USE OF SWITCH PROFILE DATA FOR ENHANCED ANALYSIS OF WHEEL RAIL BEHAVIOR AT SWITCH POINT

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SUMMARY

The switch (turnout) represents a complex component of the track structure that generates high levels of vertical and lateral dynamic forces. The effects of these dynamic forces, which vary in location as the wheels negotiate the switch rail into the body of the switch, are dependent on the profiles of the switch and stock rails, which taken together with the wheel profile, defines the wheel/rail contact conditions. New generation automated switch inspection vehicles have the ability of defining these profiles accurately, through the switch, to include detailed profiles of the switch and stock rails at 1in (25mm) intervals. While traditional uses of these switch inspection vehicles has focused on measuring the switch and frog elements, more advanced analyses make use of the detailed profile information to define safety and maintenance issues.

This paper presents two such applications of wheel/rail profile analysis. The first analysis addresses wheel climb potential by utilizing measured switch point profiles together with a range of measured wheel profile data (from new to worn). By superimposing the measured switch rail and wheel profiles and by calculating the L/V threshold for wheel climb as a function of gauge face wear angle, it is possible to better understand the vulnerability of switches to wheel climb derailments. This includes identifying locations where there exists a potential for wheel climb as well as determining the maximum allowable L/V ratios through the switch.

The second analysis addresses the actual wheel/rail contact stresses on the switch and stock rails, with analytic results to include location and magnitude of these contact stresses. The analysis utilizes a procedure for aligning the wheelset profile to the measured switch rail profile pair as it varies through the turnout. Using the point-by-point alignment, the analysis determines the location of the wheel/rail contact patch and then calculates the magnitude of the contact stress profile in that patch. The result is a map of the wheel/rail contact stress as a vehicle moves through the turnout.

INTRODUCTION

The switch (turnout) represents a complex component of the track structure that generates high levels of vertical and lateral dynamic forces. The effects of these dynamic forces, which vary in location as the wheels negotiate the switch rail into the body of the switch, are dependent on the profiles of the switch and stock rails, which when taken together with the wheel profile, defines the wheel/rail contact conditions. Until recently, accurate information on the profile of the switch rails, the frog, and other rail surfaces of the turnout were difficult to accurately obtain. However, new generation automated switch inspection vehicles now have the ability of defining these profiles accurately, through the switch, to include detailed profiles of the switch and stock rails at 1in (25mm) intervals [1, 2]. While traditional uses of these switch inspection vehicles have focused on measuring the switch and frog elements [1, 2], this is only the first application of this new and valuable information about the rail surface profiles throughout the turnout. Accurate rail surface information, to include the switch point and the frog running surface condition (and profile) lends itself
to more advanced analyses which take advantage of this detailed profile information to define safety and maintenance issues.

This paper presents two such applications of wheel/rail profile analysis. The first analysis addresses wheel climb potential by using switch point profile data together with measured wheel profile data (for a range of new and worn wheel profiles). By superimposing the switch rail and wheel profiles and then calculating the L/V threshold for wheel climb as a function of gauge face wear angle, it is possible to better understand the vulnerability of switches to wheel climb derailments. This is a major derailment category that concerns every class of railway operations from heavy axle load freight, to high speed passenger, to transit (metro) operations. Such an analysis allows for the identification, within the switch, of those locations where there exists a potential for wheel climb. It also allows for the determination of maximum allowable L/V ratios through the switch, which is important in vehicle design as well as in understanding vehicle/track interaction through the turnout.

The second analysis addresses the actual wheel/rail contact stresses on the switch and stock rails, with analytic results to include both location and magnitude of these contact stresses. Such an analysis can determine the location of the wheel/rail contact patch as a function of actual switch and frog profiles and then calculate the magnitude of the contact stress profile in that patch. The result is a map of the wheel/rail contact stress as a vehicle moves through the turnout. Such a map becomes a valuable tool in defining maintenance activities such as grinding or component replacement.

**WHEEL/RAIL FORCES AND WHEEL CLIMB AT THE SWITCH POINT**

According to the Federal Railroad Administration (FRA) Safety Data [3], switch point conditions represent the largest category of derailment causes, accounting for 39% of the turnout-caused derailments reported in 2013.

Dynamic wheel climb is a class of derailments most commonly associated with high levels of lateral loads and corresponding high L/V ratios (ratio of Lateral wheel/rail force to Vertical wheel/rail force). Dynamic wheel climb has been reported for all modes of rail operations including freight, passenger, and transit operations. Significant research has been directed towards the mechanisms associated with dynamic wheel climb derailments with derailment criterion developed by such researchers as Nadal [4], Weinstock [5], and others [6, 7].

