STP-PT-036

BOLTED FLANGED CONNECTIONS IN ELEVATED TEMPERATURE SERVICE

ASME STANDARDS TECHNOLOGY, LLC

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FOREWORD

The early research in design and analysis of bolted joints was conducted in the 1930s and 1940s and this work led to flanged joint design rules, such as the ASME Section VIII, Division 1, Appendix 2 method that was introduced in the 1940s and has remained largely unchanged since that time. The need for improvement in the design of high temperature flanged joints was identified to ASME and this project was funded by ASME to examine the requirements for high temperature in the flange material creep range flange design.

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ABSTRACT

The intent of the project is to examine the requirements for high temperature flange design and provide guidance for inclusion of design methods into the modern ASME pressure vessel design codes. While the fundamentals of high temperature flange design using code equations were included in the assessment, the initial starting point for the project was to formulate guidelines for FEA of the creep problem, based on comparison with relatively scarce flange creep test data. A literature research was conducted to review the fundamental study in high temperature flange joints, especially with respect to papers including experimental verification of results. In addition, the subject of gasket creep behavior was examined.

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1 INTRODUCTION

The early research in design and analysis of bolted joints was conducted in the 1930s and 1940s; this work led to flanged joint design rules, such as the ASME Section VIII, Division 1, Appendix 2 method that was introduced in the 1940s and has remained largely unchanged since that time. Other international methods of design have been introduced recently, most notably the CEN EN-13555 method. However, none of the current methods address design of a bolted joint in the creep range. The requirement for the design of high temperature joints was identified during the initial development of the design methods, but unfortunately a concise design method was never documented in a code or standard. This is somewhat understandable, given the myriad of complexities involved with analyzing the significance of creep on a bolted joint. In fact, the present design methods are also inadequate even when addressing low temperature operation that involves creep and relaxation of the components [1]. Unfortunately, even with the more powerful analysis methods available today, the researchers of the 1930s actually appeared to be closer to resolving high temperature flange design than more recent research efforts.

In part, this lack of advances in the design of high temperature flanges is probably due to the fact that most flanges do not operate in the creep range for the materials of construction and therefore the vast majority of flanges have given admirable service. In addition, a creep “failure” of a flange is most likely to be a relatively small leak which is easily rectified by re-tightening the bolts during operation. Such “failure” does not often warrant management attention and therefore does not garner industry attention as an issue requiring resolution. It may also be easily demonstrated that industry, as a whole, has learned to accept bolted joint leakage [2] and therefore relatively little effort has been directed towards reducing the frequency of joint leakage. The need for improvement in the design of high temperature flanged joints was identified to ASME and this project was funded by ASME, starting in August 2007, to examine the requirements for high temperature (in the flange material creep range) flange design.

The intent of the project is to examine the requirements for high temperature flange design and provide guidance for inclusion of design methods into the modern ASME pressure vessel design codes. Throughout the project, it was kept in mind that high temperature flange joints are a relatively small portion of the flange population, and that improvements in Finite Element Analysis (FEA) and computing power are now to the point where very large non-linear creep problems can be solved relatively easily. Therefore, while the fundamentals of high temperature flange design using code equations were included in the assessment, the initial starting point for the project was to formulate guidelines for FEA of the creep problem, based on comparison with relatively scarce flange creep test data. It is recognized that these guidelines may actually be the most appropriate implementation of high temperature flange design, due to the inherently critical nature of most high temperature flanges.

The following literature search looked at fundamental research in high temperature flange joints, especially with respect to papers including experimental verification of results. In addition, the area of the mechanical effects of temperature on bolted flange was included, as any assessment of flange creep must be made at the initial operating stress conditions, rather than at the ambient conditions. In addition, the subject of gasket creep behavior was examined. This subject has had extensive research across a variety of gasket types, but there is very little tie-in with actual high temperature (creep regime) behavior of the bolted joint.

2 LITERATURE RESEARCH

2.1 High Temperature Joint Behavior

The primary driver for initial efforts in high temperature flange design came from improvements in the steam power generation industry that resulted in higher steam temperatures and pressures being used in the late 1920s and early 1930s. The initial work of Baumann [3] in 1930 looked at the creep of bolts and flange components of the joint and proposed a joint "life" relationship that accounted for the relative creep strength and relative flexibility of the bolt and flanges (Figure 1). The concept of joint life was related as a measure of time before flange leakage, rather than flange or bolt mechanical failure. The figure demonstrates the significant effect of the relative flexibilities on the joint life.

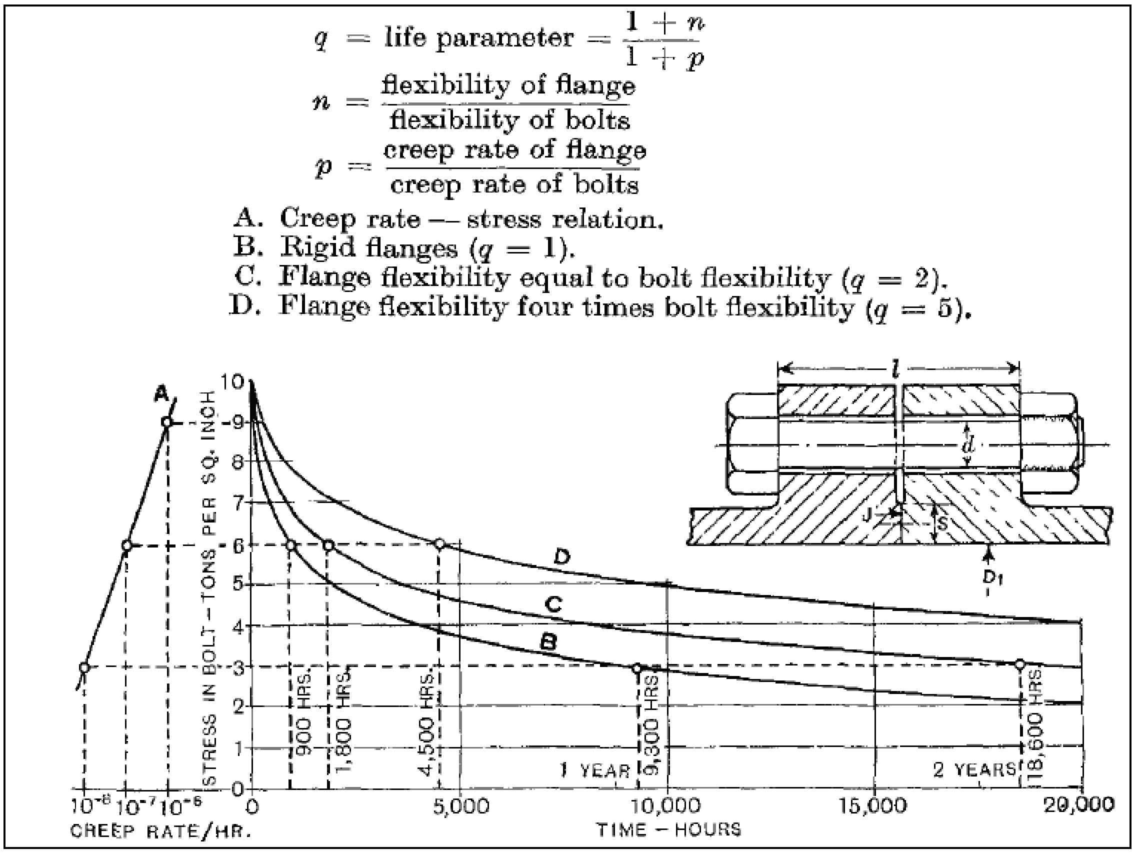

Figure 1 – From Baumann [3], page 1336

Baumann also made several conclusions from his examination of joint creep characteristics, including:

- That a bolt stress that induced a creep strain rate of less than 10^{-8} would give satisfactory life
- That if the steam load alone induced a creep strain rate of greater than 10^{-8}, visible leakage would occur in less than one year
- That flange flexibility is very desirable. A flange with similar flexibility as compared to the bolts will double the creep life of the bolt
- That with a rigid flange the life is independent of the bolt length or flange thickness.

However, as noted by Baumann, his treatment assumed tensile creep in series of the joint components. In the case of the flange, the creep is in bending and therefore the assumptions made by Baumann were conservative by comparison to the real case.

The creep of flange joints was further studied by Bailey [4] in 1935 and the concept of stress relaxation of components in series was developed further using a creep life relationship under a tensile diminishing stress (Figure 2). By using this relationship and the methods outlined in Appendix I of the paper, it is possible to determine the remaining stress (f) in each joint component from an initial stress (f_0) for a given time period (t). However, once again this assumed that the bolt and flange creep were identical in nature, which is not the case, as the flange is predominantly in bending and the bolt predominantly in tension.

