ABSTRACT
As higher demands are placed on North American railroad infrastructure by heavy haul traffic, it is increasingly important to understand the factors affecting the magnitude and distribution of load imparted to concrete crosstie rail seats. The rail seat load distribution is critical to the analysis of failure mechanisms associated with rail seat deterioration (RSD), the degradation of the concrete surface at the crosstie rail seat. RSD can lead to wide gauge, cant deficiency, and an increased risk of rail rollover, and is therefore of primary concern to Class I Freight Railroads in North America. As part of a larger study aimed at improving concrete crossties and fastening systems, researchers at UIUC are attempting to characterize the loading environment at the rail seat using matrix-based tactile surface sensors (MBTSS). This instrumentation technology has been implemented in experimentation utilizing both field conditions and novel laboratory facilities, and has provided valuable insight to further previous RSD research. This paper will compare data collected from laboratory and field MBTSS experimentation to explore the effect of fastener wear on the behavior of the rail seat load distribution. The knowledge gained from this experimentation will be integrated with associated research conducted at UIUC to form the framework for a mechanistic design approach for concrete crossties and fastening systems.

INTRODUCTION
As the demand in North America for high-performance, low-maintenance railroad infrastructure continues to increase, concrete crossties and elastic fastening systems are becoming increasingly common. Concrete crossties are typically used in areas of high curvature and steep grades on lines that experience high-speed or heavy-axle load traffic (1). Because of the increasingly common application of concrete crossties and elastic fastening systems in these high-demand environments, it is important to understand the factors contributing to common performance failures of concrete crossties and fastening systems. One of the most common failures of concrete crossties is the degradation of the concrete material directly below the rail, in the area of the crosstie known as the rail seat. This degradation is commonly referred to as rail seat deterioration (RSD), or rail seat abrasion (RSA). Figure 1 illustrates a typical severe instance of RSD, with the depth of wear increasing towards the field side of the rail seat. RSD has become a problematic failure for concrete crossties since it was first observed in the 1980’s, and is often found in regions of steep grades, high curvature, and the presence of moisture (1). If left untreated, RSD may

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lead to accelerated wear of the fastening system, wide gauge, excessive rail cant, and an increased risk of derailment due to rail rollover (1).

According to a survey of North American railroad industry representatives, RSD is considered the most critical problem with concrete crossties and fastening systems. Additionally, it was ranked as the area of crosstie and fastening system research most in need of research (2). As part of a larger research project funded by the Federal Railroad Administration (FRA) investigating common failures with concrete crossties and elastic fastening systems, researchers at the University of Illinois at Urbana-Champaign (UIUC) are investigating the failure modes associated with RSD. Previous research has identified five failure mechanisms that may result in RSD: abrasion, crushing, freeze-thaw cracking, hydro-abrasive erosion, and hydraulic pressure cracking (1). Of these five failure mechanisms, four are affected by the distribution of load at the crosstie rail seat, the exception being freeze-thaw cracking. Therefore, researchers at UIUC have undertaken an effort to better understand the distribution of the rail seat load, the factors that affect it, and its effect on rail seat deterioration. Previous research has highlighted the effect of pad modulus, fastening system type, particle intrusion, and RSD on the rail seat load distribution (3, 4, 5). Researchers at UIUC hope to incorporate the findings on RSD failure mechanisms with other FRA-funded research to generate a framework for the mechanistic design of concrete crossties and their fastening systems, in which components are designed from expected outputs and observed relationships. It is believed that such a design approach would establish a clearer procedure for designing crossties and fastening systems, resulting in fewer service failures and higher reliability of the track structure and its components (6).

