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EFFECT OF BOLT STRENGTH AND LOADING CONDITIONS ON A 7/8" BOLT WITHIN AN END-OF-CAR ARRANGEMENT

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ABSTRACT

A cause of failure within end-of-car (EOC) arrangements for cushioned cars with F-shank couplers is that of the yoke bolt failing in shear. This mode of EOC failure is of particular concern due to the concealed nature of the bolt not easily allowing for early detection of the onset of failure. To this end, a finite element analysis (FEA) was performed on a 7/8" bolt and F-bracket assembly to determine the stress state developed within the bolt in an effort to understand the potential cause or causes for the bolt failure. Several parameters, including bolt strength, bolt preload (initial torque), and external loading were varied to determine their effects on bolt performance. The subsequent results indicate that both inherent strength and initial preload have a significant effect on whether a bolt can effectively withstand the various external loading conditions encountered in the field. In addition, it is also apparent that some of the simulation loading scenarios analyzed contain the potential to initiate bolt shearing during operation. From these results, some failure mechanism theories are proposed to describe the type of failure encountered by each bolt grade, either ductile or brittle depending on the inherent material properties.

INTRODUCTION AND PROBLEM DESCRIPTION

One cause of failure within end-of-car (EOC) arrangements for cushioned cars with F-shank couplers is that of the yoke bolt failing in shear. This type of failure has the potential to preclude other more serious events such has coupler pullout and the total deterioration of the EOC arrangement. Therefore, it is critical that the failure flow of this arrangement not initiate within the bolt. Since the bolt is concealed during normal operation, early detection of any potential flaws or cracks is difficult for a railway operator. Developing a system that allows for the more visible, less consequential components to fail first is coveted. This analysis is concerned strictly with the bolt, or pin for zero preload, in a determination of how certain characteristic variations may have the ability to eliminate premature bolt shearing in the field and thus allowing a theoretical failure to initiate in a more opportune location.

Utilizing the finite element analysis (FEA) software package ANSYS Structural [1], a 7/8" bolt and F-bracket assembly was analyzed to determine the effect of bolt strength (SAE Grades 2, 5, and 8), bolt preload (torque), and external loading on the response of the assembly, concentrating on the stress distribution developed within the bolt itself. The bolt grade was found to be decisive in allowing the structure to remain in the elastic regime during the various external loading scenarios, as should be intended. In addition, the magnitude of the bolt torque is essential to the final stresses incurred within the bolt. From these two observations it can be shown that the bolt should have a high material strength (preferably Grade 8 or equivalent), while also eliminating any excessive preload (essentially forming a pinned connection).

EXTERNAL LOADING AND GEOMETRY MODEL

For these simulations, three main external loading configurations were considered. The initial loading condition utilized is defined in the AAR Specification S-4021 [2] as a 3,000 lb pull

force and a downward 250 lb vertical force, both applied through the centerline of the trainline support casting as shown in Figure 1. This free body diagram shown in Figure 1 displays the geometric components along with the external and torque loading locations. To examine a proposed increase in the horizontal component of the load, a second loading case involving a 6,000 lb pull force and a downward 250 lb vertical force was also analyzed. The final loading scenario considered the effect of the 6,000 lb force applied at an angle of 45 degrees to the horizontal, thereby having a resultant force of 4,250 lbs horizontally and 4,250 lbs vertically downward. The external loading variations are shown in Figure 2 and are represented by a remote force within the FEA simulation as shown in Figure 3.

