## CHAPTER 6: COST ANALYSIS

CHAPTER **6**

# COST ANALYSIS

## 6.1 INTRODUCTION

The purpose of this chapter is twofold. First, a basic engineering economic cost calculation method to compare various noise mitigation options is described. Although simple, the basic method should cover most of the scenarios faced by the transit engineer. The cost model may be familiar to engineers and controllers normally faced with the problem of procuring equipment or developing certain programs. The approach here is to motivate transit system planners, controllers, and operators to take the "long view" with respect to noise control treatment selection. The second purpose of the chapter is to present representative cost data (1995 dollars) for various system components used to control noise. Manufacturers and contractors should be consulted for more precise figures when considering treatment alternatives, because prices for materials and labor can vary widely depending on volume, locality, and labor rates.

## 6.2 EQUIVALENT UNIFORM ANNUAL COST MODEL

An economic model is developed to provide a means for comparing the cost of noise control treatments on a uniform basis. The model incorporates initial expenditures for equipment and products, annual maintenance, operational costs, and salvage values. The methodology is implemented in the computer software package to facilitate cost comparisons. A feature of the software package is a graphic representation of the cash cost per year needed for a specific treatment.

### 6.2.1 Description of Model

The primary goal of *engineering economic analysis* is to compare the costs of various technical options, which often have widely disparate expenditure schedules and life cycles, on an "apples-to-apples" basis. To do this, the *annual cost method* (also known as the *capital recovery method)* is used to estimate the amount of money one would have to pay annually during the life of the treatment alternative if all of the necessary funds were borrowed at the beginning of the treatment program. This amount is termed the *equivalent uniform annual cost* (EUAC). The annual cost method implicitly assumes that whichever treatment alternative is chosen, it will continually be renewed up to the life of the longest option considered.

The method does not include a monetary value for the benefit realized by the reduction of noise provided by the various components under consideration, but gives the transit engineer some indication of the costs associated with various alternatives which can then be used in conjunction with other relevant factors to arrive at an informed decision.

The basic method allows for one initial expenditure, a constant yearly expenditure, three periodic expenditure schedules, three singular expenditures, and a final scrap value. The life span of the alternative must be specified, as well as information about the periodic and singular expenses. Finally, an interest rate must be estimated to account for the time value of money.

### 6.2.2 Fundamental Assumptions

The annual cost method makes many assumptions, some of which may be poor approximations of reality. Nevertheless, they are made to obtain a solution. The assumptions should be kept in mind, and their implications for the results of the analysis should be understood.

Key assumptions used in this economic cost model are as follows:

- Disregard inflation,
- Assume a constant (fixed) interest rate,
- Disregard nonquantified factors,
- Assume that funds are available, and
- Assume all funds earn interest at the effective rate until used.

Following is a brief discussion of each assumption.

#### *6.2.2.1 Disregard Inflation*

Accounting for a constant inflation factor in an economic model is undesirable because it adds complexity to the calculations *without affecting the relative results*. If constant inflation were accounted for, the monetary values at the end of the analysis would be closer to those actually realized (within the limits of the gross assumption of constant inflation), but the ranking of the various alternatives on a cost basis would be the same. As was stated previously, the goal of engineering economic analysis is to obtain such a ranking, and the monetary values obtained are only indicative of the actual costs that would be incurred.

#### *6.2.2.2 Constant Interest Rate*

Although interest rates fluctuate regularly, a constant effective rate can be reasonably well estimated over any given number of years. Therefore, this assumption, which greatly simplifies the mathematics, is reasonable as long as a good estimate is made.

> IMPORTANT NOTE REGARDING INTEREST RATE: The interest rate a bank offers comprises two components: one component accounts for the "price" paid for the convenience of borrowing the money and the other accounts for inflation. The price component is sometimes referred to as the *real* interest rate. Thus,
>
> Bank Rate = Real Rate + Inflation Rate
>
> Because this analysis disregards inflation in cost calculations, only the real interest rate should be used. For most economic conditions, the real interest rate is about 3 to 4%.

#### *6.2.2.3 Disregard Nonquantified Factors*

The economic model being developed here is a cost model, which is meant to reflect the expenditures one could expect by pursuing any of the various options analyzed. If some element or ramification of a system under consideration does not have a quantified cost associated with it, it cannot be included in the calculation. For example, even if the alternative with the longest life has the lowest EUAC, it may not be the best choice if technological innovations are expected to eclipse the usefulness of the component long before its expected life has been reached. This potentially decisive factor should be considered by the transit engineer, but there is simply no way to include it in the cost analysis being described here, unless the cost can be estimated quantitatively.

#### *6.2.2.4 Availability of Funds*

This analysis is used to determine the equivalent uniform annual cost over the life of the project. The component with the highest up-front cost might actually be the least expensive in the long run. For this reason, no technically viable solutions should be excluded from the analysis, and one should assume that sufficient funds will be available for any technically satisfactory option. If this assumption does not turn out to be correct, and alternatives with large initial expenditures are ruled out, the analysis will at least serve to gauge how economically efficient the remaining options are.

#### *6.2.2.5 Funds Earn Interest Until Used*

This assumption accounts for the fact that money which is not spent can be "put to work" earning interest payments for future cost obligations. Alternatively, interest does not have to be paid on money which is not borrowed right away. Either way, postponing expenditures will be beneficial if the amount paid down the road is less than the money saved by postponing the expense plus the interest accrued on that amount.

### 6.2.3 The Method

The EUAC model accounts for the following cash flows:

- Initial expenditure,
- Yearly expenditures (inspection, maintenance, etc.),
- Periodic expenditures (e.g., an operation which occurs biannually),
- Singular expenditures (e.g., a one-time machine rebuild), and
- Final salvage value recovery.

To calculate the EUAC of a system or component, the present value (or present worth) of each of these expenditures is calculated, the present values are summed, and the sum is amortized over the life of the item being considered. These calculations are facilitated by the use of three elementary accounting discount factors to account for the time value of money: the *future single-payment present worth*, the *uniform series present worth*, and the *capital recovery* factors. Each discount factor is described below.

#### *6.2.3.1 Future Single-Payment Present Worth*

This factor converts a future single-payment amount to its present value. The factor, designated by P/F, is calculated as

$$P/F = (1+I)^{-n}$$

where $I$ is the annual interest rate and $n$ is the number of years in the future in which the payment will be made. For example, at 5% interest the present value of a \$100 payment to be made in three years is $\$100*(1.05)^{-3} = \$86.38$.

#### *6.2.3.2 Uniform Series Present Worth*

This factor converts an annuity payment for $n$ years to the present value of those payments. The factor, designated by P/A, is calculated as

$$P/A = [(1 + I)^n - 1]/[I*(1+I)^n]$$

where $I$ is the annual interest rate and $n$ is the number of years for which the annuity is paid. For example, at 3% interest the present value of a $25 annuity for 5 years is $\$25*[(1.03)^5 - 1]/[0.03*(1.03)^5] = \$114.49$.

*6.2.3.3 Capital Recovery*

This factor converts a present value to an annuity payment over $n$ years. The factor, designated by A/P, is calculated as

$$A/P = [I*(1 + I)^n]/[(1 + I)^n - 1]$$

where $I$ is the annual interest rate and $n$ is the number of years for which the annuity is paid. For example, at 8% interest the annuity payment over 30 years for a present value of \$250,000 is $\$250{,}000*[0.08*(1.08)^{30}]/[(1.08)^{30} - 1] = \$22{,}206.86$.

### 6.2.4 A Simple Example

Consider the EUAC for two components, each of which lasts 3 years. Component A costs \$100 and requires no other expenditures, and Component B costs \$50 and requires \$19 worth of maintenance per year. Both components are worthless at the end of the 3 years, and the real interest rate is 10%

For Component A, the EUAC is simply the \$100 purchase price (already at present value) amortized over the 3 years by the capital recovery discount factor:

$$\text{EUAC(A)} = \$100*[0.10*(1.10)^3]/[(1.10)^3 - 1] = \$40.21$$

For Component B, the EUAC is the \$50 purchase price plus the present value of the three \$19 yearly expenses all amortized over the 3 years. First, calculate the present value of all the expenses:

| | | Present Value |
|---|---|---|
| Purchase Price | \$50 | = \$50.00 |
| First Year Expenses | $\$19*(1.10)^{-1}$ | = \$17.27 |
| Second Year Expenses | $\$19*(1.10)^{-2}$ | = \$15.70 |
| Third Year Expenses | $\$19*(1.10)^{-3}$ | = \$14.27 |
| | | \$97.24 |

(NOTE: The present value of the three yearly expenses, \$47.24, could also be calculated using the P/A discount factor.)

Then amortize the present value over the 3 years:

$$\begin{aligned}\text{EUAC(B)} &= \$97.24 * [0.10 * (1.10)^3]/[(1.10)^3 - 1] \\ &= \$39.10\end{aligned}$$

Thus, Component B, despite having a higher nominal cost (\$50+ 3*\$19 = \$107), is the less expensive alternative over the 3-year period because the \$50 saved in the beginning accrues enough interest to cover the three \$19 annual expenses and leave a little excess.

If this example is repeated with an interest rate of 5%, EUAC(A) = \$36.70 and EUAC(B) = \$37.40. At the lower interest rate, the \$50 saved does not earn enough interest to cover the maintenance payments; therefore, Component A, despite its higher initial cost, is the better value, illustrating the importance of estimating an accurate figure for the real interest rate. Again, the real interest rate is the difference between the total "bank" interest rate and the inflation rate.

### 6.2.5 Another Example—Rail Grinding Machine

The data for two machines (perhaps rail grinding machines) being considered for purchase are as follows:

| | Machine A | Machine B |
|---|---|---|
| Expected Life (years) | 5 | 10 |
| Initial Cost (\$) | 700,000 | 1,500,000 |
| Final Salvage Value (\$) | 50,000 | 50,000 |
| Yearly Expenses (\$) | 100,000 | 50,000 |
| Overhaul Expenses | | |
| Year | N/A | 6 |
| Parts and Labor Less Core (\$) | N/A | 180,000 |

Assume an effective annual real interest rate of 3%.

The present value of the various expenditures for Machine A is

| | | Present Value |
|---|---|---|
| Purchase Price | \$700,000 | = \$700,000 |
| Salvage Value | $\$50{,}000*(1.03)^{-5}$ | = –\$43,100 |
| Yearly Expenses | $\$100{,}000*[(1.03)^5 - 1]/[0.03*(1.03)^5]$ | = \$458,000 |
| | | \$1,114,900 |

This figure, amortized over the 5-year life of the machine, is the EUAC:

$$\begin{aligned}\text{EUAC(A)} &= \$1{,}114{,}900 * [0.03 * (1.03)^5]/[(1.03)^5 - 1] \\ &= \$243{,}400\end{aligned}$$

The present value of the various expenditures for Machine B is

| | | Present Value |
|---|---|---|
| Purchase Price | \$1,500,000 | = \$1,500,000 |
| Salvage Value | $\$50{,}000*(1.03)^{-10}$ | = −\$37,200 |
| Yearly Expenses | $\$100{,}000*[(1.03)^{10} - 1]/[0.03*(1.03)^{10}]$ | = \$426,500 |
| Overhaul Expenses | $\$180{,}000*(1.03)^{-6}$ | = \$150,700 |
| | | \$2,040,000 |

Amortizing over the 10-year life of the machine, the EUAC is

$$\begin{aligned} \text{EUAC(B)} &= \$2{,}040{,}000 * [0.03 * (1.03)^{10} - 1] / [(1.03)^{10} - 1] \\ &= \$239{,}200 \end{aligned}$$

At 3% interest rate, Machine B is the more economically efficient option because the lower yearly costs over the life of the machine more than compensate for the higher initial expenditure and the overhaul expenses. At a higher interest rate, say 5%, this would not be true because the savings that come with purchasing the initially less expensive Machine A would accrue enough interest over time to make the higher yearly expenses worthwhile.

Note that although the life of Machine A is only half that of Machine B, a second machine is required at year 6 to continue grinding to year 10. Even with the need to purchase a new machine, the EUAC provides a direct comparison of annual costs for the machines, because the future costs for the second machine would produce the same current EUAC, assuming that all costs change according to the real interest rate. Note that including inflation may change this conclusion.

The selection of the machine may be influenced by nonquantified factors: the more expensive machine may be able grind rail to closer tolerances and provide certain computer-controlled grinding features that the less expensive machine may not. At the lower interest rate, the decision to select the more expensive machine is further motivated by the nonquantified features. At the higher interest rate, the higher EUAC for the more sophisticated machine is balanced to some extent by the nonquantified features, which must be factored into the overall decision-making process. Again, cost should not be the only criterion for selection.

## 6.3 REPRESENTATIVE COSTS

This section provides representative cost data for the noise control measures being discussed in this manual. As was discussed previously, projected cost calculations require the use of a constant real interest rate. Five percent is assumed for all of the calculations in the estimates that follow, unless otherwise noted.

As indicated, the costs are only representative. Noise control treatments and equipment are rarely off-the-shelf items, and there is some flexibility in costs depending on availability, quantities ordered, level of technological innovation, and patents. The user should therefore contact manufacturers directly to obtain up-to-date costs for initial procurement, installation, maintenance, materials, and salvage. Further, arriving at direct cost comparisons requires knowledge of the number of vehicles using a particular line and the length of the line, two factors which may vary substantially from property to property. For example, BART has approximately 140 mi of track and on the order of 500 vehicles operating in 10-car consists. A light rail system may have as little as 20 mi of track and 20 or 30 vehicles operating in 1- to 3-car consists. Thus, there are large differences in scale which must be considered. The tables provided here attempt to provide some uniformity in presentation, but direct comparison between onboard and trackwork treatments is very system-specific.

Ancillary cost savings resulting from application of noise control treatments are not directly considered. For example, re-tiring of resilient wheels need not require replacement of the aluminum center, thus providing a cost savings at tread renewal time. There may be reductions in truck vibration and shock loading resulting from use of resilient wheels, which may lead to savings in traction equipment maintenance and reduction of failure rates, but these cost savings are nonquantifiable at present. Rail grinding may actually extend rail life by reducing shock and vibration and by optimizing metal removal rates. These cost savings, if quantifiable, can be worked into the model by simple subtraction from the EUAC or present values.

The costs are obtained in part from the literature, with an adjustment for producer price index changes; from the survey; and from discussion with transit engineers. Costs were considered in 1974 with respect to the MBTA Pilot Study (*1*); again by Kurzweil et al. in 1981 (*2*); by Saurenman et al. in a study involving the SEPTA system (*3, 4*); and in a study of damped wheels at MTA NYCT (*5*).

### 6.3.1 Onboard Treatments

Onboard treatments include special wheels, undercar absorption, and other systems and components attached to the transit vehicle. Onboard treatments also include the tuning and maintenance of these components, such as by wheel truing.

#### *6.3.1.1 Wheel Truing Machines*

The cost of a wheel truing program depends on the number of vehicles serviced and the interval between truing. The labor costs involved also depend greatly on the type of lathe used (above-floor versus below-floor), although the initial cost of the two types is comparable. Labor for above-floor lathes can be 3 to 7 times that for below-floor lathes. Another type of wheel truing machine is the below-floor milling machine, which is less expensive than the lathe and is considered by some shop personnel to be less accurate than the lathe. The use of below-floor milling machines is also considered by some transit personnel to require greater metal removal when trimming flanges than the use of the lathe-type machine. The capital cost necessary for a wheel truing program is approximately $700,000 to $1,500,000, with the lathes at the high end of the range. The lathes can be expected to last for at least 10 years and, assuming a 10% salvage rate, have a salvage value of $100,000. For a system with 700 vehicles, the yearly cost for the program would be $300,000 to $400,000. As shown in Table 6-1, the EUAC for this program would be approximately $472,000.

*6.3.1.2 Dry-Stick Lubrication*

Dry-stick friction modifiers and flange lubricators are gaining popularity at light rail transit systems for reducing rail wear and stick-slip. The costs for dry-stick lubrication vary. Because this treatment is still nascent, new products may be expected to come on the market and the cost of existing products might be expected to decline with increasing usage in the years to come. For these reasons, it is especially important to contact suppliers directly for accurate price quotes and product information.

Below is some cost information (1995) provided by several transit agencies and product suppliers. Metro-Dade County reports a capital cost of $80,000 and maintenance cost of $15,000 per vehicle per year. The EUAC for this system would be $25,000 if it lasted 10 years and $21,000 if it lasted 20 years. WMATA indicated that onboard flange and tread lubrication was estimated to cost about $500 per truck per month, or about $12,000 per vehicle per year. This agency did not relay any capital costs. One supplier indicates that the cost of combined flange lubricant and contact friction modifier includes a one-time cost of $800 to $1,000 for brackets, plus 1 hr for installation time and a product cost of about $1,500 per vehicle per year, assuming four wheels lubricated per vehicle. Based on a survey of various transit systems, the overall costs are $1,400 per vehicle per year for friction modifier tread lubricant and $1,200 per vehicle per year for flange lubricant. Thus, there appear to be wide disparities between costs of various onboard lubrication products. Costs should be reviewed with the manufacturer and verified.

*6.3.1.3 Resilient Wheels*

Resilient wheels include PCC, Acousta-Flex, Penn Bochum, and SAB. Costs range from $1,600 to $4,800 per wheel, or roughly $13,000 to $38,000 per vehicle. In general, there should be no extra yearly costs incurred with the use of resilient wheels relative to solid steel wheels. Many resilient wheels have replaceable treads and can be overhauled so that the tire centers can remain in service for decades. Costs for replacement treads are on the order of $1,000 per tread plus mounting labor. The EUAC for a set of eight resilient wheels costing $3,600 per piece and lasting 30 years, with two rebuilds, is approximately $2,285 per vehicle per year, assuming the present value of a rebuild is $1,000 per wheel. Assuming that a vehicle travels about 200,000 mi per year,

**TABLE 6-1 COST DATA FOR ONBOARD TREATMENTS**

| Treatment | Capital Cost - $ - | Maintenance Cost - $/yr - | Material Cost/yr - $/yr - | Salvage Value - $ | Useful Life - Yrs |
|---|---|---|---|---|---|
| Slip/Slide Control | 5,000 - 10,000/veh | 200 | 200 | 0 | 30 |
| Wheel truing | 1 mil | 200 | 500 | 100,000 | 10 |
| Vehicle skirts | 5,500/veh | 0 | 0 | 0 | 30 |
| Undercar absorption | 3,500 - 5,300/veh | 0 | 0 | 0 | 30 |
| Door seals | 1,000/veh | 100 | 0 | 0 | 10 |
| Dry Stick Friction Modifier | 500/veh[1] | 300 | 1,400 | 0 | 30 |
| Dry Stick Flange Lubricant | 500/veh[1] | 300 | 1,200 | 0 | 30 |
| Resilient wheels | 30,000/veh[2] | 1,000[3] | 1,000[3] | 100 | 30 |
| Visco-elastic damped wheels | 6,400 - 20,000/veh[2] | 0 | 0 | 0 | 30 |
| Vibration absorbers | 4,000 - 5,600/veh[2] | 0 | NA | 0 | 30 |
| Ring dampers | 560/veh[2] | 0 | NA | 10 | 10 |
| Steerable trucks | 8,000/veh | unknown | 0 | 0 | 30 |

Notes:
1 Cost based on installation of both flange lubrication and tread friction modifier stick holders at 4 wheels
2 Assumes 8 wheels per vehicle. Costs for light rail articulated vehicles will be higher
3 Based on $1,000 per replacement tread and $1,000 for labor.
NA Not available

the cost per mile would be on the order of a penny per mile, which is comparable with the cost of automobile tires on a per-mile basis. (The above numbers are for illustrative purposes only and may not correspond to actual costs.)

#### *6.3.1.4 Wheel Vibration Absorbers*

Wheel vibration absorbers may cost between about $500 and $700 per wheel. There is no significant maintenance cost, though there may be a cost associated with removal of the absorbers from condemned wheels and remounting on new wheels. Salvage value may be considered nil. There may be a nonquantified cost savings in the way of reduced rail corrugation, though this has not been verified. The vibration absorbers can probably be reused, though this should be checked with the manufacturer. Replacement wheels should be machined and tapped to allow mounting of the dampers without the need for machining at the shop. Replacement wheels should be the same design as the original wheels, because the wheel vibration absorbers may be tuned to the specific modal resonances of the wheel. This may place a restriction on available wheel manufacturers, which might add to the cost of the wheels.

#### *6.3.1.5 Visco-Elastic Dampers*

Visco-elastic damped wheels include those which have visco-elastic ring dampers or constrained layer damping. Costs range from $500 to $2,000 per wheel, or $8,000 to $16,000 per vehicle. There are no maintenance costs, though there may be some nonquantified cost impact on wheel inspection or maintenance. Some researchers have assumed that the dampers may be reused once when wheels are replaced, though the cost to transfer the dampers is not known. The cost of transferring the dampers may be mitigated to some extent by having the manufacturer supply the wheels predrilled and machined, ready to accept the dampers. This would be facilitated if the dampers were originally supplied by the wheel manufacturer as part of a new wheel procurement. Salvage value of the dampers should be assumed nil, though the damper manufacturer might be interested in the metal or other reusable cores.

#### *6.3.1.6 Ring Damped Wheels*

Costs for steel ring dampers are on the order of $30 to $50 per wheel, including machining, when supplied as part of new wheel procurements. This damping treatment is perhaps one of the most economical and requires no maintenance, and dampers are relatively easy to replace, though replacement does not appear to be necessary. Ring dampers might also be reusable with suitably machined replacement wheels, though this should be checked. Salvage value is essentially the salvage value of carbon steel, unless they are reusable by other transit properties.

#### *6.3.1.7 Undercar Absorption*

Although undercar absorption yields only modest reductions in noise levels, it does have two strong advantages: it is relatively inexpensive, and its benefits are realized throughout the system. For simple application of glass fiberboard, the cost is $10 to $15 per square foot, or $3,500 to $5,300 per vehicle for 50% coverage. For 75% coverage the cost would be $5,300 to $7,900 per vehicle. If a high degree of shaping and/or protection were required, these estimates could increase substantially. There is no specific maintenance requirement, though there may be an impact on vehicle maintenance if the treatment is not placed judiciously.

Clear areas under the vehicle, over the trucks, are good locations. Areas over HVAC or other auxiliary equipment would probably not be accessible to treatment. Treatment, which can applied to the interior surface of fixed or demountable skirts, would not impact vehicle maintenance. Plumbing and wiring should not be obscured, though these can be relocated, which would increase the cost of the treatment. There is no salvage value. Useful life should be equivalent to that of the vehicle, here assumed to be 30 years.

#### *6.3.1.8 Skirts*

In 1980 the cost of retrofitting a transit vehicle with two full-length skirts and undercar absorption was about $12,000. Assuming a doubling of producer prices between 1980 and 1995, current costs would be roughly $24,000. However, skirts and absorption would likely be part of an overall vehicle procurement and thus might be supplied at considerably lower costs than based on 1980 estimates. Vehicle skirts are being supplied with 27 articulated Tri-Met vehicles for a cost of about $150,000, or about $5,550 per vehicle or $1,850 per truck. There are no maintenance costs associated with skirts, though there may be an increased cost associated with removal of the skirt for maintenance of the vehicle. There is no salvage value, other than that which might go with salvage of the vehicle.

#### *6.3.1.9 Slip-Slide Control*

Cost figures for slip-slide control are not easily obtained, primarily because slip-slide controls are standard equipment on many vehicles. GO Transit (Toronto) indicated that the cost of slip-slide control is $2,500 per vehicle per year. Other estimates received by an older eastern rail transit system ranged from about $5,000 to $10,000 per vehicle for retrofit. Another East Coast transit system has received estimates of about $5,000 per vehicle for retrofitting existing transit vehicles. Maintenance costs were not obtained. Salvage value should be assumed nil or, at most, scrap. Cylinders and actuators, however, may have some core value. Advances in technology may eliminate any salvage value for the electronic control systems. The useful life is assumed to be that of the vehicle, here assumed to be 30 years. However, rebuilding of actuators and cylinders on a scheduled basis may be anticipated, though much of this cost may be necessary regardless of whether slip-slide control is employed.

### 6.3.2 Trackwork Treatments

Trackwork treatments include rails and rail support components and rail maintenance procedures such as rail grinding and joint tightening. EUAC estimates for all of the trackwork treatments discussed below are presented in Table 6-2.

