## CHAPTER 8 TRACKWORK TREATMENTS

CHAPTER **8**

# TRACKWORK TREATMENTS

## 8.1 INTRODUCTION

This chapter presents a discussion of trackwork treatments for wheel/rail noise. Trackwork treatments include those treatments that would be applied directly to the track or between the rails, as distinguished from application to the structure supporting the track; in this latter case, the treatment would fall under the heading of wayside treatments. Thus, aerial structure sound barriers would not be considered a trackwork treatment. Trackwork treatments include not only materials such as rail inlays, but procedures such as rail grinding.

The principal proven trackwork treatments include rail grinding, for which an extensive discussion summarizing the basic philosophy of profile grinding and how it applies to transit systems is provided. Wayside lubrication is of particular importance for controlling wheel squeal. Continuous welded rail is effective in reducing or eliminating rail-related impact noise. Other treatments include rail joint welding, vibration damping systems, certain types of resilient fasteners and special trackwork, and hardfacing. Dry-stick lubricants are discussed with respect to onboard treatments in Chapter 7.

## 8.2 RAIL GRINDING

Rail grinding combined with wheel truing is the most effective and important means for controlling wheel/rail noise and maintaining track in good working condition. With ground rail and trued wheels, tangent track wheel/rail noise is comparable with the combined noise from traction motors, gears, fans, and undercar air turbulence, and is usually acceptable to wayside receivers, provided they are not located in close proximity to the track. Indeed, when complaints concerning wheel/rail noise do occur, excessively rough or corrugated rail is usually involved, producing a harsh sound with identifiable pure tones.

A case in point is provided by the Los Angeles light rail system, which received complaints concerning wayside noise from aerial structure track located about 75 ft from the nearest homes. The track was part of the new Green Line, and, although the rail was ground prior to startup and passby noise levels were within design criteria, a periodic grinding pattern was evident in the rail, and there was some residual roughness that was not removed by the horizontal axis grinder. After grinding with a new profile grinder, wayside noise levels were reduced by about 6 dB, and members of the neighborhood expressed the opinion that the wayside noise was acceptable and no higher than that from automobile traffic passing on the boulevard between the neighborhood and residences. Thus, no extraordinary measures were needed at this location to satisfy the community. Enough cannot be said in favor of an effective rail grinding program, which must be accompanied by an equally effective wheel truing program.

### 8.2.1 Overview of Rail Grinding

Rail grinding is the removal of rail metal from the running surface of the rail head through the use of rotating grinding stones or wheels, or grinding blocks. The grinding stones are operated by high horsepower grinding motors mounted on rail grinding equipment, which may consist of multiple cars. Depending on size, as few as eight or as many as 120 grinding motors (and thus stones) can be mounted on the grinding equipment. Grinding blocks are mounted on special grinding cars and are simply run over the surface of the rail head at moderate vehicle speed. Most of the discussion presented below concerns rotary stone grinders.

Other grinding block systems use oscillating blocks but these have not been used to any extent in North America. They have, however, been used in Austria and other European countries.

The actual position of the grinding motor on the rail head is controlled by the angle of the grinding motor. By rotating the grinding motor with respect to the rail head, the contact band moves across the rail head. The relative position of this contact band or grinding “facet” (which is of the order of ¼ in. wide) is a function of motor angle. (Note: the relationship is not symmetrical about the rail head because of the cant of the rail.) Larger grinding facets have been observed, but accompanied by periodic grinding pattern in the facet, suggesting excessive grinding force and tool chatter. Further, using multiple, narrow facets produces a lower average roughness than would a single or double facet grinding profile.

The range of angles through which the grinding motors can rotate varies with machine type and design. Figure 8-1 illustrates three sets of angle ranges corresponding to three

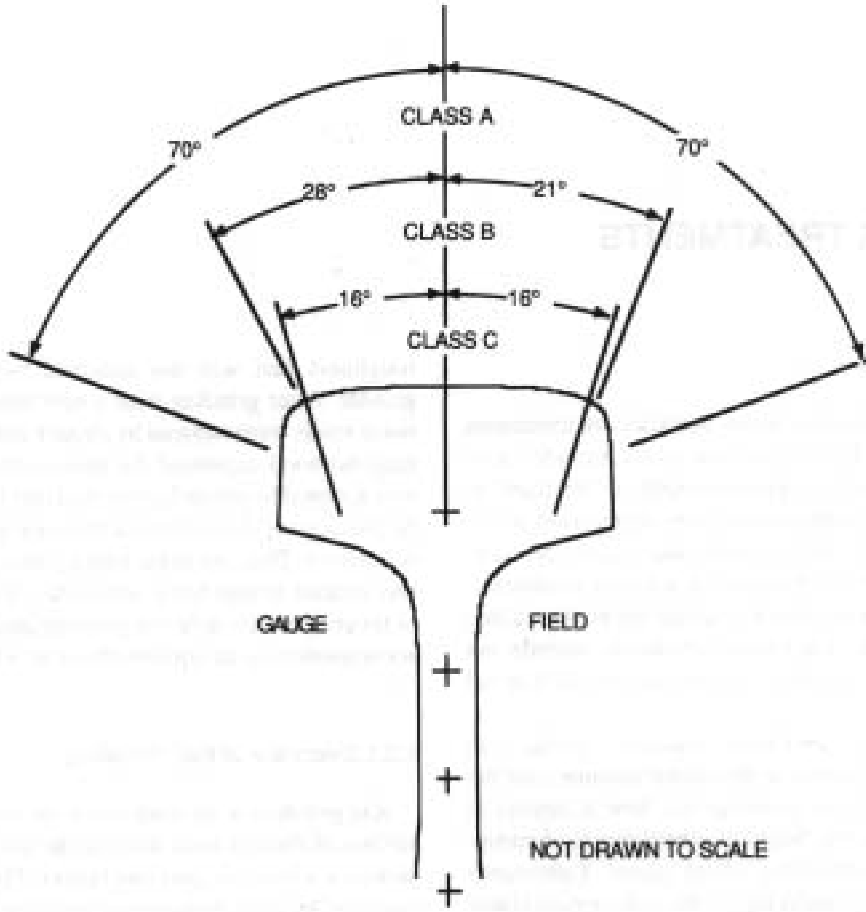


**FIGURE 8-1 RANGE OF MOTOR ANGLES FOR DIFFERENT EQUIPMENT CLASSES (ZETA-TECH, INC.)**

classes of grinding equipment. As can be seen from this figure, Class A has the widest range of motion and Class C, the narrowest.

Metal removal refers to the depth of metal removed by the grinding equipment in a single pass of the equipment, and is usually given in thousandths of an inch. The different factors that influence the amount of metal removed per pass by a grinding machine are

- Grinding power,
- Grinding speed,
- Power setting of individual motors,
- Location on rail head,
- Specific type of defect (for isolated defects),
- Grinding stone composition, and
- Effectiveness of grinding equipment with respect to active long wave grinding and automatic load control.

Of these parameters, the grinding power, equal to the number of grinding motors multiplied by the horsepower of the motors, is probably the most important. Figure 8-2 illustrates a direct correlation between grinding horsepower and metal removal. The largest machines, such as the 1960 horsepower and the 2400 horsepower machines, are used primarily on heavy freight railroads. Transit and commuter rail equipment usually have a smaller number of motors, with a lower total grinding horsepower in the range of 240 to 480 horsepower.

Rail grinding has been used by freight railroads and transit systems since the late 1930s for the elimination of rail surface defects. Those early applications used relatively unsophisticated rail grinding cars for the elimination of corrugations, engine burns, and batter at rail ends. Subsequent applications of rail grinding extended rail grinding to almost all types of rail surface defects to include corrugation, joint batter, weld batter, engine burns, flaking and shelling, and the grinding of mill scale from new rail.

Traditional defect elimination or rail “rectification” remained the primary focus of rail grinding from the 1930s until the 1980s. In recent years, traditional grinding practice has evolved from the defect elimination approach to the recently emerging rail “maintenance” or “preventive” grinding approaches. The latter approach does not allow surface defects to develop to any significant extent, but rather attempts to eliminate the development of these surface

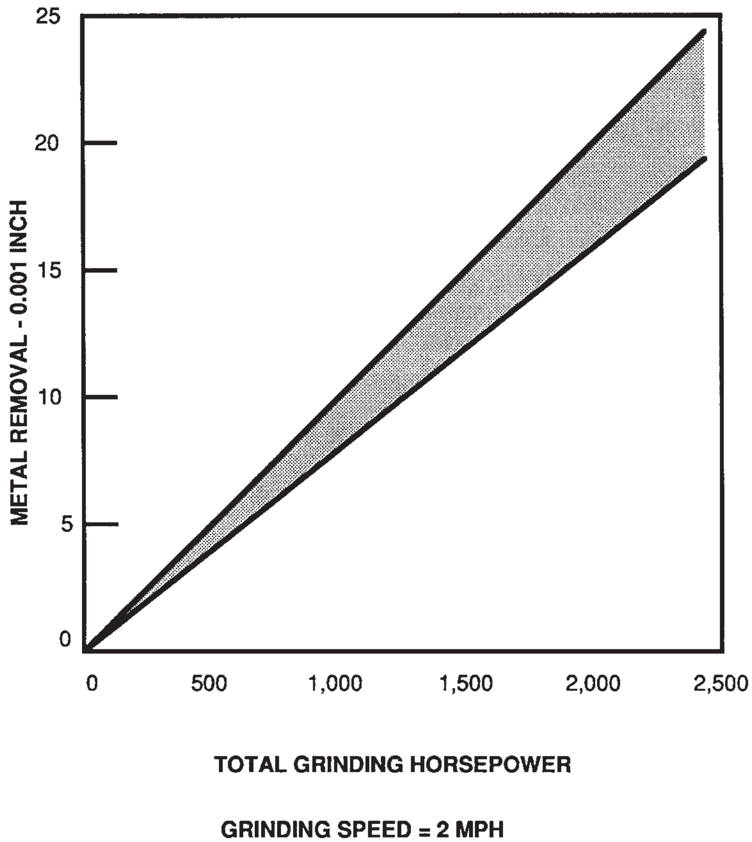


FIGURE 8-2 RELATIONSHIP BETWEEN GRINDING HORSEPOWER AND METAL REMOVAL (ZETA-TECH, INC.)

defects before they emerge on the rail head. The preventive grinding approach also makes extensive use of rail profile grinding techniques to control the shape of the rail head and the wheel/rail contact zones.

The evolution from traditional grinding to the emerging practices of profile control and maintenance grinding has resulted in a significant broadening of rail maintenance, and has introduced the potential for increasing the service life (and thus reducing the cost) of rail. Profile control and maintenance grinding have also led to significant improvements in wheel/rail dynamic interaction and the reduction of wheel/rail forces in both the vertical and horizontal planes. This reduction in dynamic interaction (and forces) results in improved ride quality, noise reduction, and reduced damage to both the track structure and the rolling stock. Specific improvements in rail and wheel performances and lives also occur.

Rail grinding, as required by transit systems, can be divided into two broad categories, based on the specific

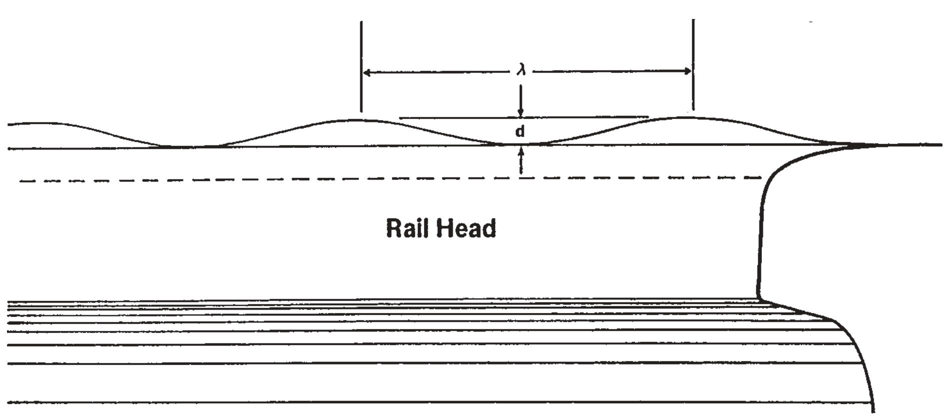


FIGURE 8-3 RAIL GRINDING WAVELENGTH

objective and method of achieving that objective. These will be described in the following sections.

### *8.2.1.1 Surface Defects (Level 1)*

Control and/or elimination of defects on the top surface of the rail head is a traditional area of rail grinding that has direct impact on transit systems. Because these surface defects represent locations where vertical wheel/rail dynamic activity is initiated, their elimination or control results in a reduction in noise, vibration, and vertical rolling and impact forces. While defect removal grinding has traditionally been one of the remedial type actions, i.e., elimination of defects after they appear on the rail head, earlier and more aggressive grinding have led to better control of this class of defects and the consequential reduction of their adverse impact on overall operations and costs.

Surface defects normally manifest themselves on the top central area of the rail head. Thus, this type of grinding usually requires relatively limited grinding motor positioning capability and pattern adjustment. The focus of this grinding is generally the removal of metal in this top central area of the rail head as illustrated by the Class C coverage of the rail head in Figure 8-1. These surface defect grinding applications include grinding of the following classes of rail surface defects:

*Corrugations*. A primary focus of grinding is the removal of rail corrugations with a special emphasis on roaring rail (very short wavelength or "short pitch") corrugations with wavelengths between 1 and 3 in. (Longer wavelengths will also require elimination, when encountered.) These corrugations will vary in depth from 0.005 in. (the smallest depth that can be measured with a traditional straight-edge and taper gauge) to 0.010 in. and greater, and they are located within a broad band at the center of the rail head (for roaring rail and long wave corrugations). In most cases, the traditional motor angle range of ±16 degrees is adequate to remove these defects, which can extend for long lengths of track.

For corrugations with a wavelength greater than 10 in. (the diameter of the grinding wheel) a long wavelength grinding system is required to effectively reduce the amplitude of the corrugations (Figures 8-3 and 8-4). This does not appear to be a significant type of corrugation on transit systems except that long wavelength corrugations have been identified with higher speed operations. Further, long wavelength corrugations will contribute to groundborne noise and vibration, and their removal should be included in any grinding program.

*Welds*. Weld grinding reduces wheel/rail impact noise and helps to control corrugation. Welds are ground either to remove high/low welds from the welding plant (see new rail) or to remove batter that occurs at the surface of the weld, generally at or near the fusion zone or the heat-affected zone of the weld. This occurs frequently near field welds, where the surface hardness can vary significantly from the parent metal, through the heat-affected zones and fusion zones. The depth of the batter can range from 0.005 in. to over 0.050 in., and its length from several inches to 36 in. (the standard length for measuring joints or welds). Note: a long wavelength grinding feature may be required to effectively grind this class of defects. Battered welds can extend (transversely) across the top of the rail head; however, they can normally be covered by a motor angle range of ±16 degrees.

