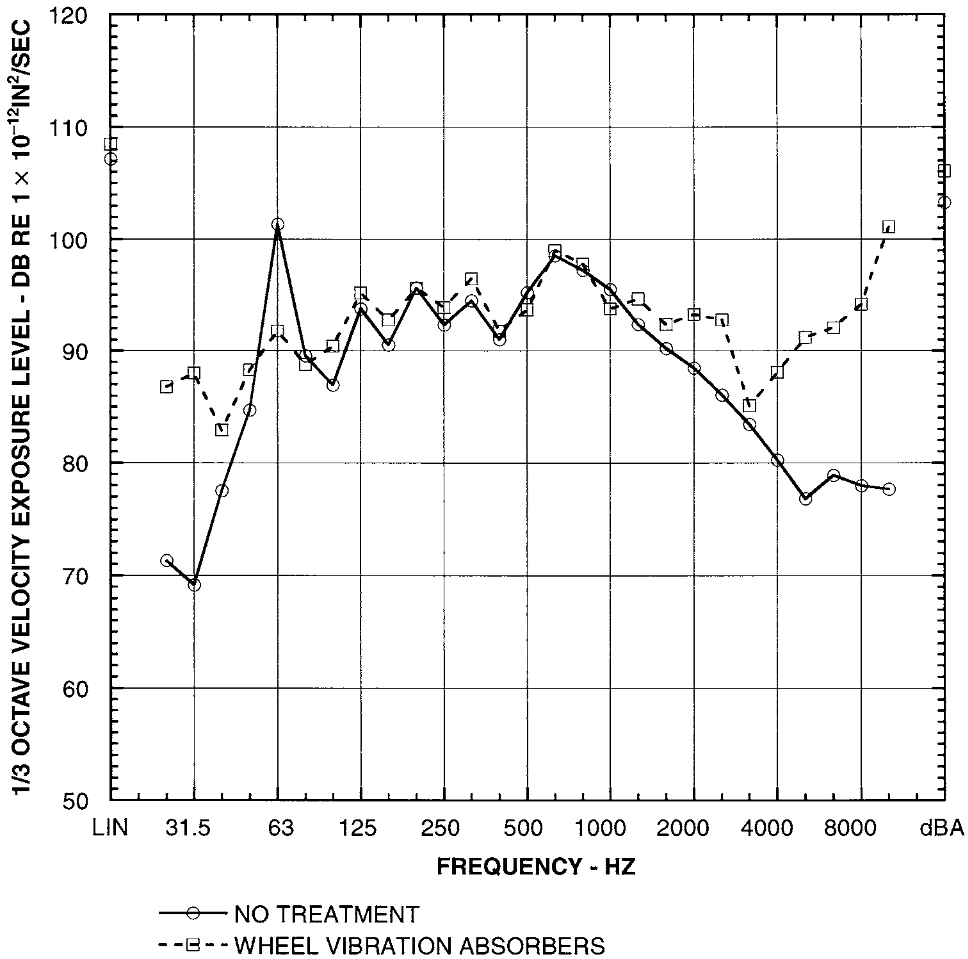

Figure 27. Low rail vertical vibration exposures for 25 mph test vehicle at East Portal.

TABLE 6 Summary of under-car noise levels at short radius curves

	1,250-1,600 Hz Squeal		
Curve	**Untreated dB RE 20 μP**	**Treated dB RE 20 μP**	**Noise Reduction - dB**
Burnside	102	97	5
Gateway	100	95	5
Sunset	100	94	6
Beaverton	91	88	3
Ruby Junction Yard	104	81	17
Energy Average	101	94	6