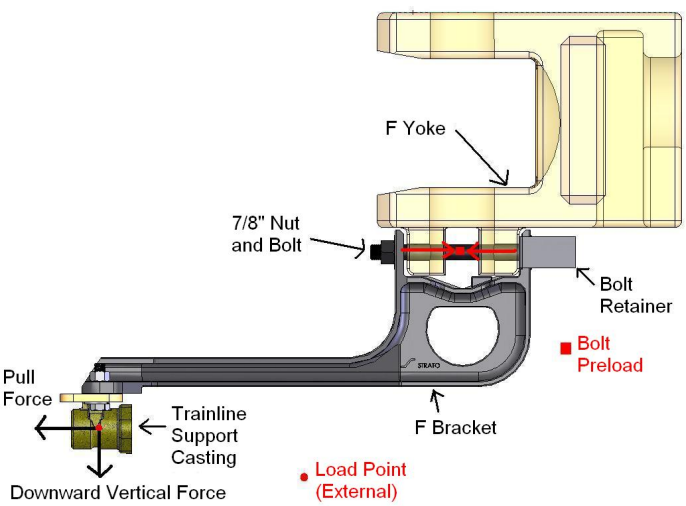

Figure 1. FREE BODY DIAGRAM WITH LOADING LOCATIONS

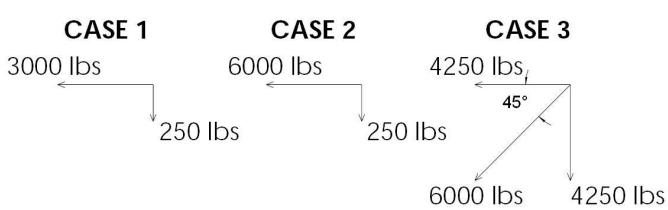

Figure 2. THREE EXTERNAL LOADING CASES

For each of these loading cases, the supports were kept identical to maintain consistency within the FEA simulations. To reduce the computational time and to narrow the focus onto the bracket and bolt interface, the remaining components of the EOC were represented by various boundary conditions and were omitted as shown in Figure 3. For example, the bolt retainer was neglected and the nut and bolt were assumed to behave as a single geometric feature. The interactions between the yoke and the assembly were modeled using a variety of support conditions, including cylindrical supports and compression only supports as shown in Figure 3. To represent the relationship between the bolt and the bracket two types of contact were utilized, a linear "bonded" contact and a nonlinear frictional contact. Between the bolt head and the bracket and the nut and the bracket it was assumed that the contact was bonded, thereby negating any relative motion between these contacting surfaces thus describing a linear contact condition. The bolt shaft and bracket contact was assumed to behave as a frictional contact ($\mu = 0.15$) allowing for a change in contact area. A changing contact area is a nonlinear phenomenom and will require more computer resources within an FEA simulation.

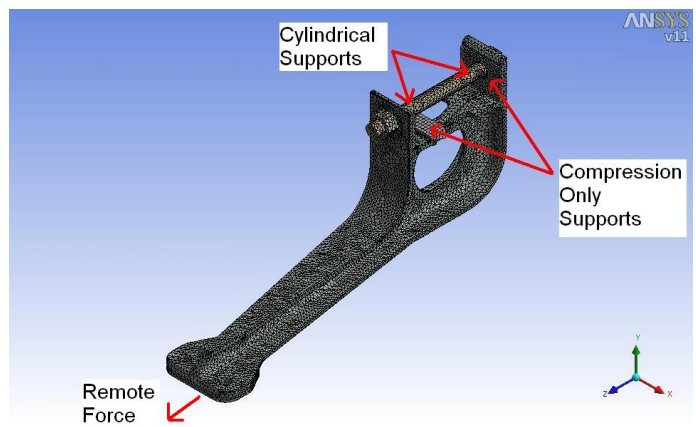

Figure 3. FEA MESH WITH BOUNDARY CONDITIONS

MATERIAL PROPERTIES AND RESULTS CORRELATION

Combining the meshed geometry with the appropriate boundary conditions, the various FEA simulations were analyzed utilizing a Grade 8, a Grade 5, and a Grade 2 bolt. The relevant material properties are shown in Table 1 with the units of pounds per square inch (PSI). The nonlinear material properties of each grade were considered by including a true stress-true strain curve within the software, thus allowing for the consideration of the plasticity effects experienced after material yield. This comparison is shown in Figure 4. In addition to the material nonlinearity, the frictional contact described previously contains inherent nonlinearities concerning the number of elements

Bolt Grade	σ_Y (PSI)	σ_U (PSI)
2	36,000	74,000
5	92,000	137,000
8	130,000	171,000

Table 1. YIELD LIMIT (σ_Y) AND TRUE ULTIMATE STRENGTH (σ_U) OF EACH BOLT GRADE

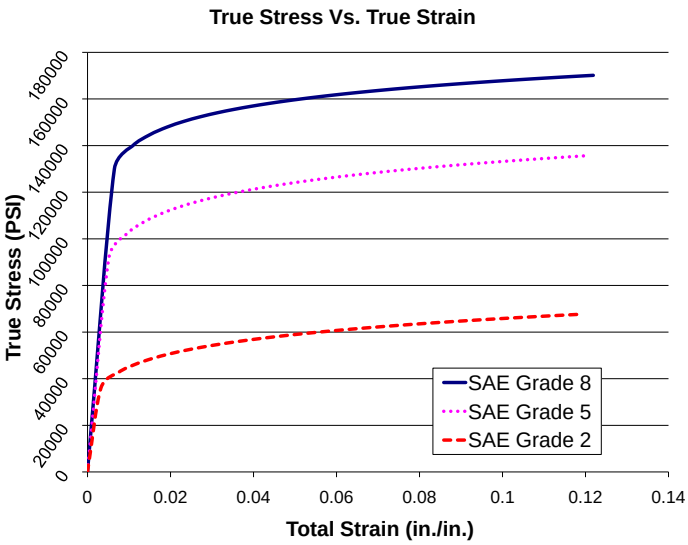

Figure 4. TRUE STRESS VS. TRUE STRAIN FOR EACH BOLT GRADE

that are in contact at any time point. For these particular analyses, expanding the solution beyond the elastic regime is vital in determining how the stress is redistributed once portions of the assembly undergo yielding. Considering the application, these bolts can realistically experience loading that causes permanent deformation within the material thereby necessitating the inclusion of nonlinear behavior within each analysis. To display the correlation between the theoretical results obtained and an actual deformed bolt extracted from the field, a representative contour plot and an image of a deformed bolt are shown in Figure 5. The location of theoretical maximum stresses and the location of actual failure correspond very well. All of the actual failed bolt samples obtained have a deformation pattern similar to the one shown in Figure 5.