Taking the law $C = Af^n$ to apply to the material over a time t in which the stress falls from the initial value f_0 to the value f, application of equation (9), Appendix I, p. 234, to the case of simple tension gives

$$t = \frac{1}{(n-1)EA}\left(\frac{1}{f}\right)^{n-1}$$

since $x_0 = f_0/E$ and $c_0 = Af_0^n$ and $f = \dfrac{1}{[(n-1)EAt]^{\frac{1}{n-1}}}$

Figure 2 – Early Stress Relaxation Relationship from Bailey [4], page 149

One of the interesting points to come from this analysis was that there was little advantage to tightening the joint to a value exceeding twice the final stress at "failure." Similarly to Baumann, failure of the joint was defined as leakage, rather than mechanical rupture. However, this relationship is somewhat misleading due to the limitations regarding the assumptions of similarity of the bolt and flange creep and also, as detailed by Bailey, the fact that the actual value is significantly higher than two, as there is a reduction in bolt and flange stress corresponding to the reduction in material Young's modulus and yield strength with increasing temperature. Therefore, the initial stresses at the start of creep are significantly lower than the initial assembly stress and it is the stress at the start of creep to which the factor must be applied. It was also stated in the paper that the sum of the individual component elastic and creep strains at any given time would be equal to the initial elastic strain imparted by the assembly load. Bailey suggested in the paper that a suitable life for a joint might be 100,000 hrs without maintenance and 10,000 hrs with maintenance (re-tightening).

Bailey continued his work in the field of high temperature bolted joint behavior as chairman of the Institute of Mechanical Engineers Pipe Flanges Research Committee that studied flange creep from 1936 to 1954. The focus of the study was to provide additional flange ratings to the British Standard BS10 for standard piping flanges in order to accommodate the increasing pressures and temperatures associated with steam power generation. The first report presented by the committee [5] detailed tests on a standard BS10: Table T, 8 inch flange at 900°F to 1000°F and 1450 psig steam pressure. The flanges were assembled, heated, steam applied internally and held steady at the operating conditions until noticeable leakage from the joint occurred. The gaskets, when used in these tests, were either metal or a very thin asbestos fiber (1/64 inch thick) and therefore the influence of the gasket on the joint behavior was neglected. The paper includes a diagram illustrating the concept of bolt stress relaxation (Figure 3) where the elastic relaxation of the bolt strain during the test (A-G') is shown relative to the creep strain of the bolt and other joint components. Measurements of the bolt lengths and flange deflections both before and after the tests were used to determine the proportions of the creep strain and elastic strain reduction occurring for the joint being tested. The tests indicated that "Nut Distortion" accounted in general for less than 10% of the overall bolt creep occurring. Nut Distortion was the term used to describe the embedment and localized creep occurring in the nut to flange and nut to bolt thread regions. This value was determined as the residual deformation when the remaining elastic and creep strain at the end of the test was subtracted from the initial assembly strain of the bolt. The paper also recommends a preliminary soak at temperature and re-tightening of the bolts after the soak as a way of increasing the life of the joint due to the fact that the majority of the relaxation occurs in the first portion of the joint life.

Bailey continued his work on high temperature flange behavior by establishing elastic stress and deflection relationships for the joint components based on ring theory in Part I of Bailey [6]. In Part II of the same paper he examined the high temperature behavior and listed a "Life Factor" for the joint (until leakage) that was equal to the ratio of the component sum of the initial elastic strains divided by the sum of the component creep strains, multiplied by the stress relaxation relationship outlined in Figure 2. The method outlined was based on the assumption that the joint life was determined by the relative component deformation (both elastic and creep). That is, once the creep deformation had reduced the elastic deformation by an appreciable amount, leakage would occur. He also found that, depending on the relative flexibility and creep rates of the pipe wall and flange hub, it may either reinforce the flange against creep or actually increase the creep deformation of the flange due to excessive diametrical growth of the vessel wall due to creep. Bailey also assumed that the effect of creep associated with shear stress was negligible and examined the effect of bolt holes in increasing the creep of the flange by testing perforated strips of mild steel at various ratios of drilled hole spacing. Bailey suggested the use of a factor (λ) to adjust for the effect of bolt holes on the strength of the flanges. The factor is a basic multiplier on flange deformation and was found, for standard BS piping flanges, to be approximately 1.17. He points to the need for creep results on flanges with the effect of bolt stress relaxation occurring. Initial results presented on C-0.5Mo material indicated that the 100,000 hr life of a joint corresponded to a similar stress level for a tensile creep test with 0.1% strain rate in 100,000 hrs.

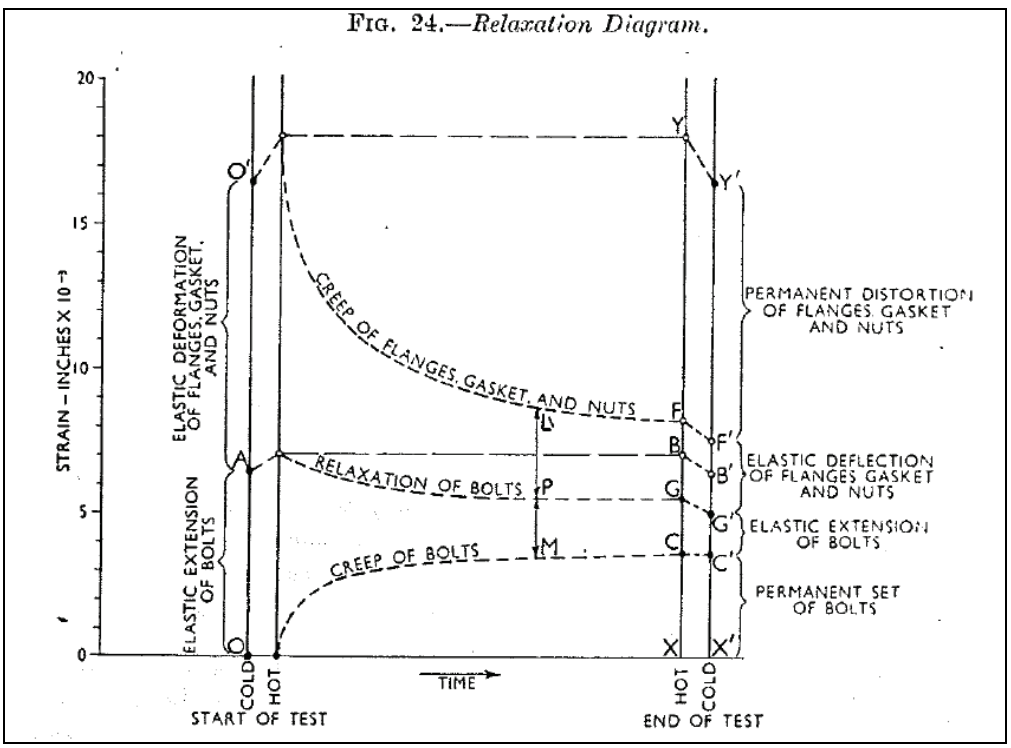

Figure 3 – Relaxation Diagram from Gough [5], Fig. 24, page 263

In a 1938 paper, Waters [7], who was one of the original developers of the methods presently used in the ASME code, developed an assessment of the effect of creep on loose-ring flanges and predicted that, on an allowable stress basis, the fact that the flange was creeping in bending would mean that the stress relaxation occurring at extreme fibers of the ring would result in an improvement in stress distribution (lower peak stresses) and therefore justification could be made that high temperature flanges could, in fact, be thinner than their low temperature counterparts. However, he also noted that it was likely that flange deformation and subsequent leakage would limit the life of the joint and it would be necessary to increase the flange thickness to a point where the deformation during the flange life did not result in leakage. The paper outlines a method of assessing the creep of the joint by finite time increments and accounting for the stress distribution in the flange ring at each time increment. It is suggested that assessment of the flange behavior may be possible by using a combination of tensile and bending creep tests. The paper also makes the same conclusions regarding the sum of the elastic and creep strains as the previous papers. However, similarly to the previous papers, the method neglects strain hardening effects in the proposed creep relationship.

A similar approach is used by Marin [8] to address creep in a loose-ring flange, however, similarly to previous papers, the outlined approach neglects the effects of strain hardening. The second report [9] of the Institute of Mechanical Engineers Pipe Flange Research Committee was released in 1939 and contained substantial additional test results on gaskets, stud bolts and on both actual flanges and on stacked loose rings. The creep tests on a variety of 0.75 in. diameter stud Ni-Cr-Mo and Cr-Mo bolt alloys under constant stress showed that the compression of the 0.125 in. thick mild steel washer was less than 0.001 in. in all cases and that the creep/embedment of the nuts contributed between 10% and 20% of the total bolt creep. The Ni-Cr-Mo bolt materials were also subject to impact test after creep to examine whether the time at temperature had led to embrittlement of the material. The results demonstrated that specimens subject to no stress exhibited as-new fracture toughness, but that stressed specimens subject to creep damage showed significant loss of toughness. In addition to the bolt tests, 0.0156 in. thick and 0.0312 in. thick asbestos gaskets were tested for both creep and leakage. The creep tests showed that there was initial gasket “flow” with significant reduction in gasket load until a temperature of 300°F, but after this temperature, the gasket stabilized and no further significant creep was witnessed.

The flange creep tests were conducted on 8 inch BS10 Table T flanges with alloy bolts. The flanges were tested under similar conditions to the first report tests, but in this case the steam leak rate was measured and “failure” of the joint was indicated when the leak rate reached 150 grams/hour. It should be noted that in these tests some leakage was witnessed from the early stages of the test, so this quantitative measure of “failure” was nominally selected. The tests were conducted at a range of temperatures and the time to leakage “failure” measured for each case. The flanges were retightened and re-tested in order to generate the effect of successive re-tightening on the time to failure. The results of these tests are shown in Figure 4 and it can be seen that the time to failure increases significantly with successive re-tightening of the bolts. In addition, it can be seen that the relationship between life and operating temperature appears adequately represented by a linear (temperature) vs. log(life) relationship, enabling short term tests to be extrapolated to determine the 100,000 hr temperature limit. In this case, the temperature limit for a life of 100,000 hrs was determined to be 835°F.

