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HIGH TEMPERATURE TESTING AND ANALYSIS OF A CLASS 300 BOLTED FLANGE JOINT

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ABSTRACT

When class 300 flange bolted joints are held at temperatures in the material creep range, it is documented that the bolt loads can relax. Tests and analysis are being performed with the goal of developing a validated FEA simulation approach to predicting the impact of creep on the bolt loads. The bolt load and gasket geometry are evaluated upon bolt up and after being heated to 1100 deg F. Tests are performed with and without a gasket to separate the impact of the gasket relaxation and flange material creep. The results of the tests and analysis approaches will be presented.

INTRODUCTION

Requests are not uncommon for valves with bolted flange joints rather than butt-weld ends in applications with temperatures in the creep range of materials. Three of the applicable ASME codes have warnings when designing for these temperatures. The ASME B16.34[1] (Valve Design) code states that flanges are expected to relax when in the creep temperature range, resulting in leaks. It further advises to increase wall thickness above the minimum wall requirements. The ASME B31.1[2] (Power Piping Design) code states that the accelerated creep damage should be evaluated. The ASME B16.5[3] (Flange Design) code warns of creep decreasing bolt load and increasing the likelihood of leaking. However, despite these warning, there is no guidance given for the design process to avoid problems with creep.

The Boiler and Pressure Vessel code, Section VIII Division 2 Part 4[4] provides some guidance, with stress limits in the creep range which are based on minimizing creep strain and flange rigidity guidelines. However, the stress limit guidelines are not based on a specific prediction of flange leakage. In particular, the stress criteria do not include consideration of the changes in the gasket. Gasket relaxation and degradation has been studied[5-7] but there is limited data for use in FEA analysis to evaluate the changes in gasket loading and thickness as a function of time at a range of temperatures in the creep range. One study focuses on the creep relaxation of the flange material.[8]

The goals of this work are to (1) evaluate the impact of the gasket behavior on bolt load in the creep temperature range and (2) develop a validated FEA simulation tool for predicting creep relaxation of a gasketed bolted flange joint between a valve body and a pipe. This will be accomplished with a sequence of tests where a bolted flange joint is heated to 1100°F and the behavior of the bolts evaluated.

FEA SIMULATION

The FEA simulation is performed with the model shown in Figure 1. This is an NPS12 class 300 valve. For simplicity, a quarter model is used, while a half model, including the portion of the valve to the left hand side of the valve portion shown, will be needed for any final simulations so that any interaction between the inlet and outlet geometry is considered. The valve and pipe were WC9 and the bolts were Inconel.

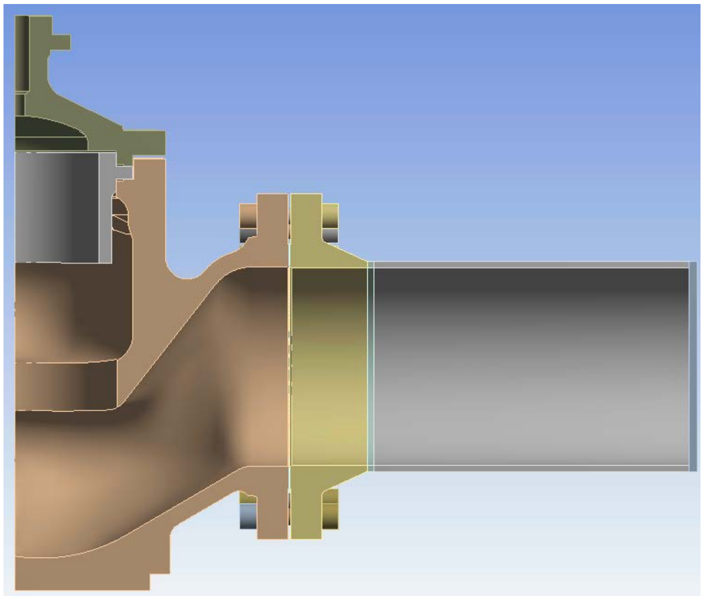

Figure 1: Valve and pipe geometry.

The bolts are modeled explicitly using a pre-tension tool, so that thermal expansion of the bolt material can be incorporated into the simulation. At each time increment in the solution, the bolt pre-tension tool is used to evaluate the bolt load. For the results shown here, the creep of the bolt was not modeled.

