EFFECTS OF INCREASED GROSS RAIL LOAD ON 36-INCH DIAMETER FREIGHT CAR WHEELS

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ABSTRACT

This paper discusses the effects of gross rail load (GRL) on the performance of 36-inch diameter freight car wheels in North American service. Association of American Railroads (AAR) Car Repair Billing (CRB) data are reviewed to show the types of wheel removals that typically occur in North American freight service. Historical trends in CRB data are noted, with emphasis on service-related wheel removals. The root causes of several wheel defects are briefly described. Wheel/rail contact stresses are reviewed for wheels with "ideal" and "non-ideal" tread profile conditions. Wheel/rail impact loads are discussed with particular emphasis on the possible negative effects of such loading on wheel failures such as shattered rims. Thermal loading of the wheel due to tread braking is considered and AAR S-660 finite element analysis computer simulations are performed for four different 36-inch one-wear wheel designs using 100-ton and 125-ton loading conditions. Finally, recent work relating the effects of drag braking to the level of residual compressive wheel rim hoop stress is reviewed. Throughout the paper recommendations are provided regarding practices that will benefit wheel safety and where the authors feel that additional industry research work will provide benefits.

INTRODUCTION

There is no question that the North American freight railroad service environment is among the most demanding in the world. Higher axle loads, long grades with drag braking, improved car utilization and large wheel/rail impacts all result in a severe test for railroad wheels every day in service.

In recent years North American railroads adopted 286,000 pounds gross rail load (GRL) as the maximum allowable load for 100-ton service, and a majority of freight cars currently operate at this level. Such cars use 36-inch diameter wheels. A smaller percentage of cars now operate at 315,000 pounds GRL (125-ton service), but use larger 38-inch diameter wheels. The allowable 100-ton load was previously 263,000 pounds GRL. Wheel service-related problems, particularly shelling and spalling, have increased in recent years, and arguably some of this increase can be attributed to the higher GRL.

Recent discussions within the railroad industry to further increase the generally accepted GRL to 315,000 pounds GRL (125-ton service) led the authors to write this paper. There is a desire on the part of some to use the cheaper 36-inch wheel in 315,000 pound GRL service rather than the more expensive 38-inch wheel. Increased GRLs will affect many mechanical and maintenance of way components, including wheels.

AAR 2000 CRB DATA

A more complete discussion of year 2000 AAR CRB data is included in another paper to be presented at this RWMEC conference (Sullivan, Lonsdale and Kezmarsky, 2001). However, data for selected "Environmental" wheel removals are contained in Table 1. These service related removals (which numbered about 98,000 wheels, or 21% of the nearly 470,000 total removals in year 2000) include shelling, built-up-tread, slid flats, out-of-round and thermal cracks.

Why Made Code	Number Of Wheels
67 = Out-of-Round	4,660
74 = Thermal Cracks	5,609
75 = Tread Shelled	53,060
76 = Tread Built-Up	9,175
78 = Tread Slid Flat	25,564

Table 1. Selected CRB Environmental wheel removals.

DESCRIPTION OF SELECTED WHEEL DEFECTS

In this portion of the paper attention will be paid to the causes of environmental wheel defects. It is the authors' position that the number of wheels removed for such service-related defects is unacceptable and resources should be allocated to reduce such failures. Standard Steel therefore supports AAR Wheel Research Consortium efforts.

Environmental wheel removals represent a large opportunity for maintenance related savings. A financial analysis presented in another RWMEC paper (Sullivan, Lonsdale and Kezmarsky, 2001) shows a $32 million cost for 2000 environmental CRB wheel removals. Further, a potential 10-year savings of $228 million (net present value) was calculated if only AAR CRB removals were considered. If certain assumptions are made and all North American wheel removals are considered, a $572 million net present value is calculated.

True wheel shelling

True wheel shelling is a rolling contact fatigue phenomenon that leads to damage on the wheel tread and eventually small pieces of the tread break off. Thus, true shelling is not related to wheel sliding, braking systems or braking ratio, and is limited to certain specific heavy-haul service lanes. It is now generally accepted that most shelling wheel removals in North America (why made code 75) are actually for wheel *spalling*.