#### *6.3.2.1 Rail Grinding (In-House)*

The cost of any rail grinding program depends on the number of track-miles to be maintained and the interval at which grinding is performed. An in-house grinding program can be economically efficient if sufficient grinding is done to warrant the capital expenditure necessary to instigate it. Most small- and intermediate-sized transit systems can operate a successful program with one grinding machine with between 8 and 20 grinding stones. In cases where grinding time is limited by nonrevenue periods, and lengthy grinding on a frequent basis is necessary to control rail corrugation, more than one grinder may be necessary.

The capital cost of a grinding machine depends on the number of grinding stones, track gauge, and support equipment (such as fire suppression systems). However, doubling the number of grinding stones cuts the grinding time in half, so that substantial labor cost savings might be obtained, and grinding during limited time periods will be facilitated by initially purchasing a 16- or 24-stone grinder rather than an 8-stone grinder. The cost of the equipment can also depend on special considerations such as clearance problems on some light rail systems.

Modern rail grinding machines have life expectancies of 10 to 15 years. Variable costs for rail grinding include labor (five-person crew), replacement of worn grinding stone sets, maintenance, and fuel. Initial capital costs for an in-house grinding program range from $700,000 to $1,300,000. Yearly operational costs vary between $150,000 and $300,000. For a $1,000,000 program with $200,000 in operational costs and a 5% salvage rate, the EUAC would be $326,000 for yearly grinding. If grinding were done every 3 years, the EUAC would be $189,000.

#### *6.3.2.2 Rail Grinding (Contracted)*

For those transit systems which do not grind often enough to warrant an in-house grinding program, contract grinding is an option. Estimates for contract grinding vary widely, from $1,000 to $7,000 per track-mile depending on the amount of track to be ground, grinding time availability, condition of the track and rail, and drayage costs. For a system with 140 track-miles, the average cost of grinding on a yearly basis would be $560,000 ($4,000 per track-mile). This is considerably more than the cost of running an in-house grinding program. If, however, grinding were to be done only once every 3 years, the EUAC for contract grinding would drop to approximately $180,000, thus making contract grinding a feasible alternative to an in-house program. However, grinding intervals should be dictated by rail wear rates, because prolonging grinding may lead to increased rail material removal and excessive replacement cost to maintain low noise condition.

**TABLE 6-2 TRACKWORK TREATMENT COSTS**

| Treatment | Initial Cost $ | Yearly Maintenance Cost $ | Yearly Material Cost | Salvage Value $ | Useful Life - yrs - |
|---|---|---|---|---|---|
| Rail Grinding (in house) | 0.7 mil to 1.3 mil | 0.27/tr-ft[1] | NA | 0.05 | 10 to 15 |
| Rail Grinding (contracted) | NA | NA | NA | NA | NA |
| Defect Welding & Grinding | NA | NA | NA | NA | NA |
| Joint Welding | 4.1 mil | 85/tr-ft | NA | 0.1 mil | 10 |
| Joint Maintenance | 0 | 0.35 to 0.50/tr-ft | NA | NA | NA |
| Lubrication | $10,000/tr-curve | 50,000/curve | $200/tr-curve | 0 | 15 |
| Hard facing | 30 | Not available | Not available | 0 | 2 |
| Rail Vibration Absorbers | 20 to 40/tr-ft | 0 | 0 | 2 | 30 |
| Rail Vibration Dampers | 20/tr-ft | 0 | 0 | 0 | 30 |
| Trackbed Absorption | 100/tr-ft | 0 | 0 | 0 | 20 |

Notes: 1 Based on 1 year grinding interval for 140 miles of track at a cost of $200,000 per year. Actual ground track per year will be considerably less.

NA Not available

*6.3.2.3 Joint Tightening and Maintenance*

Joint tightening to minimize gaps is expected to cost about $7 per joint per year in addition to the cost of normal maintenance. There would be reduced rail joint impact and wear, which may reduce batter and extend track life.

*6.3.2.4 Joint Welding*

The costs of rail joint welding and finishing are expected to be about $450,000 per track-mile, or $85 per track-foot. For 39-ft rail sections, the cost would be about $1,660 per rail joint. A cost of $4,100,000 is estimated for the rail welding car, though this would likely be a contractor operation. More exact cost data should be obtained from rail contractors.

*6.3.2.5 Hardfacing*

The cost of hardfacing, whereby a very hard material is let into the rail head running surface to control rail wear and corrugation, ranges from $15 to $40 per rail-foot, or $158,400 to $422,400 per track-mile. The cost depends on how much rail is hardfaced. To date, most hardfacing applications within the United States and Canada were for rail wear control at curves and not for corrugation reduction on tangent track. Assuming 20% of a 140-track-mile system were hardfaced and that the hardfacing would last 10 years, the EUAC for this would be approximately $1,165,000.

*6.3.2.6 Trackbed Sound Absorption*

Trackbed sound absorption may be an effective noise control measure for limited spacial applications such as stations and subways. Absorption in the form of 3-lb per cubic foot glass fiberboard encased in 3-mil-thick Tedlar, protected by perforated glass fiber or a powder-coated metal sheet, could be installed for a minimum cost of about $10 per square foot. At this price, it would cost approximately $300,000 to treat an underground station with a 750-ft-long platform (assuming treatment would extend half the length of the platform into the tunnel at each end of the station at each track and that the treatment width is about 10 ft wide). Note that ballast provides substantial sound absorption which may be very important in subways.

*6.3.2.7 Defect Welding and Grinding*

No data have been collected from transit systems on defect welding and grinding. Assuming a crew of four are required (flagger, welder, grinder, and supervisor), the cost of defect welding and grinding should about $150 per hour. Assuming that the time required to weld and grind a defect is about $^1/_2$ to 1 hour, the cost per defect would be about $75 to $150 per defect. If numerous defects exist, the cost should be balanced against rail replacement and or extensive rail grinding to remove defects. Equipment costs have not been determined.

*6.3.2.8 Rail Vibration Absorbers*

No cost data have been obtained for rail vibration absorbers. However, considering their size and design, costs could be on the order of $50 to $100 per absorber. Assuming that one absorber is installed every 5 ft (every other direct fixation fastener), the cost would be about $20 to $40 per track-foot. Maintenance should be negligible, though rail replacement would be complicated by the need to remove the absorbers. Tightening of absorber mountings might be necessary, analogous to normal joint maintenance. The absorbers should be reusable on new rail. Salvage value should be equivalent to that for scrap, unless they can be sold to another property. Useful life should be 30 years or more.

*6.3.2.9 Rail Vibration Dampers*

No cost data have been obtained for rail vibration dampers. However, their structure suggests a cost comparable to that of an inexpensive resilient direct fixation fastener consisting of a rolled plate and elastomer pad. Assuming that one damper is installed every 2.5 ft, or between successive direct fixation fasteners, the cost could be on the order of $40 per track-foot. Salvage value at the end of its useful life, assumed to be 30 years, is likely equivalent to that for scrap steel. Scrap value of $0.50 per damper is assumed, or about $0.40 per track-foot.

**6.3.3 Wayside Treatments**

Wayside treatment costs are summarized in Table 6-3. Some of the wayside noise control options have a EUAC on the order of $10 per foot of track, while rail grinding may be on the order of $1 per foot, a considerable cost savings in favor of trackwork maintenance.

*6.3.3. Station Sound Absorption Treatment*

The cost for station sound absorption treatment is typically about $10 per square foot, based on a design consisting of 1 to 2 in. of glass fiber sound-absorbing board with a perforated cover. However, special architectural designs, lay-in acoustical tile ceilings, etc., may be more costly. There is little maintenance cost, because the installations are considered permanent. There may be replacement costs as units become damaged as a result of other maintenance activities. There is no salvage value. Underplatform treatment will typically cost about $10 per square foot for glass fiber sound absorbing board.

**TABLE 6-3 WAYSIDE TREATMENT COSTS**

| Treatment | Initial Cost - $ - | Yearly Maintenance Cost - $/yr - | Yearly Material Cost - $/yr - | Salvage Value - $ - | Useful Life - yr - |
|---|---|---|---|---|---|
| Subway wall treatment | 250/tr-ft [1] | 0 | 0 | 0 | 30 |
| Station treatment | 7 to 10/sq ft | 0 | 0 | 0 | 30 |
| Fan and Vent shaft treatment | 10/sq ft | 0 | 0 | 0 | 30 |
| Sound barrier walls | 200/tr-ft [3] | 0 | 0 | 1/Sq ft | 30 |
| Berms | 200/tr-ft [3] | NA | 0 | 1/Sq ft | 30 |
| Receiver treatments | 200 to 800/tr-ft [2] | NA | 0 | 0 | 30 |

Notes: 1 Based on upper half of 17 foot diameter circular tunnel treated at $10/sq.ft.
2 Assuming one receiver every 25 feet (densely populated) at $5,000 to $20,000 per receiver.
NA Not available

*6.3.3.2 Subway Wall and Ventilation Shaft Treatments*

Subway wall treatment consisting of spray-on cementitious sound absorption typically costs on the order of $7 to $10 per square foot, installed. There are no maintenance costs or salvage value. Useful life should be 20 to 30 years.

*6.3.3.3 Sound Barriers*

Costs for sound barriers are on the order of $15 to $20 per square foot and vary according to local labor and material costs. Barrier heights range from 6 to 12 ft depending on proximity to the track. Per 100 ft of a 9-ft-high wall, the construction cost would be around $16,000. Such a wall can be expected to last 30 years or more; therefore, the EUAC per foot of wall is approximately $10.

*6.3.3.4 Receiver Treatments*

Receiver treatments include enhancing the building facade and provision of forced air ventilation or air conditioning. There may also be instances in which interior noise control is advisable. The costs associated with receiver treatments range from about $5,000 per dwelling to $20,000 per dwelling, depending on whether forced ventilation is required, the number of windows treated, local codes, labor rates, and material costs. Also, there may be unknown costs related to pest damage and operating costs of forced air ventilation. For these reasons, receiver treatment should be the last treatment to consider.

## 6.4 REFERENCES


1. Kurzweil, L.G., R. Lotz, and E.G. Apgar, *Noise Assessment and Abatement in Rapid Transit Systems* (Report on the MBTA Pilot Study), Transportation Systems Center, Urban Mass Transportation Administration, U.S. Department of Transportation, UMTA-MA–06–0025–74–8 (September 1974), pp. 3–10.
2. Kurzweil, L. G., and L. E. Wittig, *Wheel-Rail Noise Control–A Critical Evaluation* (Interim Report), prepared by Bolt Beranek & Newman, Inc., for Transportation Systems Center, Urban Mass Transportation Administration, U.S. Dept. of Transportation, UMTA-MA–06–0099–81–1 (January 1981), pp. A–2.
3. Saurenman, H. J., R. L. Shipley, and G. P. Wilson, *In-Service Performance and Costs of Methods to Control Urban Rail System Noise* (Final Report), prepared by Wilson, Ihrig & Associates, Inc., and DeLeuw Cather & Co. for Transportation Systems Center, Urban Mass Transportation Administration, U.S. Dept. of Transportation, UMTA-MA–06–0099–80–1 (December 1979), pp. 6–1 to 6–8.
4. Shipley, R. L., and H. J. Saurenman, *In-Service Performance and Costs of Methods to Control Urban Rail System Noise* (Initial Test Series Report), prepared by Wilson, Ihrig & Associates, Inc., and DeLeuw Cather & Co. for Transportation Systems Center, Urban Mass Transportation Administration, U.S. Department of Transportation, UMTA-MA–06–0025–78–7 (August 1978).
5. Manning, J. E., *In-Service Evaluation of Four Types of Damped Subway Car Wheels* (Unpublished Report), Cambridge Collaborative, March 22, 1985.

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# CHAPTER 7 ONBOARD TREATMENTS

CHAPTER 7

# ONBOARD TREATMENTS

## 7.1 INTRODUCTION

This chapter discusses various vehicle treatments for controlling wheel/rail noise. Much of the information is based on work conducted by UMTA (now FTA) during the late 1970s and early 1980s, a period that may be termed the “golden age” of rail transit noise control research. This early work concerned primarily wheel truing, slip-slide control, resilient wheels, and damped wheels. Vehicle noise control has advanced marginally since the 1970s, though noise control investigations have been continued by various operating agencies, and papers have appeared in the literature from time to time.

Treatments or provisions which may now be considered standard at modern rail transit systems include wheel truing and provision of slip-slide control systems, which, in combination with effective rail grinding, result in low levels of wheel/rail rolling noise on tangent and moderately curved track. With these basic noise control provisions, wheel/rail rolling noise is normally not a significant environmental problem. There remains, however, wheel squeal, for which no entirely satisfactory solution has yet been developed. Resilient wheels, damped wheels, and flange lubrication have been developed to address wheel squeal, all of which provide some degree of control.

The car body design, though not normally considered with respect to wheel/rail wayside noise (*1*), is very important in controlling vehicle interior noise, especially in subways, where wheel squeal and howl caused by stick-slip vibration of the wheels and rail corrugation are capable of reaching 100 dBA within the vehicle. Vehicle skirts recently have been provided with some new transit vehicles, such as at the Denver system, though there is little other experience with this type of treatment, and the degree of noise reduction is limited to a few decibels.

Recently, researchers experimented with active noise control to reduce car interior noise. The success and practicality of active noise control remains to be seen, given the complexity and distributed nature of wheel/rail interaction. Nevertheless, active noise control treatments bear watching. For example, there are certain piezo-electric damping treatments that might be useful in controlling wheel squeal. This manual describes the characteristics and performance of various treatments that may be applied to a transit vehicle, whether standard or not, relying on published reports, general literature, and in-house data collected by the authors.

## 7.2 WHEEL TRUING

Wheel truing is the front-line defense against wheel/rail noise. Without an effective wheel truing program, transit system efforts at noise control are severely hampered, and substantial additional cost beyond that which might reasonably be expected for a well-maintained transit system would be required. Without wheel truing, rolling noise levels are roughly 5 to 10 dB above normal rolling noise levels of well-trued wheels and smooth ground rail. Additional noise control provisions such as sound barriers and receiver treatments might be needed if wheel truing (and rail grinding) are not performed. Higher sound barriers and more effective sound insulation of dwellings might be required at systems with poorly maintained wheels than at systems that are well maintained. Wheel truing has systemwide effectiveness and thus may provide greater benefit per dollar than a fixed noise control treatment. There are also ancillary benefits to wheel truing which should not be ignored, such as control of ground-borne noise and vibration, improved truck dynamics, and reduced wear.

### 7.2.1 Wheel Truing Machines and Manufacturers

There are three types of commercially available wheel truing machines. One type is a lathe which cuts material from the wheel tread with a stationary cutting bar as the wheel rotates about its axis, represented by the Hegenscheidt wheel truing lathes. The second type of wheel truing machine is based on a milling machine concept, which removes tread material with a rotating cutter head held in a stationary position as the wheel rotates incrementally. An example is the Simpson-Stanray milling machine, an underfloor milling machine which does not require removal of the wheel or axle set for truing. A third type is the belt grinder, which has been used by TTC. BART experimented with the belt grinder during or before startup, but was unable to make it work successfully. Noise reduction performance data for belt grinders were not obtained for this study and are not considered further.

Wheel truing accuracy has a direct bearing on wheel/rail noise control, for two reasons. The first is that uneven cutting will increase wheel/rail noise and groundborne noise and vibration. The second is that wheels not trued to the same dimensions will be subject to excessive slip and wear, thus increasing wheel roughness and noise. This is particularly important with respect to monomotor trucks, where all four wheel must be trued to exacting tolerances.

The lathe is considered by some transit vehicle maintenance engineers to be the most accurate of the wheel truing machines. However, the milling machine is usually less expensive than the lathe and is reasonably efficient to operate. A discussion with BART engineers indicated that truing with the lathe type of truing machine requires less metal removal than truing with the milling type of machine when trimming flange throats.

### 7.2.2 Noise Reduction Effectiveness

Wheel truing tests at SEPTA indicated measurable and consistent reductions of noise on both tangent and curved track. Wheels trued with an under-the-floor milling machine were 0 to 2 dBA noisier than new wheels trued with a lathe, considered at the time to be the result of the cutter marks left by the cutting bar. The effect of the cutting marks on noise levels lessened after a few days of running. The cutting marks left by a milling-machine-type truer are on the order of a fraction of $^1/_8$ in. in dimension, so that the frequency of any noise associated with the cutting marks should be in excess of 5,000 Hz at normal vehicle speeds. Further, more recent measurements conducted at BART indicate that wheels trued with a milling-machine-type truer are no noisier than wheels trued with BART's wheel truing lathe.

Tri-Met employs a Hegenscheidt wheel truing lathe to true each wheel on a semiannual basis. Tri-Met uses Bochum 54 resilient wheels on Bombardier vehicles. Noise levels are maintained at about 80 dBA at 50 ft from tangent ballast-and-tie track for 55-mph two-car trains when the rail is in good condition, without corrugation.

On rough rail, the full noise reduction benefits of wheel truing are not fully realized, because the roughness of the rail dominates the noise generation process. However, wheel truing will still produce noise reductions on unground rail that is in good condition, free of corrugation and other surface defects. This is especially true if the untrued wheels have wheel flats and other surface defects.

The SEPTA tests at curved track showed that wheel squeal levels with new and trued wheels are essentially equivalent, suggesting that wheel truing as performed by SEPTA has little effect on wheel squeal.

Maintaining a wheel truing schedule sufficient to control wheel roughness and reduce wheel concavity, or false flanging, and operating it with a well-trained and conscientious machinist is likely to be more important with respect to noise control than the actual type of wheel truing machine, though one should consider metal removal rates for each type of machine and the type of truing required. Systems which have numerous curves may experience a greater degree of flange wear as opposed to tread wear and thus require trimming of the flange on a relatively frequent basis. In this case, the lathe type of truer might be the most economical if it removes less material than the milling-machine-type truer. Manufacturers should be questioned closely on the characteristics of their types of machines before making a final selection. Larger systems may wish to consider having both types of truing machines.

### 7.2.3 Wheel Tread Profiles

The effect of wheel tread profile on wheel rail noise is not clearly known. The principal concern in selecting a tread profile should be minimizing rail corrugation rates, optimizing the contact patch geometry and stiffness, obtaining acceptable ride quality, and controlling wheel and rail wear. Increasing the wheel taper is usually associated with an increase in possibly undesirable hunting of the truck. (BART uses a cylindrical wheel profile to effectively eliminate hunting.) The wheel tread and rail head profile control the contact patch size and shape, which in turn have an effect on corrugation growth rates, fatigue, and possibly spin-slip or roll-slip motion of the wheel. For example, squaring up the contact patch so that the lateral dimension is comparable with the longitudinal dimension has been claimed to reduce corrugation growth rates at the Vancouver Skytrain by reducing spin-slip corrugation (*2*). The Skytrain system employs a UTDC steerable truck with small diameter conical wheel treads. The conical taper has been conjectured to excite spin-slip motion in the presence of high wheel/rail conformity, and reduction of conformity by rail grinding and wheel truing appears to be beneficial.

Contact width is believed to influence rolling noise at tangent track, regardless of corrugation, as discussed in Chapter 10. One school suggests that wide contact widths induce spin-creep noise, while another school indicates that wide contact widths tend to average out rail roughness across the rail head.

Measurements at BART suggest that a narrow contact width is most desirable. The wayside noise produced by a single BART vehicle traveling at 80 mph on tangent ballast-and-tie track with smooth ground rail is about 80 dBA at 50 ft, making the BART vehicle one of the quietest. In fact, the wheel/rail noise component is comparable to noise from other sources such as the traction motors, cooling fans, and aerodynamic sources under the car. This conclusion is based on data taken at the BART Hayward test track, where the rail head was ground to produce a contact patch width of about $^5/_8$ in., and measurements indicated that the rail head ball radii were 8 in. and 11 in. The narrow contact width and the cylindrical tread profile used by BART limit spin-torques, thus limiting spin-slip. The small contact patch area also reduces the contact patch stiffness, which further reduces noise above the contact resonance frequency. Tread wear will produce a certain degree of concavity in the tread profile which will increase wheel/rail conformity, as is readily observable at BART mainline track. Maintaining a narrow contact width may not be practical without frequent wheel truing.

Conversely, theoretical analyses indicate that the roughness averaged over the rail head with wide conformal contact is lower than with a narrow contact width (*3*). The reduction of average roughness is of greater significance with respect to reducing rolling noise than reducing contact stiffness by reducing the contact width, as described in Chapter 4. Wear and corrugation rate reduction should be overriding concerns in selecting contact patch size and shape, because both of these adversely affect wayside noise levels.

The contact strip should be centered over the rails' cross-sectional center of gravity to minimize vibration couples acting on the rail. This would reduce the tendency of the rail to undergo bending of the rail web and thus reduce noise associated with this mode of vibration. However, at least one system (Vancouver Skytrain) chooses to vary the contact strip position to reduce rutting of the wheel tread. Using a cylindrical wheel with a canted rail, such as used at BART, tends to shift the contact patch to the field side of the rail head, thus inducing lateral bending vibration of the rail web.

Wheel squeal at large radius curves can be controlled to some extent with conical wheel treads and longitudinal compliance in the primary suspension. During curving, a rolling radius differential may be developed with conical tapers and may allow the high rail wheel to travel faster than the low rail wheel, thus reducing or eliminating longitudinal creep. However, wheel squeal is caused primarily by lateral slip, in turn caused by crabbing of the axle set, as discussed in Chapter 4. To control lateral slip, some longitudinal compliance is required in the journal suspension to allow the axles to align themselves with the curve radius and reduce lateral slip. For radii greater than about 700 to 800 ft, wheel squeal from typical transit vehicles is not expected, regardless of tread profile, due to the nature of the friction-creep curve. Regardless of curve radius, systems with cylindrical wheel treads may experience greater incidence of wheel squeal and roll-slip-induced rolling noise than systems with tapered wheel treads.

### 7.2.4 Cost-Benefit Considerations

Wheel truing results in greater wheel tread life, lower truck shock and vibration, and lower wheel/rail noise. This translates directly into longer life cycles for wheels, reduced maintenance of trucks and truck-mounted equipment, reduced ballast pulverization, reduced maintenance of trackwork, and lower noise mitigation costs. The benefits of wheel truing are supported by the fact that almost every major rail transit operator has an effective wheel truing program. Costs for wheel truing machines are discussed in Chapter 6.

## 7.3 BRAKING SYSTEMS

Substantial noise level differences exist between vehicles with tread-braked systems and disc-braked systems. Further, slip-slide control systems provide an effective tool for maintaining wheel condition, thus minimizing noise and maintenance costs.

Noise levels for tread-braked systems are reported to be about 10 dB higher than for disc-braked systems, due to wear of the tread caused by brake shoes. Composition tread brakes roughen the wheel tread less than cast iron tread brakes and thus produce about 3 to 5 dBA lower noise levels than cast iron tread brake systems (*4*, *5*).

Recent data collected for intercity trains in Europe indicate a direct correlation between wheel roughness and noise measured at 0.5 and 1 m from the near rail at about rail height. Some of the results of 1/3-octave band data are presented in Figure 7-1 for 89-mph trains measured after varying periods of wear. The data are not entirely representative of wayside noise, because efforts were made to exclude rail radiated noise. The data indicate that vehicles relying solely on disc brakes produce the lowest rolling noise levels. A close second, however, are vehicles with disc brakes and sintered tread brake blocks. The sintered blocks produced a concave wheel tread surface which was suggested as a reason for the slightly higher noise levels relative to those for the disc-brake-only system, even though the wheel running surface appeared to be smoother for the disc- and sintered-block-braked vehicles than for the disc-brake-only vehicles. Surprisingly, vehicles with disc and cast iron block brakes produced higher noise levels than vehicles with cast iron block brakes only. The peak at about 630 Hz for vehicles with disc and cast iron brakes is related to polygonalization of the tread at a wavelength of about 6.3 cm (*6*).

Experience to date suggests that vehicles with disc brakes only have the smoothest wheel treads and produce the lowest levels of noise compared with vehicles with tread brakes of any type.

