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WHEEL/RAIL NOISE CONTROL - A CRITICAL EVALUATION

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16. Abstract

This is the first interim report of a project whose goal is to optimize the performance of selected methods for the control of wheel/rail noise on urban transit systems. The project is part of the Urban Rail Noise Abatement Program managed by the Transportation Systems Center under the sponsorship of the Urban Mass Transportation Administration.

Noise and vibration are the major sources of environmental impact from urban rail transit operations, and is a concern for both new and existing systems. One of the primary sources of noise on rail transit systems is wheel/rail noise, or, the noise emitted by the wheels and rails as a result of their interaction. The purpose of this report is to carefully review and summarize the available information on each of the known or conceptualized methods for controlling wheel/rail noise and to identify requirements for further research, development, and testing. The report discusses the acoustical performance, costs, potential, or actual problems of these methods and suggestions are made for resolving uncertainties in the available data. This review is also particularly intended to help direct the remaining work to be performed under this project. In addition, a cost-effectiveness analysis is carried out to help in the selection of specific noise control methods for further study. The rationale for selecting each such treatment is presented.

17. Key Words
Cost Effectiveness; Damped Wheels; Noise and Noise Control; Noise Monitoring Systems; Rail Grinding; Rail Lubrication; Rapid Transit Noise; Resilient Wheels; Welded Rail; Wheel/Rail Noise; Wheel Truing

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PREFACE

This report is the first interim report under a project whose goal is the optimization of wheel/rail noise and vibration control. The project is being performed by Bolt Beranek and Newman Inc. (BBN) under contract to the U.S. Department of Transportation. The project is part of the Urban Rail Noise Abatement Program managed by the Transportation Systems Center (TSC), Cambridge, Massachusetts under the sponsorship of the Office of Rail and Construction Technology of the Urban Mass Transportation Administration (UMTA), Office of Technology Development and Deployment.

Significant contributions to this report were made by David A. Towers, Robert L. Bronsdon and Alan J. Berger of BBN and Clifford A. Woodbury III, John Edgar, and P. William Brath of Louis T. Klauder and Associates (subcontractors on this project). The BBN Program Manager is Paul J. Remington and the Technical Monitor at TSC at the time this report was written was Robert P. Kendig.

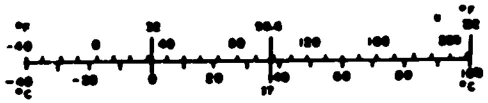
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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1. INTRODUCTION

Noise and vibration are the major sources of environmental impact from urban rail transit operations, and is a concern for both new and existing systems. One of the primary sources (in many cases, the primary source) of noise on rail transit systems is wheel/rail noise; i.e., the noise emitted by the wheels and rails as a result of their interaction.

In recent years, considerable effort has gone into understanding the mechanisms by which wheel/rail noise is generated (see, for example, [1,2]) and evaluating the methods by which it can be controlled (for example, [3]). Because of the diversity of test methods, transit equipment design and condition, and other noise determining factors (not all of which are presently understood) there still remains uncertainty as to the effectiveness and practicality of many of the wheel/rail noise control techniques [4].

The purpose of this report is to carefully review and summarize the available information on each of the known (or conceptualized) methods for controlling wheel/rail noise and to identify requirements for further research, development and testing in this area. In particular, this review is intended to help direct the remaining work to be performed under this project.

Technical Background

Wheel/rail noise is generally categorized into three types: squeal, impact, and roar.

Squeal (or screech) is the term used to describe the intense noise consisting of one or more tones that is associated with rail vehicles rounding curves of small radius. The excitation producing squeal results from the fact that, as a rail vehicle rounds a curve, its wheels, because they are constrained by the truck, cannot run tangent to the rails, i.e., the axles cannot take up radial positions in the curve. This situation causes what is called "crabbing," where the wheel both rolls along the rail and slides laterally across the rail head. The sliding of the wheel across the rail head results in time varying forces being applied to the wheel and rail (due to the sticking and slipping at the interface); the resulting vibration of the wheel produces the intense narrow-band noise [5].

Impact is the term used to describe the "clickety-clack" or banging noise that occurs when wheels cross rail joints or other discontinuities in the rail such as frogs, or when flat spots on the wheels roll over the rail. When the wheel encounters one of these discontinuities there is a brief rapid change in vertical velocity. This change in vertical velocity results in a large force at the wheel/rail interface that excites the wheel and rail into vibration and causes them to radiate sound.

Roar, or rolling noise, is the noise that dominates on tangent continuous welded rail in the absence of wheel flats. There is strong evidence that it is produced by the small-scale roughness on wheels and rails. As the wheel rolls over the roughness and encounters small bumps and valleys, it must (as when it encounters rail joints) either rise up over the bump, push the rail down out of the way, or do a little of both. The result is a force at the wheel/rail interface that excites the wheel and the rail and causes them to radiate sound.

Organization of Report

The initial sections of this report summarize the available information on wheel/rail noise control techniques. The sections are organized by treatment location: wheel treatments are contained in Section 2; rail or track treatments are in Section 3; and truck or vehicle-mounted treatments are in Section 4. The information given for each noise control method includes:

1. A description of the treatment.
2. Measured or analytically estimated effectiveness.
3. Non-acoustical benefits; e.g., extends wheel life.
4. Potential or experienced problems associated with its use.
5. Suggestions for future research, development, or testing.

Section 5 describes and discusses the potential uses and benefits of various monitoring systems. These include wheel roughness, rail roughness, and vehicle noise monitors.

In Section 6, a cost-benefit evaluation of the treatments, applied individually and in combination, is presented. The purpose of the analysis presented in this section is to rank order, or prioritize, the treatments in order to aid in the selection of treatments for further research. The cost data and methodology used for this purpose may not be adequate for use as a basis for transit property noise control decisions for reasons delineated in Section 6.

The final section summarizes the needs for future research, development and testing of wheel/rail noise control techniques and suggests priorities.

2. WHEEL TREATMENTS

2.1 Resilient Wheels

Resilient wheels are wheels in which the metal tire is structurally isolated from the wheel hub, generally by an elastomeric material. The resilient material acts in two ways: it provides damping to the wheel which results in reduced vibrations of the wheel at its resonant frequencies (most notably those frequencies associated with squeal noise); and it serves to isolate the vibrations of the wheel rim from the wheel hub, resulting in reduced dynamic forces applied to the rail and reduced accelerations of the axles, truck components and car body.

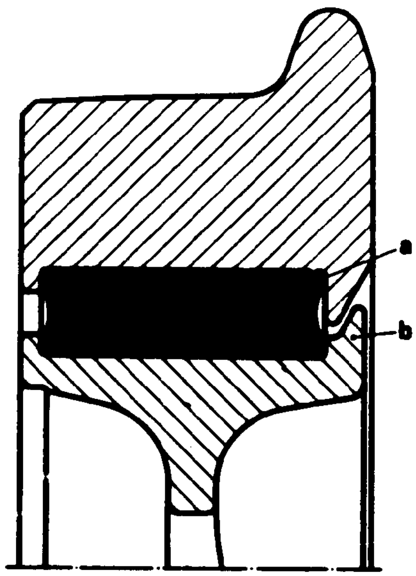
Resilient wheels have undergone continuous development since their invention in 1899. By the mid 1930's there were 40 differing designs for resilient wheels [6]. Currently there are three resilient wheel designs available for use in rapid transit systems:

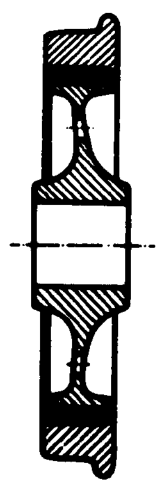
1. The Penn Cushion (Bochum) wheel, shown in Fig. 2.1, is known in Europe as the "Bochum 54." It is available in the U.S. from Penn Machine Company, Johnstown, PA.
2. The SAB wheel, shown in Fig. 2.2, is manufactured by Svenska Aktiebolaget Bromsregulator of Malmö, Sweden and marketed in the U.S. by American SAB Company, Inc., Chicago, IL.
3. A newly developed SAB V-Wheel, shown in Fig. 2.3, is also being marketed by American SAB Company, Inc.

■
Rubber-cushioned wheel, patent design Bochum 54, showing limiting flange
a) rubber segment
b) limiting flange

■ ■
Rubber-cushioned wheel with one circle of rubber segments for underground railway. View of flange face showing limiting flange

■ ■ ■
Rubber-cushioned wheel for underground railway. View of outer face showing the rubber segments





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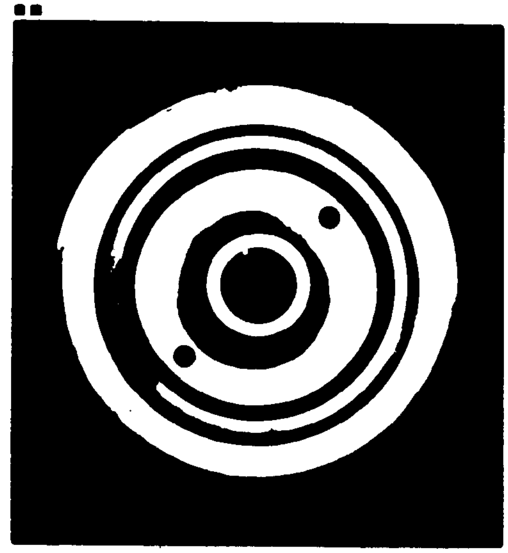

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FIG. 2.1. **PENN CUSHION (BOCHUM 54) WHEEL [7].**

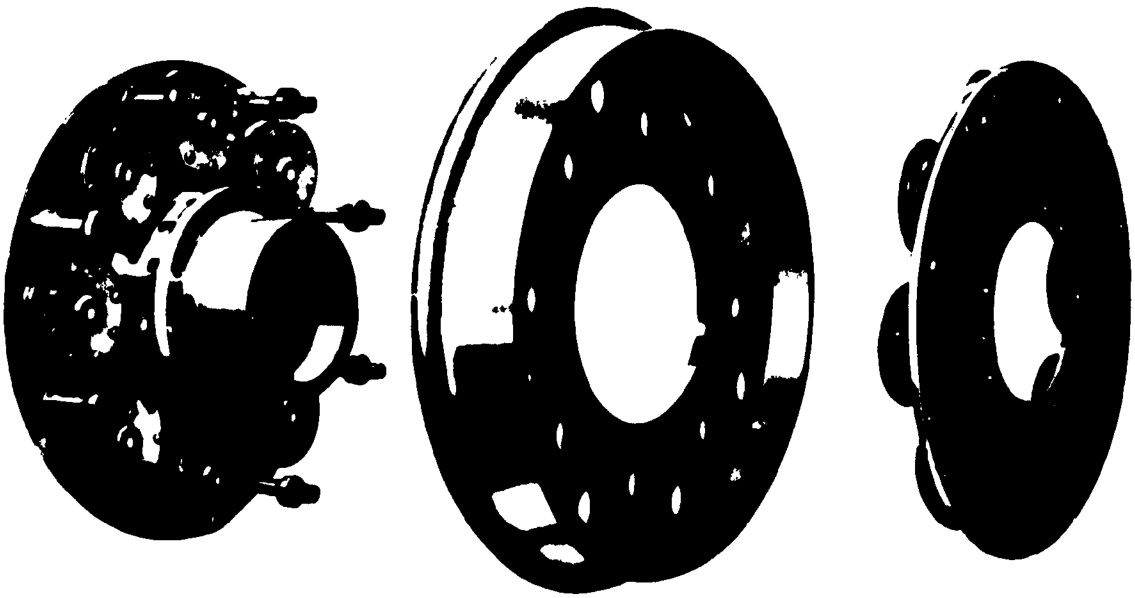


FIG. 2.2. THE SAB RESILIENT WHEEL [8].

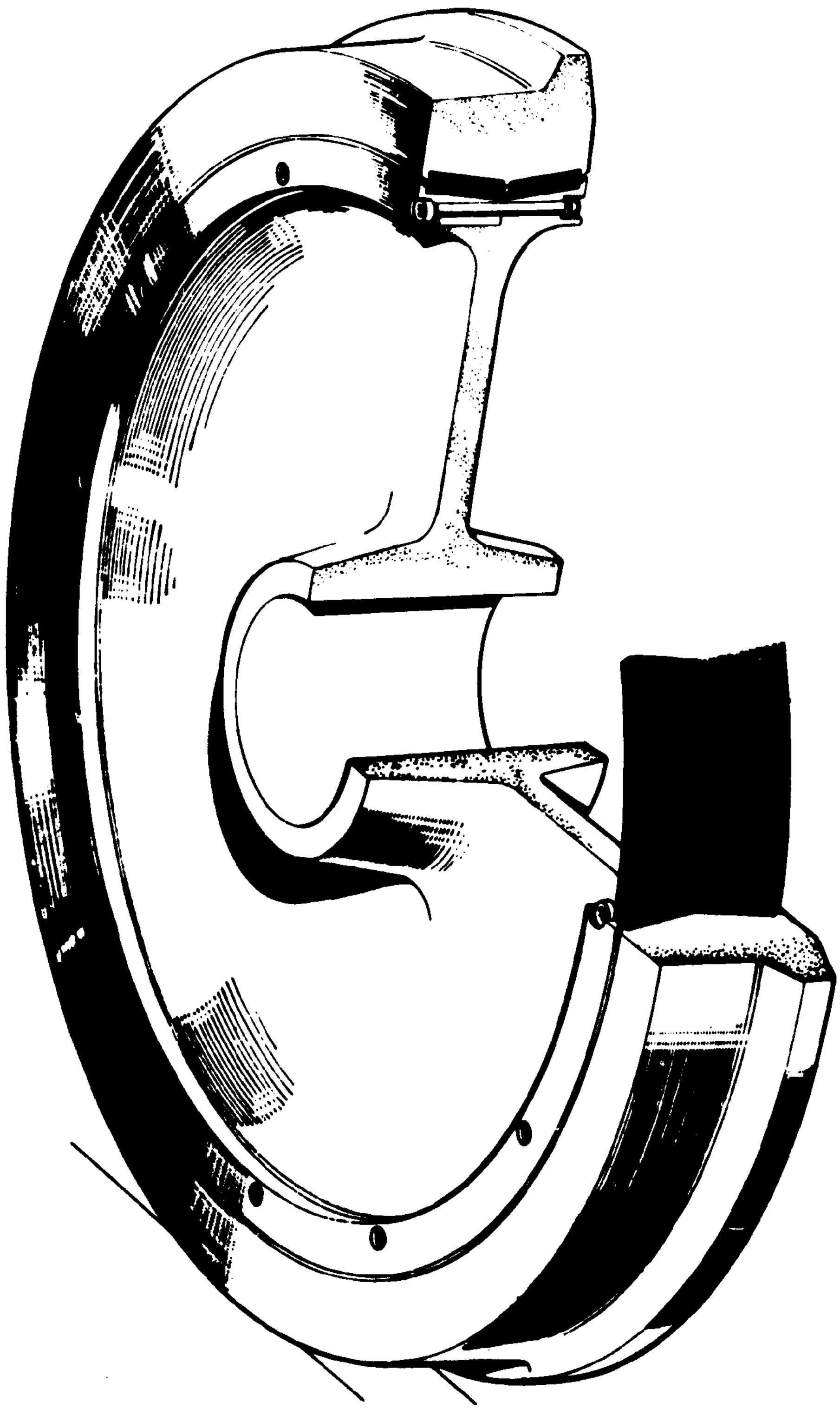


FIG. 2.3. THE SAB V-WHEEL [9].

A fourth type of resilient wheel, the Acousta Flex wheel as shown in Fig. 2.4, is in current use on the Light Rail Vehicles (LRV's) in Boston; however this wheel is no longer being manufactured and its use is being discontinued.

The Penn Cushion Wheel (see Fig. 2.1) is constructed with the rim and center plate separated by a number of closely spaced blocks, locked in place by compression. Current conductors are installed between the tire and the centerplate in the spaces between the rubber blocks. When the tire is worn out, the whole wheel can be returned to the manufacturer for re-tiring, or re-tiring may be done in a car shop with a tire replacement machine. There are over 90,000 wheels of this type in use worldwide [11]. However, Penn Machine is cautious about placing this wheel on cars with tread brakes, because of possible overheating of the wheel with resulting damage to the rubber blocks.

The SAB wheel [12], consists of four parts (see Fig. 2.2). The monobloc tire is suspended on rubber pads arranged in pairs circumferentially which are compressed between the pressure disc and the wheel center. Bolts and studs give required axial pre-compression of the rubber blocks and distribute the wheel load evenly throughout the wheel. The electrical connection is made by a number of braids (not shown in the Figure). When the truck design has inboard axle bearings, worn tire replacement is accomplished in much the same manner as changing an automobile tire. On trucks with outboard axle bearings, the tires may still be replaced without dismounting the wheel [8]. SAB has delivered over 40,000 of these wheels to railways, light and heavy rail transit, and mining operations as of January 1978.

The SAB V-Wheel (see Fig. 2.3) consists of a rimmed wheel center and a tire. Between these sections are two insert rings of synthetic rubber. Conical surfaces and bolts provide a pre-compression of the rubber rings to maintain a secure friction bond between rubber and steel. Deflection limiting flanges provide

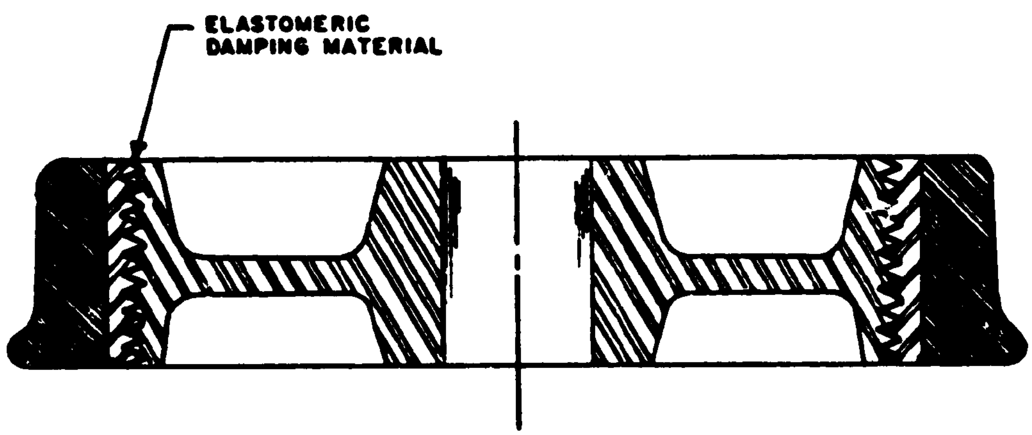


FIG. 2.4. ACOUSTA FLEX RESILIENT WHEEL [10]

protection against axial overload. As with the normal SAB wheel, wheels on trucks with inboard axle bearings can be replaced as on automobiles. These wheels have just recently been tested at the Hague Tramways in Holland [13] and at the Gothenburg Tramways in Sweden [14].

Acoustical Performance

Numerous tests of resilient wheels have been performed on both American and European transit systems. The acoustic results from tests are summarized in Table 2.1. From this summary one can ascertain that the resilient wheels all substantially reduce, and in some cases eliminate, wheel squeal noise with wayside reductions generally larger than in-car. The effect of these wheels on roar (rolling noise) and impact noise is much less noticeable, typically about 1 dBA reduction in the wayside and between 0 and 5 dBA in the car. The higher reduction observed in the car may be due to the attenuation of the vibration into the car, thus reducing the structureborne noise as well as the airborne noise. In the case of wheel squeal, the noise enters the car predominantly through an airborne path.

Problems Experienced with Resilient Wheels

The testing and in-service use of resilient wheels has identified several problems associated with their use.

The Acousta Flex wheel has experienced bonding failures on the standard light rail vehicles (LRV's) on the MBTA, on the SOAC* during testing on the Greater Cleveland RTA, and during a testing program at SEPTA [15]. In all these instances, the bond between the elastomer and the wheel rim failed, allowing the tire to begin unscrewing from the wheel hub. This additionally caused the breaking of the electrical shunts. On the MBTA LRV's the electrical shunts are also failing from what is believed to be mechanical fatique [23].

*U.S. DOT State-of-the-Art Car

TABLE 2.1. SUMMARY OF TEST RESULTS FOR RESILIENT WHEELS.

Test Location and Track Type	Wheel Type	Noise Reduction Relative to Smooth Standard Wheels dBA		Comments
		Wayside	In-Car	
SEPTA [3,15]				
tangent jointed and welded track both on elevated structure with tie and ballast track and in subway with wood ties embedded in concrete	Acousta Flex	0 to 2	0 to 2	Tests made at several speeds from 40 to 80 km/h (25 to 50 mph)
	Penn Bochum	0 to 2	0 to 1	
	SAB	0 to 1	0 to 2	
43m (140 ft) radius curve: jointed low rail, welded high rail on at-grade tie and ballast track	Acousta Flex	8 to 10	1 to 2	Virtually eliminated squeal, particularly at high frequencies
	Penn Bochum	8 to 10	1 to 2	
	SAB	3 to 4	0 to 1	Virtually eliminated high frequency squeal but introduced lower level squeal in 1250 Hz 1/3 octave band
BART [16]				
ballast and tie tangent track on embankment	Acousta Flex	0 to 1	0 to 3	Tested at 96 and 129 km/h (60 and 80 mph)
	Penn Bochum	0 to 1	1 to 3	
161.5 and 164.6m (530/540 ft) radius curves in a subway tunnel	Acousta Flex	-2 to 9	-2 to 5	Tested at 29 and 56 km/h (18 and 35 mph). Noise spectra for Acousta Flex wheels had more low frequency than either solid wheels or Bochum wheels.
	Penn Bochum	2 to 16	2 to 18	
BART [17]				
152m (500 ft) radius curve; at-grade tie and ballast track	Acousta Flex	5 to 9	-	The lower and higher values were measured 9.1m (30 ft) and 1.5m (5 ft) from the track centerline, respectively. Tests run at 24 km/h (15 mph).
	SAB	6 to 13	-	
London Transport [18,19]				
tangent track in subway tunnel	Penn Bochum	-	4.6	Speeds up to 64 km/h (40 mph)
	SAB	-	3 to 4	The lower reduction was obtained with the car vents closed; the higher with the vents open.
Transportation Test Center (Pueblo) [20]				
tangent, tie and ballast track (both jointed and welded)	Acousta Flex	2	0 to 1*	Tests performed with SOAC at speeds from 32 to 129 km/h (20 to 80 mph) *In-car noise dominated by air conditioning blowers. Wayside measurements at 15m (50 ft).
Toronto Transit Commission [21,22]				
tangent track	Penn Bochum	Insignificant	5	Speeds from 16 to 84 km/h (10 to 50 mph)
curve	Penn Bochum	See Comment		Subjective elimination of wheel squeal
Gothenburg Tramway [14]				
90m (295 ft) radius curve	SAB V-Wheel	5 to 7†	0 to 6†	†These represent reductions relative to the conventional SAB resilient wheel.

The PATCO rail transit system tested a car set of Bochum wheels in 1974 [15]. After three days of service, one truck was removed after the wheels became overheated as a result of a hand brake being applied. The other truck set remained in service for six months. PATCO concluded that:

a. the possibility exists of overheating the rubber blocks with tread brakes, especially if given a dynamic brake failure at 121 km/h (75 mph)

b. Bochum wheels cannot be used with Budd Pioneer III restrained center pivot trucks because the tire limiting flange (see Fig. 2.1) strikes the wheel hub before the car begins to turn into a curve. PATCO observed marks on the flange from striking facing frogs on curves. Lateral flexibility allowed wheels to be out of gauge on curves. The manufacturer has subsequently developed and patented a design (December 1976) to reduce this lateral displacement.

c. The copper shunt strap connecting the hub to the tire showed signs of fatigue, probably from side motion resulting from (b) above.

The Penn Bochum wheels have also been tested by the Toronto Transit Commission [21]. They found that after 13 km (8 miles) of snow-brake operation (light application of the tread brakes while running, in order to melt ice from the wheels and brake shoes under certain weather conditions), the rim temperatures reached 163°C (325°F) vs. 99°C (210°F) for standard wheels and produced a burning odor in the car. A test was run for about 16 km (10 miles) with the dynamic brake cut out (simulating dynamic brake failure). The rim temperature reach 193°C (380°F) vs. 132°C (270°F) for standard wheels.

Detailed examination of the wheels by the manufacturer showed that the rubber had not deteriorated either chemically or physically. The wheels were put back into service without changing any of the rubber blocks.

During a testing program at SEPTA [15], two Penn Bochum wheels experienced damage to the rubber blocks after a dynamic brake failure required exclusive use of the mechanical tread brake. The manufacturer determined that initial imperfections of two blocks, not detected during manufacturing, were increased due to the combination of the resulting high wheel temperatures (between 177°C (350°F) and 232°C (450°F) on the two damage wheels) and the in-service compression stresses. As a result of the SEPTA tests, the Penn Machine Company has decided not to market the Penn Bochum wheel to transit systems using tread brakes [24].

The only reported problems uncovered for the SAB wheels occurred during the SEPTA test program [15]. The handbrake on one of the test cars was applied while the car was in revenue service. This applied the tread brake to the two wheels nearest the handbrake. By the time this was discovered, the rubber inserts on one wheel had disintegrated and the shunts were broken off. The cover plate for the rubber inserts had dropped and was resting against the wheel flange. In spite of this, it appeared that the wheel was still structurally sound as no failures were found in the main bolt system. The damage to the other wheel was less severe. Some disintegration of the rubber inserts was noticeable, but the shunts remained intact and the cover plate had not dropped.

Both of these wheels experienced temperatures in excess of 275°C (527°F). It is likely that a solid steel wheel subjected to the same environment would also have been damaged.

Non-Acoustical Benefits of Resilient Wheels

In addition to their acoustical benefits, data exists indicating that resilient wheels have or result in:

a. increased wheel life over that of standard wheels by up to 50% [15,19,21];

b. reduced impact forces at joints by 40% for a locomotive with axle-hung motors [12,25,26,27];

c. reduced dynamic wheel/rail forces on continuous welded rail by about 20% [12,26];

d. reduced acceleration levels on axles, truck components, and undercar equipment by 6 to 20 dB [6,12,22,26,28,29,30];

e. reduced ground vibration levels by 4 to 10 dB over the frequency range from 40 to 250 Hz [3,25]

The above test results imply (but no explicit data have been uncovered) that resilient wheels also:

a. extend the life of the axle and truck mounted components;

b. require less frequent truing than standard wheels;

c. increase rail life;

d. reduce track and truck maintenance requirements;

e. provide a smoother, more comfortable ride for the passenger

Suggestions for Future Work

Future work on resilient wheels should emphasize design improvements which would make them acceptable for use on North American systems with tread-braked vehicles. Resilient wheels offer significant potential benefit for reducing not only squeal noise, but also ground borne vibration and wheel/rail interaction forces. The potential longer life of resilient wheels together with the implied reduced requirements for truck and track maintenance make resilient wheels potentially more cost effective (lower life cycle costs) than solid steel wheels. Improved data on these non-acoustical benefits must be obtained and incorporated into life-cycle cost equations before a more realistic cost/benefit evaluation of resilient wheels is possible.

Regarding the design of resilient wheels, improved elastomers or improved cooling of the elastomers must be provided to avoid overheating by tread brakes under most operating conditions. Even under conditions of overheating resulting from accidental overuse of hand brakes, there may be an advantage of the resilient wheels over solid steel wheels. Truly excessive use of the hand brake would destroy even solid steel wheels. As long as the resilient wheel design allows the wheels to retain their structural integrity when overheated (i.e., the wheel tire remains on the hub) it would be more efficient to replace the rubber and tire on the resilient wheels than an entire steel wheel.

Perhaps the best way to achieve an acceptable design for tread-braked systems is for the manufacturers to work directly with the transit authorities to develop the improved designs jointly.

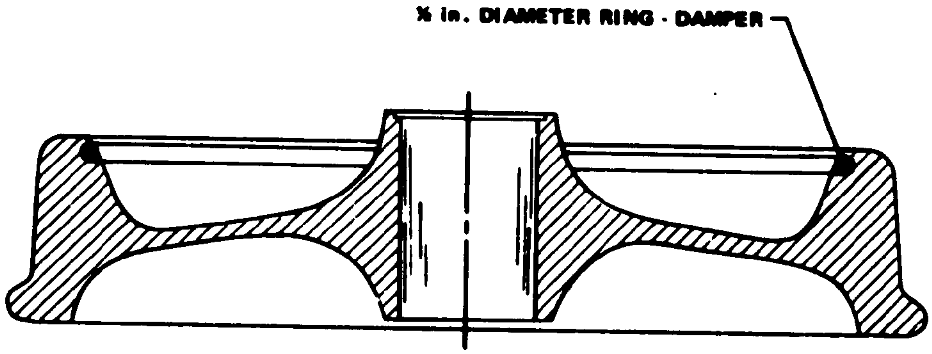
2.2 Damped Wheels

The mechanical response of a railcar wheel to wheel/rail interaction forces may be altered by increasing the internal damping of the wheel. As pointed out for resilient wheels, increased wheel damping helps suppress the pure tones characteristic of wheel squeal. It is believed that if the wheel damping exceeds the "negative damping" resulting from the stick-slip mechanism, responsible for squeal, no wheel squeal will occur [31,32,33].