A Norton model was used for the creep in the valve and pipe line material:

$$\dot{\varepsilon}_{CR} = C_1 \sigma^{C_2} \tag{1}$$

where $\dot{\varepsilon}_{CR}$ is the creep strain rate, σ is the stress, and C_1 and C_2 are constants. The constants are determined by fitting available data[9] to Eq. 1. These are done at three temperatures, and then the model is interpolated for temperatures between those for which values are available.

The modulus of elasticity, Poison's ratio, and coefficient of thermal expansion are taken from the Boiler and Pressure Vessel code. The bolting material, as well as the valve and pipe material are represented with bi-linear elastic curves, with a slope beyond yield of 2% of the modulus of elasticity.

The simulation is run in five steps (1) assembly bolt and gasket loads at room temperature, (2) pressure at room temperature, (3) raise temperature to 1100^{o} F, (4) creep for 100 hours, and (5) cool to room temperature in 30 hours. Large deflections are allowed. Creep is only active for steps 4 and 5.

The simulations have been performed for a valve body and pipe made from WC9 material and Inconel bolts. Figure 2 shows the equivalent stress at 1100^{o} F before creep is started. Figure 3 shows the equivalent stress after 100 hours of simulated creep. The stress in the valve body, as well as in the pipe have relaxed significantly: a factor of 2 in the valve body and a factor of 1.6 in the pipe.

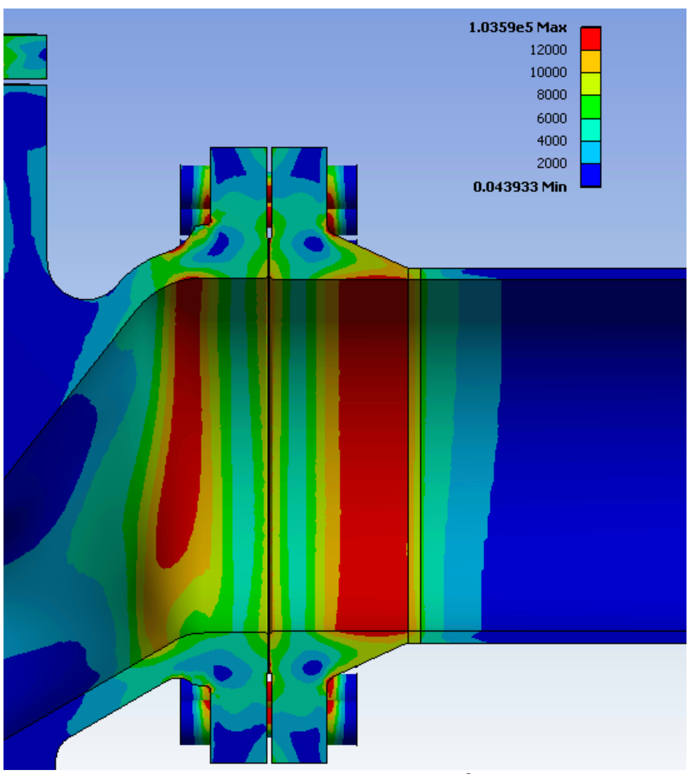

Figure 2: Equivalent stress at 1100^{o} F (before creep)

The bolt loads are initially 33,180 lb. At 1100^{o} F they decrease to an average value of 11,816 lb, then after 100 hours of creep they decrease to an average of 9,066 lb. Once cooled the bolt load is 26,305 lb, a loss of 26% of the original bolt load because of the plastic creep strain. Figure 4 shows the bolt load during the 100 hours of creep.

