Wheel Rim Residual Stress Measurement Using Ultrasonic Testing

Francois Demilly
Manager-Research & Development
Valdunes

Valerio Del Fabbro
Manager-Quality Assurance
Valdunes

Cameron Lonsdale
Technical Manager-Railway Products
Standard Steel

Emmanuel Vallart
Engineer-Inspection
Valdunes

Tom McCabe
Manager Engineering-North America
Railroad Friction Products Corp.

Blake Caldwell
Manager of Dynamometer Tests
Railroad Friction Products Corp.

SUMMARY

The use of ultrasonic methods to measure the residual hoop stress in the rim of railway wheels is well known. This paper provides a description of how the residual compressive hoop stress is formed in the rim and reviews relevant railway service environment issues. Conventional destructive testing and ultrasonic methods for evaluation of the residual stress state of the wheel rim are compared. The relationship between the "K coefficient" used in ultrasonic testing and the manufacturing process is reviewed. This paper also describes the testing of new wrought wheels during the manufacturing process and discusses stress measurements for wheels exposed to severe drag braking conditions in freight service.

The paper also describes wheel rim residual stress measurement results taken during a series of dynamometer drag and stop braking events. Dynamometer brake applications were determined using actual North American freight train service data to simulate the thermal loadings experienced by an H36 wheel over time in service. Thermal brake loading sequences appropriate for a hypothetical 315,000 lb (143 t) gross rail load (GRL) coal car were used during testing.

The influence of heavy thermal brake loading on wheel rim residual hoop stress is discussed. The effect of heavy car brake loading on brake shoe wear is also reviewed. Finally, the paper briefly discusses how residual stress measurements can improve safety and reduce costs for railroads.

INTRODUCTION – RIM QUENCHING AND SERVICE ISSUES

The process of wheel rim quenching provides beneficial residual compressive hoop stresses in the rim that inhibit the formation of service related fatigue cracks. 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, fully 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.

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.[1] 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[2].

Although the number of thermally related wheel failures has greatly diminished, such failures of curved plate, rim quenched wheels remain a possibility. Stone, Pellini and Harris noted that low stress wheels fail due to fatigue cracks that originate at the point of the flange[3]. Stone and Carpenter[4] later found that low stress, heat-treated wheels can resist at least 60 HP (45kW) of thermal input from a brake shoe without damage. A stuck brake, a handbrake left on, or excessive drag braking could cause a stress reversal condition in the rim of a curved plate, heat treated wheel.

Finite element analysis (FEA) allows for evaluating the effects of mechanical and thermal loads on the subsequent stresses in various wheel locations. Thermal loads from tread braking are found to be much more severe for a worn wheel rim than for a wheel with a new rim. This is due to the fact that a wheel rim with a smaller volume of material reaches higher temperatures during braking and the plate subsequently flexes more.

INCREASED WHEEL THERMAL LOADS

With the allowable maximum freight car gross rail load (GRL) having increased from 263,000 lb (119 t) to 286,000 lb (130 t) in recent years, and with 315,000 lb (143 t) now being discussed, the possibility of additional wheel thermal failures exists. The additional thermal brake shoe energy necessary to stop a heavier freight car will be transferred to the system's brake drum, which is the wheel. There has been some recent discussion in North America regarding use of the H36 design freight car wheel in 315,000 lb (143 t) gross rail load service instead of using the more expensive 38-inch (965-mm) diameter wheel and associated larger axle. If an H36 wheel, currently designed and analyzed for 100 ton loading (263,000 lb or 119 t GRL) using the AAR S-660 finite element analysis method, is exposed to 125 ton (315,000 lb or 143 t GRL) loading, this results in a 20% increase in thermal loading. Such thermal loading is potentially detrimental to wheel performance.

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[5]. 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[6] 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 MEASUREMENTS

Early work to measure the residual stress in wheel rims was destructive and centered upon the use of saw cut testing. Such testing can be performed using various strain gauges on the wheel surface to measure relaxation, or simply as a way to note the change in saw cut gap with saw cut travel distance. The latter method, used by North American railroads, is qualitative and does not produce an absolute stress magnitude.

Hole drilling, where a hole is drilled into the wheel surface to a depth of approximately 0.080 inches (2 mm), uses strain gauges to detect relaxation around the hole. The hole drilling technique provides a magnitude for stresses at the wheel surface, but not through the wheel rim cross section. Other stress determination methods used include a totally destructive sectioning technique perfected in Europe and a torch cut/relaxation method utilized by Valdunes.