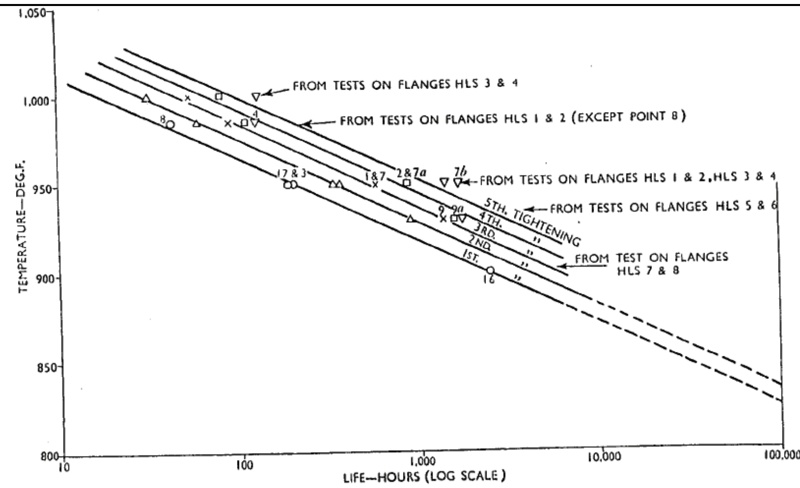

Fig. 11. Life-Temperature Relationships for Successive Tightenings on 8-inch Flanged Joint at 1,400 lb. per sq. in.

Where figures are attached, actual test points are represented.

○ First tightening (virgin material). △ Second tightening. × Third ,, □ Fourth tightening. ▽ Fifth ,,

Figure 4 – Joint Life Diagram from Tapsell [9], Fig. 11, page 448

The tests did not exhibit a significant difference between dismantling the joint versus simply tightening the bolts after leakage for the no gasket and thin gasket cases. In addition, the advantage of the stress relaxation mode of the joint versus a constant stress test (from which allowable stresses in the creep range are determined) is demonstrated by comparison between the tensile creep tests on the bolt material and joint leakage tests (Figure 5). It should be noted that in this test the results are not directly comparable, in that the joint leakage test bolts were initially tightened to a higher stress (21.6 tons/in² vs. 15 tons/in²) and the tests were conducted at 1000°F versus 975°F for the tensile tests. This difference in temperature corresponds to approximately a halving of the creep life in tensile tests. Therefore, the fact that the bolt strain associated with the flange leakage tests is significantly lower than the tensile test demonstrates the advantage of the flange configuration in creep versus the tensile creep case and the overly conservative nature of using allowable stress values based on a tensile creep test. In addition, it can be seen that the successive re-tightening of the flange did not contribute to accelerated creep damage.

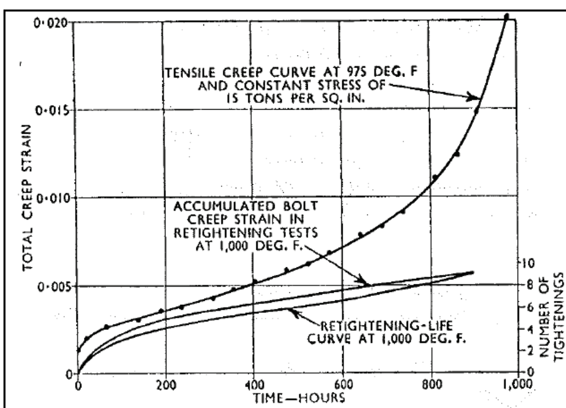

Fig. 13. Comparison between Retightening-Life Curve, Accumulated Bolt Creep-Strain Curve, and Creep-Strain Curve in Tensile Test

Figure 5 – Bolt Creep Comparison from Tapsell [9], Fig. 13, page 450

Unfortunately, after the release of these last papers, the Second World War commenced and advances in the field of high temperature joint design were delayed until follow-up research occurred in the early 1950s. The paper by Kerkhof [10] examined a method of analysis for both ambient temperature and creep design and verified the method versus experimentation on a 44 inch OD flange. Kerkhof proposed a design based on a load factor (α) that was a multiplier placed on the hydrostatic load acting on the joint. He determined that, from the conclusions of Bailey [6], if the initial assembly bolt load was adjusted to account for the relaxation in gasket load, since the stress distribution in the flange improves due to creep, only the ambient temperature design case needed to be assessed. Since the bolt load is adjusted by the load factor (α), the flange design also becomes heavier with increasing temperature, which should compensate for the component relaxation (Figure 6). However, the load factor would change for each joint configuration and so, while this approach appears simple at first glance, it would require extensive testing of different configurations in order to obtain the load factor relationships.

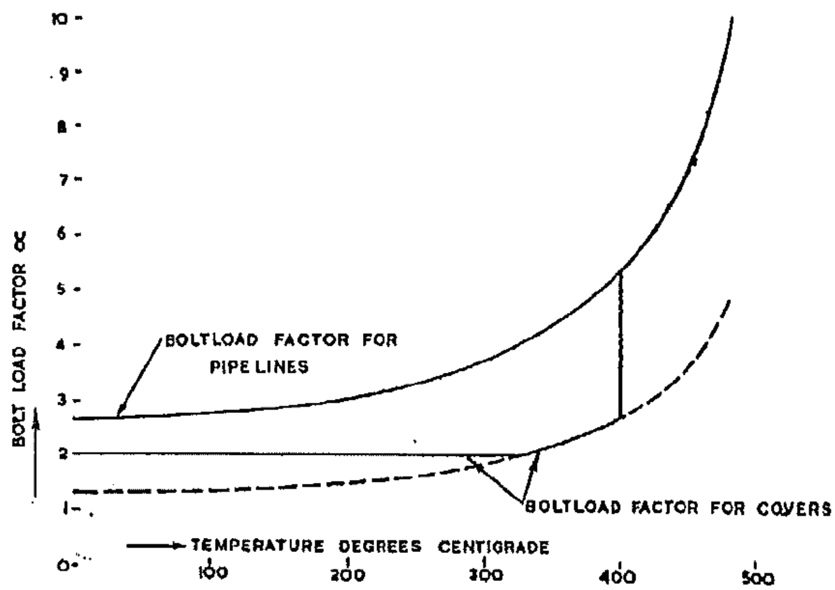

Fig. 2. Curves showing the bolt load factor "α" in relation to the temperature.

Figure 6 – Bolt Load Factor Graph from Kerkhof [10], Fig. 1, page 152

$$t^m = A(1/f^{n-1} - 1/f_0^{n-1}) \quad . \quad . \quad . \quad . \quad (1)$$

$$\frac{df}{dt} = \frac{-Bf^N}{(bf_0 - f)^p} \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $A = B/E^{p+1}$. In the above equations t is time, f is stress, f_0 being the original stress, E is Young's modulus, and A, n, B, p, and N are constants, $b = 1 - (Ea/f_0)$.

Figure 7 – Relaxation Relationships from Johnson [11], page 431

The third and final report [11] from the Institute of Mechanical Engineers Pipe Flange Research Committee contained extensive additional testing of the creep of both full size and 0.4 scale model 8 inch carbon steel and C-0.5Mo BS10 Table T flanges. The initial part of the report focused on the elastic behavior of flanges. It was found that the change in flange stiffness at 575°C was only around 80% to 60% compared to the ambient temperature stiffness. Since the reduction in Young's Modulus is around 65% for this material and temperature, this indicates that either an additional variation due to the nature of the loading or some error or unaccounted factor in the testing. In the second part of the report, two relaxation relationships were used to curve fit relaxation data on loose ring flanges (Figure 7). The first relationship follows a time hardening theory and was used to fit most of the test results. The second relationship follows a strain hardening theory and was necessary in only a few cases tested. The mean curve of the carbon steel flanges was found to fit the time hardening equation with the values of [m=0.6 and A=2.5x10^3 tons^3/minute]. There are stress relaxation curves for the tested flanges presented in the paper (Figure 8), which will be useful for comparison to proposed analysis methods. One interesting aspect of this figure is that, for the time period tested, the results did not show significant convergence between the higher initial stress state and the lower initial stress states, as might be expected. This means that for this configuration, assembling to a higher initial stress will result in a significantly longer life.

The final result of the Pipe Flange Committee testing was that a good relationship describing the stress relaxation of the bolted joints examined was not discovered and the focus of the testing remained on the joint life vs. temperature curves presented in the second report (Figure 4). These test results were used to establish the applicable operational temperature limits for the BS10 Table T flanges. For the alloy flanges in BS10, further testing was not conducted, but rather a combination of extrapolation from the carbon steel flanges and successful operational history of existing joints was used to define the allowable pressure-temperature limits. One of the issues in establishing the relaxation behavior of the joint was that it is dependent on the load history of the joint and therefore operational loading variations and the initial stress state of the flange have a significant effect on the relaxation behavior.

In an addendum to the third report [11], R.W. Bailey provided a treatment of the associated theory used to estimate the creep relationships for the flanged joint. As part of the discussion, the aspects affecting deformation at temperature are discussed, including reduction in Young's Modulus and material Yield Strength. The effect of the reduction in yield strength is examined as an "advantage," in that the extreme stresses in the flange ring are reduced as the yield decreases with temperature (Figure 9). The path A-B demonstrates the reduction in stress (and therefore creep) as the stress relaxes from yield and ambient to yield at 450°C. This advantage can be used to explain the need to only address design at ambient temperature (per Kerkhof [10]). However, it will also result in an initial loss of bolt load, and so cannot be considered an advantage from the flange leakage life perspective. In addition to outlining the theory used in the assessment of the test results, Bailey presented a relationship for assessing the effect of the bolt holes on the flange ring creep. However, he also concluded that the effect of the reduction of strength due to the presence of the bolt holes was small.