Thermal mechanical shelling

Thermal mechanical shelling, known to exist in some service lanes, is a process that requires elevated temperatures in conjunction with contact stresses. In thermal mechanical shelling, the fatigue cracking is a result of elevated tread temperature (which reduces the strength of the tread material) and the high contact stresses, which would do no damage if the wheel tread was at its room temperature strength.

Spheroidized pearlite at the tread surface, resulting from brake shoe heating, can be an important clue when looking for thermal mechanical shelling. No martensite is formed during thermal mechanical shelling and the cracks initiate at the surface. Often, these cracks are found on wheels with heat checks and the cracks appear to be closely related. In the case of spalling, the crack network is either perpendicular or parallel to the surface.

Wheel spalling

Wheel spalling occurs in service after the wheel slides on the rail and patches of martensite are formed on the tread. The wheel slide generates very high temperatures at the contact patch and the steel is austenitized. However, the large heat sink of the remaining cold wheel quickly quenches the small tread patch and untempered, hard, brittle martensite is then formed. The combined effects of the localized contact patch heating during the slide (compressive upset plastic deformation) and the 4% volume increase due to the martensitic transformation result in a tensile stress field around the martensite patch. Subsequent in-service loading leads to eventual cracking and fracture of pieces from the wheel tread.

Due to the brittle nature of martensite and the stress concentrations formed when pieces come off the tread, wheel spalling can result in severe progressive damage to the wheel. The appearance of shelling, spalling and thermal mechanical shelling can be very similar, particularly if the spalling extends around the wheel tread from repeated sliding and skidding, but spalling can often be corroborated by slip marks/patches on the wheel tread.

Built-up tread and slid flats

For built-up tread (why made code 76), the AAR (2001) specifies that “A wheel is condemnable whenever the tread has built-up metal 1/8 inch or higher on the wheel tread.” This defect is the responsibility of the car owner. Slid flat wheels (why made code 78) are the responsibility of the handling line and are condemnable if the flat spot is “two or more inches in length” or there are “two or more adjoining spots each 1-1/2 inch or over in length.” Clearly, these defects can cause high impact loads and damage to wheels, rails, lading and the car.

Out-of-round wheels

To be condemnable under AAR rules, out-of-round wheels must register at least 90,000 pounds on a wheel impact load detector and have a verified out-of-round “runout” of 0.070 inches (AAR, 2001). The effects of wheel/rail impacts from such wheels will be discussed later in the text, as will thermal cracking of the wheel tread.

AAR APPROVAL FOR WHEEL DESIGNS

Currently the AAR requires that an S-660 finite element analysis (FEA) computer simulation be conducted to determine the stress levels present in new wheel designs (AAR, 1998). Mechanical loads (vertical and lateral) and braking (thermal) loads are applied to the wheel using the simulation. The resultant stresses are then compared to the AAR database of stresses for those wheel designs that are successfully performing in railroad service. Based upon this comparison of stresses, new wheel designs are accepted or rejected by the AAR.

We note that 100-ton loading conditions in AAR S-660 used to analyze and approve wheel designs are appropriate for 263,000 pounds GRL. Thus, 36-inch wheels now operating at 286,000 pounds GRL were approved using loading criteria that are about nine percent less than actually found in service.

GRL AND MECHANICAL WHEEL LOADS

Table 2 shows the wheel load for various different GRLs, assuming an eight-wheel freight car.

Gross Rail Load, pounds	Wheel Load, pounds
263,000	32,875
286,000	35,750
315,000	39,375

Table 2. GRL and wheel load.

WHEEL/RAIL CONTACT STRESSES

The wheel/rail contact patch is normally about the size of the U. S. dime, and therefore the contact stresses at that location can be quite high. Using the wheel load data shown in Table 2 and the method of Magel and Kalousek (1996), contact stresses for a 36-inch wheel and a 38-inch wheel were calculated and are shown in Table 3. For all calculations the same “ideal” wheel/rail profile conditions are assumed. Note that 38-inch wheels have slightly lower maximum contact stresses than 36-inch diameter wheels. This is due to the larger longitudinal radius of the 38-inch wheel.

