TECHNOLOGY DIGEST

TD-16-054
©NOVEMBER 2016

TIMELY TECHNOLOGY TRANSFER

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Evaluation of Wheel-Rail Contact Stresses at Crossing Diamonds

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Summary

A team of engineers from Transportation Technology Center, Inc. analyzed the potential wheel-rail contact conditions at crossing diamonds. Under a project sponsored by the Association of American Railroads, a relative comparison of the wheel-rail contact conditions in tangent track, high angle tread bearing crossing diamonds, and high angle flange bearing crossing diamonds was conducted. The results show that contact stresses on either type of crossing diamond are more severe than in tangent track. In addition, the contact stresses for both cases are well above the yield strengths of wheels and rails. Thus, the elastic analysis can only be used for making relative comparisons of the various cases analyzed.

The more severe conditions require consideration for using premium materials and designs for crossing diamonds in order to provide safe and economical operations. Longer-term, it is expected that wheel-rail contact stresses will be similar for flange bearing and tread bearing crossing diamonds. However, wheel-rail contact stresses for flange bearing may be higher than those for tread bearing until the wheel flanges are worn to a more conformal shape. This is largely an implementation issue that will be eliminated when enough flange bearing frogs are installed. The use of an initial groove in the flange bearing running surface should reduce the initial running surface wear, while the wheels become more conformal.

To improve initial performance of flange bearing frog crossing diamonds, an initial concave groove should be considered for installation on all flange bearing running surfaces. This initial groove will help keep vehicles centered on the track and flanges centered on running surface components.

This *Technology Digest* is one of a series on research, development, and evaluation of potential improvements to flange bearing frog crossing diamonds. The series will include frog steels, foundation designs, and flange bearing ramp designs. This project is being conducted by TTCI under the AAR Strategic Research Initiatives Program.

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INTRODUCTION

TTCI continues to evaluate conditions of flange bearing frog crossing diamonds and turnout frogs used in heavy axle load freight operations. The improved dynamic performance of these designs is significant in reducing required track surface and frog bolt maintenance. Reductions in slow orders related to crossing diamond conditions have also been significant in locations where full flange bearing frog crossing diamonds have been implemented.

The industry is still in the early stages of learning how to maximize the efficiency of flange bearing ramps, frog, and crossing diamond system designs. This is evident in the high initial wear rates of flange bearing ramps and frogs. Also evident is the uneven wear and dynamic loading seen on some crossing diamonds.

This *Technology Digest*, one of a series on the current flange bearing research and development work, describes some of these issues and the design improvements developed to address them.

FBF DIAMOND PERFORMANCE ISSUES

While flange bearing frog crossing diamonds perform well dynamically as compared to conventional tread bearing frog crossing diamonds, there are still some areas for improvement. These can include relatively rapid changes in wheel-rail contact conditions, running surface grade, track stiffness, and track alignment. Mitigating the potential deleterious effects of these necessary transitions is the goal of this project.

The first flange bearing frog crossing diamonds were installed with non-conformal level (in cross section) running surfaces. This was done to more easily accommodate the range of wheel back-to-back spacing and wheel flange worn shapes found in revenue service. The first few crossing diamonds would serve to guide designers by observation of the worn shapes of the running surfaces.

While this approach made sense from a running surface design perspective, the lack of wheelset guidance contributed to higher dynamic forces and wear rates on some components. When flange bearing, the vehicle loses its ability to center itself laterally in track. Centering is usually accomplished by the tapered wheel tread profiles running on conventional rails. In this case, the rolling radius difference resulting from a lateral excursion of the wheelset will guide the wheelset back towards track center. There is no such self-centering mechanism for flange bearing.

Figure 1 shows an example of non-centered running on flange bearing ramp at FAST. The wheelsets are moved towards the left rail (not shown in the photograph). This is most likely due to the nearby left hand curve that the vehicles have just traversed in coming towards the photograph. Note the wheel flange marks and incipient wear near the guardrail on the right rail side. Further into the ramps, there is evidence of corresponding flange marks near the left running rail. As the guardrail does its work, the wheelset is moved to the right. However, the wheelset still may not be centered within the 1 7/8-inch flangeway.