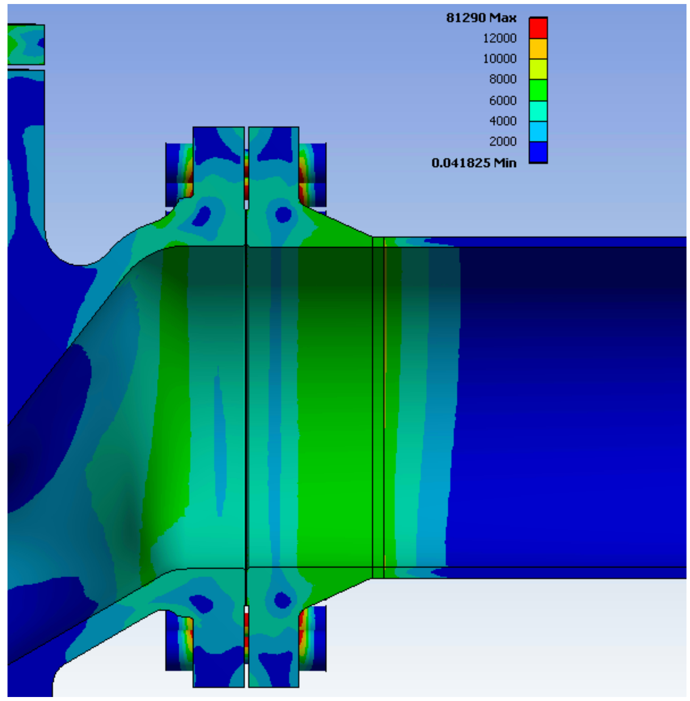

Figure 3: Equivalent Stress after 100 hours of creep at 1100^{o} F.

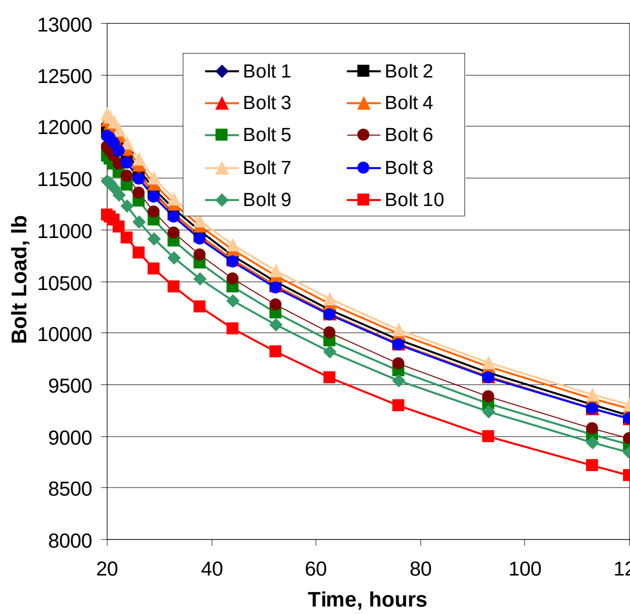

Figure 4: Bolt load during 100 hours of creep at 1100^{o} F.

These simulations are performed without a gasket because a well defined gasket creep model is not available. Given published examples of the permanent strain in gaskets at high temperatures,[5-7] one could expect the loss in bolt load to be greater than that estimated here.

TEST DESIGN

The goal of the testing is to validate the FEA simulation model with particular focus on the creep model and the influence of the flange gasket. During the test, the elongation of the bolts and deformation of the flanges are measured.

Figure 5 shows a schematic of the NPS12 test body which consists of a cast portion and a forged portion. The forged flange is welded to a short pipe section. A cast portion is used because the creep in the cast material is expected to differ from the forged material. The forged flange and the pipe are welded with a butt weld. The forged flange is designed to meet the rigidity requirements of the Boiler and Pressure Vessel code, which results in a longer transition length than required by the B16.5 Flange code. Figure 6 shows a comparison of the transitions from the flanges. The forged flange has a longer transition compared to the minimum allowed by B16.5.

The structure is not pressurized, because from FEA analysis, the bolt loads dominate the stress near the flange. Thus the test was simplified by not including the requirement to maintain a pressure seal during the high temperature testing. The casting, forging, and pipe are constructed from WC9. The spiral wound gasket has Inconel 600 windings with a Thermiculite filler.