In the discussion of the third report [11], J.A. Stafford provided additional stress relaxation test results (Figure 10) on another material obtained by constant strain test that clearly show a convergence between high initial stress and low initial stress states within the test time-frame. For this test case, the conclusion would be that assembling the joint to the highest stress level will not result in a significant increase in joint leakage life and will, therefore, only lead to additional creep damage with no advantage. Another interesting fact that can be seen is that there is variability in the test results regarding re-tightening and that, depending on the initial stress state, re-tightening may or may not improve the obtained life of the joint. However, it is likely that some or all of these conclusions (drawn for a simple tensile test) may not be directly applicable to the flange (bending stress) case.

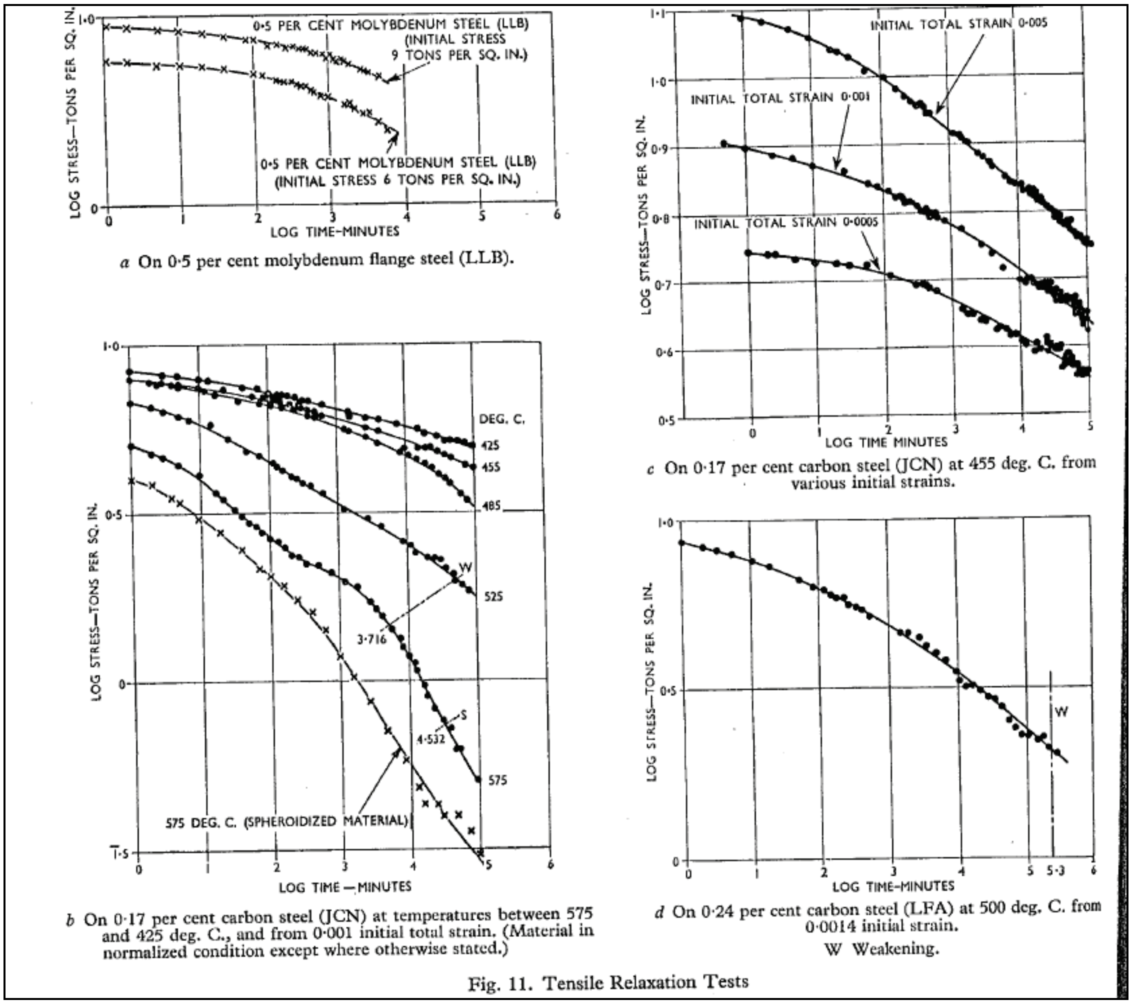

Figure 8 – Flange Ring Relaxation Graphs from Johnson [11], Fig. 11, page 432

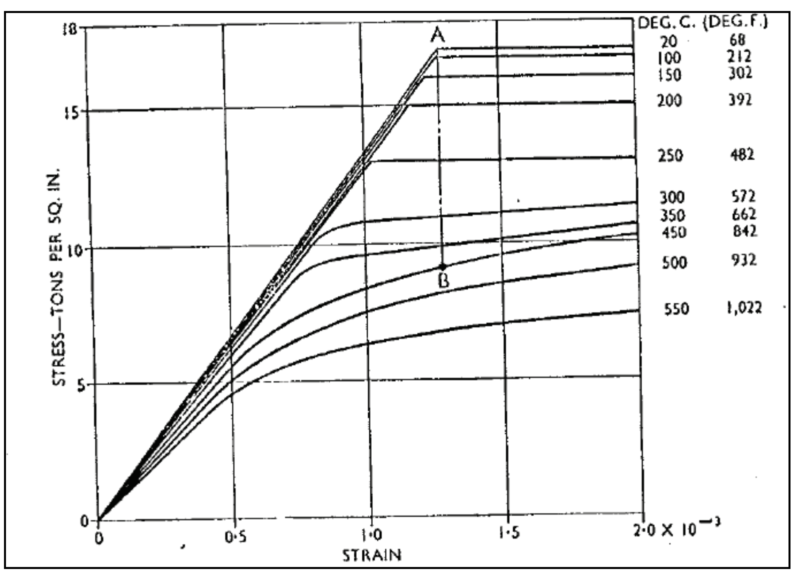

Figure 9 – Carbon Steel Pipe Flange Strength vs. Temperature from Johnson [11], Fig. 28, page 448

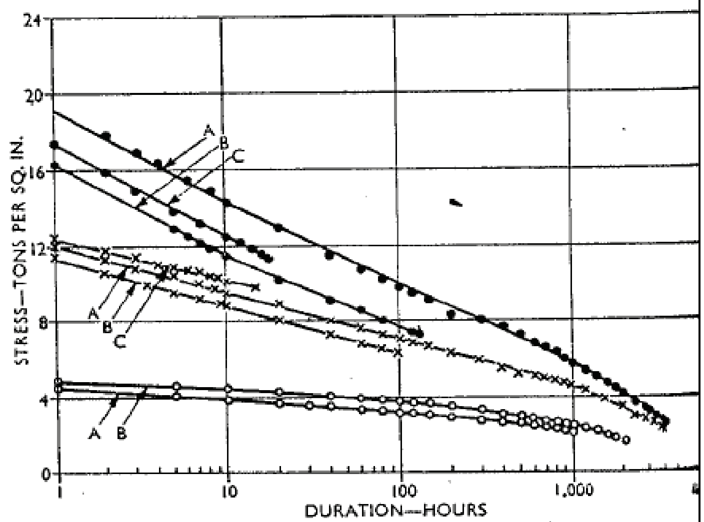

Fig. 33. Stress–Relaxation Tests—Constant Strain
Durehete 900 Steel

Effect of re-application of critical stress temperature of test
950 deg. F.

A Initial application of stress.
B First re-application of stress.
C Second re-application of stress.

—●— Initial stress 24 tons per sq. in.
—×— Initial stress 14 tons per sq. in.
—○— Initial stress 5 tons per sq. in.

Figure 10 – Tensile Test Relaxation Graphs from Johnson [11], Fig. 33, page 456

In 1956, Lake et. al. [12] proposed an alternative design method for flanges and addressed the behavior of the joint at temperatures in the creep range by applying a relaxation factor to the initial assembly stress for the ambient temperature design. The paper lists change in Young's modulus, material yield strength, creep and relative component expansion as contributing to this factor. The factor used in the paper for carbon steel and low alloy flanges is obtained from empirical (flange rating) data. There is no additional explanation of the rationale behind using the flange rating data.

Bernhard [13] summarized the application of the previous Pipe Flange Committee findings [11] into modification of the design and ratings for the British Pipe Flange Standard BS10. The paper discusses the considerations that went into extending the results across other flange sizes and materials than those originally tested. The results for determining the required thickness of different size flanges relied on the time to leakage graph produced by testing 8 inch flanges (Figure 4). It was determined that a relationship to adapt the test results to other applications was not apparent, with even the 0.1%/100,000 hr creep test results originally suggested by Bailey giving overly conservative results. A comparison of the results of constant stress creep tests on two different types of carbon steel led to the determination that the creep test results matched the data reasonably well only if offset by 20°F, which is a significant amount.