Non-destructive methods to measure stress are clearly desirable and many efforts to apply measurement techniques have taken place over the years. Numerous investigators have studied the application of ultrasonic residual stress measurements to wheel rims, and the literature is replete with important contributions, some of which are referenced here[7,8,9,10,11].

THEORY OF BIREFRINGENCE

Use of ultrasonic techniques to measure wheel rim residual stress is based upon the concept of birefringence. The speed of polarized sound waves, or alternatively the time of wave flight, through the rim section allows for calculation of residual hoop stress. Several systems are now commercially available including those produced as the result of German, French, Polish and North American work.

Valdunes owns and uses two ultrasonic residual stress measurement devices produced by a French manufacturer, Metalscan. One of the devices uses piezo-electric transducers (PET) while the other uses electromagnetic acoustic transducers (EMAT). However, both systems operate using the principle of birefringence. For the residual stress measurement systems used by Valdunes (PET and EMAT), the appropriate equations are[11]:

$B\sigma = K(\sigma_2 – \sigma_3)$ and $Bm = Bo + B\sigma$, where

$B\sigma$ = the birefringence due to stresses
K = a proportionality factor in MPa/(m/s)
$(\sigma_2 – \sigma_3)$ = stresses within the two planes of maximum polarization.
Bm = the measured birefringence
Bo = birefringence due to elastic anisotropy

Since anisotropy means that a material has directional properties, there can be variation in properties within a material due to forging, heat treating, etc. Valdunes found the effect of structural anisotropy to be very slight for their wrought wheel rims and they use a proportionality factor K of 1 m/s = 43 MPa for Class C steel[11]. Normally the K factor must be determined for each different grade of steel and the manufacturing process being used. For example, the K coefficient for cast wheels is expected to be significantly different than for wrought wheels due to material texture and grain structure affecting ultrasonic wave speeds through the rim.

RIM HOOP STRESS GRADIENT

As there is a gradient of residual hoop stresses within the rim that changes with depth below the tread surface, any nondestructive measurement made from the front or back rim face produces an average of the bulk residual hoop stresses across the rim width. Also, residual stress values are most compressive near the tread for new wheels since this area receives the most beneficial effect from rim spray quenching. As the distance below the tread increases towards the rim ID, the magnitude of measured average compressive hoop stress decreases.

A radial section plot showing the as-manufactured residual hoop stress condition of the rim after quenching, tempering and cooling (provided by Gordon[12]) is shown as Figure 1.

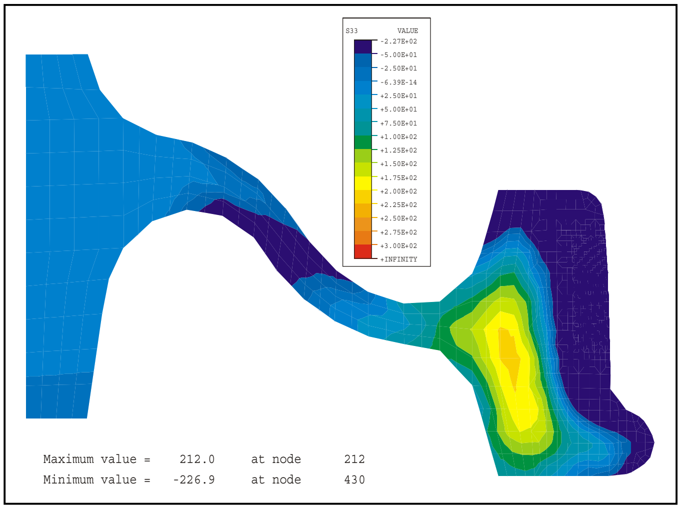

Figure 1. Radial section plot showing the as-manufactured residual hoop stress[12].

ELECTROMAGNETIC ACOUSTIC TRANSDUCER (EMAT) ADVANTAGES

The advantages of EMAT ultrasonic residual stress measurement systems include:

- Easier to use
- No couplant is needed
- No need to manually turn the transducer
- The computer based system provides a stress value rather than a wave speed

RESIDUAL STRESS TESTING OF NEW WHEELS AND RAILWAY SERVICE TESTING

Valdunes uses an EMAT system to measure the residual compressive hoop stress in the rim of newly manufactured wheels if required by customer specification. Ten bulk stress measurements are made at the same clock position on the wheel rim along a radial line on the back rim face. Measurements extend from a location corresponding to near the tread surface (most compressive) to near the rim inside diameter (least compressive) and a stress profile of the rim is then obtained.