INSTRUMENTATION TECHNOLOGY

To characterize the distribution of load at the crosstie rail seat, researchers at UIUC have utilized matrix-based tactile surface sensors (MBTSS). The MBTSS system used by UIUC is manufactured by Tekscan® Inc. and consists of rows and columns of conductive ink which, when pressed together by a load applied normal to the contact plane, output a change in resistivity at each intersection of a row and a column. This output, termed a “raw sum”, can be interpreted as the pressure exerted on the sensor at a given intersection when given the total applied load. MBTSS simultaneously outputs the area over which this load is applied. This is termed the “contact area” of the load and is calculated from the number of sensing locations that indicate an applied load. Data is collected from the entire sensing area at a maximum rate of 100 Hz. The data is calibrated during analysis using a known or assumed input load. Previous experimentation at the University of Kentucky (UK) and UIUC have shown that MBTSS are susceptible to shear and puncture damage. To protect the sensors, layers of biaxially-oriented polyethylene terephthalate (BoPET) and polytetrafluoroethylene (PTFE) are secured to both sides of a sensor that has been trimmed to fit the rail seat. The assembly is then installed between the rail pad assembly and the concrete crosstie rail seat (Figure 2) (7).
FIELD EXPERIMENTATION
Field experimentation was performed at the Transportation Technology Center (TTC) in Pueblo, Colorado, USA; a railroad research and testing facility that consists of 48 miles (77.2 km) of track with variable geometries and operating conditions. A section of 15 new concrete crossties with Safelok I shoulders and fastening systems was installed on the 13.5 mile (21.7 km) Railroad Test Track (RTT) in a section of tangent track. Eight rail seats, on five crossties, at this site were instrumented with MBTSS, as shown in Figure 3. Five consecutive rail seats (located on the near rail, rail seats 1N through 5N) were chosen in an attempt to fully capture the vertical load distribution, and to investigate the effect and variability of support conditions in a group of crossties. Additionally, three consecutive rail seats on the opposite rail (located on the far rail, rail seats 2F through 4F) were selected to provide further information on load transfer, and to examine the variability of support conditions across a single crosstie.

Although the rail pad assemblies and insulators were replaced prior to field experimentation, the clips were not. At the time of experimentation, the clips had been subjected to 5 million gross tons (MGT) of traffic and 3 cycles of removal and reapplication. It was hypothesized that the wear of the clips, especially due to the 3 reapplications, significantly reduced the applied toe load. This reduced the ability
of the fastening system to resist rail rotation under lateral load, which may affect the distribution of rail seat loads.

LABORATORY INSTRUMENTATION
Laboratory experimentation was performed at the Research and Innovation Laboratory (RAIL) at Schnabel, a facility owned by UIUC, operated by the Rail Transportation and Engineering Center (RailTEC), and used for conducting research on railroad infrastructure systems and components. Experiments were conducted using the Track Loading System (TLS), a loading frame which can accommodate a 22 foot (6.7 m) section of track with full depth substructure, shown in Figure 4. A track segment of 11 concrete crossties with Safelok I shoulders was constructed to Class I specifications and instrumented to replicate field experimentation. Figure 5 shows a plan view of the MBTSS installation on the TLS. Five consecutive rail seats on the TLS were instrumented with MBTSS to fully capture the vertical load distribution.
During laboratory experimentation, any clips that were removed were replaced with new clips to maintain an unworn condition. Although other sources of variation exist between the TLS and the RTT, it is believed that the health of the fastening system had the greatest effect of the possible variables between laboratory and field rail seat load distribution results for identical loading environments.

During data analysis, rail seats 3 and 5 were observed to have significantly lower rail seat loads than those recorded on rail seats 1, 2, and 4. Although the entire track structure was tamped using a pneumatic hand tamper prior to experimentation, gaps measuring several millimeters in height had developed under rail seats 3 and 5, resulting in poor load transfer at the cross-tie-ballast interface. This resulted in significant skewing of the data when results from all five rail seats were averaged, leading to an underestimation of the rail seat pressures. The data from rail seats 3 and 5 were therefore excluded from further analysis; the analysis subsequently presented in this paper is based on averaged data from rail seats 1, 2, and 4, the properly-supported cross-ties.

**EXPERIMENTATION LOADING ENVIRONMENT**

The application of loads during field experimentation was accomplished using the Track Loading Vehicle (TLV). The TLV is owned by the Association of American Railroads (AAR) and operated by the Transportation Technology Center, Inc. (TTCI). The TLV can be used to study a variety of scenarios including wheel climb derailments, vertical modulus, lateral track strength, gage widening, and wheel/rail force relationships. An instrumented wheelset is attached to vertically- and laterally-oriented actuators, which are attached to the frame of a modified rail car. The TLV’s ability to apply controlled vertical and lateral loads to the rail using realistic loading conditions and application made it an ideal tool for the purposes of this experimentation.