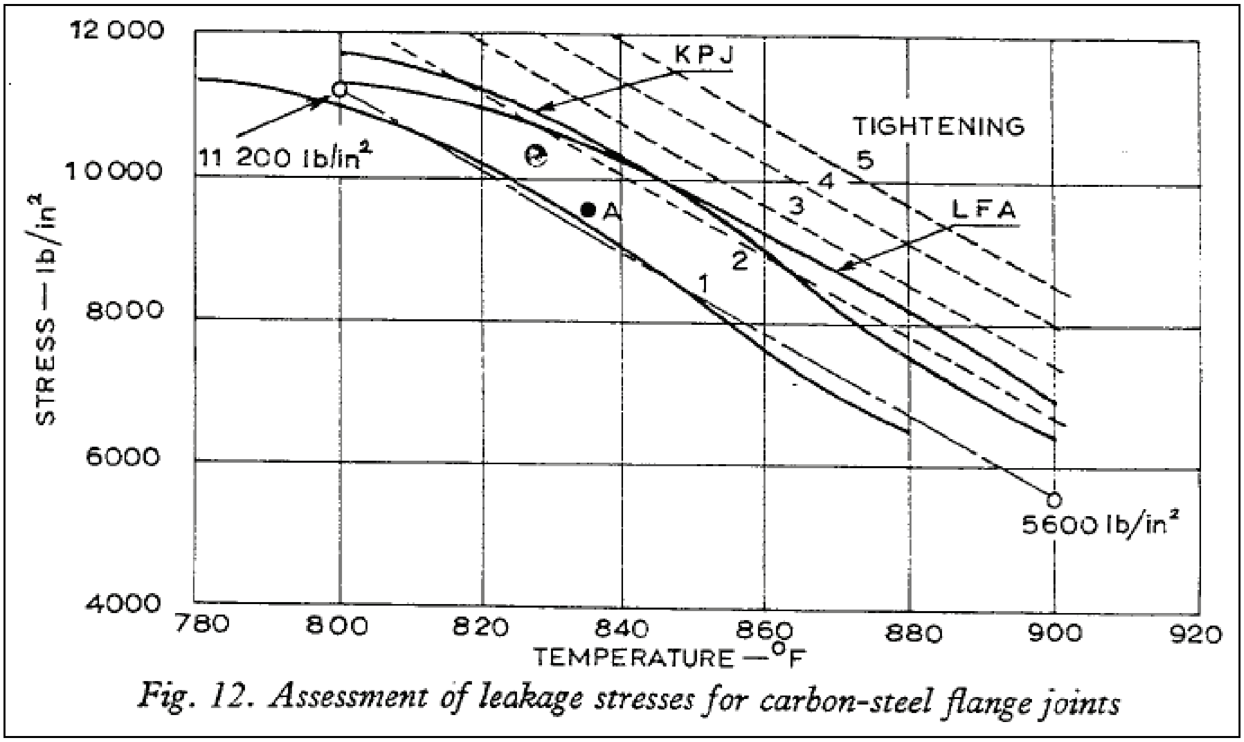

Figure 11 – Comparison of Flange Test Results vs. Creep Tests from Bernhard [13], Fig. 12, page 122

The BS10 flange thickness for a given size was determined by the surface stress on the loose ring flange, when adjusted for the effects of temperature on material yield. The effect of material yield versus temperature was accounted for by the adjusting the flange deformation based on the flange ring surface stresses by a factor "*c*" which accounted for the re-distribution of stress through the loose ring flange with increasing temperature. For the elastic stress case (linear stress distribution), the value of "*c*" was taken as 0.167. With the favorable re-distribution of the stresses at higher temperatures, the value of "*c*" increased to 0.21 at temperatures corresponding to flange creep.

Downey et. al. [14] revisits many of the concepts previously discussed regarding creep/stress relaxation behavior of high temperature joints and also discusses some mechanical failure history associated with these joints in the steam power industry. The analysis of the failure cases and theory concludes that the bolt rupture that occurred was likely a combination of one or more of the following factors; material embrittlement, notch effects due to the threads, non-uniform bolt temperature leading to localization of the creep strain in the high temperature region or non-uniform bolt cross-section, leading to localization of the creep strain in the higher stressed region of the bolt. Both of the last two factors may lead to failure due to the fact that even though the creep strain prior to leakage is limited to less than the initial elastic strain, these two factors will tend to concentrate the creep strain over a small length of the bolt and, therefore, lead to mush higher strain occurring (potentially to the point of failure) at that localized region. The main conclusion of the paper with regards to flange design is that creep relaxation tests at constant total strain in an extensometer may conservatively be used to determine the reduction in bolt load versus time for a given flange design. However, this approach neglects the flange bending creep regime (which will make it overly conservative) and also the effect of nut relaxation (which may make it non-conservative). A tensile test specimen configuration for the stress relaxation tests that accounts for the bolt and flange creep is outlined in the paper.

An excellent summary of the difference between creep and relaxation for design of high temperature bolted joints is provided in Cooper et. al. [15]. The paper details both time-hardening and strain-hardening relaxation rules in an understandable illustrative figure (Figure 12). In addition, the different creep characteristics of some commonly used flange and bolt materials are presented in order to illustrate that any method of analysis of creep or relaxation data must consider the different material characteristics, such as the significance of primary creep rates (Figure 13). An interesting conclusion regarding bolt over-stress for elevated temperature joints in that the bolt will relax to approximately two-thirds of yield, even at only a few hundred degrees F operating temperature, if it is tightened to a value above that. The paper points to the inadequacy of present design methods in considering creep and also indicates that insufficient bolt stress relaxation data is available to facilitate flange design in the creep region.

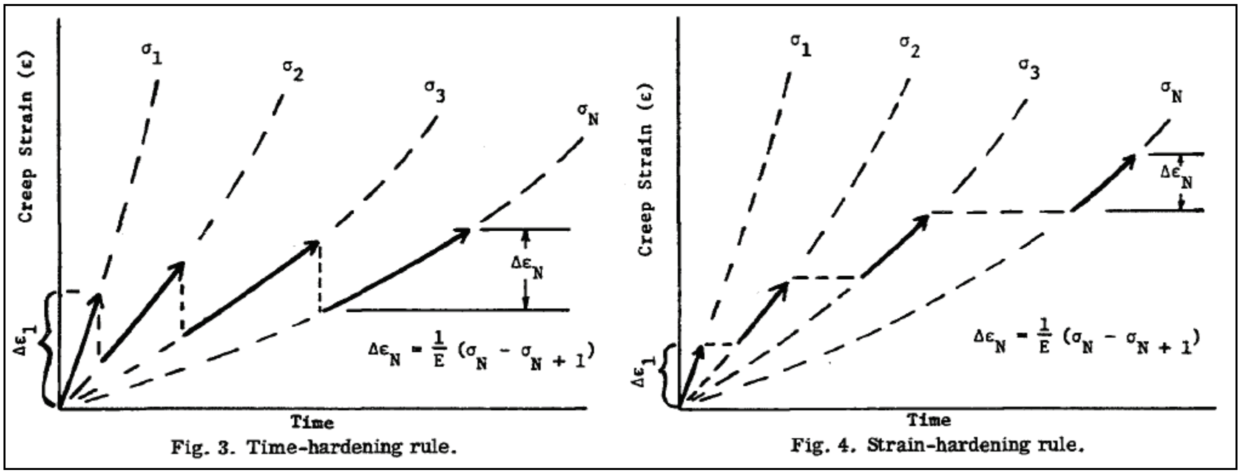

Figure 12 – Illustration of Relaxation Rules from Cooper, et. al. [14], Fig. 3 and Fig. 4, page 133

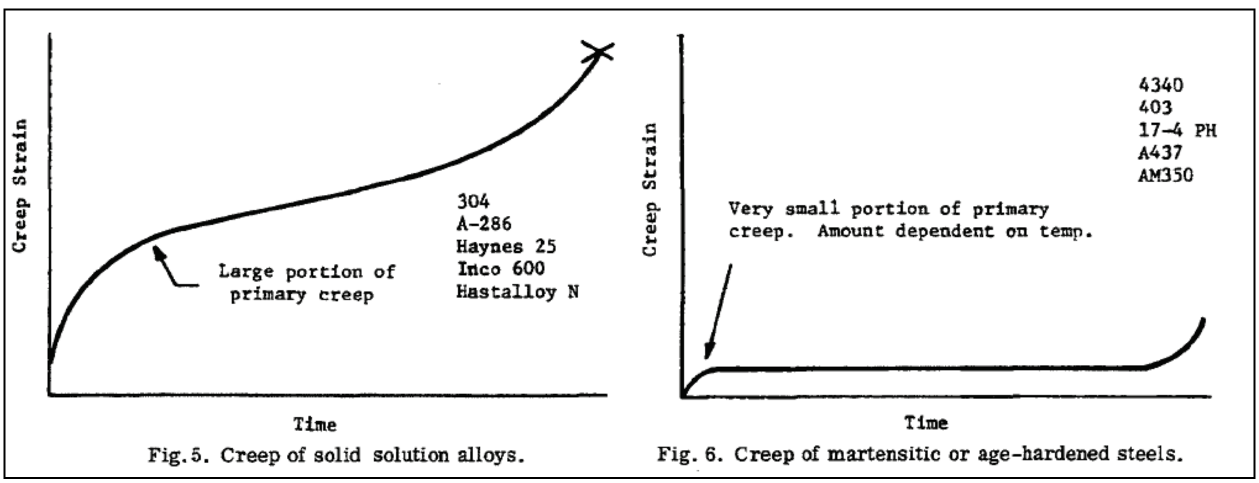

Figure 13 – Creep Characteristics of Different Materials from Cooper, et. al. 14), Fig. 5 to Fig.7, page 133

Fessler et. al. [16] detail an early use of FEA and a Bailey-Norton creep equation to examine the creep behavior of a 1-1/4 Cr flange material. The bolts in this case were assumed to be the same material as the flanges. The FEA approach initially used a time hardening relationship for the stress relaxation, but in the end a strain hardening approach gave better results. The FEA stress results were compared to a 0.4 scale photo-elastic model of the flange to verify the FEA model. No gasket was included in the model and a sensitivity analysis was performed on the bolt creep strength (as a multiple of the flange creep strength) to determine the effect of that on time to leakage. The creep predictions using FEA appear to be conservative by comparison to the test results presented in Johnson [11], as leakage is predicted after only 50,000 hrs (Figure 14). This conservative result may have been due, in part, to the fact that the redistribution of the stresses in the flange ring due to reduced material yield as temperature increased was neglected in the FEA treatment. The FEA used in this paper was the first of the references to include the effects of the attached hub and pipe on the creep behavior.