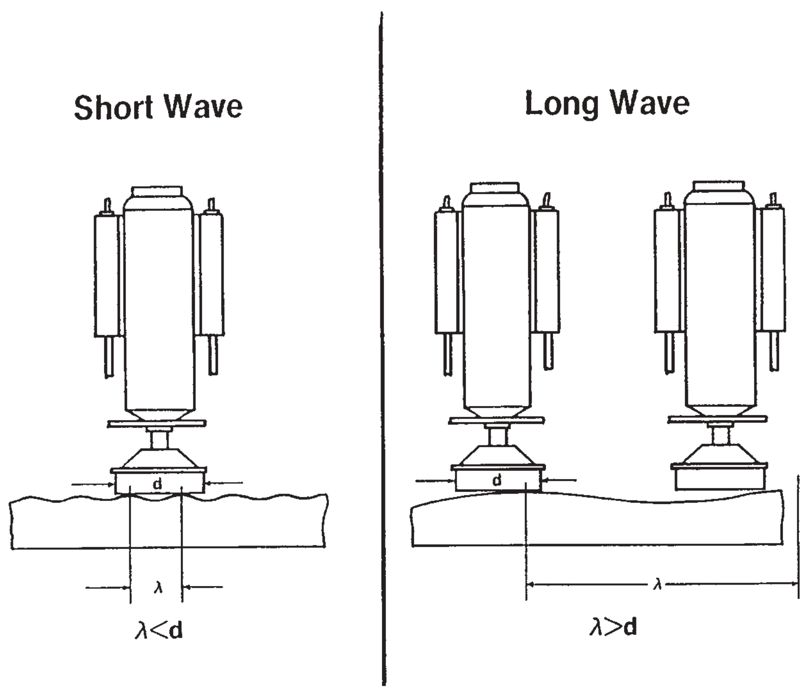


**FIGURE 8-4 EFFECT OF LONG WAVELENGTH CORRUGATION ON RAIL GRINDING**

*Battered Joints*. While joints are not a significant problem on transit systems (they are usually replaced by field welds), they do exist and they should be monitored for batter. Joint batter, which may contribute to unnecessary impact noise due to excessive gap and/or misalignment, is traditionally measured over a 36 in. length (18 in. on either side of the joint), and, as such, it has a long wavelength character, thus requiring a long wavelength grinding system for effective grinding. Joint batter can range from 0.010 to 0.250 in., although grinding out joint batter deeper than 0.050 to 0.060 in. is generally uneconomical. (Repair by welding is usually required for greater depths.) Battered joints usually extend across the top of the rail head, but they can normally be covered by a motor angle range of $\pm 16$ degrees.

*Wheel Burns*. Wheel burns may contribute to excessive impact noise. Wheel burns are not a major concern on transit systems, but they should be removed from the surface of the rail head. Wheel burns range in depth from 0.010 to 0.250 in., but grinding out wheel burns deeper than 0.050 in. is generally uneconomical. (Repair by welding is usually required for greater depths.) Wheel burns are normally found on the top central area of the rail head, and are normally covered by a $\pm 16$ degree motor angle range. A related problem concerns imperfections in the rail running surface due to repeated stopping of wheels at the same location, which may be exacerbated by automated train control systems and friction brake application at the end of deceleration.

*New Rail*. New rail is usually ground within the first several months of installation to remove rail surface mill scale, rail surface roughness, surface blemishes, and high/low welds from the welding plant. Mill scale is a very fine layer on the surface of the rail head, less than 0.010 in depth, and is usually removed by a light grinding pass. Rail welds are generally high (most railroad welding plant standards try to eliminate low welds, and consequently accept high welds), usually 0.005 to 0.020 in., and therefore rail grinding is very effective in their elimination.

Block grinding as practiced by the CTA and TTC involve moderate speed grinding passes with abrasive blocks. Relatively little material is removed with each pass, thus requiring multiple passes. An advantage of block grinding is that grinding trains may be interspersed between revenue service trains. The disadvantage of block grinders is that rail profile is not maintained, and a large number of passes may be required to remove even minor surface defects.

Horizontal axis grinders employ a grinding stone positioned so that the edge of the grinding stone contacts the rail. This type of grinder cannot grind a profile into the rail head as can a vertical axis grinder, though it can remove surface defects.

### *8.2.1.2 Profile Grinding (Level 2)*

Rail profile grinding refers to the method of controlling and maintaining the shape of the rail head (hence the term "profile") by grinding the head of the rail. Profile grinding goes beyond the basic defect removal approach of conventional grinding and addresses the control of the shape of the rail and the associated interaction between the wheel and the rail. Profile grinding is relatively new to transit properties, being done only at a few, such as Vancouver, MARTA (*1*) and Los Angeles. The MBTA has recently completed profile grinding on much of the Blue and Green Lines, with clearly positive results. Thus, profile grinding has not had extensive review with respect to noise control at U.S. transit systems, but it is gaining in popularity.

Shaping the rail head and influencing wheel/rail interaction are major differences between simple defect removal and profile grinding approaches. Traditional defect elimination grinding tends to flatten the rail, while profile grinding grinds a specific contour or profile into the rail head. (Contour grinding is used to restore the original shape or profile of the rail head, while profile grinding is used to give the rail head a special profile other than its original rail profile.) Through control of the rail head shape, the locations of wheel/rail contact, and thus the interaction between the wheels and the rail head, can be controlled. Profile grinding has a substantial influence on ride quality, by controlling truck hunting. Profile grinding has been used at the Vancouver Skytrain as a noise reduction feature to reduce wheel/rail conformity and spin-slip, resulting in lower rail wear and corrugation rates, and lower noise due to both rail surface degradation and spin-slip. Kalousek has suggested that spin-slip contributes to wayside noise, even without significant wear or corrugation (*2*).

Elimination of surface defects, if present, is the necessary first step in profile grinding. Thus, for rail with surface defects and plastic flow, profile grinding can be a three-step process:

1) The initial step consists of one or more grinding passes which eliminate any surface defects present.
2) The second step consists of one or more grinding passes which effectively reshape the deformed rail head.
3) The third and final step (if necessary) grinds the final rail head profile.

#### 8.2.1.2.1 Grinding at Curves

Rail profile grinding at curved track, as is currently practiced in North America, encompasses three general areas of rail maintenance:

- Control of gauge face wear (and lateral wheel/rail curving) of the high rail on curves and on tangents (as applicable).
- Control of short wave corrugations, on the low rail on curves.
- Control of gauge corner surface fatigue, to include both spalling and shelling, on the high rail on curves.

While all three purposes can be achieved through the proper use of rail profile grinding, they generally cannot all be addressed simultaneously. Thus, the profile that is best suited for the control of one of these maintenance areas may not be the best for the other two problem areas, even though it may be possible to derive benefit in all three areas by proper profile selection. Therefore, prioritizing rail problems for a given track location and selecting the optimum rail head profile for that problem location is necessary to get the greatest benefit from profile grinding.

The use of rail profile grinding to control wheel/rail interaction, wheel/rail contact, and (thus) rail wear at curves was developed and introduced by the mining railroads of Western Australia during the late 1970s. The focus of this research and subsequent implementation was the reduction of gauge face wear of rail on low and moderate curvature track, through the optimization of the "steering" of conventional three-piece freight car trucks. The results were the development of a set of asymmetric rail head profiles (i.e., asymmetric about the center line of the rail head), with a separate profile for the high and low rails of the same curve. In addition, for tangent track, where "hunting" wear was noted, special tangent profiles were developed to control this form of wheel/rail behavior, and the resulting rail head wear.

The initial profile grinding concept was designed to make use of the steering of the conventional three-piece freight car truck generated by the conicity of the wheelset. This generates a shifting of the wheelset ("outward") toward the outer or high rail of the curve, resulting in the outer wheel riding on the larger radius portion of its tread, and the inner wheel riding on the smaller radius portion of its tread. The difference between these radii, known as the "rolling radius differential," compensates for the difference in length, around the curve, between the high and low rails. In addition, this rolling radius differential generates a longitudinal creep force (which is, in fact, a partial wheel slippage in the longitudinal direction) which tends to align the axles into a radial position (with equally distributed misalignment of the two axles). The result of this is a degree of self-steering that reduces flanging on relatively shallow curves, and has the potential for eliminating flanging on curves less than 3 degrees (based on a 1:20 wheel conicity). This has a beneficial effect in reducing lateral creep and thus wheel squeal at curves. (Note: the effect of this self-steering is reduced when the conicity of the wheel set is reduced (i.e., for 1:40 wheels). (Systems such as BART and Chicago CTA which employ cylindrical wheels cannot benefit from this self-steering effect. Further, systems that do not true wheels sufficiently and have concave wheel tread profiles may not benefit from this self-steering effect.)

In order to maximize this rolling radius differential, and still maintain sufficient wheel/rail contact area to avoid excessive contact stresses, the profiling approach presented in Figure 8-5 was developed. This profile shifts the wheel/rail contact patch on the high rail toward the gauge side of the rail head. On the low rail, the contact zone is moved toward the field side of the rail head. In this manner the rolling radius differential is emphasized, and lateral forces generated by the three-piece truck while curving are reduced. As a direct result, lateral gauge face wear is also reduced.

Profile grinding to modify contact patch location and increase rolling radius differential is in a sense similar to gauge widening, but does not actually affect the gauge. Contact patch shifting may be considered by some to be preferable to gauge widening. Gauge widening will reduce wheel

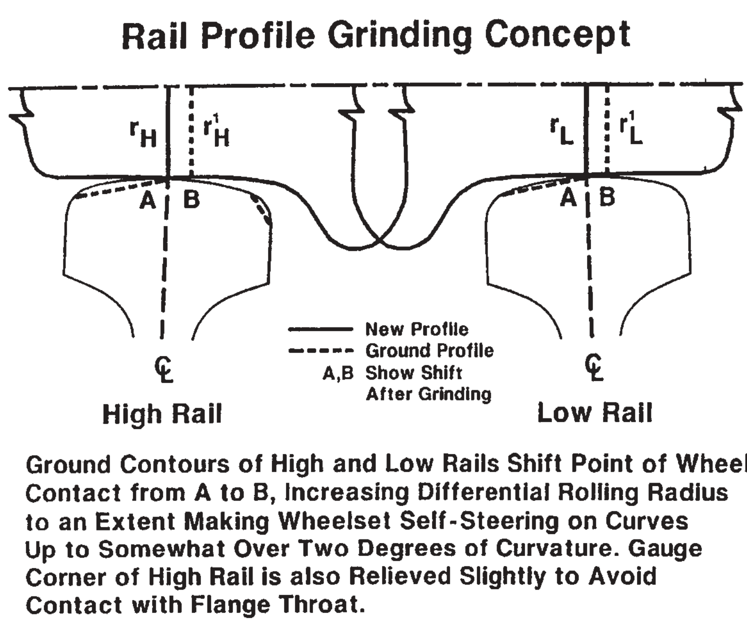


FIGURE 8-5 PROFILE GRINDING TO IMPROVE CAR STEERING

squeal for conically profiled wheel treads, by allowing the axle set to position itself such that the circumference of surface contact on the field side is larger than the contact line circumference on the gauge side. Tight gauge would force the contact patch locations close to the fillets of the flanges, limiting the ability of the wheel set to effect differential wheel rotation. However, tight gauge would introduce other problems, such as excessive flange and gauge face wear, since the truck must necessarily travel through the curve, with axles not perpendicular to the rail, so that tight gauge should not be a part of normal track design. Excessive gauge widening at short radius curves, however, will promote truck crabbing, and thus severe angle of attack and resulting flange wear, evidenced by a sharp gauge corner. Squeal also accompanies this type of curving performance, though it is not clear if the squeal is due to flange and gauge face rubbing, lateral creep, or both. There are some transit engineers who suggest that gauge widening is not desirable at curves because of increased lateral slip, provided that a tight gauge condition does not occur, and some have suggested that gauge widening is a hold-over from steam locomotive railroading days when very long wheel bases were involved. Thus, asymmetrical profile grinding to effect rolling radius differential appears to be preferable to gauge widening at curves. (However, see the discussion in this chapter concerning gauge widening.)

Rail profiles deteriorate with traffic. In one set of tests, the profiles lasted only 10 MGT (in non-lubricated heavy axle load freight operations) and were completely gone after 20 MGT of traffic. Profile life can be extended with head hardened, fully heat treated, or alloy rail. Further, rail profile life will be longer with light-axle-load rail transit vehicles than with heavy-axle-load freight. However, rapid gauge face wear occurs at light rail short radius curves, and may require reprofiling on a relatively frequent basis. Reducing crab angle to a minimum and preventing flanging by maintaining standard gauge throughout the curve may alleviate this problem if a tight gauge condition can be avoided.

#### 8.2.1.2.2 Control of Corrugations

A second area where benefit has been derived from profile grinding is in the area of corrugation control at curves. In the case of North American applications, this refers to the control of the heavy-axle-load short-wavelength corrugations found on the low rail of curves. These corrugations generally have *wavelengths in the range of 12 to 24 in.* on wood tie track. This is not usually the case for the roaring rail or short-pitch corrugations at transit systems, where the wavelength range is between 1 and 4 in.

These freight railroad corrugations are generally associated with high contact stresses generated when the false flange of a worn wheel runs on the field side of the low rail. This contact, which is counter-formal (i.e., the curvature of the two bodies in contact are opposite to each other), causes significantly higher wheel/rail contact stresses than the other (conformal) wheel/rail contact configurations. When this high contact stress is located near the field side of the low rail, severe plastic deformations and corresponding short wavelength corrugations can result.

Profile grinding has been used to control these short wavelength (“freight”) corrugations on North American freight railroads. By grinding the field side of the low rail to shift the contact point toward the center of the rail head, the high stress producing false flange contact is avoided. In recent tests on North American freight railroads, profile grinding to control the regrowth of corrugations was found to be significantly more effective in slowing regrowth of corrugation than conventional (defect elimination) grinding patterns. However, while this technique has been shown to be effective for control of freight type corrugations, there is no real evidence of its use or benefits for the control of corrugations typical of transit or passenger (high speed, lighter axle load) operations.

Again, the short wave freight corrugations referred to here differ in appearance (wavelength and amplitude distribution), and, perhaps, in initiation mechanism, from the roaring rail or short pitch class of corrugations most commonly associated with transit and passenger operations. (Control of short pitch corrugation via rail grinding is discussed in Chapter 10 to this report.)

A 16-stone profile grinder is preferable to an 8-stone profile grinder for grinding corrugations, due to the metal removal required. Production grinding to remove corrugations may require 20 passes with an 8-stone grinder, while a 16-stone grinder can accomplish the task in one-half the time (*3*).

#### 8.2.1.2.3 Control of Gauge Corner Fatigue

The third area of benefit associated with rail profile grinding is in the control of rail surface fatigue, and in particular fatigue defects at the gauge corner of the rail head. This includes both surface fatigue defects, such as spalling, and subsurface fatigue defects, such as gauge corner shelling, commonly found on heavy axle load freight operations. In a severe flanging condition, such as at a sharp curve, single-point contact between the throat of the wheel and the gauge corner of the rail will frequently result. *In the case of heavy-axle-load freight traffic*, this type of contact generates very high contact stresses in the region of the gauge corner of the high rail. These high stresses can result in gauge corner fatigue problems, including cracking and spalling.

To relieve these high contact stresses, grinding of the gauge corner of the rail can shift the wheel/rail contact points away from this corner and into a more central location on the rail head. The grinding required to shift this contact away from the gauge corner requires grinding on the gauge corner of the high rail. This grinding of the gauge corner can result in a decrease in both surface fatigue spalling and subsurface fatigue shelling, by wearing away the surface fatigue damaged rail steel, and relocating the point of maximum rail stress before fatigue damage can initiate a failure defect.

In the case of sharper curves, where flanging takes place, a second contact point between the flange of the wheel and the gauge face of the rail can occur, thus generating "two-point" contact between the wheel and the rail. This change in wheel rail contact, from one-point to two-point contact, can result in a deterioration in truck curving performance, and a corresponding increase in wheel/rail flanging forces. The result of this can be an increase in gauge face wear and, perhaps, noise, if no other action is taken. *Therefore, this type of gauge corner profile grinding should be used primarily in those areas where rail fatigue and not rail wear is the dominant rail failure mode.*

Single-point contact will result in the most efficient curving performance of the wheelsets and the lowest level of lateral wheel rail forces. Reduced wheel squeal might also be a possible benefit, though this has not been demonstrated. Thus, in the case of light-axle-load passenger operations, such as with transit systems, where fatigue failure at the gauge corner is not a problem, but rail and wheel wear is of primary import, *profile grinding for fatigue control through the initiation of two-point contact should be avoided.*

#### 8.2.1.2.4 Tangent Track

Reducing wheel/rail conformity by profile grinding can reduce corrugation growth rates on tangent track. Reduction of conformal contact across the rail head reduces the tendency for spin-slip behavior. Conversely, increasing conformal contact may average rail roughness over the contact patch, thus reducing the effective rail roughness (*4*). These are two conflicting arguments that need to be resolved. For the present, reducing conformal contact across the rail head appears to be most desirable from the standpoint of reducing rail corrugation rate.