## 7.4 TRACTION FAULT DETECTION AND SLIP-SLIDE CONTROL

Traction fault detection and slip-slide control systems are particularly effective in inhibiting formation of wheel flats, especially in wet weather, where friction braking can easily lock wheels and induce slip. Thus, traction fault detection and slip-slide control systems are some of the most effective means available to the vehicle manufacturer to control wheel flats and, thus, rolling noise (*7*). Traction fault detection systems have been used on at least 30 older MTA NYCT vehicles and are designed to reduce slip during braking by monitoring and servoeing both brake cylinder pressure and motor currents. Slip-slide or spin-slide control systems monitor axle speeds and adjust braking or tractive effort on each axle to equalize axle speeds and thus control slip during both

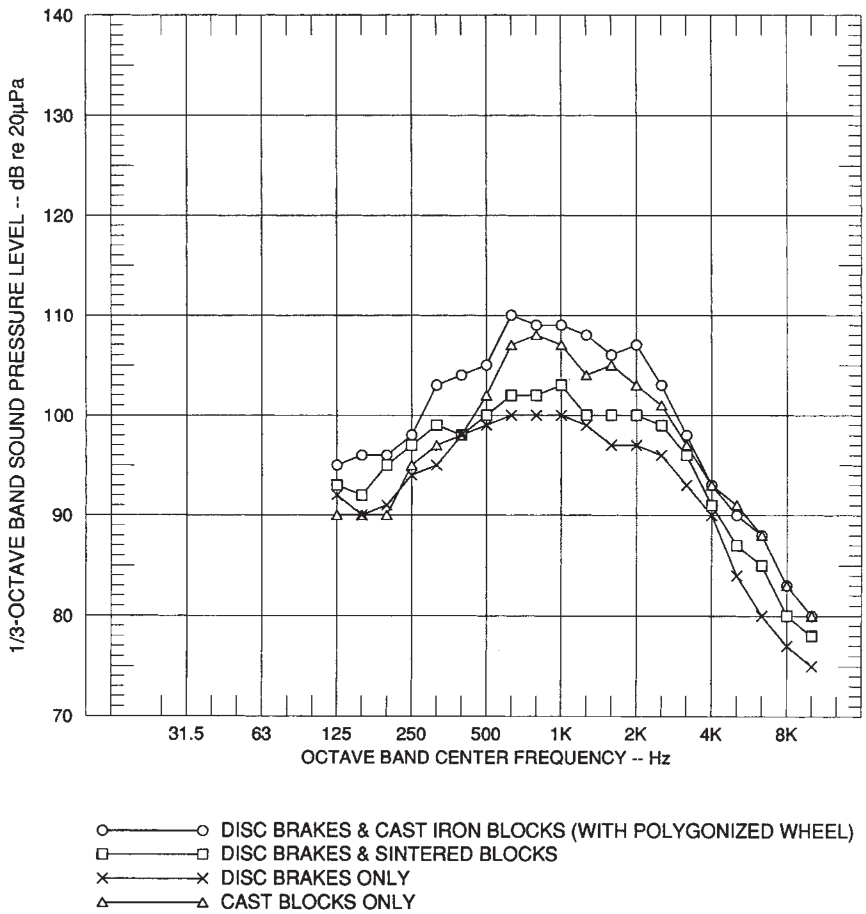


**FIGURE 7-1 EFFECT OF BRAKING SYSTEMS ON WHEEL/RAIL ROLLING NOISE - AVERAGE NOISE LEVELS AT 0.5 AND 1 METER FROM NEAR RAIL OF 89 MPH INTERCITY TRAINS**

braking and acceleration. Slip-slide control systems are thus more advanced than traction fault detection systems.

### 7.4.1 Effectiveness in Preventing Wheel Flats

The MTA NYCT reports that 50% fewer wheel flats occur with both traction fault detection systems and slip-slide control systems (survey questionnaire) (*8*). Metro Dade County reports that its slip-slide control system is not 100% effective and that small flats occur at times (survey questionnaire). Other transit systems may experience less than 100% effectiveness. A 50% reduction of wheel flat occurrence translates to a 50% reduction in wheel truing effort and a significant increase in tread life. Frequent wheel truing may result in excessive removal of tread material, thus reducing tread life.

### 7.4.2 Costs

The costs for dynamic brake control systems with traction fault detection or slip-slide control systems are not easily determined, primarily because these braking systems usually come standard with the vehicle; thus their costs are not broken out. SEPTA indicates that the costs for a microprocessor controlled system with active sensing is $10,000 per vehicle, though they are in a bidding process for the system (*9*). The MBTA has recently received bids for slip-slide control retrofits of between $5,000 and $10,000 per vehicle. A cost of $10,000 per vehicle should be assumed unless other data are obtained to the contrary.

## 7.5 RESILIENT TREADS

Resilient treads have been proposed to reduce contact stiffness and thus vibration and noise (*10*). The resilient tread consists of a steel band surrounding the wheel rim. A portion of the rim is relieved by machining a channel, which is bridged by the tread, as shown in Figure 7-2. The resilience is caused by bending of the tread over the relieved area.

### 7.5.1 Performance

Resilient treads have been considered theoretically and experimentally under laboratory conditions, but no practical

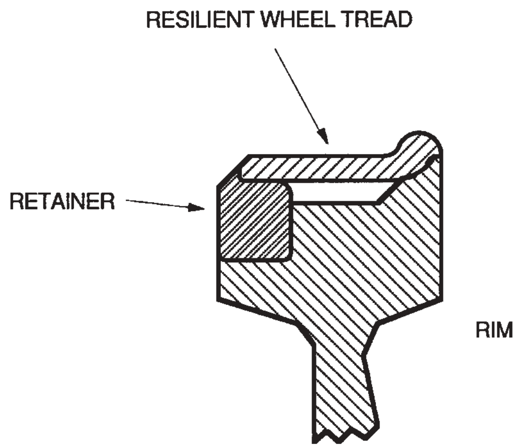


**FIGURE 7-2 RESILIENT TREAD DESIGN (REMINGTON, 1983)**

examples of resilient tread technologies for rail application have been implemented. From a theoretical perspective, greater compliance at the wheel contact area may result in lower noise levels at high frequencies, and laboratory tests with rolling rigs indicate that the A-weighted noise radiated by wheels fitted with resilient treads is reduced in level by 3 to 6 dB (*11*).

### 7.5.2 Costs

Cost data have not been obtained for resilient tread designs. Costs would include additional machining of the tire, including drilling and tapping, and addition of the resilient tread and retaining rings. Costs may be offset to some extent by replacing the tread, rather than by grinding. However, the tires would still be subject to flange wear, which might limit tread life. The flange and tread cannot be turned or milled to a significantly smaller diameter, thus decreasing the life of the tire. There may be possibilities for improving the design, however. For instance, the tread and flange might be forged as an integral unit, so that both tread and flange would be removed together. An added feature is that wheel truing might no longer be needed, because a wheel could be reconditioned simply by changing the tire in situ. The tread would then be salvaged, and the material might be reused in forging new treads. (Replaceable tires are a selling point for resilient wheels such as the Bochum 84 or SAB wheels.) Further, wheel diameters would be carefully controlled by tread replacement rather than by truing. Considering the cost of truing, including labor and capital expenditure, there may be some cost advantages associated with a replaceable resilient tread over conventional wheels.

## 7.6 DAMPED TREADS

High damping alloys have been investigated as a means of controlling contact patch resonance and corrugation. There has been some success demonstrated in roller rigs in the laboratory, though there has been no field application to date (*12*). High damping alloys for treads are not considered a practical noise control option for the purposes of this project.

## 7.7 NITINOL TREAD WHEELS

A nickel-titanium (Nitinol) tread consists of a band of Nitinol alloy measuring, perhaps $^3/_8$ in. thick by 2 in. wide, as illustrated in Figure 7-3. Nitinol is a superelastic material which has a lower modulus of elasticity than steel, can undergo considerable recoverable strain, and has a negative coefficient of thermal expansion (*13*). This potentially exciting material deserves additional study and testing for transit application.

### 7.7.1 Noise Reduction Performance

Nitinol alloy treads have been studied theoretically and evaluated under laboratory conditions with a roller rig to determine their potential for reducing rolling noise (*14*). The laboratory tests indicate that Nitinol treads may reduce wheel/rail rolling noise by 3 to 5 dB relative to standard steel wheels. Of particular value may be elimination of stick-slip vibration at curves, which may substantially reduce or eliminate wheel squeal (*15*).

### 7.7.2 Wear Reduction Performance

The Nitinol treads were produced by Raychem Corp. and subjected to laboratory evaluation of wear and friction-creep characteristics at the Illinois Institute of Technology (*16*). Nitinol provides a lower material modulus, thus reducing contact stiffness, and improves wheel/rail adhesion by way of greater wheel/rail conformity and resistance to lubricating effects of rail surface contamination. The effectiveness of Nitinol as a wear-resistant material has not been demonstrated in transit application, but laboratory tests indicate that wear is reduced with the Nitinol tread by way of contact stress reduction.

### 7.7.3 Costs

The costs of Nitinol in 1978 was about $500/lb, thus making it relatively expensive compared with other noise control provisions. Today, costs are considerably lower, about $20 to $50/lb, making the material cost of a 30-in.-diameter Nitinol tread about $350 to $860. Machining costs are probably about $20 per tread, so that the overall cost per tread would range between $400 and $1,000 per tread. The replacement rate due to wear is not known, though the wear properties are claimed to be excellent. The tire is likely to be condemned due to normal flange wear rather than wear of the Nitinol tread. A Niti-

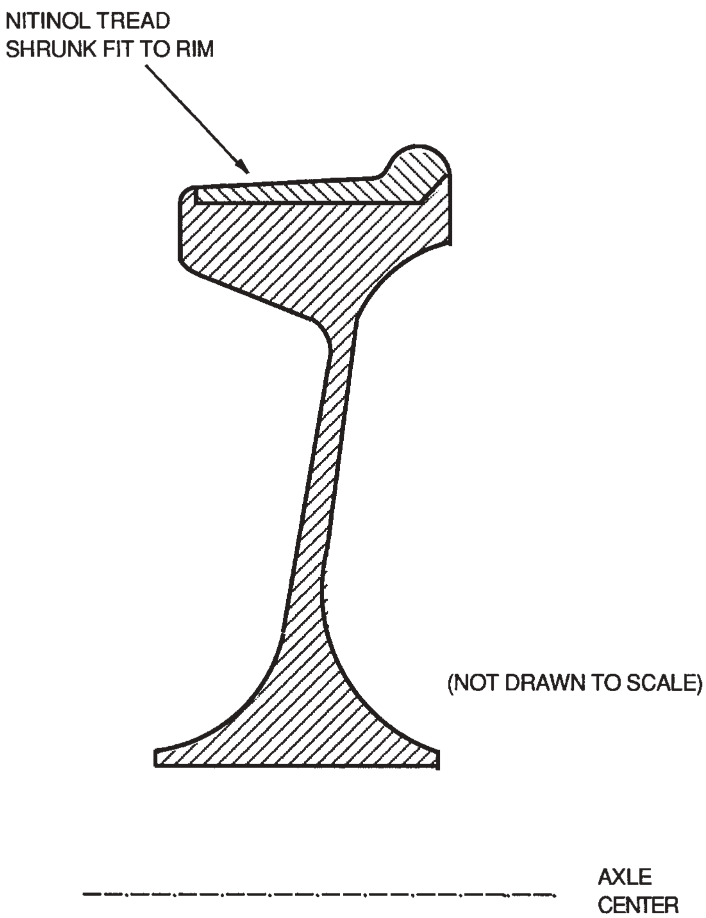


FIGURE 7-3 CONCEPTUAL DESIGN OF NITINOL TREAD

nol tread with integral flange may be attractive and could be replaced rather than trued by conventional techniques.

As with the resilient tread design discussed above, wheels might be reconditioned simply by replacing the tread. In this case, the reconditioning process would involve cutting the existing tread from the rim, cryogenically cooling a new tread, expanding the supercooled new tread with a hydraulic tread expander, placing the new tread over the wheel rim, and allowing the tread to come to room temperature, whereupon it shrinks over the rim, ensuring a snug mounting, without retainers and mounting bolts. This would represent a major change from conventional shop practice, for which cost impacts are not known. Cost impacts may well be favorable.

## 7.8 RESILIENT WHEELS

Resilient wheels are used on many light rail transit vehicles and some heavy rail transit vehicles for reducing wheel squeal at curves. Resilient wheels are constructed with a resilient element between the tire and wheel center. The resilient wheel is particularly effective in reducing wheel squeal noise, because the material damping of the elastomer springs may overcome the negative damping of the friction force versus creep velocity characteristic. In addition, the tread of the resilient wheel is not rigidly constrained, so that the tread may follow the rail without undergoing lateral slip for significant distances, thus preventing stick-slip vibration. The resilient element also offers some vibration isolation between the tire and wheel center, which may be beneficial in reducing noise, if only slightly, and reduces the unsprung mass of the wheel set at frequencies between about 100 and 500 Hz.

### 7.8.1 Products

There exist a number of suppliers of resilient wheels, attesting to their success and acceptance in the industry. Examples include the Presidents Conference Committee (PCC), Bochum, SAB, and Acousta-Flex wheels. The Bochum and SAB wheels are used widely in the United States by light rail transit vehicles. The various wheels that have been incorporated at transit systems are described below.

*Presidents Conference Committee (PCC) Wheels.* The PCC wheel, the first resilient wheel, was constructed of steel discs and rubber inserts intended to reduce shock and vibration impacts on vehicle trucks, or bogies. The wheel was introduced by the PCC of the American Transit Association. The PCC wheel was employed on the PCC cars used at many street car systems throughout the United States, but was not used widely on other vehicles.

Variants of the PCC wheel include the Penn Machine Co. PCC Standard Resilient Wheel with $^{3}/_{16}$-in. static deflection and the PCC Super Resilient Wheel with $^{3}/_{8}$-in. static deflection, achieved by employing rubber in shear. This degree of deflection causes heat buildup in the elastomer, with a possible loss of rolling efficiency at high speeds and heavy loads, and possible failure when tried on heavy rail subway systems (*17*). However, the Chicago CTA, the MTA NYCT, and other systems have had success with these wheels. The Chicago CTA replaced the PCC wheels with solid wheels because of cost and parts availability, rather than problems with reliability (*18*).

*Acousta Flex.* The Acousta Flex wheel, developed by Standard Steel, working with the Bay Area Rapid Transit District (BART) in the mid 1960s, features a 6061-T6 aluminum alloy center threaded with a carbon steel rim with silicone rubber elastomer separating the rim and center in the threaded area, as illustrated in Figure 7-4. The design is

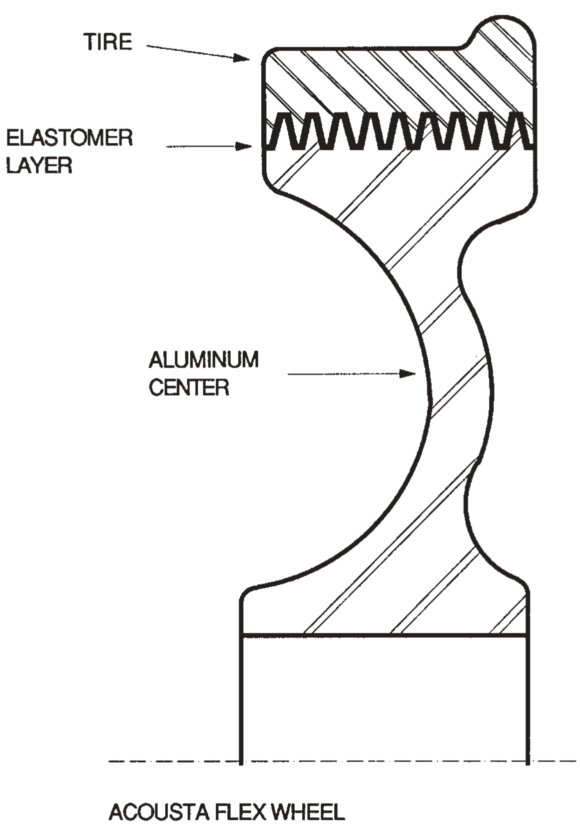


**FIGURE 7-4 ACOUSTA FLEX RESILIENT WHEEL TESTED AT BART**

completed with a carbon steel tire shrink-fitted to the rim, electrical shunts, and a steel locking ring. The tire may be removed and replaced with another to extend service life of the wheel. The silicone rubber is $^{3}/_{16}$ in. thick and is injected into the threaded region between the center and rim.

The Acousta Flex wheel was mounted on the State-of-the-Art-Car (SOAC) developed by the UMTA and was delivered to light rail transit systems at San Francisco and Boston. A set of Acousta Flex wheels was mounted on Car 107 for testing at BART and was subjected to revenue service at BART on a single vehicle for a number of years.

*Bochum.* The Bochum resilient wheel, illustrated in Figure 7-5, was initially developed in Europe by Bochumer Verein A.G. (a subsidiary of Fried Krupp Huttenwerke A.G.) and is one of the most widely used of the resilient wheels at U.S. transit systems. The Penn Bochum radial deflection is designed to be on the order of 0.040 in. to 0.050 in., primarily to reduce shock and vibration of axle and truck-mounted equipment, based on work conducted for the PCC. Penn Machine Co. is licensed to manufacture and/or distribute the wheel in the United States. The Bochum wheel comes in two types: the Penn Cushion Wheel Bochum 54 and the Penn Bochum 84.

The Penn Cushion Wheel Bochum 54 employs rubber blocks in compression between a steel tire and aluminum center. The rubber elements are not vulcanized to the rim or tire. Internal electrical shunts are incorporated between the tire and rim, though some systems provide an external strap between the tire and rim, which eliminates the possibility of pitting of the interior surface of the tire caused by electrical arcing or resistive heat buildup. There is a concern that pitting would introduce stress concentrations and possibly contribute to fatigue of the tire. SEPTA has observed that there appear to be problems with the electrical continuity caused by failure of electrical leads resulting from tire rotation. BART has run a set of Bochum wheels with external shunts for several years on a single car without difficulty. The Bochum wheel weighs considerably less than a standard wheel, because of the aluminum center.

The Penn Cushion Wheel Bochum 84 is similar to the Bochum 54, except that a removable ring is employed to retain the elastomer and tire tread, thus allowing retreading without pressing the wheel from the axle or requiring demounting of the truck and axle. The manufacturer indicates that the Bochum 84 tire may be removed entirely with hand tools. The Bochum 84 wheel is currently used by the MBTA and will be used by the Portland Tri-Met for the center trucks of the new low-floor-height vehicles now in procurement. The tire of the Penn Bochum wheel may be returned to the factory for replacement when worn excessively.

Some of the more interesting and noteworthy benefits of the Bochum wheel are listed by the manufacturer's representatives as follows (*19*):

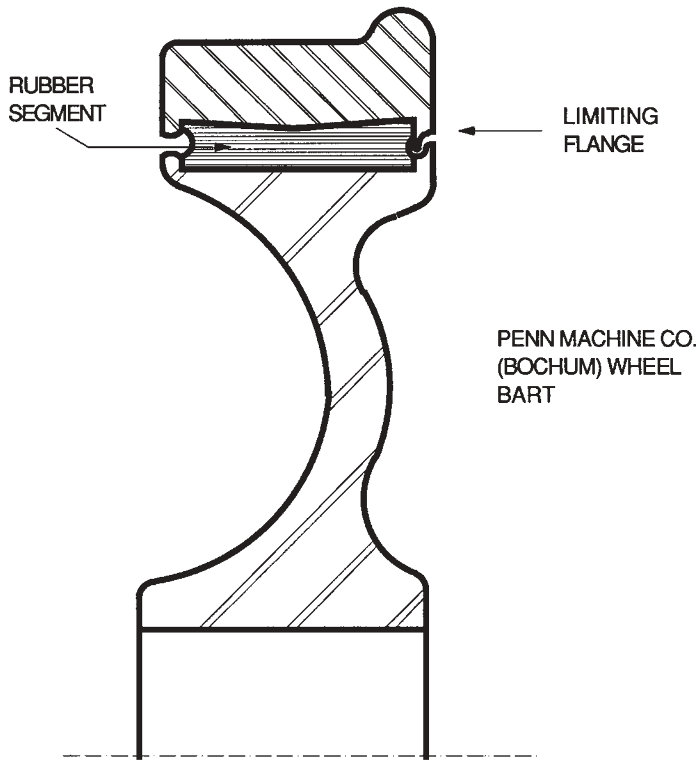


**FIGURE 7-5 BOCHUM WHEEL USED AT BART ON CAR 107**

- High-frequency squeal almost completely eliminated at curves in excess of 100-ft radius,
- Substantial elimination of flat spotting,
- Substantial reduction of impact forces,
- Reduction of flange and rail (gauge face) wear,
- Annular spring effect on acceleration and braking,
- No recorded cases of rail corrugation, and
- No recorded wheel failures of the 50,000 in use as of 1975. (A failure was reported for tread braked vehicles at SEPTA during testing (*20*).)

In contradiction to no recorded cases of rail corrugation, rail corrugation has been observed on embedded girder rail and ballasted track with 115-lb/yd rail at the Portland Tri-Met. SEPTA has also observed rail corrugation at embedded track.

The above list is impressive and deserves further evaluation before selection by a particular transit system. For example, rail corrugation is a common occurrence at many light rail transit systems employing resilient wheels, including the Bochum wheel, though corrugation has not been attributed directly to the resilient wheel.

*Penn Super Cushion Wheel.* The Penn Machine Co. provides a high compliance resilient wheel featuring a rubber-in-shear isolator. The design is in use at the Greater Cleveland RTA, SEPTA, and Toronto Transit UTDC systems. No performance data have been obtained for this wheel.

*Bochum Composite Resilient and Damped Wheel.* VSG also provides a vibration absorber system which is attached to the wheel tire to augment the squeal noise reduction effectiveness of the Bochum resilient wheels. No test data have been obtained, but the combination is expected to reduce wheel squeal occurrence more than a single resilient or damped wheel would.

*SAB Wheel.* The SAB wheel, illustrated in Figure 7-6, provides greater vertical compliance than the Bochum wheel, without sacrificing lateral stiffness. The SAB wheel is used on a number of light rail systems, including SEPTA, MBTA, and the San Jose LRT.

### 7.8.2 Noise Reduction Effectiveness

Comprehensive measurements of noise reduction effectiveness of resilient wheels were made in 1972 at BART during the Prototype Car 107 tests (*21*), in 1979 at SEPTA (*22*), and in 1982 at the MBTA Green Line (*23*). Additional tests have been conducted by the London Transport (*24*). These data are discussed below with respect to tangent and curved track.

#### *7.8.2.1 Tangent Track Performance*

Tests conducted at SEPTA indicate that resilient wheels are largely ineffective in reducing running noise on tangent

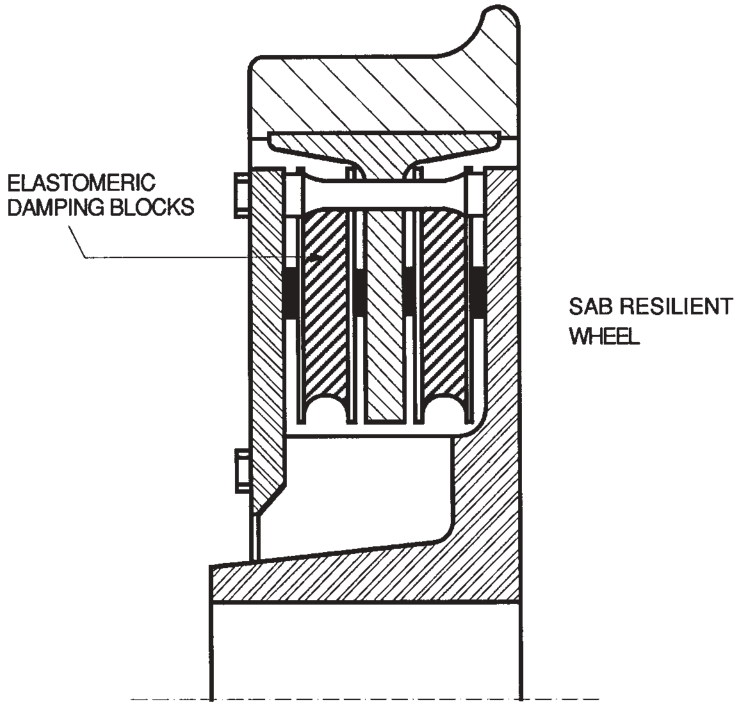


**FIGURE 7-6 SAB RESILIENT WHEEL**

track (contrary to some manufacturers' claims) (*25*). The wayside noise reductions were 0 to 2 dB on tangent track with resilient wheels relative to solid wheels. The data obtained at BART indicate that the noise reduction on tangent ballast-and-tie track with smooth ground rail was on the order of 0 to 1 dBA, similar to the results obtained at SEPTA. The interior noise was described subjectively as lower with the Penn Bochum wheel than with the standard wheel. At the MBTA, test data indicate no significant difference between standard solid and resilient Acousta-Flex and SAB wheels at the Huntington Avenue Station for train speeds on the order of 25 to 30 mph.