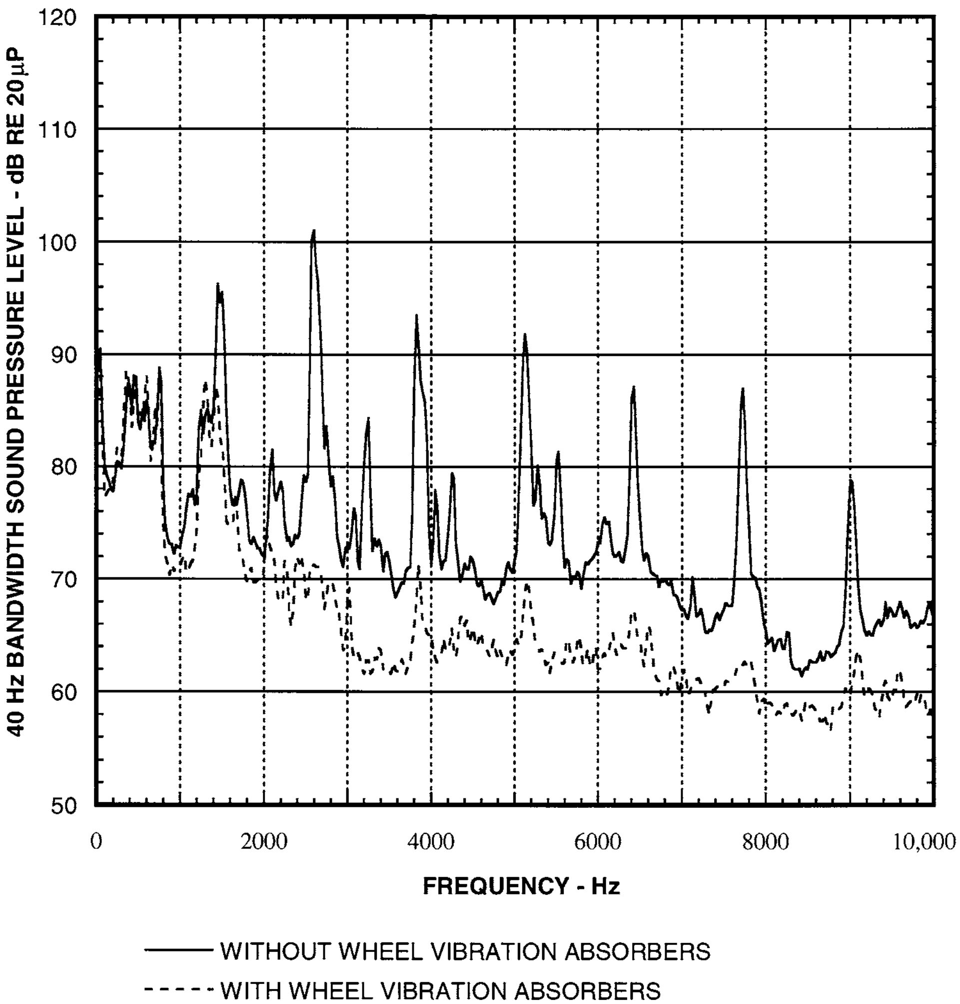

Figure 28. Narrowband spectra for leading Truck A at Ruby Junction Yard with and without wheel vibration absorbers.

absorbers, the spectra are rich in spectral peaks associated with curving noise. With the vibration absorbers, these peaks were very much reduced in level. Even so, there is still a peak at about 1,400 Hz with the absorbers at both trucks. The very pronounced peaks occurring at 2,600 Hz without the absorbers is almost entirely gone with the absorbers installed.

In spite of the unpredictability of curving noise, there appears to be a correlation of the occurrence of wheel squeal between the leading and trailing trucks and, perhaps, the center truck. That is, if squeal occurs at the leading truck, it usually occurs at the trailing truck, though at a lower level. This suggests that rail condition, humidity, or other parameter common to both trucks may be affecting the wheel squeal process, as opposed to random steering performance of the trucks. This relationship has not been investigated, but suggests an avenue for additional study.

Curving noise data were recorded at the Main Street curve in Hillsboro, located just west of the Main Street Bridge. The data contain primarily high-frequency stick-slip noise, without a significant squeal component at 1,600 Hz or 2,500 Hz. The A-weighted levels were tabulated and subjected to a regression analyses over train speed, the results of which are summarized in Table 7. The noise reduction was 7 dB for the leading truck at a speed of 25 mph. Noise reductions at higher speeds were generally lower, possibly because of increased rolling noise. The typical noise reduction ranged from 3 to 6 dB over the various train speeds.

