

Techniques for Controlling Rail Corrugation

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ABSTRACT

Corrugation on low rails in curved track is a common and significant problem on North American railways. Research findings on probable causes are described and practical methods of keeping this problem under control are suggested. The role of wheel and rail contact stresses in the plastic deformation of rail surface, a necessary condition for corrugation development, is examined, and the use of high-strength steel and rail profile grinding to control rail corrugation through reducing or preventing rail surface plastic deformation is discussed. Results of recent field trials of rail corrugation control using a rail profile grinding technique are presented.

Current interest in heavy freight and high-speed passenger train operations has brought increasing attention to the century-old problem of rail corrugation--a form of periodic rail surface deformation that causes severe vibration in vehicles and track. Vibration in turn generates noise, an environmental problem that has become a sensitive issue in densely populated areas. In terms of vehicle and track maintenance, vibration can cause severe damage to wheels and rails as well as to ties and ballast.

Railway rail corrugation is generally classified into two groups according to wavelength. Long-wave corrugation has wavelengths on the order of one-half to one tie spacing, and a peak-to-peak amplitude from 0.020 to 0.070 in. or more. In North America it is common to heavy freight train operations such as coal lines. At the trough of the corrugation, where the heavily loaded wheels (30,000 lb or more) repeatedly pound the rail, severe plastic flow and fatigue damage are often observed.

Short-wave corrugation has wavelengths in the range of 2 to 4 in., and peak-to-peak amplitude is usually less than 0.005 in. It is more common in high-speed passenger train operations and in light rail transit systems. This problem (called "roaring" rail) has long been of considerable concern in Europe. With the recent stepped up development of light rail transit systems, this problem is receiving increased attention in North America. Short-wave corrugation does not usually cause serious damage to vehicles and track structures, at least in the short term. However, the noise and vibration, especially in underground transit systems, are particularly troublesome because of the increasing sensitivity of the public.

In North America, long-wave corrugation is a common problem on low rails in curved track and requires periodic grinding. The cost of grinding could run into millions of dollars for a major railroad. Without grinding, however, the consequences would be far worse: speed restrictions or increased rail fatigue failure, or both, due to severe wheel and rail vibration. In some heavy-haul railway operations with high axle loads (up to 33 tons) and high traffic density [up to 50 million gross tons (MGT) a year on single track], corrugation could grow from a few thousandths of an inch to between 0.020 and 0.040 in. in 6 months. At this level there is usually severe wheel- and rail-contact-generated track vibration, and speed restrictions are sometimes necessary (actual imposition depends on, among other things, the sensitivity of a particular track forma-

tion to vibration). To maintain the required track capacity, these slow orders must be removed as soon as possible. If rail grinding is not available, the rails will have to be replaced. Thus, if slow orders are not acceptable and rail grinding is not available, corrugation could drastically reduce the service life of rails. A cost survey by Roney (1) suggests that the cost of rail corrugation to the Canadian railways is about \$30 million (1984 dollars) a year.

The focus of this paper is mainly on the North American long-wave corrugation problem, with identification of practical measures that could help reduce its magnitude.

CORRUGATION THEORY

To find a permanent solution to the problem of rail corrugation, its cause or causes must be identified. To date there have been a number of theories--some based on general observations, others based on in-depth study. Of the latter, there are two categories: contact resonance and stick-slip. The contact resonance theories consider resonance vibration of the wheel and rail system as the primary cause of rail corrugation. Sources of resonance frequency may be found in various possible vibration modes of wheels, axles, rails, and rail and tie systems. The stick-slip theories concentrate mainly on the frictional contact instability between wheels and rails during vehicle movement through curves. These theories, although not yet proven, could be used to explain the corrugation phenomenon in curved track under heavy axle load. These theories are reviewed in the light of the North American rail corrugation problem.

Theories on the stick-slip phenomenon explain that the contact between wheel and rail is not stable but alternates from stick to slip. During the stick cycle rail wear is small, but during the slip cycle significant wear occurs because of relative wheel and rail sliding under very high contact stresses.

