Controlling Rail and Wheel Wear On Commuter Operations

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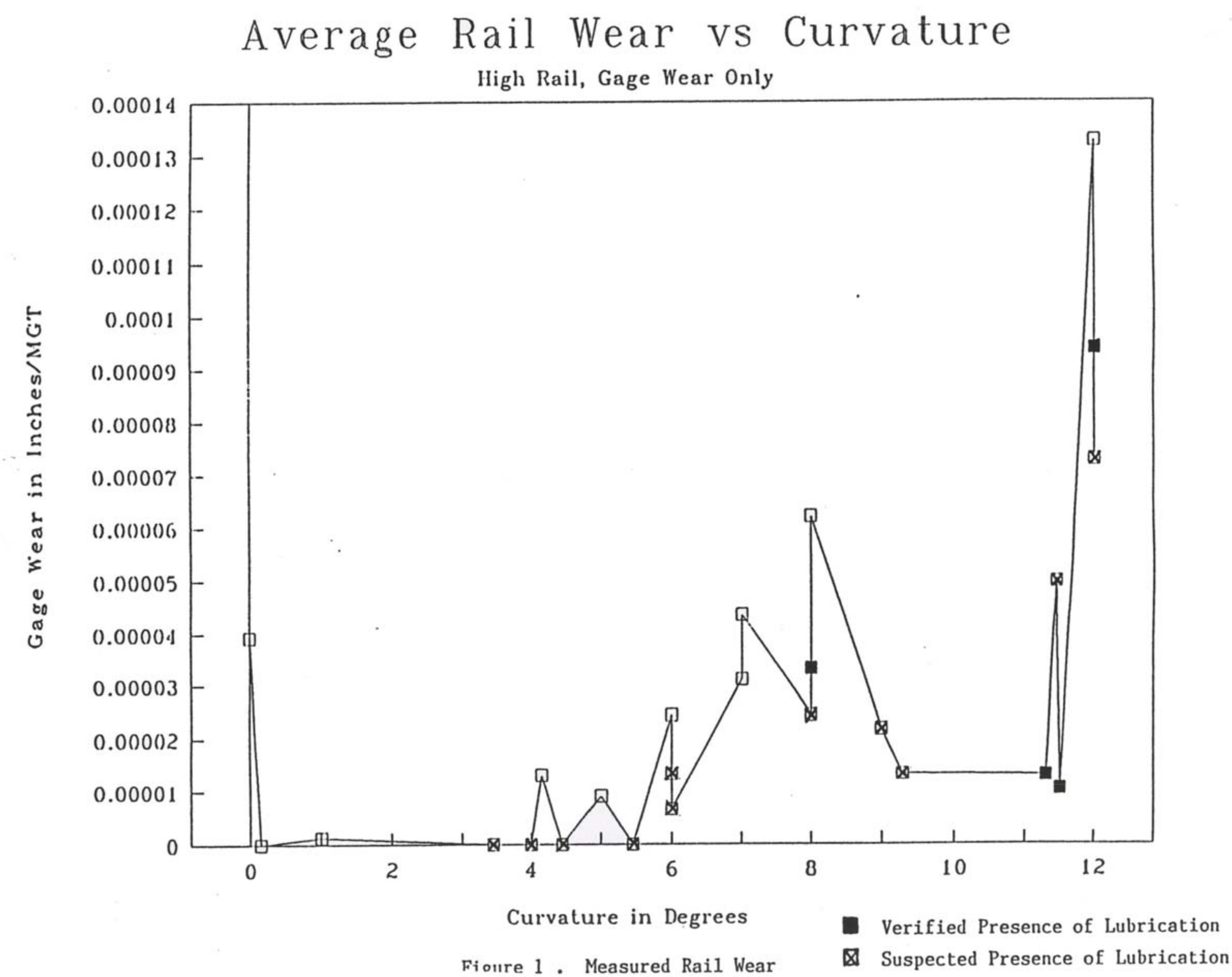
Introduction

The Regional Railroad Division of the Southeastern Pennsylvania Transportation Authority (SEPTA) operates approximately 450 route miles of commuter operation in the Philadelphia metropolitan area. In 1991, the railroad division began experiencing an unusually high rate of wheel and rail wear, particularly on sharp curves on the commuter rail lines. This wear was predominately wheel flange/rail gage face wear with an apparent correlation between the increase in wear on both the wheels and the rails.

The cause of the increase in wear was not clear or well defined, and as a result the most appropriate form(s) of corrective action was difficult to determine. In order to effectively address this issue, a comprehensive examination of the wear mechanisms, wheel and rail, was undertaken, together with a de- tailed study of the dynamic interaction between the wheel and the rail. As part of this detailed study, a series of specific activity areas were defined, and each of these areas were ad- dressed both individually and as part of the overall integrated study.

These specific study areas included the following:

1. Field investigation of rail and wheel wear.
2. Computer simulation of wheel/rail interaction and wear,
3. Analysis of wheel and rail profiles.
4. Analysis of rail grinding effects and requirements.
5. Evaluation of rail lubrication effectiveness.
6. Evaluation of track geometry (super-elevation and unbalance) effects.
7. Assessment of critical measurements and tolerances.
8. Economic benefit analysis.