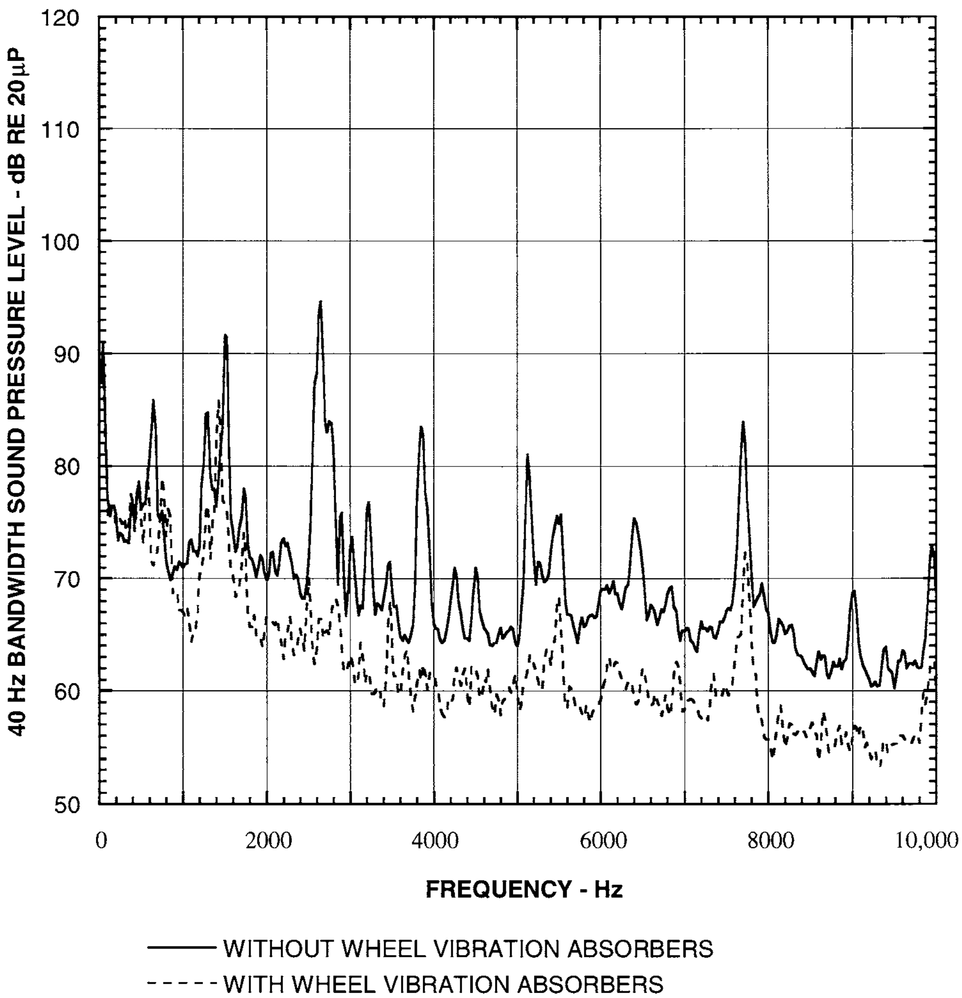

Figure 29. Narrowband spectra for trailing Truck B at Ruby Junction Yard with and without wheel vibration absorbers.

TABLE 7 Regression analysis of under-car noise levels at Hillsboro Main Street curve

Date	Treatment Condition	Regression Coefficients		A-Weighted Sound Level - dB RE 20 µP		
		Intercept	**Linear**	**25 mph**	**35 mph**	**45 mph**
Lead Truck						
27-Jul	Untreated	97.6	0.279	104.6	107.4	110.1
30-Jul	Treated	79.8	0.557	93.7	99.3	104.9
	Difference			10.9	8.1	5.3
26-Aug	Untreated	98.4	0.179	102.9	104.6	106.4
24-Aug	Treated	93.3	0.257	99.7	102.3	104.9
	Difference			3.1	2.4	1.6
	Average Difference			7.0	5.2	3.4
Trailing Truck						
27-Jul	Untreated	86.7	0.45	98.0	102.5	107.0
30-Jul	Treated	80.9	0.479	92.9	97.6	102.4
	Difference			5.1	4.9	4.6
26-Aug	Untreated	96.9	0.171	101.1	102.9	104.6
24-Aug	Treated	91.4	0.271	98.1	100.9	103.6
	Difference			3.0	2.0	1.0
	Average Difference			4.1	3.4	2.8
Grand Average				5.6	4.3	3.1