King and Kalousek (2) are among a number of researchers who explain the stick-slip theory in the context of wheelset curving behavior. This theory hypothesizes that during curve negotiation, the effective rolling radius is the same on both wheels, even though the wheel on the high rail (outer) side has a longer traveling distance. To accommodate this difference, the wheel on the high rail side must

rotate faster than the one on the low rail side. Because the wheelset is an integral component, this instantaneous rotational speed differential places the axle in torque until the torsional reaction becomes greater than the adhesion forces at the wheel and rail interface. The axle then twists back, resulting in wheel and rail slip. The stick-slip cycles set the axle into torsional vibrations which in turn perpetuate the stick-slip cycles, causing corrugation to increase.

The assumption of equal effective rolling radius on high and low rail sides does not hold, because, for a conical wheel tread in a flanging configuration, the wheel on the high side travels on a larger wheel radius. The stick-slip theory is, however, still valid if a torque of significant magnitude exists on the axle.

Various studies into the curving behavior of three-piece freight trucks have shown that large creep forces exist between wheel and rail interfaces, and that for a given wheelset, especially the leading one, the creep force orientation is such that a large torque is applied to the axle (3-5). Figure 1 shows an example of a force diagram for a leading wheelset as predicted by a steady-state, truck-curving computer model (6). The magnitude of torque on the axle is about 22,000 lb/ft in a 3-degree curve. Apart from the existence of a sufficiently large torque, the occurrence of stick-slip further requires that wheel and rail friction have a falling characteristic at high creepage and that the wheel and rail creepage condition be severe enough to give an operating point in the falling portion.

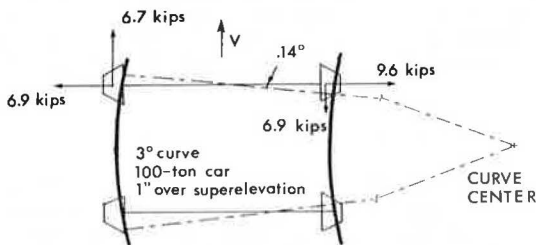


FIGURE 1 An example of force diagram for the leading wheelset in a curve.

Figure 2 shows some sample friction-creepage curves for various wheel and rail contact conditions. It is evident that a falling characteristic due to creep saturation occurs at a few percent creepage for all cases except those with water and oil contamination. However, most experimental results for the lower range of creepage, such as those in Figure 3, show a somewhat different characteristic. These indicate that creep saturation begins at about 1 percent creepage but show no falling characteristic up to 3 percent creepage.

Figure 4 shows the resultant creepage on low rail in a range of curves. These data are derived from computer simulation of the curving behavior of a three-piece freight truck with 35-ton axle load. Generally creepage increases with the degree of curvature and is about 1 percent in 8-degree curves.

These data (from Figures 3 and 4) indicate that stick-slip as a result of friction and creepage falling characteristics is not likely. It may be suspected, however, that in the field other conditions, such as wheel and rail contamination and vibration, may exist but are insufficiently accounted for in the laboratory. These may create the necessary falling characteristics for stick-slip to occur. In a recent paper, Clark (7) reported experi-

FRICTION-CREEP CURVE

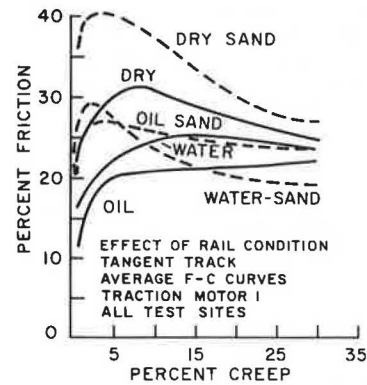


FIGURE 2 Average friction-creep curves.