Profile grinding is also used to control wheel wear, or rutting. The Vancouver Skytrain employs contour grinding to vary the contact patch location from one section of track to another, thus spreading the wear on the wheel surface. This results in lessened wheel/rail conformity which would otherwise result because of wheel wear. The LACMTA is using similar procedures on the Blue Line and Green Line.

#### 8.2.1.2.5 Noise Reduction Effectiveness

Rail grinding affects two causes of wheel/rail noise: (1) rail roughness due to surface defects and general surface roughness and (2) rail corrugation. General surfacing of the rail by grinding reduces the random roughness produced by general wear, and measurable broad band noise reductions may be obtained by reducing this roughness. For example, measurements at SEPTA (*5*) indicated that rail grinding produced small but consistent noise reductions on tangent track, limited by propulsion system noise, and little or no reduction of squeal noise at curved track. Rail grinding at continuous welded rail sections produced 0 to 2 dBA noise reduction at the wayside. However, there was little evidence of corrugation, pitting, or spalling which would be removed by grinding. Corrugation induced noise is one of the most severe forms of wheel/rail noise. Figure 8-6 compares noise levels for different rail corrugation depths and wavelengths with noise levels for uncorrugated rail. Even low corrugation amplitudes of 0.002 in. are capable of producing the characteristic roaring rail or wheel/rail howl, roughly 10 dB higher than noise levels for smooth ground rail (*6*).

Corrugated rail noise may be exacerbated by loss of contact between the wheel and rail, producing chatter and an irritating howling sound. The literature suggests that loss of contact occurs on high-speed systems when the corrugation depth exceeds about 40 microns, or about 0.0016 in., peak to valley. At lower speeds, greater corrugation amplitudes would be required to induce contact separation. These magnitudes are supported by rough calculations. For example, if an 800-lb wheel is displaced at a zero-to-peak amplitude of 0.001 in. at 500 Hz, the resulting acceleration of the wheel is about 25 g, and the reaction force of the wheel is roughly 20,000 lb, twice the static load for a typical transit vehicle wheel. Looking in the other direction, only 3.5 mil displacement of a 1-yd length of 115-lb rail at a frequency of 500 Hz would produce a dynamic load of 10,000 lb, comparable with the static contact load. These numbers are controlled by contact stiffness, rail bending stiffness, and masses, but their order-of-magnitudes indicate that small corrugation amplitudes are sufficient to overcome contact static load. Further, they are consistent with the evidence for the (nonlinear) relationship between noise level differences for corrugated rails with amplitudes of 0.002 and 0.004 in. and uncorrugated rail shown in Figure 8-6.

One might expect that corrugation rates would be highest when dynamic wheel/rail contact forces approach or exceed the static contact force. At low vertical contact force, lateral creep and abrasive wear of the rail surface may occur. Contact separation might also contribute to plastic flow and differential periodic work hardening. *Thus, the problem of maintaining wheel/rail contact and controlling corrugation rate is greater with transit vehicle wheel/rail static contact loads of 10,000 lb than for heavy haul freight wheel/rail contact loads of perhaps 25,000 to 30,000 lb. Rail grinding is thus more critical and necessary on a transit system than on a heavy haul freight system to control corrugation.*

Regardless of whether or not contact separation occurs, rail corrugation produces unnecessarily high and objectionable noise levels both at the wayside and inside the vehicle. The noise reduction achievable with rail grinding depends, of course, on the degree of wear and corrugation existing prior to rail grinding.

*BART*. BART is concerned about vehicle interior noise during passage through tunnels, specifically the Transbay Tube, and with wayside community noise. The rails in the Transbay Tube are subject to corrugation, with depths on the

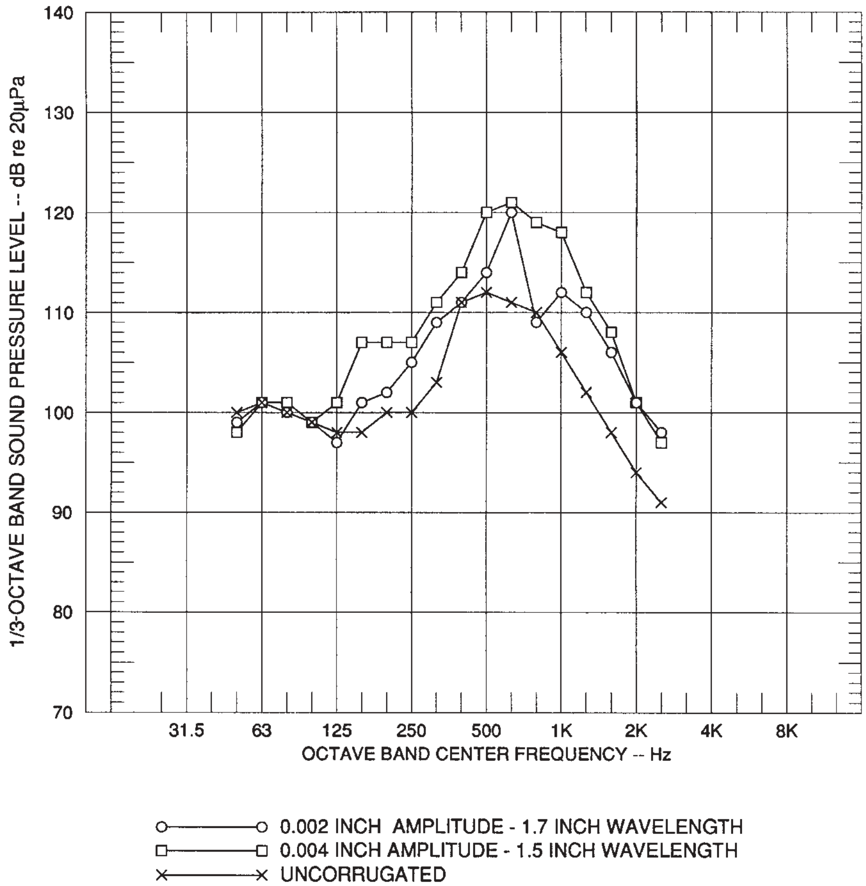


**FIGURE 8-6 EFFECT OF CORRUGATION AMPLITUDE ON WHEEL/RAIL NOISE**

order of 0.010 in. (questionnaire return) (*7*). Grinding frequency is low because of low track availability. Other subway sections are in similar condition. BART has made great progress recently in grinding sections of track, enabled in part by purchase of a Pandrol-Jackson grinder that can travel to and from grinding sites at reasonable speed, thus maximizing grinding time during nonrevenue hours.

Figure 8-7 illustrates the noise reduction at corrugated BART ballast-and-tie track due to rail grinding at Level 1, that is, grinding to remove corrugations. In this example, a very pronounced peak occurs in the 1/3-octave band spectrum at 500 Hz, which, coincidentally, corresponds to a wheelset resonance frequency. After rail grinding, the discrete frequency component was reduced by about 13 dB, resulting in a much smoother spectrum, without the harshness normally associated with corrugated rail. The A-weighted levels were reduced from about 98 dBA to about 79 dBA. With the elimination of the harsh sound due to the peak at 500 Hz, the qualitative effect of rail grinding is considerably greater than indicated simply on the basis of A-weighted noise reductions.

A second example of rail grinding effectiveness at a BART concrete aerial structure is provided in Figure 8-8. As at the ballast-and-tie section, the corrugation produced a substantial pure tone concentrated at the 500 Hz octave band. The grinding removed the corrugation, but introduced a short pitch roughness which produced a peak in the noise spectrum at about 800 Hz. An inspection by this author of BART's rails indicates a short wave periodic pattern on the rail head, which would produce the discrete frequency component at high frequencies for moderate train speeds. The pattern appears to be produced by the grinding operation, which includes grinding two large, ⅞- to 1-in. wide facets to remove surface defects, and then grinding a 45-degree bevel at either side of the rail head. Thus, no radius is provided at the rail head, relying, instead, on subsequent wear to improve the rail head profile. Further, the contact patch boundary may easily coincide with the edge between the bevel and the ground running surface. Where the 45-degree bevel is not used, the field side of the wear strip may run onto unground portions of the rail, and the edge of the strip may be irregular. Scratch marks and regular waves produced by tool chatter and debris accumulation by the grinding stone are identified as possible causes of whistling rail, though grinding induced noise is believed to be reduced to some extent by wear. SPENO is taking steps to identify and reduce or eliminate the cause of post-rail grinding whistle (*8*).

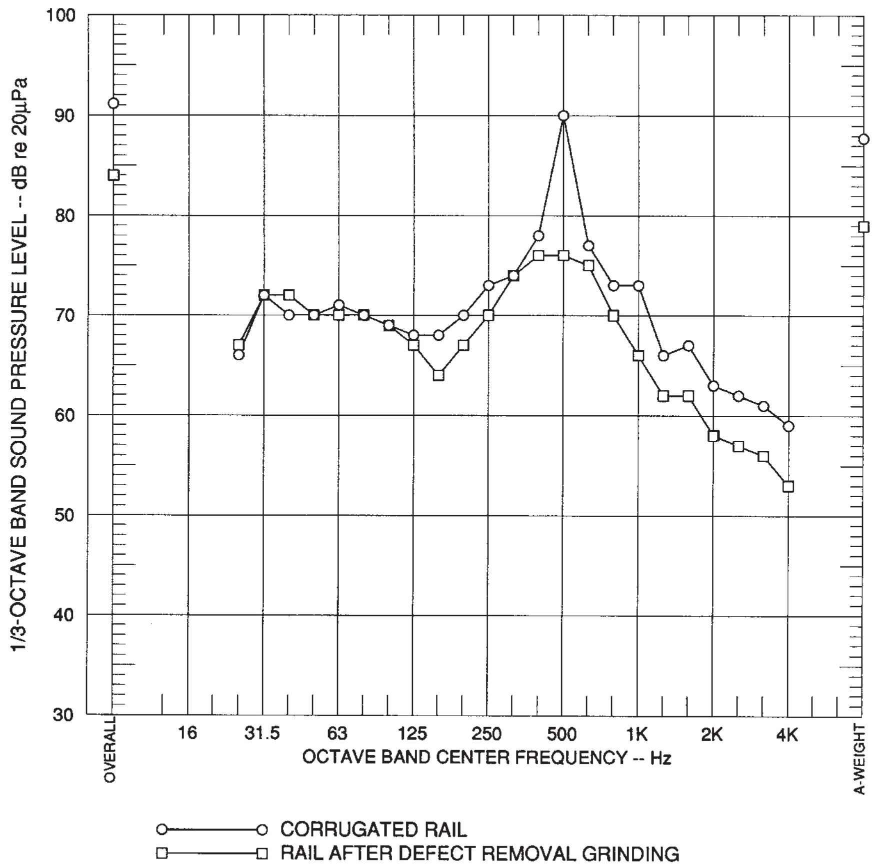


**FIGURE 8-7 WAYSIDE NOISE FROM BALLAST-AND-TIE TRACK BEFORE AND AFTER DEFECT REMOVAL GRINDING**

*Shinkansen (Japan)*. A periodic wave introduced by rail grinding is reported for the Shinkansen system in Japan (*9*). The wavelength appears to be about 1.3 in., similar in length to that at BART. Trains traveling at about 70 mph over this grinding wavelength would produce a discrete frequency component at about 1,000 Hz, also similar to the frequency component obtained at BART. Rail grinding at the Shinkansen line is done with SPENO grinding cars with rotary stones with 6 or 8 heads. Rail head defects are corrected as soon as discovered with a grinding machine which can treat 2 to 3 m of rail in one operation. Mainline sections of track are ground at 12- to 24-month intervals, and rails in noise sensitive areas are ground twice per year. Material removal is about 0.03 mm, or about 1 mil, per grinding pass. The grinder makes 3 to 5 round trips on the same section of rail, removing a total of about 0.3 to 0.5 mm, or about 10 to 20 mils. Projecting ahead for a 6-month grinding interval in noise sensitive areas, the grinding operations would evidently remove about 0.4 in. of rail in 10 years.

However, wayside noise levels at the Shinkansen increase at about 2 dBA per year, requiring removal of only about 50 μm, or 1.3 μin., every 12 months to maintain optimally low wayside noise. (Wayside wheel/rail noise levels at 25 m from the Shinkansen track for 230-km/hr trains are indicated to be about 65 to 66 dBA on ballast-and-tie track and 68 to 69 dBA on slab track, with a 2-m sound barrier wall and 60 kg/m rail. These noise levels are surprisingly low considering the speeds involved. The environmental noise limit standard is 75 dBA.) Rail welds are carefully ground to remove imperfections and thus impact noise (*10*).

*Portland Tri-Met*. Tri-Met has recently ground entire sections of track to reduce wayside noise levels due to corrugation. Specific areas of concern include ballast-and-tie sections where train speeds of 55 mph occur. Grinding was performed with a LORAM grinding machine with horizontal axis grinding stones, which remove corrugation and rail defects, but do not profile the rail head. Wayside noise levels were on the order of 90 dBA at 50 ft from the track center before rail grinding, and were reduced to about 80 dBA after grinding. However, a little over a year after grinding, noise levels increased to about 86 to 87 dBA, with re-emergent visible corrugation. Only a limited amount of material was removed, so that work hardening may not have been ground out. The only other prior grinding was before startup in the 1980s.

*MARTA*. MARTA is engaged in an “aggressive” rail profile grinding program. MARTA recently purchased a Fair-

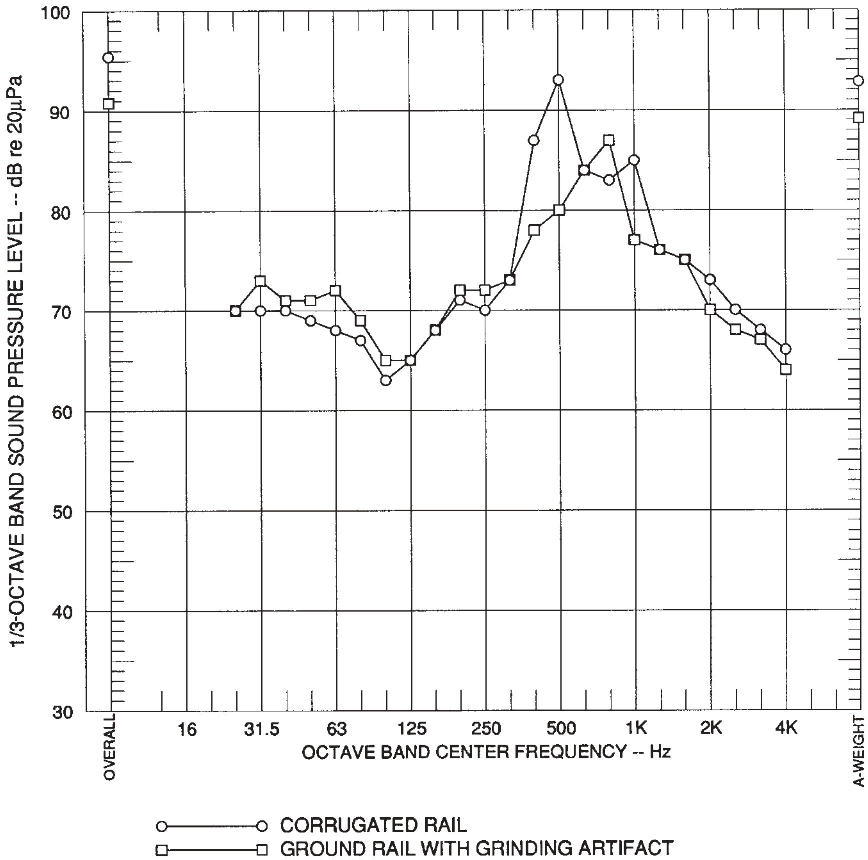


**FIGURE 8-8 EFFECT OF GRINDING ARTIFACT ON WAYSIDE NOISE AT 50 FEET FROM CONCRETE AERIAL STRUCTURE WITH DIRECT FIXATION FASTENERS**

mont RGH8 grinder with eight 20-horsepower hydraulic grinding motors. Grinding axes can be set and controlled from the operator's console, and a computer can be used to set up to 99 grinding profiles. Six-inch stones are used for turnouts and 10 stones are used for tangent track. Grinding is done at 80% of maximum horsepower for regular grinding, and 40% of maximum horsepower for polishing or finishing work. MARTA intends to remove about 0.002 in. of material from tangent track rails on a yearly basis. Material removal at turnouts is on the order of 0.005 in. per pass at the rail head and 0.015 in. per pass at the gauge corner. Grinding is performed to achieve total relief of the gauge and field corners and maintain a contact patch width of 1 to 1.5 in. MARTA can evidently maintain rail in good condition with relatively limited grinding of tangent track, no doubt because of its grinding program employed since startup. MARTA's first grinding was performed before revenue service to remove mill scale, rust, and minor mill defects. Subsequently, MARTA contracted to have about 0.002 inch ground off on an annual basis, though no profiling was performed. In 1987, MARTA experienced problems with spalling, and engaged in profile grinding with excellent results, and profile grinding has been employed ever since. As a result of MARTA's grinding program, rail corrugation has not emerged, even on direct fixation track where corrugation is often experienced, such as at BART. MARTA's continual attention to rail maintenance clearly illustrates the benefit in preventing rail corrugation and, by extension, excessive noise (*11*).