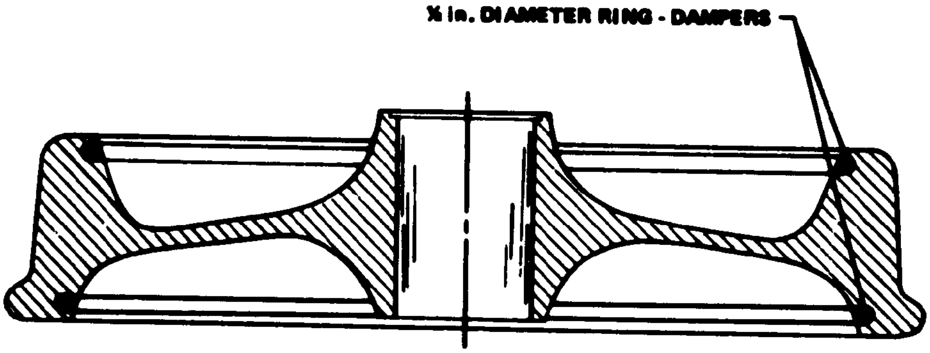
Some of the early attempts at constructing damped wheels involved annular inserts of such materials as lead [34], hardwood, and steel, either pressed into matching depressions in the rim or loosely inserted into cut-outs. Filling hollowed-out steel tires with gravel was also considered [32]. More recently, several different damping treatments have been developed and tested with considerable success. These include ring dampers, tuned dampers, constrained layer dampers, and high damping alloy wheels. Of course, the resilient wheels already discussed are also highly damped wheels.

Ring-damped wheels get their name from the damping ring, usually steel, which is snapped into a semi-cylindrical groove cut into the inner surface of the wheel rim, on either the flange or field side of the wheel (see Fig. 2.5). The rings tested in service to date have been made from steel rods, about 1.3 cm (0.5 in.) in diameter which are sprung into the groove. The ends of the ring may be left loose (as at SEPTA [3]), welded to each other (as done on London Transport [35]), or connected with an adjustable tensioning device (as tested previously by Boeing Vertol Company [36] and currently under test by the Ontario Ministry of Transportation and Communication [37]). In order for the damping ring to be effective, it must be free to vibrate in the groove. It is therefore important not to weld the ring to the wheel.

A new configuration of the ring-damped wheel is the Sumitomo force-fitted ring-damped wheel. This wheel has damping treatment placed under the rim on both sides of the wheel. The damping treatment consists of a steel ring screwed to the wheel rim. A layer of damping material covers this inner ring and an outer restriction ring is then press-fitted into place over the damping material. The press-fit is tight enough so that normal expansion of the wheel rim will not allow the damping treatment to fall off.



a. Ring-Damper on Field Side [3].



b. Ring-Damper on Field and Flange Side [3].

FIG. 2.5. VARIOUS CONFIGURATIONS OF RING-DAMPED WHEELS.

Practical tuned dampers applied to railway wheels have only been developed in the last few years. Fried. Krupp Hüttenwerke AG in Germany has developed and tested several versions [38,39,40]. To damp the resonant wheel modes, resonant vibration absorbers are fitted to the wheel rim with mounting rings as shown in Figs. 2.6 and 2.7. The steel absorbers consist of blades of different thicknesses, separated by plastic or elastomeric materials, which vibrate as cantilevered beams and whose resonance frequencies are "tuned" to the resonant vibration frequencies of the wheel. The vibration energy of the wheel is absorbed by the blades and converted into heat in the elastomeric material; hence the name tuned dampers.

By appropriately "tuning" the dampers, either the axial or radial modes of the wheel can be damped thus optimizing noise reduction for squeal or rolling noise, respectively. In tests on the Deutsche Bundesbahn (German Federal Railways), 32 dampers each weighing 0.45 kg (1.0 lb) (16 on each side of the wheel) were used [39]. This resulted in a 5% increase in the weight of the wheel. The damped wheel tested in Hamburg (see Fig. 2.7) had only 10 absorbers all placed on one side of the wheel. The absorbers suffer no wear and can be removed from worn wheels and mounted onto new ones.

Examples of the two principal types of constrained-layer damped wheels are shown in Fig. 2.8: a rim damped wheel and a center plate (web) damped wheel. As with the tuned damper, the vibrational energy of the wheel is reduced through conversion to heat in the elastomeric constrained layers.

Constrained layer damping treatments applied to wheel rims which have been tested on transit vehicles include a two-layer ring developed by the B.F. Goodrich (BFG) Company, Ohio [42,43] and a five-layer ring (see Fig. 2.8a) developed by the Soundcoat Company, Brooklyn, NY [41,44]. The BFG damping system consisted

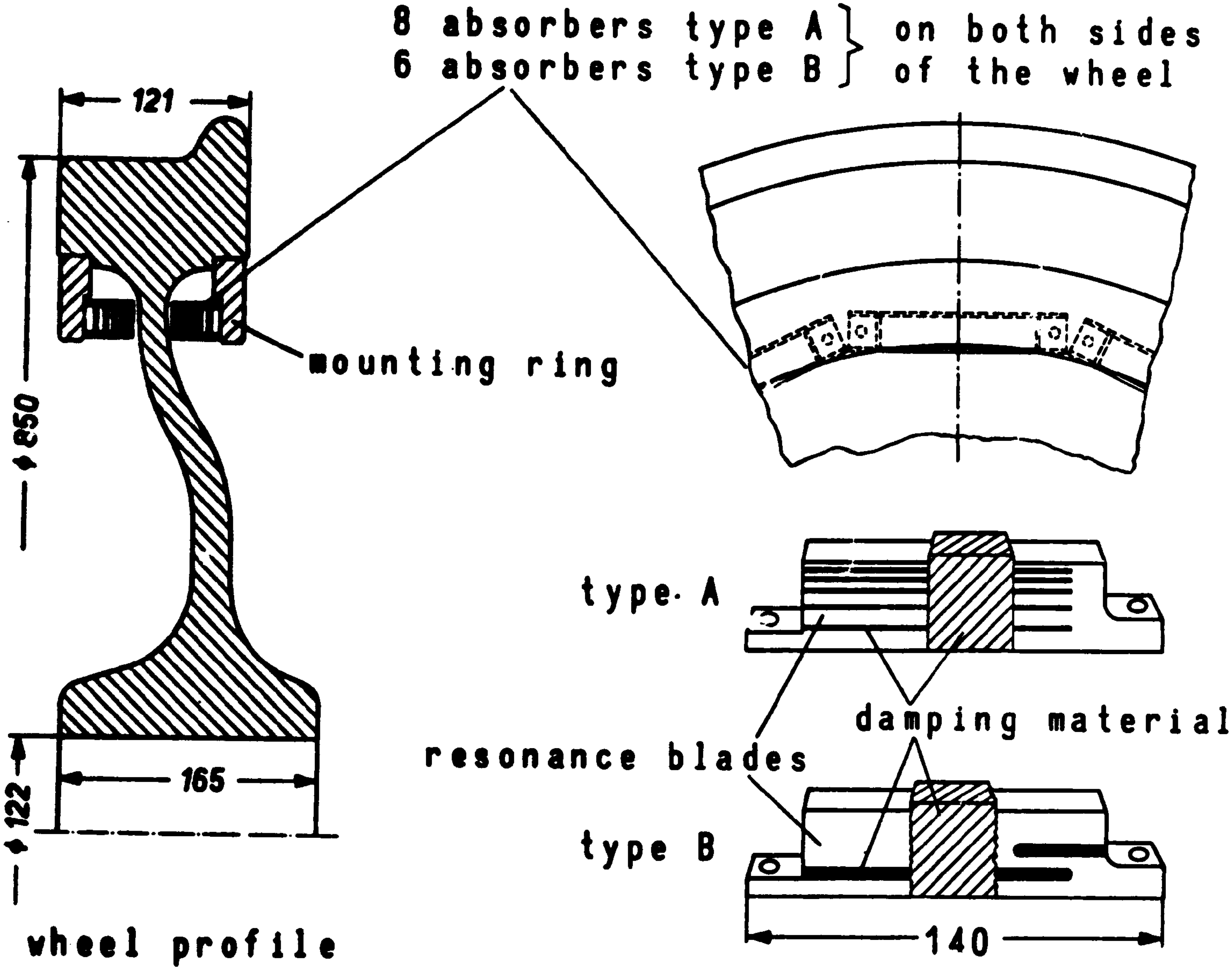


FIG. 2.6. WHEEL WITH TUNED DAMPERS TESTED IN BERLIN [38].

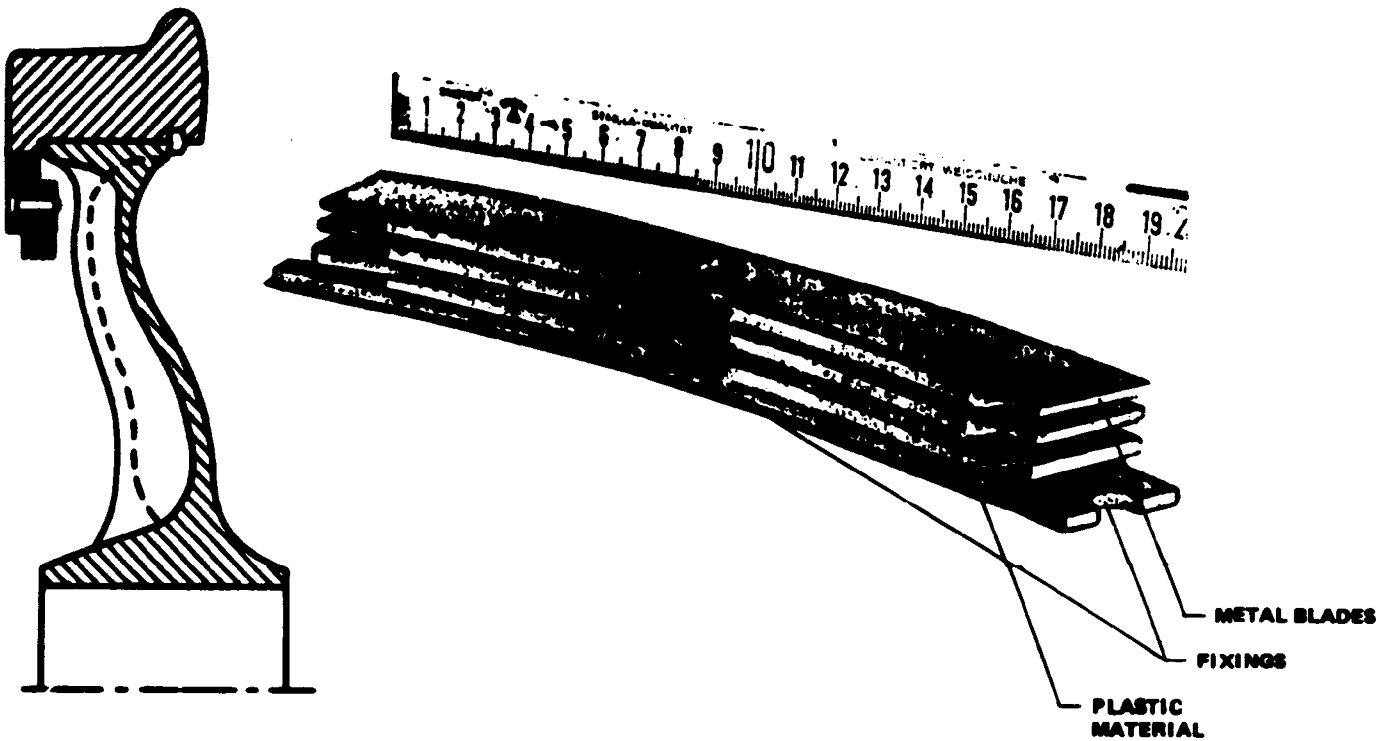
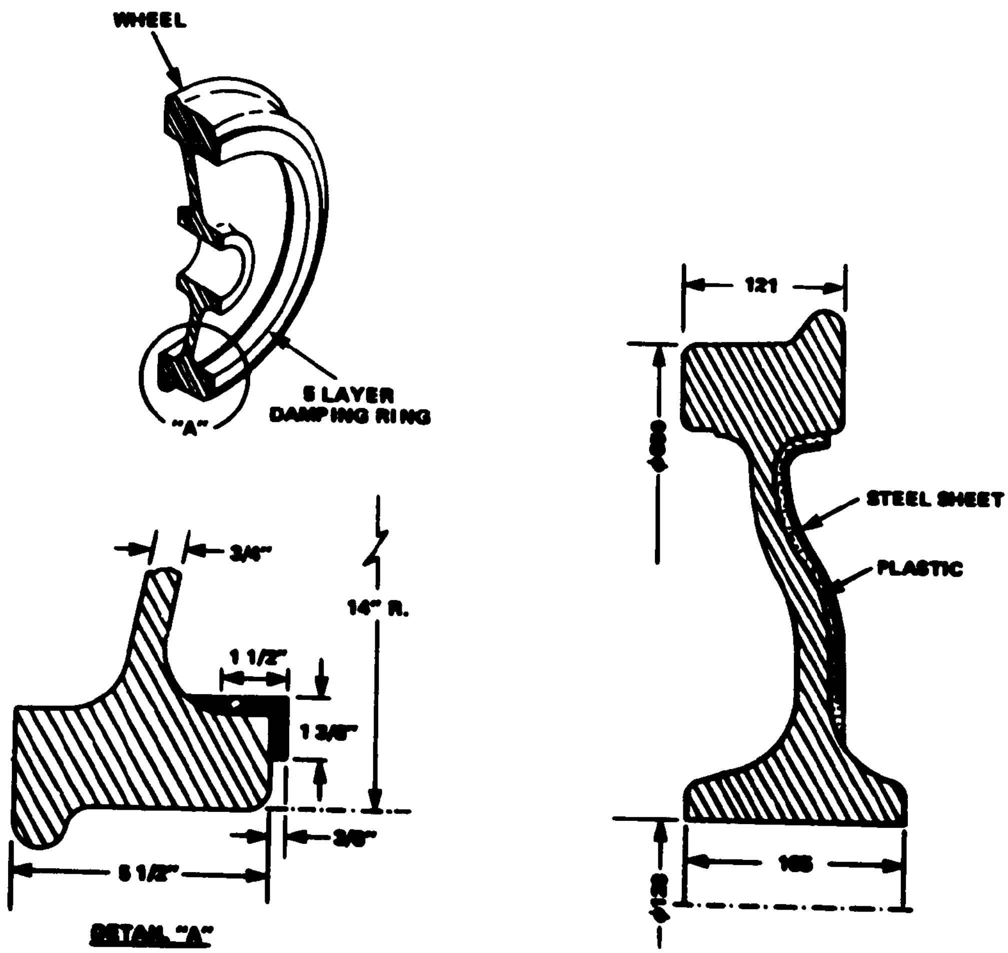


FIG. 2.7. WHEEL WITH TUNED DAMPERS TESTED IN HAMBURG [40].



(a) RIM DAMPING [41]

(b) CENTER PLATE (WEB) DAMPING [38]

FIG. 2.8. EXAMPLES OF TWO TYPES OF CONSTRAINED LAYER DAMPING.

of a 1.3 cm (0.5 in.) thick layer of BFG "Deadbeat" damping material bonded inside the wheel rim and covered with a 1.3 cm (0.5 in.) thick steel ring. The ring was designed so as not to exceed 4% of the wheel weight.

Examples of effective wheel web damping treatments include an unconstrained single layer treatment [44] and a four-layer constrained treatment [16,44] weighing 4.3 kg (9.5 lb) with a total thickness of about 0.6 cm (0.25 in.), both developed by Soundcoat, and a two-layer treatment (see Fig. 2.8b) developed by Fried.Krupp Hüttenwerke AG in Bochum, Germany [35].

To the best of this author's knowledge, the only high damping alloy wheel tested to date has been a wheel whose hub and web were formed from Incramute, a high loss factor copper alloy developed by the International Copper Research Association (INCRA). The wheel tread was steel. A small Personal Rapid Transit (PRT) test vehicle was outfitted with a set of four 35.6 cm (14 in.) diameter Incramute wheels to evaluate their performance [45].

Acoustical Performance

The results of numerous field and in-service tests of damped wheels are summarized in Table 2.2. Virtually all of the damping treatments tested substantially reduced wheel squeal. In particular, the tuned damper and constrained layer damping on the wheel rim totally eliminated the squeal tones. The magnitude of the reduction in A-weighted levels when cars travel around curves depends not only on the effectiveness of the damping treatments in reducing wheel squeal, but also how much the untreated wheels squeal and what the rolling and impact noise levels are when squeal is removed.

When ring-damped wheels were tested on the Chicago transit system (CTA) [3,47,48], the squeal component in the 8000 Hz 1/3 octave

TABLE 2.2. SUMMARY OF TEST RESULTS FOR DAMPED WHEELS.

Test Location and Track Type	Type of Damping	Noise Reduction Relative to Standard Wheels (dBA) Wayside	In-Car	Comments
SEPTA [3,46]				
All tangent track, welded and jointed	Ring-damped	-1 to +1	-2 to +2	The average noise level was unchanged by the ring-damped wheels
Curve	Ring-damped	2 to 11	2 to 3	Eliminated most of the audible squeal
Curve	Ring-damped rings stuck in grooves	0 to 5	--	Due to corrosion and/or brake dust contamination, rings became stuck in the grooves
CTA [3,47,48]				
Tangent jointed track	Ring-damped	1 to 2	0 to 1	Tests performed on both solid steel and aluminum-centered ring-damped wheels on both 2000 and 2400 series cars
Tangent welded track	Ring-damped	0 to 2	0	
Curve	Ring-damped	3 to 8	0 to 9	
London Transport [3,49]				
Tangent	Ring-damped	0	0	
PATCO [3,50]				
Curve (91.5m radius)	Ring-damped	3 to 17	--	Tests made at speeds from 8 to 24 km/h (5 to 15 mph)
BART [16,44]				
Tangent, welded, tie and ballast track on embankment	Constrained layer damping on web	0	0	4 layer damping treatment
162 and 165m (530 and 540 ft) radius curves in subway	Constrained layer damping on web	0 to 10	0 to 11	The undamped wheels did not always squeal thus the damped wheels provided little or no reductions in some cases
TTC [44]				
61m (200 ft) radius curve	Single layer damping on web	12 to 15	--	Tested at speeds from 21 to 24 km/h (13 to 15mph)
PATH [15,44]				
27m (90 ft) radius curve	Constrained-layer damping on rim	up to 32	--	5 layer damping treatment. Eliminated squeal. Measured 1.25m (4 ft) from wheel on inside of curve.
German Federal Railways (DB)[39]				
tangent, tie and ballast track, at-grade. Tested on both smooth and corrugated rail	tuned dampers	6	--	Tested at speeds from 160 to 250 km/h (100 to 155 mph). 32 absorbers per wheel tuned to radial modes. Mic placed 5 cm (2 inches) from wheel rim.
Berlin Public Transport Co. [38,39]				
series of curves on line 1 of Berlin System	tuned dampers	15	--	28 absorbers per wheel. Complete elimination of squeal. Mics placed under cars near trucks at track centerline
tangent, smooth continuous welded rail	tuned dampers	3 to 5	--	Tested at speeds from 30 to 70 km/h (19 to 43 mph). Mics placed under cars near trucks at track centerline
	constrained layer on web	3	--	
curves on line 1 in Berlin	constrained layer on web	10	--	Reduces occurrence of squeal by 70% compared to number of occurrences with undamped wheels. 2 layer treatment.
Hamburg Subway [40]				
69m (226 ft) radius curve on elevated structure	tuned dampers	up to 25	--	Complete elimination of squeal. 10 absorbers per wheel. Mic placed 2m (6.5 ft) from track centerline.
Toronto Transit Commission [42,43]				
curve	constrained layer on wheel rim	20	--	Eliminates squeal. 2 layer damping treatment
tangent	constrained layer on wheel rim	up to 2	--	2 layer treatment.
Pullman Standard Test Track [45]				Test track is for a PRT rail car with .36m (14 inch) diameter wheels.
9m (30 ft) radius curve	high damping alloy wheels	2 to 5	--	Wheels did not eliminate squeal.
tangent welded track	high damping alloy wheels	1 to 2	--	
rail joint with a .36 cm (.14 in) height mismatch across joint	high damping alloy wheels	0	--	Peak overall sound pressure level 7.6m (25 ft) from track centerline was measured. Tests made at speeds from 8 to 40 km/h (5 to 25 mph)

band was eliminated at all speeds tested. However the in-car A-weighted sound levels were dominated by squeal in the 1600 Hz 1/3 octave band. While the ring damper did reduce this component of squeal by 16 dB at a low speed of 8 km/h (5 mph), at 24 km/h (15 mph) it had no effect on the 1600 Hz component (nor on the squeal component in the 3150 Hz 1/3-octave band).

In tests at SEPTA [46], the rings were very effective at reducing the high frequency squeal components until they became "rusted" into the grooves. When replaced with new rings, the squeal was again significantly reduced. In another test at SEPTA, ring-damped wheels with rings in the field side, on the flange ride, and on both sides of the wheel were evaluated. For all practical purposes, the three configurations performed equally well.

The effectiveness of the ring dampers, and in fact any of the damping techniques, in suppressing squeal is related to the amount of damping they provide at the squeal frequencies. Table 2.3 presents a compilation of measured damping (% critical) for a variety of wheels. Note that at the lower frequencies, the resilient wheels have significantly greater damping than the ring-damped wheels. Note also that numerous configurations of the ring-damper have been evaluated. It is not clear from these results whether the mass of the ring (relative to the wheel rim mass), the damping properties of the ring, the area over which the ring contacts the groove, ring tension, or some other parameters are most important in optimizing damping. One of the highest damping configurations was the worn wheel with the steel damping ring. In this case the ring groove was cut deeper into the rim than on the new wheels. This, or the reduced tread mass, may have resulted in the high damping values.

TABLE 2.3. PERCENT CRITICAL DAMPING FOR VARIOUS DAMPED AND RESILIENT WHEELS.

Type of Wheel[1]	Frequency, Hz 500	600	1000	1560	2000	2720	4000	5180	8000
Standard [46]	.035	-	.018	-	.015	-	.007	-	.007
Standard [51]	-	.016	-	.008	-	.025	.028[2]	.013	-
Worn Standard [51][3]	.025	-	.017	-	.006	.005	.003	-	-
Steel damping ring [46][4]	.042	-	.021	-	.10	-	.07	-	.032
Steel damping ring [51][5]	-	.072	-	.256	-	.123	.309[2]	.115	-
Steel damping ring (worn wheel) [51][3]	.052	-	1.60	-	1.74	.71	.92	-	-
Hollow ring [52]	-	.022	-	.029	-	.079	.072[2]	.099	-
Hollow ring filled with oil [52]	-	.065	-	.055	-	.100	.117[2]	.146	-
Aluminum ring [52]	-	.022	-	.103	-	.261	.133[2]	.092	-
Copper/lead ring [52]	-	.055	-	.317	-	.483	.333[2]	.219	-
Extra loose steel ring [37][6]	-	.130	-	.092	-	.109	.073[2]	.049	-
Tight steel ring, 50μ strain [37]	-	.018	-	.074	-	.124	.228	.310	-
Tight steel ring, 150μ strain [37]	-	.015	-	.011	-	.067	.035	.030	-
Steel ring sprayed with anti-rusting agent[37]	-	.068	-	.088	-	.133	.347	.389	-
Steel ring coated with teflon [37]	-	.041	-	.035	-	.181	.229	.78	-
Incramute Wheel [45]	-	-	.25	-	.15	-	.1	-	.1
Penn Bochum Wheel [46]	.6	-	.7	-	.36	-	.2	-	.17
Penn Bochum Wheel [53][7]	.51	-	.31	-	.83	-	.48	-	-
Acousta Flex Wheel [46]	.6	-	.25	-	.28	-	.17	-	.15
Acousta Flex Wheel [51][8]	.61	-	-	.93	-	.35	.43	.41	-
SAB Wheel [46]	.13	-	.08	-	.043	-	.035	-	.028
SAB Wheel [53][9]	.24	-	.11	-	.083	.081	.051	.039	-

[1] Unless otherwise noted, the wheels are new. All ring dampers are 1.3cm (.5 in) in diameter.

[2] Data taken at 3920 Hz.

[3] Frequencies for worn wheel data are 520, 1280, 2240, 3240, and 4240 Hz, respectively.

[4] Data taken at SEPTA [46] are with rings on field side of wheel.

[5] All ring damped wheels tested by the Ontario Ministry of Transportation and Communication [37,51,52] had their groove on the flange side.

[6] Average data for 4 different length rings placed in regrooved wheel.

[7] Data taken at 434, 1240, 2200, and 4600 Hz, respectively.

[8] Data taken at 460, 1350, 2450, 3670, and 4960 Hz, respectively. Damping values from four tests have been averaged.

[9] Data taken at 433, 1190, 2100, 3070, 4090, and 5160 Hz, respectively.

As with the resilient wheels, the damped wheels provide little or no reduction of roar and impact noise, typically on the order of 0 to 2 dBA. One notable exception appears to be the wheels with tuned dampers. In tests on the Berlin transit system [38, 39], they provided 3 to 5 dBA reduction in roar noise. The dampers on the wheels tested on the Deutsche Bundesbahn [39], were tuned to the radial modes of the wheel and provided about 6 dBA noise reduction on newly ground, slightly corrugated, and heavily corrugated tangent welded tracks. This was measured with a truck-mounted microphone aligned radially with the wheel and placed 50 mm (2 in.) from the wheel tread. Tests were made at speeds from 160 to 250 km/h (100 to 155 mph).

A potential side benefit of wheel damping may be the reduction of brake squeal. In tests by the ORE [101], wheel damping reduced brake squeal from tread-braked wheels by up to 12 dBA.

Problems Experienced with Damped Wheels

After ten months of service on SEPTA, the ring dampers were found to be tightly bound in the wheel grooves, with an accompanying loss of effectiveness [3]. The rings appeared to have rusted in the grooves. The problem was also recently observed to have occurred on the SOAC car at the Transportation Test Center [54]. The CTA has not experienced this problem during several years of experience with ring-damped wheels. It is important to note, however, that the CTA cars are disc braked, whereas the SEPTA cars and SOAC cars are tread braked. The higher wheel temperatures and brake dust from the tread brakes could be contributing to this problem.

The placement of the groove on ring-damped wheels tested at SEPTA, requires the removal of material from the wheel tire and reduces the useful life of the wheel when the rings are placed on the field side (see Fig. 2.5a). The CTA has developed drawings for new replacement wheels which provide an extra 1 cm (3/8 in.) of metal in the tread so that the field side ring grooves will not reduce the wheel life [24].

By placing the groove on the flange side, the permissible wear of the tire is not affected. However, several transit authorities have raised the question of the grooves causing stress concentrations in the wheels that could lead to wheel failures (particularly with the grooves on the flange side of the wheel). London Transport, the only system to have widely applied ring-dampers, has not experienced problems resulting from stress concentrations at the grooves for the dampers [35]. However, plans are presently underway at PATH and NYCTA to evaluate, by means of finite element stress analysis, the possibility of the stress concentration caused by the groove leading to crack formation and wheel failure.

Constrained layer damping has the drawback of covering a portion of the wheel surface - either on the rim or on the web depending on the treatment configuration. This interferes with the visual inspection of the wheels. In a test of a five-layer rim damping treatment at PATH, the damping retaining ring fell off along the tracks due to an adhesion failure. Since the rings were assembled in the shop with "C" clamps, control of bonding of the rings was marginal [11,15]. In these same tests, the treatment interfered with the wheel turning machine operation. This same treatment has more recently been evaluated on the Paris Metro (RATP) and was sufficiently successful for them to order 800 rings [55]. Transit authorities in the U.S. have expressed concern about the effect of possible high wheel temperatures on the damping material.

Tests of the tuned dampers over a two year period in Berlin showed no degradation in the performance of the tuned-damper wheel with time [24]. The wheels must be handled with slings; picking up with hooks can damage the dampers. The damping material can withstand temperatures up to 200°C (392°F). Berlin is currently outfitting 26 new subway cars, and Hamburg 100 subway cars, with these wheels [24].