BOLT GRADE COMPARISON, PINNED CONNECTION

Initially, the effect of bolt strength (or grade) was studied to develop an understanding of how the various material yield limits affected the overall stress distribution under identical loading

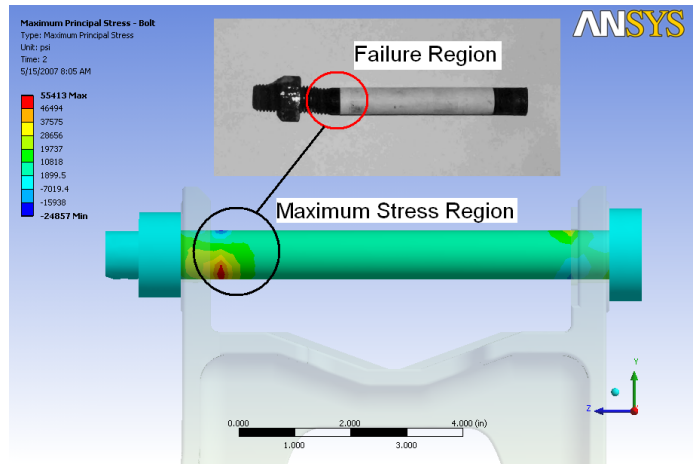

Figure 5. CONSISTENCY BETWEEN THEORETICAL AND ACTUAL RESULTS

conditions. For these first trial simulations, only the three external loading conditions shown in Figure 2 were considered; bolt preload was neglected. Table 2 displays the maximum principal stress (Max. Prin. Stress) and the plastic strain for each bolt grade under the three loading scenarios with no preload, thus modeling the bolted connection as a pinned connection. The yield strength is shown for reference.

Load Case 1 (S-4021: 3000 Pull, -250 Vertical)			
Bolt	Max. Prin. Stress (PSI)	Plastic Strain (in./in.)	Yield Stress (PSI)
Grade 8	37865	0	130000
Grade 5	35842	0	92000
Grade 2	35842	0	36000
Load Case 2 (S-4021: 6000 Pull, -250 Vertical)			
Bolt	Max. Prin. Stress (PSI)	Plastic Strain (in./in.)	Yield Stress (PSI)
Grade 8	89608	0	130000
Grade 5	83293	0	92000
Grade 2	58453	0.0024	36000
Load Case 3 (S-4021: 4250 Pull, -4250 Vertical)			
Bolt	Max. Prin. Stress (PSI)	Plastic Strain (in./in.)	Yield Stress (PSI)
Grade 8	84530	0	130000
Grade 5	85412	0	92000
Grade 2	63912	0.0009	36000

Table 2. BOLT STRENGTH COMPARISON

Figure 6 depicts the table in a graphical form, displaying the nonlinearity of the structure after surpassing the yield limit. From Figure 6 you can see that each bolt grade develops the same maximum tensile stress for each of the three load cases to the left of the yield limit line(dotted). This area is a linear region where the stress state developed in each bolt grade is identical assuming identical external loading conditions. The difference in each bolt grade is then found by analyzing the inherent safety factors associated with the variance in material properties. Above the dotted line, the behavior diverges from a linear pattern and begins to

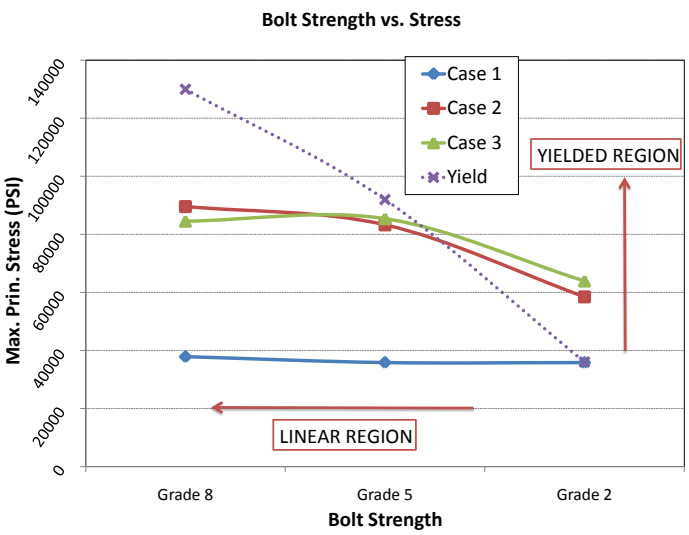

Figure 6. BOLT/PIN STRENGTH VS. MAX. PRIN. STRESS (PINNED CONNECTION)

