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NUCARS® Simulation of a Wheel Impact Load Cracked Rim Detector

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Summary

In cooperation with BNSF Railway Company (BNSF), Transportation Technology Center, Inc. (TTCI) is investigating the application of Wheel Impact Load Detectors (WILD) on curved track for the capability to detect cracked rim wheel defects in a moving train. For the initial step, TTCI performed an investigative NUCARS®* analysis to predict wheel lateral position for a variety of conditions such as friction, curvature, and wheel profile. The intent was to identify suitable conditions that would contribute to the maximum lateral axle shift on the curves. It was determined that lubrication was the most important factor for causing flanging and that curvatures of approximately eight degrees and above would cause the most consistent lateral shift.

The Wheel Impact Load Cracked Rim Detector (WILDCARD) concept is an adaptation of existing WILD technologies, but applied to curved track where the wheels on the low rail are likely to be riding outboard of the tapeline, closer to the rim face. In this investigation, TTCI used NUCARS® modeling to determine the likely wheel contact position for a hypothetical hopper car in both loaded and empty states. Initial studies modeled segments from the High Tonnage Loop (HTL) test track located at the Facility for Accelerated Service Testing (FAST) so that a suitable test site could be established. Wheel positions during normal FAST operations and during testing at 40 mph on the HTL should represent a realistic case for WILDCARD operation. Worn wheel and rail profiles were then modeled to understand the effects of worn wheel profile and how it might influence the wheel position during curving. For the purposes of testing at the Transportation Technology Center (TTC), this analysis confirms that satisfactory results should be achievable for a WILDCARD placed on the HTL curves with top-of-rail lubrication.

This work is a part of the Association of American Railroads' (AAR) Strategic Research Initiatives (SRI) program.

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

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BACKGROUND

Up to 74 percent of broken wheels fail in service absent of high impacts as measured on traditional WILD systems.[1] Wheel failures such as vertical split rim (VSR) and shattered rim crack (SRC) often exhibit damage at or near the edge of the tread. Wheel defects that occur outside of the tapeline are not reliably detected by WILD. Figure 1 shows an example of an SRC wheel that never exceeded WILD impact limits.

Figure 1. Example Cracked Wheel Damaged near the Rim

INTRODUCTION

As a part of the AAR SRI program, and in cooperation with BNSF railroad, TTCI is researching methods to detect broken rim wheels. Existing WILD systems have identified a great many high impact wheels. But WILD does not find all wheels that break. As wheels are removed, especially VSR wheels, operators have observed little correlation between high impact wheels and VSR-type failures. Yet the VSR wheels often exhibit the tread damage typical of wheels removed for high impact. The question arises, why aren't these wheels discovered by WILD? Upon investigation, it is apparent that the VSR wheel failures occur predominantly outboard of the tapeline. Conventional WILD systems on tangent track do not reliably capture this failure mode. The proposed method of detecting wheel surface defects outboard of the tapeline is being called WILDCARD. This report outlines initial steps taken to configure a system for concept demonstration at FAST.

TESTING ENVIRONMENT

A first step in the WILDCARD development was to determine a suitable location for prototype testing at TTC. To reliably detect wheels on both sides of the train, curving in both directions is needed, such as S-curves or reverse curves. For testing and concept demonstration, where conditions can be controlled, testing on a single rail is sufficient. Modeling the curves through Section 7 and Section 25 of the HTL will allow comparison of wheel positions on two typical curves. The curve in Section 7 has a radius of 1,145 feet and superelevation of 4 inches. Balance speed is 34 mph on this 5-degree curve. Similarly, the curve of Section 25 immediately follows Section 7 and has a radius of 956 feet and 5 inches of super elevation. Balance speed is 38 mph on this 6-degree curve.

NUCARS® RUN ENVIRONMENT

NUCARS® is a lumped parameter modeling software that allows the user to model and accurately predict all aspects of railcar dynamics. The run environment requires building the railcar model, modeling the track that the car will run on, and specifying operation parameters such as speed and friction at the wheel/rail interface. The empty and loaded hopper car models provided with NUCARS® were selected. AAR-1B wide-flange wheel profiles and new 136RE rail profiles were considered for this simulation. The coefficient of friction (μ) was initially set to 0.5 for both tread and flange for the left and right wheels. This represented the dry rail, high friction case. Simulations were run from 25 mph to 50 mph in 5 mph increments for both the empty and loaded car models.

MODEL RESULTS

Simulation models considered in this research effort represented a generic car that is typical of use for freight service in North America. Results are not meant to represent a specific operating condition, but rather to represent a general case that should be typical of railcars with three-piece trucks. The case of flanging on the high rail represents the maximum lateral shift and is considered the best case for WILDCARD placement. As expected, leading and trailing axles perform differently. The lead axles (axle position 1) tend to flange more reliably than the trailing axles (axle position 4). Figure 2 shows wheel lateral position for the leading axle on a dry 6-degree curve at 40 mph for a hopper car.