The work of Waters [7] and Fessler [16] was summarized and extended by Kraus, et. al. [17]. In this paper, a relaxation relationship for creep in bending of the flange ring is presented and analyzed using set time increments to determine the joint stress relaxation behavior. The effect of neglecting hardening and also including strain hardening were compared to both the Waters results (no hardening) and the Fesseler FEA results (strain hardening). The approach presented did not include the effect of the attached hub and pipe and so was found to be less conservative than the Waters approach, but about twice as conservative as the Fessler FEA results, which included the hub and pipe.

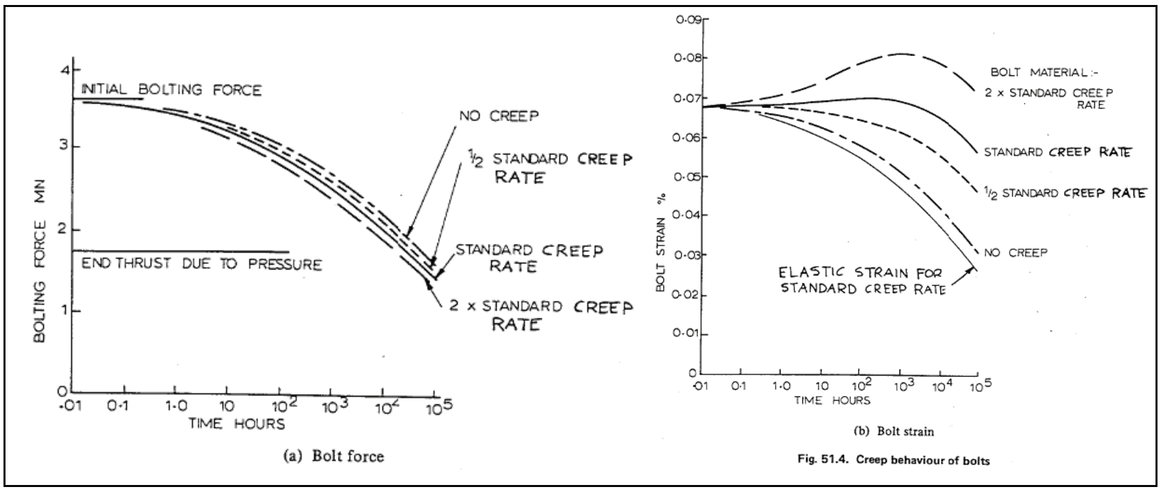

Figure 14 – FEA Bolt Load Relaxation Results from Fessler, et. al. [16], Fig. 51.4, page 45

In Maile et. al. [18] the stress relaxation behavior of bolts and bolted pipe joints is examined using both test and FEA. The creep laws examined include a Garafulo-Blackburn equation with two hardening terms and a Bailey-Norton type equation. The Garafulo-Blackburn equation was used to try to capture primary creep effects, however when compared to data on a simple bolt and cylinder test arrangement, the relationship gave a non-conservative estimate of the remaining bolt load at longer test times (Figure 15). The use of a Bailey-Norton relationship gave better long term results only when it is adjusted for initial relaxation (Figure 16). This is somewhat indicative of the difficulty in resolving flange joint creep, as it can be seen that the FEA prediction of stress relaxation on a bolt and cylinder arrangement, which is simpler than the full bolted joint, is not that accurate without additional adjustment. There are test results (both deformation and strain) presented for creep tests utilizing standard DIN 2510 flanges (Figure 17), which will provide an excellent point of comparison for any proposed methods of design. In addition, again they give some insight into the difficulties involved in accurately predicting flange behavior, as the spread of the results is relatively large (deformation between 0.7 mm to 0.9 mm), indicating sensitivity to the many variables involved. The final results presented in the paper are a comparison of FEA and experimental results (Figure 18), in which it can be seen that the FEA under-estimated the remaining bolt load by almost half.

However, this is a conservative estimate and therefore such under-estimation would probably be acceptable in a design procedure, keeping in mind that it is probably unrealistic to expect to accurately predict the stress relaxation in a flange joint. The final conclusions of this paper point to the need for greater creep and relaxation test data in order to accurately predict flange behavior.

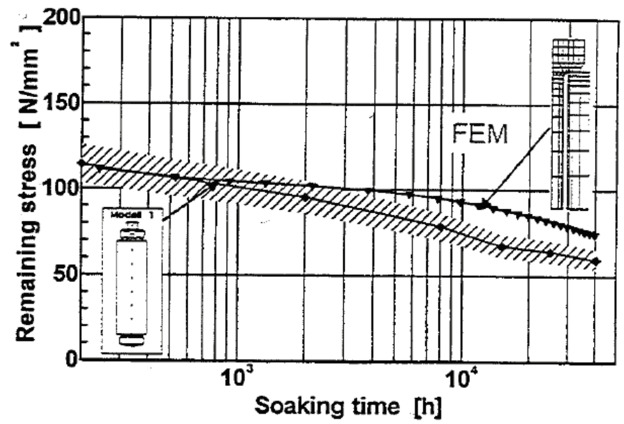

Fig 7: Comparison of experimental and calculated remaining stresses in bolting joint model No1, 12Cr-steel (X 19 CrMoVNbN 11 1), 550 °C.

Figure 15 – FEA Bolt Load Relaxation Results from Maile, et. al. [18], Fig. 7, page 156

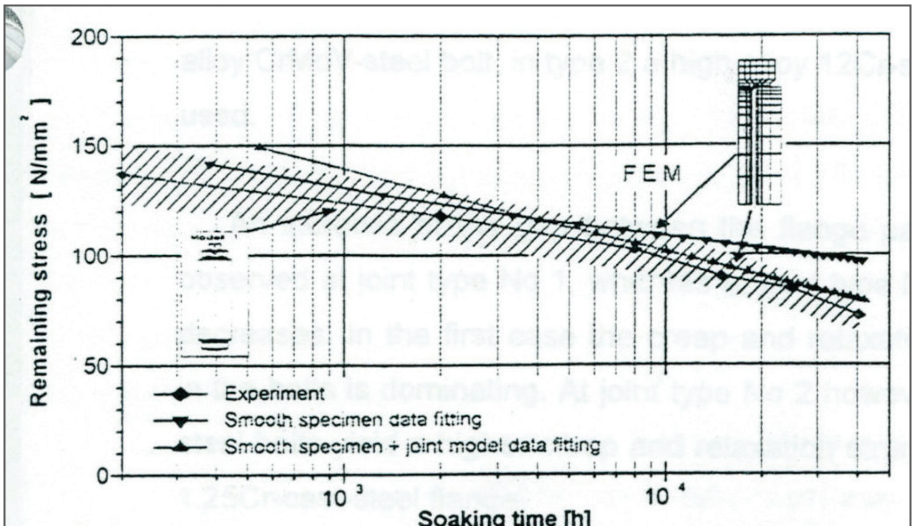

Fig. 8: Comparison of experimental and calculated remaining stresses in bolting joint model No1, 12Cr-steel (X 19 CrMoVNbN 11 1), 550 °C. Norton-Bailey creep law.

Figure 16 – FEA Bolt Load Relaxation Results from Maile, et. al. [18], Fig. 8, page 157

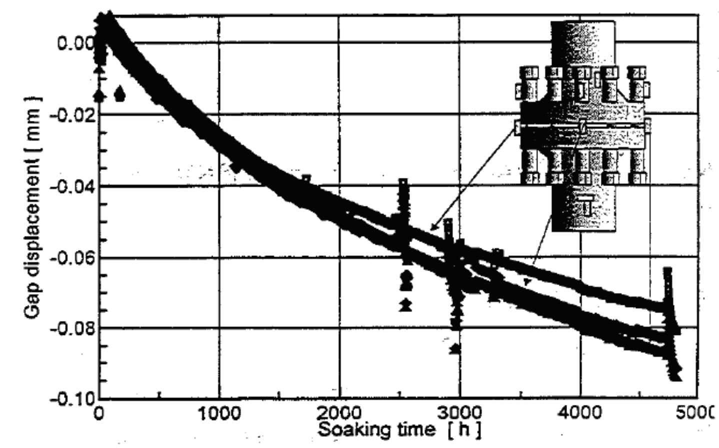

Fig. 13: Measured displacement, type No 2.