GRL (pounds)	36" wheel maximum contact stress (psi)	38" wheel maximum contact stress (psi)
263,000	157,400	154,600
286,000	161,900	159,000
315,000	167,200	164,200

Table 3. Maximum contact stresses, "ideal" tread profile.

Wheel/rail conditions can be far from ideal in actual railway service. If wheel/rail contact occurs at the flange root of the wheel, or at a false flange on the tread that has developed due to wear, contact stresses can become extremely high. High contact stresses can lead to additional wheel defects, such as shelling, in service. Thus, reprofiling of wheel treads to avoid adverse wheel/rail contact situations will become even more critical if the allowable GRL is increased to 315,000 pounds.

Maximum contact stress values for 36-inch wheels with "non-ideal" false flange and flange root contact conditions, again using the equations provided by Magel and Kalousek (1996), are shown in Table 4. Note that these are very high contact stresses that will damage the tread surface and could lead to additional fatigue cracking in the wheel rim. For example, the 315,000 pounds GRL false flange contact stress is approximately *four times* the yield strength of AAR Class C wheel rim steel.

GRL (pounds)	36" wheel maximum contact stress, false flange (psi)	36" wheel maximum contact stress, flange root (psi)
263,000	379,400	368,200
286,000	390,100	378,600
315,000	402,900	391,000

Table 4. Maximum contact stresses for "non-ideal" wheel/rail profile conditions.

WHEEL IMPACT LOADING

The effect of wheel impact loading from tread defects on wheel life is currently not well understood. Impact fatigue associated with dynamic, high strain rate loading is significantly different than fatigue associated with normal fatigue loading. A paper to be presented at the 13th International Wheelset Congress (Stone, Lonsdale and Kalay, 2001) suggests that the fatigue endurance limit (service stress to insure infinite fatigue life) of wheels under repeated impact fatigue loading is 1/3 lower than the endurance limit for normal in-service fatigue loading. This hypothesis is based on other published work showing that impact fatigue loads are more damaging for steels than are "normal" fatigue loads.

Consider that a 36-inch wheel makes one complete revolution every 9.4 feet in service. Thus, if such a wheel has a tread defect that causes a dynamic impact load of 100,000 pounds, the wheel will experience 562 impact fatigue cycles of 100,000 pounds per mile. This clearly is not beneficial for wheel "health."

A paper by researchers at Canadian National (CN) Railway (Clegg and Blevins, 1996) reviewed their experience with wheel impacts. CN has seen that out-of-round wheels with "healed" shells, where tread metal flows over and smooths tread craters, can lead to impacts as high as 199,000 pounds. They also note that wheel impacts over 100,000 pounds are ten times more common in winter than in summer. CN found that 87% of their wheel impact readings (for 32,040,106 total wheels examined) were less than 39,000 pounds. The highest impact wheels, with 150,000 pounds impact and above, made up only 0.0007% of the total. However, as noted by the authors, 0.0007% still represents a significant number of wheels. Since the CN impact detectors evaluated 32,040,106 wheels, 0.0007% wheels with the highest impact translates to approximately 225 wheels. Further, using data in the paper, it was noted that approximately 0.038% (more than 12,000 wheels) of the total CN wheels had impacts greater than 100,000 pounds. Approximately 3.5% of the wheels (more than 1.1 million wheels) had impacts greater than 50,000 pounds. All of these wheels experienced stresses well above loads associated with the normal static wheel load 35,750 pounds for a 286,000 pounds GRL car and the situation will be exacerbated if GRL is increased.