Suggestions for Future Work

Ring dampers appear to have the potential for being a simple, cost effective method for the control of squeal noise. However, a means must be found to prevent the rings from becoming bonded, or adhering, to the grooves. Laboratory tests [37] have indicated that an anti-rusting agent sprayed on the ring does not reduce its damping effect on the wheel. A simple in-service evaluation of this method could be performed on SEPTA since they are still running the test vehicles with the rings now "frozen" in place. If the bonding of the rings to the wheels is due to foreign material such as brake dust, use of a corrosion resistant ring may not solve the problem; a coating of some kind may be required to keep foreign material out of the ring groove [3].

In order to make the ring damper more effective, its damping characteristics at low frequencies (<2000 Hz) must be improved. Research is needed to better understand the nature of the damping mechanism so that the damping can be optimized. The importance of the ring mass, mechanical properties, tension, surface roughness, and groove location and geometry need to be investigated. With an optimized ring, it may be possible to achieve a performance approaching that of the tuned damper. Finally, the safety of cutting the grooves on the flange side of the wheel must be assessed.

Of the constrained layer treatments, the rim damping treatment appears to have the greatest potential. The results of their in-service use on the Paris Metro should be carefully monitored and reviewed. Similarly, the in-service performance of the tuned-damper wheels in Berlin and Hamburg should be closely followed. Ideally, an in-service evaluation on a U.S. transit system of an optimized ring-damper, a constrained layer rim damping ring, and a tuned damper should be performed. The wheels should be placed on vehicles of the same kind, put into the same train consist, and monitored over a one to two year time period.

Of the wheel damping treatments, the tuned damper appears to show the best potential for reduction of roar noise. This treatment could possibly assist in reducing wayside noise on new transit systems for which curve squeal and joint impact noise are not problems. An evaluation of the tuned-damper, designed to reduce roar noise, should be performed on one of the new U.S. systems.

2.3 Spoked Wheels

Spoked wheels have a smaller noise-radiating area than solid web wheels and as a result should be less efficient radiators of wheel noise. Tests performed in Philadelphia in 1964 [56] found that the spoked wheels reduced the in-car noise by 3 dBA and 2 dBB on an elevated structure, increased the in-car noise by 2 dBA and 0 dBB in a subway tunnel, and reduced exterior noise by 5 dBB on an elevated structure. The report, however, states that "since the noise level varies with the speed, roadway and track conditions, it can be stated conclusively that this test did not establish the exact decibel improvement attributable to spoked wheels."

More recently [57], the French National Railways (SNCF) outfitted a six car train with two car sets of spoked wheels, two car sets of monobloc steel wheels, and two car sets of steel wheels with a shrunk-on steel tire. Microphones were placed near the trucks of each car and the noise levels measured on repeated runs over a test section at 80 and 110 km/hr (50 and 68 mph). The solid (monobloc) steel wheels were found to be the quietest with the spoked wheels and tired-wheels being 2 and 3.5 dBA noisier, respectively.

The only literature found which attributed a significant noise reduction potential for a spoked wheel was a theoretical and laboratory evaluation of the two idealized wheels shown in Fig. 2.9 [58]. The investigation found that the spoked wheel radiated 8 to 10 dB (no weighting specified) less noise. This was believed to be due to both the lower vibration levels (3 to 5 dB) on the spoked wheel (the spokes were thicker than the solid web) and the smaller radiating area. These results were obtained with the wheel contacting a horizontal "rail" at the center of the tread. When the rail was canted by 1/10, the contact with the wheel moved 2.5 mm (0.1 in.) from the center of the tread face and the resulting noise levels increased by about 7 dB for the spoked wheel. No similar data were presented for the solid web wheel.

No data have been found on the effect of spoked wheels on wheel squeal; however, it is not expected that squeal noise levels would be significantly reduced.

In addition to the questionable acoustical effectiveness of spoked wheels, they have some negative aspects which must be considered. For example, the spokes would probably need to be highly stressed and, therefore, would be subject to fatigue failure. Also, the wheels have a circumferentially nonuniform radial impedance which could lead to parametric excitation as the wheel rolls on a rail [1].

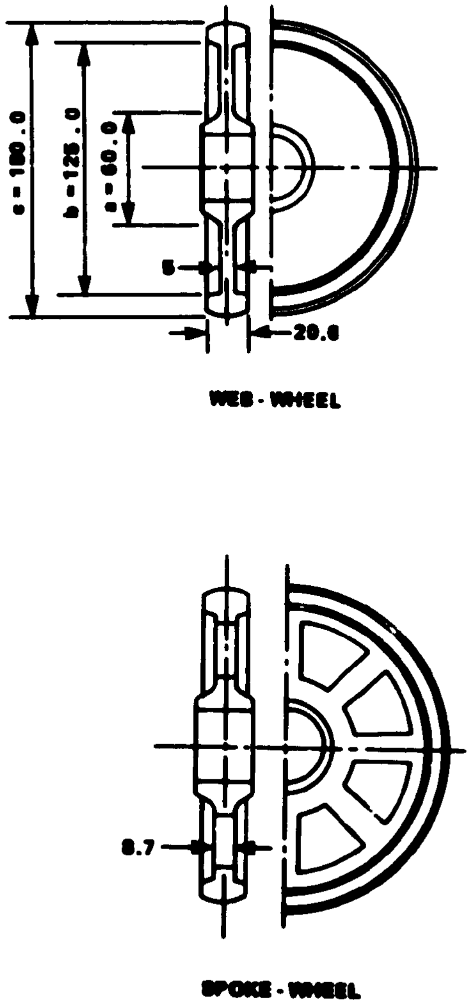


FIG. 2.9. IDEALIZED WHEELS EVALUATED THEORETICALLY AND EXPERIMENTALLY [58].

While many questions remain concerning spoked wheels — how thin can the spokes be? What shape would minimize wheel vibration and noise reduction? What are their in-service problems and wheel life? — it is not believed that future research should emphasize this approach to wheel/rail noise reduction.

2.4 Resiliently Treaded Wheels

Wheels incorporating some resiliency in their tread can potentially reduce noise by two mechanisms. A more resilient tread will have a *lower contact stiffness,* enabling the tread to deform about any wheel or rail irregularities, thereby reducing the excitation applied to the wheel and rail. Furthermore, a *larger area of contact* between the wheel and rail will result if the compliance of the tread is increased. The area of contact acts like a filter, effectively filtering out those wheel and rail surface irregularities having wavelengths on the order of, or less than, the dimensions of the contact area. Since the resiliently treaded wheels will have a larger contact area, they will more effectively filter out the excitation due to these irregularities.

Two types of resiliently treaded wheels have been envisioned: a thin-tread wheel such as that shown in Fig. 2.10, and a nonsteel-treaded wheel, such as the Nitinol (nickel-titanium alloy) treaded wheel (also called the Tinel wheel).

The thin-tread wheel, shown conceptually in Fig. 2.10, achieves its resiliency (lower contact stiffness) by making the tread from a thin steel ring. The ring is supported at its edges by the body of the wheel and is allowed to deflect in bending into the cavity in the body of the wheel. Preliminary estimates indicate [59] that for a tread thickness of 1.78 to

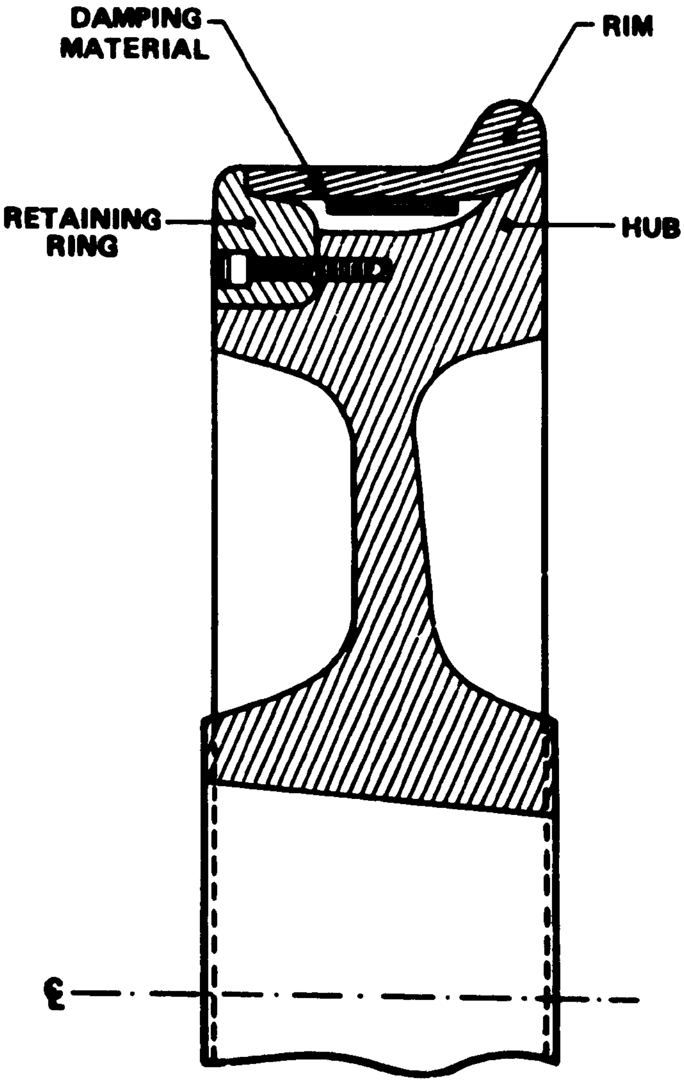


FIG. 2.10. RESILIENTLY TREADED WHEEL CONCEPT.

2.03 cm (0.7 to 0.8 in.) noise reduction of about 7 dBA would be achieved with manageable stresses in the tread.

Damping can be added to the wheel by use of nonstructurally supporting damping material placed under the tread ring (as shown in Fig. 2.10).

Braking presents a special problem that only in-service tests can fully evaluate. Heating caused by tread braking has been known to cause deterioration of the elastomeric materials in existing resilient wheels. The thin steel tread used in this concept may worsen these problems. However, since the damping material included in the concept shown in Fig. 2.10 is not load bearing (i.e., it does not structurally support the tread), any potential failure or degradation of this material due to overheating would not lead to wheel failure. Furthermore, the use of disc brakes in place of tread braking is being seriously considered in new subway car construction. Disc brakes would eliminate the tread heating problem. As currently envisioned, the treads on these wheels would be replaced rather than trued to maintain a sufficiently smooth running surface. Finally, although the enlarged contact area may result in improved traction (which could reduce the incidence of wheel flats and rail burns), it may also lead to high tread stresses. Consequently, tread durability is a serious concern. Future research should focus on developing improved conceptual designs for these wheels, which should then be fabricated and tested, first in the laboratory and then at a test track.

Titanium treaded wheels were first proposed by Remington, et. al [1]. Titanium has a lower elastic modulus than steel but a higher tensile strength. As with the resiliently treaded wheel, the resulting increase in the size of the wheel/rail contact area would reduce roar noise. This reduction, however, has been estimated to be small, about 1 dBA [2].

More recently, the Raychem Corp. has investigated the use of a nickel-titanium alloy, called Nitinol, for the wheel tread. Scale model tests of this wheel have indicated roar noise reductions of 2 to 4 dBA and elimination of squeal noise [60].

Using a roller rig, the adhesion of the Nitinol-treaded and steel wheels on a steel rail were investigated for a variety of surface conditions (dry, wet, and fuel contaminated surfaces) [61]. For a perfectly dry surface or a wet surface, the two metals behaved similarly, with steel performing slightly better. However, with the surface contaminated with diesel fuel, Nitinol seems to be superior. For example, on the fuel contaminated rail, the Nitinol alloy can attain a tractive coefficient above 0.18 while the steel only approaches 0.06. A large degree of variability in the Nitinol friction creep curve was observed in the scale model tests and the specific behavior recorded cannot at present be explained.

The wear characteristics of a Nitinol wheel rolling on a steel rail appear considerably superior to those of a steel wheel. In the scale model experiments, the wear rates were measured through a microscopic analysis of the surface. The results show that the wear produced by the Nitinol on the dry steel was 1/14 that of steel on steel. Furthermore, the wear of the Nitinol wheels themselves was about 1/4 of that observed for the steel wheels. In addition, the observed rate of increase in surface roughness is not as rapid for the Nitinol [61]. These results are for pure rolling. The wear properties of Nitinol on steel under sliding is not as favorable [62]. It is uncertain what effect the Nitinol wheels might have on wheel flats, or on wheel tread wear in curves.

Future research on the Nitinol-treaded wheel should clarify the results of the previously performed scale model tests in which the wheel/rail contact area may not have been properly scaled. In addition, a thorough life cycle cost analysis is needed to determine whether the possible long range benefits resulting from reduced wheel and rail wear can offset the high initial cost of the wheel (about $3000 per wheel).

The possibility of using a polyurethane coating on the wheel tread has been considered [63,64]. Although it was estimated that 14 dB reduction in roar noise might be achieved (based on tests with a 2 cm (3/4 in.) thick polyurethane coating on a rail) [63], calculations [64] showed that commercially available polyurethane (in 1973) could not be used because the rate of heat generation in the tire was at least twice the rate at which the heat could be dissipated without exceeding temperature limitations of the material. The author noted, however, that the possibility remained open that a better polyurethane would become available or that epoxy based compounds might be used instead of polyurethane. An attempt should be made to determine whether new plastics or other materials might be suitable for this purpose. If such a wheel were to be feasible, a means for providing electrical continuity between the rail and the wheel would need to be incorporated into the design.

2.5 Wheel Covers

Noise barriers close to the wheel's radiating surfaces (and lined with absorptive materials) may effectively control wheel noise radiation. One design, developed by the Deutsche Bundesbahn [65], is shown in Fig. 2.11. In order for the wheel cover not to become a significant noise radiator itself, it must be isolated from the wheel. The wheel in Fig. 2.11 was reported to have

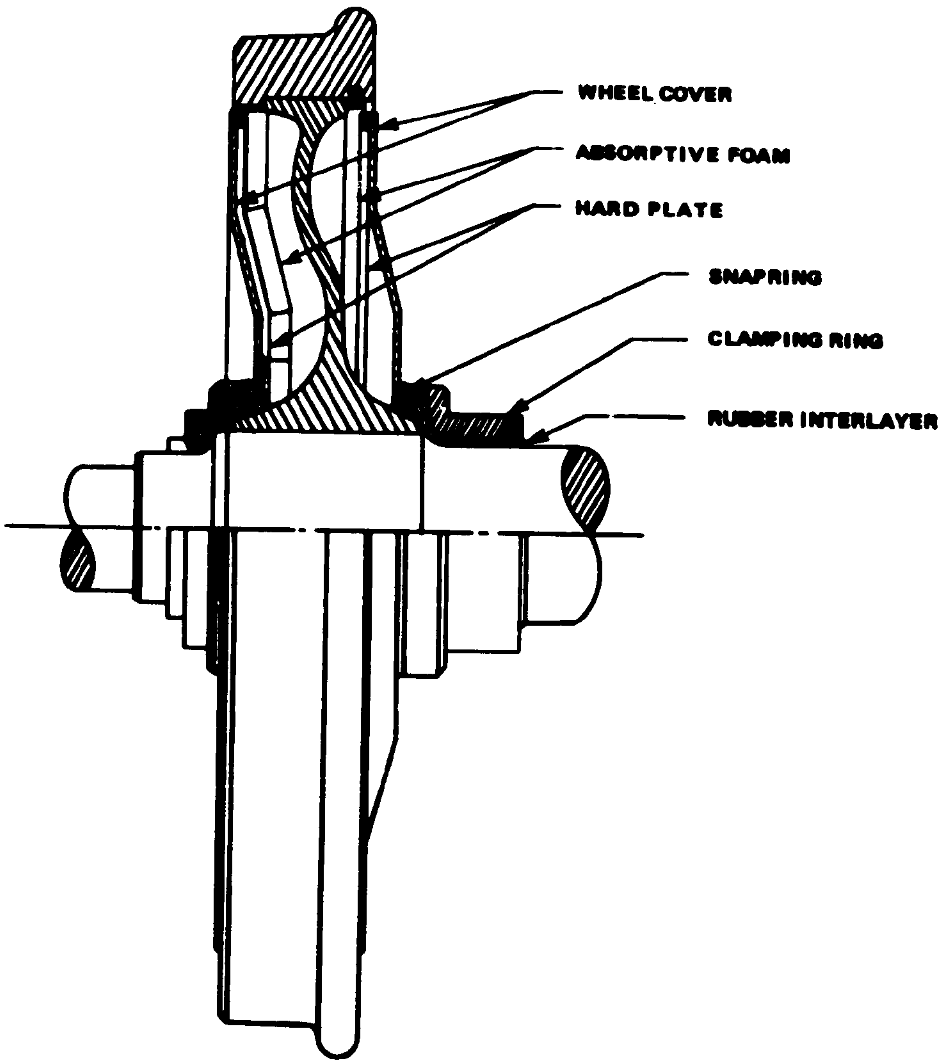


FIG. 2.11. GERMAN WHEEL COVER DESIGN [65].

resulted in a 4 dBA reduction in noise with the wheel cover alone, and a 5 dBA reduction with the absorptive foam and the wheel cover [65]. The article did not indicate whether this reduction was obtained with the wheels on a roller rig test machine or on a train; however it is assumed that the former is the case and that the measured reductions apply only to the noise from the wheel. It is also not known whether the noise reduction reported represented roar or squeal noise.

This treatment does not appear to be practical for use on U.S. transit systems because it would prevent visual inspection of the wheel. Furthermore, there may be interference with guard rails, frogs, etc. It does, however, appear to be a valuable research tool in that it provides a method for determining the importance of the wheel web (vs. the wheel tire) in noise radiation and for better understanding the relative importance of wheel and rail in noise radiation.

2.6 Wheel Truing

Wheel truing has been used throughout the history of railway operation to restore the wheel tread and flange to the proper profile or to correct tread defects such as flats, shelling and spalling. Methods for wheel truing used today include: underfloor machines (both milling and lathe cutter type) that true the wheels without having to remove them from the vehicle; above floor lathes which require the removal of the axle from the truck; belt grinders which can only be used to correct small tread defects [e.g., flats of length less than 2.5 to 3.8 cm (1 to 1.5 in.)]; and abrasive brake shoes which are occassionally substituted for the normal tread brake shoe on the vehicle which is then run at moderate speeds with the brakes applied in order to remove small flats.

The underfloor lathe can true about 4 axle sets (8 wheels) per 8 hour shift although one newer model machine can true 8 to 10 axles per shift [66]. Because the axles need to be removed from the trucks when truing on an above floor lathe, the number of axles sets which can be trued in an 8-hour shift is only between 1 and 2 (2 to 4 wheels) [15]. About 5 axles per day can be trued on the belt grinder [67]. The replacement of the normal brake shoes with the abrasive brake shoes and the running of the car to smooth the wheel is about a two-hour procedure [15].

Acoustical Performance

The noise reduction achievable from wheel truing is very dependent on the condition of the wheels before truing. In tests on SEPTA [3], only moderate noise reductions (0 to 5 dBA) were obtained, primarily because the wheel treads had no visible flat spots. In contrast, tests on the SOAC at the U.S. DOT's Transportation Test Center [20] showed reductions of 10 to 11 dBA for a car with 12 flats of length less than 2.5 cm (1 in.) and 8 flats greater than or equal to 2.5 cm (1 in.). A reduction of 8 to 9 dBA was observed on a car with 10 flats of length less than 2.5 cm (1 in.) and 3 flats longer than or equal to 2.5 cm (1 in.). A summary of various test results for wheel truing is given in Table 2.4.

The data show no consistent difference between lathe-trued and milling-cutter-trued wheels. The wayside measurements tend to indicate that the new (lathe-trued) wheels are 1 to 2 dBA quieter than the milling-cutter-trued wheels. However, the trend seems reversed for the in-car measurements.

On the NYCTA, the wheel roughness seems to increase to a greater extent than an SEPTA (see Table 2.4). This is believed

TABLE 2.4. SUMMARY OF WHEEL TRUING RESULTS. THE RESULTS SHOWN ARE THE LEVELS WITH WHEELS IN A "TEST CONDITION" RELATIVE TO WHEELS IN A "REFERENCE CONDITION" [Adapted from Ref. 3].

Track Type	Wheel Condition		Change in Sound Level (dBA)	
	Test	Reference	Wayside	Car Interior
SEPTA [3]				
Ballast & tie, welded	New[1]	Worn, 12 Mo.	-2 to -3	0 to -5
Ballast & tie, welded	Trued[2,3]	Worn, 12 Mo.	+4 to -2	+4 to -2
Ballast & tie, welded	Worn, 12 Mo.	Worn, 24 Mo.	0	0
Ballast & tie, jointed	New[1]	Worn, 12 Mo.	-1 to -2	0 to -3
Ballast & tie, jointed	Trued[2,3]	Worn, 12 Mo.	0 to -2	+3 to -1
Ballast & tie, jointed	Worn, 12 Mo.	Worn, 24 Mo.	0 to -1	0 to -2
Subway, welded	New[1]	Worn, 12 Mo.	-	0 to -3
Subway, welded	Trued[2]	Worn, 12 Mo.	-	-4 to -5
Subway, welded	Worn, 12 Mo.	Worn, 24 Mo.	-	-2 to -4
Subway, jointed	New[1]	Worn, 12 Mo.	-	0 to -3
Subway, jointed	Trued[2]	Worn, 12 Mo.	-	-4 to -5
Subway, jointed	Worn, 12 Mo.	Worn, 24 Mo.	-	-2 to -4
Curve	New[1]	Worn, 12 Mo.	-4 to -5	-3 to -4
Curve	Trued[2]	Worn, 12 Mo.	-2 to -5	-2 to -6
Curve	Worn, 12 Mo.	Worn, 24 Mo.	0	0
WMATA [3]				
Ballast & tie, welded	Flatted	Smooth	+9 to +10	-
NYCTA [68]				
Ballast & tie, welded	Smooth	Worn, 3-8 Mo.	-5 to -8	-
TRANSPORTATION TEST CENTER [20]				
Ballast & tie, welded	Smooth	Flatted	-8 to -11	-

[1]New wheels have been trued on a lathe.

[2]Wheels were trued on an underfloor milling cutter truing machine.

[3]Wheels were tested immediately after truing. The noise levels were higher than before truing apparently because the cutter marks had not been smoothed off.

to be due to the use of cast iron brake shoes in the older NYCTA cars as opposed to the composition tread brake shoes in SEPTA. The wheels quickly become spotted with dime size flats and truing results in a greater noise reduction than on "unspotted" wheels.

Wheel truing also appears to have a beneficial effect on wheel squeal, reducing the levels by 2 to 6 dBA relative to wheels with 12 months of wear.

The data from SEPTA indicate that the wheels require a short run-in period after truing before the full beneficial effect of the truing can be realized. Before the end of the run-in period (on the order of 2 days to 2 weeks), the wheels may create more noise than before they were trued (if wheel flats had not been present) [3].

Non-Acoustical Benefits

In addition to reducing wheel/rail noise, wheel truing has been found to reduce ground vibration levels above 100 Hz by up to 10 dB at SEPTA [3]. Removing large defects from the wheel surface also reduces wheel/rail loads and hence would most likely reduce track and truck component failures and maintenance requirements.

Suggestions for Future Research

Too frequent truing, or truing when wheels are not sufficiently rough will lead to reduced wheel life. Therefore, it is important to develop suitable criteria for when to true wheels. In an effort to reduce spotted wheels in the NYCTA system, the following criteria have been adopted [3]:

1. Any wheel with a flat spot of 2.5 cm (1 in.) or greater in length shall be reported for immediate truing.

2. Any wheel with a series of flat spots of 1.9 cm (3/4 in.) to 2.5 cm (1 in.) in length in which the total length of all spots in one quadrant (1/4 of the total circumference) of the tread is 10.2 cm (4 in.) or greater shall be reported for truing as soon as practical.

These and other suitable criteria should be reviewed with a goal of developing optimized criteria for truing for wheel/rail noise control. These criteria should not only indicate when to true, but also how to true. For example, dime sized flats can be removed more cost-effectively with belt grinders or possibly with abrasive brake shoes than with underfloor truing machines. Above floor lathes appear to be the least cost effective.

As a possible aid to developing an effective wheel truing program, a wheel flat or roughness monitor should be utilized. Using such a monitor (see Section 5.1) the Toronto Transit Commission is finding about 10 wheel flats per day [69]. For their fleet size and average annual mileage, this corresponds to a rate of wheel flat development of about 1 flat for every 100,000 wheel-miles traveled. This is essentially the same rate of wheel flat development reported by PATH [70]. Extrapolating to the NYCTA, this rate of flat development would result in about 75 wheel flats per day. Clearly it would be advantageous to find methods for preventing flat formation. Such methods might include slip/slide protection (see Section 4.1), replacement of cast iron brake shoes with composition shoes (see Section 4.6), improved methods of rail or wheel flange lubrication (see Section 3.6) and improved wheel metallurgy.

In order to understand the relative importance of wheel truing and rail grinding in reducing noise (since a smooth wheel on a rough rail is probably no quieter than a rough wheel on a smooth rail) and to understand more fully the benefits of surface finish, measurements of wheel and rail roughness before and after truing and grinding should be taken in conjunction with noise measurements. This data can be used to validate the analytical models developed by BBN [2], which in turn can be used to develop wheel truing criteria.

2.7 Wheel/Rail Interface Geometry: Wheel Profile, Rail Cant, Wheel/Rail Gauge

Very little is known about the effects of wheel profile, rail cant, and wheel/rail gauge differences on noise generation. These geometry variables are very interrelated and it is known that they can greatly affect truck hunting and wheel and rail wear, both of which in turn can affect wheel/rail noise.

Various combinations of wheel profile and rail cant can be used to improve hunting performance while maintaining satisfactory guidance with minimum flange contact, in turn reducing the noise and wear resulting from the impact of the wheel flange on the gauge side of the rail. Experiments indicate that a wheel taper of 1:40 with a corresponding rail cant of 1:40 is a favorable combination for preventing truck hunting at high speeds [71,72].

In a laboratory experiment with an idealized railway wheel (see Fig. 2.9), changing the slope of the rail surface from 0 (horizontal) to 1/10, thus moving the wheel/rail contact from the center of the tread face to a point 2.5 mm (0.1 in.) from the center of the tread, resulted in a 7 dB increase in the wheel rolling noise [58]. The hypothesis presented was that the out-of-plane excitation of the wheel resulted in the increased noise.

Truing wheels to a worn or "Heumann" type profile has been found to decrease wheel wear and increase the intervals between needed wheel truings [73]. The Hamburg transit system has reported that truing their wheels to a worn profile has increased the intervals between truings from 250,000 km (155,000 mi) to 595,000 km (370,000 mi) [74]. No data have been reported on the effect on noise of truing the wheels to a worn profile. However, data from SEPTA [3] (see Table 2.4) indicated that wheel squeal levels increased by 2 to 5 dBA from new or trued condition to worn condition. It is not clear whether the wheel profile or the surface roughness is primarily responsible for this increase. Furthermore, even the differences in noise resulting from conical and cylindrical wheel profiles are not known.

Keeping the difference between wheel gauge and rail gauge to a minimum will reduce truck hunting; existing system differences vary all the way from 0.6 cm (0.25 in.) to greater than 2.5 cm (1 in.). It is claimed that the smaller the difference, the less the noise created, but the greater the rail wear [71]; however, this has not been verified in controlled experiments. In fact, the opposite effect on noise has been reported to occur on the Greater Cleveland RTA (GCRTA) rapid transit system [75]. This system has a track gauge of 1.429 m (4 feet 8-1/4 in.), 0.6 cm (1/4 in.) tighter than standard, with wheel gauge set for standard track gauge. The resulting decrease between wheel and track gauge appears to excite flange modes of the wheel which result in flange "singing." Inspection of the wheel revealed that the fillet between the flange and tire on cars at the GCRTA is a smaller radius than on other systems, confirming that the flanges receive more excitation at the GCRTA than was observed on other systems [75].

Because of the strong interrelationships among these wheel and rail geometric variables and their unknown effects on noise, future research should emphasize controlled and in-service experimentation using alternate wheel profiles, and wheel/rail gauge clearances to assess their effects, most notably on squeal noise. The in-service tests of the best configurations from a noise point of view will have to be performed over a sufficient period of time to determine their effect on system operations and maintenance.

3. RAIL AND TRACK TREATMENTS

3.1 Rail Grinding

Rail grinding is used by both rapid transit systems and railroads primarily for removing mill and weld imperfections from new rail and for reprofiling and removing corrugations, flaking, head cracks and rail burns (due to wheel slip) from worn rail. Two types of rail grinders are available: abrasive block grinders where abrasive bricks are pulled along at revenue speeds [32 to 64 km/hr (20 to 40 mph)] while being pressed on the rail surface; and grinding wheels which rotate as they are pulled slowly [3 to 5 km/h (2 to 3 mph)] along the track. An abrasive block grinding train such as that used on the CTA (which has 14 abrasive bricks on each rail) requires about 110 passes over a rail section to fully smooth the surface [15]. A rotating grinding stone train such as that manufactured by Speno Rail Services Inc. with 12 grinding wheels over each rail can remove about 0.0038 cm (0.0015 in.) of steel per pass and requires about 2 to 3 passes to smooth the rail [15].