Figure 17 – FEA Bolt Load Relaxation Results from Maile, et. al. [18], Fig. 13, page 158

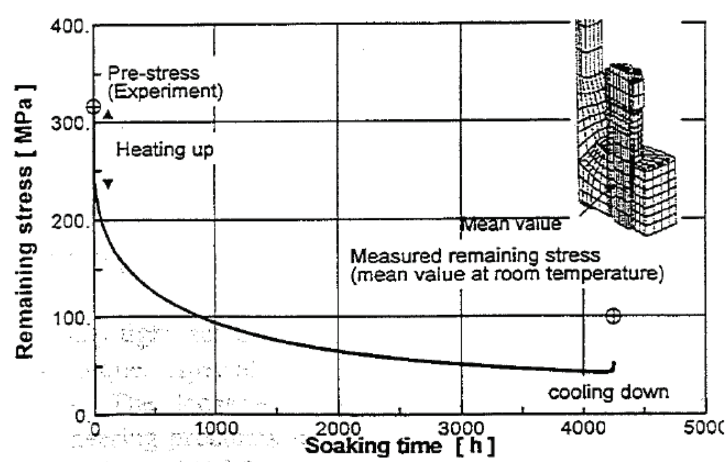

Fig. 15: Comparison between calculated remaining stress and measured remaining stress. Joint model type No 1.

Figure 18 – FEA Bolt Load Relaxation Results from Maile, et. al. [18], Fig. 15, page 159

Further details of testing similar to that discussed in [18] are presented in Gengenbach [19]. Test results for 600°C, 10,000 hr tests on 9% Cr flanges with Nimonic 80A bolts are presented and it is shown that good agreement between the test results and the use of inelastic FEA can be obtained.

Two papers by Nechache et. al. [20], [21] examine the use of finite time interval analysis similar to that used in previous papers to determine the creep behavior of a flange connection, but neglect one or more of the joint components. The creep properties used do not relate to actual materials, do not include any account of either strain or time hardening and no experimental verification was performed.

2.2 Mechanical Effects of Temperature on Joint Behavior

In early research, there were several noteworthy papers on the subject of high temperature flange leakage or temperature driven bolt load variations, such as:

- Bickford, et. al. [22] which examines the leakage of a particular reboiler flange.
- Winter, et. al. [23] which details FEA results for transient temperature effects on several piping flange sizes.
- Nau, et. al. [24] which uses FEA to examine the effect of bolt temperature lag during temperature transients.
- Singh, et. al. [25] which presents an analytic methodology to assess the radial growth of the flanges and tubesheet in a heat exchanger joint.
- Dudley [26] which outlines an analytic approach to solving the stresses and deflections of flanges due to a temperature difference between the flange ring and vessel shell.
- Sawa, et. al. [27] uses a finite cylinder heat transfer theory to examine temperature distribution in small flanges.

However, in general these papers only address certain aspects of the effects of temperature and none of them developed a comprehensive method for the assessment of the distribution of temperature in the flanged joint. The determination of the component temperatures must be considered the starting point for any method aimed at assessing the effects of temperature on flange joint operation. By building on the early work of Wesstrom [28], Brown [29], [30], [31], [32], [33], [34], [35] outlined an analytical method of determining both the temperature and associated bolt load during both steady state and transient conditions. This work was summarized in a Welding Research Council (WRC) Bulletin [36]. Using the approaches outlined in the WRC bulletin, the initial operating state of a raised-face type bolted flanged joint can be determined.

2.3 Code Status

The only pressure vessel design code that specifically addresses creep in the context of the design of bolted joints, other than by adjustment of the material allowable stresses in the creep range, is EN1591-1 and EN13555. However, the creep considerations are only with respect to the gasket relaxation and therefore do not appear to address operation in the flange or bolt material creep range. In addition, the methods implemented in both of the codes do not effectively describe the effects of gasket creep/relaxation on the bolted joint behavior [1], [37].

Mention of the effects of bolt load relaxation and the need to account for the effect in design and selection of assembly bolt load is made in BS 4882:1990 [7]. BS 4882 lists some relaxation data and makes reference to other sources of relaxation data. The previous edition (1973) had graphs of relaxation data for common bolting materials, but these have been removed from the current edition.

2.4 Gasket Creep Behavior

Thorn's report [39] is an early paper on the relaxation properties of asbestos sheet gasket materials. Farnam [40], Smoley, et. al. [41] and Marchand, et. al. [42] examine the relaxation characteristics of sheet gasket materials, including the effects of elevated temperature. Bazergui [43] presents the short-term (initial) relaxation of various gaskets and determines a suitable creep expression for describing the relaxation of the gaskets. Marchand, et. al. [44], Bouzid, et. al. [45], [46], [47], [48] and Nagy [49], [50] address the relationship of gasket relaxation with respect to joint interaction.

Nau [51] wrote a general paper on the operational characteristics of gaskets, in which the various mechanisms of gasket load relaxation versus time are described and a design theory presented. Kockelmann, et. al. [52] presents relaxation data for flexible graphite and metal insert graphite gaskets, the effects of temperature on relaxation are also examined. Latte, et. al. [53] and Bouzid [54] present the relaxation and leakage properties in modified PTFE gasket materials. Vignaud, et. al. [55] addresses the relaxation characteristics of spiral wound gaskets, including a comparison between two gaskets made by different manufacturers. Nassar et. al. [56] and Alkelani et. al. [57] examine the effect of creep on SBR gaskets in a simple bolted joint and propose a visco-elastic creep model to describe the relaxation behavior.

2.5 Material Relaxation Behavior

There are many different singular sources of relaxation data for different materials under different test conditions; however it is most often the case that the data does not match the conditions that one wishes to analyze. It is generally necessary to build an overall picture of the relaxation behavior of a given material across several test conditions in order extrapolate appropriate material properties for the conditions being analyzed. For that purpose, it is obviously easier to use data that has been compiled into a standard format. For this project, on such resource used was ASTM DS 60 [58]. This document has an extensive compilation of data from various sources for a large number of material types.

3 CREEP BEHAVIOR

3.1 Definition of Creep Law and Material Properties

One of the common problems mentioned throughout the literature is the lack of material creep/relaxation data for the determination of the joint behavior. Additionally, there are a multitude of creep laws that have been used for the evaluation of creep and relaxation behavior. Therefore, the first step in determining a method of evaluating the creep/relaxation of a joint is the selection of an appropriate creep/relaxation relationship and the determination of the availability of the required material data for modeling purposes. Unfortunately, while creep laws such as the Omega method (API 579-1 / ASME FFS-1 [59]) are codified and have published properties for many commonly used material types, they may not be appropriate for the determination of the creep/relaxation behavior of a bolted joint, necessitating an alternate approach.

The second common notable point taken from the literature was the effect of strain (or time) hardening on the actual behavior of the joint. The omission of this effect leads to an over-estimation of the loss of bolt pre-load when higher bolt loads are compared to lower bolt loads using the same creep/relaxation law. This effect is apparent when the Omega method is used to predict the relaxation in uniaxial stress versus actual measured uniaxial relaxation test data (ASTM DS-60 [58]) for both Carbon Steel at 842°F (Figure 19) and 932°F (Figure 20). It can be seen that for certain combinations of stress and temperature the Omega method is a good predictor of residual stress vs. time. However, it is also evident that the lack of strain hardening leads to over prediction of relaxation from higher initial stress levels. In addition, at low stress and especially temperatures (572°F for example) where the test results still show a substantive loss of stress (in the range of >30% of initial preload), the Omega method would predict no relaxation to occur at all. Therefore, even if the method were modified to incorporate a strain hardening term, further modification would be required to capture lower temperature relaxation behavior.