It is very possible that shattered rim wheel defects, and other defects such as vertical split rims, are initiated by large impacts caused by built-up tread, shelling/spalling, out-of-round wheels and slid flats. Wheels could be reprofiled and returned to service with defects present below the tread surface. If this is indeed the case, the 2,000-wheel increase in AAR CRB out-of-round wheels between 1999 and 2000 becomes even more significant. Further, if allowable loads are increased to 315,000 pounds GRL, the number and severity of high impact loads will surely increase. Although shattered rims remain a very small portion of total wheel removals, they can lead to derailments and therefore remain an important safety concern.

Interestingly, shattered rim-like defects have recently been created on a dynamometer at the technical center of a major wheel manufacturer (Berge, 2000). The defects have been generated at very low mileage levels (15,000 miles) for applied rolling wheel loads above 140,000 pounds. In one test case, 36,000 pound wheel loading was applied for 40,000 miles and was followed by 110,000 pound loading for 6,000 miles. Since an internal flaw was discovered at 6,000 miles using ultrasonic testing, wheel loading was then returned to the 36,000 pound level for 6,700 miles until failure occurred. Although the applied load was not a dynamic one as would be experienced by a wheel with a flat spot in actual railroad service, the results suggest that higher load levels can indeed influence shattered rim initiation.

The safety benefits of ultrasonically testing wheels that are reprofiled are obvious. If a wheel contains a crack that is detected during testing at a wheel shop, that wheel can be removed and will not fail in service. Standard Steel supports such testing and notes that BNSF and UP are performing this kind of work.

THERMAL LOADS FROM BRAKING

In addition to supporting the mechanical loads of the freight car, wheels serve as the system's brake drum. Thermal loads are extremely important to wheel performance and are responsible for generation of stresses in the plate.

As previously mentioned, AAR approves wheel designs on the basis of a FEA computer simulation performed using the S-660 standard. This document specifies the mechanical and thermal loads that should be applied to the wheel for various GRLs. For example, the 100-ton (263K GRL) loading condition requires application of a 35 HP thermal load to the wheel tread for 20 minutes. The 125-ton (315K GRL) loading case uses 41.92 HP for 20 minutes. Clearly the thermal load associated with 315K GRL is much more severe than that used for 263K GRL. The S-660 FEA is conducted for wheels using the new minimum rim thickness and the condemning rim thickness. For 36-inch one-wear freight car wheels these rim thickness values are 1.5 inches and 7/8 inches, respectively.

Thermal failures in North American railway service have become increasingly rare in recent years. This is due to the adoption of curved plate, heat-treated wheel designs by the Association of American Railroads (AAR) in 1989. Previously, with inferior straight plate, untreated wheels in service, thermal failures were much more common. Stone et al. (1999) reported that the number of wheel related derailments in North America has fallen by an order of magnitude since production of rim quenched and curved plate, low stress wheel designs became mandatory. Thermal failures occur if an excessive amount of heat is imparted to the wheel rim and stress reversal takes place. When the beneficial compressive residual hoop stresses from the manufacturing process are made tensile, protection against service initiated fatigue cracks is no longer present. Sudden wheel fractures are caused when a pre-existing wheel rim crack is exposed to a sufficiently high tensile stress.

FINITE ELEMENT ANALYSIS – 4 WHEEL DESIGNS

If a one wear 36" freight car wheel is used at 315,000 pounds GRL, this represents a 20% increase in loading over the loading used to approve the wheel design. To demonstrate the effects of increased GRL on freight car wheels, four different typical one-wear wheel designs, produced by different manufacturers, were analyzed at Standard Steel using ANSYS© FEA software and the AAR S-660 procedure. All wheels are currently used in North American freight service.

Each wheel design was analyzed using 100-ton and 125-ton loading conditions at the condemning rim thickness (7/8-inch). Only the worn rim condition was analyzed since this will give "worst case" results. Thermally (braking) generated stresses are greater for the worn rim condition than for wheels with a new rim condition. Wheels are referred to as A, B, C and D to avoid identification of the manufacturers.

Results of the FEA simulations for wheels A, B, C and D are shown in Tables 5, 6, 7 and 8, respectively. The minimum and maximum radial stresses, along with their wheel location, are shown in columns two and three. Column four shows the maximum Von Mises stress and its location for each wheel. The far right column shows the maximum tread temperature reached during the course of the simulation. For location designations, note that FHF is "front hub fillet area," BRF is "back rim fillet area," and FRF is "front rim fillet area."