The application of loads during laboratory experimentation was accomplished using the TLS. For this experimentation, vertical loads were applied using two hydraulic actuators, and the lateral loads were applied using a single manually-operated hydraulic jack as shown in Figure 4. The vertical loads were applied to both journals of a standard 36 inch (91.4 cm) wheelset through standard journal adaptors, and the lateral loads were applied towards the West (instrumented) Rail.

The testing procedure in both the field and the lab consisted of applying loads to both rails with the loading axle centered above each instrumented cross-tie. Vertical loads were applied to each rail at increasing magnitudes from 0 to 40,000 lbf (178 kN) at 5,000 lbf (22.2 kN) increments. In the field, gauge-widening lateral forces were applied at a 20,000 lbf (88.9 kN) vertical load, resulting in Lateral over Vertical (L/V) force ratios ranging from 0.0 to 0.6 at 0.1 increments, and at a 40,000 lbf (178 kN) vertical load, resulting in L/V force ratios ranging from 0.0 to 0.5 at 0.1 increments, followed by a final increment of 0.05, resulting in a final L/V force ratio of 0.55. In the lab, lateral forces were applied at 10,000 lbf (44.5 kN), 20,000 lbf (88.9 kN), 30,000 lbf (133 kN), and 40,000 lbf (178 kN) vertical loads, resulting in L/V force ratios ranging from 0.0 to 0.6 at 0.1 increments at all four vertical loads.

**MODELING RAIL SEAT LOAD ECCENTRICITY**

To better understand the cause of RSD’s signature triangular wear pattern, Volpe modeled the effect of lateral load on the rail seat load distribution. The rail and rail seat are assumed to be infinitely stiff...
bodies, and concepts from the design of building footings are used to describe the change in load distribution as lateral load increases. Volpe considered the eccentricity of the overturning resultant about the center of the rail base, and determined that if the eccentricity is within the middle third of the base (i.e. the absolute value of the eccentricity is less than or equal to one-sixth the width of the rail base), the load distribution is trapezoidal, with the pressure at the gauge side of the rail base calculated by Equation 1, and at the field side calculated by Equation 2:

\[ p_g = \frac{p}{b} \times (1 - \frac{6e}{b}) \]  
\[ p_f = \frac{p}{b} \times (1 + \frac{6e}{b}) \]  

where, \( p_g \) = Pressure on gauge side of rail base
\( p_f \) = Pressure on field side of rail base
\( P \) = Centerline vertical load
\( b \) = Rail base width
\( e \) = Eccentricity, the applied moment divided by the vertical load (M/P)

If the eccentricity is beyond the middle third of the rail base, the distribution is triangular, with the pressure at the field side of the rail base calculated by Equation 3:

\[ p_f = \frac{2P}{3 \times \left( \frac{b}{2} - e \right)} \]  

where, \( p_f \) = Pressure on field side of rail base
\( P \) = Centerline vertical load
\( b \) = Rail base width
\( e \) = Eccentricity, the applied moment divided by the vertical load (M/P) (9)

The expression for eccentricity can be redefined in terms of the applied lateral and vertical loads and the lever arms with which they act, relative to the centerline of the rail base. As described above, the critical eccentricity at which the gauge side of the rail seat becomes unloaded is one-sixth the width of the rail base. It is assumed that when critical eccentricity is achieved, both the lateral and vertical loads are applied directly above the rail seat at the gauge face of the rail. Therefore, at critical eccentricity the two expressions are equal, as shown in Equation 4:

\[ e_{crit} = \frac{b}{6} = \frac{M}{P} = \frac{L_{crit} \times h_g - V_{crit} \times \frac{w_h}{2}}{V_{crit}} \]  

where, \( e_{crit} \) = Critical eccentricity resulting in triangular load distribution
\( b \) = Rail base width
\( M \) = Applied moment relative to rail base center
\( P \) = Centerline vertical load
\( L_{crit} \) = Critical lateral wheel load
\( V_{crit} \) = Critical vertical wheel load
\( h_g \) = Height from rail base to gauge face
\( w_h \) = Width of rail head