European railroads are using rim ultrasonic stress testing for wheels that experience severe tread braking in service. The goal is to find wheels with excessively tensile hoop stresses and remove them from service. The French National Railways (SNCF) condemn wheels in tread braked mountain service at either +350 MPa or +400 MPa depending on the type of wheel[13]. The German Railways (DB) condemn wheels if the rim hoop stress is +300 MPa or greater[14].

The Belgian National Railways (SNCB) and Valdunes participated in a joint field study that periodically measured rim hoop stress for wheels in service on freight cars with severe tread braking. The cars operate over mountainous terrain from Belgium to Italy. Rim hoop stresses for a conventional European wheel design (ORE) and a new wheel design (Mountain Safety Wheel) were compared. The Mountain Safety Wheel was found to have more favorable rim hoop stresses than the ORE wheel for the same service and mileage.

DYNAMOMETER TESTING AT RAILROAD FRICTION PRODUCTS CORPORATION

Event recorder data from seven coal trains operating on three major railroads in Western North America were analyzed for all brake applications – speed when brake applied, speed when brake released, brake pipe reduction, and duration of application. The brake applications were grouped into light slowdowns or stops (minimum to 7 psi (0.5 bar) reduction), medium slowdowns and stops (8 psi to 12 psi (0.6 bar to 0.8 bar) reduction), heavy slowdowns and stops (13 psi (0.9 bar) or greater reduction), emergency stops, and grades. The number of each type of application was prorated to correspond to about 1,200 miles (1,930 km) or three to four days of operation. See Appendix 1 (Event Recorder Summary).

These data include one 16 mile (26 km) grade, with sections up to 2.4%, west of the Rocky Mountains. This grade is run at an average speed of 20 mph (32 km/hr), typically with a 12 psi (0.8 bar) brake pipe reduction, which is enough to balance the lighter portions of the grade. Dynamic brake is added to control speed on the steeper portions.

For this test, only the higher energy braking events (medium and heavy stops from higher speeds and all grades) were used. These applications were further consolidated into a dynamometer test procedure, shown in Appendix 2 (EMAT Testing Dyno Procedure).

Brake shoe forces for the stop tests were determined by converting the brake pipe reduction into shoe force; based on a 315,000 lb (143 t) gross rail load car with a design braking ratio of 13%, which is the maximum allowed under current AAR rules. Braking ratio is defined as the ratio of brake shoe force to loaded car weight, at a nominal 65 psi (4.5 bar) brake cylinder pressure resulting from a 30 psi (2 bar) brake pipe reduction from 90 psi (6 bar).

The target Equivalent Wheel Load (EWL) was 39,375 lb (17.9 t), that is 315,000 lb (143 t) divided by 8 wheels. The actual test EWL was 41,103 lb (18.6 t), 4% higher, as this was the closest EWL obtainable on the dynamometer.

The grade tests were based on a train of 110 cars and two locomotives on a 2.2% grade, with each locomotive capable of 75,000 lb (334 kN) dynamic brake retarding force. Brake shoe force was manually controlled to provide the retarding force to balance the grade. With dynamic brake, this was calculated to be 655 lb (2.91 kN). Without dynamic brake, this was calculated to be 825 lb (3.67 kN). To simulate direct release pneumatic brake operation, the shoe force was incrementally increased to compensate for friction fade, but was not decreased, even in the event of an increase in brake shoe friction.

COBRA® V-474 AAR H4 brake shoes were used. The shoes were not machined to match the wheel nor were they worn in before the test. When compared to the current AAR specification for high friction composition brake shoes, M-926-99, the stops in this program are well within the range of the current specification, but the grades are significantly more severe (see Appendix 2).

The highest energy stop in the program is the emergency stop from 60 mph (97 km/hr) at 4,940,000 ft-lb (6.70 MJ), compared to 7,030,000 ft-lb (9.53 MJ) for the 80 mph (129 km/hr) stops in Specification M-926. The shoe forces for the stops are also similar to those in M-926. The emergency stop is at 6,150 lb (27.3 kN) NSF, as compared to 6,020 lb (26.8 kN) NSF for the M-926 Heavy Stop series, which consists of three stops each from 20, 40, 60, and 80 mph (32, 64, 97 and 129 km/hr). The test program was conducted with an Equivalent Wheel Load of 41,103 lb (18.6 t), 25% higher than Specification M-926, which is based on a 100 ton car, and so has an EWL of 32,875 lb (14.9 t)[15].