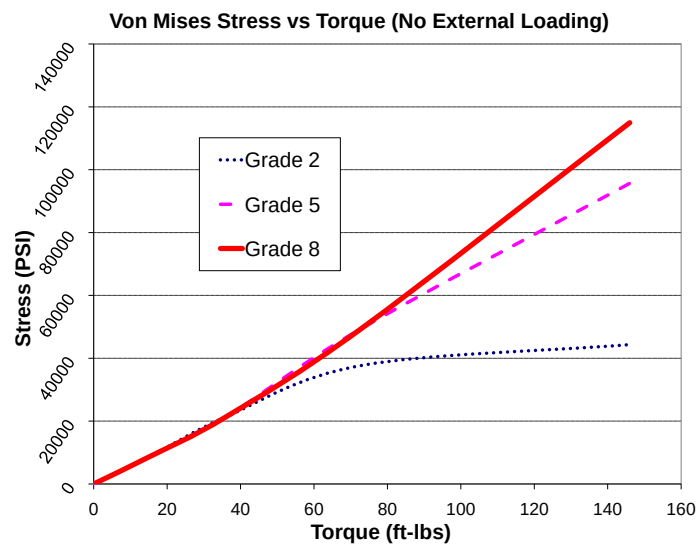

Figure 7. EFFECT OF BOLT STRENGTH AND THE APPLIED TORQUE

behave in a nonlinear manner signifying a stress in excess of the yield. For the pinned connection, you can determine that for all three loading cases, the Grade 8 bolt maintains stress levels well below the yield limit and can be considered in an acceptably safe zone with a safety factor (SF) of approximately 1.5. The Grade 5 bolt stress is in the vicinity of yielding and can have the potential to plastically deform the structure for both loading scenarios greater than that of the current S-4021 standard. For the Grade 2 bolt, even the standard S-4021 loading contains the potential to permanently deform the bolt. These results confirm that external loading in excess of that described in S-4021 requires a Grade 8 bolt to remain securely in the elastic regime.

PRELOAD VARIATION

The next step involved the variation of bolt torque to include the effects of bolt preload on the external loading cases, thereby changing the pinned connection to that of a bolted connection. For the FEA simulation, the bolt preload is a cumulative compressive force applied by the bolt on the bracket thus creating a tensile stress distribution within the bolt. Preload variation was examined for only the first two load cases using the following force values: 1000 lbs, 2500 lbs, 5000 lbs, and 10000 lbs. These correspond to torques of 15 ft-lbs, 36 ft-lbs, 73 ft-lbs, and 146 ft-lbs, respectively, according to the equation $T = K * F_i * d$, with $K = 0.2$, F_i the bolt preload and d the bolt diameter [3]. To display the relationship between bolt preload and bolt strength, Figure 7 plots the effects of applied torque on the maximum Von Mises stress encountered without the inclusion of any external loading.

From Figure 7 the only data set maintaining its linearity is that for the Grade 8 bolt, therefore only this material will remain below the yield limit for all values of applied torque analyzed. The Grade 5 bolt will diverge from a linear relationship at some torque value greater than 73 ft-lbs; using polynomial interpolation this value is approximately 138 ft-lbs of torque. For the Grade 2 bolt, a torque of greater than 36 ft-lbs will cause plastic deformation, with an interpolated value of roughly 70 ft-lbs. Therefore, a Grade 2 bolt will yield if 70 ft-lbs is applied before the assembly is placed in service. For a preload of 138 ft-lbs, the Grade 5 bolt will yield before it is subjected to any external loading. The effects of external loading and bolt preload are shown for each of the bolt grades in the Figures 8, 9, and 10. Only the first two loading cases were considered for this particular analysis as mentioned previously. The bars represent the maximum stress value obtained for each loading scenario. The lines represent the yield and the true ultimate strength of each bolt material in kips per square inch (KSI).

Each of the three charts (Figures 8, 9, 10) display the maximum principal stress plotted versus the applied torque. For each value of torque, three conditions were considered; only preload, load case one, and load case two. The three charts illustrate the loading combinations where the stresses exceed certain thresholds, either the yield limit or the true ultimate strength. The chosen stress values represent the peak tensile stress developed within the material, namely the maximum principal stress. The cut plane where these values are located is shown in Figure 11. As a note, a stress value greater than the true ultimate strength does not necessarily preclude fracture or total failure of the entire structure, only that the stress experienced locally is in excess of the theoretical ultimate limit. In conjunction, a value above the yield does not necessarily imply significant permanent deforma-