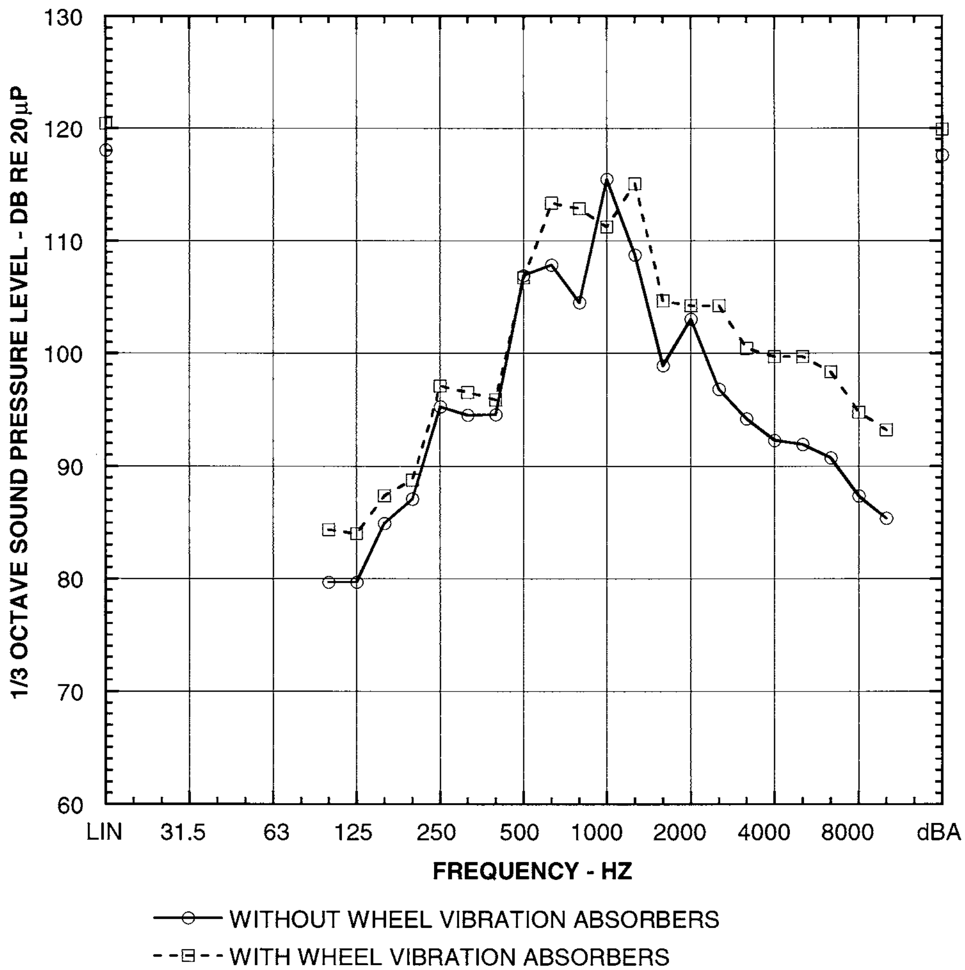

Figure 30. Root-mean-square sound pressure levels beneath test car at Banfield Expressway curve—average of all three trucks.

The noise reductions were greater during July than during August. A wayside lubricator was installed on the eastbound and westbound tracks just west of the tangent track test section, east of the Main Street Bridge, and there is a possibility that the lubricant migrated to downtown Hillsboro and back along the eastbound track to this curve, a distance of perhaps 3 mi. Such tracking would reduce stick-slip noise at the Main Street curve. Before the lubricant was applied, the noise reductions at the leading truck were 5 to 11 dB, depending on train speed. After application, the difference was of the order of 2 to 3 dB.