**Figure 1: Number of derailments at turnouts based on causes**

Dynamic wheel climb derailments are most commonly associated with sharp curves, where high levels of lateral wheel/rail force are generated. Similarly, wheel climb derailments are found on turnouts, particularly in the curved portion of the turnouts, where high levels of lateral wheel rail forces have been developed.

Among those factors that have been reported to contribute to this class of wheel climb derailments is the angle of the gage face of the rail, usually the outside or high rail of the curve [8]. This angle is often found on rails subject to gage face wear, i.e. outside or high rails, where this gage face wear can result in the development of an angle \( \phi \) between the gage face and the vertical (Figure 2). As this angle increases, the potential for a wheel to climb the gage face of the rail increases. This wheel climb will occur when the net "upward" component of the lateral (L) and vertical (V) wheel/rail forces, parallel to the gage face, is greater than the resistance to that force due to the normal force component N (Figure 2) and the corresponding coefficient of friction f, i.e. N \( \bullet f \) [8]. Thus wheel climb can occur when the following relationship occurs (Figure 3):

\[
\frac{L}{V} < \tan (\beta - f')
\]

(1)

Where:

- \( L \) = Lateral wheel/rail force
- \( V \) = Vertical wheel/rail force
- \( \beta = 90 - \phi \) (see Figure 3)
- \( f' = \tan^{-1}(f) \)

and

\( f \) = coefficient of friction

**Figure 2: Wheel/worn rail interaction [8]**

\[\text{Derailment causes at turnouts in 2013}\]

<table>
<thead>
<tr>
<th>Cause</th>
<th>Number of Derailments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Switch Point Gap</td>
<td>36%</td>
</tr>
<tr>
<td>Frog</td>
<td>4%</td>
</tr>
<tr>
<td>Switch Point</td>
<td>15%</td>
</tr>
<tr>
<td>Switch Mechanism</td>
<td>4%</td>
</tr>
<tr>
<td>Stock Rail</td>
<td>1%</td>
</tr>
<tr>
<td>Other</td>
<td>18%</td>
</tr>
</tbody>
</table>

**Figure 3: Wheel/rail forces and wheel climb at the switch point**
Thus accurate measurement of rail profile, such as by the Automated Switch Inspection Vehicle (ASIV) (Figure 4 [2]) is needed to properly ascertain the condition of the rail, to specifically include this gage face wear angle. Using this accurate rail profile data and overlaying the different types of wheel profiles (from new to worn wheel profiles) it is possible to understand the vulnerability of turnouts to wheel climb derailments. Thus the above defined wheel climb methodology can be used to identify which areas of an examined turnout have the potential for wheel climb derailment in terms of worn rail L/V ratio limit [9]. Maximum allowable L/V ratios through the turnout can also be determined which can be use in both turnout and vehicle design. Such applications included calculating maximum allowable loads and corresponding speed limits.

The following methodology was applied in order to determine critical points through the turnouts [9]:

1. Overlay the wheel and the rail profile to determine the wheel/rail contact point with the top of the stock rail touching the tread of the wheel
2. Define the wheel flange face to be in contact with the rail gage face
3. Determine the contact point coordinates on the rail gage face at each point along the switch rail
4. Calculate the switch point face angle at each point along the switch using the coordinates of the contact points
5. Apply the worn rail L/V ratio formula to calculate L/V ratios at each section of the turnout
6. Plot the maximum allowable L/V ratios along the switch rail with different lubrication levels of the rail based on actual rail gage face angle values along the switch rail

The following examples illustrate how this methodology is applied [9].

EXAMPLE 1: MODERATELY WORN WHEEL OVERLAID ON “GOOD” CONDITION HIGH SPEED (#26) TURNOUT

This example uses ASIV measurement profile data from a "good" switch point in a number 26 (high speed) turnout that includes rail cross-section profiles from the beginning of the switch point to the end of the turnout in one inch increments. This switch point is 444 inches (37 ft) long. As noted
above, the analysis focused on calculating the L/V ratio threshold for wheel climb as a function of gage face wear angle through the switch point. Noting that the first few inches of the switch point pose the highest risk of L/V issues, the analysis of the first 10 inches was performed at one inch increments. Past the initial 10 inches, to the end of the switch point, the interval between analyses was increased to a maximum of 25 inches.

Figure 7 illustrates an overlay of the moderately worn wheel (Figure 6) on the “good” switch point at the beginning of the switch point (x= 0°).