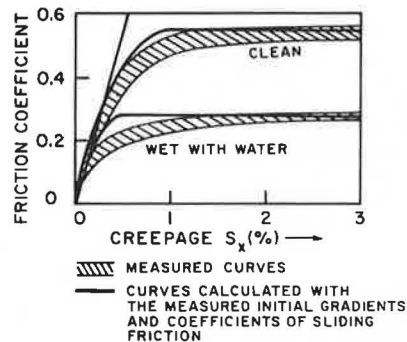


FIGURE 3 Comparison of the longitudinal traction/creep relationship of clean rollers and rollers wet with water.

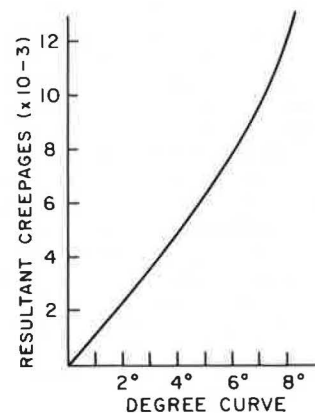


FIGURE 4 Resultant creepage versus degree curve at balance speed.

mental work to verify the stick-slip theory. The results show that for a creepage condition greater than 1 percent, considerable vibration occurred, and in some cases a corrugation-like pattern appeared on the rail surface.

PLASTIC DEFORMATION

Stick-slip vibration may create periodic variation in the tangential force component at wheel and rail

contact. However, for this force to generate periodic plastic deformation as observed in typical North American corrugated rails, the force magnitude must exceed the elastic limit of the rail steel. Thus, it is necessary to know the magnitude and nature of the wheel-loading environment at the wheel and rail interface, as well as the resistance of the rail steel to plastic deformation under actual track operations. A sample calculation that illustrates the magnitude of the wheel-loading environment relative to the strength of the rail steel follows. For a 100-ton car with 36-in. diameter wheels, running at slightly under balance speed (1 in. over-super-elevation) in a 3-degree curve, the wheel loading on the low rail is:

1. Vertical:

Static wheel load	32,000 lb
Vertical load transfer	3,200 lb
Steady-state wheel load	35,200 lb
Dynamic load factor (assumed)	10%
Dynamic wheel load	38,720 lb

 Compressive contact stress:
 - average wheel and new rail (tread is slightly hollow with a concave radius of 30 in.) 195,000 psi
 - new wheel and new rail (tread is 1 in 20) 227,100 psi
 - worn wheel and new rail (tread is hollow with false flange radius of 4 in.) 430,400 psi
2. Tangential (estimation based on steady-state curving simulation of three-piece truck):

Lateral creep force	8,000 lb
Longitudinal creep force	7,000 lb
Resultant creep force	10,600 lb
3. Ratio of tangential to vertical force 0.275

Both standard carbon and low-chrome alloy rails are used in North America. Carbon rails are used mainly in tangent track and shallow curves (e.g., less than 3 degrees). Low chrome rails are used in sharper curves as a measure against the more severe wheel-load environment. Yield strengths of standard carbon and chrome rails are about 75,000 psi and 100,000 psi, respectively. Even after taking into account a constraint factor of 1.7, these yield strengths are much lower than the estimated range of contact stresses (195,000 to 430,000 psi). Therefore, plastic deformation of rail would occur under the first wheel passage. For subsequent wheel passages, the residual stress created under earlier wheels improves the resistance of the rail steel to further plastic deformation. This phenomenon, called shakedown, has been the subject of intensive research in the field of contact mechanics. Johnson (8), in a recent comprehensive review of this subject, proposed an estimate for the elastic and shakedown limits applicable to rail steel in rolling contact as shown in Figure 5. According to this estimation, the shakedown limit for standard carbon and chrome rails, in the absence of tangential force, is 202,000 psi and 270,000 psi, respectively. However, if a tangential force of 0.275 times the vertical is also present, these shakedown limits are reduced to about 150,000 psi and 200,000 psi for standard carbon and chrome rails, respectively.