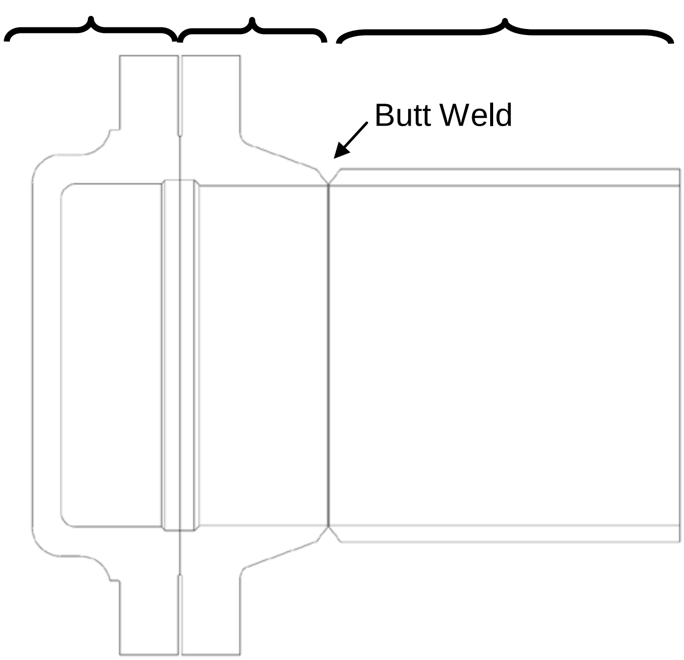

Figure 5: Schematic of the test specimen.

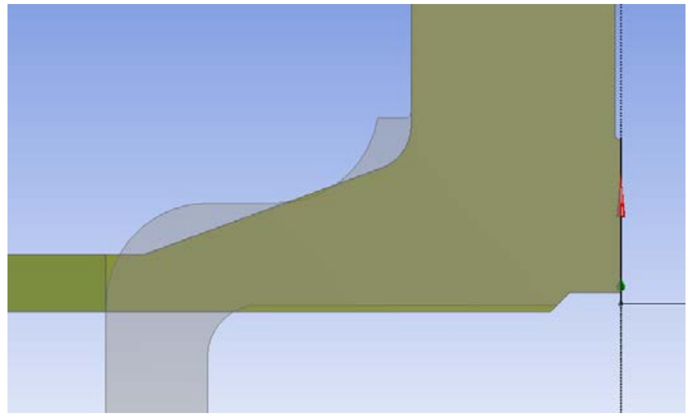

Figure 6: Comparison of the two flange transitions. Grey is the cast side.

Four samples were constructed for tests with two bolting and two gasket configurations. The four test configurations are: (1) Inconel Bolting NO Gasket, (2) Inconel Bolting WITH Gasket, (3) B16 Bolting NO Gasket, (4) B16 Bolting WITH Gasket. These are referred to as assemblies 1,2,3 and 4 respectively. The assumption is that the Inconel bolts will experience small amounts of creep compared to the B16 bolts.

Assemblies 3 and 4 will have high temperature strain gages applied in the region of high stress near the flange to give additional data points for comparison to the FEA simulations.

The measurements made are: (1) bolt length and (2) flange flatness and rotation. These measurements are performed with the structure at room temperature for the following cases (1) assembly with 30 lbs of bolt load, (2) assembly bolt load, (3) after 100 hours of creep testing, (4) unloading the bolts to 30 lbs and (5) after disassembling the structure. Measurement of the bolt length after disassembly will allow separation of the bolt length change caused by a decreased bolt load because of flange and gasket creep and bolt material creep.

A coordinate measuring machine was used to measure several critical dimensions of each assembly, the bolt length and the slope of each flange surface. The slope was determined from measurements taken on the flange between bolts at several radial positions.

The bolts are tightened using a tensioning system and performed in a star pattern with the four passes:

- Pass 1: hand tight (not to exceed 20% of final load)
- Pass 2: 20%-30% of final load
- Pass 2: 50%-70% of final load
- Pass 3: 100% of final load

Several options have been considered for measuring the bolt load. Ultrasonic systems were evaluated, but rejected because the measured values were based on an estimate of the wave speed in the bolt. However, the wave speed is dependent on creep. Thus after creep testing, the ultrasonic system would need to be recalibrated with the bolts after they creep. One recommended practice to measure bolt length is to drill the ends of the stud and then glue in ball bearings. This gives a very consistent surface for the micrometer. However, because of the high temperatures, the ball bearings were attached to the ends of the micrometer, and holes were machined into the ends of each bolt.