Fiqure 1 . Measured Rail Wear

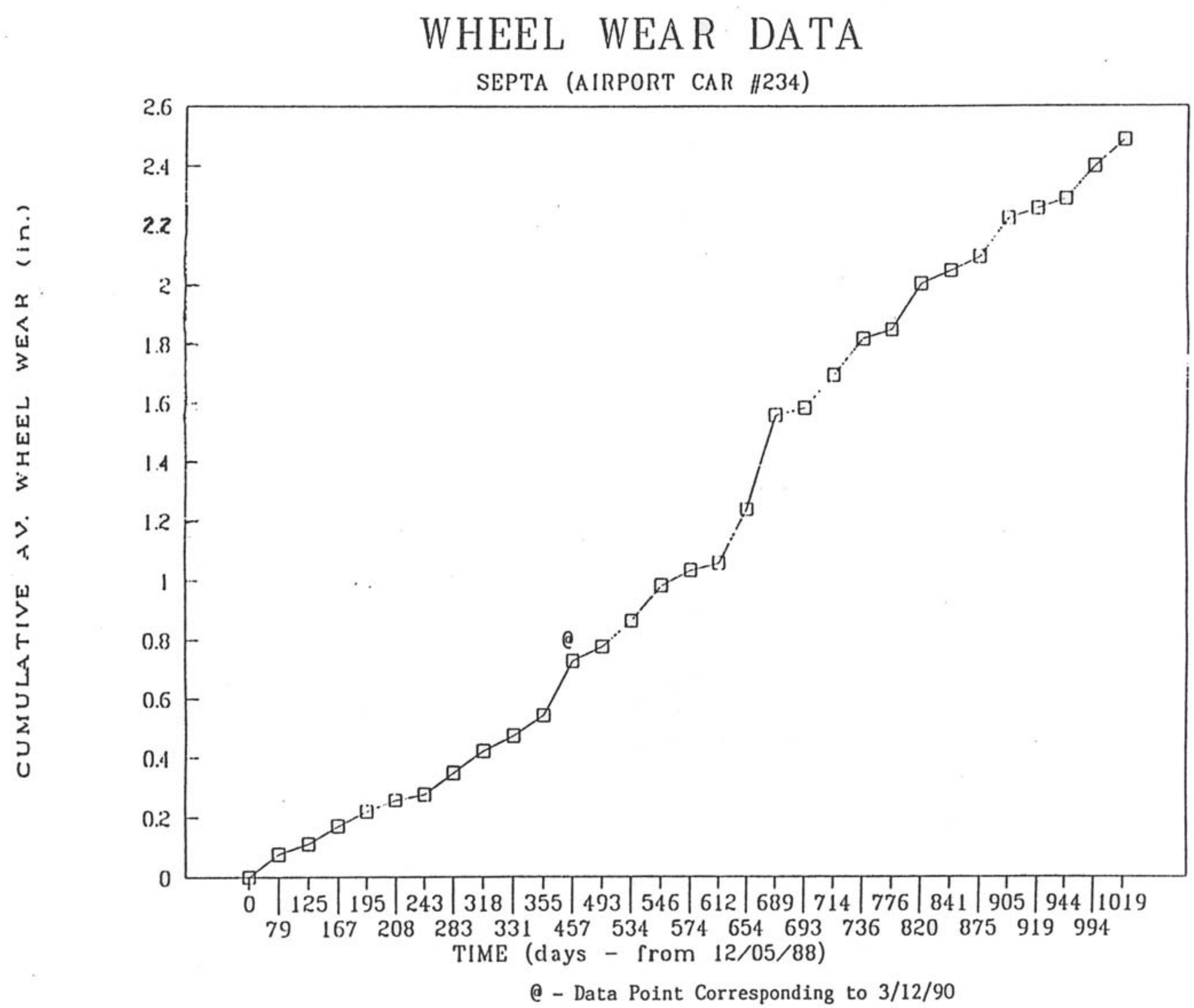


Figure 2. Wheel Wear Data; Car No. 234, 12/88 - 8/91

This paper presents a summary of each of these activities together with some of the overall finding and recommendations of this study. It should be noted here that this study was specific to the conditions and operations of the Railroad Division of SEPTA, and any extrapolation of these results to other properties should be done with extreme caution.

Field Investigation of Rail and Wheel Wear

Severe rail wear was observed and measured by SEPTA's Regional Railroad Division particularly on track with moderate to severe curvature. This observed rail wear was primarily gage face wear, with the specific mode of rail wear being abrasive wear as manifested by the rough surface of the rail gage face and the presence *of* visible metallic particles on the base of the rail, tie plates, etc.

In an effort to define and evaluate the rail wear, a series of wear measurements were taken on a broad range of curves within the Railroad Division. Rail wear measurements were made using a hand gauge at multiple locations on each measured curve. Figure 1 presents some representative measurement data, converted to units of rail wear (in inches/MGT). Specifically, Figure.1 presents the gage face wear rate as a function of curvature for the measured rails. Note: while there appears to be a trend towards the expected rate of wear increase with curvature, there are several well defined points which show either more or less wear than expected. Several of these points were determined to be locations where rail lubrication was present. Analysis of the data, combined with inspection of the track and rail conditions, deter- mined that non-uniform lubrication practice was a major cause of the variation in the rail wear rate.

In a manner analogous to that of the rail, severe wheel rail wear was also experienced by SEPTA's Regional Railroad Division. This had manifested itself in extremely short periods between wheel truing and wheel replacement.

The primary mode of the wheel wear was flange wear, and appeared to correspond to the high rate of gage face wear ob- served on the rail.

In order to analyze the wheel wear, the set of cars that operated in a dedicated service, i.e. the SEPTA airport line, were examined using wheel measurement data maintained by SEPTA during the car's regularly scheduled inspections.