*WMATA.* WMATA grinds rail on an annual basis or whenever corrugation depths reach 0.010 in. (survey questionnaire). Discussions with one of the engineering staff at WMATA indicate that rail corrugation has not been a serious problem at most of the WMATA system. WMATA owns a LORAM horizontal axis grinder. Defect removal rail grinding appears to be effective in controlling rail degradation and wear, even though the rail is not re-profiled.

*TTC.* The Toronto Transit Commission regularly grinds with block grinders at 6-week intervals (survey questionnaire). Grinding operations are performed at heavy rail subway sections, embedded light rail track, and the Scarborough automated guideway system. Rail corrugation is evidently under control.

*CTA.* The Chicago CTA employs block grinding techniques between revenue trains. Block grinders are reputed to provide the smoothest of rail surfaces, and thus the lowest noise, compared with rotary stone grinders (*12*). However, block grinders are incapable of profiling the rail, and grinding must be conducted often, though grinding may be performed at high speed.

*LACMTA*. The Los Angeles County Metropolitan Transit Authority has recently procured a Fairmont Tamper profile grinding machine. Noise reductions achieved with the new grinding at the Blue Line were 12 dB on rail that had been ground within the 3 years prior to profile grinding. Noise reductions of about 6 dB were obtained at the Green Line at a location where vigorous community reaction was experienced as a consequence of a pure tone in the wayside noise spectrum, even though the rail had been ground prior to start-up by a horizontal axis grinder. The community complaints were received immediately after start-up, in spite of the fact that wayside noise levels were within the design criteria, and in spite of the presence of an active railroad and a 4-lane boulevard between the residences and the Green Line. After profile grinding, the residents indicate that the noise is no higher than the local automobile traffic and acceptable. This is a success story that should not be minimized, since additional and relatively expensive site specific treatments were being considered prior to profile grinding.

#### *8.2.1.3 Rail Grinders*

Following below are examples of commercial rail grinding manufacturers.

*LORAM*. The LORAM Rotra grinder is a grinder (LR Series) with horizontal axis rotary stones, capable of removing surface defects and corrugation, but incapable of profile grinding. LORAM also provides vertical axis grinders capable of profile grinding. The LORAM LR grinder can grind short radius curves and embedded track, which is particularly useful at light rail systems such as Tri-Met, LA Blue Line, and others. (Loram Maintenance of Way, Inc., Hamel, MN 55340)

*Pandrol-Jackson: Production Transit Grinder*. This is a vertical axis rotary stone grinding train capable of removing both rail defects, corrugations, and reprofiling, and is capable of grinding switches and frogs. The grinding train consist may be configured with between 8 and 24 stones. Up to 7 transverse profiles can be ground under computer control. Computerized longitudinal and transverse profile measuring systems are provided, with the ability to overlay transverse profiles with desired grinding profiles. Wavelengths up to roughly 6 ft can be removed, which is particularly important for groundborne noise and vibration control. The grinder can travel to the grinding site at 50 mph on certain grades. Pandrol-Jackson also offers a software package called Graphical Grinding Plan Software (Pandrol-Jackson, Inc., Hoffman Estates, IL 60195).

*Fairmont Tamper*. Fairmont Tamper offers a variety of grinders. The Model RGH8 (used by MARTA) is called a Precision Maintenance Grinder, which restores rail head profiles and grinds switches, turnouts, grade crossings, and other areas not ground by larger grinders. The grinding speed is 6.5 mph. The grinder has 8 independently positioned stones. A second grinder may be connected to form a 16-stone grinder operated by a singe person, which reduces the grinding time required to remove a certain amount of metal. Options include embedded track capability, pattern storage and retrieval for up to 99 patterns, and a “rail corrugation recorder system” that displays grinding data before and after grinding. (Fairmont Tamper, Cayce-West Columbia, SC 29171-0020)

SEPTA has developed a split truck design for negotiation of curves with radii as low as 33 ft in transit, though its minimum curve radius grinding capability has not been determined (*13*).

#### *8.2.1.4 Economics of Rail Grinding*

A primary benefit associated with increased grinding is a significant reduction in the rate of wear of the rails in track. This benefit is due to profile grinding which controls the steering of the wheelset around a curve. Improving wheel/rail contact geometry and steering can significantly reduce rail (and also wheel) wear. A second benefit is also associated with the reduction and/or elimination of rail surface defects such as corrugations, pits, spalls, battered welds, etc. These defects have the effect of reducing rail life, particularly if they get so deep that they can not be removed. Defects cause excessive wayside and interior noise, and excessive vibration of the trucks and truck mounted equipment. While profile grinding can be used to reduce these defects, they can also be removed by conventional “defect” grinding techniques.

Additional benefits associated with profile grinding include a reduction in wheel wear rate which is commensurate with a reduction of rail wear rate. In addition, elimination of corrugations and surface defects will provide benefits in the area of surfacing, energy (fuel) consumption, and maintenance. These latter benefits are associated with the reduction in wheel/rail vibration and impact associated with the elimination of rail surface defects.

The analysis presented here is for the economics of conventional (defect) grinding for the elimination of rail surface defects (corrugations, etc.) only. For the purpose of this economic analysis, grinding will be accomplished through the use of conventional grinding machines such as a 20-motor, fully adjustable self-propelled grinding machine of the type being increasingly used in transit environments. The analysis is conservative, since it will compare *no grinding* to two levels of grinding: conventional (defect) grinding and profile grinding (which also includes reduction in wear of the rail and wheel). The analysis is based on that performed previously for a major transit system (SEPTA).

For the purpose of this analysis, three levels of grinding will be analyzed:

Level 0: No Grinding
Level 1: Defect Grinding: Elimination of corrugations and other surface defects
Level 2: Profile Grinding: Control of wheel/rail contact and wear as well as surface defects (thus this includes benefits beyond that associated with corrugation elimination).

The analysis performed was on 79 mi of curved track on a large transit system which was analyzed as a function of traffic density ranging from under 1 MGT per year (corresponding to less than 200 cars per week) to a maximum of 7 MGT per year (corresponding to several thousand cars per week). The average density is approximately 3.5 MGT (or approximately 1000 cars per week).

Based on the above densities, and a detailed breakdown of curvature, the following annual rail replacement needs were determined for the curved track on the system. (Note: tangent track was *not* considered in this analysis since grinding will not significantly affect the rate of replacement of tangent track, except where tangent track noise due to corrugations and surface defects is a factor. Thus, the analysis presented here is conservative and *underestimates* the savings associated with rail grinding on tangent track where corrugations or other surface defects are present. The rail replacement requirements presented here are not the full system requirements but rather the curve requirements only.)

The following analysis assumes a low to moderate level of lubrication on these curves.

Annual Curve Rail Replacement Requirements; Curve Only Division: (total standard and premium rail)

| | |
|---|---|
| Grinding Level 0: | 3.3 mi |
| Grinding Level 1: | 2.3 mi |
| Grinding Level 2: | 1.8 mi |

Thus, a conventional (defect) grinding program can save approximately 1 mi of rail per year, and a profile grinding program can save 1.5 mi of replacement rail each year.

Converting these mileages into costs: for a conventional level of grinding, rail only savings, amount to annual savings of approximately $240,000. For profile grinding, savings associated with rail only, amount to approximately $370,000 per year. (Note: these savings do not include any wheel cost savings, surfacing savings, or fuel (energy) savings.) If only a moderate level of wheel life extension is claimed for profile grinding (a very conservative set of values), then savings of between $155,000 and $290,000 per year could be achieved, based on the level and extent of grinding. Additional savings, associated with a reduction in vibration and impact to the track, translates into reduced surfacing requirements on the corrugated track (or track with other surface defects). Likewise, elimination of surface defects reduces the energy loss in the truck suspension system associated with the vibrations induced by these surface defects. Reduction of vibration of trucks and truck mounted equipment may reduce maintenance requirements and equipment failures. The costs (and thus savings) associated with these areas depend on the level of severity of the surface defects. Analysis of two levels, moderate surface defects and severe surface defects, generates savings as shown in Table 8-1. Thus, total savings could range between $400,000 and $650,000 per year. To this, one must add the non-monetary benefit of reducing wheel/rail noise impacts on the community and transit patrons.

Note that grinding costs and savings vary by type of grinder, local labor rates, materials, available track time, and other factors which cannot be considered here. Each transit system must review costs prior to implementation or modification of a rail grinding program.

However, there is a cost associated with providing effective grinding, which will be briefly reviewed here. Rail grinding can be provided by a full service grinding contractor, or by the railroad's own forces using a purchased rail grinding machine. This analysis will be based on a full service rail grinding contractor providing a 20-stone-self-propelled rail grinder with fully adjustable motors, such as would be most effective for profile grinding. (Note: if the grinder used does not have fully adjustable motor capabilities, then the cost of profile grinding can be significantly higher due to the need to constantly adjust motor positions, which is a time and labor intensive activity for grinding motors that are manually adjustable.)

Noting that the pass-mile requirements for the curved track only are between 360 pass miles (defect grinding) and 540 pass miles (profile grinding) per year, the cost of grinding can be determined based on availability of track time for the grinder. Assuming 4 hours of working time (spark time) exclusive of travel and clearance time, a cost per pass mile of approximately $500 can be achieved using a contract service. This translates into an annual grinding cost of $185,000 (defect grinding only) to $277,000 (profile grinding). These costs are based on having a defined rail grinding program on hand when the contract grinder is available and having a fully defined pattern, program, and all support functions on hand.

These costs and savings are summarized in Table 8-2. As can be seen from Table 8-2, the net savings due to grinding

**TABLE 8-1 SAVINGS ASSOCIATED WITH LEVEL OF GRINDING**

| | Moderate Defects | Severe Defects |
|---|---|---|
| Surfacing Savings | $ 2,000 | $ 4,800 |
| Energy Savings | $ 3,000 | $ 6,000 |

**TABLE 8-2 ECONOMICS OF RAIL GRINDING FOR VARIOUS LEVELS**

| | Defect Grinding - Level 1 | Profile Grinding - Level 2 |
|---|---|---|
| | Annual Savings/Cost | Annual Savings/Cost |
| Rail Savings | $243,079 | $ 367,119 |
| Wheel Savings* | $ 0 | $ 293,091# |
| Surfacing Savings** | $ 4,806 | $ 4,806 |
| Energy (Fuel) Savings** | $ 6,310 | $ 6,310 |
| Total Savings | $ 254,195 | $ 671,326 |
| Cost of Grinding | $ 184,500 | $ 276,750 |
| Net Savings | $ 69,695 | $ 394,576 |
| Return on Investment | 38% | 143% |

* Savings from profile grinding only

** Savings from corrugation/surface defect control (also achieved by profile grinding)

\# Maximum savings

NOTE: This does not include savings on tangent track with corrugations.

varies significantly, based on whether conventional or profile grinding is performed. For conventional grinding, the net savings is on the order of $70,000 per year corresponding to a return on investment (ROI) for grinding of approximately 38%. For profile grinding, the net savings ranges between $250,000 and $400,000 per year. This corresponds to an ROI of between 90% and 143%. Finally, if rail savings *only* were considered, the net savings due to grinding would still be between $60,000 (defect grinding) and $90,000 (profile grinding) per year, corresponding to an ROI of approximately 33%.

Grinding costs at SEPTA include a capital cost of approximately $700,000 to $800,000 for an 8-stone profile grinder with computer control. In addition, a capital cost of $200,000 should be added for an environmental car which vacuums up the grinding dust and debris. Operating costs include labor and benefits for a grinder, mechanic, foreman, and two flagmen, plus material costs for roughly two sets of grinding stones per 4 hours of grinding. With this configuration, SEPTA is able to grind an average of 500 ft of track per day, including multiple passes (*14*).

#### *8.2.1.5 Limitations*

Care must be exercised in grinding wood tie-and-ballast or wood tie track on aerial structures to ensure that fire is not caused. The practice at some systems is to follow the grinder and douse fires that may occur. On these systems, less intensive grinding may be appropriate. Some grinders may have difficulty negotiating curves in tunnels, or may be unable to grind rail at very short radius curves. Vertical axis grindings may be unable to grind embedded girder rail without special provision.

## 8.3 GAUGE WIDENING VERSUS NARROWING

Discussions with various track designers indicate that gauge widening appears to be a holdover from steam locomotive days, and is not specifically necessary to prevent excessive flange wear. In fact, gauge widening promotes crabbing, since the natural tendency of a truck is to crab its way through a curve, with the high rail wheel of the leading axle riding against the high rail. On the other hand, gauge widening allows the development of a rolling radius differential where tapered wheel profiles are used, which can be further accentuated with asymmetrical grinding of the rail head as described above. SEPTA has observed that the wheel and rail gauges used on trolley systems typically vary by only ⅛ in. The slight variation in gauges necessitates widening in curves to prevent the flanges from binding.

The first known short radius curved track with gauge narrowing has recently been installed. In the summer of 1996, the Portland Tri-Met has replaced the embedded girder rail at some downtown 83-ft radius curves with low carbon steel rail set in cork-impregnated Icosit with ¼-in. gauge narrowing. The running surface and gauge face of the rails were treated with Riflex and Eteka 5 hardfacing, respectively. No squeal has been observed with this newly replaced rail, where before, squeal was a regular occurrence (*15*). The Portland Tri-Met also employs resilient Bochum wheels, which further help to control squeal relative to solid steel wheels.

## 8.4 DOUBLE RESTRAINED CURVES

Double restraining rails can be employed to reduce crab angle and promote turning of the truck at gauge widened curves. In this case, the high rail wheel can be brought away from the high rail by the low rail restraining rail, and the low rail wheel can be moved toward the high rail by the high rail restraining rail, thus reducing crab angle and lateral slip, as well as preventing flange contact with the high or low rail. The restraining rail separation would have to be controlled to prevent binding of the wheel set, or climbing of the flange onto the restraining rail. Further, the restraining rails may be liberally lubricated to prevent squeal from developing because of friction between the wheel and restraining rail. This technique has not been studied in detail, but may represent an avenue for further investigation of wheel squeal noise control. SEPTA has indicated that restraining rails produce squeal, and that it has obtained a significant noise reduction with high rail gauge face and restraining rail lubrication.

## 8.5 RAIL JOINT WELDING

Rail joint welding reduces impact noise otherwise generated by wheels rolling over the rail joint gap. The MTA NYCT is involved in extensive rail joint welding on at-grade structures, though not on elevated structures. Apart from noise reductions, conversion to continuous welded rail reduces track degradation and maintenance. Field welding involves welding the joint and grinding the running surface very carefully to avoid impact noise caused by dips. Thermite or electric arc welding is used. Flash butt welding has been used to avoid removing or replacing the rail. Automated flash butt welding cars can be procured for this purpose (*16*).