The 10-minute grades in the program are at 52 HP (39 kW), while the 45-minute grade is at 35 HP (26 kW) with dynamic brake and 44 HP (33 kW) without dynamic brake. The grades in specification M-926 are run with a constant net shoe force, at 20 mph (32 km/hr), for 45 minutes. The Heavy Grade is run at a NSF of 1,450 lb (6.45 kN) and requires the brake shoe to provide a minimum retarding force of 400 lb (1,780 N). This correlates to a minimum shoe friction of 0.28 and a minimum horsepower of 21 HP (16 kW). The AAR approved COBRA® brake shoe passes the M-926 Heavy Grade with some margin to spare and typically generates 29 HP (22 kW) minimum. Therefore, the test program 45-minute grade, even with dynamic brake accounted for, was at 67% higher power than the AAR Heavy Grade minimum, and 21% higher than typical results.

In order to test a new H36 wheel on the dynamometer, the bore was sleeved to a smaller than normal diameter and six holes were drilled through the hub. This allowed for mounting of the wheel on the machine. Additionally, the back hub area was specially machined and a tread profile with no taper was produced. This flat tread taper, as opposed to the normal 1:20 AAR freight car wheel taper, facilitates testing of brake shoes on the dynamometer.

A total of twelve braking sequences were applied to the wheel to determine the effect of repeated heavy braking on the level of residual hoop stress in the wheel rim. As shown in Appendix 2, each sequence was made up of seven individual steps. Residual stress measurements were taken after the third, fourth, seventh, eighth, tenth and final (twelfth) sequences.

RESULTS AND DISCUSSION

Following the braking sequences selected for measurements, 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 1. 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.

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 1. Average EMAT rim hoop stress.

Note that the "Near ID" measurements do not change significantly throughout the twelve braking sequences. However, the "Near Tread" values become much less compressive with additional braking cycles. This suggests that repeated thermal loads from service braking can adversely affect the stress-state of the wheel rim. As indicated in Appendix 2, the longest grade (step four in a sequence) in each of the twelve braking sequences was 45 minutes long at a speed of 20 miles per hour (32 km/hr). Such a long grade would tend to deliver the most energy input to the wheel tread of all the steps. Table 2 shows the average tread temperature, maximum tread temperature, average horsepower input and maximum horsepower input experienced by the wheel for the long grade (step four) during each of the twelve braking sequences.

While the target retarding force, and therefore horsepower, was the same for all the 45-minute grades (except No. 8), the actual horsepower varied since we did not reduce shoe force during the grade, even in the event of an increase in shoe friction. Thirty-five horsepower (26 kW) seemed to be a critical level. If the dynamometer operator was a little too aggressive in increasing shoe force to maintain retarding force as friction started to fade, he would experience a rapid drop in friction, forcing a further increase in shoe force. A subsequent increase in shoe friction, perhaps due to fresh material being exposed to the wheel as the shoe surface delaminated under the high force and temperature, resulted in a higher than desired retarding force and horsepower. In service, this could happen to individual wheels in a train, as a result of the difference in shoe force and friction from wheel to wheel.

Long Grade No.	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 simulates operation without dynamic brake

Table 2. Long grade (step 4) test results.

Initial wheel temperatures were different for each of the long grades due to differences in cooling between steps. All of the long grades except the final one were 45 minutes long. For the final grade, the time was only 25 minutes. The dynamometer operator stopped the test when he felt the brake shoe had failed. As shown in Tables 1 and 2, an increase in the residual stress value near the tread (becoming less compressive or more positive) is generally associated with a large power input to the wheel tread. Note the large increase in residual stress when comparing the value obtained after sequence 4 and the value obtained after sequence 7. The same effect is seen when the value obtained after sequence 7 is compared to the value obtained after sequence 8. For an average braking power input of more than 40 HP (30 kW), an increase in residual stress is seen in Table 1. Also note that the amount of residual compressive stress in the rim worsened throughout the testing period, which was equivalent to approximately 40 days in service for the H36 wheel. Additional testing time was not available. However, it is important to see if further repeated thermal loads would cause stress reversal in the rim near the tread, or deeper in the rim section, particularly for a wheel with rim thickness near the condemning limit.