Of the nine cars that normally operated on the airport lines, six of the cars had sufficient data as to allow for a detailed wheel wear analysis. Figure 2 presents the wear data from one of the cars, number 234, during the period between December 1988 and August 1991. It can be clearly seen in this Figure that the wear rate during the last 18 months (March 1990 to August 1991) was significantly greater than that during the first 15 months (December 1988 through April 1990). The other five cars similarly exhibited high wear rates during 1990-1991, corresponding to Figure 2.

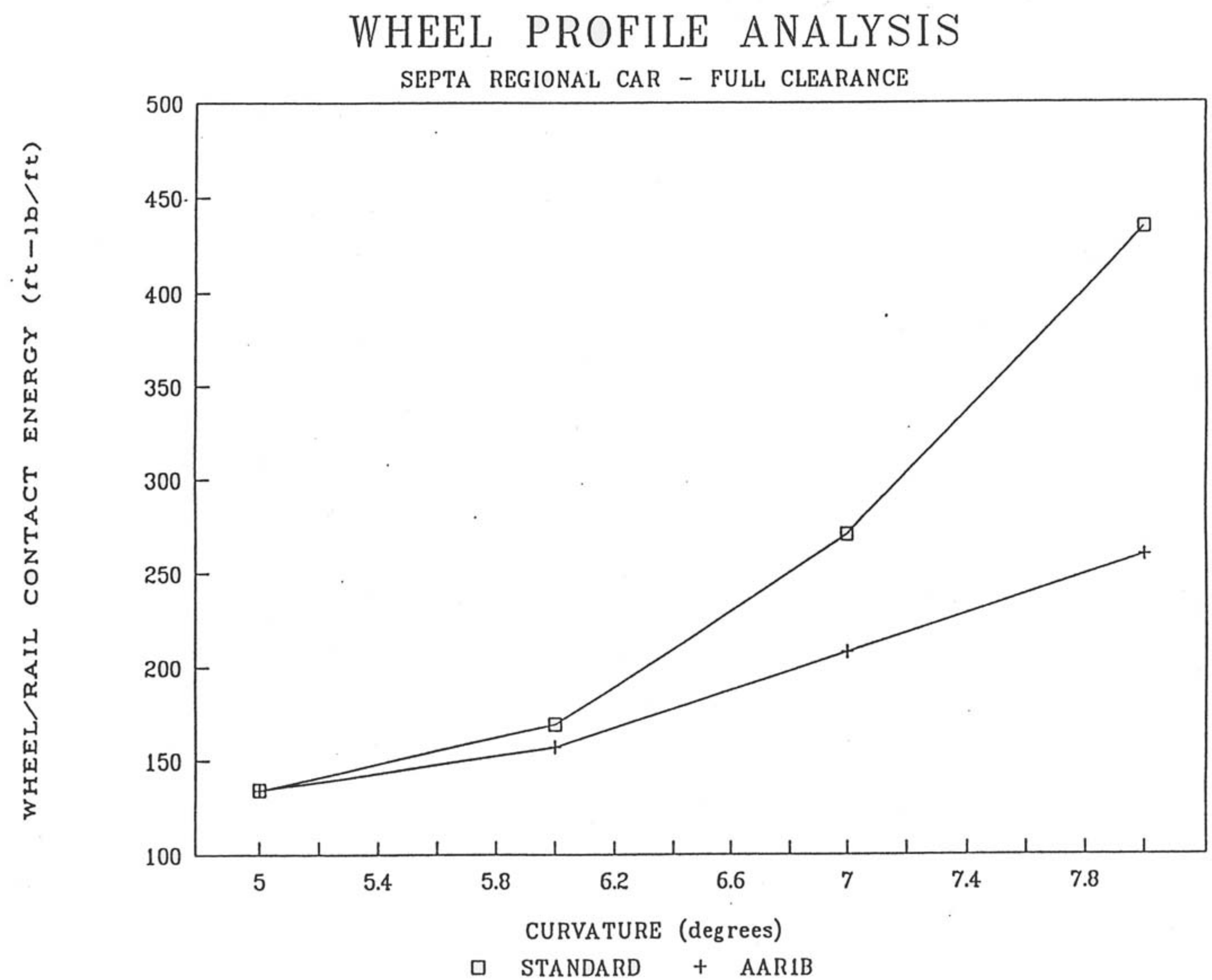


Figure 3. Comparison of Wheel/Rail Contact Energy for Different Wheel Profiles

Detailed Vehicle-Track curving Model

After the initial determination of the high incidence of wheel and rail wear, an analysis of the wheel/rail curving interaction was performed using ZETA-TECH Associates, Inc.'s VEHMOD vehicle-track curving model. This theoretical model has the capability of analyzing the curving behavior of a complete transit vehicle around a range of curvatures, and was used to examine the wheel/rail interaction as a function of the key track and vehicle parameters, to include both wheel and rail profiles.

The model calculates the following interim and final output parameters:

- Wheel/rail forces
- Energy dissipated at flange *(a* flange wear)
- Energy dissipated at tread *(a* tread wear)
- Forces between key car/truck components

The model is sensitive to all key design and operating parameters including:

- Car design
- Component stiffness
- Component geometry/tolerances
- Track geometry (curvature, elevation, . . .)
- Speed spiral geometry
- Wheel and rail profiles

The model was used to simulate operation of SEPTA car equipment across a full range of curvatures, wheel and rail profiles, and other key variable parameters. The specific results of each set of analyses will be discussed in the appropriate section of this paper.

Analysis of Wheel/Rail profiles

One key area of evaluation of wheel/rail dynamic interaction, wheel/rail forces, and resulting wear, was the effect of the profiles, both wheel and rail.