### 8.5.1 Noise Reduction Effectiveness

Joint welding and grinding would be expected to similarly reduce noise to levels consistent with continuous welded rail, provided that the rail is of similar quality and smoothness. A good rule of thumb is that welding of rail joints and grinding will reduce A-weighted noise levels by about 5 dB. Measurements at SEPTA (*17*) indicate about a 3 to 4 dBA wayside noise reduction after the first grinding of jointed ballast-and-tie track, ostensibly caused by improved rail joint alignment, while reductions due to rail grinding at other sections of track produced 0 to 2 dB noise reduction. The 3 to 4 dB improvement is indicative of the noise reduction that may be achieved by welding and grinding rail joints.

### 8.5.2 Site-Specific Conditions

The strength of steel elevated structures must be considered when deciding whether to convert to continuous welded rail. Thermal loads induce rail shrinkage or expansion, which necessarily must be resisted by the elevated structure, unless a provision is made for controlled slip via the rail clips. Rail welding has not been considered for the MTA NYCT elevated structures for this reason. However, at concrete or composite steel and reinforced concrete aerial structures, such as at BART, continuous welded rail is used, evidently without adverse effect. Longitudinal slip is designed into the rail clip assemblies to limit longitudinal track loads on the structures. Rail replacement procedures may also play a role in welding or use of welded rail. At the MTA NYCT, rail is replaced in panels consisting of two rails, cross-ties, and fasteners. This approach minimizes installation time and avoids interruption of train service. Replacement of continuous welded rail requires cutting the rail, installing a new section, and re-welding. At curves, high rail wear may necessitate frequent rail replacement, in which case continuous welded rail may not be economical.

### 8.5.3 Costs

The cost of field welding has been estimated at about $250 per joint in 1974, or about $13 per track foot (*18*). Applying a producer-price-index (PPI) ratio of 2.5 for prices today relative to 1974 indicates that current prices should be in the range of $600 to $700 per joint, or about $30 to $35 per track foot.

## 8.6 RAIL VIBRATION ABSORBERS

Rail vibration absorbers consist of resonant mechanical elements which are attached to the rail flange to absorb vibration energy. Though rail vibration absorbers have been proposed as noise control treatments from time to time, they have not been employed within the United States. They have been tested and employed in Europe. The rail vibration absorber is a treatment that may prove valuable at certain site specific locations.

### 8.6.1 Products

Deutsche Aerospace provides a rail dynamic absorber consisting of a multiple leaf unit weighing about 50 lb which is clamped to the underside of the rail. The technology is licensed to AEG Pittsburgh Daimler Benz Aerospace. AEG is now operating under the name of Adtranz. The model number of the absorber is AMSA 7.

### 8.6.2 Noise Reduction Effectiveness

Data provided by the manufacturer's representative indicate about 3 to 5 dB rail vibration reduction at 1/3-octave band frequencies between 300 and 2,000 Hz for 111 km/hr Deutsche Bundesbahn transit trains on tangent track. Absorbers were mounted on each rail, one between each rail fastener. The weight of each absorber is 50 lb. If one were to simply increase the weight of 115/yard rail by 50 lb per yard, a reduction of rail vibration above the rail-on-fastener resonance frequency and the wheel's first radial resonance frequency of 500 to 600 Hz by about 3 dB might be expected, simply from the added mass. The rail vibration reduction exceeds 3 dB at frequencies above about 700 Hz, and above this frequency the reduction is about 2 dB greater than what might be expected on the basis of mass ratios. Nevertheless, the vibration absorbers would be expected to reduce vibration transmission along the rail, which reduces the effective noise radiation length of the rail, and may help to control so-called "singing rail." Rail vibration absorbers might be effective in eliminating the pinned-pinned resonance of the rail due to discrete fastener supports. There is a distinct possibility that the absorbers might be useful in reducing rail corrugation rates.

No data were provided for wheel squeal noise reduction, but the possibility may exist that squeal noise might be reduced. However, theory of wheel squeal suggests that rail damping treatment should be of little effectiveness in reducing wheel squeal. Experiments are needed to determine any squeal noise reduction that might be obtained.

### 8.6.3 Site Considerations

A number of site specific considerations must be considered. The possible problems listed below are hypothetical, because of the lack of experience with application at U. S. transit systems.

*8.6.3.1 Elevated Structures*

There may be some reticence to use vibration absorbers clamped to the underside of rails on aerial structures or open deck steel elevated structures such as in New York, because of concern over safety for people underneath the structure who might be struck by falling absorbers. This problem should be controllable through use of some method of positive retention and inspection. Further, addition of 50 lbs per rail between each tie may cause concern over weight, a concern at older transit systems such as the MTA NYCT and Chicago CTA.

*8.6.3.2 Temperature*

Deutsche Aerospace did not provide an indication of the temperature range of the AMSA 7 vibration absorber. However, absorbers utilizing an elastomer element and optimized for moderate to high temperatures may lose a portion of their effectiveness at low frequencies. This may be of little importance during winter months when windows in cold climates are kept closed.

*8.6.3.3 Snow/Ice*

The leaf vibration absorber such as provided by Deutsche Aerospace would appear to be susceptible to freezing in subfreezing weather with snow.

*8.6.3.4 Ballast-and-Tie Track*

Vibration absorbers may be impractical on ballast-and-tie track unless they can be positioned clear of ballast. Further, the ballast-and-tie track may provide substantial energy absorption without vibration absorbers, so that the addition of the absorber would provide little additional noise reduction. However, a "singing rail" phenomenon at 500 to 1,000 Hz has been observed at ballasted track with concrete ties and Pandrol clips, where there is little absorption of vibration energy along the rail. In this case, vibration absorbers might prove very effective.

### 8.6.4 Cost

The costs of rail vibration absorbers is difficult to establish for the U.S. market, due to lack of use. However, they should not cost more than a typical resilient direct fixation fastener. Recent estimates for the cost of a conceptual vibration absorber design were about \$50 to \$100 per absorber. Assuming that absorbers are placed between every other fastener pair, the cost per track foot would be about \$20 to \$40 per track foot. If greater noise reduction is needed, requiring an absorber in each fastener bay, the cost would be \$40 to \$80 per track foot.

## 8.7 RAIL DAMPERS

A rail damper is a visco-elastic constrained layer damping system applied to the rail web to control wheel squeal, such as the Phoenix AG Noise Absorber Type A-4.1, illustrated in Figure 8-9. This product is held against the rail web with a spring clip, which reaches under and about the rail foot. The treatment can be applied with minimal disturbance of track, provided that it may be made short enough to fit between the

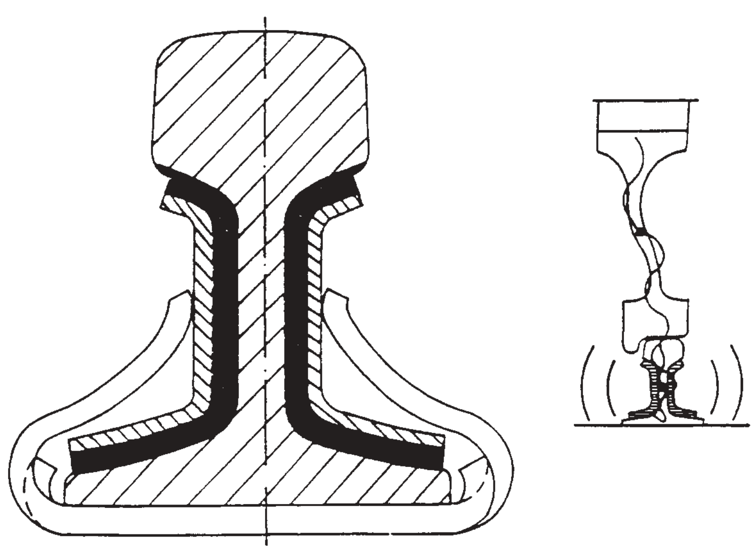

FIGURE 8-9 RAIL VIBRATION DAMPERS (PHOENIX AG TYPE 799323)

track fixation. The product has been produced in Europe for type S 41, S 49, and S 54 rails. Installations include:

Stuttgarter Strassenbaum (October 1991)
Societe De Transport Charleroi (Belgium, May 1992)
Leipziger Verkehrsbetriebe (September 1992)
Stadtwerke Frankfurt (September 1992)

The product has been installed and tested at the MBTA Green Line Government Center Station (*19*). The results of the tests are illustrated in Figure 8-10, which indicate that the damping treatment was effective at eliminating a component of squeal at 1,600 Hz. A second component of squeal at 4,000 Hz was not controlled, with the result that the A-weighted noise reduction was not great. Even so, the energy averaged A-weighted noise reductions were on the order of 2 to 5 dB, and a qualitative improvement should be obtained with elimination of the 1,600 Hz component. Identification and treatment of the source of the 4,000 Hz component of squeal would yield further A-weighted noise reductions, with further qualitative improvement.

Extension of these test results to other systems represents an intriguing wheel squeal noise control possibility. However, additional testing at other systems with different curve radii and different types of wheels is recommended. For example, testing of solid wheels at ballast-and-tie curves of radii on the order of 300 ft would be desirable to determine effectiveness for heavy rail systems.

## 8.8 RESILIENT RAIL FASTENERS

Resilient rail fasteners are effective in controlling wheel/rail noise to the extent that they provide vibration isolation between the rail and structure and reduce looseness in the rail fixation relative to standard ballast-and-tie track with tie plates and cut-spikes. They are not normally considered as a noise reducing treatment, however, except where vibration isolation is needed to control structure borne noise radiation from aerial structures or groundborne noise and vibration. These latter types of noise are not specifically within the jurisdiction of this manual, though aerial structure noise control is discussed below with respect to fastener design. There may even be a tendency for increased levels of wayside noise with resilient direct fixation fasteners relative to, for example, ballast-and-tie track, due to lack of sound absorption normally provided by the ballast. The design of resilient direct fixation track and resilient fasteners may have a significant effect on rail corrugation growth rates, though insufficient field test data exist to adequately define desirable characteristics for minimizing corrugation. Accordingly, those fastener characteristics which might have an influence on corrugation are discussed below.

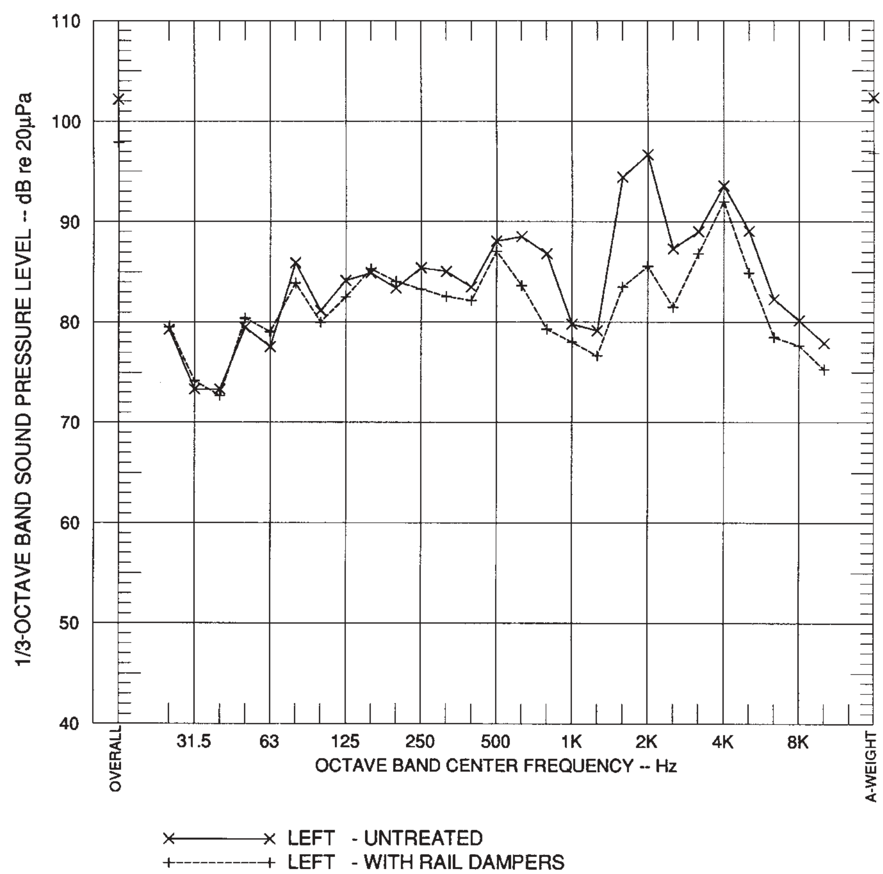


**FIGURE 8-10 WHEEL SQUEAL NOISE LEVELS AT THE LEFT-HAND MEASUREMENT LOCATION, 10 FEET FROM THE NEAR RAIL**

### 8.8.1 Characteristics

Rail fastener characteristics are described below.

#### *8.8.1.1 Static Stiffness*

The static stiffness is the principal design parameter used to describe a resilient direct fixation fastener. The static stiffness of modern direct fixation fasteners ranges from about 50,000 lb/in. for Clouth's "Cologne Egg," or its equivalent provided by Advanced Track, to about 110,000 lb/in., represented by the Lord Corporation fasteners supplied to BART and LACMTA. Earlier fasteners, such as the TTC or BART Landis fastener, have stiffnesses on the order of 250,000 lb/in. or more. Fasteners consisting of rolled top plates and thin neoprene pads have very high stiffnesses, often in excess of 1,000,000 lb/in. These fasteners should probably not be classified as resilient, because their stiffness may exceed the roadbed stiffness, especially on aerial structures.

#### *8.8.1.2 Dynamic versus Static Stiffness*

An important factor in resilient fastener vibration isolation performance is its dynamic stiffness. The dynamic stiffness is normally significantly greater than the static stiffness, because of the hysteretic nature of elastomers. Failing to account for the dynamic stiffness may result in insufficient vibration isolation. Further, the dynamic stiffness may vary considerably from one design to the next, even if the static stiffnesses are equivalent, because of variations in elastomer properties.

The dynamic stiffness is normally described in terms of the ratio of dynamic-to-static-stiffness. The dynamic stiffness of modern resilient direct fixation fasteners manufactured with natural rubber elastomer exhibit a ratio of dynamic-to-static stiffness of about 1.4 or less. Fasteners with synthetic rubber or high durometer elastomers exhibit ratios as much as 1.7 or 2, and are undesirable for vibration isolation.

Very stiff fasteners may impede rail motion at frequencies in the range of 250 to 500 Hz, thus reducing noise radiation by the rail. On the other hand, the rail-on-fastener resonance frequency for very stiff fasteners may be within the range of corrugation frequencies associated with short pitch rail corrugation, and thus may contribute to rail corrugation.

#### *8.8.1.3 Top Plate Bending*

The typical resilient fastener has a cast or rolled top plate bonded to an elastomer element. The resonance frequency of the top plate in bending is typically about 600 Hz or higher, and the stiffness of the fastener increases dramatically with frequency prior to this resonance, and thereafter decreases sharply at the resonance frequency. Depending on the mass of the top plate and stiffness, the input mechanical impedance of the fastener also reaches a minimum at the rigid-body resonance of the plate on the fastener stiffness, which usually occurs at frequencies below 500 Hz for modern fasteners. The frequency range above 500 Hz is also comparable with the pinned-pinned mode of rail vertical vibration due to discrete rail support. The wheel also exhibits lateral vibration modes and a radial anti-resonances above 300 or 400 Hz. These various modes of vibration may combine to exacerbate wheel/rail noise, especially on direct fixation track. Moreover, all of these frequencies are comparable with frequencies associated with short-pitch corrugation. Increasing the top plate resonance frequency in excess of about 1,000 Hz to separate the top plate resonance frequency from rail corrugation frequencies would be consistent with good design practice to reduce corrugation, though this has not been substantiated. The resilient fasteners currently being installed at BART at the new extensions were specified to have a top plate resonance in excess of 800 Hz to avoid coincidence with the rail corrugation frequency typically occurring at about 500 Hz. Fasteners utilizing a cast top plate with thick rail seat area provide high top plate stiffness. Examples include fasteners supplied to BART, LACMTA, and MTA NYCT. (See the discussion concerning top-plate bending and rail corrugation in Chapter 10.)