Although Volpe did not specify an equation to convert lateral load into an overturning moment, they did state that they assumed that the rail pad assembly did not distribute the rail seat load and that the fasteners provided no contribution to the moment (i.e. through clamping force applied to the rail base) (9). Thus, it can be assumed that the only contributions to the moment are the lateral and vertical load eccentricities, as described above. The equation for the critical eccentricity can therefore be rearranged to obtain an expression for the critical L/V force ratio, as shown in Equation 5:
where, \( \frac{L}{V_{\text{crit}}} = \frac{L_{\text{crit}}}{V_{\text{crit}}} = \frac{\frac{b}{2} + \frac{w_h}{2}}{h_g} \) (5)

If the dimensions of a 136RE rail section, the rail size used in both field and laboratory experimentation, are used, the critical L/V force ratio can be calculated as 0.37. At this L/V force ratio, the gauge side of the rail seat will become unloaded, and the load distribution will develop a significant concentration on the field side of the rail seat.

This calculated critical L/V force ratio can be compared to results from field experimentation with MBTSS. All eight instrumented rail seats experience a loss of contact area at a “threshold” L/V ratio. When a 40,000 lbf (178 kN) vertical load is applied, this threshold L/V occurs between 0.3 and 0.4, which agrees with the calculated critical L/V force ratio derived from the Volpe model (Figure 6). However, when the vertical load is reduced to 20,000 lbf (88.9 kN), this threshold L/V occurs between 0.2 and 0.3: the predicted critical L/V now overestimates this threshold (Figure 7).

![FIGURE 6. Loss of Contact Area under 40,000 lbf (178 kN) Vertical Wheel Load](image)
This behavior of the contact area may be a result of the reaction against the lateral load. A higher vertical force would increase the available frictional force at the rail base-rail pad interface, which counter the lateral wheel load. At the critical L/V force ratio, the lateral load may overcome this frictional force, causing the rail to slip to bear against the insulator post and cast-in shoulder. Once this slip occurs, this new bearing point may become a pivot about which the rail can rotate. A lower vertical wheel load would reduce the capacity of the rail pad frictional force, resulting in slip at a lower lateral load.

A second contributing factor could be the location of the wheel/rail contact patch. Below the threshold L/V, the contact patch is located on the head of the rail rather than the gauge face, and only shifts once the lateral load overcomes frictional forces at the wheel/rail interface. Under a 20,000 lbf (88.9 kN) vertical load, these frictional forces would be reduced, meaning that less lateral load is required to cause the wheel to slip laterally. This slip relative to the head of the rail would reduce the lever arm with which the vertical load acts to resist the overturning moment. More detailed analysis of the resisting moments applied by the vertical wheel load and fastening system toe loads could further refine this model.

**CONTACT PRESSURE AND ABRASION**

Researchers at UIUC conducted several representative experiments to examine the effect of abrasion on interactions at the rail pad-rail seat interface. These experiments involved the construction of a bi-axial loading frame which could apply vertical and lateral loads to specimens representing rail pads of varying material and rail seats. To establish a loading regime, they calculated the average pressure on a rail seat, assuming uniform load distribution over the entire rail seat. This value was estimated to be between 400 psi and 1,800 psi (2.76 MPa and 12.41 MPa, respectively), depending on the applied vertical load. Their findings indicated that in all loading cases, a specimen of nylon 6-6 (a typical rail pad material) will produce abrasion given repeated, small-displacement slip at the rail pad-rail seat interface (10).

To compare this assumption to data from field experimentation, we will examine the average change in rail seat pressure observed during field experimentation. It is assumed that each rail seat supports half of the vertical wheel load applied directly above it, and that the other half is distributed across the four crossties closest to the point of loading (i.e., two crossties to either side of the center crosstie). This approximation is derived from both experimental field data and literature on rail seat load magnitudes (6).
Therefore, for a 40,000 lbf (178 kN) vertical wheel load, we can approximate the total rail seat load as 20,000 lbf (88.9 kN), which will remain constant as the L/V force ratio increases.