The test sample was suspended in the furnace from the open pipe end, so that there is no bias included in the results by supporting the weight on a portion of either flange. Figure 6 shows a picture of the test setup with assembly 1 suspended in the kiln.

The temperature used to control the kilns was measured on the assemblies, near to where strain gages will be attached. This is done to ensure that the material around the strain gages does not heat up too fast and possibly causing the gages to become unbonded. The assemblies were heated to 1100^{o} F in approximately 50 hours, held at that temperature for 200 hours and then removed from the kiln. The bodies cooled to room temperature within 48 hours. The temperatures on the bodies were logged so that the same temperature time profile could be used in the FEA simulations.

Figure 6: Assembly 1 suspended in the kiln before heating.

RESULTS

Assemblies 1 and 2 have been tested to date. Data continues to be processed. Available results are presented.

Figure 7 shows the percentage change in bolt load after heating based on the bolt length measurement. The values for all 16 bolts were measured. Given the resolution and repeatability of the length measurements a variability of ±5% is assumed for the bolt load. On average, Assembly 1 lost 82% of the initial bolt load and Assembly 2 lost 81% of the initial bolt load, however, there are more bolts that lost more load with Assembly 2. Based on these measurements alone, the impact of the gasket is not clearly quantifiable.

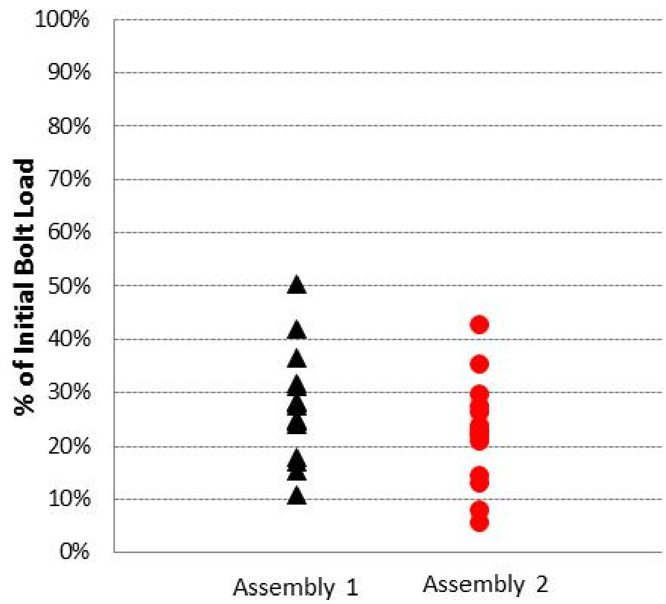

Figure 7: Percentage of initial bolt load after heating for Assemblies 1 and 2.

Figure 8 shows the CMM measured deflection of the flange faces. The deflection was offset to set zero at the inner most points. This data shows that the flange surface is flat before heating and loading (PRE Heat UNLOADED) and then the flange angle changes with loading (PRE Heat LOADED). After heating, the flange angle has increases (POST Heat LOADED) and then when the load is removed (POST Heat UNLOADED) there is permanent strain in the flange angle, returning to near the angle of the fully loaded flange.

The data of the flange surface was converted to an angle by a linear regression curve fit to each data set. Figure 9 shows the angle between the flange faces between two of the sixteen bolts. These results show the flanges rotate upon loading before heat, the angle increasing with heating, and the angle decreases after unloading the bolts. However, after heating and unloading the bolts, the flange faces do not fully return to the original angle, showing a permanent creep strain. The lower flange rotation for Assembly 2, suggests that the bolt relaxation with the gasket includes flange rotation as well as gasket thickness change, whereas in the case of the metal to metal contact of Assembly 1, the rotation is the primary source of bolt load loss. However, the data in Figure 9 is for the surface between two bolts. There are fifteen other surfaces that need to be processed, and given the spread in the bolt load results in Figure 7, there is expected to be significant spread in the measured flange rotation angles. Thus, making more definitive conclusions regarding the impact of the gasket requires further analysis of the data.