*7.8.2.2 Curved Track Performance*

A comparison of squeal noise reductions achieved with the Penn Bochum, Acousta-Flex, and SAB wheels is presented in Table 7-1 for tests conducted at BART, SEPTA, and MBTA. The Penn Bochum, Acousta Flex, and SAB wheels are all very effective in reducing wheel squeal and howl on smooth ground rail at curves. The BART Car 107 tests indicate reduction or elimination of 500 Hz wheel/rail howl at curves with ground rail. Measured reductions of car interior noise were 12 to 18 dBA at a 540-ft radius curve for the Penn Bochum wheel on ground rail, with similar noise reductions outside the vehicle. The Acousta Flex wheel provided noise reductions of 0 to 2 dBA on unground rail at the same curve. At a shorter 530-ft radius curve, the tests indicated car interior noise reductions of 3 dBA for the Penn Bochum on ground rail and 5 dBA reduction for the Acousta Flex on unground rail. At the squeal frequencies above 1,000 Hz, the noise reductions were greater, as shown in Table 7-2.

Examples of octave band noise levels obtained for BART Car 107 are provided in Figure 7-7. The peak at 500 Hz is due to low-frequency squeal or howl, believed to be caused by vibration of the tire in a transverse mode. The Bochum wheel was effective in eliminating the 500 Hz component. The AcoustaFlex wheel is less resilient than the Bochum and was ineffective on unground rail and marginally effective on ground rail in reducing the peak at 500 Hz.

The SEPTA tests indicate that the Bochum wheel was the most effective resilient wheel in reducing wheel squeal at frequencies above 1,000 Hz, followed by the Acousta Flex and then by the SAB. This squeal mode is believed to be caused by bending vibration of the tire, with little strain energy storage in the web or axle, a mode that may be contrasted with the vibration mode at 500 Hz. The SAB wheel was less effective than the former two because it had a lower loss factor than those of either the Penn Bochum or Acousta Flex wheels.

Wayside curving A-weighted noise reductions were 8 to 10 dBA for the Penn Bochum and Acousta Flex wheels and 3 to 4 dBA for the SAB wheel. At the squeal frequencies,

**TABLE 7-1 NOISE REDUCTION EFFECTIVENESS OF RESILIENT WHEELS AT CURVES - NOISE LEVELS WITH RESILIENT WHEELS RELATIVE TO THOSE WITHOUT**

| System | Track Type | Wheel Type | Relative Noise Level | | |
|---|---|---|---|---|---|
| | | | Wayside | Interior | At Squeal Frequency |
| | | | dBA | dBA | dB |
| SEPTA | All Tangent Track, Welded and Joined | Acousta Flex | 0 to -1 | 0 to -2 | -- |
| | | Penn Bochum | 0 to -1 | 0 to -2 | -- |
| | | SAB | 0 to -1 | 0 to -2 | -- |
| | Curve Track | Acousta Flex | -8 to -10 | -1 to -2 | -5 to -30 |
| | | Penn Bochum | -8 to -10 | -1 to -2 | -20 to -30 |
| | | SAB | -3 to -4 | 0 to -1 | 0 to -30 |
| BART | Ballast& Tie, Continuous welded Rail | Acousta Flex | 0 to -2 | 0 to -2 | -- |
| | | Penn Bochum | 0 to -2 | 0 to -2 | -- |
| | Curve, DF Track in subway | Acousta Flex | -3 to -9 | -1 to -5 | -3 to -25 |
| | | Penn Bochum | -9 to -16 | -8 to -18 | -15 to -30 |
| LONDON TRANS-PORT | Tunnel | Penn Bochum | -- | -5 | -- |
| | | SAB | -- | -3 | -- |
| MBTA | Lechmere Station Loops | Acousta Flex | -13 | -11 | -- |
| | | SAB | -14 | -17 | -- |

typically above 1,000 Hz, the Penn Bochum produced a 20 to 30 dBA noise reduction, followed by the Acousta Flex with a 5 to 30 dB reduction, and then by the SAB with a 0 to 30 dB reduction. The reduction by 30 dB at the squeal frequency is due to elimination of the squeal. London Transport reports a total noise reduction of 3 and 5 dBA at curved track with the use of the Penn Bochum and SAB wheels, respectively.

Recent measurements were conducted at Sacramento RTD and at the Los Angeles Blue Line (*26*), both with Bochum resilient wheels and onboard lubrication with HPF and LCF dry-stick lubricant. The curve at Sacramento is embedded ballast-and-tie track, with concrete pavement between the rails. At Los Angeles, specifically Long Beach, the track is resiliently supported with an elastomer pavement between the rails and at either side. The results at Sacramento indicate that squeal occurs at the 82-ft radius curves, while at the 100-ft radius curve, squeal occurrence is very low and limited to a pure tone at about 500 Hz. At the 90- and 100-ft radius curves at the Los Angeles Blue Line, squeal is nonexistent.

How much of the squeal noise control at the Blue Line curves is due to the Bochum wheel and how much is due to the HPF and LCF onboard lubrication is not known, and squeal is observable at larger radius ballast-and-tie track curves in the Blue Line maintenance yard. The combination of resilient wheels with onboard lubrication, $^1/_4$ in. wider wheel gauge, $^1/_2$-in. track gauge widening at curves, 115-lb/yd rail, elastomer grade crossing, and resilient rail support appears to be effective for controlling squeal.

Portland Tri-Met vehicles use resilient Bochum wheels on monomotor trucks. Measurements were conducted at Tri-Met 82-ft radius embedded track curves at a time when oil drop lubricators were used on two cars of the fleet. The embedded track consisted of girder rail embedded in a solid urethane elastomer, poured in a concrete trough. Wheel squeal with the Bochum wheels was not observed during periods of high humidity and dampness of the rail, but was significant during dry periods. The squeal at Tri-Met was not very different from that observed at the Sacramento 82-ft radius curve. At the larger radius ballast-and-tie curves, the squeal was also significant.

In summary, the effectiveness of resilient wheels in controlling squeal is mixed at short radius curves. However, squeal appears to be well controlled on curves of 100 ft or larger radius by a combination of resilient wheels, onboard tread lubrication with a friction enhancer, tighter gauge, and rubberized grade crossing.

### 7.8.3 Site-Specific Conditions

Discussed below are certain site-specific limitations that may apply to selection of resilient wheels.

#### *7.8.3.1 Tread Brakes*

The measurements at SEPTA demonstrated that resilient wheels with elastomer springs are not entirely compatible with tread braked systems due to heat buildup caused by friction between the brake shoe and tread. Both the SAB

TABLE 7-2 SQUEAL NOISE REDUCTIONS OBSERVED FOR VISCO-ELASTICALLY DAMPED WHEELS AT SHORT RADIUS CURVES

| TEST CONDITION | | | | A-WEIGHTED LEVEL | OCTAVE BAND FREQUENCY - HZ | | |
|---|---|---|---|---|---|---|---|
| Curve Radius | Speed | Location | Rail | | 500 | 1,000 | 2,000 |
| feet | mph | | | dBA | dB | dB | dB |
| 540 | 35 | Interior | Unground | 0 | 1 | 1 | 8 |
| | | | Ground | 7 | 7 | 5 | 12 |
| | | Exterior | Unground | 0 | 0 | 0 | 10 |
| | | | Ground | 6 | 5 | 5 | 10 |
| | 18 | Interior | Unground | 2 | 2 | 2 | 7 |
| | | | Ground | 9 | 4 | 8 | 10 |
| | | Exterior | Unground | 5 | 4 | 3 | 13 |
| | | | Ground | 4 | -1 | 5 | 3 |
| 530 | 18 | Interior | Unground | 6 | 0 | 6 | 22 |
| | | | Ground | 0 | 1 | 0 | -4 |
| | | Exterior | Unground | 10 | 3 | 6 | 19 |
| | | | Ground | -1 | -2 | -1 | -2 |

Note: Negative values indicate higher noise levels.

and Penn Bochum wheels failed in this manner. Two of the elastomer blocks of one of the Penn Bochum wheels contained defects which were exacerbated by heat caused by exclusive use of the tread brakes after failure of the dynamic braking system. The SAB wheels on one of the axles suffered severe damage after application of the hand brake (ostensibly for extended periods of time) during revenue service. The Acousta Flex wheel was removed during testing due to bond failure, possibly caused by incomplete bonding during manufacture. The problems with the resilient wheels were discovered before any structural failure of the wheels.

There are several examples of the successful use of resilient wheels on vehicles with tread brakes. One is the original PCC streetcar, in use since the 1930s. Prior to 1974, London Transport evaluated SAB and Bochum wheels for several years of revenue service. The SAB wheels were run 367,000 km on transit cars with tread brakes, and 53,000 km of service were with a car with no dynamic braking. The Bochum wheels were operated for 212,000 km of revenue service with no problems caused by heat from braking (*27*).

*7.8.3.2 Disc Brakes*

BART, which uses disc brakes on all vehicles, ran Bochum resilient wheels on Car 104 for roughly 10 to 12 years, with one rebuild, and the Acousta Flex wheels on Car 107 for less time, without serious problems (*28*). There were some difficulties with shunts, which were solved by adding external cable shunts. Interestingly, the original Bochum wheels tested in 1972 have been overhauled and are now in service on rehabilitated Car 596, the only car of the BART fleet of 500 or more cars with resilient wheels.

*7.8.3.3 Truck Shock and Vibration Reduction*

A perhaps very valuable ancillary benefit of resilient wheels is the reduction of truck shock and vibration, the reason for the development of the PCC resilient wheel. BART Cars 104 and 107, with resilient wheels, are believed to have experienced less truck maintenance problems than the other vehicles, though the information is entirely anecdotal and without corroboration with written record. BART has experienced significant failure rates during the early years of operation of cars fitted with rigid wheels, though car reliability with standard wheels is much improved and considered acceptable. At the very least, the survivability of Cars 104 and 107 appears to be unimpaired by the use of resilient wheels. Again, these observations are purely anecdotal and not supported by quantitative data.

*7.8.3.4 Corrugation*

The Cleveland LRT RTA reports (survey questionnaire) the occurrence of ripple corrugation at station stops with resilient wheels on light rail vehicles and no problems with solid steel wheels on the heavy rail system. (No wheel/rail noise problems are reported, however.)

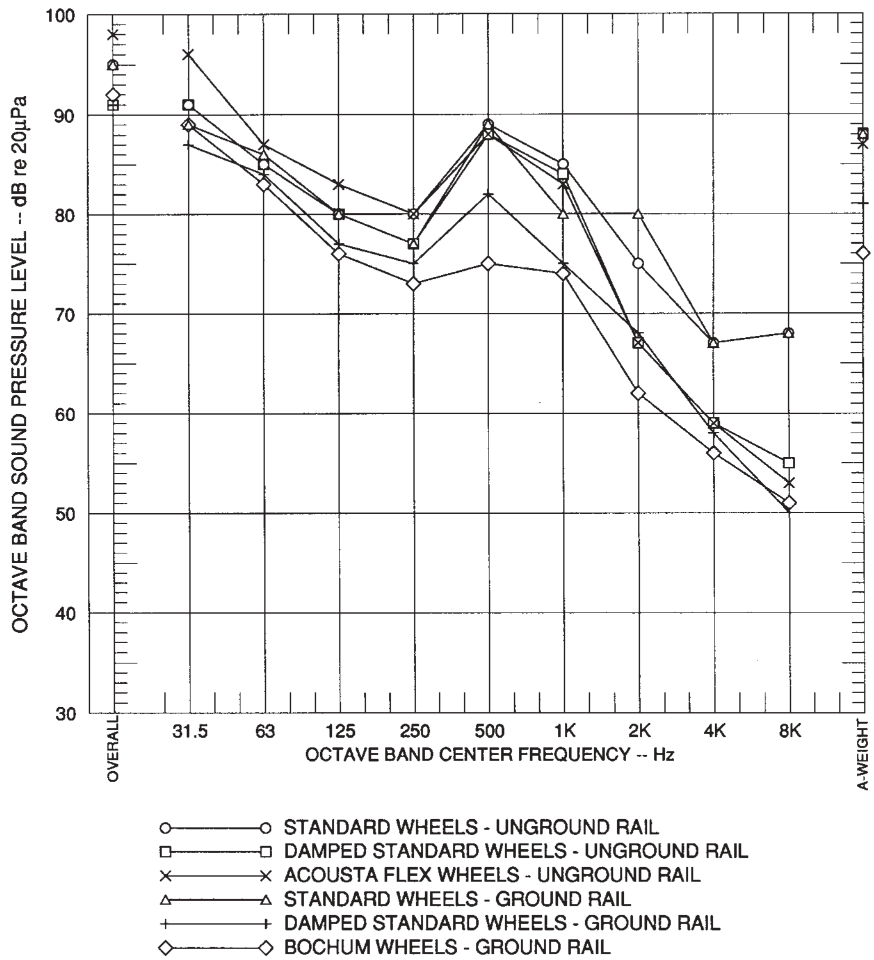


**FIGURE 7-7 CAR INTERIOR NOISE LEVELS FOR BART PROTOTYPE CAR 107 IN BOX SUBWAY ON 540-FT RADIUS CURVE - 35 MPH**

The Pittsburgh system is evidently experiencing corrugation at various sections of track and also uses the Bochum wheel. Part or all of the problem may be related to deferred rail grinding, which might otherwise control this problem.

There appears to be significant rail corrugation at certain light rail embedded tracks incorporating girder rail embedded in solid urethane elastomer. Examples include the Portland Tri-Met, where significant groundborne noise and vibration from corrugated embedded track has been observed (*29*), and the system at Calgary, Alberta. Both of these systems employ Bochum resilient wheels. Subsequent rail grinding appears to have alleviated this problem. Portland is designing new embedded tracks to have a lower rail support modulus than that of urethane elastomer embedded track, by employing resilient rail fasteners. How well the resilient embedded track works with the resilient wheels is not known yet, but rail corrugation appears to be under control at the Los Angeles Blue Line, which has a resilient rail support for the embedded track (*30*).

TTC received numerous complaints concerning ground vibration after introduction of the Canadian light rail vehicles with Bochum resilient wheels. TTC vigorously investigated the problem and decided to keep the Bochum wheel and control ground vibration with rail grinding and wheel truing (*31*). The problem was identified with high radial stiffness, and resilient wheels with lower radial stiffness evidently exhibited less tendency to become rough and generate ground vibration. However, TTC determined that wheel truing and maintaining rolling diameter tolerances for all wheels was sufficient to justify retaining the Bochum wheel (*32*).

### 7.8.4 Costs

The costs for Bochum wheels range between $1,800 and $3,000 per wheel, depending on size and type (*33*), and costs vary from procurement to procurement. The cost of a replacement tread is about $1,000, delivered. Labor for tread replacement may also be about $1,000 per wheel, depending on local labor rates and shop practices. The tread replacement cost is driven by a need to remove the wheel from the vehicle. The new Bochum 84 wheel is designed to allow tread replacement in situ. Costs for the SAB and other resilient wheels have not been established.

There may be significant cost savings achieved with resilient wheels with respect to reduced truck maintenance resulting from reducing truck shock and vibration. The PCC resilient wheel was developed with this in mind.

## 7.9 DAMPED WHEELS

Damped wheels reduce noise by absorbing bending vibration energy stored in the tire and wheel web. Damped treatments come in a variety of configurations, the most economical of which is the ring damper, a frictional (or Coulomb) damper consisting of a steel ring let into the inner diameter of the wheel rim or tire. Other configurations include visco-elastic damping rings, vibration absorbers attached to the wheel rim, and constrained layer dampers attached to the rim. Most of these are effective in controlling wheel squeal at curves, but have little effect on wheel/rail noise on tangent track.

An advantage of the damped wheel relative to the resilient wheel is that wheel squeal can be controlled without concern over stability of the wheel. The tread of the resilient wheel tire can deflect relative to the wheel center, which may not be acceptable for certain types of track. Stuttgart is evidently replacing resilient wheels with rigid wheels fitted with vibration absorbers.

Vibration absorbers attached to the tire and designed to absorb vibration energy over a range of frequencies from 400 Hz to 5,000 Hz can be effective in reducing wheel squeal. Usually, high-frequency squeal at frequencies above about 1,000 Hz is most significant. However, the low-frequency transverse vibration of the tire at roughly 500 to 1,000 Hz might be associated with corrugation formation, and damping of these vibration modes might reduce rail corrugation rates and associated noise.

A study of the vibration modes of the wheel will aid in the identification of appropriate damping treatments. Constrained layer dampers must be applied at locations involving maximum bending strain. Vibration absorbers, on the other hand, should be mounted at locations of maximum vibration velocity. For transverse bending of a wheel, maximum bending strain and velocity occur at the tire. Significant bending of the wheel center may also be involved, in which case a constrained layer damping treatment would be effective if applied to the web of the wheel, but a vibration absorber would not be effective if mounted at the web, or worse, at the hub.

For radial vibration deformation of the wheel, constrained layer damping treatments would be ineffective if applied to the wheel center, but a vibration absorber applied to the rim with effective axis in the radial direction would be effective. A vibration absorber capable of absorbing both radial and transverse vibration at the tire appears to be a very attractive approach to controlling both radial and transverse modes of wheel vibration.

### 7.9.1 Products

Commercially available damping systems include the following.

*Krupp Tuned Vibration Absorbers.* The Krupp tuned vibration absorber wheel has been offered by Krupp-Stahl AG of Germany since the late 1970s. The design, illustrated in Figure 7-8, consists of multiple stainless steel dampers bolted to a steel rim, which, in turn, is bolted to the rim or tire of the wheel. A variant of the design tested at the MTA NYCT was bolted to the field side of the tire with T-slots machined into the tire. Other variations of mounting have been developed and investigated. The damper assembly includes variable thickness damping blades, each tuned to specific modal frequencies of the wheel. The weight of the damper assembly is 44 lb, but some material is removed from existing tread for mounting, limiting the overall weight increase to less than 10 lb. At the MTA NYCT, 40 lb of material were removed from the wheel for testing, giving a net weight increase of about 4 lb (*34*).

*Adtranz.* Wheel vibration absorbers are provided by MAN/GHH and have been distributed by AEG Transportation (now Adtranz) for Deutsche Aerospace. The manufacturer's literature indicates that these units are very effective in reducing noise at resonance peaks above 500 Hz. AEG

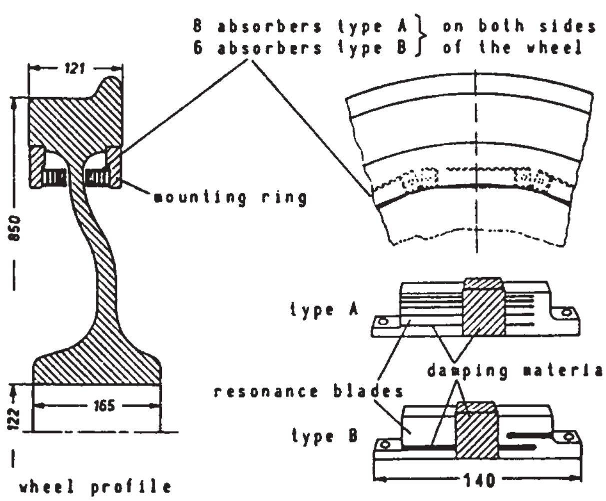


**FIGURE 7-8 KRUPP TUNED VIBRATION ABSORBER**

has marketed a vibration absorber called a broad band wheel noise vibration absorber, developed by Innovative Noise Control Technologies (INCT). Both Adtranz and INCT are part of Deutsche Aerospace AG (DASA) and Daimler Benz Group.

The vibration absorber is a fin-type absorber with multiple bending elements mounted on the tire and extending radially inward toward the center of the hub. Adtranz indicates that the absorber design has been in use in Europe since 1980, including by the ICE. Currently, Adtranz is offering a block vibration absorber (referred to as VICON) which is compact and lightweight and may be tuned to wheel squeal frequencies. Tests conducted by Adtranz at the WMATA Metro indicate that these block absorbers are effective in reducing wheel squeal at 250-ft radius curves.

*Bochum Composite Resilient and Damped Wheel.* VSG, through Penn Machine, provides a resilient Bochum wheel with vibration absorbers attached to the tire to augment the squeal reduction effectiveness of the Bochum resilient wheel. The damper system consists of leaf springs and is applied to the Bochum 54 and Bochum 86 wheels. As discussed above, the Bochum wheel does not completely eliminate wheel squeal at short radius curves, and addition of properly tuned vibration absorbers to the tire would be expected to significantly reduce remaining squeal noise.

*Ring Damped Wheels.* A generic ring damped wheel consists of a mild steel ring let into the inner surface of the steel rim, as illustrated in Figure 7-9. This configuration is in use at the Chicago CTA and is effective in controlling wheel squeal (*35*). The damping arises from Coulomb friction between the ring and the confining groove. No drilling and tapping are required to mount the damper.

*Visco-Elastic Ring Damped Wheel.* A variant of the generic ring damped wheel is the Soundcoat Co. ring damped wheel. The ring damper consists of a 0.625-in.-diameter steel rod coated with Soundcoat DYAD damping treatment. The ring is let into the inner diameter surface of the steel tire and bonded in place. At New York, the ring was installed in a standard MTA NYCT wheel with a groove cut for the purpose on the field side. An advantage of the visco-elastic ring damped wheel over the generic Coulomb ring damper is that damping action will not be impeded by contamination between the ring and tire or freezing due to corrosion. The visco-elastic ring damper, dependent on bending of the tire tread for effectiveness, would not effectively control transverse rotational rigid vibration of the tire.

*Sumitomo Ring Damper.* The Sumitomo ring damper consists of a damping layer bonded between two concentric steel rings. The assembly is force-fitted to the interior surface of the wheel rim at the field side, positively retained by four screws. This type of damper would be most effective in controlling bending vibration (squeal above 1,000 Hz) of the tire and ineffective in controlling rigid transverse rotational vibration of the tire.

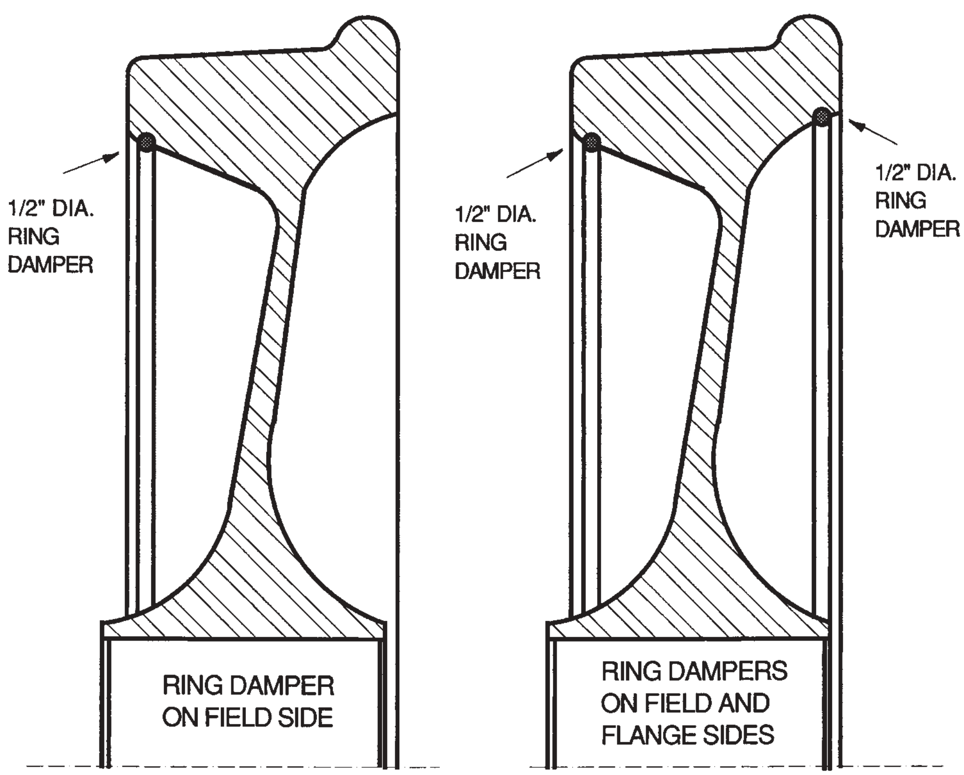


**FIGURE 7-9 RING DAMPED WHEELS**

*Soundcoat Constrained Layer Damped Wheels.* The Soundcoat Co. provides a constrained layer damper consisting of a layer of DYAD damping treatment adhered to the side of the rim of the wheel and constrained with a steel angle rolled into a ring, with positive mechanical retention. This configuration has evidently been implemented at the Paris Metro and was tested at the MTA NYCT. The damper would be effective in controlling bending vibration of the tire and ineffective in controlling rigid transverse rotational vibration of the tire.