These estimates indicate that:

1. Standard carbon rails would be plastically deformed under the passage of most wheels, and
2. Chrome rails would be plastically deformed under the passage of new and worn (i.e., with false flange) wheels. For average wheels, the likelihood

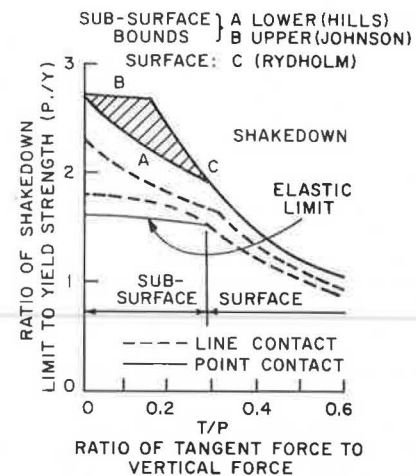


FIGURE 5 Elastic and shakedown limits in rolling contact.

of plastic deformation is less than certain, because the shakedown limit under 0.275 creepage is only slightly below the contact stress.

For a track carrying 50 mgt of 32-ton axle loads per year, and assuming a wheel condition distribution of 15 percent new, 75 percent average, and 15 percent worn, the plastic deformation cycle for standard carbon rail and chrome rail would be about 1,500,000 and 500,000 times a year, respectively.

Thus, curved track is generally too hostile an environment for the standard carbon and low alloy rail steels currently in common use. These rails would likely corrugate in the presence of vibration.

PRACTICAL SOLUTION

The preceding discussion indicates that low rails in curves are subject to such a severe loading environment that unless hardened rails are used, there is potential for widespread corrugation. High wheel loads (due to heavy cars) and creepage (due to poor curving of three-piece trucks) place the present rail steel on the threshold of plastic deformation. With wheel and rail contact being steel on steel (low damping), and with limited adhesion (stick-slip), there is potential for vibration. Uniformly spaced tie support could add to the vibration problem as well. Knowing that these two factors, that is, plastic deformation and contact vibration, are essential in the development of corrugation, the solution obviously is to eliminate or reduce their severity. To achieve this goal when there is no option of replacing existing trucks with ones of steerable design, or replacing existing track with a continuously supported structure, or reducing axle load, the author believes the following remedial actions could be taken:

1. Profile grinding. At present, rail is ground periodically to remove corrugation. This grinding operation can be adapted to produce an optimal rail profile as well, without additional cost. The objective is to create a rail profile that conforms to existing wheel profiles, given the existing range of track gauge error. On the low rail, the primary aim is to ensure that wheel and rail contact takes place away from the false flange should the wheel be worn and the track gauge widen. If this is achieved, a potential contact stress of 430,000 psi under worn wheels can be reduced to 195,000 psi, making the low