Figure 2. Axle Position 1 for a 6-degree Curve at 40 mph

The worst case for lateral shift was axle position 4 and high friction conditions. High friction conditions cause longitudinal traction that forces the trailing axle toward the low rail. Figure 3 shows wheel lateral position for trailing axle on a dry 6-degree curve at 40 mph for a loaded hopper car.

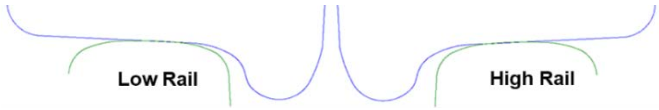
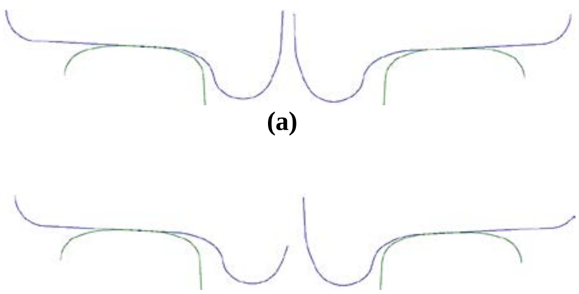


Figure 3. Axle Position 4 for a 6-degree Curve at 40 mph with Dry Friction

EFFECTS OF LUBRICATION

Top-of-rail lubrication reduces the longitudinal traction forces which tend to center the trailing axle. In general, lower friction is favorable for flanging on all axles. As a result, the best location for installation of the WILDCARD system will be on a lubricated curve. Figure 4 shows wheel lateral position for trailing axle on a 6-degree curve at 40 mph with and without lubrication.



(a)

(b)

Figure 4. Axle Position 4 on a 6-degree curve. (a) Without Lubrication (μ = 0.5); (b) With Lubrication (μ = 0.35)

EFFECTS OF TREAD WEAR

After running initial models with new wheel/rail profiles, the effect of wheel profile was investigated to determine its influence on flanging. For this analysis, flange contact was considered any lateral displacement greater than 0.5 inch. Near-flange contact, considered suitable for the purpose of impact measurement, was any displacement greater than 0.3 inch. Five progressively worn wheel profiles were used along with lubricated friction levels of 0.35.

The model stabilized quickly in each curve, reaching a steady state after approximately 100 feet of the curve. An average of displacements over the central 200-foot section of curve was taken. Since the simulation was symmetric, little to no variation was seen in the displacements on the right-hand versus the left-hand curve. Wheel profile wear demonstrated a very strong influence on low rail contact point position. The observation is that narrower flanges generally coincide with more hollow tread, which tends to increase lateral offset and place the contact point further outboard. Figure 5 shows wheel lateral position for trailing axle with varying wheel profile.

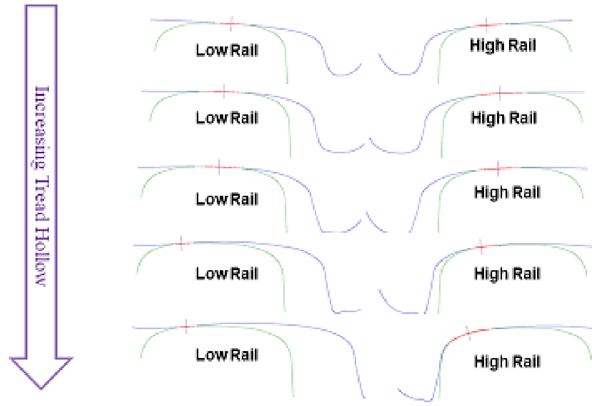


Figure 5. Effects of Tread Wear on Axle Position 4 on a 6-degree Curve with Lubrication (μ = 0.35)

EFFECTS OF CURVATURE

The next set of runs simulated 1-degree increments of curvature from 2 degrees through 10 degrees. An underbalanced condition of 1.5 inches was assumed, and the elevations for each curve were based on typical running speeds in such a curve. A minimum of 30 mph was used, as this is the typical minimum operating speed of tangent WILD systems. The track model used was a 300-foot spiral into a right hand curve of 400 feet at the prescribed curvature and elevation, followed by a 500-foot spiral into 400 feet of the opposite curvature and elevation. Figure 6 shows the result.