FEA simulations were performed on the test body. Given the symmetry of the body, only a slice of the geometry from midway between two bolts and through a half of one bolt was simulated, Figure 10. The Norton creep model discussed earlier was applied to the structure and no creep model was applied to the bolts. The temperature profile from the test was used and creep was turned on in the model at 800^{o}F. Thermal expansion was included. Figure 11 shows the simulated bolt load as a function of time. The blue curve shows the temperature and black curve the bolt load. Zero time is set for when the sample reached 1100^{o}F. The simulation was run until the samples reached room temperature. The measured change in bolt load is included with the red triangles. The simulations predict loosing 45% of the bolt load, which is significantly less than the measured loss.

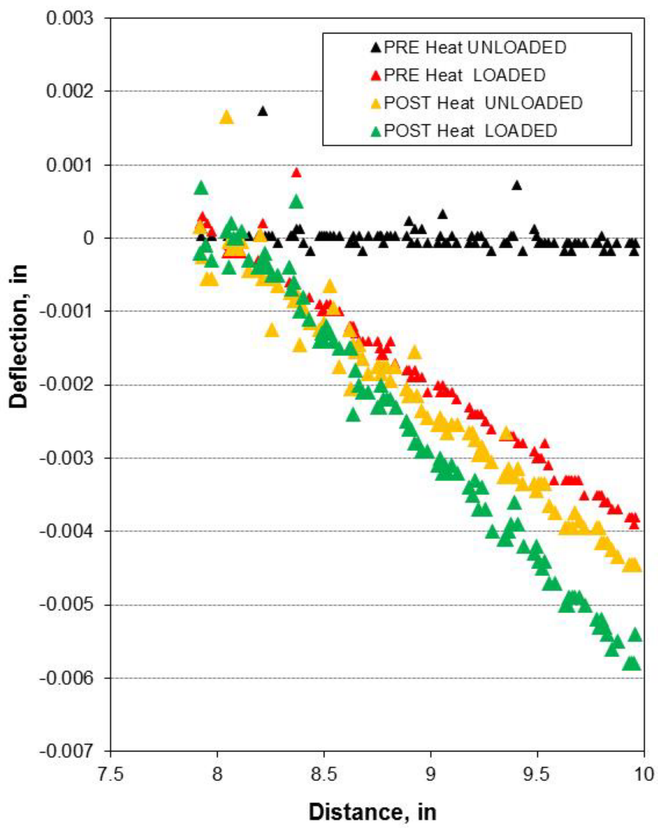

Figure 8: Deflection of one of the flanges in Assembly 1.

SUMMARY AND CONCLUSIONS

While the codes for valve and flange designs provide warnings regarding the impact of creep on flange joint behavior, they do not provide design guidelines. Similarly, since the BPV code does not provide specific models for creep analysis, a validated creep analysis of a bolted flange joint needs to be developed. The work presented here provides test data showing the relaxation of the bolt load when holding a joint at 1100^{o} F for 200 hours. The test data clearly shows the flange rotation once loaded and then the increased rotation after heating the sample into the creep range. The results also show that the flanges do not relax back to their original rotation angle after releasing the bolt load, showing the creep strain in the region of the flange. Thus the bolt load is correlated to permanent rotation of the flanges caused by creep.

There is also evidence that with the gasket the bolt load loss is a combination of the flange rotation and additional gasket compression. However, further analysis of the data is needed to confirm and quantify this.