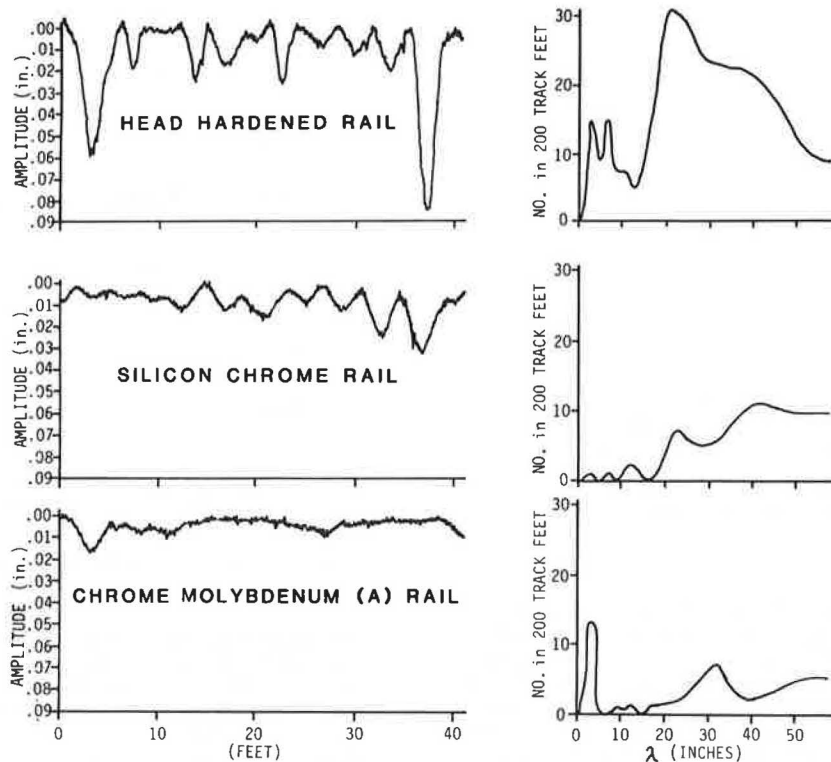


FIGURE 6 Typical initial rail profiles.

rail less susceptible to plastic deformation. This technique will be described in the case study.

2. Grinding of new rail. The surface of new rail normally is not smooth. Apart from mill scales, there are irregularities with wavelengths from a few inches to several feet. An example is shown in Figure 6. This roughness can excite wheel and rail vibration at a time when the rail steel has not yet hardened to its potential. Early removal of surface roughness would reduce the initial plastic deformation and produce a smoother hardened rail surface at a later stage.

3. Superelevation. Track should be elevated at balance or at a slightly deficient level to avoid wheel load transfer to low rails. In a mixed-traffic situation, superelevation should be designed to accommodate the heavier traffic categories.

4. Heat-treated rail. Heat-treated rails such as the head-hardened type could be used to improve rail resistance to plastic deformation. With a yield strength (2 percent proof stress) of about 125,000 psi, they would be able to resist plastic deformation under most wheel loads except the false flange type. When used with a profile grinding program, which is effective in avoiding false flange contact, heat-treated rails would provide satisfactory resistance against plastic deformation, and hence rail corrugation.

5. Rail lubrication. The standard practice of lubricating the gauge face of the high rail effectively transfers some of the creepage from low rail to high rail. Figure 7 is a three-piece truck curving simulation result showing the reduction of longitudinal creepage and creep force on the low rail as the friction factor on the high rail gauge face decreases. This has the dual effects of reducing the likelihood of wheel slip and increasing the shake-down limit of the low rail. Daniels and Blume (9) reported a correlation study at FAST in which rail corrugation growth was much lower during the periods when rail lubrication was used than during the

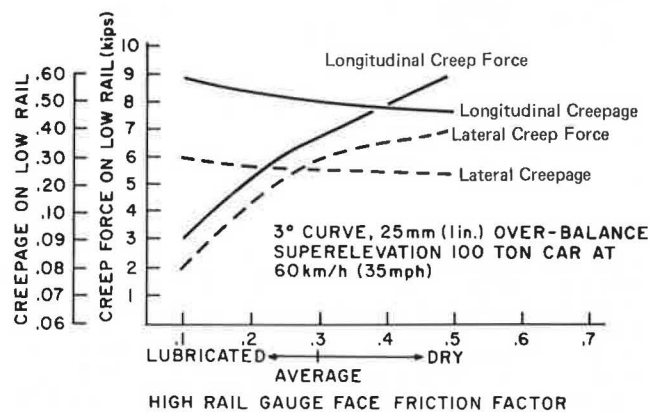


FIGURE 7 Effect of rail lubrication on creepages and creep forces.

periods without lubrication (Figure 8). Because trackside lubricators are already being used on most railways, a field trial to validate this effect could be carried out quite readily.

There are also other measures that may be less readily achievable but are nonetheless feasible:

1. Wheel remachining. More frequent wheel inspection and remachining would reduce the percentage of defective wheels such as those with false flange or flat spots. If this action is coordinated with a program of profile grinding and the use of heat-treated rail, the problem of rail corrugation would be greatly reduced. For several years the Hamersley Iron and Mount Newman Mining Railways in Australia (33-ton axle loads) have been experiencing minimum rail corrugation problems using the preceding strategy.