Data were taken at a large radius curve with, presumably, corrugated rail along the Banfield Expressway with and without wheel vibration absorbers installed. The noise spectra with the absorbers installed were very different from those for the untreated wheel, a result that is not reflected in the tangent track noise data. The reason for this difference in behavior has not been determined. Figure 30 includes one-third octave sound levels measured beneath the test car negotiating a large radius curve at a speed of about 55 mph. The data were consistent for both the July and August tests. That is, similar spectra were obtained in both July and August, and the energy averages of the results for both series of tests are presented in the figure. Without the wheel vibration absorbers, there is a pronounced peak at 1,000 Hz. With the wheel vibration absorbers, there appears to be a splitting of the peak frequency into one peak at roughly 630 or 800 Hz and into another more well-defined peak at 1,250 Hz. The wheel vibration absorber would not be expected to produce this degree of frequency splitting. Additionally A-weighted and overall noise levels are greater with the absorbers installed than without, which would not be expected with additional damping provided by the absorbers. No explanation has been obtained for these results.

CHAPTER **9**

CONCLUSIONS AND SUGGESTED RESEARCH

This project involved the most extensive testing and demonstration of wheel vibration absorbers conducted to date within the United States; in the case of rail vibration absorbers, no prior U.S. testing had been conducted. The current tests indicate the nature of their performance on modern light rail transit systems with resilient wheels. Valuable information concerning wheel/rail noise radiation have been obtained, which can be applied to further development of the these and other noise control technologies.

Provided below is a summary of the conclusions of the study and possible directions for further research and development.

RAIL VIBRATION ABSORBERS

The rail absorbers were moderately successful in reducing wayside noise. The test results indicate the following:

- The rail vibration absorbers eliminated the pre-passage and post-passage singing rail at tangent track, thus qualitatively reducing wayside noise impact. The elimination is obvious to the listener and indicates that effective noise control treatments need not necessarily reduce the maximum A-weighted level. The singing rail spectrum is characterized by pass-bands and stop-bands of vertical bending, lateral bending, and torsional vibration. Of these, vertical bending is the most significant with respect to rolling noise. The low-frequency limit of the first pass-band in vertical bending corresponds to the first pinned-pinned mode of bending vibration. At curved track, pre- and post-passage noise caused by stick-slip excitation of the rail was also reduced.
- The rail vibration absorbers substantially reduced rail vibration levels over a broad range of frequencies prior to, during, and after passage of the train. This reduction may have a benefit with respect to rail corrugation control. The reduction was obtained at both tangent and curved track.
- The rail vibration absorbers did not reduce either wayside maximum levels or SELs at tangent track, indicating that wayside noise during train passage was radiated primarily by the wheels at tangent track. Wayside maximum A-weighted noise levels at tangent track were actually slightly higher with the rail vibration absorbers than without absorbers by 1 or 2 dB, possibly because of a resonance of the absorber mounting plate and rail flange at about 1,250 Hz.

The rail vibration absorbers are still in service at the Portland Tri-Met's East Portal curve. The opinion expressed by some at Tri-Met is that the absorbers seem to be helping to control the stick-slip noise problem at this curve.

WHEEL VIBRATION ABSORBERS

The tests indicate the following:

- The wheel vibration absorbers reduced wheel squeal noise exposure at short radius curves, but did not eliminate squeal. The probability of occurrence and some moderate reduction of the maximum level was obtained. In some cases, wheel squeal was not observed with the absorbers attached. However, there were also events where no squeal was observed for the untreated wheel.
- The wheel vibration absorbers did not appear to reduce stick-slip noise at the wide radius test curve at the East-Portal curve at the Portland Tri-Met.
- There did not appear to be a joint noise reduction benefit of the combined treatment of wheel and rail vibration absorbers. That is, the combination of wheel and rail vibration absorbers did not produce a reduction greater than that which would be expected on the basis of individual respective performance.
- The wheel vibration absorbers had no significant effect on rolling noise at tangent track, though there was a very minor reduction of about 1 dB at about the 1,000 Hz one-third octave, suggesting that some very minor modification of the wheel noise radiation characteristic was obtained.
- Rail vibration was significantly lower over a broad frequency range with the treated wheels relative to untreated wheels. This result is contradictory to the lack of measured noise reductions. Modal analyses of radial tire responses indicate a reduction of the input mechanical impedance at anti-resonance frequencies of the tire. Reduction of the tire's input mechanical impedance at

its anti-resonance frequencies may have reduced wheel/rail vibration forces without reducing tire vibration at other frequencies. To the extent that wheel/rail contact vibration forces are reduced by the wheel vibration absorbers, as indicated by reduced rail vibration, there may be a benefit with respect to rail corrugation control.