Wheel A	Minimum Radial Stress (ksi)	Maximum Radial Stress (ksi)	Maximum Von Mises Stress (ksi)	Maximum Tread Temp. (°F)
100-ton	71 (FHF)	102 (FHF)	88 (FHF)	735
125-ton	86 (FHF)	123 (FHF)	107 (FHF)	861

Table 5. 36-inch wheel FEA results, Wheel A.

Wheel B	Minimum Radial Stress (ksi)	Maximum Radial Stress (ksi)	Maximum Von Mises Stress (ksi)	Maximum Tread Temp. (°F)
100-ton	65 (FHF)	97 (FHF)	83 (FHF)	855
125-ton	78 (FHF)	117 (FHF)	100 (FHF)	1,003

Table 6. 36-inch wheel FEA results, Wheel B.

Wheel C	Minimum Radial Stress (ksi)	Maximum Radial Stress (ksi)	Maximum Von Mises Stress (ksi)	Maximum Tread Temp. (°F)
100-ton	43 (BRF)	75 (BRF)	76 (FRF)	832
125-ton	50 (BRF)	91 (BRF)	92 (FRF)	975

Table 7. 36-inch wheel FEA results, Wheel C.

Wheel D	Minimum Radial Stress (ksi)	Maximum Radial Stress (ksi)	Maximum Von Mises Stress (ksi)	Maximum Tread Temp. (°F)
100-ton	64 (BRF)	87 (BRF)	79 (FRF)	789
125-ton	78 (BRF)	104 (BRF)	95 (FRF)	925

Table 8. 36-inch wheel FEA results, Wheel D.

As the results show, an increase in GRL greatly increases the level of stress in the wheels. Further, with a thin rim as would be found at the condemning limit of 7/8-inch, the temperatures and stresses can be very high. All four of the wheel designs experience a large increase in stresses when loading is increased from 286,000 pounds GRL to 315,000 pounds GRL.

Also, note that the maximum and minimum radial stress levels in each wheel are tensile and quite high at 315K GRL. Radial stresses, applied in a cyclic fashion in the wheel plate and fillets, are important to plate fatigue crack formation and failures. Plate failures are currently a relatively rare occurrence in North American railway service. Wheel tread temperatures are also seen to be elevated, approaching the point where spheroidization of pearlite takes place.

The authors recognize that S-660 finite element analysis results are from a computer simulation and represent a relatively rare heavy drag-braking scenario. However, the results suggest that 315,000 pounds GRL will indeed be more damaging to wheels. Thus, we believe that further increases in service loading of 36-inch wheels to 315,000 pounds GRL should be done with great care and only after additional study by AAR and others.

WHEEL RIM RESIDUAL STRESS ISSUES

Beneficial residual compressive hoop stresses are imparted by rim quenching and help to inhibit the formation of fatigue cracks in service. For residual compressive rim hoop stresses to result from the rim quenching operation, plastic (permanent) deformation must take place. Austenitized wheels are rotated inside a ring fixture that sprays water on the tread and flange areas, thereby creating a harder, pearlitic microstructure and the desirable compressive hoop stress. When the water spray quenches the hot, austenitic wheel rim, the outer rim fibers cool and shrink inwards. However, the steel below the quenched region is still hot (thus larger than at lower temperatures) and has a reduced yield strength at that temperature. The inner fibers of the rim and the plate are upset in compression by the colder, shrinking, outer rim fibers and yielding occurs. Upon subsequent cooling and shrinking of the inner rim fibers and plate, these areas are smaller than they were originally due to the compressive yielding. However, such areas try to fit into a larger space to maintain continuity. This results in the lower part of the rim and the plate being in tension while the outer portion of the rim is in compression and a residual hoop stress gradient is present in the wheel rim.