This analysis will consider three different quantifications of pressure: uniform, average, and maximum. Uniform pressure is calculated by assuming a uniform distribution of the rail seat load across the entire area of the rail seat, the same method as was used to determine input loads for the previously mentioned abrasion tests. Average pressure is calculated by distributing the rail seat load over the observed contact area, the actual portion of the rail seat engaged in load transfer during a given test. Maximum pressure is calculated by determining the conversion factor between the total raw sum recorded by MBTSS and the input load, and then applying this conversion factor to the sensing location with the highest raw sum.

Figure 8 illustrates the change in uniform, average, and maximum pressure due to increasing L/V force ratio at a constant vertical wheel load. By definition, the uniform pressure is unaffected by the change in L/V and remains constant at 556 psi (3.83 MPa), because it assumes that the contact area does not change. Below the previously observed threshold L/V force ratio (between 0.3 and 0.4), average pressure plots very close to the uniform pressure, indicating that the entire rail seat is loaded. Beyond the threshold L/V force ratio, average pressure increases 45% to 817 psi (5.63 MPa), or 47% greater than the uniform pressure. The maximum pressure is considerably higher than both the uniform and average pressures, starting at 1,179 psi (8.13 MPa), or 112% greater than the uniform pressure. It increases to 1,996 psi (13.76 MPa) at 0.5 L/V, a 69% increase from the 0.0 L/V case, or 259% higher than the uniform pressure. Though not illustrated in Figure 8, the location of the maximum pressure trends toward the field side of the rail seat as the L/V ratio increases; this coincides with a shift in the centroid of loading from the center of the rail seat 1.88 in (47.78 mm) toward the field side, or 1.25 in (31.75 mm) from the field side shoulder at 0.5 L/V.

Figure 8 shows that the assumed range of 400 to 1,800 psi (2.76 MPa and 12.41 MPa) used in the abrasion experimentation at UIUC adequately bounds the average pressure exerted on the rail seat under typical loads for North American heavy-axle freight traffic. However, it does not capture the maximum pressures observed at high lateral loads (above 0.4 L/V force ratio). Wear patterns of mild or newly-formed RSD suggest that RSD first develops at these areas of extreme pressure, expanding as the...
loss of material becomes more severe. More detailed analysis of the characteristics of abrasion at higher pressures, perhaps exceeding 2,000 psi (13.79 MPa), and how they differ from the characteristics of lower-pressure abrasion could lead to a better understanding of abrasion as a failure mechanism for RSD.

EFFECT OF FASTENER WEAR

Figure 9 compares the qualitative effect of L/V force ratio under a constant 40,000 lbf (178 kN) vertical load for three separate cases. The first case represents the common design assumption that the rail seat load is distributed uniformly across the entire rail seat. By definition, this distribution is not affected by L/V force ratio. The second case represents a typical rail seat load distribution for a rail seat with new fasteners, as illustrated by data from experimentation on the TLS. Although there is some concentration of load on the field side of the rail seat, the fasteners are able to restrict rail rotation to 0.31 degrees or less. This results in very little change in rail seat load distribution. The final case represents a typical rail seat load distribution for a rail seat with worn fasteners, as illustrated by data from field experimentation on the RTT using the TLV. The ability of the clips to restrict rail rotation is reduced, allowing rail rotations up to 0.52 degrees, which results in significant concentration of the rail seat load along the field side of the rail seat. Further, this excessive rail rotation results in a complete unloading of the gauge side of the rail seat at L/V force ratios above the previously mentioned threshold L/V force ratio of 0.4. Figure 9 also shows the change in pressures exerted on the rail seat: the increased rail rotation in the worn fastener case results in higher pressures than the new fastener case, as illustrated by the accompanying pressure scale.

![Figure 9. Qualitative Effect of L/V Force Ratio on Rail Seat Load Distributions under 40,000 lbf (178 kN) Vertical Wheel Load](image)

Figure 10 illustrates the quantitative effect of L/V force ratio and fastener health on contact area, the area of the rail seat that is engaged in load transfer. The data has been normalized to the contact area seen under a 40,000 lbf (178 kN) vertical and 0 lbf lateral wheel load combination. Therefore, the percent of contact area at 0.0 L/V force ratio describes the effect of vertical load, while the change in percent contact area for each data series describes the effect of L/V force ratio for each vertical load magnitude. The use of new fasteners results in a consistent increase in contact area for all vertical load magnitudes of between 0.58% and 1.75%. It is hypothesized that this increase is due to deformation of the rail pad assembly as the rail rotates under higher L/V force ratios. By contrast, the worn fastener case exhibits a loss of up to 42% of initial contact area once the L/V force ratio exceeds the aforementioned critical
“threshold” value. These data support the hypothesis that the ability of the worn fasteners to restrict rail rotation was reduced, which resulted in the observed lower contact areas under worn fasteners.