### 8.8.2 Rail Damping

Remington has developed a theory of rail fastener performance which includes damping provided by the elastomer, thus predicting a reduction of rail vibration and rail radiated noise. At least part of the noise reduction achieved at steel elevated structures by resilient fasteners relative to standard tie plates and wood ties with cut spikes is due to this damping effect. Just as rail vibration absorbers absorb rail vibration energy, resilient elastomer fasteners also may absorb vibration energy. The degree of vibration and noise reduction is dependent on fastener design, but there is a possibility of tuning the fastener to enhance its vibration absorbing properties by exploiting the 1/4-wave resonance of the elastomer (*20*). No fastener has yet been constructed to maximize its vibration energy absorbing capability, but such an approach remains attractive. Such a fastener could evidently be constructed with an elastomer thickness on the order of 1 to 2 in.

The top plate bending resonance might conceivably be exploited to absorb vibration energy at the resonance frequency. Although removing the coincidence between rail corrugation frequencies and top plate bending resonance frequency appears to be desirable, tuning the top plate bending resonance and elastomer damping to maximize energy absorption at 500 to 1,000 Hz is attractive. Further study is required.

### 8.8.3 Products

Resilient fasteners are currently supplied for U.S. transit systems by a number of firms, including Lord Corporation; Advanced Track/Goodyear; American Track Systems, Inc.; Transit Products, Inc.; Landis Sales, Inc.; Clouth Gummiwerke; and Phoenix USA.

### 8.8.4 Steel Elevated Structures

Resilient fasteners have been incorporated into the wood tie trackwork on steel elevated structures and very soft fasteners have been used at composite concrete deck and steel box girder aerial structures to control noise. Very soft fasteners provide the rail with greater support compliance, allowing the rail to vibrate at higher amplitude than would be the case with very stiff fasteners, but provide greater vibration isolation between the rail and supporting structure. The net effect of reducing fastener stiffness on wheel/rail A-weighted noise may be nil or limited to a few decibels. For example, wayside noise levels at MTA NYCT steel elevated structures appear to be about 1 to 2 dBA lower with soft fasteners of static stiffness on the order of 100,000 lb/in. static compared to resilient fasteners with static stiffness greater than 200,000 lb/in. (*21*). Similar results have been obtained at BART (*22*). Low frequency structure radiated noise is reduced at WMATA and MTA NYCT over a limited frequency range on concrete deck/steel box composite aerial structures, though the A-weighted noise reduction is limited (*23*).

Resilient rail fasteners with positive retention rail clips, such as Pandrol clips, eliminate clatter between the rail and tie plate, thus reducing impact forces and noise on older systems which otherwise use wood ties and standard tie plates, such as the MTA NYCT and CTA. MTA NYCT has replaced all steel elevated structure track with resilient rail fasteners, wood ties, and new rail. The noise reduction benefit obtained just by eliminating the standard tie plate in favor of resilient rail fasteners is on the order of 3 to 6 dBA, based on single event noise exposure measurements (*24*).

The radiation of noise from steel elevated structures is very complex, involving not only wheel/rail interaction but structural responses that are difficult if not impossible to quantify. The most comprehensive effort in this regard includes work developed by Remington (*25*). By far, the most effective treatments for elevated structure noise control appear to be elastomer fasteners combined with continuous welded rail or jointed rail with tight rail joints, rail grinding, and wheel truing. Since retro-fit of most of the steel elevated structures with resilient fasteners, the MTA NYCT has received very few complaints concerning noise (*26*).

### 8.8.5 Costs

Costs for resilient fasteners vary considerably from procurement to procurement, depending on quantities ordered, and the degree of qualification testing required. Typical costs range between $50 and $100 per fastener, including anchor bolts, anchors, and clips. Benefits may include reduced rail maintenance with direct fixation track relative to ballast-and-tie track, and reduced track maintenance for wood ties and resilient fasteners on older structures relative to standard tie plates with cut-spikes.

### 8.8.6 Site-Specific Conditions

Site-specific conditions include the following:

> *Temperature range and longitudinal slip requirements*. High temperature variations may induce substantial longitudinal expansion and contraction of the rail, which may be of serious concern with respect to elevated structures. Rail buckling must be considered. Fastener rail clips are normally specified to allow longitudinal slip to accommodate rail expansion and contraction.
>
> *Corrosion and electrolysis*. Direct fixation fasteners are effective insulators. However, condensation and moisture within tunnels, coupled with electrolysis, may cause fasteners to deteriorate, for which certain remedies have been developed, such as galvanizing or coating with rust inhibitors. Stray currents must also be considered.

## 8.9 SPECIAL TRACKWORK

Special trackwork includes switches, turnouts, and crossovers. Significant impact noise may be generated by wheels traversing frog gaps associated with special trackwork. Impact noise may be controlled by grinding the frog to provide as smooth a transition as possible for each wheel to pass from one side of the flangeway to the other. Special frogs, including moveable point, swing nose, and spring frogs, have been developed to minimize impact forces by eliminating the fixed gap associated with the frog. Because the frog gap is the major cause of the increase in noise when a train passes through a turnout, the use of special frogs to reduce special trackwork noise is a practical noise control provision for many transit systems.

### 8.9.1 Types of Frogs and Manufacturers

The various types of frogs and their manufacturers are discussed below.

#### *8.9.1.1 Rail Bound Manganese Frogs*

Rail bound manganese frogs are used at many transit systems, and serve as the baseline for the purpose of discussing noise control. These frogs include a fixed gap and conventional rail wrapped around a manganese center. This places the wear resisting element at the discontinuities of the frog, which reduces wear, but generates the impact noise previously discussed when wheels traverse the fixed gap. In addition, there are two joint gaps just before and after the manganese insert which also produce noise. The typical cost of a rail bound manganese frog is approximately $6,000 (1994 dollars).

#### *8.9.1.2 Welded Vee Frogs*

Welded vee frogs are fixed gap frogs constructed from standard rolled rail. The welded vee holds together two tapered rails with a continuous longitudinal weld. This design eliminates the two joint gaps associated with the rail bound manganese frog, but does not eliminate the fixed gap.

#### *8.9.1.3 Flange Bearing Frogs*

Flange bearing frogs provide support to the wheel flange while traversing the frog gap in embedded track. A properly installed frog supports the flange, maintains the wheel height through the frog, and reduces the impact forces associated with the wheel traversing the gap. The depth of the flange support below the top of rail is critical in providing a smooth transition through the gap. If this support is too high or too low, then the transition is not smooth and the impact noise is not eliminated.

*8.9.1.4 Moveable Point Frogs*

Moveable point frogs, also known as swing nose frogs, are perhaps the most effective at eliminating the impact noise associated with fixed gap frogs. Modern movable point frogs have been developed in Europe for high speed railways, and current applications in North America include limited use on the AMTRAK Northeast Corridor, limited use on heavy freight railroads to reduce frog maintenance, and use on transit systems such as Vancouver ALRT and Detroit CATS where the small diameter wheels cannot safely traverse a frog gap. The gap of the frog is eliminated by laterally moving the nose of the frog in a direction corresponding to the direction of train travel. The moveable point frog generally requires additional signalling, switch control circuits, and an additional switch machine to move the point of the frog. The additional cost of the moveable point frog has been estimated to be as high as $100,000 when the additional design, construction, material, and maintenance costs are considered. Manufacturers of moveable point frogs include (1) Voest-Alpine, (2) Cogifer, and (3) Balfour-Beatty.

*8.9.1.5 Spring Frogs*

Spring frogs also eliminate the impact noise associated with fixed gap frogs for trains traversing the frog in a normal tangent direction. The spring frog includes a spring loaded point which maintains the continuity of the rail running surface for normal tangent operations. For diverging movements, the normally closed frog is pushed open by the wheel flange. There may be additional noise associated with trains making diverging movements, because the train wheels must still pass through the fixed portion of the frog. Thus, use of these frogs in noise sensitive areas where a significant number of diverging movements will occur will not significantly mitigate the noise impacts associated with standard frogs. The additional cost of a spring frog is on the order of $12,000, compared to approximately $6,000 for a standard frog. Manufacturers of the spring frog include Voest-Alpine.

### 8.9.2 Noise Reduction Effectiveness

There are limited data regarding the noise reduction achieved with moveable point or spring frogs relative to standard frogs. Typical wayside noise levels for standard frogs are 6 to 10 dBA higher than normal rolling noise levels on tangent track with continuous welded rail, although levels as high as 15 dBA over normal have been measured at SF Muni for older style embedded track turnouts for operations at speeds less than 25 mph at a distance of 25 ft. This increase may be speed related, because the wheel/rail rolling noise generated at low speed is considerably less than for higher speed operations, so that impact noise may be dominant at low speeds. The difference between impact noise and rolling noise is less for high speed operations than for low speed operations, and impact noise may be masked by rolling noise at high speed. This appears to be true for AMTRAK operations on the Northeast Corridor, where wayside noise measurements showed virtually no difference for operations on standard ballast-and-tie track with a #20 rail bound manganese frog and a #30 movable point frog. However, impact noise from BART trains traversing well used special trackwork at 70 mph is quite audible.

In July 1992, a series of tests of both wayside noise and vibration were performed at the BART test track to obtain a quantitative evaluation of the reduction of noise due to the use of a movable point frog. Measurements were made at the same location on different dates with standard ballast-and-tie track, a #10 rail bound manganese frog and a #10 moveable point frog. The frogs were installed on the near rail of the test track, though entire turnout assemblies were not installed. Wayside noise test results obtained at 25 ft and 50 ft from the track centerline indicate that train operations on the moveable point frog totally eliminated the impact noise, but generated a strange howling or whining noise as each wheel passed over the frog. The cause of the aberrant noise is unknown and is uncharacteristic of other movable point or spring frogs which have been evaluated on a qualitative basis. The noise could be due to improper installation.

Only limited noise data for spring frogs have been obtained. Interior noise data obtained by WIA indicate that the spring frogs installed at a crossover in the subway connector of the Howard-Dan Ryan Line in Chicago are effective at controlling impact noise.

No quantitative data for the noise characteristics of modern flange bearing frogs have been located, although the mechanism for reducing impact noise of light rail trains operating at slow speeds on embedded track in streets appears promising.

Some manufacturers have claimed that wayside noise is reduced with the use of welded-Vee frogs. Measurements made on the BART Concord Line for in-service conditions indicate that wayside impact noise due to trains traversing welded-Vee frogs are generally 6 to 10 dBA greater than for standard ballast-and-tie track. This increase is typical of the rail bound manganese frog, which suggests that the welded-Vee frog does not provide effective impact noise reduction.

### 8.9.3 Costs

As previously indicated, moveable point frogs have been used on a limited basis by mainline heavy haul railroads to reduce maintenance costs. However, as previously indicated, the capital costs of moveable point frogs are greater than any of the others discussed here. The cost of a turnout with moveable point frog can be on the order of $100,000 higher than the cost with a standard frog.

Reduced maintenance costs are expected for the frog for rail transit applications as well as for freight, although frog maintenance is generally not a major cost factor for transit systems. Increased maintenance associated with the additional switch machine and other signalling equipment may negate the benefit of reduced maintenance of the frog for transit system operations.

The cost of the spring frog is estimated to be about $12,000, or $6,000 more than the cost of a standard rail bound manganese frog. Thus, the cost of a spring frog is considerably less than that of a moveable point frog.

### 8.9.4 Site-Specific Conditions

The ability to use moveable point frogs, spring frogs or flange bearing frogs is dependent on a number of factors, including:

- Turnout size and train speed – High turnout speeds may be accommodated with moveable point frogs. Thus they are applicable to large turnouts. Spring frogs must be limited to turnouts of low train speed (approximately 15 mph) for diverging movements, and are most practical for turnouts of short radius. Although the use of flange bearing frogs is not speed limited for most transit use, the noise reduction benefits will be most readily realized for low speed operations typical of light rail transit operating on embedded track.
- Spring frogs are generally unsuitable for modern signalling systems which require positive directional indicators.
- Spring frogs will not eliminate the impact noise of the wheels traversing the frog gap for diverging movements, and thus the noise reduction capability where diverging movements are routine is limited.
- Spring frogs, unless heated, are not practical in subfreezing weather. At the CTA, spring frogs are exclusively used in subway applications.

## 8.10 RAIL LUBRICATION

This section concerns wayside lubrication approaches to controlling wheel squeal and wear at curves. Onboard lubrication is discussed in Chapter 7, and much of the discussion presented there is also applicable to this section.

There are two views concerning the generation of squeal, both of which may be correct. The first and most attractive theory holds that wheel squeal is caused by lateral creep and stick-slip of the tread across the rail head, with inhibition of squeal by flange contact (*27*). The second and popular theory is that wheel squeal is caused by flange contact and associated stick-slip excitation of the rail and wheel. Except for inhibition of squeal by flange contact, both of these squeal-generating mechanisms may exist, though the physics of the latter theory are not described in the literature. The former squeal generating mechanism is described in Chapter 4.

### 8.10.1 Background

When a vehicle travels around a curve, the flange of the outside leading wheel of the each truck bears against the gauge face of the outside rail (high rail) of the curve, causing side or gauge face wear. The flange contact at the high rail induces a torque on the truck, causing the truck to crab in the curve, possibly inducing trailing wheel flange contact at the low rail, and increasing lateral creep across the rail head. Thus, substantial wear may occur at the gauge face of both the high and low rails. In addition, a stick-slip mechanism can be introduced, whereby friction-induced vibration at the wheel tread/rail head interface can generate noise, i.e., squeal. Thus, lubrication, as used in the rail transit environment, is generally used to address one or both of these two problems, i.e., wear and noise. Many transit properties apply lubricant to reduce the squeal noise noted above, although reduction of rail and wheel wear is the primary criterion for lubrication.

Lubrication is fundamentally limited in controlling wheel squeal, because tread and rail running surfaces cannot be lubricated without loss of adhesion and braking effectiveness. Loss of braking effectiveness may result in wheel flatting, which produces excessive rolling noise, a counterproductive result of improper lubrication. As noted above, the most attractive theory of wheel squeal holds that lateral creep of the tread across the rail running surface is the primary source of squeal, rather than flange contact, yet only the flange may be lubricated, and then only lightly, although lubrication of the flange may improve curving, reduce crab angle, and, thus, reduce squeal.

Aural and visual observations at the MBTA Green, Blue, and Red Line's restrained curves indicate that wheel squeal is associated with substantial lateral creep and flange contact, even with lubricated restraining rails (*28*). The possibility exists that adjustment and lubrication of the low rail restraining rail to prevent excessive crabbing and flange contact at the high rail might reduce the propensity for lateral stick-slip between the rail running surface and tread. At double restrained curves, the high rail restraining rail can be adjusted to further reduce crabbing angle and flange contact at the low rail. There is no limitation on lubrication of the restraining rails, since these contact the back side of the wheel tire and flange, and exploitation of lubricated restraining rails to control curving appears to be worth pursuing for squeal reduction. Wheel squeal also occurs at the Sacramento RTD 82-ft-radius restrained curves without high rail flange contact (*29*), suggesting that lubricating restraining rails will not entirely solve the problem of wheel squeal.

To reduce wear, lubrication with a low coefficient of friction lubricant between the wheel flanges and the gauge face of the rail is commonly used. Lubrication of the gauge face of the high rail in curves has been used over 50 years to reduce the rate of wear of the rail, particularly gauge face wear in curves. However, only in the last decade has hard research data become available to support the significance of benefits associated with rail lubrication. These benefits include not only wheel and rail wear reduction, but also reduction in energy (fuel) consumption.

Lubrication procedures for noise reduction are not as well defined as for wear reduction. Some properties rely on a layer of lubricant between the wheel flange and the rail gauge face to reduce slip-stick around the curve and thus reduce noise. Another approach that has been advocated is the application of a high friction lubricant, or friction modifier, to the wheel tread, to modify the friction versus creep curve of the wheel tread and the top of the rail head, an approach intended to directly reduce the stick-slip squeal mechanism noted above. (These latter procedures are discussed with respect to onboard treatments.) One railroad property has experimented with hand application of the friction modifier to the rail head to control squeal. An intriguing possibility is automatic application of friction modifiers to the wheel tread by wayside applicators as the wheel enters a curve, though such equipment are not available.