Acoustical Performance

Table 3.1 summarizes the results of numerous tests of rail grinding, both in the U.S. and in Europe. As with wheel truing, the effectiveness of rail grinding is highly dependent on the rail surface condition before grinding. Summarizing the information in Table 3.1, it appears that grinding normally worn (uncorrugated) tangent track (either welded or jointed) results in about a 1 to 3 dBA noise reduction. Grinding new rail (which has mill and rust on the running surface) reduces roar noise by about 6 dBA. Grinding corrugated rail reduces roar noise by 10 to 15 dBA. On curves, grinding has no consistent effect on wheel squeal - in some case the levels increase, in others they decrease.

TABLE 3.1. SUMMARY OF RAIL GRINDING RESULTS

Test Location and Track Type[1]	Rail Condition Before Grinding	Wheel Type and Condition	Reduction in Noise Level Due to Grinding (dBA)	
			Wayside	Car Interior
SEPTA [3]				
Tangent welded, tie and ballast	Worn, 1 to 2 yr	Standard & resilient (new)	0 to 4	0 to 1
Tangent jointed, tie and ballast	Worn, 1 to 2 yr	Standard resilient and ring-damped	0 to 4	0 to 3
Tangent welded, concrete invert in subway	Worn, 1 yr	New/trued standard	-	0 to 2
		Worn standard	-	0 to 1
		Resilient	-	1 to 3
Tangent jointed, concrete invert in subway	Worn, 1 yr	New/trued standard	-	0 to 2
		Worn standard	-	0 to 1
		Resilient	-	1 to 3
43 m (140 ft) radius curve, tie and ballast	Worn, 1 yr	New/trued standard	-2 to +3	0 to 3
		Worn standard	-2 to +6	2 to 3
		Ring-damped	2 to 3	0 to 1
BART [16,76]				
Tangent welded, tie and ballast.	New, no wear	New standard	6 to 9	4 to 5
Tangent welded, direct fixation on aerial structure	Slightly worn[1]	Resilient	1 to 2	-
161.5 & 164.6 m (530 & 540 ft) radius curve in subway	New, no wear	New standard	-7 to +7	-9 to +6
		New visco-damped	-8 to +10	-6 to +8
CTA [77]				
Tangent welded, tie and ballast	Worn	Trued standard	0 to 3	1 to 2
		Trued standard[2]	4	2 to 3
Tangent welded, tie and ballast	New, no wear	Trued standard	3 to 4	-
		Trued standard[2]	7	-
MUNICH [78]				
Tangent welded, tie and ballast	New, no wear	Standard	6 to 7	-
SNCF [79]				
Tangent, tie and ballast	Corrugated	Standard	4 to 5[3]	-
			7 to 9[4]	-
			8 to 11[5]	
DB [39][6]				
Tangent welded, tie and ballast	Corrugated	Standard and	13 to 15	-
	Slightly corrugated	tuned-damper	6 to 8	-

[1]After grinding, small indentations were still found at welds and occasional small corrugation was found, possibly due to chattering of the rail grinder [76].

[2]This data was taken using a 2000 series car with a modified truck which included extra soft journal sleeves.

[3]Reduction after 20 passes with an abrasive brick grinding car.

[4]Reduction after 40 passes.

[5]Reduction after 60 passes.

[6]These tests compared noise levels on slightly corrugated and corrugated track to those on smooth ground rail. There were no measurements made on the same track before and after grinding.

Non-Acoustical Benefits of Rail Grinding

Measurements in SEPTA [3], BART [80] and TTC [25] indicate that rail grinding reduces ground vibration levels by 2 to 10 dB. In addition, since wheel/rail forces are being reduced (particularly in the case of rail corrugations), rail and wheel failures, and track and truck maintenance requirements may be reduced.

Grinding is needed after rail welding in order to realize the full benefit of the welding. Built up metal at rail welds can lead to secondary batter if not ground off. This occurs when the wheel "hops" over the weld and impacts the rail a short distance beyond, causing an indentation (secondary batter).

Problems Experienced with Rail Grinding

The NYCTA will not allow grinding of rail on open deck elevated structures because of the danger of falling metal from a broken grinding stone, and because of the potential fire hazard [24]. Yet the CTA uses their abrasive brick grinding train to grind on their elevated lines [15]. This possibility, or the use of other types of grinding methods [see 81],should be investigated.

PATCO's experience with rail grinding [15] includes occassional chipping or breaking of grinding stones by guard rails, occasional grass fires along surface track (they do not use water spray provisions usually accompanying grinding), and generation of grinding dust in subways which is removed during routine and annual cleanups. In spite of these problems, PATCO places no major restrictions on grinding.

Recommendations for Future Research

As with wheel truing, criteria must be developed for when and how to perform cost-effective rail grinding for noise control. The differences and relative merits of the two methods of grinding need to be better understood. Satisfactory methods for grinding rail (where corrugated) on open-deck elevated structures must be developed and/or demonstrated.

Finally, the principle reason rail grinding is performed on rapid transit systems appears to be for the control of rail corrugations. Systems which do not get corrugation generally do not grind rail. The causes of rail corrugation are still unknown. A careful, in-depth study into the cause or causes of rail corrugation on a specific transit system might yield useful guidelines for preventing corrugation on that system. This would have significant implications for cost savings on track and vehicle maintenance.

3.2 Rail Welding

Shop welding of rail sections together into long strings of continuous welded rail (CWR) or field welding of new or previously jointed rail has become common railroad practice and is being adopted by most transit authorities. In Europe, CWR is considered the most important development of the last 30 years in helping to reduce track costs [74]. It has been reported that CWR decreases average rolling resistance by up to 10% [24]. London Transport shop welds 18.3 m (60 ft) rails into 91.4 m (300 ft) lengths of CWR (the longest that can be moved conveniently through curves and interchanges on their system). The CWR is connected in the track using bolted joints. The mating surfaces of both the joint bars and the rails are machined for a smooth, tight fit, and high strength bolts are used. Regular maintenance is performed as needed. Only one welded joint failure has been reported in 30 years and the bolted joints are considered very convenient for track maintenance [74].

Most European transit systems are using field welds to connect lengths of shop welded rail. Thermite welding is widely used, but electric arc welding is also used. Good results are claimed with both methods. Different transit systems report expected weld failure rates of 0 to 30 per year [74]. Lately, flash butt welding has been used on some systems to convert track from bolted joints to CWR without removing and replacing rail. The NYCTA is purchasing an automated flash butt welding car for just this purpose.

Acoustical Performance

Although the primary reason for using welded rather than jointed rail is to reduce track maintenance costs, there is a clear acoustical benefit associated with its use. A summary of test data comparing noise levels on welded and on jointed track is presented in Table 3.2. Although the results vary, probably due to the condition of the joints being tested and the surface condition of both the jointed and welded rail, there is generally a noise reduction, typically between 4 and 10 dBA. It should be noted that new or well maintained joints, with no vertical mismatch between rail ends under load, do not produce significantly different noise levels from those of welded rail (see Section 3.3).

Some problems do exist with welded joints, most noteably field welded joints. Possible failures have been reported to be a concern [74]. As mentioned in Section 3.1, proper grinding of welded joints must be performed shortly after field welding in order to avoid secondary batter. In order to optimize the advantage of welded rail, it is important to keep the hardness and wear characteristics of the weld as close to as possible to those of the adjacent rail sections and the width of the weld should be kept to a minimum, otherwise differential wear at the joint

TABLE 3.2. SUMMARY OF NOISE REDUCTION DATA COMPARING WELDED RAIL WITH JOINTED RAIL.

Test Location and Track Type	Wheel Type and Condition	Noise Level Difference Between Jointed Track and Welded Track (dBA)	
		Wayside	Car Interior
SEPTA [3]			
Tangent, tie and ballast on elevated structure	New resilient (3 types)	2 to 6	3 to 6
	New and worn standard	2 to 6	2 to 4
	New ring-damped	3 to 5	2 to 4
Tangent, direct fixation in subway	New and worn standard and new resilient	-	1 to 3
CTA [77]			
Tangent, tie and ballast:			
worn jointed and worn welded	Trued standard, standard truck	4 to 6	3 to 5
worn jointed and worn welded	Trued standard, modified truck[1]	8 to 9	-
worn jointed and smooth welded	Trued standard, standard truck	7 to 11	1 to 5
worn jointed and smooth welded	Trued standard, modified truck[1]	11 to 14	-
NYCTA [68]			
Tangent, tie and ballast	Worn and trued standard	5 to 10	-
VARIOUS SYSTEMS [82]	Standard	4 to 10	-

[1]Tests performed with a modified 2000 series truck with extra soft journal sleeves.

will eventually lead to roughness and an increase in noise and wheel/rail loads.

Certain maintenance operations become more difficult with CWR. Replacing damaged sections of track requires that the rail be cut and a new section fitted and rewelded. However, this is offset by the reduced incidence of rail failure and rate of track degradation with CWR.

Suggestions for Future Work

Very little additional research is needed for evaluating the acoustical benefits of welded rail. The primary need for research in this area has to do with finding ways of using welded rail on older steel elevated structure where significant noise reduction is needed. There are mixed opinions concerning the feasibility of welding rail on these structures. The primary concern is for the possibility of track buckling. However, there have been successful tests on both the NYCTA [83] and the CTA [24] where jointed rail on open-deck steel elevated structures has been replaced by welded rail. This issue needs to be resolved and a way found to utilize welded rail on these structures.

3.3 Track Maintenance: Rail Joints, Ballast Cleaning, Track Geometry

Tests on newly laid jointed track have shown that a smooth, tightly bolted joint makes no more noise than a welded joint [20]. The Paris Metro [84] reports that their long welded rails are linked up by "expansion devices" (translation from French) which do not cause any additional noise *when well maintained.* When poorly maintained, these joints result in an increase of 12 dB in the peak A-weighted sound level for trains passing at 100 km/h (62 mph).

Tests made at SEPTA [15] showed a 1 to 2 dBA reduction in wayside noise when the joint bars were replaced in an effort to better align the rail ends. The rail was not changed nor was it ground for this test. Grinding the aligned joints resulted in another 1 to 3 dBA reduction for a total of 2 to 5 dBA.

When joints are not well maintained, the repeated wheel loads result in loose bolts; wear of mating surfaces which, in turn, allow differential movement of the rail ends; broken bolts in some cases; and battered rail ends. These, in turn, lead to more severe impacts from passing wheels which accelerate the degradation process and significantly increase noise levels (See Section 3.2).

On London Transport, 91.4 m (300 ft) lengths of CWR are connected in the field with bolted joints. Bolts are oiled and torqued regularly to specified tension. The joints are regapped as needed in the early spring of every year, to minimize the chance of buckling and reduce rail-end batter and noise. It is generally agreed that it pays to lubricate and tighten track bolts periodically, to avoid rapid and excessive track degradation when bolts are loose [74].

Other track maintenance procedures (aside from rail grinding) which can effect noise include ballast cleaning and track geometry maintenance. Ballast is an effective absorber of acoustic energy [85,86] and this property can be maintained by keeping the ballast from becoming contaminated. Maintaining good track geometry reduces the likelihood of flange impact and associated noise [1].

Suggestions for Future Work

Analytical and experimental work [2, 87] has indicated that vertical misalignment of the rail heads at joints (under load) is one of the principal sources of impact noise. The gap between adjoining rail ends has negligible effect, provided the vertical alignment is perfect. Furthermore, for a "step-up" joint (in which the wheel runs into the end of the rail it's moving onto), the impact noise continues to increase with speed, whereas for a "step-down" joint, the impact noise levels off above a critical speed. Consequently, rail joint maintenance practices should be studied to assure that they are compatible with avoiding step-up joints, and appropriate maintenance schedules or criteria should be developed to avoid significant increases of noise at joints. This work should further seek to define the relative costs and benefits of increased joint maintenance, use of special joints (see Sec. 3.4), and use of continuous welded rail.

3.4 Smooth Transition Rail Joints

Joints that smooth the transition of the wheel from one rail segment to the other have the potential of reducing impact noise. Recent track designs call for special trackwork to be installed with bonded joints, and insulated joints are bonded and bolted [74].

Insulated joints represent a special problem. The insulating material, because of its softer composition, wears rapidly, causing a depression in the continuity of the rail which leads to rail end batter and increased noise and vibration [71]. The elimination of noise from this source must be further studied. A diagonally cut (beveled) joint, at considerable increase in cost, may be one method of solving this problem. The pre-assembled glued insulated joint, used by some European properties should receive consideration [71].

Tests of 45° beveled joints have been performed on the RATP [84]. The results from these tests, not found in the published literature, should be obtained and reviewed. Other configurations of the joint mating surfaces should be investigated to determine if improved wheel load transfer between rails and reduced impact forces and noise can be obtained.

Some modern rail joint systems use "Huck" bolts which are claimed to resist normal loosening effects at joints better than traditional threaded fasteners [88,89]. This, and other fastening methods which might reduce joint maintenance requirements need to be further investigated.

3.5 Special Track Work Designs

Trains passing over special trackwork (switches, crossovers, etc.) create noise and vibration levels similar to and often exceeding those for travel on jointed rail. These result from the impacts at the mechanical joints and points, and the banging and squealing brought on by the numerous changes in direction generally associated with this type of trackwork. Some of these noises can be controlled by careful design and installation [71]. Bonded or welded joints should be used whenever possible. Abrupt direction changes may be reduced by the use of lower angle turnouts. Rail lubrication (see Section 3.6) should assist in eliminating the sharp squeals at short curved sections of track within special trackwork.

A moving point frog has been tested on the Paris Metro [84]; however, the first noise measurements were inconclusive because it was impossible to distinguish between the noise from the trains passing the point and passing the nearby joints. Information on this and any other type of low-noise design components for special trackwork should be collected and reviewed, primarily for making the information available to track designers and transit engineers.

3.6 Rail and Wheel Flange Lubrication

Lubrication of the gauge side of rails in curves, the side of the restraining (check) rail, or the wheel flange at curve locations is used widely in the U.S. and Europe primarily to reduce wheel and rail wear. In addition to manual lubrication, automatic track-side and vehicle-borne lubricators have been used, each of which can be designed to apply the lubricant to either the wheel flange or the rail [90-93]. In the U.S., trackside rail lubricators are most commonly used.

Many benefits of such lubrication have been claimed (and often demonstrated). These include prolonged life of old curve rail, extended life of new high rail in curves, reduction of realignment and regauging costs, reduction of wear on wheel flanges, and reduction of wheel squeal.

Acoustical Performance

Tests to determine the effectiveness of rail and flange lubrication in reducing wheel squeal have obtained varied results. Some of these test results are summarized in Table 3.3. In some tests it appears that lubrication of the gauge side of the high rail can reduce squeal; in others, it appears that lubrication of the gauge side of the rail (without lubricating the top of the rail) provides no reduction in squeal; still other tests indicate that only the top of the low rail need be lubricated to eliminate squeal. *In all tests where lubrication occurred on top of the rails, squeal was reduced or eliminated.* In most tests, not only

TABLE 3.3. SUMMARY OF RAIL LUBRICATION TEST RESULTS.

Test Location and Curve Radius	Description of Lubrication and Comments	Noise Reduction (dBA)
PATH [11] 35 m (115 ft) radius curve with check rail	a) Automatic lubrication of side of high rail only	4 to 16
	b) Small amount of grease applied to top of high rail at mid portion of curve (grease residue on side not removed)	16 to 18
	c) Grease applied to top of high rail over half of curve length (residue on side not removed)	12 to 27
MBTA [94] 61 m (200 ft) radius outer, 55 m (180 ft) inner track	Outer track (both rails) lubricated with spray of water and oil mixture. Inner track left dry. Reduced both frequency of occurrence and loudness of squeal.	Average of 9
Germany [95] 140 m (459 ft) radius, no check rail reported	a) Lubrication of gauge side of rail only	0
	b) Water spray on top of inner rail	Eliminated squeal
	c) Water spray on top of both rails	Same as for inner rail

the magnitude of squeal is affected but also the frequency and duration of the squeal. On the CTA [15], where rail lubrication has been used for over 40 years to reduce wear on curves, it has been found to have a marked but erratic noise reduction effect.

Problems Associated with Lubrication

In spite of the many beneficial aspects of rail and flange lubrication, numerous problems are associated with their use:

1. Excess grease from lubricators can "crawl up" or be splashed by the wheels to the top of the rail. If a train is braked and wheels lock at the greased section, wheels on the greased rail slide along the rail without any damage while the other wheel on the axle develops a flat as it slides over ungreased rail [90]. In addition, the loss of traction may become a safety problem.
2. Water-based spray on top of rails, which appears to reduce squeal and evaporates so that traction is quickly regained, has other negative side effects. It increases the wear on the wheel and rail [2, 96]; it freezes in winter (although this can be overcome by adding some anti-freeze to the water); and there is some concern that it can cause rot in wood ties.
3. Automatic lubricators appear to require considerable maintenance in order to keep them functioning properly.
4. The viscosity and effectiveness of lubricants may vary with temperature and age.

Suggestions for Future Work

The current methods for applying rail or flange lubrication are designed to reduce wear, not noise. They specifically attempt to avoid placing any "lubricant" on top of the rail. However, current understanding of wheel squeal attributes its generation to the friction-creep characteristics on the wheel/rail running surfaces [2]. An automatic lubricator designed to reduce wheel squeal needs to be developed and tested. This will necessitate experimentation with a number of different lubricants, in order to find one with optimum performance characteristics. The ideal lubricant, which may turn out to be solid (rather than liquid) would suppress squeal without contaminating non-involved surfaces. An ideal lubricant would be a material that evaporates or disperses rapidly after application but is nontoxic, non-flammable, non-corrosive, non-environmentally damaging, and does not freeze. It may not be easy to find such a material.

In addition to finding an optimum lubricant, the method and locations of application need to be investigated. For example, should both rails in a curve or just the inner or outer rail be lubricated? Should lubrication be applied only at the beginning of a curve or at several locations along its length? Because of the site-specific nature of this treatment and the potential reduced wheel and rail wear benefits, rail lubrication appears to provide a potentially cost-effective remedy for the problem of wheel squeal.

3.7 Resilient Rail

A resilient rail is one in which the rail head is isolated from the rail foot as shown in Fig. 3.1. The isolation not only reduces the vibration transmitted from the head of the rail to

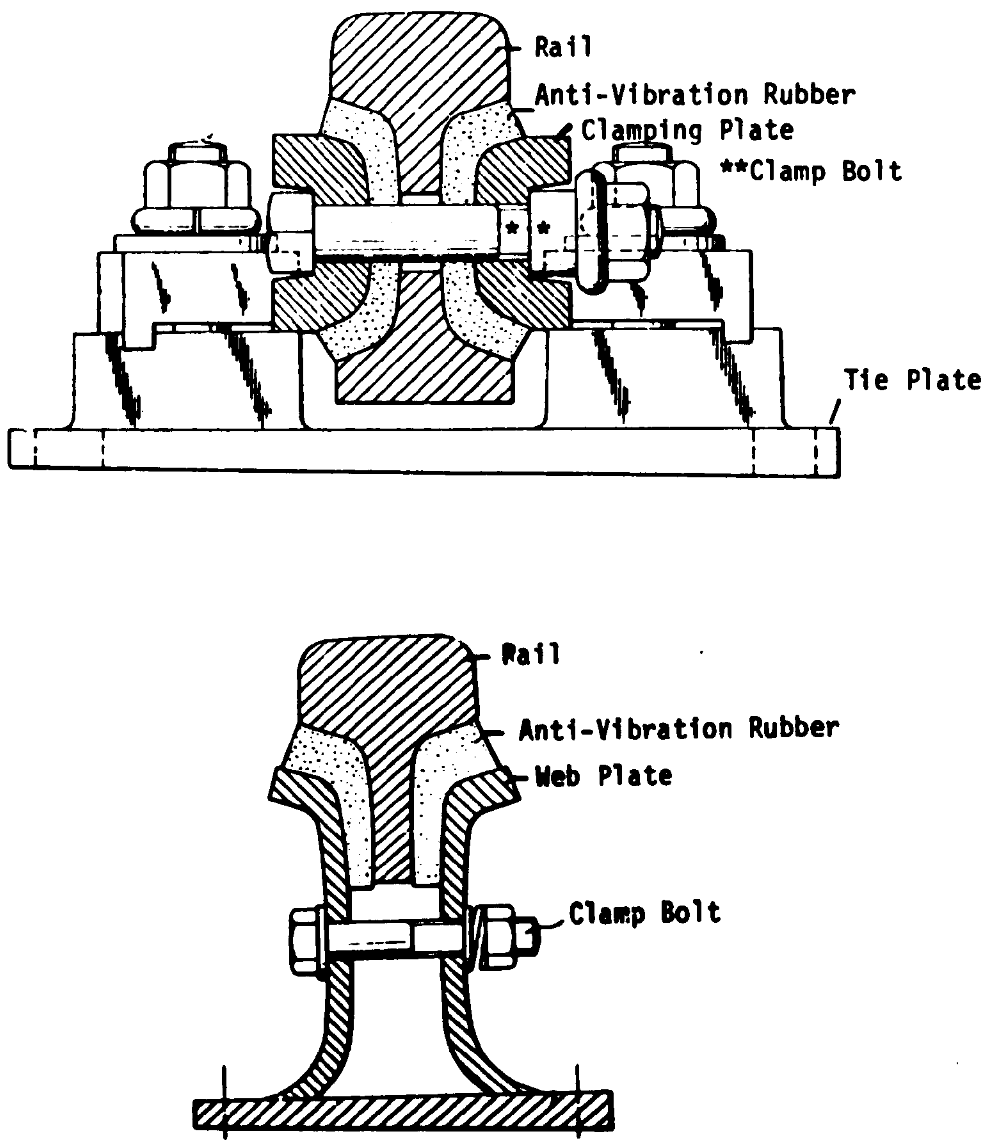


FIG. 3.1. TWO CONFIGURATIONS OF A RESILIENT RAIL [97].

the supporting structure, but also provides damping in the rail. Tests of this rail on an open-tie deck, steel girder bridge on the Japanese National Railways (JNR) [98], showed the rails to reduce wayside noise by 4 to 6 dBA. The peak vibration on the rail was reduced 6 dB directly under the wheels and 13 dB between the wheels. This means that the damping provided by the elastomer reduces the rail vibration at locations on the rail away from the wheels and thus the length of rail which radiates noise is decreased.

The durability, hardness, tensile strength, weatherability, compression set, heat buildup, stress relaxation, and spring constant of the resilient rail were evaluated in laboratory tests and found to be adequate [98]. This was followed by a series of field tests which showed the rail to be safe and effective. Designs are available for joints between resilient rail sections, transition rail from conventional to resilient rail, and provision for guard rail and trough for derailment potential [97].

Problems which may need to be addressed include the possible high initial costs of such a rail (no data available); rail for curves may have to be pre-bent at the mill; and the design would have to be demonstrated to be able to maintain gauge even if the resilient material fails [88].

This type of rail may have significant potential for noise reduction on steel elevated structures, although much the same result may be obtainable with appropriately designed resilient rail fasteners. Comparative tests of resilient rail and conventional rail with resilient fasteners on elevated structures should be performed. In addition, the resilient rails need to be evaluated in tunnel and on at-grade track to determine their potential for roar noise reduction. It is not expected that squeal noise would be greatly affected.

3.8 Rail Damping

A number of attempts have been made to reduce wheel/rail noise by applying damping material to either the bottom of the rail [17] or the sides of the rail [63,95,99-103]. Most of these attempts resulted in 0 to 2 dBA reduction in roar noise and erratic and unpredictable reduction in squeal noise.

Since most available data on wheel/rail noise seems to indicate that the wheel is at least as important in radiating noise as the rail and that resonances in the rail do not significantly affect the noise level, the main effect of rail damping would be to reduce the length of rail that radiates (see discussion in Section 3.7). Thus rail damping would not be expected to have a dramatic effect on overall wheel/rail noise levels unless radiation from the wheel were substantially reduced. Hence, it is not surprising that little or no reduction of roar noise has been obtained with damped rail.

Based on test results on curved track [101] it was concluded that damping on the rail will only lead to a reduction of airborne noise when rail vibration levels become large (> $\sim$3 g's) and the squeal frequencies in the noise spectra correspond to vibration peaks on the rail. In almost all cases, the damping of the rail will not be useful.

Based on the available data, this treatment does not appear to hold much promise for significantly improved performance. If it turns out that the rail is an important radiator of noise in particular situations, such as on an elevated structure, rail damping should then be investigated in conjunction with wheel treatments.

3.9 Resilient Rail Fasteners

Resilient fasteners are usually used to fasten rail directly to concrete (either decks on aerial structures, inverts in tunnels, or concrete ties). They have also occasionally been used with wood ties, although tie saver pads are now in common use. The effectiveness of resilient fasteners in reducing roar noise is unclear. Tests of different fasteners on an aerial structure during the design of BART [17] showed only slight differences in the wayside noise. Some fasteners resulted in increases in both rail vibration and wayside noise [by about 3 dBA]. It is not expected that resilient fasteners would have a significant effect on wheel squeal. Like resilient rails, resilient rail fasteners both isolate the rail vibration from the supporting structure, and increase the damping of the rail (if properly designed). On a steel elevated structure, such fasteners have resulted in 3 to 5 dBA reduction in wayside noise, due primarily to reduction of the vibration transmitted to the ties and steel girders [104].

It is not presently clear what optimum fastener characteristics should be for minimizing wheel/rail noise. Fastener stiffness, which affects the isolation of the rail vibration from supporting structures, is an important consideration for ground vibration in tunnels and structure-borne noise on elevated structures. For frequencies above about 300 Hz the fastener stiffness does not greatly affect the rail or wheel response. However, the damping provided by the fastener will effect the length of rail which radiates noise (refer back to Sections 3.7 and 3.8). Future research in this area should focus on determining optimum fastener characteristics for application to specific structures. This will require both analytical and experimental effort. Once optimum characteristics have been determined, designs for achieving these characteristics should be developed.

3.10 Specially Treaded Rails

There appears to be a potential for reducing roar noise, squeal noise, or both by means of covering or treating the rail running surface. As discussed in Section 2.4, roar noise can be reduced by increasing the contact area between the wheel and the rail. This can be accomplished by using a material with a lower elastic modulus than steel on the running surface of either the wheel (see Section 2.4) or the rail.

Tests of a polyurethane treaded rail on a 27.4 m (90 ft) test track at a car shop of the CTA [63] showed a 14 dB (no weighting given) reduction in rolling noise. In fact, the author commented that the actual reduction was more but could not be measured because the noise from the current collector equipment exceeded the wheel/rail noise. A section of the elastomer-covered rail was subsequently installed at the north end of the Skokie rapid transit line, but no further information was given. The actual configuration tested (shown in Fig. 3.2) used a 1.9 cm (3/4 in.) thick layer of polyurethane on an extra-wide rail head. The extra width was deemed necessary to keep the stresses in the polyurethane within acceptable limits. The configuration tested on the CTA required a fourth rail for ground return; other methods would need to be found. If the thickness of the polyurethane (or elastomer) were substantial (compared to the wheel flange height), the height of the wheel flange would have to be increased by a similar amount to ensure reliable guidance [63]. Tests on the

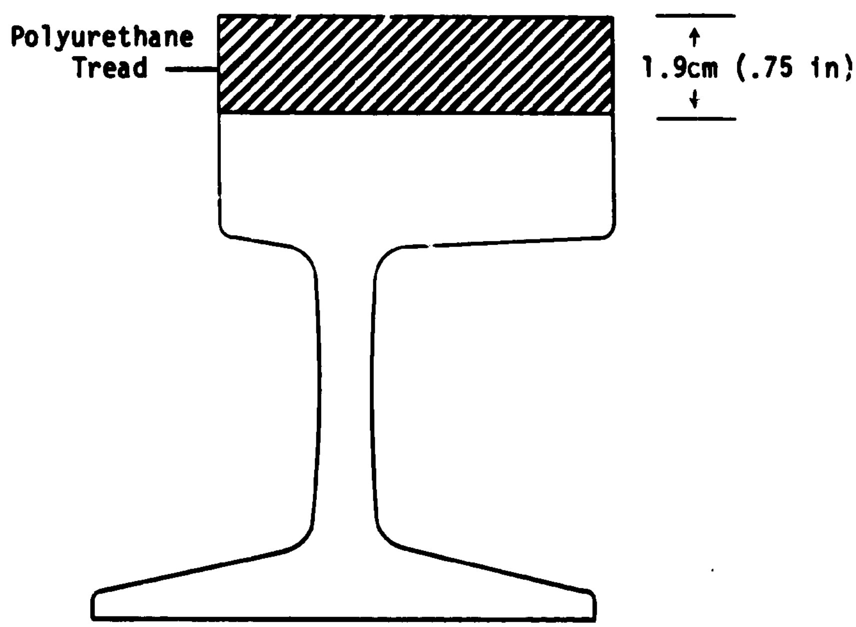


FIG. 3.2. POLYURETHANE-TREADED RAIL TESTED AT CTA [63].

adhesion characteristics of steel on the polyurethane under dry, wet, and dirt contaminated conditions indicate that it was equal to or better than steel on steel.