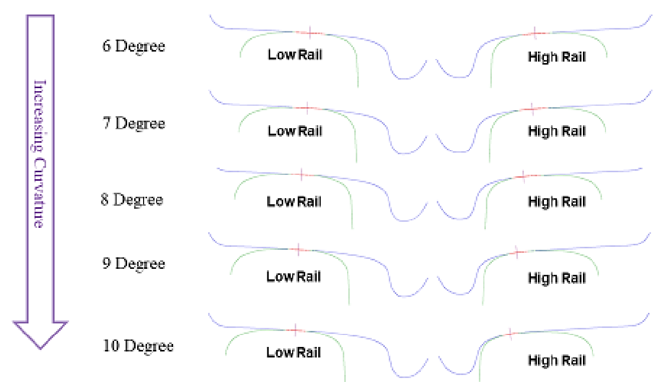


Figure 6. Effects of Curvature on Axle Position 4 on Varying Degrees Curve with Lubrication (μ = 0.35)

Figure 6 illustrates that the increasing curvature increases the likelihood of flanging on all axles. Curvatures of 8 degrees and above provide the most reliable flanging regardless of wheel profile.

ANALYSIS

The behavior of each vehicle model was quite stable throughout the simulations. There was some dynamic behavior in the spiral and spiral transitions, but no consistent flanging in these transitions. There was no substantial difference between the loaded and empty cars. Wheel lateral position was consistent during fully developed curving. Once the axles found their lateral position, they stayed there throughout the curve.

Wheel position does vary with speed, but not as much as it does by axle position on the car. Axles 1 and 3, the leading axles of each truck, tend to flange at all speeds from 25 to 50 mph on both the empty and loaded cars. Axles 2 and 4 do move laterally, but not always toward the outside rail. This is common with three piece trucks. Figure 7 shows the axle lateral offsets for the case of a loaded car with high friction at 40 mph. Notice that axle 4 (purple line) stays nearly centered on the track, with a slight inward displacement on the curve in Section 25.

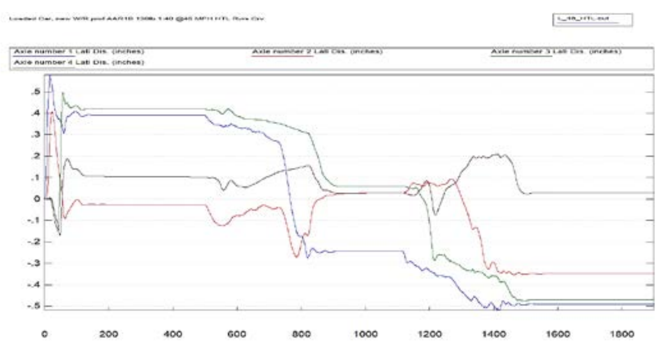


Figure 7. Axle Lateral Displacements with Dry Friction

Axle lateral position is strongly influenced by friction at the rail. Adding lubrication to the rail to reduce friction profoundly influences axle lateral position. Figure 8 shows the axle lateral displacement at 40 mph with lubrication. With top-of-rail lubrication, axles 2 and 4 now move outboard on both curves.

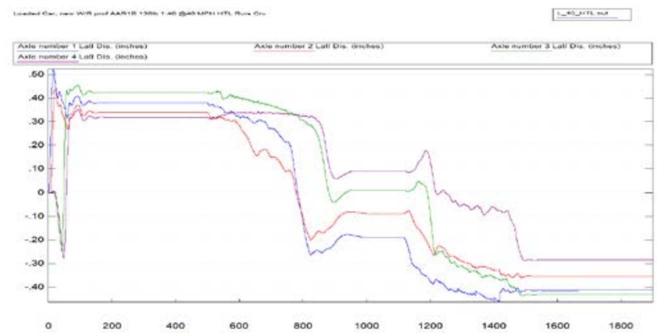


Figure 8. Axle Lateral Displacements with Lubrication

SYSTEM PLACEMENT

Based on the modeling results obtained from NUCARS®, it is recommended to place the system in a lubricated reverse curve. TTCI also recommends installing WILDCARD immediately following a traditional WILD detector in the tangent. The idea of placing WILDCARD beyond tangent WILD is to allow comparison of measurements between both systems. Wheel tread conditions at the outer edges of the rims on both sides of the train could be compared to rolling condition on the tapeline as measured by the tangent WILD.

CONCLUSION

NUCARS® modeling/simulation demonstrated that the flanging at 40 mph can be expected to occur on the outside wheels of axles 1 and 3 on the curve in section 25 of the HTL at TTC. Near flanging condition occurs on axles 2 and 4 at this speed with lubrication. With top-of-rail lubrication, the inside or lower wheel on the curve maintains contact outboard of the tapeline through the curve. The difference in wheel lateral position between axles 1, 3 and axles 2, 4 is less than 0.125 inch under these operating conditions with new wheel profiles. This is the case for either an empty or loaded car. Wheel profile alters the results, often favorably. In cases of increasing wheel hollow, the contact point for the wheel on the low rail can be expected to move outward.

REFERENCES

1. 2015 AHSC Study on Wheel Failures