The contact point coordinates are determined to be:

<table>
<thead>
<tr>
<th>X coordinates</th>
<th>Y coordinates</th>
</tr>
</thead>
<tbody>
<tr>
<td>-1.041</td>
<td>2.359</td>
</tr>
<tr>
<td>-1.035</td>
<td>2.374</td>
</tr>
<tr>
<td>-1.028</td>
<td>2.388</td>
</tr>
<tr>
<td>-1.020</td>
<td>2.402</td>
</tr>
</tbody>
</table>

Using the L/V threshold formula (1), the wheel climb threshold is calculated at each point along the switch rail as a function of the different coefficients of friction that can occur in track. This ranges from dry conditions with a coefficient of friction of 0.5 to well lubricated conditions with a coefficient of friction of 0.2. This is summarized in Table 1.

Note that as the coefficient of friction increases, the L/V threshold decreases, as expected, from the actual physical behavior. This results in an increase of risk for wheel climb (Figure 10).

Figure 8 shows the change in contact gage face angle as a function of position along the switch rail, and Figure 9 shows the resulting L/V wheel climb threshold along the switch rail.
Table 1: Summary of the good switch point/moderately worn wheel interaction within the turnout

<table>
<thead>
<tr>
<th>x(in)</th>
<th>φ(degree)</th>
<th>β = 90-φ (degree)</th>
<th>f(0.2)</th>
<th>L/V [f(0.2)]</th>
<th>f(0.3)</th>
<th>L/V [f(0.3)]</th>
<th>f(0.4)</th>
<th>L/V [f(0.4)]</th>
<th>f=0.5</th>
<th>L/V [f=0.5]</th>
</tr>
</thead>
<tbody>
<tr>
<td>250</td>
<td>11.9</td>
<td>78.1</td>
<td>0.2</td>
<td>2.33</td>
<td>0.3</td>
<td>1.83</td>
<td>0.4</td>
<td>1.50</td>
<td>0.5</td>
<td>1.26</td>
</tr>
<tr>
<td>275</td>
<td>2.5</td>
<td>87.5</td>
<td>0.2</td>
<td>4.07</td>
<td>0.3</td>
<td>2.87</td>
<td>0.4</td>
<td>2.21</td>
<td>0.5</td>
<td>1.80</td>
</tr>
<tr>
<td>300</td>
<td>-2.2</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>325</td>
<td>-1.1</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>350</td>
<td>-1.9</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>375</td>
<td>-1.4</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>400</td>
<td>-1.5</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>425</td>
<td>-2.2</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
<tr>
<td>444</td>
<td>-1.94</td>
<td>-</td>
<td>0.2</td>
<td>-</td>
<td>0.3</td>
<td>-</td>
<td>0.4</td>
<td>-</td>
<td>0.5</td>
<td>-</td>
</tr>
</tbody>
</table>

Note, the L/V threshold is always greater than 0.8, which is a traditional “safe” L/V threshold level, and in fact most of the time is above 1.0.

EXAMPLE 2: MODERATELY WORN WHEEL OVERLAID ON DAMAGED SWITCH POINT

This example uses ASIV measurement profile data from a damaged switch point in a number 10 turnout that includes rail cross-section profiles from the beginning of the switch point to the end of the turnout in one inch increments. This switch point is 191 inches (16 ft) long. Figure 10 illustrates an overlay of the moderately worn wheel (Figure 6) on the damaged switch point at the beginning of the switch point (x= 0”).

The contact point coordinates are determined to be:

<table>
<thead>
<tr>
<th>X coordinates</th>
<th>Y coordinates</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.982</td>
<td>2.500</td>
</tr>
<tr>
<td>-0.978</td>
<td>2.517</td>
</tr>
<tr>
<td>-0.973</td>
<td>2.534</td>
</tr>
<tr>
<td>-0.969</td>
<td>2.550</td>
</tr>
<tr>
<td>-0.964</td>
<td>2.567</td>
</tr>
<tr>
<td>-0.959</td>
<td>2.583</td>
</tr>
<tr>
<td>-0.954</td>
<td>2.599</td>
</tr>
</tbody>
</table>

As in Example 1, using the L/V threshold formula (1), the wheel climb threshold is calculated at each point along the switch rail as a function of the different coefficients of friction that can occur.
in track. This ranges from dry conditions with a coefficient of friction of 0.5 to well lubricated conditions with a coefficient of friction of 0.2.

Figure 11 shows the change in contact gage face angle as a function of position along the switch rail, and Figure 12 shows the resulting L/V wheel climb threshold along the switch rail.