We realize that the “rail chill” effect, where the relatively colder rail removes some of the braking heat from the wheel tread in actual railway service, makes our dynamometer results somewhat less than completely realistic. Moyer, Carpenter and Rajkumar reported that this “rail chill” effect can reduce the heat input to the tread by 18% and when combined with convection losses in service can reduce the total tread heat input by 35%[16]. However, we believe that it is very possible for a wheel to experience the thermal loads shown in Table 2 under adverse or unusual conditions. Also note that the AAR S-660 FEA standard for wheels specifies use of 35 HP (26 kW) thermal loading for 20 minutes if 263,000 lb (119 t) GRL is used, while for 315,000 lb (143 t) GRL S-660 specifies use of 41.92 HP (31.3 kW) for 20 minutes[17]. The power values used in our testing were similar to these magnitudes. If the allowable freight car gross rail load is increased to 315,000 lb (143 t), large thermal loads will become more likely in service as the amount of braking needed to stop heavier freight cars increases. Use of an H36 wheel under the repeated heavy thermal loads associated with 315,000 lb (143 t) GRL will likely lead to additional wheels with thermal cracking and non-ideal hoop stress conditions in service.

EFFECTS OF HEAVY LOADING ON BRAKE SHOE WEAR

This test program, particularly the 45-minute grade, resulted in very rapid wear, delamination and plastic deformation of the brake shoes.

Shoe B101A was run through three complete test cycles (equivalent to about 3,600 miles or 5,800 km) of service. Shoe B101F was run through one partial cycle, ending with the 45-minute grade with no dynamic brake. Each of these shoes wore about 22 in^3 (360 cm^3), which is just about half of the total usable volume of 46 in^3 (750 cm^3). This compares to typical wear of 0.4 in^3 (6.5 cm^3) on the M-926 grade tests and typical in service life of 80,000 to 120,000 miles (128,700 km to 193,100 km) on cars with a gross rail load of 286,000 lb (130 t).

Under these high horsepower conditions brake shoe friction dropped to about 0.2. In order to maintain the required 655 lb (2.91 kN) of retarding force, net shoe force reached 3,240 lb (14.4 kN) during the last 10 minutes of the 45-minute grade. For a car with an 11% net braking ratio, the midpoint of the AAR specification, this requires a 19 psi (1.3 bar) brake pipe reduction or 75% of available full service braking, when operating at 90 psi (6.2 bar) brake pipe pressure. When simulating the loss of dynamic brake on the locomotives, a net shoe force of 5,100 lb, equivalent to an emergency application, was required to obtain 825 lb (3.67 kN) retarding force to balance the grade.

The current AAR H4 brake shoe can generate up to about 30 brake HP (22 kW) while maintaining a stable friction level of above 0.28 with normal wear. This level of braking is required in current service with 286,000 lb (130 t) cars, when operating down long grades with the assistance of dynamic braking from the locomotives.

Stone, Blaine, and Carpenter[18] found that shoe wear in long grade braking increased rapidly above 30 to 40 brake horsepower (22 to 30 kW) and with time spent above 600°F (315 C). They found the increased wear was due to delamination and plastic deformation at the high shoe forces associated with the high horsepower tests.

While the Stone et al.[18] tests were run at 40 mph, our tests were run at 20 mph. The slower speed resulted in proportionately higher shoe forces. Rapid shoe wear occurred even at 35 HP (26 kW) and accelerated at higher horsepower. Wheel temperature exceeded 600°F (315 C) for long periods in all the 45-minute grades. In actual service, the wheel temperature will be reduced by the rail chill effect, helping the brake shoe. However, the brake horsepower requirement is unaffected, as it is solely a function of retarding force and speed.

The AAR has recognized that the use of heavier cars puts greater demands on the brake shoe. A task force is working on a revised brake shoe specification, which will include more severe grade braking requirements. New brake shoe materials are required to meet these demands.

BENEFITS OF WHEEL RIM RESIDUAL STRESS MEASUREMENTS

Regardless of the type of service experienced by a wheel, measurement of wheel rim residual stresses using ultrasonic techniques is one method to help improve railroad safety and control costs. Clearly, testing can be used to reduce the potential for catastrophic derailments and the associated large wrecking and repair costs. Further, wheel rim ultrasonic stress measurements could be used to help identify equipment with possible air brake system problems. If a car is found to have wheels with substantially more tensile residual hoop stress than other cars in the same class or type of service, this could mean that the brake system is not functioning properly. Alternatively, for cars that are shopped for air brake system defects, wheels with built-up-tread, slid flat wheels and perhaps spalled/shelled wheels, the wheels could be tested to determine if the rim residual hoop stress is still within acceptable limits.