In order to examine the effect of fastener wear and loading environment on pressures, it is necessary to determine the total load applied to each rail seat. For the new fasteners investigated in the laboratory, the rail seat load was calculated from internal strain gauges embedded below the crosstie rail seat. For the worn fastener case (data collected in the field), these embedment gauges were not present on rail seats instrumented with MBTSS. It was therefore necessary to estimate the rail seat load directly below the point of loading. The rail seat load in this case was estimated to be half of the vertical wheel load, based on both an extensive literature review (RailTEC 2015) and data acquired from strain gauges used in conjunction with field experimentation.

Figure 11 compares the uniform, average, and maximum pressures for the new and worn fastener cases under a 20,000 lbf (88.9 kN) vertical load, and Figure 12 compares the uniform, average, and maximum pressures for the same cases under a 40,000 lbf (178 kN) vertical load. In both figures, the new fastener average pressures plot within 50% of the theoretical uniform pressure, even under L/V force ratios as high as 0.6. This indicates that almost all of the contact area is utilized in load transfer. The worn fastener average pressures plot close to the theoretical uniform pressure below the aforementioned “threshold” L/V force ratio. Above this critical point, the reduction of contact area increases these pressures by up to 80% of their original value.

The maximum pressures observed for the new fastener case were approximately 325% higher than the theoretical uniform pressure under a 20,000 lbf (88.9 kN) vertical wheel load, experiencing no net change from 0.0 to 0.6 L/V. Under a 40,000 lbf (178 kN) vertical wheel load, the new fastener maximum pressures are inversely related to L/V force ratio, ranging from 211% to 177% higher than the theoretical uniform pressure. By contrast, the maximum pressures observed in the worn fastener case for both vertical wheel load magnitudes exhibited strong positive correlation with L/V force ratio. Again, the magnitude of maximum pressure relative to the theoretical uniform pressure is greater under the 20,000 lbf vertical load, ranging from 285% to 540% greater than the theoretical uniform pressure, than under the 40 kip vertical wheel load, ranging from 155% to 325% greater than the theoretical uniform pressure.
FIGURE 11. Effect of L/V Force Ratio on Rail Seat Pressures (20,000 lbf (88.9 kN) Vertical Wheel Load)

FIGURE 12. Effect of L/V Force Ratio on Rail Seat Pressures (40,000 lbf (178 kN) Vertical Wheel Load)
It is important to note that although none of the observed pressures approach the design compressive strength of the concrete (i.e. 7,000 psi (11)), the increase in pressure will change the characteristics of failure mechanisms associated with RSD (e.g. increased frictional force leading to more severe abrasion). It is hypothesized that RSD first develops in regions of extreme pressure and then spreads as the loss of material becomes more severe. Figures 11 and 12 show a higher maximum pressure for the new fastening system case than was observed in the worn fastener case at low L/V force ratios. During laboratory experimentation, several rail seats exhibited vertical rail seat loads in excess of the assumed 50% vertical wheel load distribution used to calculate rail seat loads for field experimentation. It is hypothesized that this is due primarily to increased rail seat load on the instrumented rail seats resulting from stiffer support conditions relative to adjacent crossties. These higher rail seat loads are hypothesized to be the cause for the initial elevation of the new fastener maximum pressure relative to the worn fastener maximum pressure.

CONCLUSIONS AND FUTURE WORK
The findings of this analysis show that as lateral load increases, the rail seat becomes more irregularly loaded, and the load distribution concentrates on the field side of the rail seat. At a critical L/V force ratio, the rail seat begins to experience a reduction in contact area. The contact area continues to reduce as lateral load is increased beyond the critical L/V force ratio; at 0.5 L/V force ratio it can be seen that the gauge side of the rail seat can be completely unloaded in the presence of worn fasteners. This further concentrates the rail seat load onto the field side, which could explain the shape of the telltale triangular RSD wear pattern.