Figure 11: Contact point face angle change in moderately worn wheel and damaged switch point

It should be noted that beyond 75in from the starting point there is no positive gage face angle, thus based on the theory, no risk of wheel climb.

Figure 12: Allowable L/V ratios through the damaged switch point turnout with different lubrication levels

Here too, the L/V threshold is always greater than 0.8, though it approaches 0.8 for a length of the switch point, and then increases to well above 1.0.

EXAMPLE 3: MODERATELY WORN WHEEL OVERLAID ON A WORN SWITCH POINT

This example uses ASIV measurement profile data from a worn switch point in a number 7.5 turnout that includes rail cross-section profiles from the beginning of the switch point to the end of the turnout in one inch increments. This turnout is 273 inches (23 ft) long. Figure 13 illustrates an overlay of the moderately worn wheel (Figure 6) on the worn switch point 16 inches back from the start of the switch point (x=16°).

The contact point coordinates are determined to be:

<table>
<thead>
<tr>
<th>X coordinates</th>
<th>Y coordinates</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.526</td>
<td>2.807</td>
</tr>
<tr>
<td>2.536</td>
<td>2.797</td>
</tr>
<tr>
<td>2.544</td>
<td>2.785</td>
</tr>
<tr>
<td>2.552</td>
<td>2.772</td>
</tr>
<tr>
<td>2.558</td>
<td>2.759</td>
</tr>
<tr>
<td>2.565</td>
<td>2.744</td>
</tr>
</tbody>
</table>

Here again the wheel climb threshold is calculated at each point along the switch rail as a function of the different coefficients of friction from dry (0.5) to well lubricated (0.2).

Figure 14 shows the change in contact gage face angle as a function of position along the switch rail, and Figure 15 shows the resulting L/V wheel climb threshold along the switch rail.

Figure 14: Contact point face angle change in moderately worn wheel and worn switch point

It should be again noted that beyond 75in from the starting point there is no positive gage face angles, and therefore, no risk of wheel climb.
In this example, the L/V threshold drops below 0.8 for the dry (unlubricated) rail condition. This occurs from 8” to 20” back from the switch point, with a minimum value of 0.63, well below the traditional “safe” wheel climb limit of 0.8. As such, this is a location with a significant risk of wheel climb considering that that region of the switch is traditionally a high wheel climb risk location. Thus, it can be seen that this analytic approach allows for the determination of locations within the turnout with a high potential for wheel climb.

CONTACT STRESSES THROUGH THE SWITCH POINT

A second type of analysis, that makes use of the ability to accurately measure the switch rail, stock rail and frog running surfaces of the turnout, looks at the wheel/rail contact stresses on the turnout running surfaces. Using this data allows for the accurate determination of the alignment of the wheelset to the running surface profile as it varies through the turnout as well as the corresponding location of the wheel/rail contact and associated wheel/rail contact stress. The result is a map of the wheel/rail contact stress as a vehicle moves through the turnout.

One such approach [11] utilizes such a wheel/rail alignment model, together with the actual running surface profiles to determine the alignment of the fixed wheelset and the associated wheel/rail contact locations. The analytic approach used assumes that the absolute difference in rolling radius of the two wheels on a fixed axle, must be less than or equal to a defined rolling radius difference threshold. For the diverging or curved leg of the turnout, the following relation needs to be satisfied for a rigid axle set:

\[
A \left(b \left(\frac{R_{\text{inner}}}{R_{\text{outer}}}, \frac{R_{\text{inner}}}{R_{\text{outer}}} - R_{\text{outer}} \right) \right) \leq R_{\text{roll threshold}}
\]

Where:
- \( R \) is the radius of curvature of a rail at contact point
- \( r \) is the rolling radius of a wheel at contact point

An iterative two phase approach was implemented to determine the contact sites (contact patches) based on the rolling radius difference. For both phases, the state of wheel/rail contact is based on elastic deformation thresholds. The contact threshold is defined as the maximum normal distance between the wheel and rail where it is assumed contact is being made based on elastic material deformation. Figure 16 illustrates a contact patch and how the normal distance between the wheel and rail is used to define deformation based contact. For phase one, an initial deformation based on a generic Hertzian analysis is used as the contact threshold. For phase two, the contact threshold is calculated based on the application of a crossed cylinder Hertzian contact stress model (Figure 17) to the contact state reached during phase one. The wheel/rail alignment and contact state is then adjusted using the modeled deformation based contact threshold. For more detailed description of the analysis approach, refer to Reference 11.