### 8.10.2 Acoustical Benefits

Characterizing squeal noise is difficult, because of the intermittent or unpredictable occurrence of squeal. There are two methods: (1) maximum level and (2) root-mean-square, or energy equivalent level, over the duration of curving. The former method addresses the audibility of squeal noise, and also addresses the degree of discomfort experienced by persons located close to the track (e.g., pedestrians at street corners, or transit patrons in vehicles with open windows). The latter procedure addresses the duration and occurrence of the squeal noise, useful for predicting community noise levels such as energy equivalent level ($L_{eq}$) and day-night level ($L_{dn}$). Both peak and energy equivalent measures should be employed, and noise reduction methods should attempt to reduce both.

From a practical point of view, the maximum squeal noise level reduction is not as important as elimination or reduction of the duration or occurrence of squeal. Once a regenerative system begins to squeal, the amplitude tends to saturate, limited only by material damping and friction in the system. Still, lubrication does tend to reduce the amplitude and duration of squeal noise, and, thus, is attractive, even if squeal is not entirely eliminated. A lubrication procedure should be deemed at least marginally successful even if it only reduces the occurrence and not the amplitude of squeal. A reduction of the occurrence or duration of wheel squeal by a factor of two will reduce wayside energy equivalent noise levels by 3 dB, even though the maximum level is unaffected.

The noise reduction effectiveness of lubrication can be substantial at curved track. Without lubrication, wheel squeal maximum noise levels may exceed 100 dBA (Toronto reports levels as high as 110 dBA, though the measurement distances are not known). With lubrication, passby noise levels have been reduced to those of rolling and auxiliary equipment noise. Thus, typical noise reductions are on the order of 15 to 25 dBA.

An automatic wayside lubricator employed at SEPTA is effective in reducing squeal at a turnaround. Both rails are fitted with flange lubricators, and there is some migration of lubricant to the rail head. Squeal is eliminated for most of the curve. However, at the end of the curve, there is some re-emergence of squeal, attributed to loss of lubricant.

### 8.10.3 Lubrication Application Techniques

The techniques of lubricating the rail head vary significantly, depending on operating environments and external factors. While rail lubrication techniques have been in use for many years, they have met with varying degrees of success. In general, the two approaches that can be utilized in rail lubrication include wayside lubrication (with lubricators that are permanently located at fixed points in the track), and onboard lubrication (lubricators that are mounted on a moving vehicle.) The discussion presented below concerns wayside lubrication systems, while onboard lubrication is discussed in Chapter 7.

Wayside lubrication is the traditional approach that has been used by railroads and transit systems for many years. Most of these lubricators use some form of mechanical applicator system, such as a wiping bar, to apply a predefined amount of lubricant to each passing wheel flange. Thus, every wheel of every train gets a small amount of lubricant applied to its flange, which in turn carries the lubricant along and applies it to the rail (or rails) for a distance beyond the lubricator. This carrying distance is limited, however, so that wayside lubricators must be located at periodic intervals along the curve. Very often the level of lubrication varies with distance from the lubricator, climate (temperature and rainfall), train speed, grease characteristics and other factors. In addition, the remote nature of wayside lubricators makes inspection and maintenance difficult. Recent developments in wayside lubricator technology include more reliable wheel sensing and applicator systems, and more specific attention on lubricator inspection and maintenance.

The wayside lubricator is fixed to the track near the beginning of a curve, and dispenses a small amount of lubricant to each wheel flange of a train passing over it. This lubricant is in turn deposited along the curve, on that part of the rail that is in contact with the wheel flange. A film of lubricant is thus applied to the high rail for the complete length of the curve. If wheel squeal is indeed produced by lateral creep of the wheel tread across the rail head, the success of flange lubrication in reducing wheel squeal may be due to unintended migration of small amounts of lubricant to these surfaces.

For very limited circumstances, hand application of lubricants can also be used. However, hand lubrication should be used on a spot basis only, since it is extremely difficult and expensive to manually apply lubricant on a regular basis through the length of a curve. Wayside lubricators have traditionally been used for wear reduction, while hand application has been used for both wear and noise reduction (and also derailment prevention).

Dry lubricants which provide a low coefficient of friction are applied to the wheel flange by stick lubricators attached to the vehicle. LCF dry lubricants reduce the coefficient of friction to about 0.06, which may be compared to 0.02 to 0.04 for liquid lubricants. A particular advantage of dry lubricants is that they may be less likely to migrate from the flange area to the running surface of the rail, and thus less likely to compromise traction.

Water spray by wayside applicators have been used at several systems to control wheel squeal. An example is the TTC, which uses water sprays during the summer months to control squeal, and rail corrugation and wear, at curves. Both the high and low rails are thus treated. The system is evidently made by the TTC. Water spray is used at the SRT system at both high and low rails during the summer (survey questionnaire return). Water spray has been reported to reduce wheel squeal by 18 dBA at a short radius curve at the WMATA system (*30*). However, water spray could not be used during winter periods of freezing weather, and was thus not utilized as a long-term noise control method.

### 8.10.4 Current Railroad and Transit Practices

Current practice in the railroad and transit industries is to lubricate the rail to reduce wheel and rail wear, and to a lesser extent noise. Actual practices have been evolving in the last decade, with an increasing use of lubrication due to the definable and measurable benefits associated with lubrication.

Virtually all of the transit systems in the United States are using wayside lubricators (though one was on an experimental basis). In addition, while a few problems with sliding trains have been reported, these appear to have been corrected by proper attention to and maintenance of the lubricators. For large properties such as MTA NYCT and CTA a large number of wayside lubricators are currently in use. Other systems also use hand lubrication for noise control.

### 8.10.5 Rail Lubrication Cautions

While the benefits of rail lubrication have been well documented, there are a few cautions in regard to the lubrication of the rail (and the wheel). While most of these caveats refer to greases, they can also apply to onboard stick lubricants which are misapplied or not properly placed. While these cautions in no way detract from the benefits of rail lubrication, there are potential problem areas that may arise as a transit system's lubrication program is expanded.

These cautions can be divided into three basic areas:

1. Over-lubrication,
2. Wheel/Rail dynamics, and
3. Wear versus fatigue.

The concerns regarding over-lubrication are generally well known. Over-lubrication has been generally defined as the condition where lubrication is present *on top* of the running surface of the rail (this is not the case with the high-friction lubricant intended to be applied to the rail head or wheel tread), though this may be the very mechanism which produces effective wheel squeal noise reduction. With over-lubrication, several classes of problems stemming from operating difficulties associated with wheel-slip can arise. These problems include the slipping of trains during stops, particularly station stops, and the stalling of trains on grades, due to the reduction of the coefficient of friction between the wheel and the rail (and the corresponding reduction in effective traction). Other operating problems associated with train handling, train action, or general operations, can also occur. Over-lubrication frequently occurs when the wayside lubricators are made to produce a higher output level than appropriate in an attempt to extend the distance covered by the lubricator (i.e., extend the "carry" of the lubricant).

In addition to these operating problems, maintenance problems due to over-lubrication can occur, including the formation of wheel burns (and their associated rail and track problems) due to wheel slippage on lubricated rail heads. Also, decreased efficiency of ultrasonic inspection equipment can occur if a layer of lubrication and dirt is built up on top of the rail.

The second class of concerns associated with lubrication has been reported recently as a result of an investigation into the effect of lubrication on wheel/rail forces. Specifically, these concerns are associated with the lubrication of one rail of either a curve or a section of tangent track. As the lubrication increases on *one rail*, the wheel/rail forces, the corresponding L/V ratios, and the associated railhead lateral deflections (dynamic gauge widening), all increase. However, when lubrication is simultaneously applied to the other rail, so that both rails are lubricated, both the forces and the corresponding deflections decrease.

The third area of concern regarding the increased use of lubrication is the emergence of rail fatigue, both surface and internal, as the dominant failure criterion for curves as well as tangent track. This is usually not a serious concern for transit systems with their lighter axle loads.

In general, the documented benefits of rail lubrication indicate that increased application of rail lubrication is economically justifiable for general rail application. In fact, lubrication offers the potential for significant savings in several areas of railway operations, including wheel and rail wear and fuel consumption. However, like any maintenance procedure, rail transit personnel should be aware of the potential problems associated with this procedure.

### 8.10.6 Economic Considerations

The economics of lubrication for rail and wheel wear are extremely significant with high ROIs (return on investment) reported for conditions where wear has been reduced by effective use of lubrication. To the extent that high noise levels due to stick-slip vibration are occasioned by high wheel and rail wear rates, rail lubrication for noise control should result in a positive ROI. For example, recent experience at SEPTA indicates that the ROI due to increased rail and wheel life is between 85 to 90% for moderate to good lubrication at curves, energy savings notwithstanding. The return on freight railroads is even greater, due to the heavier axle loads and more severe operating environment. This order of magnitude of benefit has been experienced at other properties as well (*31*).

### 8.10.7 Lubricant Products and Manufacturers

Lubricant products and manufacturers include

- Lubriquip
- Century Lubricating Oils: Centurail Track PL graphite grease
- D. A. Stuart Co., SURBOND 71 MT6 Grease
- Texaco 904
- Moliplex EP–2 non-graphite grease (WMATA)
- Exxon Corporation: Van Estan No. 10
- Superior Graphite Company, #30, #32, #37 flange lubricants
- Intek, Railube 1200 (Represented by American Track Systems)
- SWEPCO 604

Examples of automatic lubrication systems include

- Rails Co. Electro/Pneumatic (used by TTC)
- SRS Clic-o-Automatic (used by TTC)
- X Electric (Used by WMATA)

The Toronto Transit Commission lubricates both the high and low rails and restraining rail during the winter with Rails Co. Electro/Pneumatic and SRS Clic-o-Automatic wayside lubricators and SURBOND 71MT6 grease. At the surface track system, grease is manually applied during the winter, at high, low, and restraining (girder) rails at loops. At the SRT system, grease (SURBOND) is used on the high rail only during the winter, using a Clic-o-Automatic system (questionnaire return).

The St. Louis Metrolink manually lubricates the restraining rail at curves, employing a Century Lubricants Centurail Track PL graphite grease.

SEPTA manually applies a graphite grease made by various manufacturers to the restraining rails, and also employs an automatic lubrication system at one of its rail loops (questionnaire and personal observation).

The Port Authority Trans-Hudson Corporation employs track-mounted grease lubrication, and also employs Texaco 904 or Exxon Van Estan No. 10 to high and restraining rails, using automatic and manual applicators. The cost is indicated by the survey questionnaire response by PATH as $400,000 per year.

### 8.10.8 Site-Specific Conditions

Local conditions must be considered in selecting a lubrication program. These include temperature, rainfall, grade, electrical contact, and environmental pollution.

#### *8.10.8.1 Temperature*

Water sprays are not practical in exterior environments where cold weather may freeze the water. The TTC has developed a systems whereby water is used in the summer months and grease in the winter.

#### *8.10.8.2 Rainfall*

High rainfall may wash away some of the lubricant. This is particularly relevant to dry-stick friction modifiers, or other dry lubricants such as graphite, or lubricants that are carried in volatile liquids which dry after application.

#### *8.10.8.3 Track Grade*

Excessive grade may preclude use of grease as a lubricant since traction may be lost. Lubricants are normally applied to the flange to avoid loss of traction, though some lubricant necessarily migrates to the rail running surface. Water sprays may remain as a viable option, since friction is not seriously affected by the water, provided that freezing temperatures are not encountered.

#### *8.10.8.4 Electrical Contact*

There is some concern over loss of electrical contact resulting from the use of lubricants. Water sprays may induce corrosion which is not conducive to electrical contact, and might not be advisable in lightly used track or where signalling may be affected.

*8.10.8.5 Environmental Pollution*

Environmental degradation by lubricants is a serious consideration. Water sprays would likely pose less of a problem than grease or oil. Lubricants should be biodegradable to the maximum extent possible.

## 8.11 BALLASTED TRACK

Ballasted track provides substantial sound absorption and reduces wheel rail noise at both wayside and subway areas by about 5 dB relative to direct fixation track, the differences in rail dynamics notwithstanding. These assumptions are supported by measurements at the BART system (*32*). At the Chicago CTA station platforms, the noise level with ballasted track are up to 15 dBA quieter than at stations with concrete trackbeds (*33*) (though this may be related to differences in rail condition).

## 8.12 TRACKBED SOUND ABSORPTION

Installation of sound absorption between the rails, placed directly on the invert with a positive retention system, will absorb some of the sound energy radiated by the rails and wheels (as well as traction power system noise), in a manner analogous to ballasted track. This approach is not particularly attractive due to the need for cleaning, oil spills, or other contamination. Nevertheless, there may be areas where use of between rail absorption might be attractive, such as at curves for reduction of squeal noise inside the vehicle.

## 8.13 HARDFACING

Rail head treatment or "hardfacing" is the application of a hard alloy metal inlay to the rail head, gauge face, or gauge corner to inhibit excessive rail wear. The procedure involves cutting or grinding a groove in the rail surface and welding a bead of material into the groove (Figure 8-11).

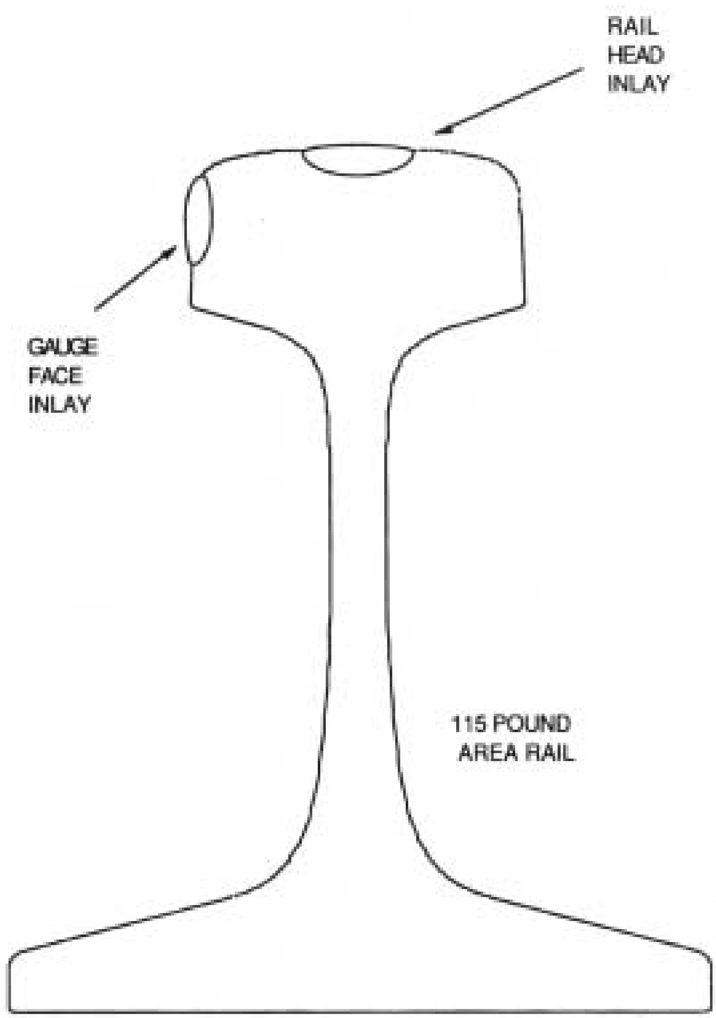


FIGURE 8-11 RAIL HARDFACING

The technique was developed by Electro Thermite GmbH, Germany, and has been used on a limited basis on transit systems in the United States, primarily for wear reduction. The technique has been in use in Europe since the early 1980s for rail corrugation control, and has received some limited use in the United States with little success for squeal noise control.

The general treatment, called by the trade name "Riflex," actually consists of three different materials:

- Riflex—used for corrugation control,
- Eteka-5—used for gauge face wear control, and
- Anti-Screech—used for squeal noise control.

The material placed on top of the rail is for corrugation and noise control (Riflex) and the material placed on the gauge face of the railhead is for control of gauge face wear (Eteka-5), particularly in sharp curves. Rail with a combination of the two, referred to as "Combi Rail," consists of weld beads on both the top and gauge face of the railhead. The Anti-Screech treatment is intended for reduction of wheel squeal. All three systems are produced in the same way, though different alloy fillers are used for each of the three.