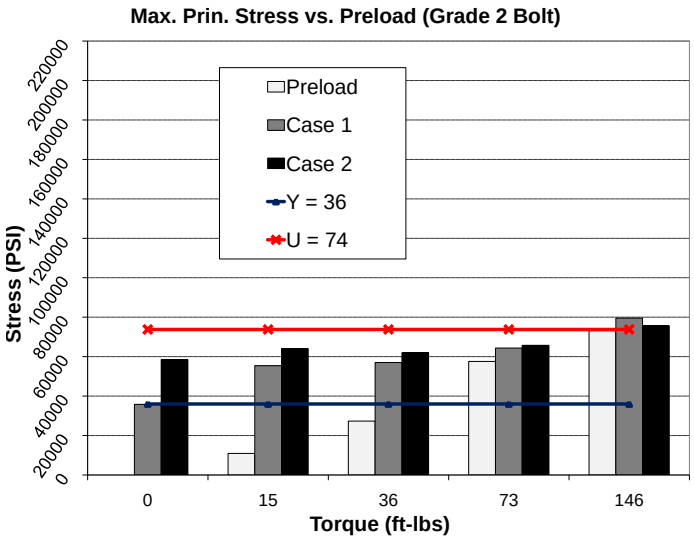

Figure 8. GRADE 2 TORQUE VARIATION

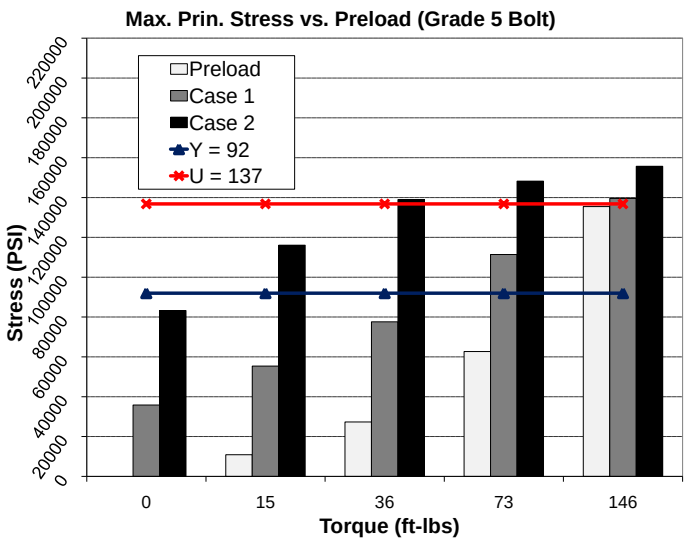

Figure 9. GRADE 5 TORQUE VARIATION

tion leading to reduced functional performance of the structure, it merely illustrates that the stress-strain relationship has entered a nonlinear regime. Another point to consider is that these values are maximums localized at a singular node within the numerical representation of the geometry, which is to say that they represent localized stress peaks and not the nominal stresses formulated throughout the majority of the structure. However, these localized peak stress locations are relevant in that if a failure occurs it will initiate in these areas.

Analyzing each individual chart it is apparent that for both external load cases, even without applied torque, the stress incurred within the Grade 2 bolt will approach the yield limit as

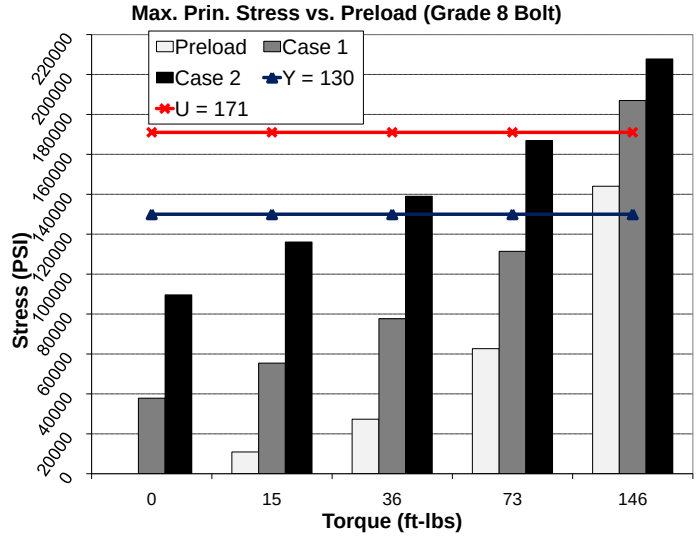

Figure 10. GRADE 8 TORQUE VARIATION

shown in Figure 8. From Figure 9, the Grade 5 bolt will exceed the yield for load case two with a torque in the vicinity of 36 ft-lbs and for load case one with a torque value near 73 ft-lbs. All three bolts experience very high, localized stresses when preloaded with 146 ft-lbs of torque, neglecting any external loading. For each bolt, the effect of this magnitude of torque alone is enough to approach the true ultimate limit and to exceed the yield point of the structure. Therefore, a torque of 146 ft-lbs is large enough to cause permanent deformation of a bolt singularly without the aide of any external loading. These results demonstrate that the magnitude of torque applied during the bracket application can have an effect on the stresses induced within the bolt during operation. For an EOC arrangement, a high compressive clamping force is unnecessary; the bolt's purpose is not to maintain contact pressure between surfaces but rather to act as a support structure for the bracket. Based on these stress results, it would be advantageous to keep the bolt preload to a minimum to prevent unnecessary stresses formulating in the bolt before entering service.

POTENTIAL FAILURE MECHANISMS

Figures 8, 9, and 10 develop some interesting possibilities concerning the failure mechanism of each bolt grade. For the Grade 2 bolt, it is apparent that significant plastic deformation is occurring along the cross section of the shaft. This can be surmised based on the stresses leveling off after a certain value, thus indicating that greater portions of the bolt area are yielding, allowing for the stress to redistribute to a greater percentage of the cross sectional area. This type of behavior indicates an underlying ductility to this particular grade bolt, where the failure of the bolt is not due to a stress above the true ultimate limit caus-

ing a fracture, but rather due to strains exceeding yield causing excessive plastic deformation across a majority of the cross section. This type of failure can be considered a ductile failure (1) where the material continually yields until the deformation is too extensive for effective operation by the component. Upon closer inspection, the deformation results agree with this scenario. The following (Figures 12, 13, 14) is a sequence of cross sectional contour results displaying the plastic deformation as a function of bolt torque for load case one, beginning with the onset of plasticity. The cross sectional cut plane is located in the region of maximum stress shown in Figure 11.