- The spectra of combined under-car rolling and curving noise at 55 mph differed significantly with wheel vibration absorbers installed relative to without at a large radius curve at Tri-Met's Banfield Expressway corridor. The wheel/rail interaction at this curve and at this speed appeared to be altered by the wheel vibration absorbers. The nature of wheel rail interaction at wide radius curves at high speed is probably very different from that at tangent track, including lateral stick-slip generated noise that does not include well-developed squeal. Rail corrugation is believed to exist at this curve on the basis of the noise data, and, if so, the vibration absorbers may have altered the non-linear contact forces related to rail corrugation.
- The modal analyses of the treated and untreated Tri-Met Bochum 54 wheels indicate that the wheel vibration absorbers reduced the resonant response of the tire in lateral bending at frequencies of 2,600, 3,800, 6,400, and 7,700 Hz. Lateral bending modes at frequencies of 500, 1,400, 5,100, and 9,000 Hz were largely unaffected. However, wheel squeal average reductions were observed at 1,400 Hz during operational testing. The effect of the absorbers on the radial response of the tire is less easily determined. However, the absorbers appeared to reduce the response of the tire at frequencies between 2,000 and 2,600 Hz and between 3,000 and 3,300 Hz.

Systems considering wheel vibration absorbers should ensure that the trucks of new vehicles provide the necessary clearance. Also, the tires must be pre-drilled and threaded prior to heat treatment. Other than clearance limitations, the absorbers are easily installed.

DISCUSSION

Observations by this author indicate that the low rail is the primary source of sustained wheel squeal from solid wheels at QR in Brisbane, Australia, while the high rail appears to be a source of short duration stick-slip noise. This latter type of noise is referred to by QR as "flanging noise." While flanging noise is a popular term, the actual mechanism is more likely stick-slip of the tire's running surface over the top of the rail, driven by lateral flange forces at the gauge face. The resultant vertical twisting moment or couple applied to the tire over a short lever arm of the order of a few centimeters virtually ensures the excitation of a broad range of lateral bending modes of the tire. Much of the curving noise observed with the Bochum 84 and PCC Super-Resilient wheels at the NJT and the Bochum 54 and 84 resilient wheels at the Portland Tri-Met tend to be of this nature, especially at large radius curves. In contrast, sustained wheel squeal from the low rail is consistent with continuous, un-impeded sliding of the lead axle tire across the rail head. (The wheels of the trailing axle tend to have lower angles of attack than the leading axle wheels, so that the trailing axle wheels are not usually significant sources of wheel squeal.) This process involves excitation of lower frequency modes of lateral bending vibration, which are more easily attenuated by tuned wheel vibration absorbers.

The demands on a wheel vibration absorber are severe for controlling high-frequency stick-slip or flanging noise because stick-slip vibration contains frequency components extending up to very high frequencies, possibly above the upper limit of the audible range at 20,000 Hz. The wheel vibration absorbers are apparently tuned to control at most a limited number of modes of tire bending, primarily at frequencies of the order of 1,000 to 7,000 Hz, judging from the modal analysis data. Where sustained wheel squeal does occur at frequencies of the order of 1,000 to 4,000 Hz, the wheel vibration absorbers appear to be effective in reducing the duration, or probability of occurrence, and level of wheel squeal. Even so, lower levels of high-frequency stick-slip flanging noise were observed with the vibration absorbers installed relative to the untreated wheel condition, though this was not borne out by testing at the East Portal curve at Tri-Met.