In the case of the rail, the concept of rail profile control (by rail grinding) was examined from the point of view of controlling rail wear, and in particular rail gage face wear [1]. The resulting analysis indicated that this approach of rail profile control by grinding was in fact appropriate for SEPTA, particularly on shallow to moderate curves. For more severe curves, where flanging will

occur, it was determined that this same approach was appropriate, but should be augmented with a one point contact between the flange of the wheel and the gage corner of the rail.

The specific profile recommendations for SEPTA included separate profiles for tangent track, high and low rails on shallow curves (3 degrees or less), and high and low rails on sharp curves.

In all cases, a rail head radius of between 8 and 10 inches was defined. In the case of tangent track, a centered contact zone, approximately 1 1/4" to 1 1/2" in width, was defined on both rails. For the shallow curves (3 degrees or less), where flanging could be controlled [1, 2], separate profiles for high and low rails were defined. For the high rail, the contact zone was set to be 0.4" to 0.6" in width located to the gage side of the rail center. For the low rail, the contact zone was set to be 0.50" to 0.75" in width located to the field side of the rail centerline.

For sharp curves, greater than 3 degrees, again separate profiles were defined. On the high rail, where flanging will occur, one point contact [2] was determined to be the most effective. The contact band on the high rail was correspondingly set to bean approximately 1.5" zone on the gage corner of the rail. (Note that the width changes slightly for new or worn rail.) For the low rail, the same contact zone as presented for shallow curves was to be maintained [2].

In a manner analogous to that of the rail, the SEPTA wheel profiles were examined from the point of view of minimizing wheel/rail wear. This analysis was performed using the wheel/rail curving model described previously.

SEPTA's current wheel profiles are standard AAR 1:20 profiles. This profile was compared to the newly developed AAR 1B profiles, from the point of view of minimization of wheel/rail contact energy at the interface between the wheel flange and the gage face of the rail.

Figure 3 presents a comparison of the energy dissipated at the wheel flange for these two profiles (AAR 1:20 vs. AAR 1B). As can be seen from this figure, the AAR 1B profile shows a reduced level of energy dissipation for a wide range of curvatures. This translates into a lower wheel/rail wear rate.

Based on these model results and additional model analysis, it was determined that The AAR-1B profile improved the effective conicity of the wheels, thus improving steering of the wheelsets.

This then reduces flanging on moderate curves. Based on this, the AAR-IB profile was recommended for control of wheel-rail wear on SEPTA. However, it was noted that the AAR-IB is more susceptible to hunting. While hunting is not currently a concern for SEPTA's present equipment, it was noted that hunting or potential hunting situations should be monitored during the initial introduction of the IB profile to insure that no new problems develop.

Rail Grinding

Rail grinding was carefully examined from the point of view of controlling and maintaining the rail head profile (see previous discussion) as well as for control of surface defects on the rail itself. Grinding has come into extensive use on railway systems to control rail surface defects such as corrugations, battered welds, engine burns, etc, as well as being part of an overall rail maintenance approach, based on the control of the wheel/rail contact zones through the grinding of special "profiles" onto the rail head. This latter use has a direct application in the control of both rail and wheel wear, as already noted.

Rail grinding was therefore used for both profile control and control or surface defects. In the former case, rail profile grinding was defined as the approach required to establish the recommended rail head profiles. Noting that any rail profile will deteriorate under traffic, it is necessary to periodically monitor the profile and to regrind the rail head (maintain the profile) when the profile deteriorates to the point where it is no longer functioning properly. If this profile maintenance is .not carried out (i.e., if the profile is allowed to deteriorate and is not restored), then the benefits of the ground profiles will no longer continue. Consequently, profile grinding must be considered an ongoing maintenance activity with periodic "maintenance" grinding required to retain the optimum rail head profile.

Based on SEPTA's traffic levels and measured rail condition, it was determined that profile grinding to restore and maintain the optimum rail profile is required one to two times a year.

In addition to profile control, grinding is also used to control surface defects such as corrugations, battered welds, engine burns, and other rail surface defects [3]. Based on the analysis of SEPTA's rail conditions, a series of recommendations for controlling surface defects by grinding was developed, see Table 1.

In addition, the required annual grinding program for SEPTA's Railroad Division was estimated to be of the order of 360 pass miles per year, based on conventional grinding (defects) only, and 540 pass miles per year, based on profile grinding (which was the recommended approach). These estimates were based on the use of a small 20 motor grinding machine that has full motor adjustability. As part of this program, a formal rail grinding planning and evaluation procedure was developed consisting of the following steps:

a. Pre-grind evaluation
b. Procedures for grinding equipment
c. Intermediate evaluation
d. Evaluation of grinding effectiveness
e. Use of measurement tools

TABLE 1 Rail Grinding Standards for Surface Defects

* Corrugations

- Grind when 50% of corrugations .012 -.015"
- Depth after grinding < 0. 005"

* Weld Conditions

- High welds grind when 25% > .015"
- Height after grinding < 0.005"
- Low welds ground when 25% > .025"
- Depth after grinding < 0.010"
- Low welds> .040" should be welded

* Other Surface Defects

- Grind engine burns when 50% > .020-.025"
- Depth after grinding < .010"
- Engine burns> .040" should be welded
- Grind new rail within 20 MGT of installation

Rail Lubrication

Lubrication plays an important role in the reduction of rail and wheel wear, and the extension of rail life in high wear locations on track. Experience on other properties, as well as controlled test data, has shown that the wear life of the rail can be increased from two to ten times with effective lubrication [4].