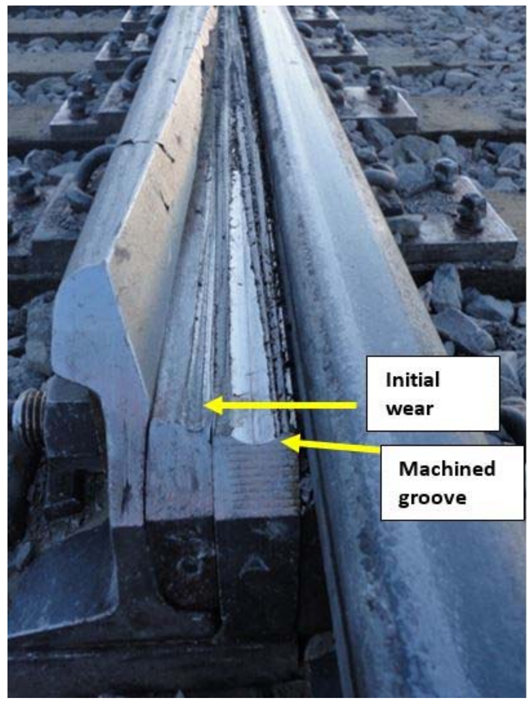

Figure 1. Non-centered Running on Flange Bearing Ramp (note wheel flange path)

It has been noted from profile measurements that wheelsets are not centered at some of the initial flange bearing frog crossing diamond installations. This may be due to adjacent track features, such as curves, small cross-level deviations or simply that the first tall flange wheels wear a groove that most other wheels follow. An uneven centering of a wheel flange on a nominal 2-inch wide bar used for flange bearing may create high sidewall stresses, which could lead to shear failure of the running surface. Figure 2 shows a vertical crack in the running surface of a flange bearing bar in another crossing diamond. This location also showed evidence of the running surface spreading laterally after the bar cracked. The guard in the photograph shows signs of wheel back contact, as well.

Figure 2. Cracked Flange Bearing Bar Associated with Non-centered Running

WHEEL-RAIL CONTACT STRESSES

Wheel-rail contact stress analysis was conducted using elastic materials assumptions to provide a gross approximation of likely wheel-rail or wheel-frog contact stresses for four cases. Four wheel-rail contact conditions, shown below, were assessed based on the Hertzian elastic contact theory. Results from this analysis can be used to assess the relative effects of various changes in wheel-rail contact conditions.

- Conventional Track Base Case: Nominal worn wheel on tangent rail
- Tread Bearing Frog Worn Case: Same case as above with a higher dynamic load—as happens at a flangeway gap. A dynamic factor of 2.0 was used in the analysis
- Flange Bearing Frog New Condition Case: Wheel flange on flat flangeway surface
- Flange Bearing Frog Worn Case: Wheel flange on flangeway with concave surface with a radius of 4 inches

The wheel profiles used in the analysis were measured from five revenue service cars that routinely run over flange bearing frog crossing diamonds (20 wheelsets, 40 wheel profiles).[1] The rail profiles used are the new 136RE profile and a pair of measured worn profiles measured on tangent track.

The wheel profiles from these five test cars showed a range of variety in shapes, as shown in Figures 3 and 4. Depending on the wheel and rail profile combinations, the contact locations vary from rail gage shoulder to slightly toward the field side even on tangent track.

The measured wheel profiles can generally be divided into two groups based on the shapes of wheel flange tip. The first group of profiles (27 of 40) had flange shapes similar to the AAR-1B wheel (Figure 3). The second group of profiles (13 of 40) had the flange backs less tapered than the AAR-1B and had wider, flatter flange tips (Figure 4). These shapes of wheel flange tip tended to widen the band of contact and to shift it towards the center of the track. Also, note that the front side of the wheel flange of both groups stayed relatively constant in shape due to frequent contact with the gage faces of rails in curves.

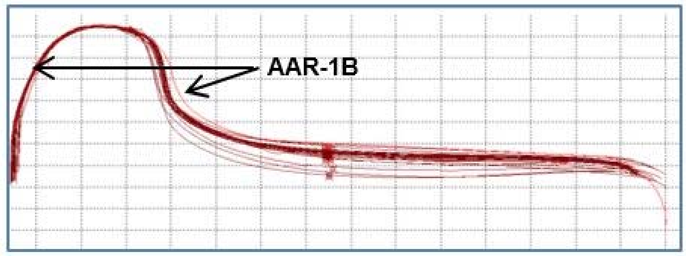

Figure 3. Wheel Flange Tips have Similar Shape as AAR-1B Wheel (Red)