*Wheel Center Constrained Layer Damper.* A second type of constrained layer damper, not commercially available, consists of a layer of Soundcoat visco-elastic damping treatment constrained between a spun aluminum dish and the gauge side of an aluminum centered wheel. The design (Figure 7-10) was evaluated at BART (*36*). The design is included in the report because it represents one of the earliest designs evaluated on a modern U.S. transit system.

*Constrained Layer Acoustic Absorber (Klockner-Werke).* The Acoustic Absorber consists of a series of concentric circular 1-mm-thick sheet metal annular discs separated by 2 mm of visco-elastic damping material, or thermoplastic. When the assembly is bolted to the wheel, metal-to-metal contact is avoided. The absorbers are manufactured by Klockner-Werke AG. No performance data have been obtained.

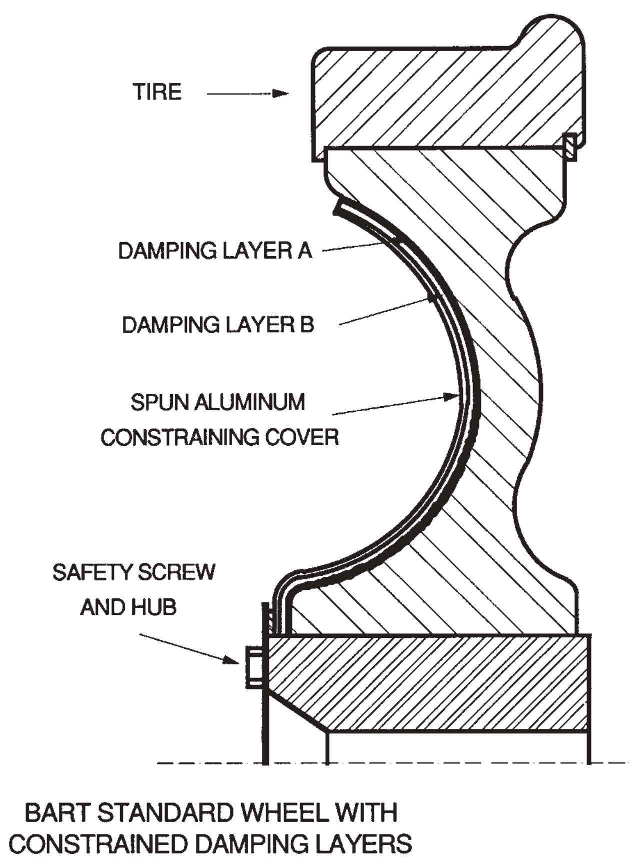


**FIGURE 7-10 BART CONSTRAINED LAYER DAMPER FOR WHEEL CENTER**

### 7.9.2 Noise Reduction Effectiveness

The noise reductions of various damped wheels were measured at BART on Prototype Car 107 (*37*), at SEPTA (*38*), and at MTA NYCT (*39*). The BART tests involved measurement of the noise reduction performance of a dish-shaped constrained layer damper on the BART Prototype Car 107, and the SEPTA tests included a measurement of the generic ring damper. The tests at the MTA NYCT included the (*1*) generic ring damper, (*2*) Soundcoat visco-elastic ring damper, (*3*) Sumitomo visco-elastic ring damper, (*4*) Soundcoat constrained layer damper, and (*5*) the Krupp tuned vibration absorber.

The description of squeal noise reduction performance is complicated by the intermittent nature of wheel squeal. Squeal occurrence is affected by humidity, contaminants, lubrication, turning radius, and so forth. When squeal does occur, it may reach levels in excess of 100 dBA at perhaps 25 ft from the vehicle, and the maximum level may vary over a range of 10 dB or more. With damping, wheel squeal noise is usually reduced, and there is usually greater uniformity of passby noise levels.

Ranges of observed A-weighted noise reductions for tangent and curved track are presented in Table 7-3 for most of the damped wheels identified above. The levels are based on the worst-case differences between maximum and minimum noise levels observed for the damped wheel and the comparable undamped wheel. Thus, the ranges are based on the minimum as well as maximum observed noise for the damped wheels and thus are exaggerated to some extent. The best indicator of performance is whether the squeal is eliminated, which is difficult to determine from the literature unless noted therein or indicated by inspection of narrow band or 1/3-octave band spectra.

#### *7.9.2.1 Products*

*BART Constrained Layer Damping Dish.* The viscous damping plate, described in Figure 7-10, was evaluated on the BART Prototype Car 107. The results indicated no significant noise reduction on tangent welded track. On curved track, the results were mixed. At a 540-ft radius curve, the squeal noise reduction at the important 500 Hz octave relative to the standard wheel was insignificant for unground rail and about 7 or 8 dB on ground rail. At higher frequencies, the noise reduction was about 10 dB in the 1,000 and 2,000 Hz octaves. A-weighted noise reductions were on the order of 4 to 6 dB, limited by the performance at 500 Hz. A summary of the observed noise reductions is presented in Table 7-2. On unground rail, the peak in the noise spectra was at about 500 Hz. After rail grinding, the peak in the squeal frequency shifted upward to 1,000 to 2,000 Hz, where the damping treatment was more effective. The reason for the frequency shift of squeal noise was not determined, nor was the rail grinding procedure described in the report.

There are certain attractive features of the dish-shaped visco-elastic constrained layer damping treatment for the wheel center. The treatment may be applied to the interior side of the wheel, allowing inspection from the field side for wheel web flaws. A second feature is that no machining is required of the tire or tread to accommodate a constrained layer dish damper. Thus, there is no stress concentration resulting from modification of the tire or machining costs for retrofit.

The dish-shaped dampers do not appear to be as effective in reducing squeal noise as the ring dampers, either visco-elastic or Coulomb type, perhaps because the mode shape associated with squeal noise primarily involves distortion of the tire or tread, as opposed to bending of the wheel center. That is, most of the strain energy associated with wheel squeal is in the tire. Finally, the dish damper at BART was not as effective as the Bochum resilient wheel in reducing curving noise.

*Soundcoat Constrained Layer Damped Wheel.* The Soundcoat constrained layer damped wheel was evaluated at the MTA NYCT and found to be effective at all frequencies. The damping treatment was applied to a standard MTA NYCT 34-in.-diameter wheel for testing. Wheel squeal was virtually eliminated, as indicated by the lack of discrete frequency squeal components in passby noise spectra, relative to the standard MTA NYCT wheel. After 17 months of service, squeal at 2,325 and 4,175 Hz was obtained. Maximum A-weighted noise levels were 82 to 86 dBA after 17 months, compared with 81 to 83 dBA when new. Degradation may be attributed to wear of the rail and wheel, which may affect tribological properties and wheel and rail contours. That is, wear of the tire tread may reduce or eliminate tread taper, thus reducing or eliminating rolling radius differential and increasing slip between the tire and rail. Another possibility is that wheel flange and/or rail gauge face wear allowed an increase in crabbing angle and thus lateral slip.

*Visco-Elastic Ring Damper.* The Soundcoat and Sumitomo visco-elastic ring dampers were tested at the MTA NYCT and found to be as or more effective than the Soundcoat constrained layer damped wheel. Both of these dampers place damping treatment directly against the tire, where it is most effective in absorbing bending vibration of the tire. The narrow band spectra for selected samples of noise obtained during curve negotiation indicate little evidence of wheel squeal components, though the resonant response of the associated vibration mode can be observed.

*Generic Ring Damped (Coulomb) Wheels.* The noise reduction effectiveness of ring damped wheels was evaluated at SEPTA and at the Chicago CTA (*40*, *41*). Tests at SEPTA indicate that the ring damped wheels exhibit a damping factor comparable to that of the SAB wheel (which is a factor of 8 to 10 less than those of the Bochum 54 and Acousta Flex wheels) at frequencies above about 1,400 Hz. Above 1,400 Hz, squeal noise was reduced, and below 1,400 Hz, little noise reduction was obtained, possibly because there was little bending strain in the tire at these lower frequencies. After several months of service, the rings became impacted or frozen in their grooves due to corrosion and foreign material such as brake and steel dust. When frozen in their grooves, the rings lose their frictional damping characteristics.

Wheel squeal noise reductions on the order of 9 to 11 dB were observed for ring dampers that were not frozen in place. With the rings frozen, the noise reduction was on the order of 3 and 5 dB for ground and unground track, respectively. The noise reduction effectiveness of ring damped wheels on tangent track at SEPTA was on the order of 2.5 dBA or less, with the maximum reductions on worn or jointed track. For continuous welded rail, the noise reductions were not significant, as has been observed at other systems. The greatest tangent track noise reduction was observed in the car interior, while wayside noise levels were reduced less than 1.6 dB.

At the Chicago CTA, interior noise reductions of 1 to 15 dB and 5 to 20 dB were observed for squeal noise components at curved track. At tangent track, running noise reductions for welded track were limited to 2 dB (*42*). Other tests indicated a 2 dB noise reduction on tangent jointed track and 0 dB on tangent welded track, consistent with the results obtained for the SEPTA system (*43*).

*Krupp Tuned Vibration Absorber.* The Krupp tuned vibration absorber is illustrated in Figure 7-8. Wheel squeal noise was inhibited by the Krupp tuned vibration absorber during tests at the MTA NYCT. Further, the vibration modes associated with wheel squeal components appear to be completely eliminated. However, tests after 17 months of service were inconclusive, because the Krupp vibration absorber was tested on a vehicle with another vehicle with standard wheels, which produced squeal. The data suggest, however, that squeal is also absent from the passby noise for the vehicle equipped with the Krupp tuned vibration absorber.

*Adtranz (AEG DASA & GHH/MNN) Vibration Absorber.* Manufacturer's data provided by Adtranz indicate substantial reduction of noise by GHH/MNN vibration absorbers at discrete frequencies above 500 Hz. The noise reduction applies to squeal at curves, and the manufacturer indicates a 20 to 30 dBA noise reduction. The wheels are about 4% heavier with the absorbers than without. AEG reports that the AEG DASA vibration absorbers reduce wheel squeal noise at the 250-ft radius curve at the WMATA West Falls Church maintenance yard (*44*). The newer block vibration absorbers currently promoted by Adtranz have been tested at WMATA with significant reduction of squeal noise.

*Constrained Layer Acoustic Absorber (Klockner-Werke).* The Klockner-Werke Acoustic Absorber was

**TABLE 7-3 NOISE REDUCTION EFFECTIVENESS OF VARIOUS DAMPED WHEELS**

| Damper | Test Location | Damper Condition | Track Type | Noise Reduction A-Weighted dBA | Noise Reduction Squeal Frequency dB |
|---|---|---|---|---|---|
| Generic Ring Damper | SEPTA | ? | Tangent Welded & Joined | 0 to 1 | -- |
| | | New | Curve | 2 to 11 | 10 to 25 |
| | | Seized | Curve | 0 to 5 | 0 to 5 |
| | CTA | ? | Tangent Jointed | 1 to 2 | -- |
| | | | Tangent Welded | 0 | -- |
| | | | Curve | 3 to 8 | 5 to 20 |
| | NYCTA | New | Curve | 21 | 24 |
| | London Transport | ? | Tangent | 0 | -- |
| Constrained Layer Dish Damper | BART | New | Tangent Welded Ballast & Tie | 0 | -- |
| | | | Curve | 10 | 25 |
| The Soundcoat Co. Visco-Elastic Ring Damper | NYCTA | New | Curve | 8 to 25 | Eliminated |
| | | After 17 months | Curve | 11 to 22 | Eliminated |
| The Soundcoat Co. Constrained Layer Damped Wheel | NYCTA | New | Curve | 7 to 24 | Eliminated |
| | | 17 Months | Curve | 11 to 18 | -- |
| The Sumitomo Constrained Layer Damping Ring | NYCTA | New | Curve | 9 to 27 | Eliminated |
| | | 17 Months | Curve | 13 to 20 | -- |
| Krupp Tuned Vibration Absorber | NYCTA | New | Curve | 8 to 25 | Eliminated |
| | | 17 Months | Curve | 12 to 19 | -- |

tested by the Deutsche Bundesbahn on 250 km/hr intercity coaches (*45*). The results of the tests indicate that the noise reduction produced by the absorber is about 5 dB for trued wheels and ground rail at tangent track when integrated over a frequency range of about 600 to 3,800 Hz. The authors indicate that the result would be similar for A-weighted noise levels. The wheel/rail roar noise reduction performance of the absorber on tangent track is one of the highest reported. At frequencies above 1,000 Hz, noise reductions on the order of 10 dB or more were obtained. However, the noise reduction provided by the absorber was small or nonexistent at frequencies below 1,000 Hz, where much of transit wheel/rail noise at tangent track occurs. Systems with tangent track A-weighted noise dominated by noise energy at frequencies less than 1,000 Hz might not benefit from such an absorber, similar to the result obtained for the BART constrained layer dish absorber. Finally, the test data indicate that the source height for the wheels with absorber is close to the top of rail, suggesting that the dominant source of noise at frequencies above 1,000 Hz may be the rail when the resonances of the wheel are effectively damped.

*7.9.2.2 Site-Specific Conditions*

*Temperature.* Vibration absorbers, dynamic absorbers, or dampers which absorb vibration energy with a visco-elastic elastomer are affected by temperature. However, no data have been supplied concerning the performance of vibration absorbers at low or very low temperature. Fortunately, the windows of residential and commercial buildings are likely to be closed during cold periods, so that the temperature dependance of the absorbers may not be particularly important. Pedestrian traffic would still be exposed to squeal. In subways, temperatures are more uniform, so that vibration absorber performance may not be lessened during the winter. Visco-elastic damping materials may be optimized for various temperature ranges. Before selecting a vibration absorber for application in cold climates, the manufacturer should provide data concerning cold weather performance.

*Dust and Debris—Freezing.* Squeal noise reduction effectiveness of SEPTA ring dampers was lost as a result of dirt and other material freezing the damping ring in the retaining groove. The Chicago CTA reports no such problem, however.

*Wheel Tire Design.* Wheel tread designs may be such that mounting of damper assemblies to the tread may compromise the fatigue resistance of the tread. Care must be taken in selecting and mounting damper assemblies. Usually, a stress analysis should be performed, and the manufacturer of the damping treatment should warrant the product against damage to the tire resulting from mounting.

*Outboard Disc Brakes.* Outboard disc brakes may interfere with mounting of damper assemblies to the exterior of the wheel. AEG mounted the DASA absorbers on the inboard side of the wheel to avoid this problem during tests at WMATA. However, they would be less easily inspected on the inboard side. Wheel vibration absorbers appear to be mounted on the flange side of the wheel in most applications, which may help protect the absorber from damage by trackside appurtenances and equipment.

#### *7.9.2.3 Costs*

The approximate cost for tuned vibration absorbers is expected to range between $500 and $1,000 per wheel. In sufficient quantities, these costs might be reduced further. Preparation of the wheels and mounting of the absorbers may cost an additional $1,000 per wheel. However, assuming that the mounting holes can be drilled and tapped in about 4 hours of shop time, the mounting cost, at $75 per hour, may be on the order of $300. Both procurement and mounting costs must be considered.

The various constrained layer damping treatments tested at the MTA NYCT cost about $1,000 to $1,200 per wheel, including mounting, at the time of the tests in the 1980s. Applying a producer price index of 1.5 would make these costs closer to $1,500 to $1,800 per wheel. However, these costs should be reduced considerably for large quantities, and costs of the order of those for block vibration absorbers would not be unexpected.

The ring damper costs about $35 per wheel. The groove may be machined into the tire during manufacturing, so that the cost for providing the groove may be minimal.

No cost data were obtained from operating U.S. transit systems, due to lack of application, except for the ring dampers.

## 7.10 UNDERCAR ABSORPTION

Undercar absorption was evaluated on the BART Prototype Car 107 in 1970 (*46*). Car interior noise data collected over the X-End truck (between the doors) at 60 and 80 mph on ballast-and-tie tangent track with ground rail indicate a 3 to 4 dB noise reduction with either standard or damped wheels. Maximum effectiveness was obtained at the 500 Hz and higher octave bands. These results are particularly encouraging because the ballast already contributes substantial sound absorption. The undercar treatment was applied over the trucks and consisted of 140 sq ft of Microcoustic duct liner of 1 in. thickness. Propulsion system noise is a dominant source of undercar noise on ballast-and-tie track for the BART vehicle on smooth ground continuous rail. Undercar sound absorption is expected to be effective in controlling noise between the underside of the vehicle floor and top of the truck.

Measurements of car interior noise at the 19th and 12th Street curves in the BART subway indicate 0 to 2 dBA noise reduction with the use of undercar absorption, including a reduction of squeal and howling noise by about 2 dB. The reduction is most significant in the 250 to 4,000 Hz octave bands. However, in the subway, most of the sound transmitted into the car interior is radiated outward from the vehicle truck area and reflected from the subway walls, and this sound energy would not be absorbed directly by the undercar sound absorption.

Wheel/rail noise that is radiated inward and upward and reflects from the underside of the vehicle will tend to find its way into the train tunnel annulus if it is not absorbed and then into the car interior, subject to transmission loss of the car body. Undercar sound absorption can remove this component, but would leave the noise radiated outward from the wheels and rail unabsorbed. Thus, no more than 1 or 2 dB noise reduction would be expected from undercar absorption over the truck area, as is supported by the data. Essentially the same result, about 3 dB reduction, is obtained with scale model predictions, though the results apply primarily to vehicles with substantial undercar equipment noise (*47*). To the extent that there is no significant absorption in subways with direct fixation track, provision of sound absorbing materials throughout the entire underside of the vehicle would, presumably, have a significant effect on reverberant sound levels in the subway.

Undercar absorption, which is relatively inexpensive, typically on the order of $10 per sq ft for glass fiberboard, would also reduce undercar equipment noise and would be effective systemwide. Installed costs for absorption may be considerable, depending on the degree of protection and shaping required. Costs were estimated to be $40 per sq ft in 1980 (*48*). More definitive cost data should be acquired before selecting undercar absorption as a treatment. Undercar absorption should be considered for obtaining additional noise control provided that interference with maintenance would not occur. Ideally, new car procurements should include undercar absorption provided by the manufacturer.

## 7.11 SKIRTS

Vehicle skirts have been proposed as a noise control treatment for both wheel/rail and traction system noise. Wheel/rail noise reductions at tangent track are expected to be minimal, because the rail and lower part of the wheel remain exposed. Earlier researchers have indicated that the noise reduction may be limited to 0 to 3 dB (*49*). Wheel squeal should be reduced to the extent that squeal noise is radiated from the upper as well as lower portions of the wheel. The interior surfaces of the skirts should be treated with acoustical absorption to enhance their effectiveness.

Wheel skirts have been incorporated into the Denver light rail transit system vehicles supplied by Siemens Duewag, though no noise reduction data were obtained for this vehicle. Portland Tri-Met plans to use low-floor-height vehicles with vehicle skirts, and unofficial measurement data indicate that noise reductions attributed to the skirts are about 2 dB.

A detailed study of vehicle skirt noise reduction in combination with undercar absorption and absorptive barriers was undertaken with model experiments and applied to measured noise levels from PATCO trains (*50*). The results are applicable primarily to vehicles with substantial undercar equipment noise, such as from propulsion system fans and gear boxes. The results do not indicate that skirts would be effective for controlling wheel/rail noise. The results for vehicle skirts in combination with absorptive sound barriers located close to the vehicle, as on an aerial structure, suggest that reduction of wheel/rail noise might be substantial, due to the narrow path required for propagation of sound between the exterior skirt surface and absorptive surface of the wall. In this case, the noise reduction of the combined treatment was predicted to exceed the sum of the noise reductions for the individual treatments by 2 dB. The total noise reductions are expected to be in excess of 10 dB. Without barriers, however, the undercar equipment noise reduction for skirts is estimated to be 5 to 9 dBA for half and full skirts, respectively, with undercar absorption. Again, for wheel/rail noise, which is not shielded by the skirt, the noise reduction may be less than 3 dB.

Test results were reported for the Shinkansen high-speed trains fitted with skirts extended to within 150 mm of the top of rail and with some sound absorption applied to the interior surface of the skirts (*51*). Some noise reduction was obtained for sections of track without barriers, and no noise reductions were obtained by the extended skirts at sections of track with barriers, suggesting that the skirts have limited usefulness for wheel/rail noise control.

Site-specific limitations may include clearance problems for third rail systems and maintenance and inspection limitations. Retrofit may be limited due to interference with truck-mounted equipment.

Costs were roughly estimated in 1980 to be about $12,000 per vehicle for two full-length skirts and undercar absorption (*52*). Assuming a doubling of producer prices, the current costs would be roughly $24,000 today. However, skirts and absorption would likely be part of an overall vehicle procurement and thus might be supplied at considerably lower costs than based on estimates of 1980. For instance, the Denver light rail system uses Siemens-Duewag vehicles with skirts, and presumable cost data have been developed. The skirts supplied to the Portland low-floor-height vehicle evidently cost about $5,000 per vehicle.

### 7.12 STEERABLE TRUCKS

Steerable trucks have been proposed as a measure for controlling wheel squeal at curves. The TTC Scarborough line and the Vancouver Skytrain employ steerable trucks manufactured by the UTDC. Steerable trucks have axles that are linked in such a way that the axles point toward the center of the curve during negotiation, which eliminates lateral creep due to the finite wheel base of the truck. Rigid axle systems require that some longitudinal slip must occur, unless the curve radius is large enough to allow a rolling radius differential to develop. In this case, the wheel tread must be tapered, and the contact patches must be offset to the low rail side with, perhaps, gauge widening to allow roll without slip. On trucks with cross-linked axles, the steering is affected by the interaction between the wheel and rail, which deforms the primary suspension sufficiently to steer the truck (self-steered). If the axles are linked to the car body, the car body provides the steering forces (force-steered), in which case the steering is accommodated by deformation of the primary suspension or by a pivot. A major advantage of steerable trucks is the reduced wheel and rail wear, reduced energy consumption, and possibly improved ride quality. Reduced wear might offset the initial costs of steerable trucks (*53*).

At short radius curves, corrugation due to stick-slip can still occur, leading to roughened wheels and thus wayside noise. Examples include the TTC Scarborough line, which experienced severe herringbone corrugation at very short radius curves. The problem was exacerbated by friction in the truck suspension which prevented complete steerability.

### 7.13 SOFT VERSUS STIFF PRIMARY SUSPENSIONS

Trucks with soft primary suspension systems appear to produce lower levels of noise than equivalent trucks with stiff suspensions. Reductions on the order of 3 to 5 dBA for wayside noise and 8 to 9 dBA for car interior noise under some conditions are reported for modified Chicago CTA 2000 series vehicles with soft journal bushing suspensions versus CTA 2200 series vehicles. (This might be due to better vibration isolation between the car body and truck components, rather than entirely due to reduction of airborne wheel/rail noise.) Typical interior noise reductions were 4 to 5 dBA for ballast-and-tie track in subways and were less with jointed track in subway and on aerial structures (*54*).

The reduction of interior noise with reduced primary suspension stiffness is understandable, due to reduced force transmissibility across the primary suspension. However, an explanation for the reduction of wayside noise is less obvious, particularly if one assumes that most of the A-weighted noise contribution is at frequencies above perhaps 125 Hz, well above the primary suspension resonance frequency. One possibility is that the truck with soft journal bushing suspensions was better able to position itself on the track, thus reducing stick-slip- or spin-slip-generated rolling noise. Less squeal noise would be expected with soft primary suspensions relative to stiff suspensions, due to better alignment of the axle with curve radii. However, this requires softness in the longitudinal direction, as opposed to the vertical direction, to allow the axles to steer themselves, a condition which has some impact on ride quality and stability.