SEPTA has been using a mix of wayside lubricators and hand lubrication for its rail lubrication needs. The wayside lubricators, which are distributed throughout the system, were frequently turned off because of concern for the lubricant (rail grease) getting to the top of the rail head and causing difficulties in train braking. The use of hand lubrication was intermit- tent and manpower intensive and did not appear (based on observations and measurement) to be applied in sufficient quantity and frequency to properly lubricate the system, given the level of traffic and degree of rail and wheel wear.

TABLE 2

Effect of Unbalance

CURVE	UNBALANCE (INCHES) 0	1	2	3	4	5
1	1.00	1.08	1.17	1.25	1.34	1.42
2	1.00	1.06	1.12	1.18	1.24	1.30
3	1.00	1.05	1.10	1.15	1.19	1.24
4	1.00	1.04	1.08	1.13	1.17	1.21
5	1.00	1.04	1.08	1.11	1.15	1.19
6	1.00	1.03	1.07	1.10	1.14	1.17
7	1.00	1.03	1.06	1.10	1.13	1.16
8	1.00	1.03	1.06	1.09	1.12	1.15
9	1.00	1.03	1.06	1.08	1.11	1.14
10	1.00	1.03	1.05	1.08	1.11	1.13

Analysis of SEPTA's lubrication practices indicated that it could be improved significantly to extend the wear life of rails in curves. It was estimated that more "effective" rail lubrication practices could extend rail wear life to two or more times that of reported curve rail life, in many cases. However, in order to achieve this level of improvement, a proper combination of lubrication techniques was required. This included replacement of the hand lubrication with more effective lubrication techniques, such as a hi-rail mounted system for lubricating curves or a more comprehensive wayside lubricator system. Such a hi-rail vehicle could apply a layer of lubricant, accurately to the gage face of the rail, continuously along the curves. Alternately, a more extensive wayside lubrication system was required, operating at a high level of effectiveness. In order to accomplish this, a strong emphasis has to be placed on lubricator maintenance with the use of dedicated lubricator maintainers.

In all cases, the level of lubrication was to be carefully monitored to insure that an adequate level of lubricant was obtained, while minimizing the movement of lubricant to the top of the rail head.

Superelevation and Unbalance

Wheel/rail forces, and particularly wheel/rail lateral forces, will increase as a function of increasing overbalance or increasing superelevation. This is clearly illustrated in Table 2, which shows the relative effect of unbalance on wheel/rail forces for a broad range of curvatures. Since wheel flange wear and rail gage face wear are directly related to the wheel/rail lateral forces, decreasing the amount of allowable unbalance will reduce the level o£ wear, all other factors constant.

However, unbalance permits the operation of higher speeds on track within a defined safety envelope. Thus, reducing allowable unbalance will directly reduce the operating speeds, and have an undesirable impact on overall operations. There is a direct and inverse trade off between operations and wear in this condition.

Noting the effect OT large unbalance values on wheel/rail wear, it was suggested the maximum amount of unbalance be limited to three (3) inches, so as to avoid the large increment in force (and thus wear) that appears to occur for unbalance values greater than 3 inches. However, this was based on wear considerations only, not safety consideration, and as such it operational requirements dictated a higher level of unbalance, this could be accommodated, provided it is within the overall safety envelope of the vehicles [5]. It is this safety criterion that should limit the maximum allowable level of superelevation, as well as elevation [6].