The Riflex treatment for rail corrugation control uses a very hard material, exceeding Rockwell C 58 and 587 BHN (Hultgren Ball). As such, the treatment is harder than alloy rails such as Cr/Si high-strength steel alloy, and is designed to support the wheel without any surface degradation, thus preventing the formation of corrugations. The Eteka-5 material is fairly ductile when applied, allowing for bending of the rail to fit tight radii curves, but quickly hardens to 470 to 500 BHN. The Anti-Screech is a babbit-like material which remains soft. All three materials can be welded into CWR strings by either electric flash or thermit welding processes.

The Riflex and Eteka-5 treatments are deposited into a groove at the top and/or gauge face of the railhead (Figure 8-11). The shop procedure includes (1) grinding a 20 mm-wide by 5-mm deep groove into the top (or gauge face) of the rail; (2) welding a bead of the alloy into the groove (and extending beyond it); (3) finish grinding the bead; and (4) roller straightening the finished rail. Pre-heat and submerged arc welding are used to deposit the Riflex alloy in two passes so that the height of the filler is well above the rail head. The filler is then rough ground with a stone at various angles and finished with a longitudinal belt sander parallel with the rail. The finished surface of the filler is between 1 mm and 2 mm above the top of the rail. The rail is then straightened, inspected for defects, and shipped. The material is intended for use with low carbon steels. All three products may be applied to embedded girder rail because no straightening is required. The welding process induces considerable distortion of the rail, which may have some adverse impact on field installation, which should be investigated with the supplier.

The hard Riflex alloys are magnetic, so that they do not interfere with magnetic rail brakes. (The babbit-like soft Anti-Screech may not be magnetic, though this has not been verified with the manufacturer.) Further, there is no known problem with signalling or current conduction.

### 8.13.1 Noise Reduction Effectiveness

Hardfacing is a relatively new and largely unproven technique for the control of noise, corrugation and/or wear (*34*), and its main attraction is evidently the control of wear at curves. However, to the extent that it is effective in controlling corrugation, it would be very effective in controlling corrugation noise. Further, reduction of wheel squeal is a significant benefit. These are discussed below.

#### *8.13.1.1 Corrugation Control*

Wear rates with and without Riflex evaluated under laboratory simulation indicated a wear rate of about 0.1 mm per 100 million gross tonnes with Riflex versus 0.5 mm per 100 million gross tonnes without Riflex, amounting to 75% to 80% reduction in wear rate (*35*). Wheel wear rate was not adversely affected. The running surface of the Riflex treatment remained narrow, while that of the untreated rail broadened very rapidly with increasing load. The wheel wear surface of the test apparatus remained narrow, resulting in minor "grooving" of the wheel tread, for wheel supported by the Riflex rail, while the tread wear band remained broad with the untreated rail. While the laboratory simulation did not represent field conditions with actual vehicles and truck dynamics, the test results indicate that substantial rates of wear reductions are achievable.

Orgo-Thermit Inc. provided numerous photographs of before and after treatment of rail running surfaces, suggesting very favorable performance of Riflex in reducing corrugation rates, or, in some cases, inhibiting formation of corrugation. Examples include the Paris Metro, Frankfurt, Hannover, Stockholm, and Darmstadt systems. These examples include both tangent and curved track, and both ballast-and-tie and embedded track. Tests on the Paris Metro showed that Riflex resisted corrugation for 3 years and 84 MGT at a location where standard rails corrugated within four days. Orgo-Thermite has indicated that Riflex is effective in controlling or inhibiting corrugation at curved track. Examples include Darmstadt, and Hannover.

To the extent that the weldment may exhibit considerable parametric variation, or variation of effective contact width, rolling noise on Riflex would be expected to exceed rolling noise on standard, well-ground rail. No data have been obtained to verify this, however.

*8.13.1.2 Wheel Squeal*

Wheel squeal is caused by negative damping resulting from a negative slope of the friction versus creep velocity curve for the wheel/rail interface. Van Ruiten (1988) indicates that wheel squeal will be inhibited if the material damping of the wheel and rail is greater than the negative damping due to negative friction-creep curve. Flattening the friction-creep curve will reduce negative damping and reduce or inhibit squeal. The friction versus creep curve can be modified by chemical treatment or hard-surfacing of the rails with the babbit-like anti-screech material (*36*).

Modification of the friction-creep characteristic through chemical means is well founded. Surface segregation of aluminum in steels, for example, can modify the frictional characteristics of the contact zone (*37*). The friction-creep slope is perhaps steepest when the metals are of identical composition, as is the usual case at modern transit systems where wheel treads and rails are both manufactured from carbon steel. However, if an alloy is introduced into the rail head, or other “contaminant,” the slope of the friction versus creep curve can be modified, and, perhaps, inhibit squeal.

Hardfacing with Riflex Anti-Screech was evaluated at curved ballast-and-tie track in one of the maintenance yards at the WMATA system. Initially, the treatment was successful in eliminating squeal, reducing passby noise levels by about 20 dB. However, after 3 months of service, passby noise levels were reduced by only about 14 dB, relative to pretreatment noise levels, and there was occasional squeal noise. After 6 months of service, “chronic squeal reappeared” (*38*). Further, conversations with WMATA personnel indicate that the treatment was effective enough in controlling noise to replace it. The cause of the loss of performance is likely due to wear of the material, allowing wheel tread contact with the native rail steel.

The CTA has a 150-ft radius curve at the Linden Terminal treated with Riflex, though this curve is not in regular service. Wheel squeal is completely absent during a test with a train operated at walking speed. However, the vehicles also incorporate ring dampers, which are effective in controlling noise at curves, so that it is not clear how much of the squeal noise inhibition is due to Riflex and how much is due to ring dampers.

The SEPTA system has employed the Riflex Eteka-5 treatment at short radius curves, and found the treatment to be of limited effectiveness with respect to noise control. However, the treatment is effective in reducing rail wear at curves, and SEPTA intends to continue the treatment, because the cost of treatment is considerably less than rail replacement costs on the order of $400 to 500 per 39-ft section of standard “T” rail.

One of the principal problems with application of Riflex to U.S. transit system rails is that it has been applied to high carbon steel rails. An effective design might include installation of low carbon steel at curves and treatment with Riflex throughout the curve and short transitions at either end. However, this approach has to be compared with simply installing head hardened or alloy rail.

### 8.13.2 Costs versus Benefits with Respect to Wear Reduction

The Riflex/Eteka-5 treatment is usually performed on the rails prior to installation in track. The major differences between the Riflex rails and the standard carbon (or premium; i.e., alloy or heat treated) rails are the initial costs and the relative lives. (Note: the term Riflex is used to refer to the entire range of materials to include Riflex, Eteka-5, Anti-Screech, etc.) Because of the special work performed on the Riflex rails, the initial cost of these rails are significantly higher than that of standard or premium rails.

The initial cost is estimated to be about $1,500 per rail section, which may be compared with $400 (year 1995) per standard carbon steel rail section. There may be benefits which further reduce the cost of the treatment, or even produce a positive return on the investment. To assess the relative benefits and costs of the Riflex and Eteka-5 rails, it is necessary not to simply look at first costs, but to examine the total costs of maintaining the rails over the life of the rails. This type of analysis, often referred to as life-cycle costing, combines the initial costs, with the value of the future maintenance activities and future replacement costs.

A present worth analysis is carried out. The present worth analysis determines the cost or value of future maintenance and replacement activities in terms of costs *today*. This has the effect of converting a future stream of costs, be they maintenance, replacement, or other, into an equivalent *first* cost, so that various options, with different costs and maintenance streams, can be readily compared. Furthermore, this comparison is made on the basis of an “equivalent” initial or first cost (i.e., all future costs are brought forward to add to the initial cost to obtain a combined equivalent total cost, at the time of initial purchase or installation.) (This cost can be expressed in terms of an annuity cost, which may be more favorable for cost comparisons, as is the approach used in Chapter 6.)

The benefits of Riflex/Anti-Screech are compared with the costs of hand lubrication at curves on a daily basis, particularly sharp curves. (Note: hand lubrication is carried out by transit systems in limited high noise level areas.)

- Cost of the Riflex is $1,500 per rail section (as opposed to standard rail which costs approximately $400 per rail)
- Daily hand lubrication of the rail by a track inspector: $1.39 per day; 260 days/year ($362 per year)
- For a rail life of 20 years (reasonable for moderate curvatures on high density lines or sharp curves on moderate density lines) the ROI is 75%.
- For a 15-year life the ROI is 40%.

Thus, this analysis indicates that Riflex can be an attractive alternative to hand lubrication for the purpose of wear reduction at curves, provided that the Riflex survives over the anticipated life. Analysis of the benefits of Riflex for corrugation control show that it can be a cost effective alternative to grinding, with an improved ROI over rail grinding when the rail has a life of greater than 15 years.

However, the noise reducing capability of Riflex has not been shown to be good or reliable, and where Anti-Screech was effective at WMATA, its life expectancy was limited to months rather than years. This experience suggests that hand lubrication may, in the end, be preferable to Riflex for wheel squeal reduction, regardless of wear reduction capabilities. The reduction of corrugation remains as a possible positive benefit. Great care should be exercised in the selection of rail head inlays for noise control purposes.

### 8.13.3 Site-Specific Conditions

Riflex evidently cannot be applied to alloy rail, and thus may have limited usefulness at existing sections of track with alloy steel rail rails. This may have been the reason for the loss of performance of the Anti-Screech at WMATA, where the Anti-Screech was reputed to have curled out of the groove.

## 8.14 RAIL HEAD DAMPING INLAY (AQ-FLEX)

Elektro-Thermit GmbH, Germany, provides rail head treatment by the trade name of AQ-Flex, consisting of a synthetic resin glued to a groove in the rail head, for the reduction of squeal. The procedure has been applied for at least a year at German rapid transit systems, and can be applied to all grades of steel. No operational data have been obtained, but two transit operators which have employed the treatment are

Stadwerke am Main
Verkesbetriebe ZGW
Hanauer Lanstrasse 345
60314 Frankfurt/Main

Chemitzer Verkehrs
Zwickauer Strasse 164
09116 Chemnitz

Elektro-Thermit indicates that there is a subjective reduction of wheel squeal, though no data have been provided.

The vulcanization process is used with all types of rails and is applied so that the wheel does not come into contact with the resin based filler material. The noise is evidently supposed to be reduced by the material damping provided by the resin inlay. The treatment is intended as a substitute for Anti-Screech inlay treatment, also supplied by the same manufacturer. There are significant questions regarding actual performance, wear, and squeal noise reduction.

## 8.15 HEAD HARDENED, FULLY HEAT-TREATED, AND ALLOY RAIL

Head hardened and fully heat-treated rail are used by many properties for controlling rail wear and corrugation at short radius curves. A recent survey indicates that heat-treated and alloy steel rails exhibit substantially lower corrugation rates (*39*). No literature have been found concerning the stick-slip characteristics of carbon steel wheels on alloy steel rails.

## 8.16 FRICTIONLESS RAIL

The MBTA has installed 119-lb/yd rail with modified rail head section at the high rail of the Government Center curve of the Green Line to control wheel squeal. Figure 8-12 illustrates the rail section used. Also included in the track is 132-lb/yd restraining rail at the low rail. Automatic flange lubrication systems lubricate the back side of the wheel flange and thus the contact between the wheel and restraining rail. The trucks have resilient SAB wheels. The MBTA indicated that there was some reduction of noise after installation of the frictionless rail, but, after a period of time, the wheel squeal returned, ostensibly because of wear (*40*). Visual inspection of the curve indicates that there is substantial crabbing of the truck as it negotiates the curve, with the lead high rail wheel flange rubbing the gauge face of the rail and the trailing low rail wheel flange rubbing against the low rail gauge face; the guard rail is not preventing flange contact at the high rail. Moreover, the crab angle, or creep angle, between the rail and wheels is severe at the leading axle. Most or all of the squeal appears to be coming from the high rail side of the truck. Immediately after installation of the frictionless rail, flanging at the high rail might have been inhibited by the restraining rail. The reduced rail head width would help to reduce flange contact by effectively increasing gauge, though track gauge was measured to be 4′–8⅞″, only 3/8″ gauge widening. The noise reduction is attributed by the MBTA to reduction of friction by the reduced rail section, rather than by or in addition to flange contact inhibition. The return of wheel squeal noise suggests that if there was a change of wheel/rail contact friction, it reverted to normal conditions. There is the distinct possibility that reduction of flange contact with the frictionless rail was the principal cause of the initial noise reduction, and that re-establishment of flange contact because of wear of the restraining rail is the cause of the re-emergence of squeal. This could be checked by reducing the flange way between the restraining rail and low rail,which would also reduce the crab angle or creep angle between the wheel tread and rail. If this is successful, the same operating conditions could be obtained with standard rail profiles, rather than frictionless rail, by simply moving the tie plate further out. However, the contact area would be wider for the standard rail section relative to the frictionless rail section. At the present, the frictionless rail does not appear to be effective over the long term.

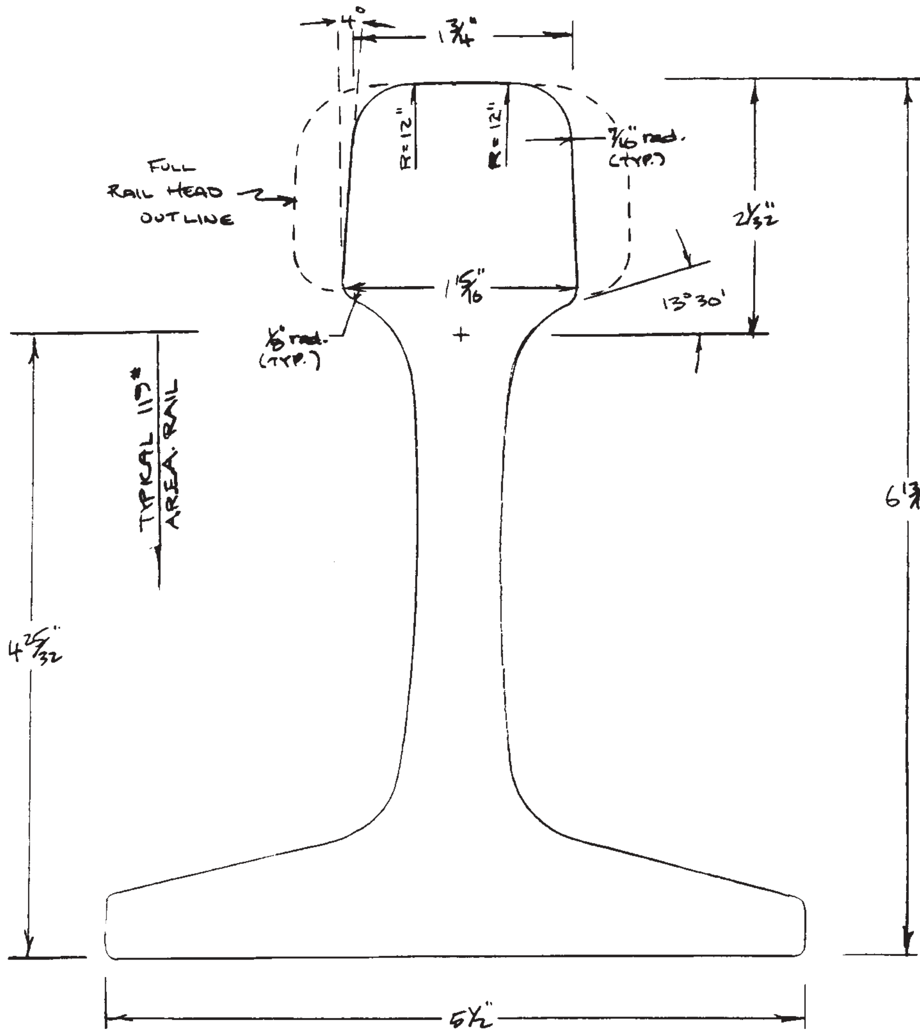


**FIGURE 8-12 FRICTIONLESS RAIL PROFILE - MBTA**

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