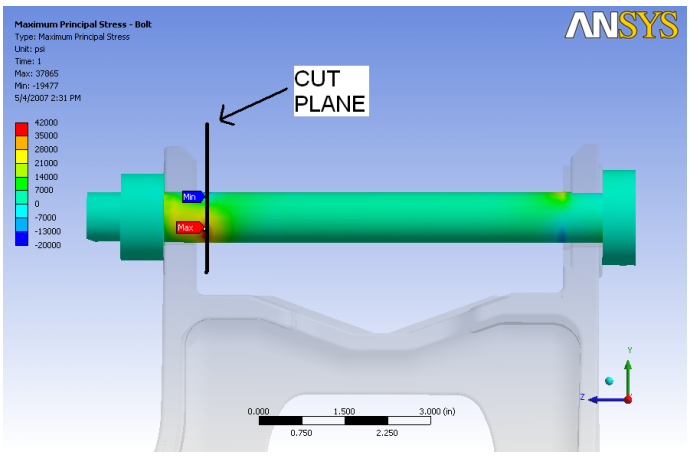

Figure 11. CUT PLANE LOCATION

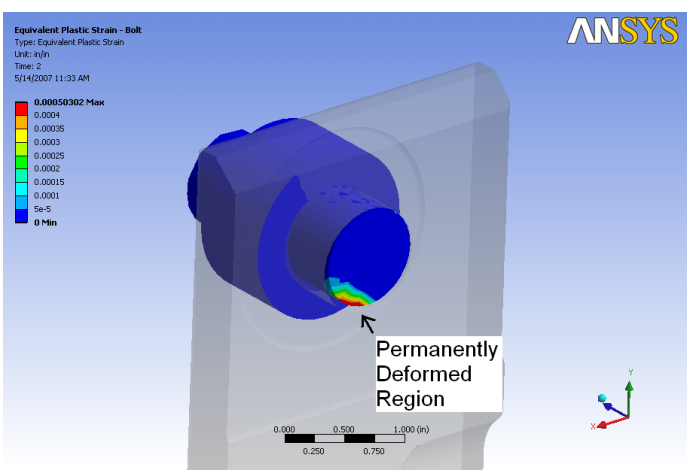

Figure 12. PLASTIC DEFORMATION: TORQUE = 36 FT-LBS (LOAD CASE 1)

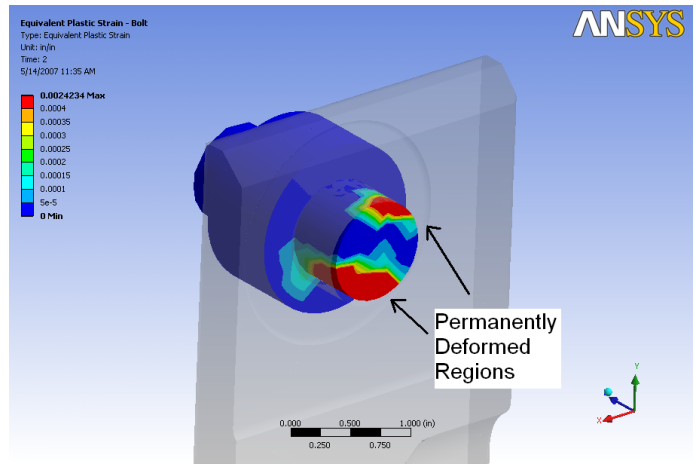

Figure 13. PLASTIC DEFORMATION: TORQUE = 73 FT-LBS (LOAD CASE 1)

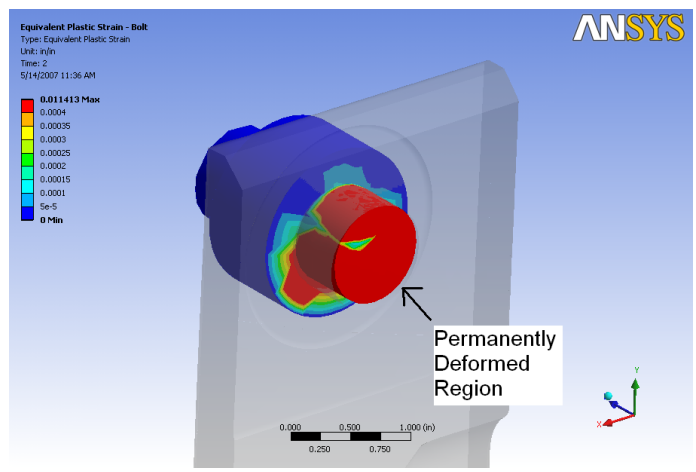

Figure 14. PLASTIC DEFORMATION: TORQUE = 146 FT-LBS (LOAD CASE 1)

The sequence for load case two follows a similar pattern of permanent deformation. As the bolt torque is increased, the percentage of yielded cross sectional area increases. This progressive increase leads to nearly the entire cross section undergoing plastic deformation. A specific criterion can then be used to determine the extent of yielding that can be withstood by a structure before it is considered to have failed. For this example, if the criterion for failure occurred when 50% of the cross section had yielded, then a torque of 73 ft-lbs would cause failure of the bolt.