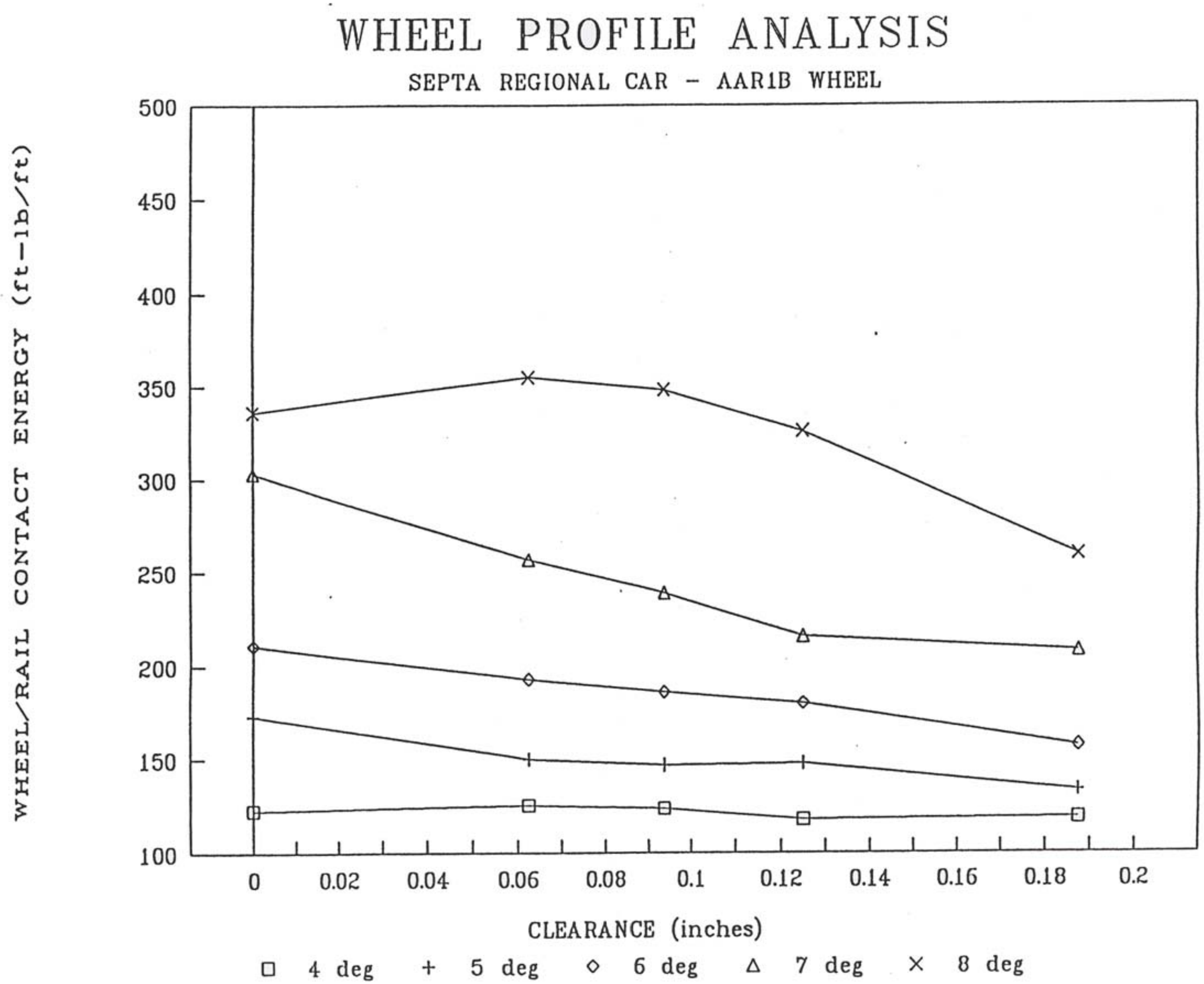


Figure 4. Effect of Pedestal Liners Wear Plate Gap on Wheel/Rail Contact Energy

Critical Measurements and Tolerances

Allowable component tolerances and wear limits can directly effect the wheel and rail wear of the system. For example, in the case of mechanical equipment, any increase in the friction between key truck and car body components can result in increased energy requirements, poor curving performance, and thus increase in wear at the wheel/rail interface.

This latter behavior was specifically observed in the center plate liners (wear plates) used on SEPTA equipment. One design of replacement wear plates was found to have a significantly increased coefficient of friction than the original wear plate. Analysis of the effect of this increased friction wear plate showed a loss in curving performance, which in turn resulted in increased lateral wheel/rail forces and increased energy at the wheel/rail interface. This increase in energy dissipated at the wheel/rail interface translates directly into an increase in both wheel flange and rail gage face wear. Based on this analysis it was recommended that low coefficient of friction center plate liners (wear plates) be used on all SEPTA equipment.

In addition to the above analyses, a parametric study of mechanical tolerances and sensitivities was performed with particular emphasis on key truck clearances, both lateral and longitudinal. Based on this analysis it was found that the gap at the pedestal liner wear plate can effect wheel/rail contact energy. This behavior is illustrated in Figure 4 which shows the effect of these gap dimensions on the contact energy for the AAR- 1B wheel profile (the wheel profile recommended in this study). As can be seen from this figure, a very small gap, i.e. less than 3/32", will have a greater level of energy dissipation for a wide range of curvatures. This translates into a higher wheel/rail wear rate. Increasing the gap to a range of between 3/32" and 3/16" for both the lateral and longitudinal gaps results in a reduction of contact energy at the wheel/rail inter- face. This in turn translates into a lower wheel and rail wear rate.

Rail Wear Standards

Rail wear is one of the key reasons for the removal of rail from track, particularly for transit and commuter track. While there have been significant changes in the criterion used to determine when rail should be replaced in track, rail wear is the dominant replacement criterion for track with light axle loadings, such as transit systems or commuter railways.

Rail wear limits have traditionally been set for jointed track to avoid the flanges of the rolling stock from striking the top of the joint bars. However, with the use of continuously welded rail and the virtual elimination of rail joints, the bending strength of the worn rail section defines the rail wear limits [7].

While permissible rail wear limits have been defined to include vertical head loss, side (gauge face) head loss, head area loss, and maximum angle of side wear, only the first two have been used on an active basis in North America because of their ease in measurement and relative simplicity of application. However, interest in the maximum angle of side wear has significantly increased on North American transit and passenger systems because of the risk of wheel climb and associated derailment of equipment. As a result, several North American properties have recently incorporated gage face wear angle limits to their existing rail wear standards [7].