In contrast, BART has one of the stiffest primary suspension systems and one of the quietest transit vehicles when the wheels are well trued and the rail is ground smooth. Further, tests at BART indicate little noise reduction by softening the primary suspension (*55*).

## 7.14 ONBOARD LUBRICATION

Onboard lubrication is used to lubricate the wheel flanges and enhance friction and tractive characteristics of the tread and rail. Flange lubrication is generally a friction reducer, designed to reduce wear and frictional forces. The tread lubrication is generally a friction enhancer, designed to flatten the friction versus creep curve and reduce or eliminate stick-slip between the tread and rail head running surfaces.

### 7.14.1 Flange Lubrication

Flange lubricant can be either in solid form (stick), grease, or oil. Of the three, grease has been found to be most effective in reducing rail and wheel wear at railroads. Solid lubricants and greases have both been used for noise reduction. However, if not properly controlled, the grease lubricant can find its way onto the top of the rail, leading to wheel-slip and poor train braking. (Note: This is also the case with wayside lubricators.) Ensuring that only the curves are lubricated is also difficult (though not always necessary). Dry-stick lubrication has become popular at light rail transit systems such as the Los Angeles Blue Line and Sacramento RTD, and Portland Tri-Met is experimenting with onboard dry-stick lubrication as well. The Portland Tri-Met system has also employed oil dispenser lubricators on two of the vehicles to control wheel squeal.

Advantages of onboard lubrication include ability to inspect and install new lubricant in the shop. A key advantage is the potential for increased control of the lubricant dispensing process and the use of equipment not kept at isolated field locations but brought back into maintenance shop areas on a periodic basis. In addition, the vehicular-mounted systems permit maintenance and inspection to be carried out in more convenient central locations, such as in yards or shops.

In recent years, a great deal of attention has been focused on improving vehicle-mounted lubrication systems. Consequently, there have emerged three distinct types of vehicle-mounted lubrication systems:

- High-rail vehicle systems (grease only),
- Locomotive- or car-mounted systems (grease, oil, or stick),
- Dedicated lubricator car (grease only).

In general, for grease application, the benefits of the vehicle-mounted systems, to some extent, are similar for all three types of systems. In all cases, applying the lubricant from a moving vehicle results in a uniform distribution of lubricant along the track. The specific locations where the lubricant is to be applied can be controlled from onboard the vehicle through either manual controls or through automatic sensing systems (for curves). In addition, lubricant can be applied continuously along the right of way, through both tangents and curves, if desired. However, a proper amount of lubricant must be applied. This necessitates frequent applications by the various lubrication systems, depending on the type of system and the amount of lubricant applied.

Vehicle (car- and locomotive-mounted) lubricators apply a predetermined measure of lubricant (grease, oil, or solid) to the wheel flange as the vehicle moves along the track. This lubricant is then deposited onto the gauge face of the rail by the wheel flange. There is no requirement for the lubricant to be carried along the track by the wheel. In fact, the required properties of the lubricant used in this type of system are that the grease remains where it is deposited and does not carry. (Thus, the direct properties of the lubricant are different for vehicleborne and wayside lubricators.)

The primary benefits of vehicle-mounted lubricator systems are noise reduction, energy savings (fuel reduction), and wheel wear reduction. The ability to reduce rail wear, particularly on sharp curves, is much more limited with vehicle-mounted systems than with wayside lubricator systems, including those systems with curve sensors, which increase the lubricant output (oil or grease) on curves.

Vehicle-mounted lubricators encompass a broad range of systems, which can be further defined by the type of lubricant used: solid, oil, or grease. In the rail transit environment, car-mounted stick lubricators can be mounted on every car in the fleet.

### 7.14.2 Oil Drop and Spray Systems

Onboard lubrication includes systems which drop oil, spray water, and apply lubricants to the wheel flange and tread. Oil drop and water spray systems have been in use for some time, exemplified by the Portland Tri-Met system's use of oil drops to lubricate the wheels of two vehicles out of the entire fleet to control wheel squeal at short radius curves.

An onboard lubrication system that uses oil as the active lubricant is the REBS wheel flange lubrication system, consisting of an oil pressurization and feed system mounted directly on the truck. The compressed air from braking systems may be used for pressurization. Lubricant is delivered to the flange through a nozzle as an oil-enriched spray with a droplet size on the order of 0.4 mm, producing a film thickness of about 0.001 mm. The lubricant is described by the manufacturer as biologically degradable and nonpolluting. No noise reduction data have been obtained, though the product is reputed by the manufacturer to reduce noise (*56*).

### 7.14.3 Dry-Stick Lubricants

Perhaps the most interesting development in wheel/rail noise and wear control are low coefficient of friction (LCF) flange lubricants and high positive friction (HPF) tread friction modifiers. An excellent review of onboard lubrication has been presented by Kramer (*57*). These onboard dry-stick lubricants have received widespread application among light rail transit systems and have been subjected to substantial operational testing, with mixed results.

Friction modifiers are distinct from conventional lubrication products. The HPF friction modifier applied to the tread running surface is not intended to reduce friction, but to modify the slope of the friction versus creep curve and reduce or eliminate the negative damping associated with stick-slip vibration and, perhaps, periodic roll-slip or spin-slip behavior which may contribute to short pitch or "roaring rail" corrugation.

LCF lubricants are applied to the flange to control flange and gauge face wear and should not be confused with HPF products, even though both of these products may be applied in dry form and supplied by the same manufacturer. Both LCF and HPF are applied in combination by several light rail systems to the flange and wheel tread, respectively, to improve traction and perhaps reduce wheel squeal and rail corrugation.

#### *7.14.3.1 Systems Using Dry-Stick Lubricants*

Systems that have tested dry-stick lubricants include the following:

- WMATA
- Los Angeles Blue Line and Green Line
- Portland Tri-Met
- BART
- Pittsburgh
- Vancouver Skytrain
- PATH (planning to test)
- Sacramento RTD
- Denver.

Certain British, European, and Asian systems, including the following, have considered, experimented with, or are using dry lubricants (*58*):

| | |
|---|---|
| Docklands Light Rail (U.K.) | RATP Paris Metro (France) |
| British Rail (U.K.) | ET Bilbao (Spain) |
| FEVE (Spain) | Helsinki Metro (Finland) |
| Renfe (Spain) | AKN Hamburg (Germany) |
| DSB (Copenhagen) | Reinsbahn Dusseldorf (Germany) |
| VAG Nurnberg (Germany) | Singapore MRT (Singapore) |
| Rostock (Germany) | |
| London Underground (U.K.) | |

The success or acceptability of the dry lubricants is not known for all of the above systems.

#### *7.14.3.2 Manufacturers*

Manufacturers of onboard lubrication products include the following:

- E/M Corp.—Manufacturer of Glidemaster.
- Kelsan Lubrication—LCF, HPF and HPF II high positive friction, and VHPF very high positive friction dry-stick treatments.
- Phymet, Inc.—This treatment, named Micropoly, is described as a high-density, miroporous polyethylene, with 65% 40-weight gear oil by weight, with particles of Teflon and molydisulfide and other additives (*59*). A Phymet product has been applied by the Metro-Dade County system (*60*).
- KLS Lubriquip—Solid-stick wheel flange lubricant. This product was tested by WMATA and discontinued (*61*).
- A&K Railroad Material—Manufacturer of Abbalube flange lubricant.
- REBS—Wheel flange lubrication for rail-mounted vehicles.

The Association of American Railroads is experimenting with a permanent film lubrication for application to railroads. The permanent film would be porous and impregnated with lubricant. No reports or test data have been obtained.

#### *7.14.3.3 Noise Reduction Effectiveness*

There are two types of noise reduction addressed by dry-lubricant technology: (*1*) wheel squeal at curved track and (*2*) noise caused by "roaring rail" corrugation on tangent or moderately curved track.

##### 7.14.3.3.1 Curved Track

Friction modifiers such as the Kelsan HPF, in combination with LCF stick treatments, may inhibit stick-slip motion at curves, thus inhibiting wheel squeal. To the extent that wheel squeal is eliminated, noise reductions on the order of 20 dBA may be obtained. The results would not be dissimilar to conventional lubrication with water, oil, or grease. In practice, the use of lubricants for noise reduction has met with significant variations in reported rates of success. Some of the reported benefits of using the higher friction dry lubricants are as follows:

*St. Louis MetroLink.* The St. Louis MetroLink employs graphite lubricator sticks for wheel flange lubrication. The sticks are supplied by Kelsan Lubricants Ltd. at a capital cost of $25,000 (*62*). This would presumably be the LCF flange lubricant. No significant wheel squeal is reported at curves. Some gauge widening is employed at curves, and the vehicles are by Siemens-Duewag with monomotor trucks and resilient Bochum wheels (*63*).

*Sacramento Light Rail.* Tests at the Sacramento Light Rail Academy Way train yard using both wayside and onboard noise measurements showed noise reductions with the combined use of an LCF and HPF lubricant applied to the flange and tread, respectively, of 5 to 28 dBA at curves. Figure 7-11 presents one such set of measurement data (*64*). In this case, the lubricants were supplied by Century Lubricating Oils. (Kelsan is now a supplier of these products.) All of the vehicles are lubricated with both LCF and HPF dry-stick lubricant, though only a single axle of the idling truck of each vehicle is so lubricated. The Sacramento RTD uses resilient Bochum 54 wheels.

Sacramento RTD indicates that before application of dry-stick lubrication, complaints concerning noise regularly occurred. After application, no or few noise complaints occurred. A major advantage of the HPF treatment is improved braking, and the Sacramento RTD is pleased with the lubricant. Additional 1/3-octave band test data collected by this author are presented in Figures 7-12 and 7-13 for 100- and 82-ft radius curves, respectively, with two-car Sacramento RTD vehicles. These data indicate the statistical variation of squeal noise as well as the energy-averaged squeal noise level during curve negotiation and the total noise exposure level for 16 vehicles, which may be used in calculating day-night or $L_{eq}$ sound levels.

The $L_{eq}$ levels are most relevant. The curves were adjacent to one another and were constructed of the same track forms, though of different radii. The gauge faces of the rails are lubricated by hand daily with a nonflowing grease (SWEPCO 604). The results clearly show frequent occurrence of squeal at the 82-ft radius curve and only moderate occurrence of squeal at the 100-ft radius curve. Squeal at the 500- and 630-Hz third octave bands occurs at both curves. Only the 82-ft radius curve produced high levels of squeal at the 1,600-Hz third octave band. Inspection of the rail at the 82-ft radius curves revealed a spotty surface, with residue presumably left by the friction modifier. The rail at the 100-ft radius curve was smoothly polished (*65*).

*WMATA.* Tests on a 300-ft radius curve at the New Carollton yard showed wayside noise level reductions of 16 to 17 dBA between dry and lubricated (stick lube) conditions. The results are presented in Table 7-4. WMATA tested the

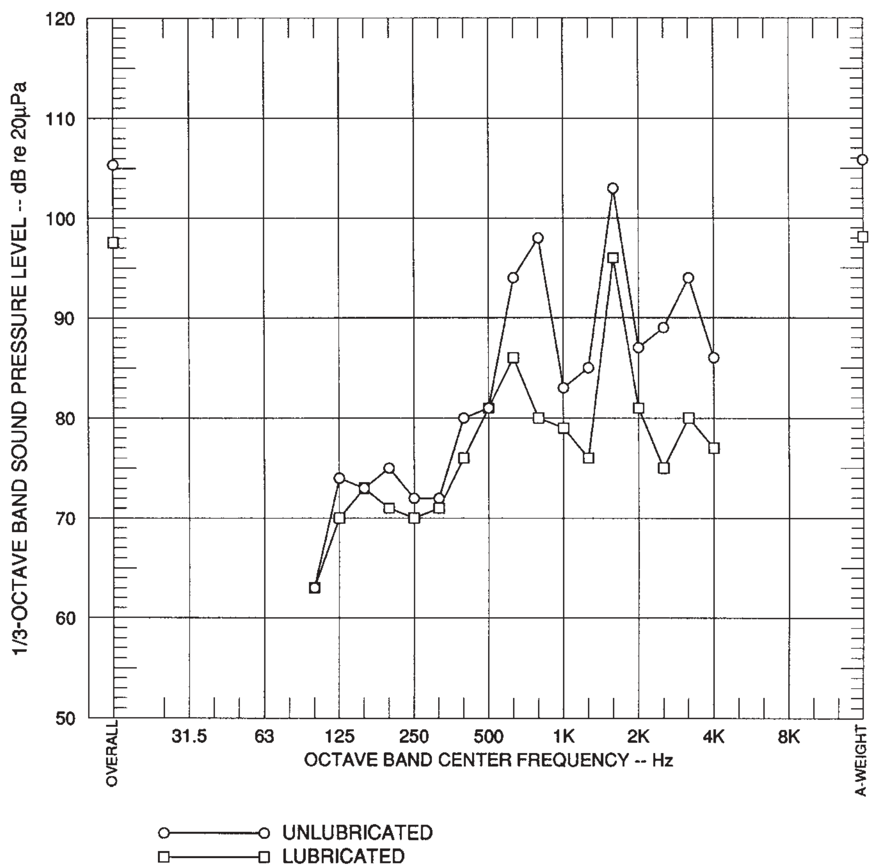


**FIGURE 7-11 WHEEL SQUEAL MAXIMUM NOISE LEVELS AT SACRAMENTO RTD LIGHT RAIL SYSTEM WITH AND WITHOUT HIGH POSITIVE FRICTION LUBRICANT**

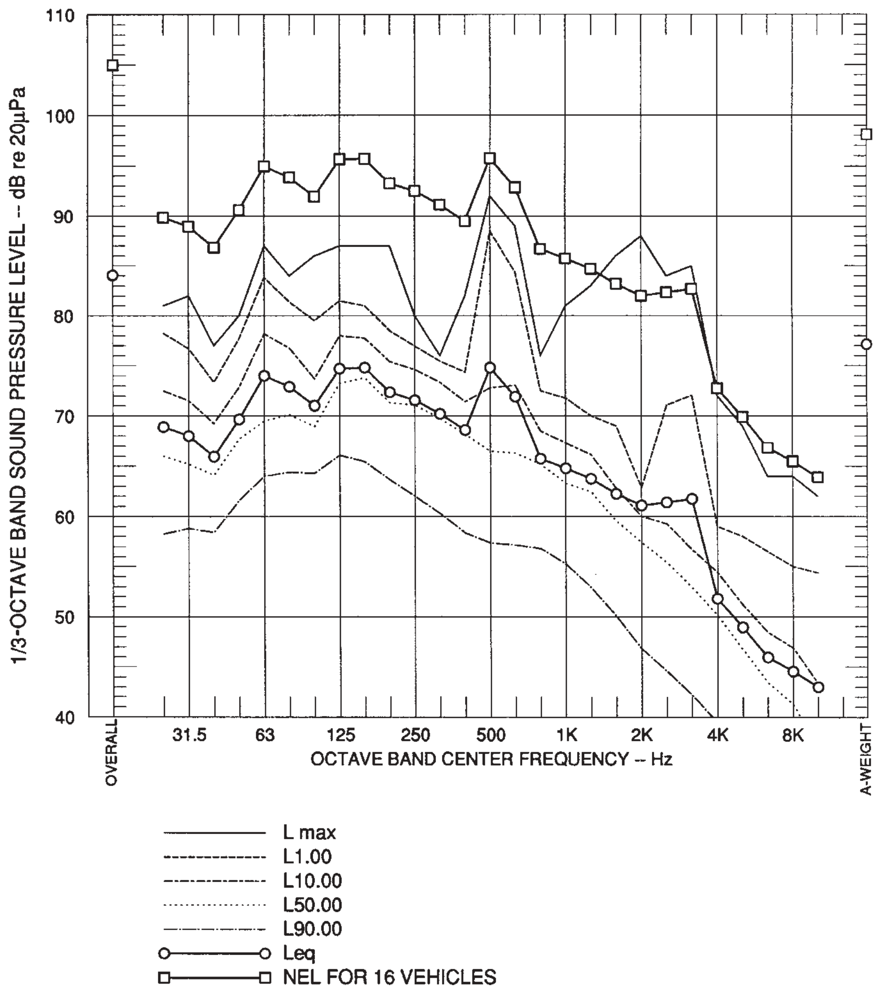


**FIGURE 7-12 CURVING NOISE AT SACRAMENTO RTD, 23 FT INSIDE OF 100-FT RADIUS CURVE - TRAINS AT 7 MPH**

Century Oils LCF Series (now distributed by Kelsan Lubrication) and KLS Lubriquip solid-stick flange treatments and found that the mounting brackets produced rattles and that the materials produced a squeal noise (*66*). Followup conversation with WMATA indicated that the noise reduction was moderate, but the cost was prohibitive, and, therefore, WMATA discontinued evaluation of the materials.

*Portland Tri-Met.* Portland Tri-Met is experimenting with combined Kelsan LCF flange lubricant and Kelsan HPF applied to the flange and tread surfaces of Bochum wheels on the center idling trucks of the entire fleet of articulated light rail transit vehicles. The lubricants were applied in late summer and early fall of 1994. No noticeable wheel squeal noise reductions were observed at short radius (82-ft) curves, nor have any noise reductions been observed at tangent track sections during the period in which the lubricators were installed. However, the weather was wet during the observation period. Tri-Met reinstalled the lubrication systems on all vehicles and had plans to conduct noise reduction tests during the summer of 1996. Tri-Met has indicated that the wheel flanges are more polished than before treatment.

*Los Angeles Blue Line.* The Los Angeles Blue Line has Kelsan LCF and Kelsan HPF dry-stick lubricators applied to the flanges and treads, respectively, on each driven truck of each of the light rail vehicles. The Los Angeles MTA indicates that there is a lack of wheel squeal at curves, which include 90- and 100-ft radius curves in embedded track. The Los Angeles Blue Line embedded track consists of 115-lb/yd rail in resilient rail supports and has Bochum resilient wheels. The wheel flange gauge is wider than standard, and $^{1}/_{2}$-in. gauge widening is employed at embedded track curves.

Observations by this author indicate that the short radius curves in Long Beach are smoothly polished and that there is a complete absence of squeal. However, squeal was observed at ballast-and-tie curved track in the Blue Line shops, which experiences less traffic than the embedded curves, but should be lubricated to the same degree on a per vehicle passage basis.

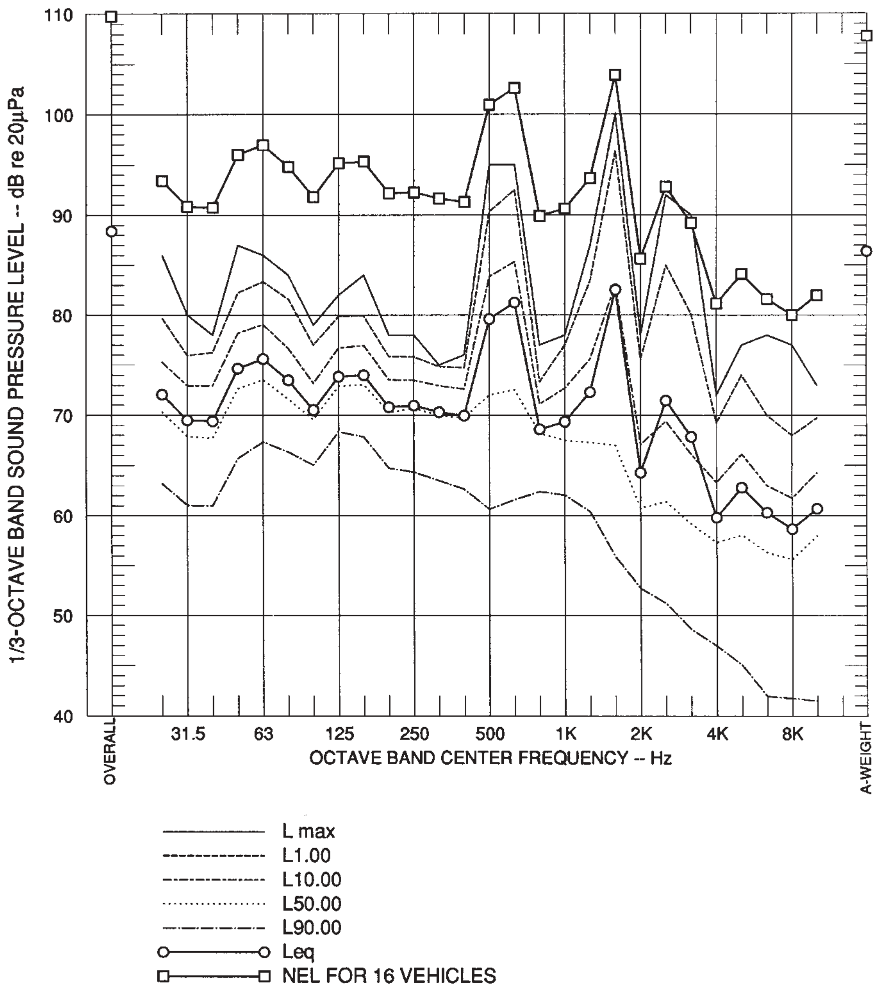


**FIGURE 7-13 CURVING NOISE SACRAMENTO RTD, 51 FEET INSIDE OF 82-FT RADIUS CURVE - TRAINS AT 7 MPH**

*Santa Clara Transportation Agency.* The Santa Clara Transportation Agency tested the Century Lubricating Oils HPF and LCF spring-loaded stick friction modifiers and found a 0 to 2 dB squeal noise reduction. Testing was discontinued when the dry sticks broke or were lost after less than 2,000 mi. The agency employs SAB resilient wheels and rigidly embedded track in concrete. As of this writing, the agency has decided to try the dry-stick lubricants again, though no data are available.

The curve radius and use of resilient wheels appear to be larger factors in controlling wheel squeal than use of the HPF or LCF flange lubricant. However, the lack of squeal at the

**TABLE 7-4 WHEEL SQUEAL NOISE REDUCTION DUE TO LUBRICATION AT CURVE - WMATA**

| TRACK CONDITION | MAXIMUM LEVEL - DBA - | STANDARD DEVIATION - DB - | SOUND EXPOSURE LEVEL - DBA - | STANDARD DEVIATION - DB - | DURATION - SEC - |
|---|---|---|---|---|---|
| DRY | 103 | 2.7 | 109 | 2.9 | 29-32 |
| LUBRICATED | 87 | 9.0 | 93 | 6.2 | 25 |
| LUBRICATED NEW APPLICATION | 79 | 4.1 | 88 | 3.5 | 24 |

Los Angeles Blue Line 90- and 100-ft radius curves suggests that either (*1*) the lubrication of two axle sets of each vehicle is more effective in reducing squeal than lubrication of a single axle set as at Sacramento or (*2*) the embedded track design at Los Angeles with resilient rail support and elastomer road surface is more effective in reducing squeal than the ballast-and-tie track with asphalt and concrete road surface used at Sacramento.

The most popular theory of wheel squeal, discussed in Chapter 5, holds that damping of the track by rubberized grade crossing should have little effectiveness in controlling squeal. The Los Angeles Blue Line employs $^1/_2$-in. gauge widening at the embedded curves, but also has a $^1/_4$ in. wider wheel flange gauge than other transit vehicles. The result is that there may be less crabbing of the truck in the curve than a normal standard gauge vehicle, which may reduce squeal. Another difference is that concavity of the tread running surface profile is evident in the tread profile of worn wheels at Sacramento, while at the Los Angeles Blue Line, wear appears to be less and the wheel treads appear to have a certain amount of taper. Thus, a rolling radius differential is promoted at the curves of the Los Angeles Blue Line, with attendant self-steering of the axles, thus, perhaps, lessening slip and squeal.

The concavity of the tread profile at the Sacramento RTD would not promote a rolling radius differential and self-steering, and thus may exacerbate squeal. At the Los Angeles Blue Line, the combined effect of wider wheel set gauge, tapered tread profile, curve radii of 90 ft or greater, resilient wheels, 115-lb/yd rail, and rubberized grade crossing and HPF tread and LCF flange lubrication may inhibit squeal.

