TECHNOLOGY DIGEST

TD-17-038

©DECEMBER 2017

TIMELY TECHNOLOGY TRANSFER

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Improved Ramp Designs for Flange Bearing Frog Crossing Diamonds

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Summary

In an effort to extend the service life of flange bearing frog (FBF) ramps, Transportation Technology Center, Inc. (TTCI) developed three ramp designs that were evaluated using numerical modeling and performance measurements of dynamic wheel loads and wear rates under heavy axle loads (HAL) at the Facility for Accelerated Service Testing (FAST). The three ramp shapes consist of:

- A two-slope ramp that reduces the slope at the most likely flange bearing locations (1:240) and replicates a longer 21-foot ramp,
- A linear ramp (1:180) replicating a 16-foot ramp, and
- A vertical curve ramp (1:180) that smooths out the slope change at the top of the ramp in attempt to reduce the negative effects from the slope change.

The numerical modeling and wear results suggest the HAL train flanges come into contact at a consistent location on the ramp, producing a minor load increase (~2 kips) and localized wear at that location. The increased wear at flange contact was more spread out for the two-slope ramp, suggesting a lower slope ramp will reduce localized wear. However, this localized wear may not be an issue in revenue service due to the wider range in flange heights.

A second observation is the wheel/rail force response was similar at the slope change at the top of the ramp for the linear and vertical curve ramps. Decreases in load were observed at the slope change for both ramps, suggesting the top of the linear ramp was rounded out by wear, and the wear data agrees with this. FBFs in revenue service will have a stiffness change at the frog, but this was not replicated in the FAST test.

The largest vertical wheel force occurred at the ramp-to-level-running-surface interface due to the discontinuity at that location. As with any track discontinuity, greater elevation differences result in greater impact loads, so ensuring the ramp and level running surface are installed at the same time will reduce this impact, assuming similar wear rates.

Recommendations from this test are to use the lowest ramp slope as economically feasible to reduce wear at initial flange contact, curve the slope angle at the top of the ramp to reduce potential dynamic effects, and ensure a smooth transition from ramps to level running surfaces.

This *Technology Digest* is one of a series on research, development and evaluation of potential improvements to FBF crossing diamonds. The series will include frog steels, foundation designs and transverse running surface profiles. This project was conducted by TTCI under the AAR Strategic Research Initiatives Program.

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INTRODUCTION

Flange bearing frog (FBF) crossing diamonds and turnout frogs have proven to be feasible for 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 crossing diamond condition-related slow orders have also been significant in locations where full FBF crossing diamonds have been implemented.

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

This *Technology Digest* (TD) is one in a series on TTCI's current flange bearing research and development work, and describes some of these issues and the design improvements developed in response.

FBF Diamond Performance Issues

While FBF crossing diamonds perform well dynamically as compared to conventional tread bearing frog crossing diamonds, there are still 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 adverse effects of these necessary transitions is the goal of this project.

The train and track form a complex dynamic system that is subject to changes in the static and dynamic properties of each. Variations in dynamic performance are manifested as uneven wear and deformation of track and running surfaces.

Development of Improved Dynamic Performance Ramps

Previous work developed the relationship between dynamic wheel loads, ramp rates, and vehicle speed for heavy axle load freight equipment.[1, 2] This work assumed simple linear ramps and showed that a lower incident angle between ramp and wheel flange is needed to minimize dynamic loads. Observation of operations on flange bearing ramps confirmed the modeling. It also showed that the transition to level running in flange bearing was important. The abrupt change in slope at the top of the ramps causes wheel unloading and subsequent oscillations of vertical load. While the same effect happens at the low end of the ramp, the effect is smaller due to the range of wheel flange heights and the longitudinal spread of locations where flanges make first contact with the running surface.

The project team brainstormed potential ramp shapes that might improve overall performance. Amongst the more obvious candidates were:

- Longer (i.e., lower slope) ramps. This is not always feasible due to space limitations. But it does always add to the initial cost.
- Ramps with vertical curves at the upper end. As is done with transition grade changes elsewhere on the railroad, vertical curves should lower vertical dynamic loads.
- Two-slope ramps. Ramps with a steeper slope at the low end and a subsequent shallower slope at the upper end may provide benefits, depending on the distribution of wheel flange heights in the vehicle fleet.

The three ramp designs described above were modeled in NUCARS® vehicle track dynamic simulation model. The ramps designs are shown in Figure 1. There are two areas of concern in designing flange bearing ramps. One is the ramp slope in the zone where most wheels will transition from tread bearing to flange bearing. In this zone, a smaller effective ramp slope will produce smaller dynamic loads. There is an obvious trade-off between higher initial cost and reduced maintenance costs and better performance due to lower slope ramps. In the simulation, the two-slope ramp has a slope of 1:240 for most of the length of the ramp. The other two ramps have a slope of 1:180 for most or all of their lengths.