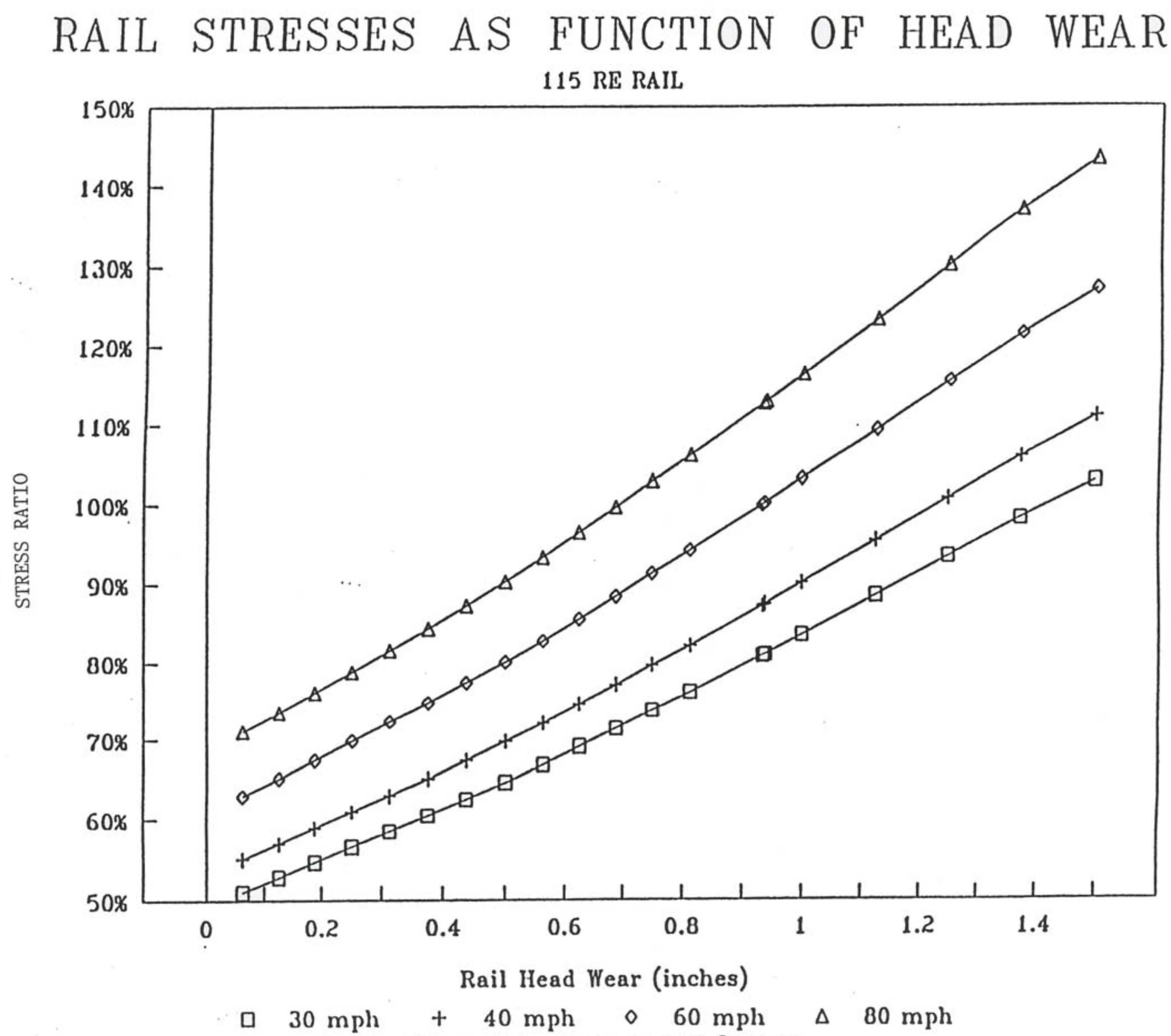


Figure 5. The Effect of Head Wear on Rail Stresses

Vertical head loss refers to the loss of metal from the top surface of the rail head. While it occurs on all track, it is a dominant mode of wear on tangents and shallow curves. vertical head loss is measured independent of any other parameter, and is taken at the center of the rail head. Gage face wear, or side head loss, refers to the loss of metal from the side of the rail head. Gage face wear occurs primarily on curved track when flanging of the wheels occurs. Gage face wear is usually measured 5/8" below the top of the rail head. Thus, its exact location is dependent on the degree of vertical head wear.

In order to properly define these limits, the strength of the rail, in bending, is calculated using the rail's beam strength and Beam on Elastic Foundation theory [7,8]. As the rail experiences wear, both head and gage face wear, it's beam strength is reduced, thus increasing the stresses in the rail caused by the rolling stock and their associated wheel loadings. Figure 5 presents the results of such an analysis for 115 RE rail showing the loss of rail strength (and thus rail head wear) as a function of maximum operating speed.

This analysis was performed for each of the major rail sections of interest to SEPTA: 115 RE, 100 ARA-B, and 132 RE. The resulting wear limits, both head and gage face wear, were determined for each of these sections and presented in Table 3. In addition, a maximum allowable gage face angle value is also presented based on wheel climb criterion (safety).

Table 3 Rail Wear Limits

Rail Section	Max Head Wear (inches)	Max Gage Face Wear (inches)
100 ARA-B	3/8 - 7/16	3/8 - 7/16
115 RE	5/8	1/2 - 5/8
132 RE	3/4	5/8 - 3/4

* Replace if wheel flanges strike top of joint bars

* Maximum allowable angle of side wear 32 degrees from the vertical

Economic Benefit Analysis

As part of this overall activity, cost-benefit analyses were performed for those key maintenance activities where significant additional cost outlays may be experienced by SEPTA. Specifically, the following maintenance areas were analyzed:

* Economics of rail lubrication
* Economics of rail grinding.

These analyses determined the benefit {in reduced component requirements and corresponding costs) of the proposed maintenance activity, together with the costs associated with effectively and adequately implementing these activities [9].

The primary benefit associated with increased lubrication is a significant reduction in the rate of wear of the rails in track. Additional benefits associated with lubrication include a reduction in wheel wear, which corresponds to the reduced rail wear, and a reduction in fuel (energy) usage [10]. Table 4 presents a detailed breakdown of the savings and costs associated with increased lubrication. These result in a Return on Investment for rail lubrication (ROI) of between 85% and 89%. Note that this does not include any energy savings.

The primary benefit associated with rail grinding is a significant reduction in the rate of rail replacement. The largest such benefit is due to rail profile grinding, while a second benefit is also associated with the reduction and/or elimination of rail surface defects such as corrugations, engine burns, battered welds, etc. [11] Additional benefits associated with grinding include a reduction in wheel wear, which corresponds to the rail wear reduction due to profile grinding. In addition, elimination of corrugations and surface defects will provide benefits in the area of track surfacing and energy (fuel) consumption.