#### 7.14.3.3.2 Tangent Track

A modest noise reduction of at least 2 dB of wayside noise at tangent track in good condition is possible, due to reduction of roll-slip or other stick-slip behavior by use of friction modifiers, even with the rail in otherwise smooth condition. This mechanism of noise generation is considered significant by a number of researchers. (See section on theory of wheel/rail noise generation.) Modification of the friction versus creep velocity curve to avoid a negative slope of friction versus creep velocity would, presumably, tend to inhibit roll-slip behavior. The HPF friction modifier is designed to do this by mixing with oxides and wear debris from the rail and tread and forming a film which provides an effective friction coefficient of between 0.2 and 0.4, with a positive friction versus creep velocity slope. A good test of the effect of solid friction modifiers on tangent track rolling noise requires extensive running in over a period of several days or weeks to allow mixing of the friction modifier with surface oxides on the rail running rail surface.

The greatest potential for rolling noise reduction with use of friction modifiers may be in reducing rail corrugation rates. Reduction of corrugation rate is theoretically possible for a treatment which eliminates the negative friction effect at the contact patch. Prevention of corrugation, either by friction modification, rail grinding, or a combination of both, can result in 10 to 15 dB noise reduction relative to poorly maintained, corrugated rail. Friction modifiers alone applied to corrugated or rough rail would probably not reduce noise, though the manufacturers indicate that a noise reduction of at least 2 dB may be expected in this case. More likely, a combination of profile grinding and application of a friction modifier to lengthen grinding intervals represents a most effective rail maintenance procedure, as indicated by the Vancouver Skytrain. Regardless of whether a friction modifier is used, there is no substitute for effective rail grinding. Following is a discussion of the experiences of various transit systems with HPF and tangent track noise control.

*Vancouver Skytrain.* The Vancouver Skytrain employs the Centrac HPF friction modifier and reports that noise levels actually go up if lubrication is suspended, even with frequent rail grinding (*67*). Further, the friction modifier is beneficial in reducing rail corrugation growth. The Skytrain vehicle has a steerable truck manufactured by UTDC with small wheel size and conical treads. Skytrain engages in frequent profile grinding and wheel truing.

*BART.* BART experimented with a solid lubricant applied to all wheels of a test train at the Hayward Test Track (*68*). Measurements of interior and exterior noise were conducted before and after treatment on recently ground tangent track in excellent condition. Approximately 80 runs were conducted by the train after treatment and prior to measuring the noise. The exterior test results are illustrated in Figure 7-14. Field test personnel indicated that the trains subjectively sounded quieter (after a period of a few days), but the objective test data indicated a 1.5 dBA increase in exterior noise at 50 mph. The interior noise test results, not shown here, indicated a 1 to 2 dB increase at 50 and 70 mph, with little change in spectrum.

The lack of agreement between subjective assessment of the noise and qualitative test data is an example of the difficulty and danger inherent in basing conclusions regarding noise reduction effectiveness on judgment, especially when changes in noise level are small or insignificant, as in the case of these test data. A 1 to 2 dB change in noise level is barely perceptible to the average person.

*Santa Clara Transportation Agency.* The Santa Clara Transportation Agency tested the Century Lubricating Oils HPF and LCF spring-loaded stick friction modifiers and found no rolling noise reduction. Testing was discontinued when the dry sticks broke or were lost after less than 2,000 mi.

*Sacramento RTD.* Wayside noise at 50 ft from a ballast-and-tie track were measured at the Sacramento RTD system. Train speeds were on the order of 50 mph. The results are presented in Figure 7-15, which indicates that wayside noise

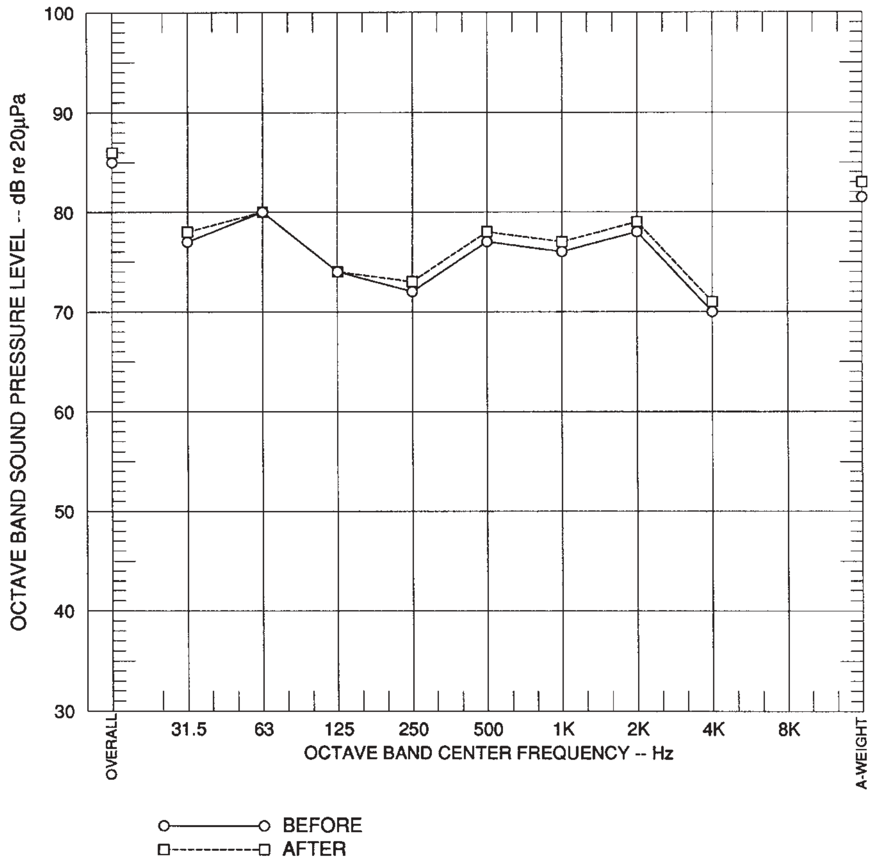


**FIGURE 7-14 WAYSIDE NOISE AT 50 FEET FROM BALLAST-AND-TIE TRACK AT BART HAYWARD TEST TRACK BEFORE AND AFTER APPLICATION OF SOLID-STICK LUBRICANT**

is on the order of 86 to 89 dBA at this section of track. Noise levels for trains at 50 ft from the adjacent track were about 83 to 85 dBA (not shown). A substantial pure tone component exists at about 630- and 800-Hz third octaves, related to rail corrugation. These results were obtained about 2 years after rail grinding with a Loram horizontal rail grinder. Major conclusions are (1) rail corrugation is not prevented by the HPF treatment, (2) corrugation is reemerging within a 2-year period after grinding, and (3) wayside noise levels are not necessarily reduced by the HPF dry-stick lubrication.

Measured wayside noise levels for systems with and without HPF treatment are compared in Figure 7-16. These data are for light rail vehicles traveling at between 45 and 55 mph on ballast-and-tie track. The data shown for Portland Tri-Met were obtained in 1989, several years after startup and prerevenue service grinding, without HPF dry-stick lubrication. The data shown for the Sacramento RTD and Los Angeles Blue Line were obtained within 2 and 1.5 years after grinding, respectively, with a Loram horizontal axis grinder and with at least 1 year of application of HPF dry-stick lubrication. (The data were collected in May and June 1995.)

The data obtained for the Sacramento RTD and Los Angeles Blue Line clearly show a pronounced peak at about 800 Hz due to a wave in the rail head, confirmed by data obtained at varying speeds. Visual observation suggests that the wave is minor rail corrugation. However, the rail grinder could conceivably have contributed to the wave as well, though one might expect that the grinding artifact would have been worn away with time. (The same horizontal axis grinder was used at the Sacramento RTD and Los Angeles Blue Line.)

During the measurement, wheel/rail howl was observed, further suggesting the presence of corrugation. The results do not indicate that the dry-stick lubrication is effective in preventing rail corrugation, nor that it is effective in reducing noise levels relative to those observed at the Portland Tri-Met system for trains running without dry-stick lubrication. This is not to say that HPF does not reduce noise or corrugation rates, only that these data do not indicate this. Tri-Met has also met with substantial corrugation noise prior to recent rail grinding efforts and experimentation with HPF dry-stick lubrication.

### *7.14.3.4 Costs*

The costs for friction modifiers vary. Metro-Dade County reports a capital cost of $80,000 and maintenance cost of $15,000 per vehicle per year for the Phymet solid lubricant

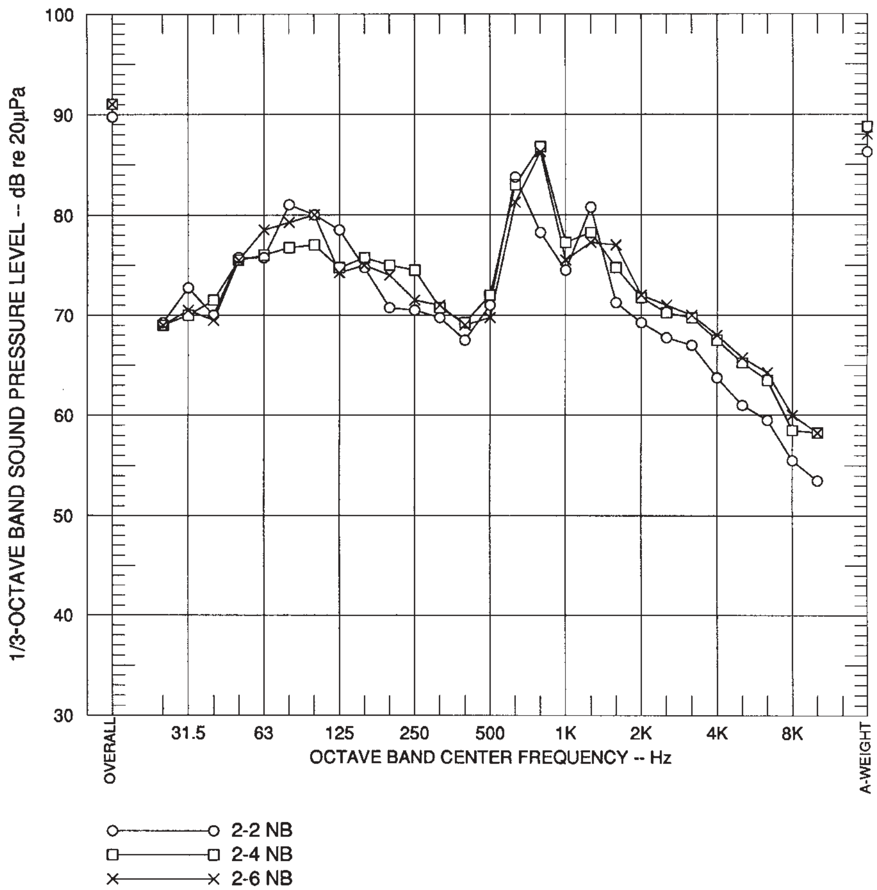


**FIGURE 7-15 TANGENT TRACK PASSBY NOISE FOR SACRAMENTO RTD TRAINS WITH RESILIENT WHEELS AND HPF DRY-STICK LUBRICATION**

system. Phymet has not been contacted regarding projected costs. However, these costs would be presumed to be competitive with those of other solid-stick lubricants.

The Bi-State Development Agency employs graphite lubricator sticks for wheel flange lubrication at the St. Louis MetroLink. The sticks are supplied by Kelsan Lubricants Ltd. at a capital cost of $25,000. This would presumably be the LCF flange lubricant.

WMATA indicated that onboard flange and tread lubrication was estimated to cost about $500 per truck per month, or about $12,000 per vehicle per year.

Kelsan Lubricants Inc. indicates that the cost of combined LCF and HPF lubrication includes a one-time cost of $800 to $1,000 for brackets plus perhaps 1 hour installation time, followed by a product cost of about $1,500 per vehicle per year, assuming four wheels per vehicle are lubricated. Product installation time is estimated to be about 10 min per vehicle during normal servicing, at 16,000- to 18,000-mi intervals. Based on a survey of various transit systems, the overall costs are $1,400 per vehicle per year for HPF tread lubricant and $1,200 per vehicle per year for LCF flange lubricant. Lubricant usage varies with degree of tread and flange roughness. Kelsan further indicated that users report that surface finish improves dramatically over time, with the result that stick usage should drop over time, and bears further investigation.

#### *7.14.3.5 Site-Specific Conditions*

There are no published data concerning site-specific conditions, though wet weather could conceivably wash much of the lubricant from the rail, thus inhibiting formation of the oxide/lubricant film over time. This may be a reason why the dry-stick lubrication experiment by Portland Tri-Met was inconclusive as of this writing.

No data have been obtained concerning the environmental hazards of dry-stick lubricant. According to one report, the HPF material consists of a polyester resin with molybdenum disulfide (*69*). The manufacturer of HPF indicates that the material is manufactured from polyester and rare earths.

An engineer at WMATA expressed a concern with respect to train signaling due to excessive resistance caused by tread lubrication. Similar concerns were identified by Kramer (*70*).

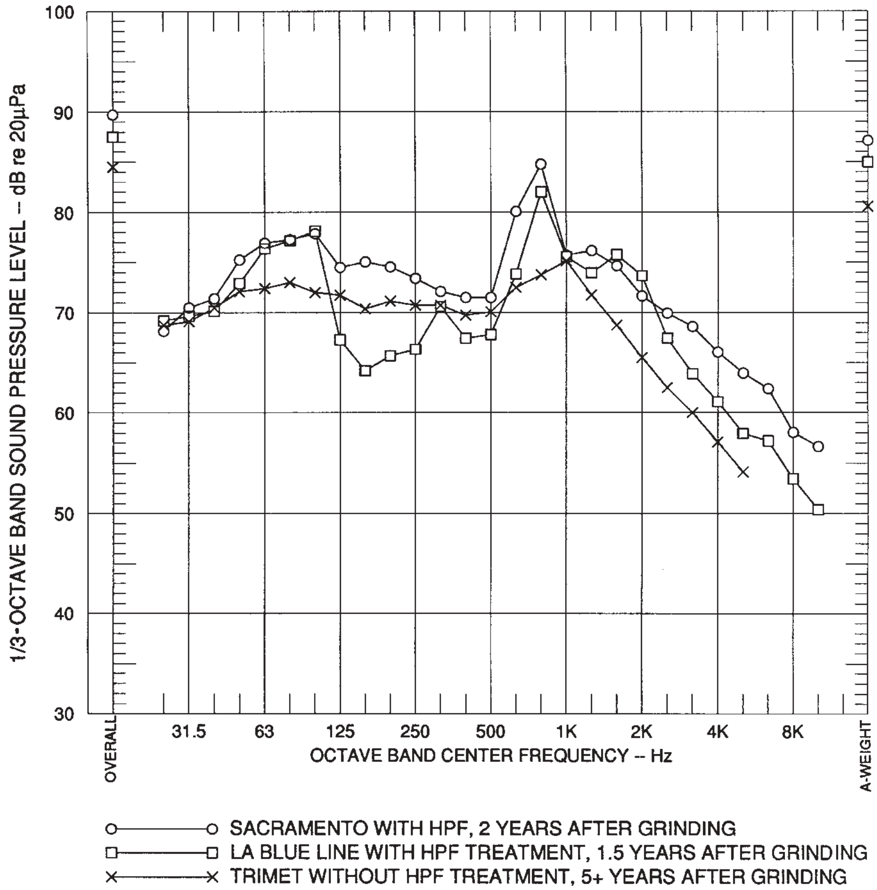


**FIGURE 7-16 COMPARISON OF WAYSIDE NOISE LEVELS FOR LIGHT RAIL TRANSIT VEHICLES ON BALLAST-AND-TIE TRACK WITH AND WITHOUT HPF SOLID LUBRICANT - RESILIENT WHEELS**

However, no difficulties with train signaling were reported by Los Angeles Blue Line or Sacramento RTD.

*7.14.3.6 Application Techniques*

Onboard application of dry-stick lubricants is accomplished by spring-loaded brackets which apply the stick lubricant directly to the flange and tread surfaces. The greatest difficulty experienced with applying dry lubricants as onboard treatment for wheel/rail noise control appears to be rattling noise from the brackets and failure of the brackets. Proper design of the brackets should eliminate these problems. Figure 7-17 illustrates the mounting brackets used at the Sacramento RTD and Los Angeles Blue Line. Both LCF and HPF flange lubricators are shown.

Care must be exercised during bracket installation to ensure that the brackets do not make noise, as was reported by some systems. The manufacturer should warrant that the brackets will not squeal or rattle during vehicle use.

## 7.15 CAR BODY SOUND INSULATION

An effective car body design is important for controlling car interior noise and patron and vehicle operator noise exposure. Although no data have been obtained, ridership may be affected by noise, and systems interested in attracting patronage and improving ridership can ill afford to have unnecessarily high car interior noise levels. Modern transit vehicles today are quiet compared with vehicles built before 1960. However, older vehicles are still in use, and, at some systems, vehicles may be operated in a subway with open windows, as with streetcars, thus exposing patrons to very high noise levels, sufficient to prevent conversation of any sort during passage through tunnels. Further, the noise from corrugated rail and wheel squeal in tunnels can be uncomfortable to patrons.

### 7.15.1 Vehicle Procurement Specifications

The primary purpose of reducing vehicle noise is to produce a reasonably pleasant environment for patrons, allow conversation with normal vocal effort, and reduce overall patron and operator noise exposure. The inclusion of vehicle noise criteria or standards as part of vehicle procurement specifications is one of the best methods of ensuring that noise exposure goals are achieved. To this end, the APTA guidelines discussed earlier in this manual are designed to ensure that state-of-the-art techniques are used to minimize

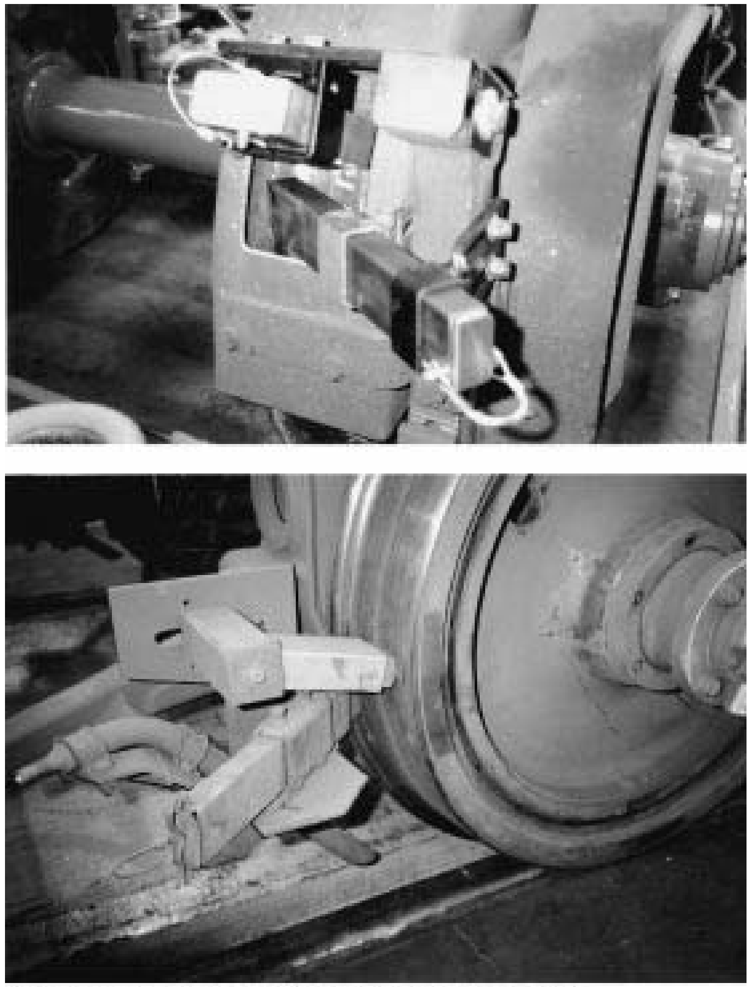

FIGURE 7-17 LCF AND HPF MOUNTING BRACKETS AT SACRAMENTO RTD (UPPER PHOTO) AND LOS ANGELES BLUE LINE (LOWER)

noise and create a reasonably pleasant interior noise environment. Meeting APTA guidelines means that at low speeds on ballast-and-tie track, the subjective rating of the interior noise is "quiet," while at high speeds in subways the level is "intrusive" but not annoying. The vehicle noise design goals given in the APTA guidelines are based on a balance between desirable and economically feasible noise environments. The noise limits provided in the guidelines are achievable and practical and are based on measurements of modern rail transit vehicle noise.

### 7.15.2 Sound Transmission Loss Requirements

Sufficient sound insulation or transmission loss of the car body floor, wall, and ceiling assemblies is necessary to achieve the design goal interior noise levels of the APTA guidelines. At speeds in excess of about 50 km/hr, noise from the air conditioning systems and other auxiliaries is dominated by propulsion system and wheel/rail noise. In subways, wheel/rail noise is normally the dominant source. Because both of these sources are outside of the car body shell, the transmission loss of the floor, walls, ceiling, windows, and doors determine the interior noise of the car, especially when operating at high speed. The transmission loss of the floor is most important for operation on at-grade ballast-and-tie track or concrete aerial structures, because the dominant noise sources are located beneath the floor, and sound is radiated away from the vehicle. The transmission loss of the walls, ceiling, windows, and especially the doors are most important for operation in subways, where the sound energy is confined to the train tunnel annulus.

The sound energy transmitted by mechanical vibration from the trucks to the body of a modern transit vehicle supported by air springs is normally far below the airborne sound energy transmitted through the car body elements. Therefore, enhancement of the transmission loss of the car body elements is one of the best available means of reducing car interior noise. Efforts directed at detecting body leaks and effectively sealing them are usually successful and desirable. Even on modern transit vehicles, door seals become worn, or they are removed during the course of maintenance. Replacement of door seals may be sufficient to return a transit vehicle to its original noise control performance.

A field sound transmission loss test is desirable to determine whether the transmission loss of the car body elements is sufficient to achieve car interior noise limits. The required sound transmission loss at each of four frequencies for the various car body elements are usually part of modern new car procurement specifications. The sound transmission loss test procedures are outlined in ASTM E 336-71, as revised, Recommended Practice for the Measurement of Airborne Sound Insulation in Buildings. (An example of sound transmission loss requirements is provided in Table 17 of that document, excerpted from the Chicago CTA transit car procurement specifications for the 2600 Series vehicles.)

### 7.15.3 Car Body Designs

The vehicle shell should be a nonhomogeneous sandwich barrier composed of two impervious barriers separated by a layer of sound absorbing material. The sandwich wall type provides superior transmission loss for the equivalent mass of the wall. The superior performance of the sandwich wall is now generally recognized and has been successfully incorporated into numerous transit vehicle designs. When the separation of the impervious layers is too small, or the areal density of one or both of the barriers is too low, there is a loss of sound isolation in the low- and mid-frequency ranges.

Figures 7-18, 7-19, and 7-20 illustrate the floor, wall, and ceiling constructions for the State-of-the-Art-Car (SOAC) tested at the Transportation Test Center in Pueblo, Colorado, and at various transit systems. A new transit vehicle utilizing these features for car body construction would meet the sound transmission loss requirements of the CTA specifications or other similar new vehicle specifications. The sound transmission loss effectiveness of these elements must not be reduced by flanking paths around windows, doors, ventilation ducts, floor penetrations for cables and pipes, through-wall penetrations for heating elements, and so forth. Flanking paths can be minimized by proper design and careful workmanship of window molds and door paths (effective brush seals) and proper sealing of any necessary penetrations through the inner liner or through the floor. Usually, sealing the floor penetrations is the most critical requirement. Older transit vehicles can be modified to significantly improve the interior noise environments. Possible improvements include sealing doors and windows and providing forced air ventilation or conditioning.

The life expectancy of the vehicle must be considered in the cost of any retrofit. If the vehicle is to be retired within 5 years or so, the cost effectiveness of acoustically improving the car is questionable. However, if there still is a long life expectancy or if the car is to be rebuilt for nonacoustical reasons, a significant improvement in the vehicle acoustics can be achieved at a relatively low cost. An extensive study done at the MTA NYCT to determine ways of quieting older subway cars showed that 10 to 15 dB noise reduction, in most cases, could be achieved inside the car by treating door seals, traction motor fan, and floor, for an estimated cost of less than $25,000 per car (1979 dollars), exclusive of air conditioning costs. However, noise from traction systems, ancillary equipment, and fans must be considered in designing an overall noise control strategy.