The control of wheel squeal at the wheel/rail interface requires the use of a material whose friction coefficient does not decrease with increased creep [2]. Nitinol, a nickel-titanium alloy (see Section 2.4) appears to be one such material. As discussed in Section 2.4, its wear characteristics in combination with steel under pure rolling are superior to those of steel on steel. It would have to be determined if this also held for combined rolling and sliding as occurs on curves. Should Nitinol or some other alloy be found to have suitable friction-creep, wear, electrical, thermal, corrosion, cost, and fatigue properties, treading the surface of a rail with this material might provide a site-specific, cost-effective solution to wheel squeal. However, at present, rail lubrication appears to be potentially far more cost-effective.

Electro-Thermite GMBH of Essen, Germany has patented some treatments for "hardfacing" of rail that are claimed to extend rail life, control rail corrugation, and reduce or eliminate squeal [105]. The process involves the welding and subsequent grinding of special very hard, low-friction steel strips onto the rail head. The manufacturer has indicated [106] that the patented rail is in use in Europe on curve track of some street car and subway systems. In tight street car curves, typically used in European cities, rail must be replaced every 4 to 6 months. The strip-treated rails are claimed [106] to extend the replacement period to about 2 years. For these situations, the treatment is justified on economic grounds alone. The elimination of wheel squeal is an extra benefit without an associated cost. This assumes that the use of the treatment does not result in a significant increase in wheel wear. The cost of applying the patented strips to the rail head is twice the cost of the rail.

Suggestions for Future Work

Polyurethane or similarly coated rail heads (or wheel treads) seem to have the greatest potential of available methods for reducing roar noise. Coating the rail heads has the advantage of providing site-specific relief if needed. However, many practical design problems including finding suitable materials, method of attachment, ground return, maintaining adequate flange guidance, etc. need to be investigated before in-service testing can be performed.

The claimed benefits of hardfacing make this treatment look extremely promising for control of wheel squeal. The current in-service application sites of this treatment should be visited and evaluated. Depending on the results of this evaluation, an in-service application, on a U.S. system which has particular problems with rail wear on curves and wheel squeal, might be recommended.

3.11 Trackbed Absorptive Treatment

Ballast is relatively effective in absorbing acoustical energy. Its sound absorption coefficient for layer thickness of 30 to 46 cm (12 to 18 in.) varies between 0.5 and 0.9 over the frequency range 250 to 4000 Hz [86]. The absorption coefficient decreases with decreasing ballast layer thickness, with values between 0.1 and 0.5 measured over the same frequency range for layers between 5 and 20 cm (2 and 8 in.) thick [107]. The absorption coefficients of the thicker ballast layers compare favorably with many materials designed to be acoustically absorptive.

Trackbed absorption is generally effective when concrete slab track is used, either in tunnels or on at-grade or aerial structures. Tests have shown that A-weighted levels near at-grade and elevated slab tracks can be reduced by 0 to 4.5 dB [typically

2 dBA] by placing ballast or other absorptive material between and next to the rails [99, 107-109]. In non-acoustically treated tunnels, the reduction can amount to significantly more, 5-10 dBA [107,110]. When ballasted track is already in use, additional trackbed absorption is difficult to obtain and will be unlikely to result in additional noise reduction.

Acoustical material used for trackbed absorption must be weatherproof (if used outdoors), fireproof, cleanable, and resistant to airflow under trains. Cleanability is important because dirt, oil and other debris may contaminate the pores of the material and reduce their effectiveness.

This treatment, although particularly effective in otherwise untreated tunnels, does not appear to be worth pursuing with further research. Its performance in tunnels can be fairly well predicted (its main effect being a net increase in the total absorption in the tunnel [110]). Also, in tunnels it may be more practical to add absorption to the walls, ceilings, or under platform areas in stations. For at-grade or elevated slab track, only small reductions are achieved and further research will not likely improve the performance.

3.12 Acoustical Barriers

Much research and testing have been done on wayside barriers for rail systems [108, 111-113]. Modern rapid transit systems such as MARTA are using barriers along the decks of aerial structures. Most of these barriers are between 1 and 2 m (3.3 and 6.6 ft) high and reduce wayside noise by 5 to 15 dBA. An absorptive barrier (one lined on the side facing the train with acoustically absorptive material) is typically 3 to 4 dBA more effective than a reflective barrier. This is primarily due to the elimination of reflections between the barrier and the side of the car. The theory and practical design for full height wayside barriers is well documented. Furthermore, this treatment is not strictly a wheel/rail noise source treatment, but rather a treatment affecting the noise propagation path. (This is also true for trackbed absorption and vehicle skirts).

One concept which still requires further research is the "rail barrier". Current theory [2] attributes a significant portion of the overall wheel/rail noise radiation to the rail. The available data have not confirmed this; however, recent measurements at the Transportation Test Center in Pueblo, Colorado [114] may clarify the contribution from the wheel and the rail. It may turn out that although the rail radiates a significant portion of the total noise, its radiation is more attenuated than that of the wheels due to its proximity to the ground. If the rail radiation is significant, a low barrier placed as near to the rail as possible could attenuate the wayside noise from this source. If used in conjunction with wheel treatment or vehicle skirts, significant reductions in wayside noise may be achievable.

The major potential problems with rail barriers are clearance and snow removal. Additionally, such barriers (or, for that matter, full-sized wayside barriers) can result in an increase in car interior noise if airborne wheel/rail noise is a significant contributor to the in-car noise. Absorptive barriers minimize this effect.

One trackbed construction technique which may act as a rail barrier, which "covers" the sides of the rail but not the top, is that used by many street car systems. The grooved girder rail used in this case is normally spiked to wood ties on ballast or cast in concrete. The pavement (which may be bituminous or concrete) is laid on both sides of the rail, up to the top of the rail. These systems are generally no quieter than systems using uncovered "T" - rails [115], however. In fact, more important may be the noise attenuation due to the absorption by the ground surface.

3.13 Restraining Rails on Curves

Very high lateral forces are generated at the wheel-rail interface on curves. A restraining (or check) rail relieves the leading outside wheel flange of this pressure and transfers it to the back of the inner leading wheel flange, reducing wear and the tendency to derail. With the use of a restraining rail on a curve, the trailing wheels experience far less side thrust on curves. The use of restraining rails shifts the requirement for lubrication primarily to the inside or non-gauge side of the flange, which has the advantage of removing the lubrication point as far away as possible from the tread.

The effect of restraining rails on wheel squeal is not known. Some data taken on a roller rig [116] indicate that flange contact between the wheel and the rail actually reduces wheel squeal. Since the restraining rail reduces this flange contact for the leading outside wheel, but introduces another type of flange contact on the back of the inner wheel, the net effect on squeal is not at all clear. The test data summarized in Section 3.6 (Rail Lubrication) was conflicting as to whether the inner or outer wheels were responsible for the squeal. This conflict may have arisen because one set of tests were performed on track with restraining rail while the outer test track had no restraining rail. Because many systems require the use of restraining rails, it is important to understand how they affect wheel squeal. It may be possible (although at this point it is pure conjecture) to reduce wheel squeal by appropriate positioning and choice of geometry of the restraining rail. This is one area where experimental work is needed.

4. VEHICLE AND TRUCK TREATMENTS

4.1 Wheel Slip/Slide Prevention Systems

This section discusses two types of systems used to prevent wheel slide which results in flat spots on the tread. Wheel flats on transit car wheel treads can increase noise levels by up to 10 dB. If the flats are very large, the car may have to be removed from service. Two types of slip/slide detection systems are discussed below. One system, known as a traction fault system, is used on a retrofit basis by the NYCTA. The other is known as a slip/slide or spin/slide detection system, and it is used on most newer transit cars.

Traction Fault Detectors

This system consists of a feedback controller which senses dynamic braking forces on the individual wheels by sensing both motor currents and brake cylinder pressure, and can relieve these loads to minimize wheel slide [68]. The NYCTA has reported a 50 percent reduction in the number of flat spots on 30 cars that were fitted with this system. Records were kept for a period of two years.

There are two ways of looking at the potential benefits of this system. First, by reducing the frequency of wheel flat generation , this detection system would allow for less frequent wheel truing. Alternatively, if the truing machines are not used to capacity, then the use of traction fault detectors should allow the transit system to adopt a more stringent criterion of allowable flat spot length. In the latter case, the average noise level of the system would decrease.

The side benefits of a traction fault detection system are reduced truck and track structure vibration levels, and longer wheel life because of less frequent truing.

Slip/Slide Detectors

This type of system is used on most newer transit cars. It is sometimes referred to as a spin/slide detector. In addition to minimizing wheel slide during braking, this system also minimizes wheel spin during acceleration. A slip/slide system consists of speed detectors on each axle, a signal detection and conditioning system, and a feedback path to the main brake control signal [117]. During braking, this system compares the rpm of each axle, and if one axle is found to be rotating at a different speed, the brakes on the corresponding truck are released and then reapplied. If the wheel continues to slide, the cycle is repeated.

The NYCTA has also reported a 50 percent reduction in the number of wheel flat occurrences on the cars equipped with this device.

The side benefits of slip/slide detectors are the same as those for the traction fault detectors listed above.

Even with slip/slide systems, wheel flats are still a major problem on most transit systems. A multiphase research program should be conducted to address this problem. The first phase would consist of making measurements to determine the extent of the wheel flat problem. This could be done by making rail or invert vibration measurements in a manner similar to TTC wheel roughness detector (see Section 5.1). The instrumentation could be temporary rather than be hard wired as it is in Toronto. The objective of this phase of the study would be to determine if wheel flats are a major contributor to the system-wide average noise level. It may be that on some systems wheel flat noise is a more important source than either rail joint noise or roar noise. If this is the case, then clearly the wheel flat problem must be solved before any significant overall noise reduction can be achieved.

The second phase of the research effort would be to understand as well as possible the mechanics of how flats occur. This study would include the response of present slip/slide systems and may be both theoretical and experimental in nature.

A third phase of the study would be a cost/benefit analysis comparing the cost of more frequent wheel truings versus the cost of better slip/slide systems. This analysis would include the value of increased wheel life if the number of wheel truings can be reduced.

Finally, if wheel flat noise is found to be a major source, and if the economics favor a more sophisticated detector (as opposed to more frequent truing), then a better slip/slide detector would be developed and tested.

4.2 Steerable Trucks

A steerable or radial truck as it is sometimes called, is a truck in which the axles are either cross linked or linked to the car body in such a manner that both axles can point toward the center of a curve. When the axles are cross linked the steering forces come from the wheel/rail interaction, and the flexibility between axles is usually achieved by shear deformation of the primary suspension. This is called a self-steered truck. When the axles are linked to the car body, which then provides the steering forces, the truck is considered to be force-steered. In that case, the flexibility can come from either shear deformation of the primary suspension or a pivot at the center of the truck. A steerable truck can prevent flange contact on curves and minimize the angle of attack between the wheel and the rail; thus, these trucks show great promise for preventing or reducing wheel squeal.

Steerable trucks for freight cars have been used for several years in other countries, and they are now becoming commercially available in the United States [118]. Railroads are interested in radial trucks because they reduce flange and rail wear, especially if the railroads have a large amount of curved tracks. Radial trucks for rapid transit systems are presently in an experimental stage of development. A few North American manufacturers are preparing prototype trucks for testing on various transit systems [119]. Because curves with smaller radii are more common on transit systems than on railroads, steerable trucks for transit systems may require more complicated linkage systems than railroad freight trucks, although one type of force-steered truck concept being developed [120] seems to hold significant potential for simplifying these linkages. A change in the wheel tread profile may also be necessary.

Only very sketchy data are presently available on the ability of steerable trucks to prevent or minimize squeal. One manufacturer claims that their design is able to negotiate curves with radii as small as 20 m (65 ft) without wheel squeal [120].

The major non-acoustical benefits of radial trucks are reduced wheel and rail wear, as well as less fuel consumption due to reduced rolling resistance on curves. The savings in reduced wear may more than offset the higher initial cost of these trucks. Another potential benefit of radial trucks is better ride quality, particularly if hunting is a present problem.

At this time, there are no major problems with switching to the use of radial trucks. The new design being developed by the Urban Transportation Development Corporation (UTDC) [120] appears to be adaptable for converting existing trucks. A change to the use of radial trucks would most likely have to be justified economically by the increased life of wheels and rail, with noise reduction a concomittant benefit.

Since prototype steerable trucks for rapid transit cars are presently under development, the major effort at this time should be in the direction of seeing that they are properly and thoroughly tested. The long term acceptance of this design will have to be justified on the basis of a reduced life cycle cost.

4.3 Reduced Truck Axle Spacing

According to the model developed for squeal nosie by Rudd [2], wheel squeal can occur for a square (i.e., parallel axle) truck when the wheel base of the truck is greater than approximately 1/100 times the radius of the curve. Thus, making the wheel base shorter should allow the truck to negotiate smaller radii curves without squeal. A truck with a 5 ft wheel base could be used, for example, on a system with curves of radii not less than 500 ft, to eliminate squeal.

This concept could only be considered for a new design. Reducing the axle spacing of a truck may be difficult because of space limitation for traction motors and other equipment. A shorter wheel base could also lower the hunting speed of the truck, and this in turn could lead to increased wheel and rail wear and perhaps even increased rolling noise.

4.4 Vehicle Skirts

Vehicle skirts are acoustical barriers attached to the sides of a transit car and extending down as far as possible to block the direct line-of-sight from the truck, wheels, and undercar equipment to the wayside. The effectiveness of vehicle skirts at reducing wheel/rail noise is limited because of restrictions on how far down the edge of the skirt can extend. In Europe, vehicle clearance specifications restrict the lowest point on the running gear (other than the wheel sets) to be located above the top edge of the rails (in all operational conditions) by the amount the spring system travels plus the wheel tire height; i.e., about 13 cm (5 in.) [121]. The vehicle skirts are also constrained by the clearance to the third rail, particularly critical for long cars on curves. To meet these clearances, some of the wheel and all of the rail will remain exposed, so that only limited noise reduction can be achieved.

Tests of vehicle skirts have been performed in Europe [99,122] and Japan [123], and scale model tests have been performed in the U.S. [124]. The wayside noise reduction obtained in the field tests ranged from 0 to 3 dBA. These were for both absorptive and reflecting skirts (see discussion on absorptive and reflecting barriers in Section 3.12). On a transit vehicle where propulsion noise may become more important than wheel/rail noise at high speeds, absorptive vehicle skirts may provide reductions up to 10 dBA [124]. In addition it may be possible to utilize vehicle skirts to aid in the heat management problem by suitably directing air flow under the car body.

Problems envisioned with vehicle skirts include: possible interference with inspection and maintenance of wheels and under-car equipment; absorptive treatment, which may be needed to avoid an increase in in-car noise, and may absorb contaminants such as oil or grease and become a fire hazard; and unless properly designed, heat buildup under the cars.

Experiments performed on a rolling rig [125] indicate that about half of the noise radiated from wheels, axles, and truck frame come from the lower half of the wheel, including the wheel/rail interface. This implies that to get the optimum benefit from vehicle skirts, they should be used in conjunction with wayside barriers. The combination of skirts and barriers were evaluated in a scale model experiment where the source of noise being modeled was the vehicle propulsion system [124]. For this particular source of noise, the vehicle skirt in combination with the barrier resulted in about 5 to 10 dBA additional attenuation over that provided by the barrier alone (except in the case of a high absorptive barrier).

Future work in this area should look into the design of vehicle skirts which address the problem of heat management under transit cars and are intended for use mainly on transit vehicles whose propulsion systems are at least as important a noise source as wheel/rail noise. For wheel/rail noise control, this technique should be tested in conjunction with a wayside noise barrier.

4.5 Undercar Absorption

Most of the discussion contained in the section on trackbed absorption (Section 3.11) also applies to undercar absorption. The difference in this case is that the absorptive material is placed under the car rather than on the track. It is primarily expected to reduce noise levels in untreated subway tunnels and in-car noise on aerial structures or other track where ballast is not used. Little or no reduction in wheel/rail noise radiated to the wayside is expected. Scale model tests on the effect of undercar absorption on wayside noise from *propulsion equipment,* for a vehicle on a concrete deck aerial structure, showed a possible 5 dBA reduction [124].

This treatment might be suitable for transit systems in which most of the track is in non-ballasted tunnel. It might also be necessary for use with vehicle skirts (see Section 4.4) to avoid any increase in car interior noise.

4.6 Braking Systems

The type of braking system used on a rail vehicle can greatly affect the surface roughness of the wheels and hence the noise. Tests in Germany [126], Switzerland [127], the Netherlands [128], and Japan [129] have clearly demonstrated that composition tread brakes or disc brakes result in significantly reduced wheel roughness, wear, and noise compared to cast iron tread brakes. Results of acoustic testing are summarized in Table 4.1. Note that the wheels braked with cast iron tread brakes are about 10 dBA noisier than disc-braked wheels, and 5 to 7 dBA noisier than composition tread-braked wheels. The high phosphorous content cast iron shoes performed almost as well as the composition shoes.

TABLE 4.1. SUMMARY OF TEST RESULTS FOR VARIOUS BRAKE SYSTEMS.

Test Location	"Noisier" Brake	"Quieter" Brake	Noise Reduction* (dBA)	Comments
German Federal Railways (DB) [126]	Cast iron, tread	Disc brake	9	The noise increase with cast iron shoes occurred within 16 brakings from 140 km/h (87 mph). All cast iron braked wheels developed corrugation of 2 to 4 cm (.8 to 1.6 in.) wavelength, 30 μm (1.2 mils) deep.
Swiss Railways [27]	"Samson" shoe (cast iron with high phosphorous content)	Disc brake	5	"Samson" blocks resulted in smearing of roughness on wheel. Average noise level difference after 10 to 25 brakings.
	Cast iron shoe	Disc brake	10	After first braking, wheels with cast iron brakes showed typical spotting which increased with number of brakings.
Japanese National Railways (JNR) [129]	Cast iron shoes	Composition shoes	5 to 7	The cast iron shoes caused corrugation on the wheel treads: 13 to 26 mm (.5 to 1 in.) wavelength and 10 to 20 μm (.4 to .8 mils) deep.

*Measured 7 to 7.5 m (23 to 25 ft) from the track, 1.5 m (5 ft) above the rail surface.

Measurements on the NYCTA, the only rapid transit system in the U.S. using cast iron shoes, showed a rapid increase in wheel/rail noise with time: 5 dBA after 3 months, and 8 dBA after 6 months [68]. This is most likely due to the use of the cast iron brake shoes.

Most of the tests performed with cast iron brake shoes reported the formation of corrugations or "spotting" around the wheel. These corrugations had wave lengths varying from about 2 to 4 cm (0.8 to 1.6 in.) and depths on the order of 10 to 30 μm (0.4 to 1.2 mils). The high phosporous cast iron shoes tended to smear the initially formed roughness while the disc braked wheels showed no such corrugation.

One recognized advantage of cast iron shoes is their ability to assure good electrical contact for train detection by track circuits [130]. Composition tread brakes or disc brakes are not as good. However, this is not a severe problem since very few transit systems run single cars and it is unlikely that all four axles on each car in a train would simultaneously develop sufficient electrical resistance at the wheel surface to lose detectability.

Some reluctance to conversion to composition shoes are based on their reputed effects on lowering adhesion, poor wet stopping capability, thermal crack inducements on wheels, and possible environmental problems because of their asbestos content. Regarding the latter concern, although the original composition block might contain up to 60% asbestos, the wear debris contains only 2% asbestos, the remainder changed to forsterite by the heat of braking [130]. The improved wet stopping performance with cast iron shoes may be due in part to the higher wheel roughness caused by their use [130].

Many existing cars using cast iron tread brakes have been converted to composition shoes by eliminating clasp brake rigging (rigging which applies braking from two shoes per wheel) [131]. This conversion usually involves converting the brake cylinder relay valve to provide 40 to 60% of the control pressure at all times. In other instances, the effective brake cylinder diameter has been suitably reduced by use of a bushing kit. This permits using normal brake cylinder pressure [131].

For demanding continuous rating duties, such as in rapid transit applications, disc braking appears to be the optimum choice.

Future work in this area should concentrate on an in-service evaluation of cast iron vs. composition tread brakes. The logical location for such a test would be the NYCTA since all of their older (pre R-44) transit cars use cast iron brake shoes. Such an in-service test would involve the conversion of several transit cars to composition tread brakes, and the running of a test train in the system composed of both cast iron and composition tread braked cars; or running several test trains of each type of braked cars. Periodic measurements of wheel roughness, wayside noise, in-car noise, and track vibrations should be made. In addition, wheel temperatures might be monitored and regular wheel inspections performed. This type of testing would not only point out the relative merits of each brake shoe, but also the costs and practical problems associated with conversion to composition shoes.

4.7 Soft Primary Suspension

This treatment probably does not reduce wheel/rail noise directly, but it can reduce the noise radiated from the trucks or car body by reducing the vibration levels of these elements. The ability of this treatment to reduce the overall noise depends on how soft the primary suspension is in the first place. In a test case, measurements of some CTA 2000 series cars with modified softer suspensions showed reduced noise levels at both the wayside and inside the car [77]. These results are summarized in the table below, but it should be kept in mind that the original primary suspension was very stiff, 8.8×10^7 N/m (500,000 lb/in.), and that it was reduced by a factor of 30. The extra soft primary used in these tests was judged not to be practical for fleet retrofit.

Table 4.2 Noise Reduction of A-Weighted Sound Level Due to Reduction in Primary Suspension Stiffness on CTA 2000 Series Cars

	Jointed Rail	Corrugated Welded Rail	Smooth Welded Rail
Wayside Tie/Ballast	0 dB	2 dB	3 dB
Interior Tie/Ballast	3 dB	6 dB	6-[illegible] dB

The major side benefit, if not the purpose, of reducing the stiffness of a stiff primary suspension is a reduction in ground vibration. Reduction of the primary suspension stiffness should also cause lower dynamic loads on the truck components and the track structure. The latter benefits should reduce maintenance costs, but it would be very difficult to put an actual monetary value on these reduced costs.

It is clear from the summary of results presented above, as well as other studies [132], that stiff primary suspension stiffness can cause increased noise and ground vibration levels. The rapid transit industry should investigate this situation. If it is determined that sufficient data are already available, these should be used to set standards on acceptable suspension stiffness. If presently available data are not sufficient, then a more thorough investigation should be initiated.

4.8 Vehicle Speed

It is clear from a number of studies that wheel/rail noise is very dependent on the speed of the vehicle [3,77,82]. The speed dependence is usually in the range from 20 to 40 times the log to the base ten of the speed, with 30 $\log_{10}$ (speed) being the most commonly quoted value. This implies that halving the speed will produce a noise reduction of 6 to 12 dB.

Reducing speed is in conflict with the desire of transit systems to provide fast travel times. Nevertheless, reducing speed may work in some limited circumstances. The New York City Transit Authority, for example, has a speed limit for the purpose of noise control along one section of an elevated structure alignment that passes a nearby school.

5. MONITORING SYSTEMS

5.1 Wheel Flat Detection Systems

One widely recognized source of increased wheel/rail noise is flat spots on the treads of the wheels. These flat spots are generally attributed to locking the wheels during braking and the subsequent sliding of the wheel along the rail. The sliding wears a depression in the tread similar to the profile of the head of the rail. Frequently (and particularly in the case of large flats), there is also a flow of metal to the rear of the contact area, resulting in a high spot on the tread just beyond the depression. With further use of the wheel, small flats tend to wear out, and the metal flow associated with large flats tends to flatten.

Noise levels from wheels with large flats can be about 10 dB louder than levels from wheels with smooth treads (see Sec. 2.6). Ground vibrations and impulsive forces to the rolling stock are also greatly influenced by wheel flats.

Wheel flat detection systems are not a quieting technique per se, but rather they are an aid in determining which wheels are in need of maintenance. The maintenance usually consists of turning, milling or grinding the surface of the tread to obtain a new surface free of imperfections.

Wheel flats can be detected by a number of methods. Visual inspection of the wheel treads is one technique that is still used by several rapid transit systems in the United States. It is, however, hard to see the entire surface of the tread without indexing the rotation of the wheels a few times. The reports on wheel flat detection systems prepared by the Office for Research and Experiments (ORE) of the International Union of Railways (UIC) states that some railroads used to position a man alongside a train to listen for flat wheels as they passed [133].

This system was later replaced by a directional microphone consisting of a free field microphone placed at the focus of a parabolic reflector. Listening, either by ear or with the aid of microphones, is more difficult in the case of rapid transit systems where there are other noise sources on the car, and where it may be desirable to detect flats in reverberant tunnels or on noisy elevated structures. In many cases it is also important to obtain more quantitative information such as the length of the flat or how much rail vibration it causes.

The discussion which follows presents a review of some wheel flat detection systems that have been used in the past or that are presently in use either in Europe or North America.

The ORE issued four reports on wheel flat detection systems in the period from October 1968 to October 1975 [133]. The purpose of the ORE study was to develop or test devices that automatically determine the length or depths of flats. The first report discusses the importance of detecting flats in terms of the increased rail stress which they cause, and discusses briefly detection by means of microphones and either accelerometers or strain gauges mounted on the rail. The second ORE report discusses a field comparison of four detection devices – two strain gauge sensing devices, a device consisting of 3 accelerometers and 2 axle detectors, and a device that electrically senses the momentary break between the wheels and the rail as the flat spot passes. The last two reports describe two versions of the electrical device in more detail.

The strain gauge device was not pursued by the ORE because of its dependence on maintaining a consistent trackbed and the difficulty of correlating the signal to the length of the flat. The accelerometer system was also not pursued because the signal did not correlate well with the length of the flat - it was hard to distinguish a long flat from a series of short flats.

The electrical devices tested by ORE were based on the principle that above a critical speed of approximately 40 km/hr (24 mph) the wheel and rail momentarily loose contact as the flat passes the rail. This is because the wheel cannot accelerate downward at a high enough rate to stay in contact with the rail. The duration of the separation is roughly proportional to the length of the flat and the weight of the vehicle, but independent of speed provided that the critical speed is exceeded. In this system a short segment of one rail is electrified with a sine signal at 100 kHz or higher. During smooth rolling with no flats the rails are electrically shorted and their relative electrical potential is small. When the electrical circuit is broken by the presence of a flat, the potential difference between the rails rises to a measurable finite value (Fig. 5.1). The duration of the increase in potential is proportional to the length of the flat.

The wheel flat detection system used by the Toronto Transit Commission (TTC) is based on measuring the acceleration of the subway tunnel invert [134]. TTC is not so much interested in the length of the flat as it is in the effect of the flat - namely, the tunnel vibration and subsequent ground vibration. The output from the accelerometer signal is transmitted via telephone lines to the Davisville Carhouse where it is displayed on a graphic level recorder. Figure 5.2 shows an example of a typical record. Wheels, or at least trucks, that cause high vibration levels are clearly identified. If the major purpose of the detection system is to detect wheels that cause high rail, tunnel, or ground vibrations, then the TTC system may be more appropriate than the systems tested by ORE.