In contrast to the Grade 2 bolt displaying extensive plastic deformation and therefore exhibiting a ductile failure pattern, the Grade 5 and the Grade 8 bolt display more brittle stress behavior. Rather than the maximum stress leveling off and redistributing, the localized peak values continue to escalate until the maximum

stress has exceeded the true ultimate strength of the material describing a type of brittle failure [4]. These two bolt grades exhibit considerably less permanent deformation, yet their stress values approach their respective failure limits. This type of behavior indicates that perhaps a brittle fracture will cause the eventual failure of the bolt, in contrast to the proposed ductile failure of the Grade 2 bolt. The next chart displays this fact by plotting the amount of permanent deformation for each value of torque in Figure 15.

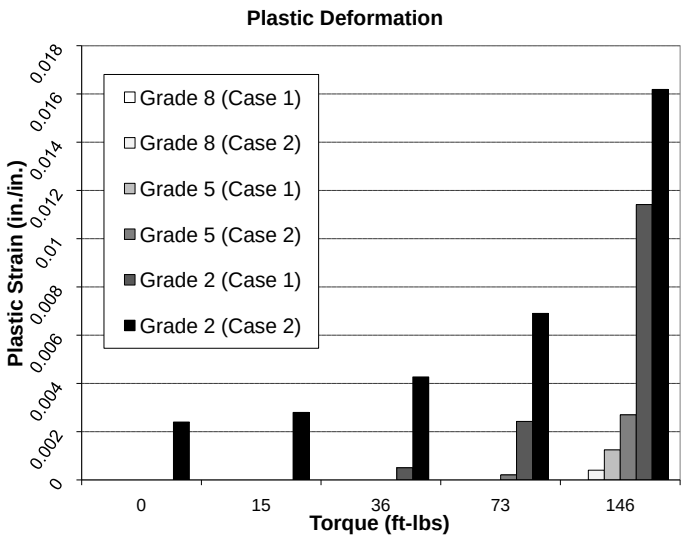

Figure 15. PLASTIC DEFORMATION VS. TORQUE (LOAD CASE 1,2)

As Figure 15 illustrates, only the Grade 2 bolt experiences any significant yielding before the torque reaches a value of 146 ft-lbs, at which point all three bolt grades will exceed the yield limit locally. Due to the high maximum principal stresses experienced, these bolts may fail in a more brittle manner where sudden fracture occurs with little overall yielding. A cross sectional contour plot displaying the maximum principle stress for load case one with a torque of 146 ft-lbs is shown in Figures 16 and 17.

The contour plots shown in Figures 16 and 17 display the localized area of high maximum tensile stress where possible fracture could initiate due to the formation of a dislocation or flaw. The results are similar for load case two, although the stresses incurred in the Grade 5 bolt begin to exceed the ultimate strength when the torque is 36 ft-lbs and for the Grade 8 bolt the stresses are very near the limit when the torque is 73 ft-lbs as opposed to the 146 ft-lbs of torque needed for load case one.

These failure mechanisms are assuming static loading conditions where the failure is caused by a singular loading event. Further expansion of these mechanisms would include analysis

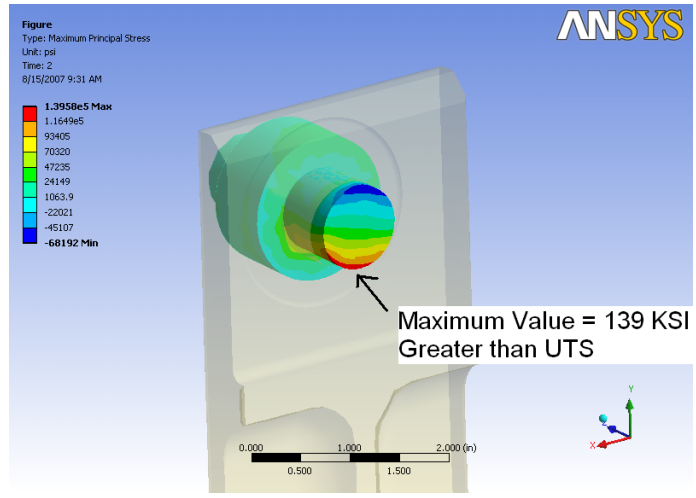

Figure 16. MAXIMUM PRINCIPAL STRESS (GRADE 5)

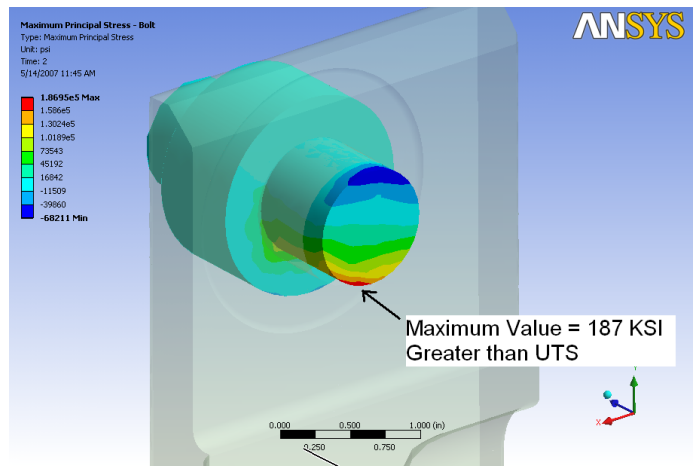

Figure 17. MAXIMUM PRINCIPAL STRESS (GRADE 8)

into how cyclic loading and fatigue affect the stress state developed in the bolt over a period of time. The impact of fatigue damage is a topic for future research.