The performance of wheel vibration absorbers is perhaps best described in terms of probabilities. In particular, the energy average of wheel squeal throughout a curve (equal to the root-mean-square level for the entire curve negotiation), or the closely related sound exposure level should be used for characterizing squeal, rather than simply the maximum level. The energy average or sound exposure level is directly related to the equivalent level, L_{eq}, and the Day Night Level, L_{dn}, common descriptors of wayside community noise. However, noise containing discrete frequency components is particularly irritating, and complete elimination of wheel squeal would be most desirable.

Tests conducted by the MTA NYCT indicate that wheel vibration absorbers applied to solid steel wheels were effective in controlling sustained wheel squeal at short radius curves. Other damping treatments were also found to be effective, including carbon steel ring dampers, ring dampers with conformal viscoelastic damping element, and constrained layer dampers. These tests, and the data reported here, indicate that treating either resilient or solid wheels with wheel vibration dampers should provide at least some benefit in controlling wheel squeal.

FURTHER RESEARCH

There are many rail transportation systems experimenting with a wide variety of wheel squeal noise control treatments, including water sprays, Teflon-based lubricants, vegetable oil-based lubricants, hand lubrication, on-board dry-stick lubrication with friction modifiers and flange lubricants, alloy composition of rails and wheels, hardness characteristics, wheel and rail profiles, gauge narrowing, rail embedment, rail running surface treatment by film deposition of synthetic materials, and others. These systems include the Portland Tri-Met, Pittsburgh, San Diego, and others. The NJT has specified that the new vehicles purchased for the Hudson–Bergen line be supplied with Bochum 84 wheels capable of using wheel vibration absorbers. San Diego is experimenting with Teflon-based wayside lubrication. QR has performed extensive investigations of wheel squeal noise and has experimented with variety of treatments, including lubrication and wheel vibration absorbers. Several German systems have experimented with wheel squeal noise control strategies. As a result, considerable knowledge is being acquired that should identify effective noise control treatments for specific operating conditions. A summary of recent experiences at these various systems would be valuable.

To the extent that rail vibration absorbers reduce rail vibration and eliminate the pinned-pinned mode of rail vibration, and, presumably, wheel/rail contact forces, they are candidates for possible rail corrugation treatments. Rail vibration absorbers may be effective at controlling chronic rail corrugation at sections of track where frequent rail grinding would otherwise be required to control noise. There are numerous examples of rail corrugation at many transit systems, and having such a mitigation tool would be extremely valuable. The rail vibration dampers now used at the Portland Tri-Met East Portal curve could be moved to a section of tangent track for long-term monitoring of corrugation rate reduction potential at little cost.

Alternative rail vibration damping treatments include the constrained layer dampers. Incorporating the damping mechanism directly into the rail support is very attractive, though this might not be effective in controlling the pinned-pinned mode unless fastener spacing is sufficiently random or if the bending moment is resisted by the fastener.

Wheel vibration absorbers might have systemwide applicability in reducing lateral tire oscillation, contact forces, and, thus, rail corrugation rates. Any test program designed to evaluate wheel vibration absorbers and rail corrugation reduction would require treatment of all transit vehicles and thus may be impractical. However, the new Hudson–Bergen Line vehicles will have Bochum wheels pre-drilled to accept wheel vibration absorbers, and this system would be an excellent candidate for rail corrugation rate control experiments. Fin absorbers are being supplied with certain new car procurements at QR and perhaps at other systems, and opportunities may exist for monitoring wheel squeal and rail corrugation rates with these types of absorbers.

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Abbreviations used without definitions in TRB publications:

AASHO	American Association of State Highway Officials
AASHTO	American Association of State Highway and Transportation Officials
ASCE	American Society of Civil Engineers
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
FAA	Federal Aviation Administration
FHWA	Federal Highway Administration
FRA	Federal Railroad Administration
FTA	Federal Transit Administration
IEEE	Institute of Electrical and Electronics Engineers
ITE	Institute of Transportation Engineers
NCHRP	National Cooperative Highway Research Program
NCTRP	National Cooperative Transit Research and Development Program
NHTSA	National Highway Traffic Safety Administration
SAE	Society of Automotive Engineers
TCRP	Transit Cooperative Research Program
TRB	Transportation Research Board
U.S.DOT	United States Department of Transportation

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