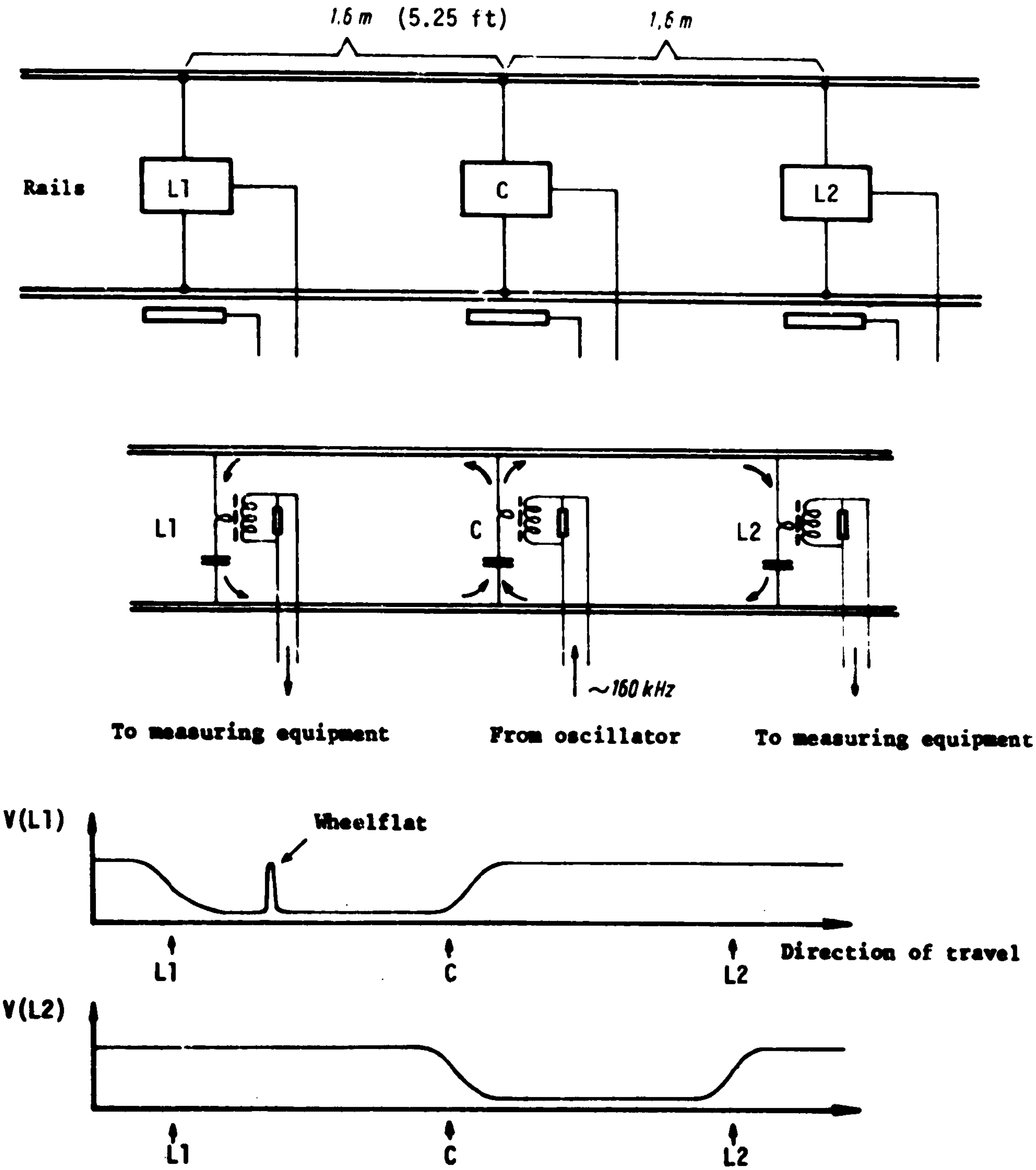
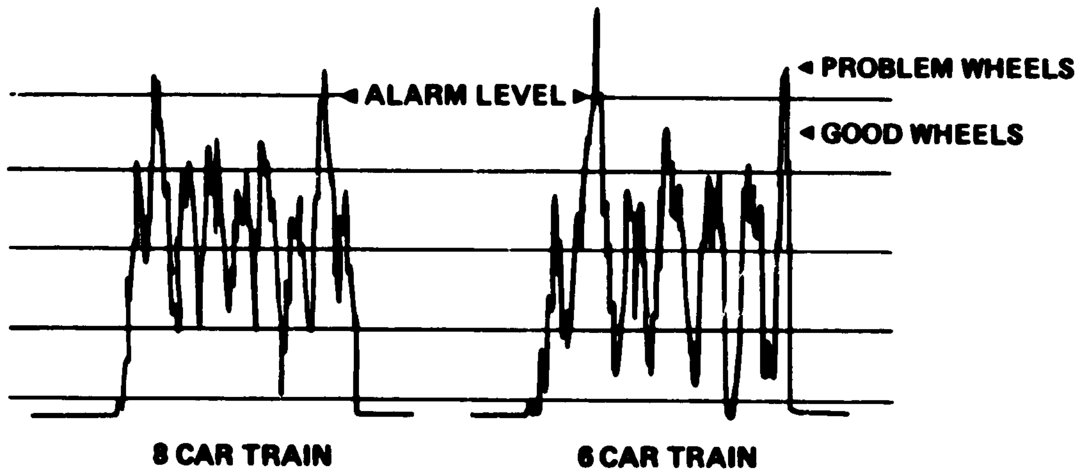
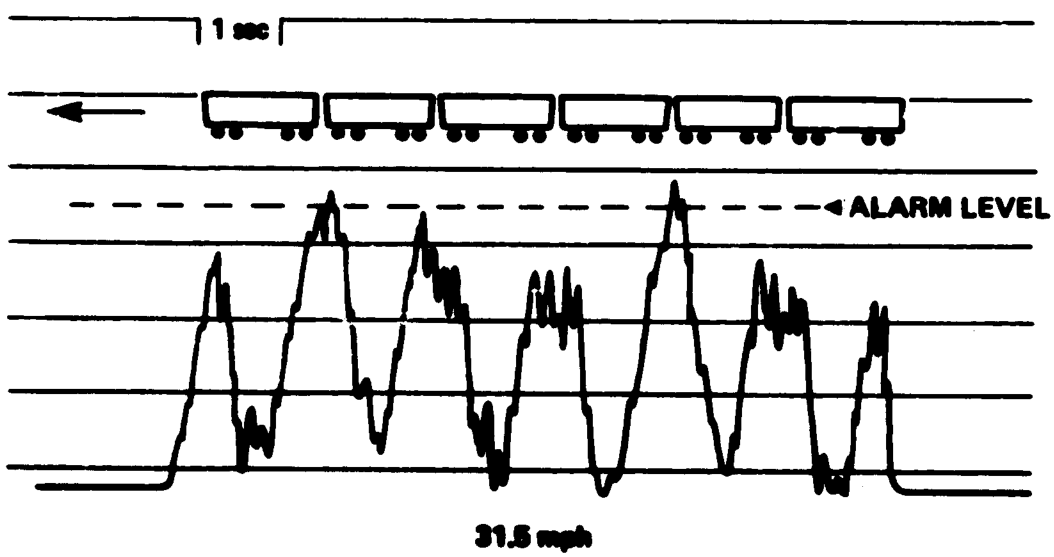


FIG. 5.1. CONFIGURATION AND OUTPUT SIGNAL OF SWEDISH WHEEL FLAT DETECTION SYSTEM EVALUATED BY ORE [Adapted from 133]



a. Typical Level Recorder Traces.



b. Expanded Level Recorder Trace.

FIG. 5.2. GRAPHIC LEVEL RECORDER TRACE OF ACCELEROMETER SIGNAL ON TTC WHEEL FLAT DETECTION SYSTEM [134].

A study of a wheel flat detection system based on an accelerometer mounted on the rail was conducted by D. Dousis for his Master of Science thesis [135]. This study was primarily concerned with computer signal processing. The data based on rolling a single wheel set on a short stretch of rail were inconclusive.

A number of wheel flat detection systems, including some of those mentioned above, are reviewed in a DOT/TSC memorandum by G. Economou [136]. This memorandum also describes a commercially available rail accelerometer system produced by IRD Mechanalysis Inc. This system is designed for use in hump yards to identify cars that should be set out for visual inspection. In addition, the TSC memorandum describes a conceptional system based on measuring the displacement of the flange relative to the head of the rail. The flange would ride against a floating rail, and at less than critical speed the floating rail would be depressed as the wheel rolled onto the flat spot.

If the purpose of the detection system is to identify wheels that cause high rail vibration levels (and subsequently high noise levels, tunnel vibration levels and ground vibration levels), then a system like that used by TTC seems most appropriate. This system may not be able to distinguish the exact wheel which is causing the problem, but it will identify a truck with a problem wheelset. Further signal conditioning or extra accelerometers may be needed if it is necessary to identify the exact wheel responsible for the high vibration level. The system should be tied into a train identification system, or a time signal, such that trains with bad wheels, can be identified at a latter time.

5.2 Rail Roughness Detection System

This is a conceptual device that would be mounted on a car, perhaps a special car, for locating sections of track that cause high wheel/rail noise levels. Experimental devices to measure rail roughness have been used by Remington and others, both as a part of the present study and as part of a previous study [2]. However, these experimental devices were not mounted on a car, and they could only be used over a short section of a single rail.

The rail roughness detector envisioned here would be mounted on a transit car or track geometry car, and be capable of measuring rail roughness at modest operating speeds of perhaps 5 to 10 mph. As in the case of the other monitoring systems, this device is not a quieting technique per se. Rather, the monitor is an aid in determining those sections of track that require maintenance, and it is the maintenance that will reduce the noise levels.

One possible design of this device would consist of accelerometers mounted on the axle journal boxes of a transit car truck. Alternatively extra small probe wheels that ride on the head of the rail could be used to measure the rail roughness. The advantage of this latter configuration is that it would minimize vibrations caused by axle bearings or gears.

Although the conceptual rail roughness detection device described above does not exist at this time, it is very similar to some track geometry measurement systems that are in operation [137]. The major difference between this conceptual device and the track geometry systems is their operational frequency range and consequently the range of wavelengths on the rail that they are capable of measuring. Track geometry systems typically measure wavelengths in the range from 1.2 to 22.9 m (4 to 75 ft), whereas a rail roughness detection device should be capable of measuring wavelengths in the range from 0.64 cm (0.25 in.) up to approximately 0.3 m (1 ft).

The major side benefit of locating sections of track with rough rail is that these sections of track are probably also responsible for high structural loads on both the rolling stock and the trackbed. Elimination of the rough track would reduce these loads.

Developing a rail roughness monitor system to operate off a transit car or track geometry car at modest speeds may be difficult. The devices must be able to withstand the shock loads of rail discontinuities such as joints, switch points, and frogs. It may not be advisable to try to develop such a detector for a transit system with jointed rail, because the rail joints would clearly dominate, if not overload, the response.

5.3 Vehicle Noise Monitor

A vehicle noise monitor is a conceptual system that would consist of a microphone mounted in a tunnel or outdoors near the track. The signal would probably be transmitted back to a central location in a manner similar to the TTC wheel roughness detector described in Section 5.1. This system would identify unusually noisy cars as they past the monitoring microphone regardless of the source of the noise. In this respect, it would have trouble distinguishing between wheel flats or a faulty air conditioner fan. If, however, this system were used in conjunction with a wheel roughness detector, then one would at least be able to know if the noise source were something other than wheel roughness.

The major potential side benefits of a vehicle noise monitor system is its ability to pinpoint exceptionally noisy on-board equipment. If the equipment is exceptionally noisy, then it is probably also in need of maintenance.

The actual implementation of a vehicle noise monitor system would be fairly straightforward. It could be designed, for example, in a manner similar to the TTC wheel roughness monitor except that the accelerometer would be replaced with a microphone. A rugged microphone transducer such as a piezoelectric type would probably be required.

A fair amount of normal variation in noise levels may be present from car to car. This is because some cars do not have all the auxiliary equipment, and because many types of auxiliary equipment cycle on and off as needed. Compressors and motor/alternators, for example, are usually put on alternate cars, and both of these items cycle on and off independently. In addition, the noise level from some sources such as traction motor fans are highly dependent on speed. Consequently, picking a noise threshold beyond which the car is not considered acceptable, may be difficult. Appropriate signal processing that could correct measured noise levels for variations in vehicle speed would eliminate speed as a variable. Alternately, the monitoring systems could be set up in a speed controlled location.

6. COST-EFFECTIVENESS EVALUATION

For the purpose of rank ordering noise control treatments for further investigation and providing guidance for the work to be pursued under the remainder of this DOT project, a cost-effectiveness evaluation was performed. The control of squeal, impact and roar were considered separately and only wayside noise control was considered.

It is important to recognize at the outset that the results of this evaluation may not be directly applicable for use by a transit authority in selecting among possible noise control treatments for their system. This is true for several reasons:

1. Since wheel/rail noise control is being emphasized, no other noise sources, such as propulsion equipment noise, are being considered. Therefore, the noise reduction achieved on an actual transit system will typically be less than is predicted here, due to the presence of other non-treated sources.
2. The cost data used here is often incomplete, and at best accounts for only direct costs. Many cost factors, such as the secondary cost savings provided by the treatments (e.g., track maintenance cost savings resulting from welded rail) were not accounted for.
3. The separate consideration of squeal, impact, and roar does not account for possible cross-benefits achieved by using a particular treatment. For example, tuned-damper wheels are assumed to provide some reduction for each type of wheel/rail noise. Although it turns out that these wheels are not the most cost-effective for any type of wheel/

rail noise, when overall wheel/rail noise control is considered (simultaneous reduction of squeal, impact, and roar), they may appear far more cost effective.*

4. The cost-effectiveness evaluation was based on a hypothetical "average" U.S. transit system that may have significant differences from any particular system. For example, the amount of curved track on the system affects the trade-off between wheel treatments and rail treatments for elimination of curve squeal. In addition, treatment effectiveness can vary between systems. For example, wheel truing effectiveness depends on wheel roughness before truing.*

In order to identify which wheel/rail noise control treatments, applied either individually or in combination, are the most cost-effective (i.e., are least costly for a specified amount of noise reduction), the effectivenesses and costs of the treatments were estimated for a generic transit system. The make-up of this generic transit system is based on the average of the track types and lengths, and vehicle fleet size on the six "older" U.S. rapid transit systems (NYCTA, PATH, CTA, MBTA, SEPTA, and GCRTA). Table 6.1 summarizes the make-up of our generic system. Since only wayside (and station) noise was considered for the purposes of the cost-effectiveness evaluation, only the outdoor and station track mileages were needed for the generic system. Furthermore, for simplicity, no distinction was made between the effectiveness of the various treatments on at-

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*A methodology for performing a more complete cost-effectiveness evaluation, which is not limited by restrictions 1,3, and 4 above was initially developed at the U.S. DOT Transportation System Center (TSC) [138], and is now being improved upon and computerized [139].

TABLE 6.1. TRACK LENGTHS AND VEHICLE FLEET SIZE ON THE GENERIC RAPID TRANSIT SYSTEM.

Outdoor Track and Station Track in Tunnel		
Tangent Welded	21.2 km (13.2 mi)	(single track)
Tangent Jointed	102.0 km (63.4 mi)	
Curve,* Jointed	15.4 km (9.6 mi)	
Total	138.6 km (86.2 mi)	
Vehicle Fleet		
Cars With Slip/Slide Protection	171	
Cars Without Slip/Slide Protection	1128	
Total	1299	

*Curves of radius less than 300 m (1000 ft); i.e., for which wheel squeal is a concern.

grade, steel-elevated, concrete-elevated, embankment or other types of track; therefore, these track categories are not listed separately in Table 6.1

Cost-Effectiveness Methodology

Five separate sources of wheel/rail noise are considered: wheel flats (W_F), random small-scale wheel roughness (W_R), random small-scale rail roughness (R_R), rail joints (R_J), and rail curves (R_C). Rail corrugation, another major source of wheel/rail noise is assumed not to be present, either because it is not a problem on the generic system or it has already been removed through rail grinding -- the only known technique for treating rail corrugation. Based on the available data (summarized in this report), the effectiveness of each of the wheel/rail noise control techniques at reducing each of the five noise sources was estimated. These effectiveness "vectors" are presented in Table 6.2.

Also presented in this table are the incremental costs of applying each treatment on the generic system. The costs for each treatment were computed for a 30-year period, and represent the costs over and above the normal rail, wheel and maintenance costs which would be necessary during the same 30-year period if the treatments were not applied. The baseline information and costing assumptions are given in Appendix A.

In order to compute the overall noise reduction achieved by applying any particular treatment to the generic transit system, the untreated noise levels from each wheel/rail noise source must be known. Three separate untreated situations were considered -- one each for squeal, impact, and roar. The corresponding untreated noise levels assumed for this study are given in Table 6.3.

TABLE 6.2. NOISE REDUCTION EFFECTIVENESS AND COSTS OF TREATMENTS.

Treatments	Wayside Noise Reduction[1] (dBA)					Incremental Cost on the Generic System Over a 30-Yr Period[2] ($M)
	W_F	W_R	R_R	R_J	R_C	
Resilient wheels	1	1	1	1	20	7.2
Ring-damped wheels	1	1	1	1	15	1.1
Tuned-damper wheels	4	4	4	4	50	5.7
Constrained-layer damped	1	1	1	1	50	2.1
Damping alloy wheels	0	1	1	0	15	7.2
Resiliently treaded wheels	7	7	7	7	10[3]	7.2
Nickel-titanium treaded wheels	3	3	3	3	50	62.4
Wheel covers	2	2	2	2	0	2.1
Wheel truing	50	4	0	0	4	2.4
Rail lubrication (on gauge side)	0	0	0	0	5[4]	1.0 (curves only)
Water spray lubrication	0	0	0	0	50	1.0 (curves only)
Rail grinding	0	0	4	1	0	2.9
Rail welding	0	0	0	50	0	27.4
Joint maintenance	0	0	0	20	0	5.9
Smooth transition joints	0	0	0	20	0	4.6
Resilient rail	0	4	4	20[5]	0	18.2
Damped rails	2	2	2	2	0	13.6
"Hardfacing" rails	0	0	0	0	50	3.0 (curves only)
Nickel-titanium treaded rails[6]	3	3	3	3	50	45.6 (curves only)
Rail barriers[7]	2	2	2	2	2	10.9
Wayside barriers	10	10	10	10	10	36.4
Slip/slide prevention	10[8]	0	0	0	0	15.9
Steerable trucks	0	0	0	0	50	10.4
Vehicle skirts	3	3	3	3	3	22.7
Soft primary suspension[9]	0	2	2	0	0	7.8
Rail barriers + vehicle skirts	10	10	10	10	10	33.6

[1]The value presented are for the reduction of the individual sources noted: W_F = wheel flats; W_R = random small-scale wheel roughness; R_R = random small-scale rail roughness; R_J = rail joints; R_C = rail curves. A reduction of 50 dBA implies that the treatment eliminates the source. Values based on data for at-grade, tie and ballast track.

[2]These are the costs over and above the normal rail, wheel, and maintenance costs during a 30-yr period.

[3]Assumes some damping provided by elastomer, although not as much as for resilient wheels.

[4]Assumes 70% of squeals are eliminated: $-10 \log(1-.7) = 5$ dB.

[5]Assumes a smooth transition joint.

[6]Assumes same effectiveness as nickel-titanium treaded wheel.

[7]Assumes rail radiates equally with wheel and that rail barrier reduces rail radiation only by 6 dB.

[8]Assumes 90% of wheel flats are eliminated: $-10 \log(1-.9) = 10$ dB.

[9]Assumes very stiff primary before treatment.

TABLE 6.3. UNTREATED SOURCE NOISE LEVELS.

Type of Wheel/Rail Noise and Track/Vehicle Description	Wayside Noise Level From Each Source* (dBA)					Overall Untreated Noise Level (dBA)
	W_F	W_R	R_R	R_J	R_C	
Squeal: at-grade, tie and ballast, curved, jointed track of radii less than 300 m (1000 ft). Vehicle speed of 32 km/h (20 mph). Unflatted wheels. Composition tread brakes.	$0^{\dagger}$	63	63	69	90	90.1
Impact:** at-grade, tangent, jointed, tie and ballast track. Composition tread-brake cars with flatted wheels, traveling at 60 km/h (37 mph).	83	71	71	78	$0^{\dagger}$	84.6
Roar: at-grade, tangent, welded, tie and ballast track. Composition tread-braked cars with normally worn but unflatted wheels, traveling at 60 km/h (37 mph).	$0^{\dagger}$	71	71	$0^{\dagger}$	$0^{\dagger}$	74

*W_F = wheel flat; W_R = random small-scale wheel roughness; R_R = random small-scale rail roughness; R_J = rail joints; R_C = rail curve squeal.

†For all practical purposes, a zero means the source in not present.

**Two additional cases were looked at for impact noise. One for which no joint noise was present (R_J = 0), the other in which no wheel flats were present (W_F = 0); the roar noise (W_R = R_R = 71) was present in both these cases.

The overall treatment effectiveness is computed by subtracting the effectiveness values given in Table 6.2 from the appropriate untreated source levels given in Table 6.3 and then summing (on an energy basis) the treated source levels to compute a new (treated) overall level. The difference between the overall untreated and treated noise levels represents the treatment effectiveness. When combined treatments are used, the individual source reductions provided by each treatment (values in Table 6.2) are summed to get the combined source reduction effect. This combined effectiveness is then subtracted from the untreated source levels. Examples of this computation are given in Table 6.4.

The cost-effectiveness is defined as the overall reduction, in dBA, provided by the treatment, divided by its cost. (The cost for a combined treatment is simply the sum of the costs given in Table 6.2 for the individual treatments.)

A computer program was written for carrying out the computations described above. The treatments and combinations of treatments applied for each type of noise situation were selected and given as an input to the program. Those treatments which were considered (either individually or in appropriate combinations) for control of each type of wheel/rail noise are indicated in Table 6.5. The results are presented in Appendix B. The analysis of the results are presented below for each type of wheel/rail noise.

Squeal Noise Control

Figure 6.1 illustrates the results of the cost vs. effectiveness computation for squeal noise control. Each dot on the figure represents a treatment (or combination of treatments).

TABLE 6.4. SAMPLE COMPUTATION OF OVERALL TREATMENT EFFECTIVENESS.

Case 1. Effectiveness of Wheel Truing in Reducing Roar Noise.						
	W_F	W_R	R_R	R_J	R_C	Overall Level
A. Untreated roar source levels, dBA (from Table 6.3)	0	71	71	0	0	74.0
B. Wheel truing effectiveness, dBA (from Table 6.2)	50	4	0	0	0	-
Treated source levels, dBA (A-B)	-50	67	71	0	0	72.5
Overall treatment effectivess (overall untreated level — overall treated level)						1.5

Case 2. Combined Effectiveness of Wheel Truing and Rail Grinding in Reducing Roar Noise.						
	W_F	W_R	R_R	R_J	R_C	Overall Level
A. Untreated source levels	0	71	71	0	0	74.0
B. Wheel truing effectiveness	50	4	0	0	0	-
C. Rail grinding effectiveness	0	0	4	1	0	-
D. Combined effectiveness (B+C)	50	4	4	1	0	-
Treated source levels (A-D)	-50	67	67	-1	0	70.0
Overall combined treatment effectiveness						4.0

TABLE 6.5. TREATMENTS CONSIDERED FOR THE REDUCTION OF EACH TYPE OF WHEEL/RAIL NOISE

Treatment	Squeal	Impact	Roar
Resilient Wheels	✓	✓	✓
Ring-Damped Wheels	✓	✓	✓
Tuned-Damped Wheels	✓	✓	✓
Constrained-Layer Damped Wheels	✓	✓	✓
Damping Alloy Wheels	✓	-	✓
Resiliently Treaded Wheels	✓	✓	✓
Nickel-Titanium Treaded Wheels	✓	✓	✓
Wheel Covers	-	✓	✓
Wheel Truing	-	✓	✓
Rail Lubrication (on gauge side of rail)	✓	-	-
Water Spray Lubrication	✓	-	-
Rail Grinding	-	-	✓
Rail Welding	-	✓	-
Joint Maintenance	-	✓	-
Smooth Transition Joints	-	✓	-
Resilient Rail	-	-	✓
Damped Rails	-	✓	✓
"Hard Facing" Rails	✓	-	-
Nickel-Titanium Treaded Rails	✓	-	-
Rail Barriers	-	✓	✓
Wayside Barriers	-	✓	✓
Slip/Slide Prevention	-	✓	-
Steerable Trucks	✓	-	-
Vehicle Skirts	-	✓	✓
Soft Primary Suspension	-	-	✓

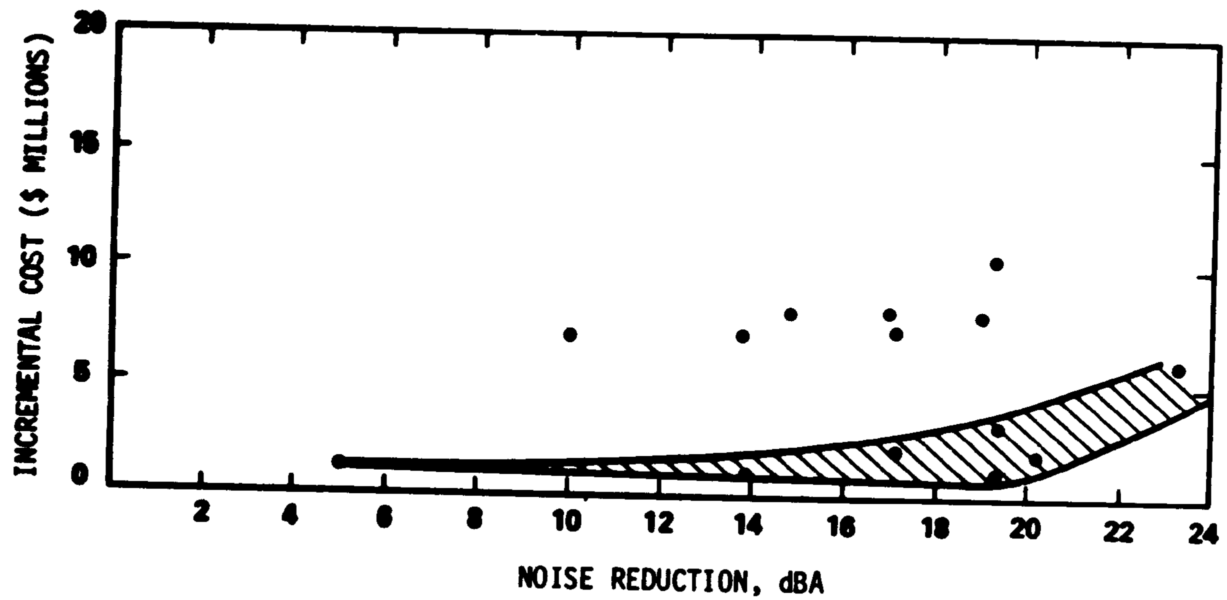

FIG. 6.1. TREATMENT COST VERSUS SQUEAL NOISE REDUCTION ON THE GENERIC TRANSIT SYSTEM. EACH DOT REPRESENTS A TREATMENT OR COMBINATION OF TREATMENTS. THOSE FALLING IN THE SHADED REGION ARE CONSIDERED TO BE THE MOST COST-EFFECTIVE.

The dots falling in the shaded area represent those treatment combinations which provide the most noise reduction for a given cost. These treatments are listed in Table 6.6.

It is at once apparent that water spray lubrication stands out as the most cost-effective treatment for elimination of squeal. Ring-damped wheels and constrained layer damped wheels (rim damping) also appear extremely cost-effective.

Those treatments which appear *not* to hold much promise for cost-effective reduction of *squeal* noise include Nitinol-treaded wheels, Nitinol-treaded rails, resiliently treaded wheels, steerable truck, damping alloy wheels, and surprisingly, resilient wheels.

In interpreting these results, it should be remembered that effectiveness, in this case, is measured only for squeal noise control and that cost savings which might exist because of reduced track, wheel, and truck wear and maintenance have not been accounted for in the cost of the treatment. This is particularly important for the steerable truck (which might have little or no incremental cost if reduced wheel and rail wear could be properly estimated and accounted for) and for resilient wheels (for which reduced track and undercar equipment maintenance costs have not been included).

Impact Noise Control

Figure 6.2 illustrates the results of the cost-effectiveness computation for the reduction of impact noise from both wheel flats and rail joints. The treatments which fall into the shaded area in the figure are listed in Table 6.7.

Wheel truing stands out as not only the most cost-effective technique, but also as a necessary one to be used in combination

TABLE 6.6. MOST COST-EFFECTIVE TREATMENTS FOR REDUCING WHEEL SQUEAL.*

Treatment	Overall Reduction (dBA)	Incremental Cost Over 30 Years ($M)	Cost-Effectiveness- (dB/$M)
Water Spray Lubrication	19.3	1.0	19.3
Ring-Damped Wheels	13.9	1.1	12.6
Constrained Layer Damped Wheels	20.3	2.1	9.7
Rail Lubrication and Ring-Damped Wheels	17.2	2.1	8.2
"Hard Facing" Rails	19.3	3.0	6.4
Rail Lubrication (Flange Lub.)	4.9	1.0	4.9
Tuned-Damper Wheels	23.3	5.7	4.1

*These are the treatments represented by the dots within the shaded region in Fig. 6.1.