CONCLUSIONS

The examination of the variation of strength, torque and external loading was performed using ANSYS to determine the stress development within the bolt in a 7/8" bolt and F-bracket assembly. This is critical in the analysis of the mechanisms for bolt shear failure experienced in freight train operation. Understanding how the stresses develop within the bolt is imperative in formulating a system of failure where the more visible components of an EOC arrangement may fail before the bolt experiences a shear failure. From the previous results, it can be shown that both bolt grade and strength, along with the magnitude of

external loading, are prominent in the extent of stresses developed within the bolt. Once these stresses were determined, the propensity for failure and the corresponding failure mode were theorized for the various bolt grades.

For the initial analysis, the bolted connection was treated essentially as a pinned connection by neglecting the preload (torque). This type of loading configuration demonstrates the effects of bolt strength and external loading on the stress distribution. For the three external load cases, only the Grade 8 bolt remains considerably below the yield limit as shown in Figure 6. Although yielding does not preclude bolt failure, it is still desirable to maintain the stresses within the elastic regime. When concerning load case one (S-4021 specification), the Grade 8 bolt maintained the largest safety factor (SF = 3.6). The Grade 5 bolt also remained well below yield with a safety factor of 2.6. In contrast, the Grade 2 bolt was on the threshold of yielding with a safety factor approximately equal to 1.0. Using this as a basis for load case one, both the Grade 5 and the Grade 8 bolt would be acceptable for use assuming a pinned connection. If the loading were increased to model load case two or three, only the Grade 8 bolt remains a viable option.

To include the effects of operator torque applied during installation, the amount of bolt preload was included into load case one and two. The value of torque was systematically increased to determine how the additional loading affects the final stress state within each bolt grade. As shown in Figure 7, the preload induced in the bolt can greatly increase the stress gradient developed, even before external loading is applied. For the Grade 8 bolt to remain below the true ultimate strength, the initial bolt torque would need to remain below 146 ft-lbs for load case one, and at or below 73 ft-lbs for load case two as shown in Figure 10. If a Grade 5 bolt was used for load case one, then the torque would need to remain at or below 73 ft-lbs, shown in Figure 9. Therefore, the initial torque applied to a Grade 8 bolt should not exceed the value of 73 ft-lbs for all loading cases in order to remain below the ultimate strength. If only load case one is considered, than this value of torque can be considered the limit for both the Grade 8 and the Grade 5 bolt.

The analogous stress results were than utilized to determine the theoretical mode of failure for each grade of bolt. It has been proposed that the Grade 2 bolt will fail in a ductile manner due to an excessive percentage of the cross sectional area undergoing plastic deformation, as shown in Figures 12, 13, and 14. This complete yielding of the structure can lead to a loss of functionality or even fracture. For the Grade 5 and Grade 8 bolt, the failure mechanism is theorized as that of a more brittle failure where the structure experiences high localized stresses with little permanent deformation as shown in Figures 16 and 17. The small occurrences of plastic yielding, shown in Figure 15, accompanied by maximum principal stresses exceeding the true ultimate limit are the basis for this assumption.

Overall, the nonlinear behavior of the 7/8" bolt and F-bracket assembly was analyzed to determine how bolt strength and bolt torque variation can impact the ability of the bolt to remain linearly elastic during operational loading conditions experienced in the field. The desire to develop a failure flow such that bolt shear is eliminated is the motivation behind the analysis. Due to this particular application where the bolt is utilized as a support rather than as a clamping mechanism, the standard torques applied to a 7/8" bolt are unnecessary. Therefore eliminating excessive or all bolt pretension would be ideal. This can be accomplished by either eliminating torque altogether with the utilization of a pin connection, or by enforcing a "hand tight" standard on bolt application. Also, the use of higher-grade bolts would be necessary if the external loading exceeded the loading standards developed within S-4021, mainly the ± 3000 pound horizontal force and the ± 250 pound vertical force. Load Case 2 represents this type of external loading.

As a note, since this analysis was performed the AAR has initiated a mandatory replacement of all 7/8" bolts with a Grade 8 pin. With this initiative, excessive bolt preload will be necessarily eliminated from the potential causes of bolt shear within operation. In addition, the bolt/pin material will be guaranteed to have the mechanical properties of the Grade 8 bolt, thus eliminating the installation of potentially inadequate Grade 5 and Grade 2 bolts. The only remaining variable to then consider is that of the external loading conditions experienced during service. If the Grade 8 pin is only subjected to the trainline support forces described in S-4021, than instances of bolt failure will decrease due to this particular component's ability to withstand those load magnitudes. In contrast, if 7/8" bolt/pin failure instances remain unchanged, further analysis will need to be performed to determine whether additional external loading scenarios are causing the failure; additionally, cyclic loading should be examined to determine the correlation bewteen fatigue damage and bolt failure.

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