TABLE 7-5 SOUND TRANSMISSION LOSS REQUIREMENTS FOR CAR BODY COMPONENTS (72)

| OCTAVE BAND CENTER FREQUENCY | ENTIRE FLOOR | WALLS AND WINDOWS | CEILING OR ROOF | DOORS |
|---|---|---|---|---|
| 250 | 27 | 23 | 23 | 14 |
| 500 | 35 | 31 | 31 | 22 |
| 1,000 | 38 | 34 | 34 | 25 |
| 2,000 | 38 | 34 | 34 | 25 |

Note: The sound transmission loss shall be averaged over each characteristic section of the car body defined in the above table, and must include the influence of all sound energy which transmits through all weak areas such as apertures, door seals, air ducts, or openings for supply and return ducts.

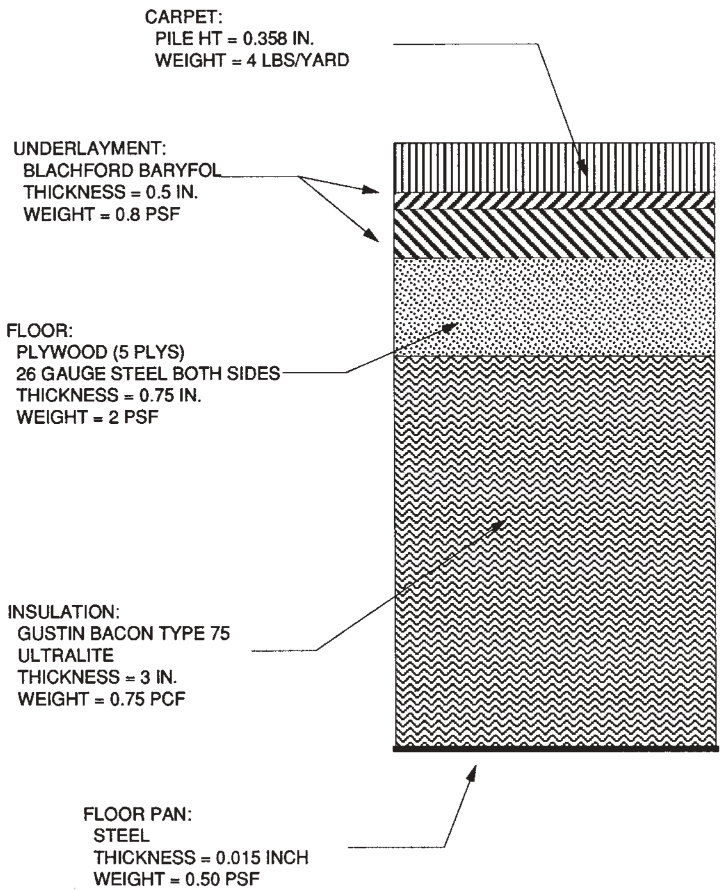


**FIGURE 7-18 CAR BODY NOISE REDUCTION FEATURES, FLOOR**

### 7.15.4 Doors

There are three types of doors typically used on rail transit vehicles:

- Sliding doors—The doors slide on an upper and lower track into and out of a pocket in the wall of the car.
- Bifold doors—Each of the two door parts is hinged in the middle and at the connection to the car wall, so that as they open or close from the center, the door simply folds out of the way against the car wall. The bifold door is not generally used in new cars due to access requirements for disabled patrons.
- Plug doors—Similar to the sliding door configuration, the doors slide on a track which takes them outside of the vehicle wall instead of sliding into a car wall pocket.

Plug doors are used on the SLRV and provide superior sound insulation characteristics due to their positive sealing nature, whereas both sliding and bifold doors allow sound leakage along the vertical edges of the door as well as at the door tracks. The sound insulation characteristics of the bifold door are less desirable than those of the other two configurations. Not only are there leaks about the periphery of the door, but the hinged joint also provides a sound transmission path. Further, the light weight of the bifold door usually does not provide sufficient sound insulation for a modern transit vehicle.

Door operation noise must be considered in the overall noise control of the transit vehicle. Sliding doors are the type in use at most newer transit systems. These doors are fast operating and are generally efficient and relatively quiet when moving. However, due to their size and weight, significant impact noise can be generated when the doors are unlocked and when the doors are closed. The bifold doors may also generate significant impact noise when unlocking or when closing, though they are typically quieter than the other two configurations.

An example of the importance of door seals in controlling noise between the doors of a BART transit vehicle is illustrated in Figure 7-21 (*71*). In this example, the A-weighted noise level between the doors of a 20+-year-old BART vehicle traveling in subway on ground rail with worn or nonexistent door seals was 91 dBA. Replacement of the seals reduced the noise level to about 85 dBA. Further, sealing the door joints and jams with aluminized duct tape reduced the

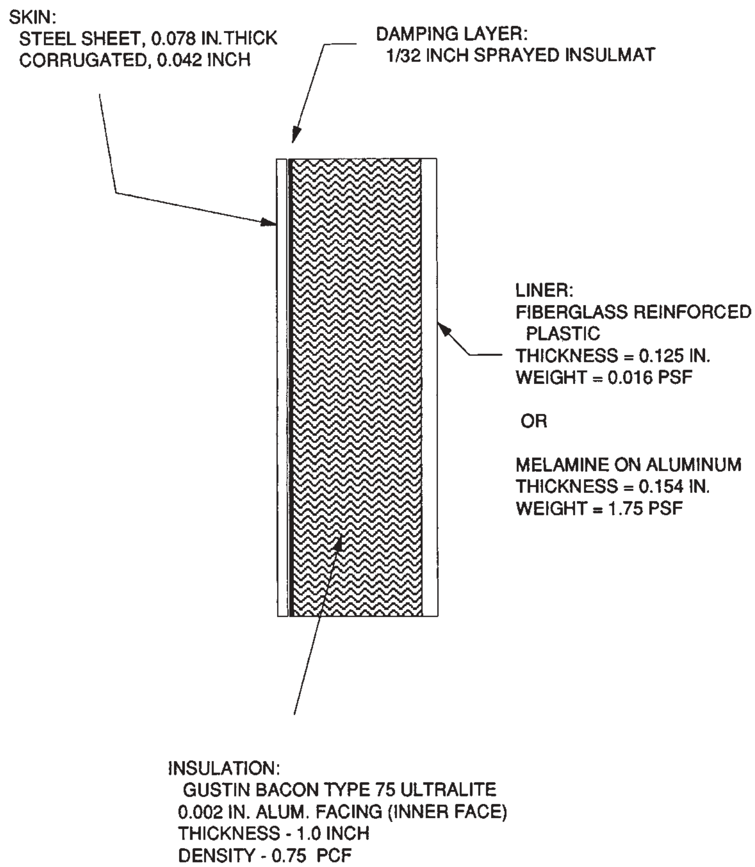


**FIGURE 7-19 CAR BODY NOISE REDUCTION FEATURES, WALL**

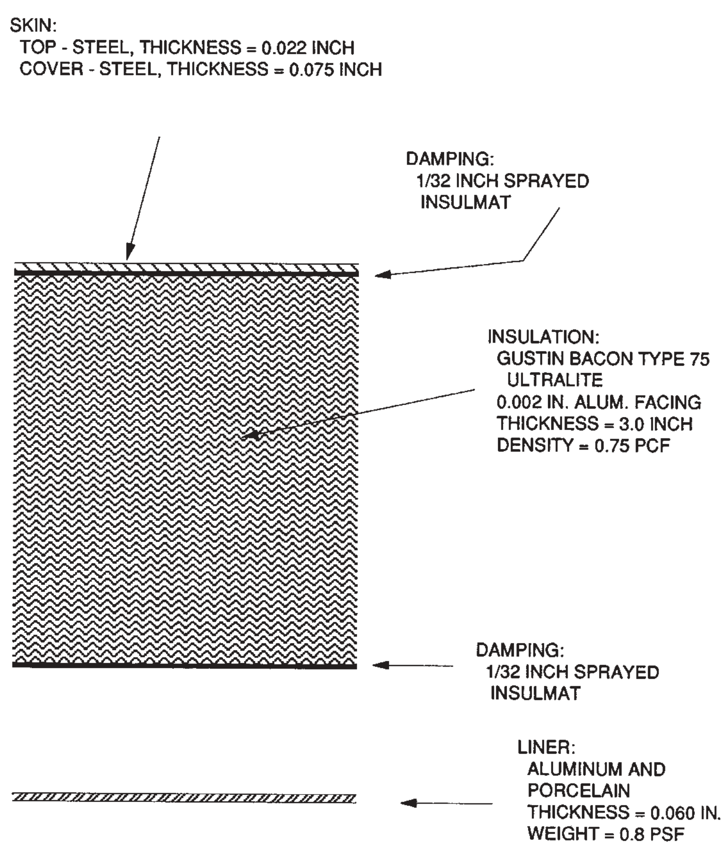


**FIGURE 7-20 CAR BODY NOISE REDUCTION FEATURES, CEILING**

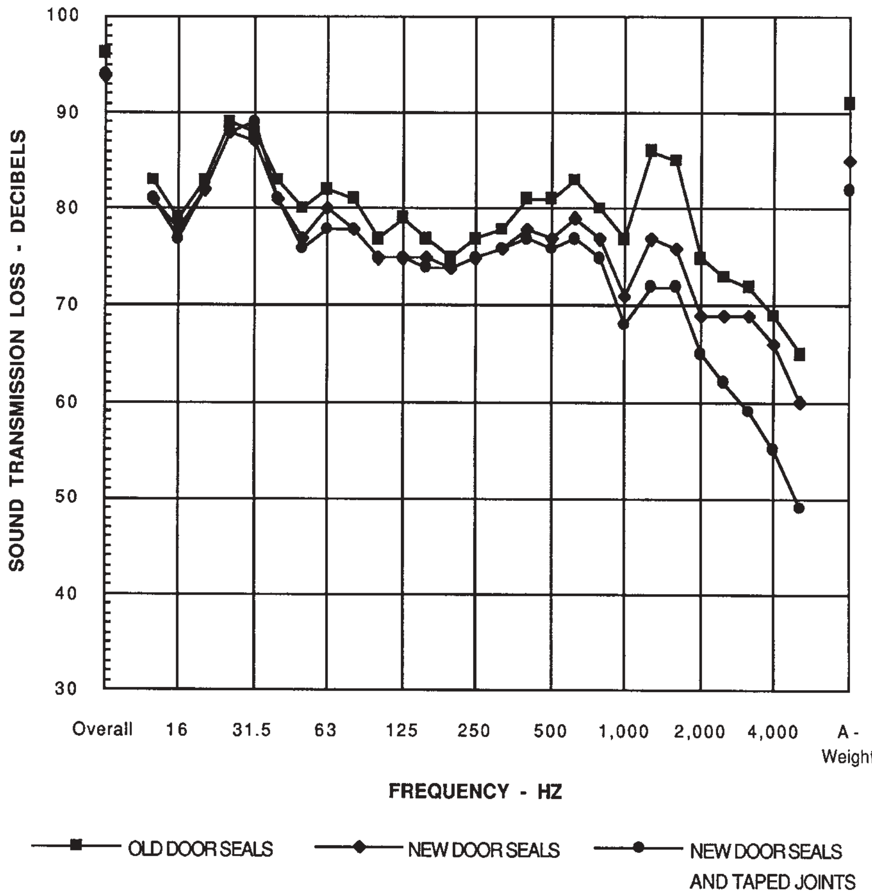


**FIGURE 7-21 NOISE LEVELS BETWEEN SLIDING DOORS OF BART CARS IN SUBWAY AT 66 MPH ON GROUND RAIL**

noise to about 81 dBA. These data clearly show the importance of door seals in maintaining an acceptable car interior noise level, both for new and existing vehicles. (The peak in the spectrum at about 1,250 Hz is due to a rail grinding pattern. Removal of this pattern would further reduce car interior A-weighted noise.)

## 7.16 REFERENCES

1. Fischer, H.M., "Noise Generation by Railroad Coaches," *Journal of Sound and Vibration,* Vol. 66, No. 3 (1979), pp. 333–349.
2. Kalousek, J., and K.L. Johnson, "An Investigation of Short Pitch Wheel and Rail Corrugations on the Vancouver Mass Transit System," *Proc., Instn. Mech. Engrs.,* Vol. 206 (1992).
3. Remington, P.J., *Wheel/Rail Rolling Noise Generation and Control,* TCRP Project C3, BBN Systems and Technologies, for Wilson, Ihrig & Associates, Inc. (September 1994).
4. Nelson, P. J., ed., *Transportation Noise Reference Book,* Butterworths, London (1987), Chapter 15.
5. Kurzweil, L.G., and L.E. Wittig, *Wheel/Rail Noise Control—A Critical Evaluation,* Bolt Beranek & Newman, Inc., for UMTA/U.S.DOT, Report UMTA-MA-06-0099-81-1, p. 81.
6. Dittrich, M.G., F.J.W. Biegstraaten, P.C.G.J. Dings, and D.J. Thompson, "Wheel Roughness and Railway Rolling Noise—The Influence of Braking Systems and Mileage," *Rail Engineering International,* No. 3. (1994), pp. 17–19.
7. Abbot, D.J., *Rail/Wheel Noise: A Rolling Stock Manufacturer's Perspective,* ABB Transportation Ltd., U.K.
8. Kurzweil, L.G., and L.E. Wittig, *Wheel/Rail Noise Control—A Critical Evaluation,* Interim Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT (January 1981), pp. 73–75.
9. Survey questionnaire.
10. Remington, P. J., N. R. Dixon, and L. E. Wittig, *Control of Wheel/Rail Noise and Vibration,* Final Report, Bolt, Beranek & Newman, Inc., for Transportation Systems Center/U.S. DOT, NTIS Report UMTA-MA-06-0099-82-5 (April 1983), pp. 289–317.
11. Remington, P. J., N. R. Dixon, and L. E. Wittig, *Control of Wheel/Rail Noise and Vibration,* Final Report, Bolt, Beranek & Newman, Inc., for Transportation Systems Center/U.S. DOT, NTIS Report UMTA-MA-06-0099-82-5 (April 1983), p. 309.
12. Suda, Y., and H. Komine, "Contact Vibration With High Damping Alloy," *Contact Mechanics and Wear of Wheel/Rail Systems,* Preliminary Proceedings, J. Kalousek, ed. (1994).
13. Duerig, T.W., K.N. Melton, D. Stockel, and C.M. Wayman, *Engineering Aspects of Shape Memory Alloys,* Butterworth-Heinemann, London (1990), p. 21.
14. Remington, P. J., N. R. Dixon, and L. E. Wittig, *Control of Wheel/Rail Noise and Vibration,* Final Report, Bolt, Beranek & Newman, Inc., for Transportation Systems Center/U.S. DOT,

NTIS Report UMTA-MA-06-0099-82-5 (April 1983), pp. 271–289.
15. Personal communication with J. Kremme, formerly of Raychem Corp., inventor of Nitinol wheel treads.
16. Kumar, S., P. Matic, A. Parmaksizglu, C. Sciammarella, and S. Rao Kilaparti, *Preliminary Investigation of Nitinol Wheel/Rail Steel Adhesion Creep and Wear Characteristics,* Final Report, Illinois Institute of Technology, for Raychem Corp., Report IIT-TRANS-78-5 (June 1978), 50 pp.
17. Quilter, R. J., and E. L. Van Sickel, "Quiet Progressive a Railroad Wheel," *Metropolitan* (March-April 1975), pp. 10–12.
18. Personal communication from W. Keevil of Chicago CTA.
19. Quilter, R. J., and E. L. Van Sickel, "Quiet Progressive a Railroad Wheel," *Metropolitan* (March-April 1975), pp. 10–12.
20. Saurenman, H.J., R.L. Shipley, and G.P. Wilson, *In-Service Performance and Costs of Methods to Control Urban Rail System Noise,* Final Report, Wilson, Ihrig & Associates, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-1 (December 1979).
21. Wilson, G.P., *BARTD Car 107 Noise Tests with Standard, Damped, and Resilient Wheels,* Wilson, Ihrig & Associates, Inc., for Parsons, Brinckerhoff Tudor-Bechtel (June 21, 1972).
22. Saurenman, H.J., R.L. Shipley, and G.P. Wilson, *In-Service Performance and Costs of Methods to Control Urban Rail System Noise,* Final Report, Wilson, Ihrig & Associates, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-1 (December 1979).
23. Rickley, E.J., and M.J. Brien, *Noise Test-Resilient Wheels Massachusetts Bay Transportation Authority Green Line,* Final Report, Transportation Systems Center, for UMTA/U.S. DOT, Report UMTA-MA-06-0099-82-4 (November 1982).
24. London Transport, Office of the Scientific Advisor, "Report on the Effectiveness of Noise Damping Rings on Train Wheels," *Investigations into Noise on Railways,* Laboratory Investigation 128/X215 (September 30, 1974).
25. Quilter, R. J., and E. L. Van Sickel, "Quiet Progressive a Railroad Wheel," *Metropolitan* (March-April 1975), pp. 10–12.
26. Nelson, J.T., Wheel/Rail Noise at the Los Angeles Blue Line, Technical Memorandum for TCRP Project C3.
27. Rutty, F. G. *Resilient Wheels—Experience of London Transport,* London Transport Report 8/74-c239 (revised September 13, 1974).
28. Personal communication with BART engineers, and personal knowledge of the author.
29. Personal observation of the author.
30. Nelson, J.T., Wheel/Rail Noise at the Los Angeles Blue Line, Technical Memorandum for TCRP Project C3 (1996).
31. *CLRV Noise and Vibration Study,* Final Report, Toronto Transit Commission, CLRV Noise and Vibration Study Working Committee (March 1983).
32. *CLRV Noise and Vibration Study,* Final Report, Toronto Transit Commission, CLRV Noise and Vibration Study Working Committee (March 1993), pp. 1–5.
33. Correspondence from Penn Machine to Wilson, Ihrig & Associates, Inc. (November 9, 1994).
34. Manning, J. E. *In-Service Evaluation of Four Types of Damped Subway Car Wheels,* U.S. DOT (draft).
35. Personal communication with W. Keevil of Chicago CTA.
36. Wilson, G.P., *BARTD Prototype Car 107 Noise Tests With Standard, Damped, and Resilient Wheels,* Final Report, Wilson, Ihrig & Associates, Inc., for Parsons, Brinckerhoff Tudor-Bechtel (June 21, 1972), p. 17.
37. Wilson, G.P., *BARTD Prototype Car 107 Noise Tests With Standard, Damped, and Resilient Wheels,* Final Report, Wilson, Ihrig & Associates, Inc., for Parsons, Brinckerhoff Tudor-Bechtel (June 21, 1972), p. 17.
38. Saurenman, H.J., R.L. Shipley, and G.P. Wilson, *In-Service Performance and Costs of Methods to Control Urban Rail System Noise,* Wilson, Ihrig & Associates, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-1 (draft).
39. Manning, J. E., *In-Service Evaluation of Four Types of Damped Subway Car Wheels,* U.S. DOT (draft).
40. Wolfe, S.L., *Noise and Vibration Test Report—2400 Series Rapid Transit Cars,* Final Report, Wilson, Ihrig & Associates, Inc. (June 1977).
41. CTA Ring Damped Wheel Noise Test Report, Boeing Vertol Interoffice Memorandum (May 19, 1975).
42. Wolfe, S. L., *Noise and Vibration Test Report—2400 Series Rapid Transit Cars,* Final Report, Wilson, Ihrig & Associates, Inc. (June 1977).
43. CTA Ring Damped Wheel Noise Test Report, Boeing Vertol Interoffice Memorandum (May 19, 1975).
44. Adtranz report concerning test results at WMATA West Falls Church yard (1996).
45. Barsikov, B., W.F. King III, and E. Pfizenmaier, Wheel/Rail Noise Generated by a High Speed Train Investigated with a Line Array of Microphones, *Journal of Sound and Vibration,* Vol. 118, No. 1 (1987), pp. 99–122.
46. Wilson, G.P., *BART Prototype Car 107 Noise Tests with Standard, Damped and Resilient Wheels: Ballast and Tie Tangent Track and Short Radius Subway Curves,* Final Report, Wilson, Ihrig & Associates, Inc., for Parsons Brinckerhoff Tudor-Bechtel (June 21, 1972), p. 14.
47. Hanson, C.E., M. Schafer, D. Towers, and K. Eldred, *Noise Control for Rapid Transit Cars on Elevated Structures: Preliminary Investigation of Vehicle Skirts, Undercar Absorption, and Noise Barriers,* Final Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-4, p. 13.
48. Hanson, C.E., M. Schafer, D. Towers, and K. Eldred, *Noise Control for Rapid Transit Cars on Elevated Structures: Preliminary Investigation of Vehicle Skirts, Undercar Absorption, and Noise Barriers,* Final Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-4, p. 33.
49. Kurzweil, L.G., and L.E. Wittig, *Wheel/Rail Noise Control—A Critical Evaluation,* Interim Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT (January 1981), pp. 78–79.
50. Hanson, C.E., M. Schafer, D. Towers, and K. Eldred, *Noise Control for Rapid Transit Cars on Elevated Structures: Preliminary Investigation of Vehicle Skirts, Undercar Absorption, and Noise Barriers,* Final Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-4.
51. Ban, Y., and T. Miyamoto, Noise Control of High-Speed Railways, *Journal of Sound and Vibration,* Vol. 43, No. 2 (1975), pp. 273–280.
52. Hanson, C.E., M. Schafer, D. Towers, and K. Eldred, *Noise Control for Rapid Transit Cars on Elevated Structures: Preliminary Investigation of Vehicle Skirts, Undercar Absorption, and*

*Noise Barriers*, Final Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT, Report UMTA-MA-06-0099-80-4, p. 32.

53. Kurzweil, L.G., and L.E. Wittig, *Wheel/Rail Noise Control—A Critical Evaluation,* Interim Report, Bolt, Beranek & Newman, Inc., for UMTA/U.S. DOT, Report UMTA-MA-0099-81-1, pp. 75–77.
54. Wilson, G. P., *Noise Levels From Operations of CTA Rail Transit Trains*, Wilson, Ihrig & Associates, Inc., for Chicago Transit Authority (May 1977).
55. Lee, P.Y., Wilson, Ihrig & Associates, Inc., Letter Report to K. Thorn-Olsen, Parsons, Brinckerhoff Tudor-Bechtel, concerning BARTD Prototype Car 107 tests with standard and soft rubber axle bushings (April 17, 1972).
56. Manufacturer's literature supplied by American Track Systems, Inc., Shrewsbury, PA.
57. Kramer, J.K., "Rail Lubrication: A Solid Future?", *Railway Track and Structures* (November 1994), pp. 31–33.
58. Century Lubricating Oils trade literature, provided by Kelsan, Inc.
59. Kramer, J.K., "Rail Lubrication: A Solid Future?", *Railway Track and Structures* (November 1994), p. 31.
60. TCRP Project C3 survey questionnaire return.
61. TCRP Project C3 survey questionnaire return.
62. TCRP Project C3 survey questionnaire return.
63. Personal communication with St. Louis MetroLink engineers.
64. Letter Report from Navcon Engineering to Century Lubricating Oils concerning acoustical measurements performed by Navcon at Sacramento, January 8, 1992 (obtained from Kelsan Lubricants, Inc.).
65. Nelson, J.T., *Wheel/Rail Noise at the Sacramento Regional Transit System,* Technical Memorandum on TCRP Project C3, Wilson, Ihrig & Associates, Inc., for Transit Cooperative Research Program (May 16, 1995), pp. 23–32.
66. TCRP Project C3 survey questionnaire return.
67. Vidan, J., B.C. Transit, private communication.
68. Wilson, G. P., Letter Report to Jerry McCleary, San Francisco Bay Area Rapid Transit District (October 29, 1993).
69. Letter Report to Jeff Meloy, Century Lubricating Oils, from M. Staiano, Staiano Engineering, summarizing the results of field tests at WMATA (January 2, 1992) (available from Kelsan Lubricants).
70. Kramer, J.K., "Rail Lubrication: A Solid Future?", *Railway Track and Structures* (November 1994), p. 33.
71. Jue, D.A., and J.T. Nelson, *BART A&B Car Rehabilitation Car Interior Noise Tests, Original B-Cars with New Doors,* Wilson, Ihrig & Associates, Inc., for Booz, Allen, Hamilton, Inc. (December 15, 1995).
72. Saurenman, H.J., J.T. Nelson, and G.P. Wilson, *Handbook of Urban Rail Noise and Vibration Control,* Wilson, Ihrig & Associates, Inc., for UMTA/U.S. DOT, DOT-TSC-UMTA-81-72, NTIS UMTA-MA-06-0099-82-1 (February 1982).

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