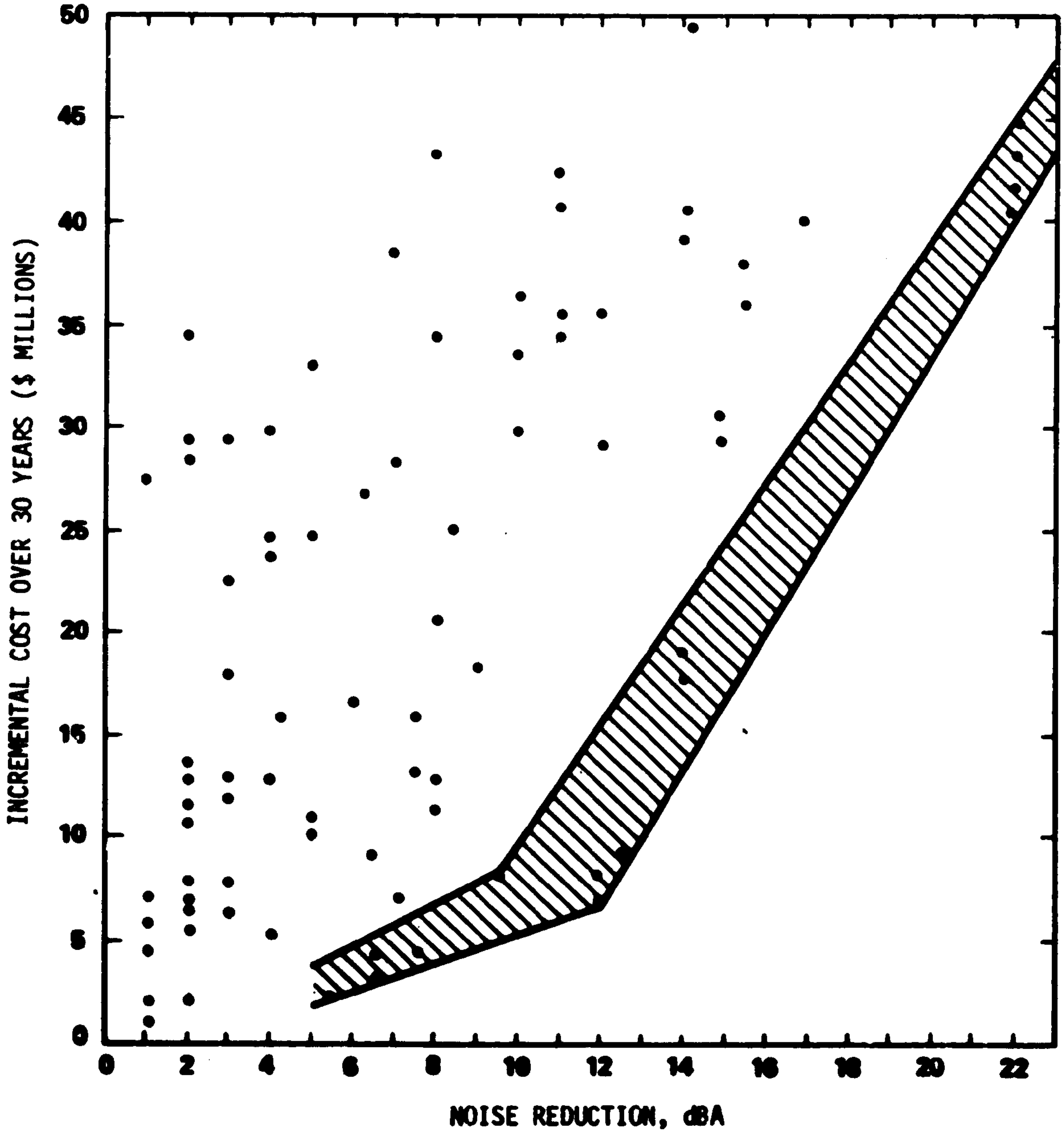

FIG. 6.2. TREATMENT COST VERSES IMPACT NOISE REDUCTION ON THE GENERIC TRANSIT SYSTEM.

TABLE 6.7. MOST COST-EFFECTIVE TREATMENTS FOR REDUCING IMPACT NOISE WHEN BOTH WHEEL FLATS AND RAIL JOINTS ARE PRESENT.*

Treatment	Overall Noise Reduction (dBA)	Incremental Cost Over 30 Years ($M)	Cost-Effectiveness (dBA/$M)
Wheel Truing	5.5	2.4	2.3
Wheel Truing and Ring-Damped	6.5	3.5	1.9
Wheel Truing and Smooth Transition Joints	12.0	7.0	1.7
Wheel Truing and Wheel Covers	7.5	4.5	1.7
Wheel Truing and Constrained Layer Damping	6.5	4.5	1.4
Wheel Truing and Joint Maintenance	12.0	8.3	1.4
Wheel Truing and Resiliently Treaded Wheels†	12.5	9.6	1.3
Wheel Truing and Tuned-Damped Wheels	9.5	8.1	1.2
Wheel Truing, Smooth Transition Joints and Rail Barriers	14.0	17.9	.8
Wheel Truing, Joint Maintenance and Rail Barriers	14.0	19.2	.7
Wheel Truing, Smooth Transition Joints, Rail Barriers and Vehicle Skirts	22.0	40.6	.5
Wheel Truing, Joint Maintenance, Rail Barriers and Vehicle Skirts	22.0	41.9	.5
Wheel Truing, Smooth Transition Joints and Wayside Barriers	22.0	43.4	.5
Wheel Truing, Joint Maintenance and Wayside Barriers	22.0	44.7	.5
Wheel Truing, Resilient Rail and Wayside Barriers	25.8	57.0	.5

*These treatments are the ones represented by the dots within the shaded region in Fig. 6.2.

†As currently envisioned the treads on resiliently treaded wheels would be replaced rather than trued.

with the other techniques. The immediate question which arises is why slip/slide prevention does not show up as among the most cost-effective. The answer probably lies in the assumptions made about the effectiveness of both wheel truing and slip/slide prevention in reducing wheel flat noise. Increasing the wheel truing rate by 50% was assumed to totally eliminate wheel flat noise which it clearly cannot accomplish. Slip/slide prevention was assumed to eliminate 90% of the flats for a computed effectiveness of 10 dB for wheel flat noise reduction. Furthermore, the costs of reduced wheel life caused by increased truing was not accounted for. The potential cost savings from reduced track and truck maintenance resulting from lower wheel/rail forces in the absence of wheel flats would be comparable for both treatments. A more complete analysis would have to account for these factors, all of which would tend to increase the cost effectiveness of slip/slide prevention and decrease that of wheel truing.

Other treatments which look attractive for impact noise control are smooth transition joints, improved joint maintenance, tuned-damper wheels, and resiliently treaded wheels. In order to get large reductions (≥20 dB), resilient rail, wayside barriers, or vehicle skirts plus rail barriers appear to be required in addition to wheel truing and joint improvement. Although ring-damped wheels, wheel covers, and constrained-layer damped wheels appear, in combination with wheel truing, to be cost-effective, they only add 1 or 2 dB to the effectiveness of truing alone and are therefore not considered appropriate treatments for impact noise control.

One track treatment noticeably absent from the cost-effectiveness table is rail welding. Several reasons may exist for this. First, the effectiveness of smooth transition joints and improved joint maintenance (20 dB) may well have been overestimated, since for all practical purposes, a 20 dB reduction in joint noise implies elimination of that source. Second, the costs used for rail welding assumed a field welding program using the existing rail and a flash butt welding train. Over the 30-year period, it was assumed that this process would be required twice. In actuality, when the rail replacement was performed, shop welded rail could be used at a much reduced cost. Finally, the reduction in track maintenance costs (other than routine joint inspection) was not accounted for in costing the welded rail. All these factors need to be more closely examined before concluding that rail welding was not a cost-effective noise reduction technique. This is especially true, since many older transit systems in both Europe and the U.S. are now, or have already, converted to continuous welded rail for other than noise reasons.

Treatments which appear *not* to be cost-effective for impact noise reduction are nickel-titanium treaded wheels, vehicle skirts (when used without rail barriers), resilient wheels, and damped rails.

The results of the cost-effectiveness computations for the situation where wheel flats are present but not rail joints lead to the same conclusions as to cost-effective treatments as presented above. For the situation where rail joints are present but no wheel flats, the only additional treatment which appears to be worth pursuing is the tuned-damper wheels.

Roar Noise Control

Figure 6.3 illustrates the results of the cost-effectiveness computation for roar noise control. The most cost-effective

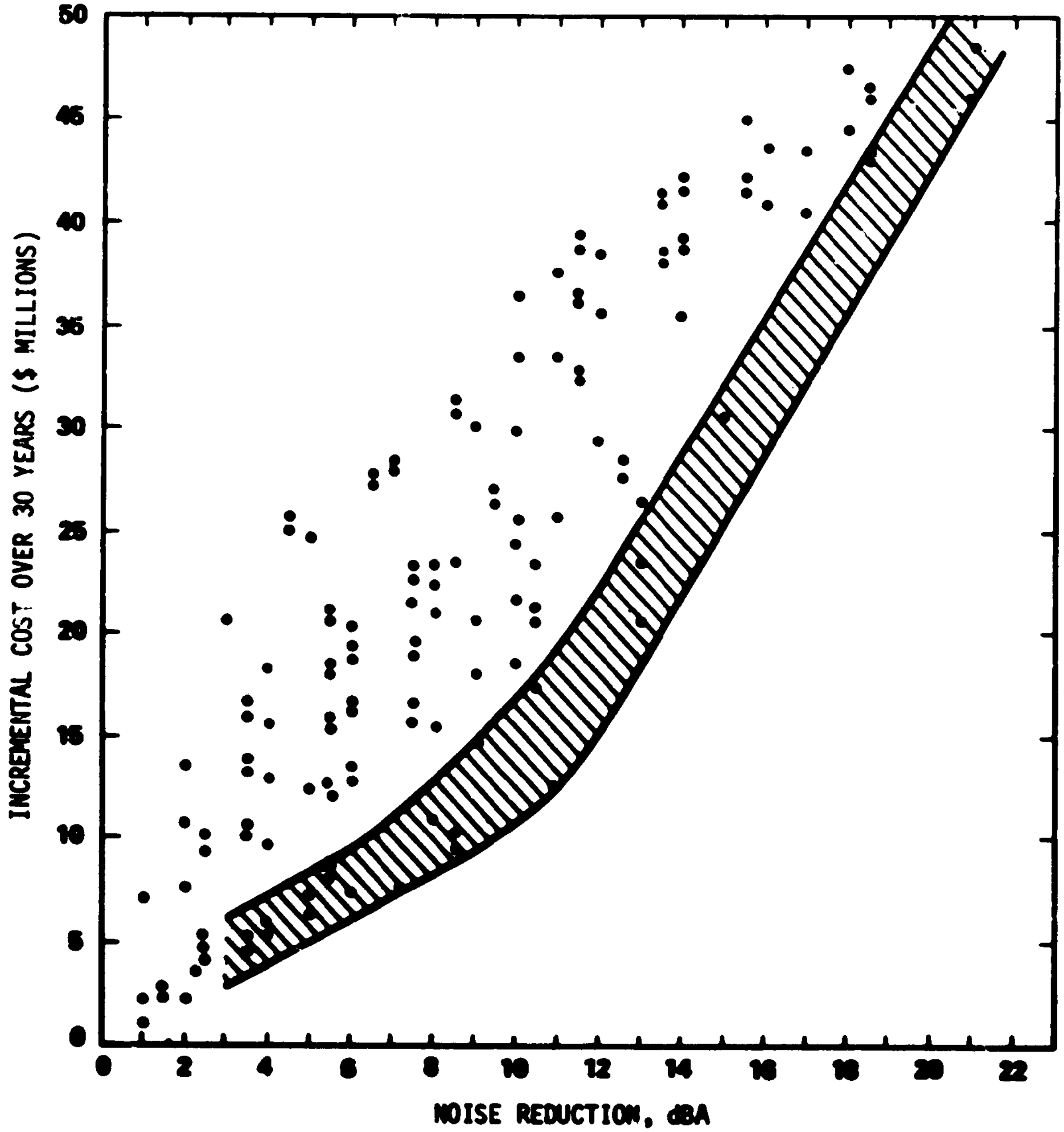

FIG. 6.3. TREATMENT COST VERSES ROAR NOISE REDUCTION ON THE GENERIC TRANSIT SYSTEM.

treatments, indicated by the dots within the shaded area of the figure, are listed in Table 6.8.

The treatments that stand out as being cost-effective (usually when used in combination with each other, but in some cases even used individually) are resiliently treaded wheels, wheel truing, rail grinding, and tuned-damper wheels. Ring-damped wheels, constrained-layer damped wheels, soft primary suspensions, and wheel covers appear in the list of the most cost-effective; however, as was the case for impact noise, these treatments only add 1 or 2 dB to the effectiveness of the treatments with which they are being combined. Because of their individually low cost per dB of reduction, they appear to be cost-effective. However, the marginal increase they provide is not considered to be worth even the relatively low increased cost.

Treatments which appear to be cost-effective when large reductions ($\geq$ 15 dB) in roar noise are required include vehicle skirts and rail barriers in combination, wayside barriers, and resilient rails. In all cases, these large reductions require the simultaneous use of wheel truing and resiliently treaded wheels, rail grinding and resiliently treaded wheels, or all three.

Treatments which do *not* appear to be cost-effective for roar noise control include nickel-titanium treaded wheels, damped rails, resilient wheels, damping alloy wheels, vehicle skirts (when used without rail barriers), and rail barriers (when used without vehicle skirts).

TABLE 6.8. MOST COST-EFFECTIVE TREATMENTS FOR REDUCING ROAR NOISE.*

Treatments	Overall Noise Reduction (dBA)	Incremental Cost Over 30 Years ($M)	Cost-Effectiveness (dBA/$M)
Resiliently Treaded Wheels	7.0	7.2	1.0
Wheel Truing and Resiliently Treaded Wheels†	8.6	9.6	.9
Wheel Truing, Rail Grinding and Resiliently Treaded Wheels†	11.0	12.5	.9
Rail Grinding and Resiliently Treaded Wheels	8.6	10.1	.8
Wheel Truing, Rail Grinding and Wheel Covers	6.0	7.4	.8
Wheel Truing and Wheel Covers	3.6	4.5	.8
Wheel Truing, Rail Grinding and Ring-Damped Wheels	5.0	6.4	.8
Wheel Truing and Rail Grinding	4.0	5.3	.8
Wheel Truing, Rail Grinding and Tuned-Damped Wheels	8.0	11.0	.7
Rail Grinding and Wheel Covers	3.6	5.0	.7
Tuned-Damped Wheels	4.0	5.7	.7
Wheel Truing and Tuned-Damped Wheels	5.5	8.1	.7
Wheel Truing, Rail Grinding and Constrained-Layer Damped Wheels	5.0	7.4	.7
Rail Grinding and Tuned-Damped Wheels	5.6	8.6	.6
Wheel Truing, Rail Grinding, Resiliently Treaded Wheels and Soft Primary Suspension†	13.0	20.3	.6
Wheel Truing, Resiliently Treaded Wheels and Soft Primary Suspension†	10.6	17.4	.6
Resiliently Treaded Wheels and Soft Primary Suspension	9.0	15.0	.6
Rail Grinding, Resiliently Treaded Wheels and Soft Primary Suspension	10.6	17.9	.6
Wheel Truing, Rail Grinding, Resiliently Treaded Wheels and Rail Barriers†	13.0	23.4	.6
Wheel Truing, Rail Grinding, Resiliently Treaded Wheels and Resilient Rail†	15.0	30.7	.5
Wheel Truing, Rail Grinding, Resiliently Treaded Wheels, Rail Barriers and Vehicle Skirts	21.0	46.1	.5
Wheel Truing, Resiliently Treaded Wheels, Rail Barriers and Vehicle Skirts†	18.6	43.2	.4
Wheel Truing, Rail Grinding, Resiliently Treaded Wheels and Wayside Barriers†	21.0	48.9	.4
Rail Grinding, Resiliently Treaded Wheels, Rail Barriers and Vehicle Skirts	18.6	43.7	.4

*The treatments listed here are the ones represented by the points within the shaded region in Fig. 6.3.
†As currently envisioned, the treads on the resiliently treaded wheels would be replaced rather than trued.

7. RECOMMENDATIONS FOR FURTHER RESEARCH AND TESTING UNDER THIS PROJECT

Table 7.1 is a list of those treatments identified in Section 6 as being among the most cost-effective for noise control or of high priority because of their combined acoustical and nonacoustical benefits. Identification of unresolved issues and research requirements for each of these treatments (as well as for the other treatments not included in Table 7.1) are contained in earlier sections. This section focuses on those treatments identified with a "yes" in the "Study Recommendation" column of Table 7.1. The explanation for not recommending study (under this project) of the other treatments contained in the table are given in the footnotes.

The following "projects" are ones that are believed to be compatible with the goals of the present program. However, funding limitations will dictate how many and to what extent these projects can be pursued.

Squeal Treatments

There are numerous wheel treatments which will effectively eliminate or substantially reduce wheel squeal. What appears to be needed is a good site-specific solution to wheel squeal. Both water spray lubrication and hard facing of rails fall into this category. For each of these, current in-service installations should be identified and their performance evaluated. Alternative "lubricants" for rail tread treatments should be investigated. Based on these findings, a detailed test plan for an in-service evaluation of each treatment should be developed. Alternately, if design changes to the existing systems seem appropriate, such changes should be recommended.

TABLE 7.1. SUMMARY OF HIGH PRIORITY TREATMENTS FOR WHEEL/RAIL NOISE CONTROL.

	Study Recommendation*
Squeal Noise Control	
Water Spray Lubrication	Yes
Ring-damped Wheels	1,2
Constrained Layer Damped Wheels	2
Rail (wheel flange) Lubrication	4
Hard Facing Rails	Yes
Tuned Damper Wheels	1,2
Steerable Truck†	4
Resilient Wheels†	4
Impact Noise Control	
Wheel Truing	Yes
Smooth Transition Joints	3
Improved Joint Maintenance Procedures	2,4
Resiliently Treaded Wheels	Yes
Tuned Damper Wheels	1,2
Slip/Slide Detectors**	Yes
Welded Rail†	4
Roar Noise Control	
Resiliently Treaded Wheels	Yes
Tuned Damper Wheels	1,2
Rail Grinding and Wheel Truing	Yes

*A "Yes" in this column indicates that the treatment is recommended for further study under this contract. The numbers 1 through 4 are explained below and represent reasons why these treatments are not recommended for further study as part of the present contract.

†These treatments did not fall into the most cost-effective category primarily because their major benefits, extension of wheel and rail life and reduction of maintenance requirements, were not fully accounted for in determining their costs. If these benefits were properly costed, it is believed that the treatments would become among the most cost-effective.

**Although this treatment did not come out as being among the most cost-effective, this is believed to be primarily due to the overestimation of the effectiveness of wheel truing relative to slip/slide prevention.

[1]Treatments are being developed by others.

[2]In-service testing is required, which is presently beyond the funding level of this contract.

[3]Treatment has limited applicability and/or other equally effective treatments are available.

[4]Primary benefits of treatment are nonacoustical.

Impact Treatments

In the area of impact noise, a detailed evaluation of the trade-off between increased wheel truing and the use of slip/slide detectors is being recommended. There are several reasons why wheel tread defects (rather than rail joints) are being emphasized. Available data appear to indicate that wheel flats or other tread defects are more significant sources of impact noise than are rail joints. Furthermore, most rail systems are converting or have converted to welded rail; insulated joints are often bonded; and special trackwork is very site-limited.

Questions which the current work would seek to answer (perhaps for one system) include:

1. How important is the wheel flat problem?
   a. What percent of the wheels on the system are flatted (or have other noise producing tread defects)?
   b. At what rate are such surface defects generated?
   c. What percent of flats which are generated work themselves out through normal wear?
2. What benefit could be obtained (systemwide) by reducing the percentage of flatted wheels?
3. How effective are slip/slide detection systems at preventing impact-producing wheel tread defects.
4. How much truing would be needed to reduce the percentage of flatted wheels by a desired amount?

5. What effect does increased wheel truing have on wheel life?

6. What percent of the truing currently being performed is done for the removal of noise-producing wheel tread defects?

It is not expected that all these questions can be answered as part of this study. Rather, sufficient information should be obtained for the design of a long-term in-service evaluation and cost-benefit study of selected wheel tread conditioning techniques and slip/slide prevention systems.

Roar Treatments

The treatments selected for further study in roar noise reduction include resiliently treaded wheels (the only truly innovative treatment being recommended), wheel truing and rail grinding. In addition, one other project is recommended, not necessarily for performance under this contract, but because of its potential overall importance to wheel/rail noise.

The proposed project for resiliently treaded wheels is organized as follows:

1. Use analytical models to estimate the performance and to define the desired properties of these wheels.
2. Develop conceptual designs.
3. Build prototype wheels for laboratory and/or field testing.
4. Evaluate the prototype wheels.
5. Make recommendations for further development and testing.

In the area of wheel truing and rail grinding, the following tasks should be performed:

1. Measure surface roughness of various in-service wheels and rails and examine the correlation between these roughnesses and noise.
2. Measure surface roughnesses achievable with various wheel tread smoothing (truing/grinding) and rail smoothing methods.
3. Develop criteria for wheel truing and rail grinding necessary for obtaining specified wheel/rail noise levels.
4. Examine new techniques for wheel truing and/or rail grinding.

One problem which has plagued some transit systems and not affected other to any significant extent is rail corrugation. Past efforts at finding causes and solutions for rail corrugation in general have not been fruitful. The only generally accepted solution appears to be rail grinding. A careful, in-depth study into the specific cause (or causes) of rail corrugation on a single selected transit system should yield useful guidelines for its prevention on that system. This would have significant implications for cost savings on track and vehicle maintenance.

APPENDIX A

TREATMENT COSTS AND ASSUMPTIONS

TABLE A.1. INCREMENTAL COSTS DUE TO TREATMENTS OVER A 30 YEAR PERIOD.

Treatment	Incremental Cost*	Comments
Resilient wheels	$ 5,540/car	Tire and elastomers replaced twice. Incremental cost is average for three types of resilient wheels.
Ring-damped wheels	$ 864/car	Assumes new rings needed when wheels are replaced.
Tuned-damped wheels	$ 4,350/car	Assumes that tuned dampers can be reused once.
Constrained layer damped wheels	$ 1,600/car	New treatment needed when wheels are replaced.
Damping alloy wheels	$ 5,540/car	Same costs assumed as for resilient wheels.
Resiliently treated wheels	$ 5,540/car	Same costs assumed as for resilient wheels.
Nitinol treaded wheels	$ 48,000/car	Assumes Nitinol tread lasts for 15 yr.
Wheel covers	$ 1,600/car	Can be reused when wheels are replaced. Neglects increased wheel inspection costs.
Wheel truing	$ 973,000 + $176/car-set of wheels trued during 30 yr	Includes cost of an underfloor truer with installation and maintenance, as well as costs for truing.
Rail lubrication	$ 15,000/curve	Includes cost of lubricator with installation and maintenance plus cost of lubricant.
Water spray lubrication	$ 15,000/curve	Same cost assumed as for rail lubrication.
Rail grinding	$ 1,125/track mile ground during 30 yr	Costs for contracted rail grinding services used.
Rail welding	$3,000,000 + $334,000/mi	Includes cost of flash butt welding car and welding in-place rail. Cost is reduced by amount saved in routine rail joint maintenance.
Joint maintenance	$ 81,000/track mile	Assumes incremental cost is twice that of routine joint maintenance.
Smooth transition joint	$ 31,700/track mile	Joints replaced with rail after 15 yr.
Resilient rail	$211,000/track mile	Assumes incremental cost is equal to cost of standard rail.
Damped rails	$158,000/track mile	Damping replaced with rail after 15 yr.
"Hardfacing" rails	$317,000/track mile	Assumes treatment extends rail life on curves by a factor of 2.
Nickel-titanium treaded rails	$4,750,000/track mile	Assumes a 30 yr rail life.
Rail barriers	$127,000/track mile	.45 m (1.5 ft) high barrier near rails. Does not include possible increased costs for track or barrier maintenance, snow removal, etc.
Wayside barriers	$422,000/track mile	1.5 m (5 ft) high barrier on each side of track Same comments on costs as for rail barrier.
Slip/slide prevention	$ 15,500/car - $11 x (no. of wheels normally trued in 30 yr)	Assumes 1/2 of the normal wheel truing is necessary.
Steerable trucks	$ 8,000/car	Assumes possibility of converting existing trucks using guided steering.
Vehicle skirts	$ 17,500/car	Neglects possible increases in undercar maintenance and inspection costs.
Soft primary suspension	$ 6,000/car	Neglects possible savings in truck and undercar equipment maintenance costs.

*These are the costs over and above the normal rail, wheel, and maintenance costs during a 30-yr period.

TABLE A.2. GENERAL ASSUMPTIONS FOR BASELINE COSTS.

1.	Rail life = 15 yr on tangent track; 5 yr on curve
2.	Wheel life = 7.5 yr
3.	Labor rate = $15/hr
4.	Average annual car mileage = 64,000 km/yr (40,000 mi/yr)
5.	Average wheel truing rate = once every 117,000 km (72,600 mi)*
6.	Assume one-third of curves on system already have rail lubrication.
7.	Standard wheel costs $745 (including inspection costs over its lifetime).
8.	Routine joint maintenance cost = $5/joint/yr.
9.	To assess future cash flows it is assumed that the rate of inflation equals the discount rate. This implies that the present value of future payments is equal to the future payment in absolute terms and all costs over time can simply be added to arrive at a total cost for a treatment.

*For noise control purposes, this rate is increased by 50%.

APPENDIX B: COMPUTED COSTS AND EFFECTIVENESS OF TREATMENTS APPLIED TO THE GENERIC TRANSIT SYSTEM.

This appendix presents the computed costs, overall effectiveness, and cost-effectiveness (ratio of overall effectiveness to cost) for individual and combined treatments. Five cases are presented: one for squeal, three for impact, and one for roar.

TABLE B.1. WHEEL SQUEAL NOISE CONTROL

TREATMENT DESCRIPTORS		Untreated Source Levels, dBA
1	RESILIENT WHEELS	$W_F = 0$
2	RING-DAMPED WHEELS	
3	TUNED-DAMPER WHEEL	$W_R = 63$
4	CONSTRAINED LAYER DAMPED	
5	DAMPING ALLOY WHEELS	$R_R = 63$
6		
7	RESILIENTLY TREADED WHEELS	$R_J = 69$
8	NITINOL TREADED WHEELS	
9	WHEEL COVERS	$R_C = 90$
10	WHEEL TRUING	
11	RAIL LUBRICATION	
12	RAIL GRINDING	
13	RAIL WELDING	
14	JOINT MAINTAINANCE	
15	SMOOTH TRANSITION JOINTS	
16	WATER SPRAY LUBRICATION	
17	RESILIENT RAIL	
18	DAMPED RAILS	
19	WAYSIDE BARRIER (5 FEET)	
20	"HARDFACING" RAILS	
21	NITINOL TREADED RAILS	
22	RAIL BARRIERS	
23	SLIP/SLIDE PREVENTION	
24	STEERABLE TRUCKS	
25	VEHICLE SKIRTS	
26	SOFT PRIMARY SUSPENSION	
27	RAIL BARRIERS AND VEHICLE SKIRTS	
28		

TREATMENTS					E(dBA)	C($M)	CE(dB/$M)
16	0	0	0	0	19.28	1.00	19.280092000
2	0	0	0	0	13.91	1.10	12.648069500
4	0	0	0	0	20.28	2.10	9.656738520
11	2	0	0	0	17.16	2.10	8.169415830
20	0	0	0	0	19.28	3.00	6.426697310
11	0	0	0	0	4.89	1.00	4.890643120
3	0	0	0	0	23.27	5.70	4.083262860
1	0	0	0	0	17.16	7.20	2.382746280
11	1	0	0	0	19.03	8.20	2.321156890
11	5	0	0	0	16.81	8.20	2.049512983
5	0	0	0	0	13.74	7.20	1.908832950
24	0	0	0	0	19.28	10.40	1.853855000
11	7	0	0	0	14.74	8.20	1.797101840
7	0	0	0	0	9.95	7.20	1.381850290
21	0	0	0	0	22.28	45.60	0.488519177
8	0	0	0	0	22.28	62.40	0.356994782

TABLE B.2. IMPACT NOISE CONTROL WHEN BOTH WHEEL FLATS AND RAIL JOINTS ARE PRESENT.