Although the modeling work performed by Volpe is relatively basic and relies on several worst-case assumptions, when comparing it to the field results presented in this paper, it provides an acceptable initial estimate for the critical L/V force ratio for a loading environment representative of North American heavy-axle load freight traffic. However, at lower vertical wheel loads, the calculated critical L/V tends to overestimate data obtained from field experimentation. With further refinement, the footing model proposed by Volpe may provide a good basis for a mechanistic calculation of the rail seat load. Consideration should be given for the effect of the vertical wheel load and elastic fastener clamping force in resisting the overturning moment created by the lateral wheel load.

The previous research conducted at UIUC provides insight into the causes, effects, and mitigation of abrasion between the rail pad and rail seat. Although the range of pressures chosen for their experimentation sufficiently bound the observed average pressure on a rail seat, it did not capture the maximum pressures observed, occurring at 0.4 L/V force ratio and above in the case of worn fasteners. Further experimentation at these higher pressures may yield more information on the initial formation of RSD. Consideration should be given to conducting similar tests to those already performed, at higher loads and reduced contact areas to simulate these high L/V scenarios.

Data from this experimentation have shown that the health of the fastening system has a significant effect on the rail seat load distribution in concrete crossties. Data collected from laboratory experimentation on a track structure with new fasteners were compared to data from field experimentation under identical loading scenarios on a track structure with fasteners that had been subjected to both 5 MGT of traffic and, more importantly, three fastening system reapplication cycles. This wear on the fasteners resulted in an average reduction of contact area by 40%, an increase in average pressure by 71%, and an increase in maximum pressure by 60%, relative to the performance of new fasteners. Further, it was shown that under the worn fastener case, the portion of the rail seat load distributed within one inch of the field side shoulder was the most sensitive to changes in L/V force ratio, accounting for up to half of the total rail seat load under high L/V force ratios. It is therefore important to consider the effect of fastening system wear when evaluating the long-term performance of concrete crossties and fastening systems. Further experimentation to quantify the effect of traffic on fastening system wear, and therefore rail seat load distribution, would therefore be beneficial to understanding the parameters critical to preventing RSD. The load distribution at the rail seat is critical to four of the five RSD failure mechanisms proposed by researchers at UIUC. Therefore, it is critical to further understand how it is affected by changes in the loading environment and track structure. Further analysis could consider the effects of crosstie support conditions, fastening system type, and RSD-induced wear of the rail seat. These findings may provide guidance in controlling the behavior of the load distribution which, in turn, could mitigate the effects of RSD.
ACKNOWLEDGEMENTS
This research was primarily funded by United States Department of Transportation (US DOT) Federal Railroad Administration (FRA). The first two authors were supported by a research grant from Amsted RPS. Additional support was provided by the National University Rail (NURail) Center, a US DOT-OST Tier 1 University Transportation Center. The published material in this report represents the position of the authors and not necessarily that of US DOT. Generous support and guidance has also been provided from the industry partners of this research: Union Pacific Railroad; BNSF Railway; National Railway Passenger Corporation (Amtrak); Amsted RPS / Amsted Rail, Inc.; GIC Ingeniería y Construcción; Hanson Professional Services, Inc.; and CXT Concrete Ties, Inc., and LB Foster Company. J. Riley Edwards has been supported in part by grants from CN, CSX, Hanson Professional Services, and the George Krambles Transportation Scholarship Fund. For providing direction, advice, and resources, the authors would like to thank Christopher Rapp from Hanson Professional Services, Inc, Mauricio Gutierrez from GIC Ingeniería y Construcción, Professor Jerry Rose and Graduate Research Assistant Jason Stith from the University of Kentucky, and Vince Carrara from Tekscan®, Inc. The authors would also like to thank Marc Killion, Tim Prunkard, and Don Marrow from the University of Illinois at Urbana-Champaign for their assistance in laboratory experimentation, and undergraduate research assistants Douglas Capuder, Zachary Ehlers, Zachary Jenkins, and Daniel Rivi for their assistance in analyzing the data presented in this paper.

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