![Figure 15: Allowable L/V ratios through the worn switch point turnout with different lubrication levels](image)

![Figure 16: Wheel/Rail contact with a detailed view of the normal distance between wheel and rail along the contact patch](image)
The results presented in this paper are based on a static vertical wheel load of 36 kips (160.13 kN) corresponding to a conventional 286,000 lb. freight car. The assumed lateral load is 7.5 kips (33 kN). In addition, the vertical wheel load was adjusted for dynamic effects using the AREMA Impact Factor based on a diverging speed in the turnout of 5 mph (8 kph) and a wheel diameter of 36 inches (0.91 m).

Using these load levels, the normal reaction force components were calculated at the contact patch and then converted to maximum compressive stresses and expected deformations using the Hertzian contact stress model noted above [12, 13, 15].

Figure 19 presents the results of the contact patch stress analysis for both a section in the straight leg (a) and one in the diverging leg (b). As can be seen in this figure, the calculated contact stress distribution is color coded to allow for overlay onto the profile sections, at the point of contact. Note the very different contact points on the straight and diverging (curved) legs of the turnout.

Figure 19: Switch rail profiles with contact stress distributions
(a) Straight leg, (b) Diverging leg

Figure 20 illustrates a more comprehensive stress mapping through the diverging leg of a number 10 turnout. This includes detailed stress maps at the switch point, frog, and closure rails. As shown in this figure, areas of high stress were located near the switch point where the wheel load transitions from the stock rail to the switch rail in a facing point movement (or from switch rail to stock rail in a trailing point movement) and near the frog point where the wheel load transitions from the wing rail to the frog nose in a facing point movement (or from the frog nose to the wing rail in a trailing point movement).
Furthermore, the diverging legs of a turnout generally have two-point contact (as illustrated by the two sets of contact illustrated in Figure 19). By contrast, the through or straight leg had only a single contact patch on the head of the rail as the rolling radius difference converges when the wheelset is close to being centered.

![Image](https://via.placeholder.com/150)

**Figure 20. Contact patch stresses through a right hand diverging leg of a #10 turnout. Stress ranges from 0 (Blue) TO 40 KSI (2758 MPA) (Red)**

(a) Frog, (b) Closure rails between switch and frog, (c) Switch, (d) Complete stress view from switch point to the end of frog

Figures 21 and 22 present the stress analysis as a moving average of the maximum contact stress for the primary contact patches found on the top surface of the rails for the straight and diverging leg of a turnout. The significantly increased level of contact stress at the switch point and frog on the diverging leg can be seen clearly in Figure 22. For the straight leg, there is no switch rail contact near the switch point as the stock rail carries the wheel beyond the point, and as a result the maximum stress levels occur at the nose of the frog where wheels have to transition across the frog gap (Figure 21).

![Diagram](https://via.placeholder.com/150)

**Figure 21: Maximum contact stress throughout the straight leg of a turnout (Left rail contains switch rail and frog)**

![Diagram](https://via.placeholder.com/150)

**Figure 22: Maximum contact stress throughout the diverging leg of a turnout (Left rail contains switch and frog)**

**CONCLUSIONS**

This paper shows the improved types of analyses of wheel rail performance through a turnout that can be achieved using accurate wheel and turnout rail profiles as measured by the latest generation automated measurement systems. Such analyses go beyond the basic measurement of the switch and frog elements, which while extremely important in defining maintenance and safety limits, represent just the first "level" of...
analyses that can be performed using this new generation of accurate surface condition data. These more advanced analyses make use of the detailed profile information to define safety and maintenance issues, such as wheel/climb thresholds and locations or wheel/rail contact stress levels. This new information, in turn, can allow for improvements in component and system design, particularly those focused on the dynamic interaction of the wheels through the complex turnout geometry. This can also lead to better turnout design, optimization of the turnout railhead profiles and improvement in those maintenance activities associated with turnout maintenance.

Ultimately, this new information can directly lead to more proactive maintenance activity for turnouts such as optimization of the switch and crossing grinding process to allow for maintenance (and “repair”) of the degraded rail profiles. This approach has been used for many years on conventional (“plain line”) rails where optimized rail profiles are ground into the rail head to reduce high contact stress levels and significantly extend the life of the rail. With the ability to measure the profiles in the turnout and analyzing the associated wheel/rail contact stresses on the turnout and its key components, this approach can now be extended to switch and crossing grinding where degraded turnout rail head profiles can be re-profiled to improve wheel/rail contact, reduce associated contact stress levels and reducing the risk of wheel climb and associated wheel climb derailments.

REFERENCES

(9) Ozturk, Samet, A Study On the Wheel-Rail Interaction at Switch Points to Reduce Derailments in turnouts, Master’s thesis, University of Delaware, Spring 2014