TREATMENT DESCRIPTORS		Untreated Source Levels, dBA
1	RESILIENT WHEELS	$W_F = 83$
2	RING-DAMPED WHEELS	
3	TUNED-DAMPER WHEEL	$W_R = 71$
4	CONSTRAINED LAYER DAMPED	
5	DAMPING ALLOY WHEELS	$R_R = 71$
6		
7	RESILIENTLY TREADED WHEELS	$R_J = 78$
8	NITINOL TREADED WHEELS	
9	WHEEL COVERS	$R_C = 0$
10	WHEEL TRUING	
11	RAIL LUBRICATION	
12	RAIL GRINDING	
13	RAIL WELDING	
14	JOINT MAINTAINANCE	
15	SMOOTH TRANSITION JOINTS	
16	WATER SPRAY LUBRICATION	
17	RESILIENT RAIL	
18	DAMPED RAILS	
19	WAYSIDE BARRIER (5 FEET)	
20	"HARDFACING" RAILS	
21	NITINOL TREADED RAILS	
22	RAIL BARRIERS	
23	SLIP/SLIDE PREVENTION	
24	STEERABLE TRUCKS	
25	VEHICLE SKIRTS	
26	SOFT PRIMARY SUSPENSION	
27	RAIL BARRIERS AND VEHICLE SKIRTS	
28		

TREATMENTS					E(dBA)	C($M)	CE(dB/$M)
10	0	0	0	0	5.52	2.40	2.300932880
10	2	0	0	0	6.52	3.50	1.863496890
10	15	0	0	0	11.98	7.00	1.711724890
10	9	0	0	0	7.52	4.50	1.671608670
10	4	0	0	0	6.52	4.50	1.449386460
10	14	0	0	0	11.98	8.30	1.443623420
10	7	0	0	0	12.52	9.60	1.304399910
10	3	0	0	0	9.52	8.10	1.175585080
7	0	0	0	0	7.00	7.20	0.972221904
9	0	0	0	0	2.00	2.10	0.952379905
2	0	0	0	0	1.00	1.10	0.909088820
10	15	22	0	0	13.98	17.90	0.781121470
10	14	22	0	0	13.98	19.20	0.728233039
3	0	0	0	0	4.00	5.70	0.701753981
15	7	0	0	0	8.06	11.80	0.683272265
10	1	0	0	0	6.52	9.60	0.679399900
14	7	0	0	0	8.06	13.10	0.615466625

TABLE B.2 (Cont'd.)

10	22	0	0	0	7.52	13.30	0.565581881
10	15	27	0	0	21.98	40.60	0.541430399
10	14	27	0	0	21.98	41.90	0.524631836
19	10	15	0	0	21.98	43.40	0.506499402
10	15	25	0	0	14.98	29.70	0.504446946
22	7	0	0	0	9.00	18.10	0.497237448
19	10	14	0	0	21.98	44.70	0.491768990
15	3	0	0	0	5.06	10.30	0.491515804
10	14	25	0	0	14.98	31.00	0.483292721
4	0	0	0	0	1.00	2.10	0.476189382
10	18	0	0	0	7.52	16.00	0.470139939
15	9	0	0	0	3.06	6.70	0.457106393
19	10	17	0	0	25.76	57.00	0.451928578
14	3	0	0	0	5.06	11.60	0.436432138
10	27	0	0	0	15.52	36.00	0.431173310
27	7	0	0	0	17.00	40.80	0.416666612
10	13	0	0	0	12.13	29.80	0.407210063
19	10	0	0	0	15.52	38.80	0.400057707
15	23	0	0	0	7.99	20.50	0.389553871
14	9	0	0	0	3.06	8.00	0.382826608
14	23	0	0	0	7.99	21.80	0.366323598
15	2	0	0	0	2.06	5.70	0.361861914
22	3	0	0	0	6.00	16.60	0.361445651
27	3	0	0	0	14.00	39.30	0.356234048
10	13	27	0	0	22.13	63.40	0.349130288
10	13	22	0	0	14.13	40.70	0.347293858
10	25	0	0	0	8.52	25.10	0.339531425
27	9	0	0	0	12.00	35.70	0.336134389
25	7	0	0	0	10.00	29.90	0.334448084
19	10	13	0	0	22.13	66.20	0.334363446
27	2	0	0	0	11.00	34.70	0.317002822
27	4	0	0	0	11.00	35.70	0.308123190
15	4	0	0	0	2.06	6.70	0.307852674
22	9	0	0	0	4.00	13.00	0.307692125
27	0	0	0	0	10.00	33.60	0.297618982
14	2	0	0	0	2.06	7.00	0.294658989
10	13	25	0	0	15.13	52.50	0.288283046
27	23	0	0	0	14.25	49.50	0.287829533
19	0	0	0	0	10.00	36.40	0.274725210
19	15	0	0	0	11.06	41.00	0.269819826
27	1	0	0	0	11.00	40.80	0.269607794
23	0	0	0	0	4.25	15.90	0.267142251
19	14	0	0	0	11.06	42.30	0.261527490
14	4	0	0	0	2.06	8.00	0.257826615
22	2	0	0	0	3.00	12.00	0.249999808
25	3	0	0	0	7.00	28.40	0.246478790
13	7	0	0	0	8.07	34.60	0.233375093
22	23	0	0	0	6.25	26.80	0.233117979
15	0	0	0	0	1.06	4.60	0.231002795
22	4	0	0	0	3.00	13.00	0.230769055
25	9	0	0	0	5.00	24.80	0.201612815

TABLE B.2 (Cont'd.)

25	23	0	0	0	7.25	38.60	0.187760668
13	23	0	0	0	8.05	43.30	0.185821934
22	0	0	0	0	2.00	10.90	0.183486037
14	0	0	0	0	1.06	5.90	0.180103876
15	1	0	0	0	2.06	11.80	0.174797704
19	13	0	0	0	11.07	63.80	0.173585866
25	2	0	0	0	4.00	23.80	0.168067129
22	1	0	0	0	3.00	18.10	0.165745730
25	4	0	0	0	4.00	24.80	0.161290228
14	1	0	0	0	2.06	13.10	0.157451367
13	3	0	0	0	5.07	33.10	0.153316569
18	0	0	0	0	2.00	13.60	0.147058662
1	0	0	0	0	1.00	7.20	0.138888570
27	8	0	0	0	13.00	96.00	0.135416642
25	1	0	0	0	4.00	29.90	0.133779187
25	0	0	0	0	3.00	22.70	0.132158488
10	8	0	0	0	8.52	64.80	0.131516032
13	9	0	0	0	3.07	29.50	0.104229773
13	2	0	0	0	2.07	28.50	0.072799239
25	8	0	0	0	6.00	85.10	0.070505261
13	4	0	0	0	2.07	29.50	0.070331468
22	8	0	0	0	5.00	73.30	0.068212794
15	8	0	0	0	4.06	67.00	0.060636012
13	1	0	0	0	2.07	34.60	0.059964691
14	6	0	0	0	4.06	68.30	0.059481886
8	0	0	0	0	3.00	62.40	0.048076886
13	8	0	0	0	4.07	89.80	0.045376150
13	0	0	0	0	1.07	27.40	0.039225485

TABLE B.3. IMPACT NOISE CONTROL WHEN WHEEL FLATS BUT NO RAIL JOINTS ARE PRESENT.

TREATMENT DESCRIPTORS

1 RESILIENT WHEELS
2 RING-DAMPED WHEELS
3 TUNED-DAMPER WHEEL
4 CONSTRAINED LAYER DAMPED
5 DAMPING ALLOY WHEELS
6
7 RESILIENTLY TREADED WHEELS
8 NITINOL TREADED WHEELS
9 WHEEL COVERS
10 WHEEL TRUING
11 RAIL LUBRICATION
12 RAIL GRINDING
13 RAIL WELDING
14 JOINT MAINTAINANCE
15 SMOOTH TRANSITION JOINTS
16 WATER SPRAY LUBRICATION
17 RESILIENT RAIL
18 DAMPED RAILS
19 WAYSIDE BARRIER (5 FEET)
20 "HARDFACING" RAILS
21 NITINOL TREADED RAILS
22 RAIL BARRIERS
23 SLIP/SLIDE PREVENTION
24 STEERABLE TRUCKS
25 VEHICLE SKIRTS
26 SOFT PRIMARY SUSPENSION
27 RAIL BARRIERS AND VEHICLE SKIRTS
28

Untreated Source Levels, dBA

$W_F = 83$

$W_R = 71$

$R_R = 71$

$R_J = 0$

$R_C = 0$

TREATMENTS					E(dBA)	C($M)	CE(dB/$M)
10	0	0	0	0	11.06	2.40	4.608426150
10	2	0	0	0	12.06	3.50	3.445777920
10	9	0	0	0	13.06	4.50	2.902271690
10	4	0	0	0	12.06	4.50	2.680049510
10	7	0	0	0	18.06	9.60	1.881273180
10	3	0	0	0	15.06	8.10	1.859286760
10	1	0	0	0	12.06	9.60	1.256273200
10	22	0	0	0	13.06	13.30	0.981971629
7	0	0	0	0	7.00	7.20	0.972221866
9	0	0	0	0	2.00	2.10	0.952379823
2	0	0	0	0	1.00	1.10	0.909088656
10	18	0	0	0	13.06	16.00	0.816263914
3	0	0	0	0	4.00	5.70	0.701753952
10	27	0	0	0	21.06	36.00	0.585006185
10	25	0	0	0	14.06	25.10	0.560168229
19	10	0	0	0	21.06	38.80	0.542789243

TABLE B.3 (Cont'd.)

22	7	0	0	0	9.00	18.10	0.497237433
4	0	0	0	0	1.00	2.10	0.476189297
19	10	17	0	0	25.06	57.00	0.439639971
23	0	0	0	0	6.97	15.90	0.438450150
27	7	0	0	0	17.00	40.80	0.416666612
22	3	0	0	0	6.00	16.60	0.361445643
27	3	0	0	0	14.00	39.30	0.356234036
27	23	0	0	0	16.97	49.50	0.342855711
27	9	0	0	0	12.00	35.70	0.336134389
22	23	0	0	0	8.97	26.80	0.334752139
25	7	0	0	0	10.00	29.90	0.334448073
27	2	0	0	0	11.00	34.70	0.317002818
27	4	0	0	0	11.00	35.70	0.308123186
22	9	0	0	0	4.00	13.00	0.307692114
27	0	0	0	0	10.00	33.60	0.297618970
19	0	0	0	0	10.00	36.40	0.274725203
27	1	0	0	0	11.00	40.80	0.269607790
25	23	0	0	0	9.97	38.60	0.258325324
22	2	0	0	0	3.00	12.00	0.249999793
25	3	0	0	0	7.00	28.40	0.246478783
22	4	0	0	0	3.00	13.00	0.230769040
10	8	0	0	0	14.06	64.80	0.216978747
25	9	0	0	0	5.00	24.80	0.201612808
22	0	0	0	0	2.00	10.90	0.183486022
25	2	0	0	0	4.00	23.80	0.168067124
22	1	0	0	0	3.00	18.10	0.165745720
25	4	0	0	0	4.00	24.80	0.161290223
18	0	0	0	0	2.00	13.60	0.147058647
1	0	0	0	0	1.00	7.20	0.138888545
27	8	0	0	0	13.00	96.00	0.135416642
25	1	0	0	0	4.00	29.90	0.133779181
25	0	0	0	0	3.00	22.70	0.132158481
25	8	0	0	0	6.00	85.10	0.070505260
22	8	0	0	0	5.00	73.30	0.068212792
8	0	0	0	0	3.00	62.40	0.048076883

TABLE B.4. IMPACT NOISE CONTROL WHEN RAIL JOINTS BUT NO WHEEL FLATS ARE PRESENT.

TREATMENT DESCRIPTORS

1 RESILIENT WHEELS
2 RING-DAMPED WHEELS
3 TUNED-DAMPER WHEEL
4 CONSTRAINED LAYER DAMPEL
5 DAMPING ALLOY WHEELS
6
7 RESILIENTLY TREADED WHEELS
8 NITINOL TREADED WHEELS
9 WHEEL COVERS
10 WHEEL TRUING
11 RAIL LUBRICATION
12 RAIL GRINDING
13 RAIL WELDING
14 JOINT MAINTAINANCE
15 SMOOTH TRANSITION JOINTS
16 WATER SPRAY LUBRICATION
17 RESILIENT RAIL
18 DAMPED RAILS
19 WAYSIDE BARRIER (5 FEET)
20 "HARDFACING" RAILS
21 NITINOL TREADED RAILS
22 RAIL BARRIERS
23 SLIP/SLIDE PREVENTION
24 STEERABLE TRUCKS
25 VEHICLE SKIRTS
26 SOFT PRIMARY SUSPENSION
27 RAIL BARRIERS AND VEHICLE SKIRTS
28

Untreated Source Levels, dBA

$W_F = 0$

$W_R = 71$

$R_R = 71$

$R_J = 78$

$R_C = 0$

TREATMENTS					E(dBA)	C($M)	CE(dB/$M)
15	0	0	0	0	5.34	4.60	1.160988960
15	2	0	0	0	6.34	5.70	1.112377060
15	9	0	0	0	7.34	6.70	1.095604350
15	7	0	0	0	12.34	11.80	1.045809250
7	0	0	0	0	7.00	7.20	0.972222157
9	0	0	0	0	2.00	2.10	0.952380762
15	4	0	0	0	6.34	6.70	0.946350634
14	7	0	0	0	12.34	13.10	0.942026667
14	9	0	0	0	7.34	8.00	0.917568646
2	0	0	0	0	1.00	1.10	0.909090482
15	3	0	0	0	9.34	10.30	0.906849437
14	2	0	0	0	6.34	7.00	0.905792758
14	0	0	0	0	5.34	5.90	0.905177839
14	3	0	0	0	9.34	11.60	0.805219762
14	4	0	0	0	6.34	8.00	0.792568661
3	0	0	0	0	4.00	5.70	0.701754294

TABLE B.4 (Cont'd.)

15	1	0	0	0	6.34	11.80	0.537334681
22	7	0	0	0	9.00	18.10	0.497237545
14	1	0	0	0	6.34	13.10	0.484011397
4	0	0	0	0	1.00	2.10	0.476190250
27	7	0	0	0	17.00	40.80	0.416666657
19	15	0	0	0	15.34	41.00	0.374159735
19	14	0	0	0	15.34	42.30	0.362660732
22	3	0	0	0	6.00	16.60	0.361445759
13	7	0	0	0	12.45	34.60	0.359766785
27	3	0	0	0	14.00	39.30	0.356234096
27	9	0	0	0	12.00	35.70	0.336134445
25	7	0	0	0	10.00	29.90	0.334448144
27	2	0	0	0	11.00	34.70	0.317002870
27	4	0	0	0	11.00	35.70	0.308123238
22	9	0	0	0	4.00	13.00	0.307692263
27	0	0	0	0	10.00	33.60	0.297619034
13	3	0	0	0	9.45	33.10	0.285435978
19	0	0	0	0	10.00	36.40	0.274725262
27	1	0	0	0	11.00	40.80	0.269607835
13	9	0	0	0	7.45	29.50	0.252472218
22	2	0	0	0	3.00	12.00	0.249999961
25	3	0	0	0	7.00	28.40	0.246478856
19	13	0	0	0	15.45	63.80	0.242130570
22	4	0	0	0	3.00	13.00	0.230769195
13	2	0	0	0	6.45	28.50	0.226243183
13	4	0	0	0	6.45	29.50	0.218573922
25	9	0	0	0	5.00	24.80	0.201612886
13	0	0	0	0	5.45	27.40	0.198829578
13	1	0	0	0	6.45	34.60	0.186356379
22	0	0	0	0	2.00	10.90	0.183486207
25	2	0	0	0	4.00	23.80	0.168067204
22	1	0	0	0	3.00	18.10	0.165745830
25	4	0	0	0	4.00	24.80	0.161290301
18	0	0	0	0	2.00	13.60	0.147058794
1	0	0	0	0	1.00	7.20	0.138888823
27	8	0	0	0	13.00	96.00	0.135416662
25	1	0	0	0	4.00	29.90	0.133779246
25	0	0	0	0	3.00	22.70	0.132158564
15	8	0	0	0	8.34	67.00	0.124485809
14	8	0	0	0	8.34	68.30	0.122116387
13	8	0	0	0	8.45	89.80	0.094074951
25	8	0	0	0	6.00	85.10	0.070505283
22	8	0	0	0	5.00	73.30	0.068212820
8	0	0	0	0	3.00	62.40	0.048076916

TABLE B.5. ROAR NOISE CONTROL.

TREATMENT DESCRIPTORS

1 RESILIENT WHEELS
2 RING-DAMPED WHEELS
3 TUNED-DAMPER WHEEL
4 CONSTRAINED LAYER DAMPED
5 DAMPING ALLOY WHEELS
6
7 RESILIENTLY TREADED WHEELS
8 NITINOL TREADED WHEELS
9 WHEEL COVERS
10 WHEEL TRUING
11 RAIL LUBRICATION
12 RAIL GRINDING
13 RAIL WELDING
14 JOINT MAINTAINANCE
15 SMOOTH TRANSITION JOINTS
16 WATER SPRAY LUBRICATION
17 RESILIENT RAIL
18 DAMPED RAILS
19 WAYSIDE BARRIER (5 FEET)
20 "HARDFACING" RAILS
21 NITINOL TREADED RAILS
22 RAIL BARRIERS
23 SLIP/SLIDE PREVENTION
24 STEERABLE TRUCKS
25 VEHICLE SKIRTS
26 SOFT PRIMARY SUSPENSION
27 RAIL BARRIERS AND VEHICLE SKIRTS
28

Untreated Source Levels, dBA

$W_F = 0$

$W_R = 71$

$R_R = 71$

$R_J = 0$

$R_C = 0$

TREATMENTS					E(dBA)	C($M)	CE(dB/$M)
7	0	0	0	0	7.00	7.20	0.972222090
9	0	0	0	0	2.00	2.10	0.952380575
2	0	0	0	0	1.00	1.10	0.909090288
10	7	0	0	0	8.55	9.60	0.891134851
10	12	7	0	0	11.00	12.50	0.879999951
12	7	0	0	0	8.55	10.10	0.847019240
10	12	9	0	0	6.00	7.40	0.810810670
10	9	0	0	0	3.55	4.50	0.789976604
10	12	2	0	0	5.00	6.40	0.781249903
10	12	0	0	0	4.00	5.30	0.754716829
10	2	0	0	0	2.55	3.50	0.729969934
10	12	3	0	0	8.00	11.00	0.727272660
12	9	0	0	0	3.55	5.00	0.710978851
3	0	0	0	0	4.00	5.70	0.701754272
10	3	0	0	0	5.55	8.10	0.685789473
10	12	4	0	0	5.00	7.40	0.675675593
10	0	0	0	0	1.55	2.40	0.647872806

TABLE B.5 (Cont'd.)

12	3	0	0	0	5.55	8.60	0.645917967
10	12	7	26	0	13.00	20.30	0.640394040
12	2	0	0	0	2.55	4.00	0.638723642
10	7	26	0	0	10.55	17.40	0.606603131
7	26	0	0	0	9.00	15.00	0.599999927
12	7	26	0	0	10.55	17.90	0.589658886
10	4	0	0	0	2.55	4.50	0.567754395
10	12	7	22	0	13.00	23.40	0.555555515
12	0	0	0	0	1.55	2.90	0.536170490
10	12	3	26	0	10.00	18.80	0.531914853
10	12	9	26	0	8.00	15.20	0.526315704
10	7	22	0	0	10.55	20.50	0.514872909
12	4	0	0	0	2.55	5.00	0.510978915
12	7	22	0	0	10.55	21.00	0.502614029
10	12	7	18	0	13.00	26.10	0.498084251
7	22	0	0	0	9.00	18.10	0.497237526
10	12	7	17	0	15.00	30.70	0.488599330
4	0	0	0	0	1.00	2.10	0.476190150
10	3	26	0	0	7.55	15.90	0.475150604
12	3	26	0	0	7.55	16.40	0.460664287
10	12	26	0	0	6.00	13.10	0.458015174
10	12	3	22	0	10.00	21.90	0.456620965
10	12	7	27	0	21.00	46.10	0.455531448
10	7	18	0	0	10.55	23.20	0.454952356
10	9	26	0	0	5.55	12.30	0.451617416
10	7	17	0	0	12.55	27.80	0.451614931
12	7	18	0	0	10.55	23.70	0.445354190
3	26	0	0	0	6.00	13.50	0.444444377
12	7	17	0	0	12.55	28.30	0.443635836
12	9	26	0	0	5.55	12.80	0.433976062
7	17	0	0	0	11.00	25.40	0.433070827
7	18	0	0	0	9.00	20.80	0.432692267
10	7	27	0	0	18.55	43.20	0.429511454
19	10	12	7	0	21.00	48.90	0.429447848
12	7	27	0	0	18.55	43.70	0.424597133
7	27	0	0	0	17.00	40.80	0.416666657
10	12	3	17	0	12.00	29.20	0.410958894
10	12	3	18	0	10.00	24.60	0.406504028
9	26	0	0	0	4.00	9.90	0.404040273
10	12	3	27	0	18.00	44.60	0.403587427
19	10	7	0	0	18.55	46.00	0.403367277
10	12	5	0	0	5.00	12.50	0.399999954
10	12	1	0	0	5.00	12.50	0.399999954
19	12	7	0	0	18.55	46.50	0.399029993
10	12	7	25	0	14.00	35.20	0.397727247
10	3	22	0	0	7.55	19.00	0.397626039
10	12	9	17	0	10.00	25.60	0.390624952
10	12	9	27	0	16.00	41.00	0.390243877
19	7	0	0	0	17.00	43.60	0.389908247
12	3	22	0	0	7.55	19.50	0.387430493
10	12	9	18	0	8.00	21.00	0.380952328

TABLE B.5 (Cont'd.)

19	10	12	3	0	18.00	47.40	0.379746821
10	3	27	0	0	15.55	41.70	0.373019062
10	12	22	0	0	6.00	16.20	0.370370314
12	3	27	0	0	15.55	42.20	0.368599404
19	10	12	9	0	16.00	43.80	0.365296777
10	3	17	0	0	9.55	26.30	0.363303993
3	22	0	0	0	6.00	16.60	0.361445740
10	9	22	0	0	5.55	15.40	0.360707436
10	12	27	0	0	14.00	38.90	0.359837152
10	7	25	0	0	11.55	32.30	0.357736673
12	3	17	0	0	9.55	26.80	0.356525909
3	27	0	0	0	14.00	39.30	0.356234081
10	9	27	0	0	13.55	38.10	0.355771519
12	7	25	0	0	11.55	32.80	0.352283366
12	9	27	0	0	13.55	38.60	0.351163063
19	10	3	0	0	15.55	44.50	0.349548198
12	9	22	0	0	5.55	15.90	0.349364411
10	26	0	0	0	3.55	10.20	0.348519072
10	3	18	0	0	7.55	21.70	0.348151829
19	12	3	0	0	15.55	45.00	0.345664326
10	12	17	0	0	8.00	23.50	0.340425499
12	3	18	0	0	7.55	22.20	0.340310570
9	27	0	0	0	12.00	35.70	0.336134434
19	10	12	0	0	14.00	41.70	0.335731395
3	17	0	0	0	8.00	23.90	0.334727999
7	25	0	0	0	10.00	29.90	0.334448133
10	9	17	0	0	7.55	22.70	0.332814742
19	3	0	0	0	14.00	42.10	0.332541551
12	26	0	0	0	3.55	10.70	0.332233086
19	10	9	0	0	13.55	40.90	0.331415519
19	12	9	0	0	13.55	41.40	0.327412903
10	12	3	25	0	11.00	33.70	0.326409474
12	9	17	0	0	7.55	23.20	0.325641956
10	27	0	0	0	11.55	36.00	0.320969291
10	12	18	0	0	6.00	18.90	0.317460261
12	27	0	0	0	11.55	36.50	0.316572443
19	9	0	0	0	12.00	38.50	0.311688289
3	18	0	0	0	6.00	19.30	0.310880791
9	22	0	0	0	4.00	13.00	0.307692234
10	9	18	0	0	5.55	18.10	0.306900244
10	12	9	25	0	9.00	30.10	0.299003288
12	9	18	0	0	5.55	18.60	0.298650209
19	10	0	0	0	11.55	38.80	0.297806557
27	0	0	0	0	10.00	33.60	0.297619022
9	17	0	0	0	6.00	20.30	0.295566429
19	12	0	0	0	11.55	39.30	0.294017661
10	3	25	0	0	8.55	30.80	0.277756319
19	0	0	0	0	10.00	36.40	0.274725251
12	3	25	0	0	8.55	31.30	0.273319311
10	17	0	0	0	5.55	20.60	0.269655082
10	22	0	0	0	3.55	13.30	0.267285321
10	1	0	0	0	2.55	9.60	0.266134873
10	5	0	0	0	2.55	9.60	0.266134862

TABLE B.5 (Cont'd.)

12	17	0	0	0	5.55	21.10	0.263265099
12	22	0	0	0	3.55	13.80	0.257601045
26	0	0	0	0	2.00	7.80	0.256410126
9	18	0	0	0	4.00	15.70	0.254776999
12	1	0	0	0	2.55	10.10	0.252959859
12	5	0	0	0	2.55	10.10	0.252959844
10	12	25	0	0	7.00	28.00	0.249999967
3	25	0	0	0	7.00	28.40	0.246478844
10	9	25	0	0	6.55	27.20	0.240988776
12	9	25	0	0	6.55	27.70	0.236638784
10	18	0	0	0	3.55	16.00	0.222180920
17	0	0	0	0	4.00	18.20	0.219780155
12	18	0	0	0	3.55	16.50	0.215448137
9	25	0	0	0	5.00	24.80	0.201612873
22	0	0	0	0	2.00	10.90	0.183486173
10	25	0	0	0	4.55	25.10	0.181469912
12	25	0	0	0	4.55	25.60	0.177925563
10	12	8	27	0	17.00	101.30	0.167818356
19	10	12	8	0	17.00	104.10	0.163304510
10	8	27	0	0	14.55	98.40	0.147915596
12	8	27	0	0	14.55	98.90	0.147167789
18	0	0	0	0	2.00	13.60	0.147058765
19	10	8	0	0	14.55	101.20	0.143823069
19	12	8	0	0	14.55	101.70	0.143115971
1	0	0	0	0	1.00	7.20	0.138888793
5	0	0	0	0	1.00	7.20	0.138888778
8	27	0	0	0	13.00	96.00	0.135416659
25	0	0	0	0	3.00	22.70	0.132158551
19	8	0	0	0	13.00	98.80	0.131578939
10	12	8	17	0	11.00	85.90	0.128055876
10	12	8	26	0	9.00	75.50	0.119205287
10	12	8	22	0	9.00	78.60	0.114503808
10	12	8	18	0	9.00	81.30	0.110701098
10	12	8	25	0	10.00	90.40	0.110619460
10	12	8	0	0	7.00	67.70	0.103397331
10	8	17	0	0	8.55	33.00	0.103071023
12	8	17	0	0	8.55	83.50	0.102453823
10	8	26	0	0	6.55	72.60	0.090287806
12	8	26	0	0	6.55	73.10	0.089670237
8	17	0	0	0	7.00	80.60	0.086848623
10	8	22	0	0	6.55	75.70	0.086590420
10	8	25	0	0	7.55	87.50	0.086341654
12	8	22	0	0	6.55	76.20	0.086022240
12	8	25	0	0	7.55	88.00	0.085851075
10	8	18	0	0	6.55	78.40	0.083608352
12	8	18	0	0	6.55	78.90	0.083078513
8	26	0	0	0	5.00	70.20	0.071225061
8	25	0	0	0	6.00	85.10	0.070505278
10	8	0	0	0	4.55	64.80	0.070291586
12	8	0	0	0	4.55	65.30	0.069753363
8	22	0	0	0	5.00	73.30	0.068212816
8	18	0	0	0	5.00	76.00	0.065789466
8	0	0	0	0	3.00	62.40	0.048076912

APPENDIX C: REPORT OF NEW TECHNOLOGY

The work reported here represents an extensive review and critical evaluation of existing and potential methods for the control of wheel/rail noise on urban rail rapid transit systems. Several new treatments developed by other organizations, for which patents exist or are pending, have been discussed. These include:

1. Internal ring-damped wheel, Standard Steel Division of Titanium Metals Corporation of America, Burnham, PA; pp. 18, 19, 30.
2. Hardfacing of rails, ELECTRO-THERMIT GmbH, Essen, W. Germany; pp. 68, 69, 107, 115.
3. Nitinol treaded (Tinel®) wheel, Raychem Corp., Menlo Park, CA; pp. 37, 38, 106, 112.

Two innovative treatments reviewed here have previously been identified and discussed by Bolt Beranek and Newman Inc., under U.S. DOT Contract No. DOT-TSC-644 (see P.J. Remington, M.J. Rudd, and I.L. Ver, "Wheel/Rail Noise and Vibration -- Volume II: Applications to Control of Wheel/Rail Noise," U.S. DOT Report No. UMTA-MA-06-0025-75-11, May 1975). These are the resiliently treaded wheel (pp. 34, 35, 36, 106, 118) and the low rail barrier (pp. 71, 110, 112). The conceptual drawing of the resiliently treaded wheel shown in Fig. 2.10 (p. 35) represents a new design that has potential for use in rail transit type systems.

One new treatment is described which may have potential for site-specific squeal noise reduction. That is the use of Nitinol (or other suitable metal alloy) as the tread (top) portion of the rail on curves (see pp. 68, 106). Initial evaluation of this treatment makes it appear to be noncost-effective because of the high expected costs of the metal alloy.

Finally, two concepts for noise-related monitoring systems are described: a rail roughness detection monitor (pp. 93, 94) and a vehicle noise monitor (pp. 94, 95).

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