The net savings due to grinding varies significantly based on whether conventional or profile grinding is performed. For conventional grinding, the net savings correspond to an ROI for grinding of approximately 47%. For profile grinding, the net savings correspond to an ROI of approximately 81%. These results are summarized in Table 5.

TABLE 4

Economics of Lubrication
Summary of Costs and Benefits
Rail and Wheel Savings Only

	Moderate Lubrication	"Good" Lubrication
Annual Savings/Cost		
Rail Savings	$109,600	$17l,873
Wheel Savings	$153,524	$293,091
Total Savings	$263,124	$464,964
Annual Cost of Lubrication		
Capital Cost	$ 27,333	$ 36,445
Maint Costs	$ 90,090	$166,320
Grease Cost	$ 21,622	$ 48,048
Total Cost	$139,045	$250,813
Net Savings	$124,079	$214,151
ROI	89%	85%

TABLE 5

Economics of Grinding
Summary of Costs and Benefits

	Defect Grinding	Profile Grinding
Annual Savings/Cost		
Rail Savings	$113,153	$169,730
Wheel Savings	$ 63,216	$153,524
Surfacing	$ 2,066	$ 4,806
Energy Savings	$ 2,776	$ 6,310
Total Savings	$181,211	$334,369
Cost of Grinding	$123,000	$184,500
Net Savings	$ 58,211	$149,869
Return on Investment	47%	81%

Conclusions

The wheel and rail wear that was being experienced by SEPTA's Regional Railroad Division was predominately wheel flange/rail gage face wear, particularly on moderate to sharp curves on the commuter rail lines.

In order to analyze the wheel/rail dynamics associated with SEPTA's operations and to evaluate the effects of changes in maintenance practices on wheel and rail wear, a vehicle-track curving computer model was used to analyze the curving behavior of a complete SEPTA rail vehicle. Based on this model analysis, together with the analysis of the wheel and rail behavior itself, a serious of recommendations for changes in maintenance practices, policies, and standards were developed.

Among the recommended areas or change were the use of improved rail and wheel profiles. In the case of the rail profiles, use of an asymmetrical rail head profile, obtained through rail profile grinding, was suggested as a means of controlling rail wear, and in particular rail gage face wear. Specifically, a profile that generates and maintains "one point contact" between the flange of the wheel and the gage corner of the rail was developed for the sharper curves, while alternate (different) profiles were developed for tangent track and the high and low rails on shallow curves (3 degrees or less).

In addition to the control of the rail profile, rail grinding was also recommended for the control of corrugations, battered welds, engine burns, and other rail surface defects.

For the wheel profiles, SEPTA's standard AAR 1:20 profile was compared to the newly developed AAR 1B profile. Based on model analysis, it was determined that the AAR-1B profile improves the effective conicity of wheel, thus improving steering of wheelset, and as a result, the AAR-1B reduces flanging, and thus wear, on moderate curves. Based on this analysis, the AAR-1B profile was recommended for control of wheel-rail wear on SEPTA.

Another area of emphasis that emerged from this study was that of proper lubrication. It was ascertained that more "effective" rail lubrication practices can extend rail wear life to two or more times that of current curve rail life, in many cases. In order to achieve this, it was recommended that SEPTA investigate and implement a hi-rail mounted system for lubricating its curve or a more comprehensive wayside lubricator system. Such a hi-rail vehicle could apply a layer of lubricant, accurately to the gage face of the rail, continuously along the curves. Alternately, a more extensive wayside lubrication system would be required. However, the effectiveness of these wayside lubricators must be increased significantly by placing a strong emphasis on lubricator maintenance.

While wheel/rail lateral forces, and thus wheel/rail wear, will increase as a function of increasing overbalance or increasing superelevation, unbalance permits the operation of higher speeds on track, within a defined safety envelope. Thus, reducing allowable unbalance will directly reduce the level of safe operating speeds, and have an undesirable impact on overall operations. There is a direct and inverse trade off between operations and wear in this condition. Noting that there are other ways to effectively control wheel/rail wear, in the case of SEPTA operations, other techniques such as increased lubrication and grinding represent a higher priority for the control of wheel and rail wear.

Mechanical equipment tolerances and wear between key truck components can likewise affect wheel/rail wear, as can an in- crease in the friction between key truck and car body components. This latter behavior was specifically observed in the center plate liners (wear plates) used on SEPTA equipment. Analysis of the effect of this increased friction wear plate showed a loss in curving performance, which resulted in increased lateral wheel/rail forces and thus an increase in both wheel flange and rail gage face wear. Use of low coefficient of friction center plate liners (wear plates) would significantly reduce this effect.

Rail wear limits were also redefined; to include: vertical head loss, side (gauge face) head loss, and maximum angle of side wear. Making use of the strength of the rail, in bending, allowable limits for both head and gage face wear were calculated for each of the major rail sections of interest to SEPTA: l15 RE; 100 ARA-B and 132 RE. In addition, a maximum allowable gage face angle value was established based on wheel climb criterion (safety).

Finally, in order to determine whether many of these "improvements" were in fact economically justifiable, a cost-benefit analyses was performed, with specifically emphasis on the economics of rail lubrication and the economics of rail grinding.

The resulting analyses showed a Return on Investment (ROI) for rail lubrication of between 85% and 89% and an ROI for rail grinding of between 47% and 81%. Thus, the recommendations for reducing rail and wheel wear appear to be economically justifiable as well as technically feasible.

Recent application by SEPTA of several of these recommendations has resulted in a noticeable decrease in the rate of wheel and/or rail wear, and it is expected that with the full incorporation of these recommended changes in practice, wheel and rail life will experience further improvements.

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