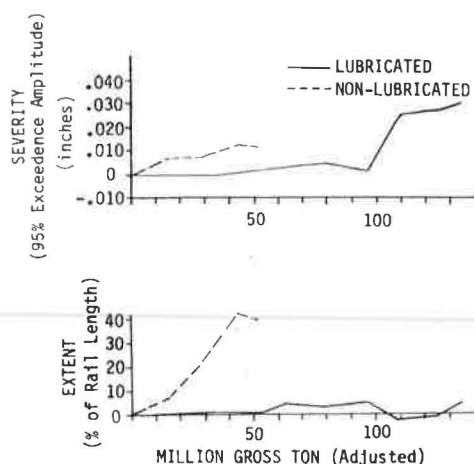


FIGURE 8 Rail corrugation growth by lubrication condition; standard carbon rail, 5 degree curve, hardwood ties.

2. Track gauge control. The biggest problem with timber track, as far as corrugation is concerned, is gauge widening, which causes false flange contact on low rails. The use of concrete ties solves this problem but introduces high dynamic load because of larger tie mass and stiff rail pad. For timber track, a practical solution could be to insert one steel tie, for example, every five tie spacings. The spacing, however, should not be uniform (i.e., not always one every five ties) as this may introduce a periodic stiffness variation. Gauge rod is another, perhaps less practical, option.

CASE STUDY: PROFILE GRINDING TEST ON CP RAIL - 1982 (10)

Outline

In May and June 1982, a rail profile grinding test was conducted on a section of curved track on the Canadian Pacific (CP) main line through the Rocky Mountains. The first objective was to reduce the problem of corrugation on low rails in curves. For this, two curves were selected: an 8-degree curve with severe corrugation (average depth 0.0595 in.) (1.44 mm) on the low rail (chrome), and a 2-degree curve with typical corrugation (average depth 0.0335 in.) (0.84 mm) on the low rail (standard carbon). Measurements of rail vibration under a 30-mph test train (three locomotives, five 100-ton cars) are shown in Figure 9. These indicate the severity of the problem, especially in the 8-degree curve.

Investigation

Track gauges and wheel profiles were surveyed in order to design a suitable rail profile to redress the problem of corrugation on low rails in curves. This survey revealed that in curves where rail corrugation occurs, the static track gauge is typically 0.25 in. wider than usual. Under traffic, there will be up to 0.25 in. of additional gauge widening due to lateral wheel forces (typical magnitude of 4 to 12 kips outward on both high and low rails). The survey of wheel profiles showed that there was a large percentage of wheels with about 0.25 in. flange wear, and a small but significant percentage of wheels having false flange (i.e., reversed tread radius on the field side). The combined effect of

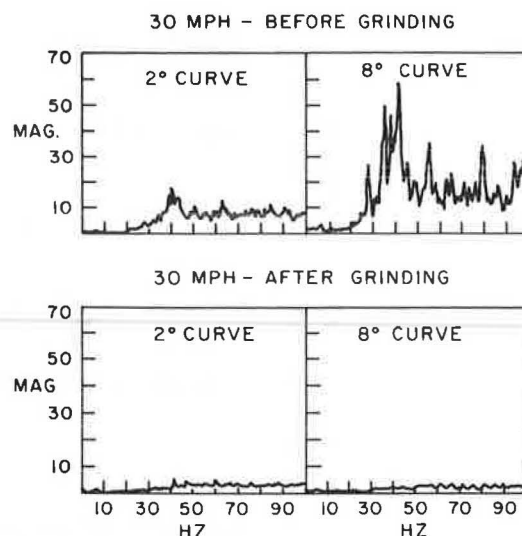


FIGURE 9 Measurements of rail vibration.

gauge widening and wheel wear creates an effective total gauge widening under wheel passage of about 0.50 to 0.75 in. (0.25 in. flange wear + 0.25 in. gauge face wear + up to 0.25 in. dynamic gauge widening). This led to a kind of false flange contact problem, as shown in Figure 10.