Installation of a residual stress measurement system in wheel truing shops is another application where railway safety and total cost containment would benefit. As previously noted, compressive wheel rim residual hoop stress has the greatest magnitude closest to the tread since the surface has been quenched during manufacture. As the wheel rim wears the magnitude of residual compressive stress becomes less compressive. Since wheels with thin rims have a less desirable response to thermal loading in service, it is advisable to determine if the wheel rim residual hoop stress is acceptable at the time of turning and remounting. Currently the history of a wheel is generally not known at the time of truing. Therefore, it is possible that thermally damaged wheels, perhaps removed from a car with defective brakes, will be turned and will reenter service. Costs can be avoided if such defective wheels are scrapped and not remounted.

CONCLUSIONS

1. Use of ultrasonic methods to measure the residual compressive hoop stress in wheel rims is well established. Wheel manufacturers and railway maintenance operations are using the techniques.

2. EMAT wheel rim residual stress measurement systems have many advantages and can be used to help improve railway safety.

3. The increased braking needs associated with 315,000 lb (143 t) GRL cars make wheel-damaging thermal loads more likely.

4. It appears that repeated, large (over 40 HP, or 30 kW), long duration (45 minutes) thermal loads increase the level of residual hoop stress in an H36 wheel rim near the tread surface and thus reduce the amount of beneficial compression.

5. The higher shoe forces and brake horsepower required with 315,000 lb (143 t) GRL cars will result in much more rapid brake shoe wear than for current loads.

6. The AAR brake shoe specification should be revised to include heavy grade requirements based on 315,000 lb (143 t) GRL cars. New brake shoe materials will be required to meet these requirements.

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Appendix 1: Event Recorder Summary For Coal Trains

Light Stops / Slow Downs
Brake Pipe Reduction < 8 psi (0.6 bar)

No. of Events	Initial Velocity (mph)		Δ Velocity (mph)	
	Nominal	Range	Average	Range
15	10	0-15	8	0-15
3	25	16-40	0	-5-0
14	25	16-40	8	0-15
7	25	16-40	25	16-40
9	50	41-60	2	-5-5
3	50	41-60	21	15-35

Medium Stops / Slow Downs
Brake Pipe Reduction 8 to 12 psi (0.6 to 0.8 bar)

No. of Events	Initial Velocity (mph)		Δ Velocity (mph)	
	Nominal	Range	Average	Range
8	10	0-15	8	0-15
8	25	16-40	8	0-15
8	25	16-40	25	16-40
4	50	41-60	8	0-15
2	50	41-60	25	16-40
2	50	41-60	50	41-60

Heavy Stops / Slow Downs
Brake Pipe Reduction 13 to 20 psi (0.9 to 1.4 bar)

No. of Events	Initial Velocity (mph)		Δ Velocity (mph)	
	Nominal	Range	Average	Range
4	10	0-15	8	0-15
1	25	16-40	25	16-40

Emergency Stops

No. of Events	Initial Velocity (mph)		Δ Velocity (mph)	
	Nominal	Range	Average	Range
1	10	0-15	10	0-15
2	50	41-60	50	41-60

Grades

No. of Events	Brake Pipe Reduction (psi)	Time (min)	Velocity (mph)
2	12	10	30
1	12	45	20

Appendix 2: EMAT Dynamometer Test Procedure

Target EWL: 39,375 lb
Actual EWL: 41,103 lb

Stop Tests					
Step	No.	Initial Speed (mph)	Net Shoe Force (lb)	Equivalent BP Reduction (psi)	Energy (ft-lb)
1	4	50	1,900	10	3,432,342
3	1	25	3,200	16	858,085
5a	1	60	6,150	Emergency	4,942,572
5b	1	40	6,150	Emergency	2,196,699

Grade Tests						
Step	No.	Speed (mph)	Time (min)	Retarding Force (lb)	HP	Comments
2	1	30	10	655	52	
4	1	20	45	655	35	
4x	1	20	45	825	44	Optional, assumes no dynamic brake
6	1	30	10	655	52	

AAR M-926-99 (Selected Tests for Comparison)

EWL: 32,875 lb

Stop Tests						
Description	No.	Initial Speed (mph)	Net Shoe Force (lb)	Equivalent BP Reduction (psi)	Energy (ft-lb)	Comments
Heavy Stop	3	80	6,020		7,027,854	Highest speed

Grade Tests						
Step	No.	Speed (mph)	Time (min)	Retarding Force (lb)	HP	Comments
Heavy Grade	1	20	45	400	21	Specification minimum retarding force
Heavy Grade	1	20	45	539	29	COBRA® typical retarding force