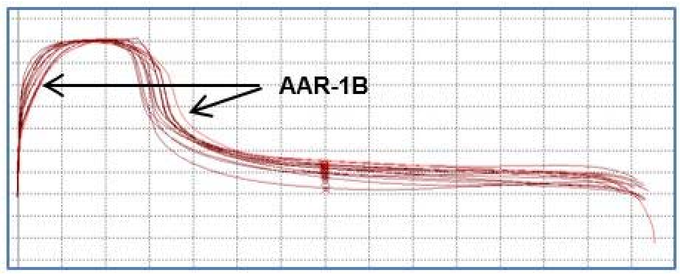

Figure 4. Wheel Flange Back Less Tapered than AAR-1B Wheel (Red)

The representative worn wheel profiles were matched to the four track cases for comparison of calculated contact stresses. The wheel-rail profile analysis software, WRTOL®,[2] was used to perform the calculations for the conventional wheel tread on rail contact stresses. A specially modified version of the software was used to perform the wheel flange in flange bearing frog contact stress calculations.

Table 1 lists the contact conditions modelled for flange bearing and the results. For the open track case and the tread bearing frog crossing diamond case, worn wheel tread and rail profiles were used in the simulation. For the flange bearing frog crossing diamond case, five combinations of flange tip shape and frog running surface shape were modelled. These ranged from the design flange tip profile (0.63-inch radius), to a worn flange tip (1.25-inch radius), to a rather flat flange tip (5-inch radius), which were matched to either a flat or 4-inch radius frog surface.

Table 1. Calculated Contact Stresses for Flange Bearing

	Wheel Flange Tip R (inch)	Flange Way R (inch)	Average Contact Stress (ksi)	Maximum Contact Stress (ksi)
Worn Flange Tip	1.25	-4.0	298.9	448.3
Worn Flange Tip	1.25	Flat	303.5	455.3
Flat Flange Tip	5.00	Flat	194.8	292.3
New Flange Tip	0.63	-4.0	406.2	609.4
New Flange Tip	0.63	Flat	455.7	683.5

Table 2. Summary of Likely Frog Contact Stresses

Case	Condition	Average Contact Stress (ksi)	Maximum Contact Stress (ksi)	Dynamic Load Factor[3]	Dynamic Contact Stress (ksi)	Comparison Ratio
Open Track	Mainline Class 4	211	317	1.3	411	1.0
Tread Bearing Frog	Worn Wheel & Frog	211	317	2.0	633	1.5
Flange Bearing Frog	New Flange	406	609	1.4	853	2.1
Flange Bearing Frog	Worn Flange	299	448	1.4	628	1.5

A vertical wheel load of 36,000 pounds was applied to represent the load of 110-ton (286,000 pounds) cars. The track gage used in the analysis was 56.5 inches with wheel back-to-back spacing of 53.05 inches.

As expected, the radius of the wheel flange tip had a larger effect on the elastic contact stress. The worn wheel flange tip with a 1.25-inch radius produced lower contact stress. The effect of adding a concave groove of 4 inches to the frog running surface on the contact stress is considered significant for new wheel flange tips, but almost nil for more worn flange tips. A pre-machined concave groove controlled the flange bearing contact path. Track engineers in the field observed that the initial running surface groove helped keep vehicles centered on the track and wheel flanges centered on the flange bearing running surfaces, which reduced lateral guard wear and vehicle dynamic lateral movement.

Making the frog running surface closely conformal to the wheel flange would also provide a further stress reduction. However, it is not recommended due to allowable variations in wheel back-to-back spacing and flange tip shapes. The wider 4-inch radius grooves would also help in funneling wheels to the lateral center of the running surface at guardrail flares.

The likely maximum contact stresses were also assessed by considering the dynamic loading factors. The typical dynamic load factors were applied to each of the open track, tread bearing frog, and flange bearing frog crossing diamond cases. These dynamic load factors were measured during testing done at FAST under the same trains. Table 2 shows a summary of the results. It provides a quick reference for likely (i.e., rough order of magnitude) contact stresses of frogs in revenue service and may serve as a guide for selection of candidate frog materials.

CONCLUSIONS

Test results showed the wheel-rail contact conditions at frogs are more severe than in open track. These more severe conditions require the consideration for using premium materials and designs for crossing diamonds. Wheel-rail contact stresses for flange bearing frogs may be higher than those for tread bearing frogs until the wheel flanges are worn to a more conformal shape. This is largely an implementation issue that will likely be eliminated when enough flange bearing frogs have been installed. The use of an initial groove in the flange bearing running surface should reduce the initial running surface wear while the wheels become more conformal.

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