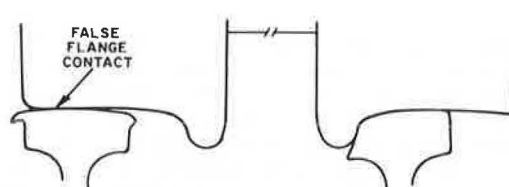


FIGURE 10 Wheel and rail contact geometry in curves with wide gauge.

Design of the Ground Rail Profile

The preceding investigation led to the conclusion that false flange contact is a primary contributing factor to the problem of corrugation on low rails. To avoid false flange contact (without resorting to the remachining of worn wheels and the prevention of track gauge widening), the field side of the low rail must be ground down so that wheel and rail contact takes place only on the gauge side. Figure 11 shows diagrammatically what must be done. In reality, grinding of the field side had to be reduced because of the high cost of grinding and short track time. A cost-effective grinding program is one that

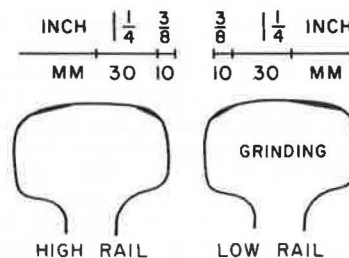


FIGURE 11 Proposed ground rail profile for curves on CP Rail.

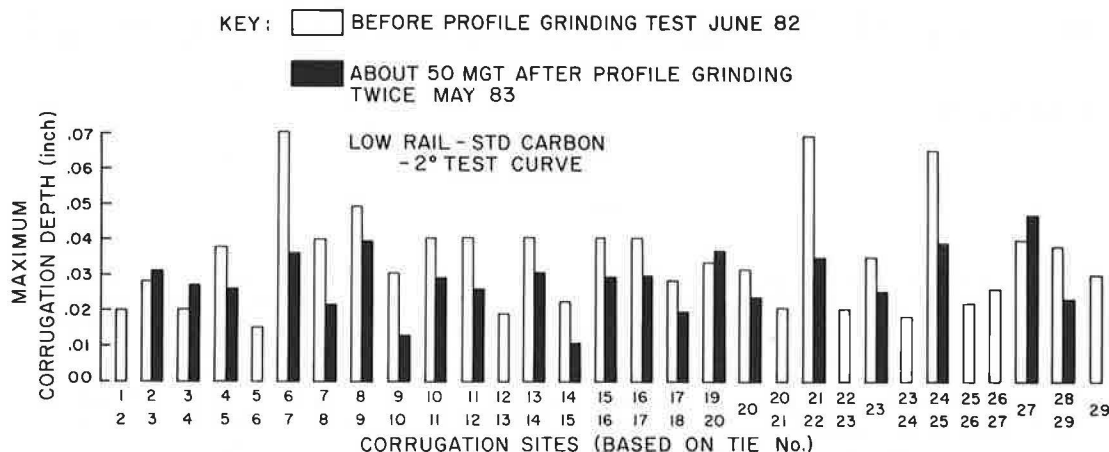


FIGURE 12 Effect of profile grinding on the formation of rail corrugation.

removes only sufficient rail metal to avoid wheel and rail contact in the false flange area between grinding sessions. At the time of the next session, this grinding can be repeated, hence excessive grinding is not necessary each time.

Test Results

Corrugation regrowth was measured periodically after the test grinding. Figure 12 shows the corrugation levels in the 2-degree test curve 1 year after the test grinding. The rail had been subjected to a total of two profile grinding sessions. The open bars represent corrugation levels existing about 6 months (about 25 mgt) after conventional (symmetrical) grinding. The full bars represent corrugation levels 6 months after profile grinding. Except at 4 locations where minor increases occurred, the remaining 26 locations showed a significant reduction in rail corrugation levels.

Further Work

Following the previously described test, CP Rail adopted the designed profile rail as a provisional standard for the main line. A more comprehensive field test involving 12 curves is being carried out. The objective of this test is to further examine the performance of the designed rail profile and to explore its refinement.

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