Of additional concern is the possibility of progressive fatigue crack damage to the wheel tread surface from thermal cracking. Certain passenger cars were found to have severe thermal cracking in service, and this situation led to an extensive investigation and research. The wheel treads were found to have been subjected to severe braking heating that led to localized stress reversal and tensile stresses at the tread surface. Subsequent finite element analysis modeling work by Gordon, Jones and Perlman (1998) showed how service loads (thermal and mechanical) affect the as-manufactured residual compressive hoop stress profile within the wheel rim section. Localized tensile hoop stresses were found at various depths below the tread surface depending on service loading conditions.

The mechanism by which localized stress reversal occurs is as follows: When brake shoes are applied to the rolling wheel tread, the surface is heated due to friction. The steel at the tread surface gets hotter, tries to expand and is constrained by the colder body of the wheel rim and plate. If the tread surface is heated to a high enough temperature by braking, the steel will have a reduced yield strength, and plastic deformation caused by expansion and compressive upsetting of steel in the hot zone is possible. After cooling and shrinking, continuity must be maintained between the locally yielded material at the tread surface and the constraining remainder of the wheel. Therefore, the material at the tread surface is now in tension. If a freight car wheel is subjected to severe braking cycles with intense heating, progressive thermal fatigue cracking could occur, particularly at higher gross rail loads and/or near the condemning rim thickness.

RESIDUAL STRESS AND BRAKING - TESTING

Recent cooperative dynamometer test work involving Standard Steel, Valdunes and a brake shoe manufacturer, to be presented at the 13th International Wheelset Congress, found that increased GRL does have a damaging effect on the wheel (Demilly et al., 2001). A wrought H36 (one wear) wheel was exposed to "real world" braking sequences downloaded from locomotive event recorders. We note that a special cylindrical tread profile was applied to this test wheel, as this tread profile is required for testing of brake shoes.

The magnitude of the braking thermal load for the sequences was increased from that associated with the current 286,000 pound GRL level to that associated with the equivalent of 328,000 pounds GRL. (Note: 328K GRL was used as the dynamometer does not allow for "fractional" load additions).

The H36 wheel was subjected to a series of braking events including stops, short grades and long grades that were included in each of 12 total sequences. The long grade step was believed to be the most damaging braking event in each sequence. Table 9 shows the temperatures and horsepower levels used for the 45 minute "long grade" (step number 4 of 7) of each of the 12 braking sequences.

Wheel rim residual hoop stress measurements were taken using a portable electromagnetic acoustic transducer (EMAT) system owned by Valdunes (Lonsdale, Demilly, Del Fabbro, 2000). Following the braking sequences selected for measurements (see Table 10), eight total EMAT measurements were made on the back rim face at the twelve, three, six and nine o'clock positions (two measurements at each location) of the wheel. The peak residual stress value corresponding to a point near the tread surface and the peak residual stress value corresponding to a point near the ID were recorded. Residual stress results are contained in Table 10. The reported values are averages of the four clock position measurements for the "Near Tread" and "Near ID" positions. No thermal cracks were seen on the wheel tread at any time.

Long Grade #	Avg. Temp. °F	Max. Temp. °F	Avg. Power HP	Max. Power HP
1	612	792	35.6	42.5
2	586	770	35.2	38.8
3	627	801	36.4	41.3
4	677	999	41.6	66.4
5	668	1021	42.9	69.3
6	604	798	36.0	42.6
7	684	935	39.4	56.3
8**	786	1102	49.2	68.1
9	648	900	40.7	55.9
10	640	839	38.8	50.2
11	741	1093	41.4	69.6
12*	564	758	45.0	56.3

*Note: This grade only 25 min., all others 45 min., ** This grade simulated operation without dynamic brake.

Table 9. Long grade (step 4) test results.

After Brake Sequence	Near Tread Stress, MPa	Near ID Stress, MPa
3	-157	-214
4	-143	-222
7	-109	-222
8	-65	-204
10	-64	-196
12	-55	-217

Table 10. Average EMAT rim hoop stress.