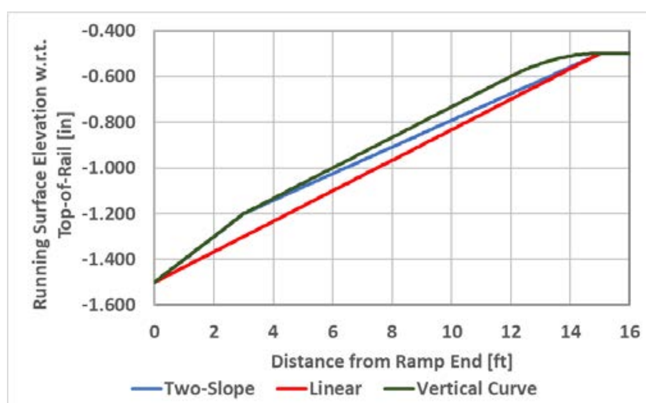

Figure 1. Profile View of Candidate Flange Bearing Ramp Designs

The second area of concern is the change in grade at the top of the ramps. As is well known in railway profile design, a transition curve is needed to connect adjacent grades. This vertical curve will help spread out accelerations and forces that result from the grade change.

* *NUCARS® is a registered trademark of Transportation Technology Center, Inc., Pueblo, Colorado*

Figure 2 provides modeling results for the three ramp designs. A loaded 315,000-pound bulk commodity car was simulated over the three ramp designs at 40 mph. Notice the characteristic dynamic load spike as wheels encounter the flange bearing ramp for each case at between 35 and 38 feet in Figure 2. This event happens a bit earlier for the two-slope and vertical curve ramps due to the steeper low end slopes of each. Also note that the magnitude of the dynamic load is slightly lower for the two-slope ramp. This is due to the lower slope at the location where the simulation vehicle becomes flange bearing. However, the effect is quite small, provided there is no large change in track stiffness at the crossing diamond as compared to open track. The lower slope in the zone where most wheels transition to/from flange bearing may ultimately provide a significant benefit in service life if it spreads the locations of wheel flange first/last contact over a significantly wider area.

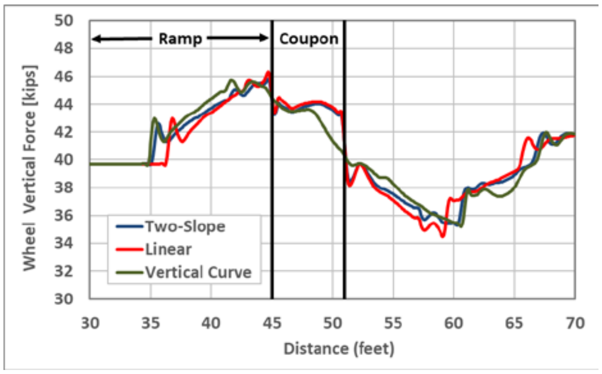

Figure 2. Simulation Results for Candidate Flange Bearing Ramp Designs

The next dynamic event of interest occurs at the top of the ramp, where wheels transition back to level running over the frogs of the crossing diamond. Notice that the vertical curve is able to reduce the maximum vertical load and increase the minimum vertical load at this transition.

Field Results

The three ramp designs modeled were built and installed in the High Tonnage Loop (HTL) of FAST for performance evaluation. The setup consisted of two sets of 16-foot ramps surrounding a 4-foot level running surface coupon. The ramps were installed as one of three experiments in a track panel. The other two experiments are ongoing: 1) evaluation of candidate frog steels; and 2) evaluation of under-tie pads to reduce track stiffness changes at transitions.

Instrumented wheelset (IWS) runs were performed on the flange bearing panel at 40 mph to compare against the numerical modeling data. The ramps in track during the IWS runs were the vertical curve in the counterclockwise (CCW) (westbound) direction with 41.3 million gross tons (MGT) and the linear ramp in the clockwise (CW) (eastbound) direction with 71.0 MGT. The level running surface coupons had 71 MGT. The two-slope ramp was replaced after 21 MGT due to a chip-out and subsequent longitudinal crack of the running surface. The chip-out was due to metal flow from an adjacent coupon. This occurred at the edge of the milled running surface groove. Lateral and longitudinal flow were under the guardrail head and went undetected until the cracking in the ramp was several inches long.

Figure 3 shows the vertical wheel loads from the leading IWS axle for both rails for the vertical curve (CCW) and linear ramps (CW) with the x=0-axis representing the start of the 16-foot ramp.

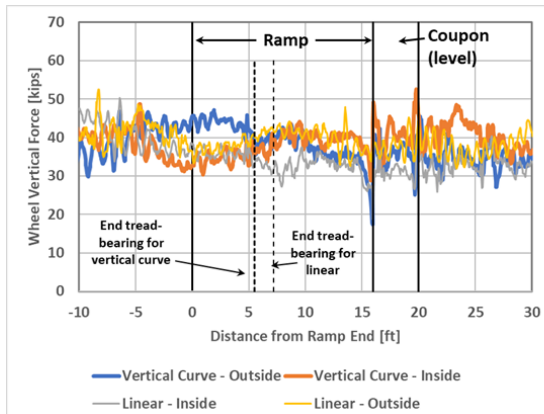

Figure 3. Measured Vertical Forces on Vertical Curve and Linear Ramps

The first observation is the flange support starts earlier on the ramp for the vertical curve (5.5 feet from ramp end) than the linear ramp (7.2 feet from ramp end). This is anticipated from the numerical modeling data presented in Figure 2 and the fact that a lower elevated ramp will result in delayed flange contact. The distance values were determined by noting when the wheel/rail contact position on the IWS switched from tread to flange bearing.