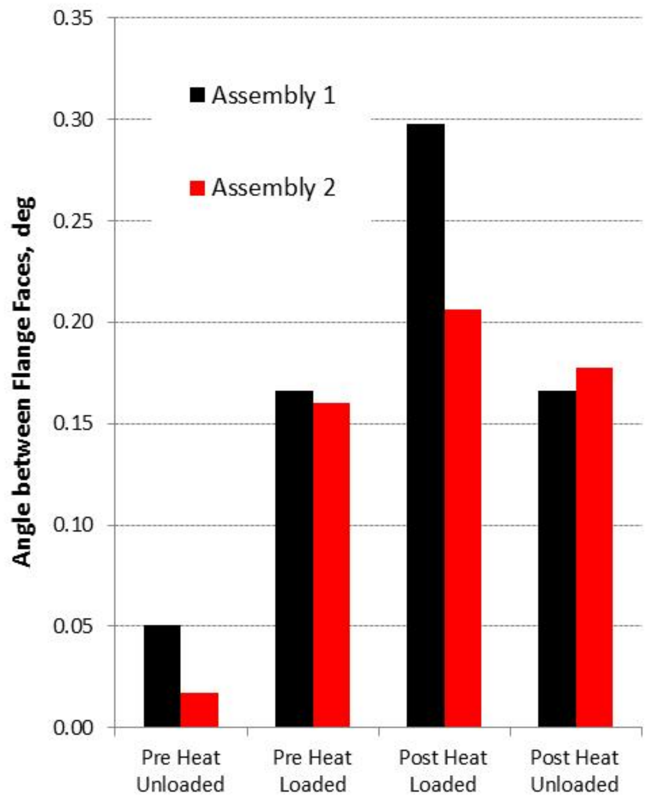

Figure 9: Flange rotation angle for Assemblies 1 and 2.

The tests were completed for Inconel bolts. Data from these tests continue to be processed for the flange rotation angle between each bolt and to evaluate creep in the bolts themselves. Tests are also underway for two identical assemblies, but with B16 bolting.

The FEA simulations using a Norton creep model underestimate the amount of bolt load loss that was measured. Additional work is needed to refine the creep model, possibly adjusting the creep model coefficients for the material, in particular for the cast side of the assembly, and to evaluate the inclusion of bolt creep.

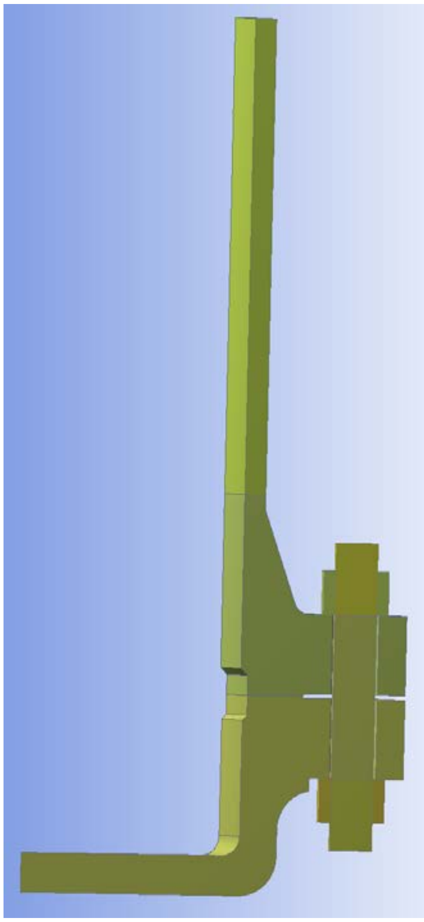

Figure 10: Geometry for the FEA simulation of tests.

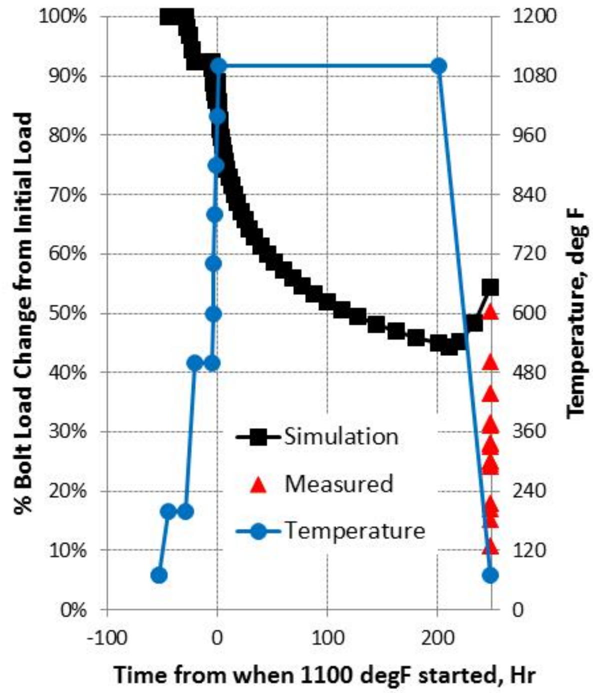

Figure 11: Simulated percent change in bolt load relative to the initial load. Measured bolt load change for Assembly 1 is included.

ACKNOWLEDGMENTS

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