The dynamometer testing (equivalent to about 30-40 days in service) found that the level of residual hoop compression in the wheel rim near the tread surface worsened during the test. This was due to long duration grades (45 minutes) and braking inputs over 40 HP. Although the rim hoop stresses did not become tensile, the level of compression was greatly reduced near the tread surface. This suggests that 315,000 pound GRL will be more damaging to 36-inch wheels than is current service.

We note that European railways are currently using nondestructive testing in car repair shops to identify and condemn those wheels with excessive tensile rim hoop stress. Ultrasonic testing systems to measure the level of residual compressive hoop stress in wheel rims are currently commercially available from a number of manufacturers. Such testing would clearly have benefits in North America where tread braking is much more severe than in Europe. Cars with braking problems could have wheels tested to determine if wheel rim stresses are tensile and therefore potentially dangerous. Standard Steel supports this type of testing, which will also allow the railroad industry to determine the magnitude of thermally related wheel problems in service.

ULTRASONIC TESTING ISSUES

As the load carried by the wheel increases, the allowable size of discontinuities in the wheel rim decreases. Currently, the AAR ultrasonic rejection standard for testing the rim of newly manufactured wheels is 50% of the amplitude response from a 1/8-inch diameter flat bottom hole (FBH). If GRLs increase, the rejection standard will surely have to be tightened to prevent an increase in shattered rims, vertical split rims, etc. However, as the rejection standard diminishes in size it 1) becomes more difficult to reliably detect the smaller indications and 2) becomes extremely difficult to produce accurate flat bottom holes in the required orientation.

To insure the best possible ultrasonic inspection for new wheel rims, Standard Steel has installed three state-of-the-art phased array ultrasonic inspection systems at the Burnham, PA, plant. This technology is thoroughly described in a paper to be presented at the 13th International Wheelset Congress. (Lonsdale et al., 2001). Standard Steel was the first user of phased array inspection systems for wheels, although such technology has been used for medical applications and the inspection of pipe welded at high speeds.

The new phased array systems allow for improved volumetric coverage of the rim, improved resolution and improved sensitivity. They also have an improved signal to noise ratio and are more flexible than older types of ultrasonic testing systems. Specialized ultrasonic inspections can be conducted at a specific depth below the wheel tread by using the beam focusing capability of the system, and ultrasonic beams can also be steered. Finally, data collection capability is possible for ultrasonic scans. If AAR adopts 315,000 pounds GRL for widespread operations, the need for the best possible ultrasonic inspection becomes even more critical than it is today.

CONCLUDING REMARKS

1. Increased freight car gross rail loads will lead to higher mechanical and thermal stresses on 36-inch diameter wheels and this makes in-service wheel failures more likely. Service related wheel removals will also likely increase.

2. Reprofiling of wheel treads eliminates tread defects and reduces contact stresses caused by non-ideal false flanges, etc. Lower contact stresses mean less tread damage and fewer wheel removals for true shelling in the long run.

3. Wheel impacts impose large dynamic stresses on wheels and such stresses may be important for shattered rim and vertical split rim formation. Higher GRL means larger and more frequent impact loads.

4. Ultrasonic testing of reprofiled wheels, particularly those wheels removed for high impact loads, is a prudent course of action and will help to improve safety.

5. 315,000 pounds GRL will mean that wheels will experience greater mechanical, thermal and impact loads than ever before. The quantitative effect of 315K GRL on wheel performance and life is not clear. More study of the effects of 315K GRL on wheels is needed.

6. Finite element analysis computer simulations show that 315K GRL loading results in higher wheel stresses and temperatures than 286K GL loading. All wheel types now in freight service will have such higher stresses and temperatures.

7. Drag braking tests of an H36 wheel using a braking load associated with greater than currently used GRL negatively affected the wheel rim residual compressive hoop stress profile within a short service time. Residual stress testing of wheel rims is being done in Europe and can help improve safety.

8. Wheel manufacturers have recognized the increasing severity of the North American freight railroad environment and have modernized production and testing

facilities to meet the challenges. Wheel redesign efforts have also taken place.

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