Once flange-supported, increases in vertical wheel load were difficult to separate from the normal force fluctuations during IWS runs. The vertical curve — outside rail (thick blue) and linear — inside rail (thin gray) appear to increase about 4 to 6 kips, matching numerical results, but no change is observed for the other two rails.

Both ramps show a decrease in vertical force about one foot from the level coupon. This was anticipated from the numerical modeling for the vertical curve, but not the linear ramp. A possible explanation is that the wheel flange wore down the slope change angle on the linear ramp, producing a natural curve at that location.

Impacts were observed at the ramp-to-level surface interface. For the lead axle, the impacts were similar with 49 kips on the vertical curve and 52 kips for the linear ramp. For the trailing axle, the vertical curve produced greater impacts with 62 kips versus 55 kips on the linear ramp. The impacts are produced because of the discontinuity. Greater impacts for the vertical curve are likely due to uneven wear because the level surface coupon was in track for 23 MGT longer than the linear ramp.

The wear along the various ramps was measured about every 1.5 feet at regular MGT intervals. The comparisons presented focus on how ramp shape influences the location of maximum wear, not magnitude of wear, because different steel materials were used. That topic will be addressed in a later TD. While the comparisons between ramp shapes involve two different materials, the data presented is believed to be representative.

Figure 4 compares the wear along the ramp at about 20 MGT. The results verify that the location of maximum wear on the linear ramp is further from the ramp end (~7.5 feet) than the two-slope or vertical curve ramps (~6 feet). This agrees with both the numerical and IWS data. A second observation is that the "peak wear" of the two-slope ramp is more spread out than on the linear or vertical curve ramps. This was anticipated from the lower slope at the expected point of initial flange contact. This would be advantageous in revenue service by spreading the wear from flange contact over a larger distance; however, the wide range of flange heights in revenue service may naturally mitigate this issue. A third observation is that the wear nearest to the level running surface coupon (~15 feet) is larger for the linear ramp and may indicate rounding of the slope change at the top of the ramp.

SUMMARY

The numerical modeling, IWS, and wear data showed general agreement on the responses between the three different ramp shapes: two-slope, linear, and vertical curve. All three shapes performed acceptably with the following comments.

First, lower slope angles (two-slope ramp) spread the wear from flange contact over a larger distance and reduce localized wear.[1] This will be most advantageous if all the wheel flanges are of similar heights.

Second, while the dynamic effects between the linear and vertical curve ramps were similar for the IWS runs, the vertical curve should perform better early in the life of the ramps, as the linear ramps have to wear to the vertical curve shape. In this initial wear-in period, the dynamic loads and any resultant differential wear will be higher.

Third, any stiffness change in the crossing diamond (from ramps to frogs) will make the dynamic loading from the longitudinal slope change larger. This effect was not present in the field test, due to having no frogs in the test panel. An additional recommendation is to ensure that the ramps and frogs are installed at the same time and are of the same material. Replacing a single part may result in abrupt elevation changes at joints that must be ground to a common elevation. Use of different materials or cross section profiles may result in running surface elevation variations due to different wear rates.

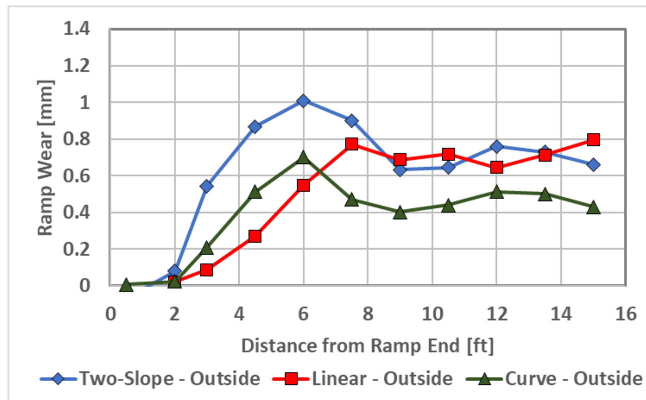

Figure 4. Measured Wear along the Ramp at 20 MGT

ACKNOWLEDGEMENTS

The financial and technical support from AAR's Strategic Research Initiative on Special Trackwork Project and TTCI is greatly appreciated.

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1. Davis, David, Rafael Jimenez, and Semih Kalay. July 2011. "Implementation Guidelines for Flange Bearing Frogs." *Technology Digest* TD-11-018, Association of American Railroads, Transportation Technology Center, Inc., Pueblo, CO.
2. Davis, David, Richard Reiff, Mike Joerms, and Semih Kalay. September 1997. "Load and Ride Quality Assessment of Crossing Diamonds." *Technology Digest* TD-97-036, Association of American Railroads, Washington, DC.