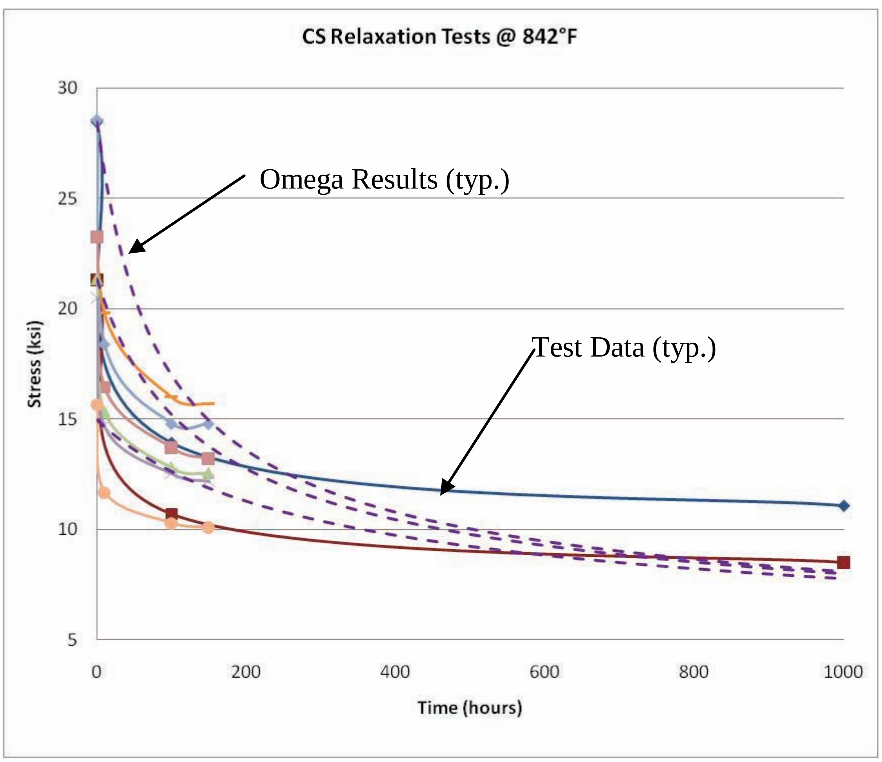

Figure 19 – Uniaxial Carbon Steel Omega Relaxation Results at 842°F

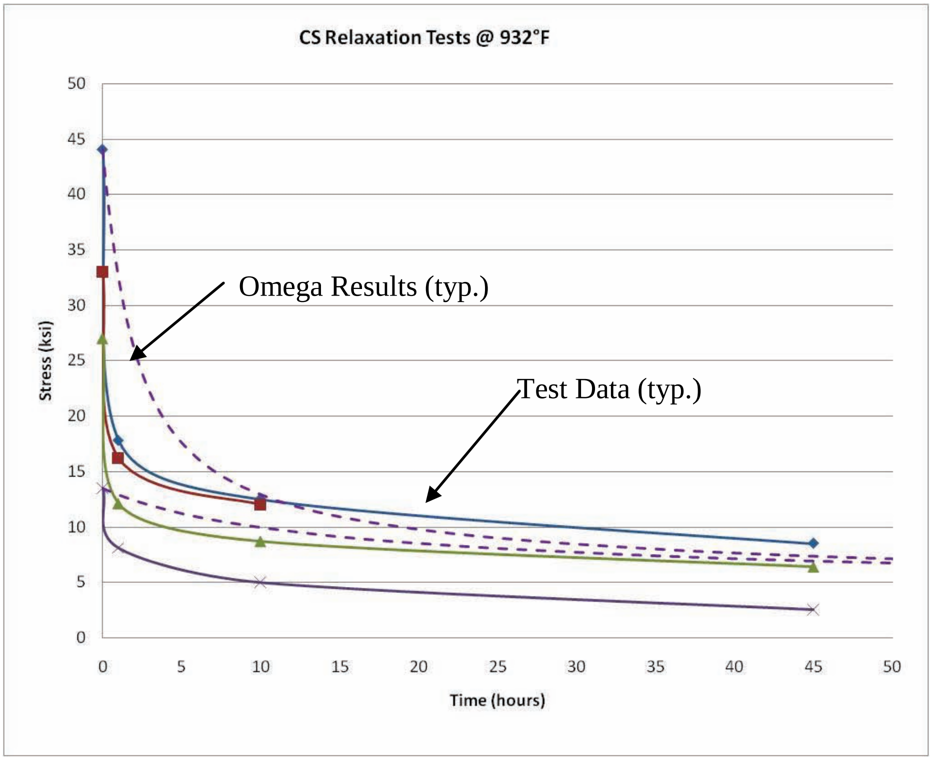

Figure 20 – Uniaxial Carbon Steel Omega Relaxation Results at 932°F

The relaxation of bolt stress at temperatures below the normal creep range is due to lower temperature mechanisms such as microplasticity (ASTM DS-60 [58] and McMahon [60]). Microplasticity occurs where the nominal cross section of the structure (bolt and flange in this case) is at a stress level below yield, but there are local regions of the microstructure that undergo yielding. This localized yield behavior, at the microstructure level, leads to a loss of initial stress under relaxation testing. Normal creep laws do not predict this type of behavior. Therefore, it is evident that the utilization of a simple creep relationship, such as the one-term modified Graham-Walles Equation (1), presented in Maile, et.al. [18] and Bolton [61] and the use of actual relaxation test data to determine the appropriate relationship will be necessary to accurately predict the creep relaxation behavior of a bolted joint.

$$\Delta\epsilon_c = A.\sigma_0^N.\epsilon_{c0}^m.dt \qquad \textbf{(1)}$$

Where: $\Delta\varepsilon_c$ is the increment in creep strain over the time increment *dt*

A, *N* and *m* are material constants, with *N* being temperature dependant

σ_0 and ε_{c0} are the initial stress and initial creep strain at the start of the time increment.

This equation was programmed using a finite-difference approach with an Excel spreadsheet to enable a fit of the equation to the available data from ASTM DS-60 [58] for carbon steel. The fit is found to be reasonable for some test data (Figure 21 to Figure 24) and more approximate for other test cases (Figure 25 and Figure 26). In addition, it should be noted that while the trend of the data is in reasonable agreement, there are relatively few test points going into each fit and the inclusion of test points at longer test times has a significant effect on the obtained fit. This indicates that it would be advisable to have controlled test data from conditions that are similar to the expected material operation. The *A* and *m* used in these fits were 5x 10^{-13} and -0.93, respectively with units in ksi, in./in. and hours. If the variability of the *N* factor with temperature is plotted (Figure 27), it can be seen that the graph has a bi-linear appearance with a transition in the temperature range where creep effects become significant for this material. There are relatively few data points (from a diverse variety of test configurations) to support this observation; however, from a mechanistic sense it would seem appropriate that the presence of a transition from Microplasticity to Creep regimes would support this observation.

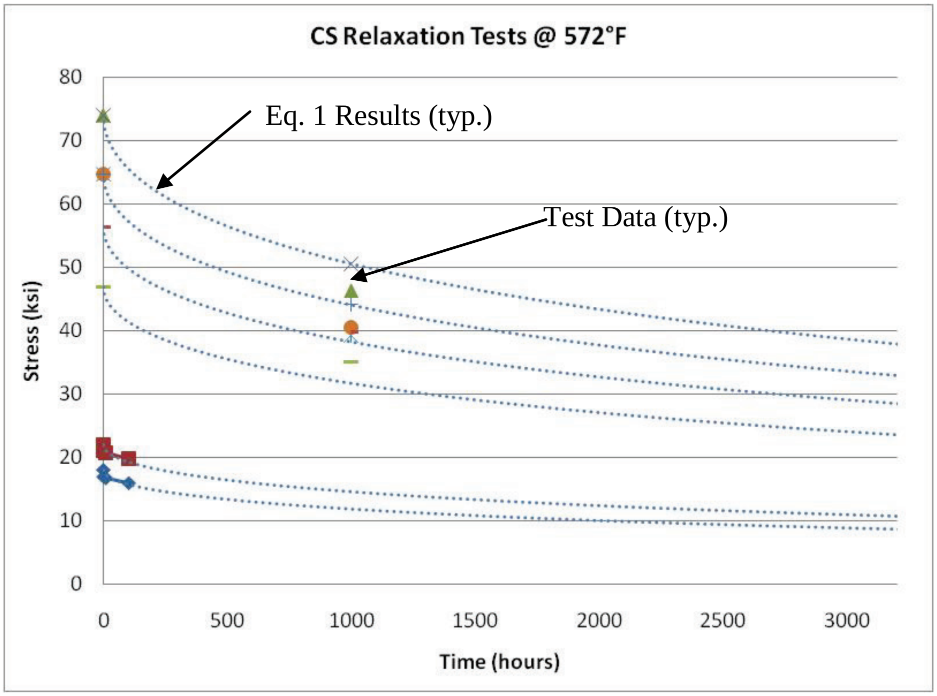

Figure 21 – Equation (1) Carbon Steel Relaxation Results at 572°F

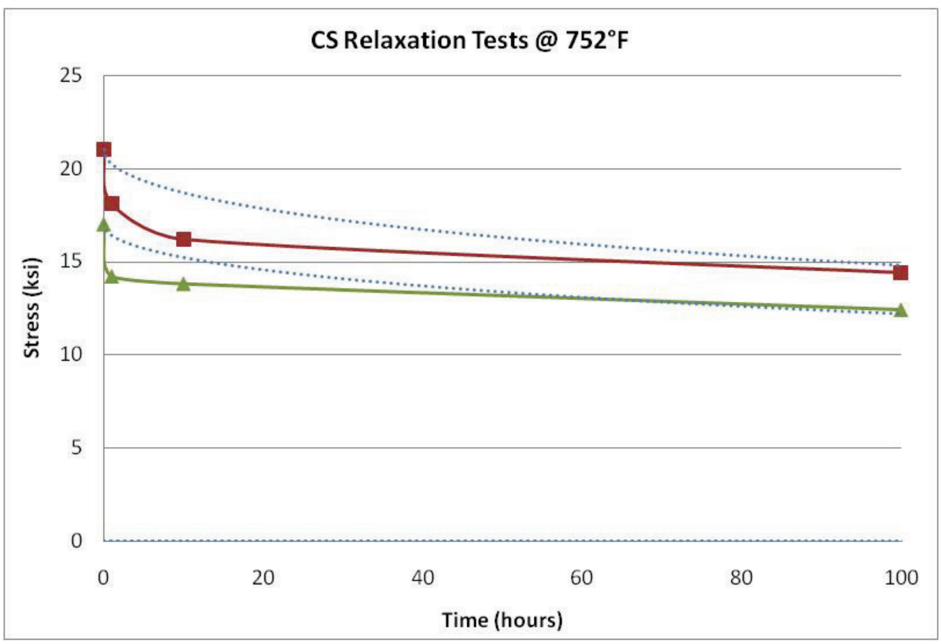

Figure 22 – Equation (1) Carbon Steel Relaxation Results at 752°F

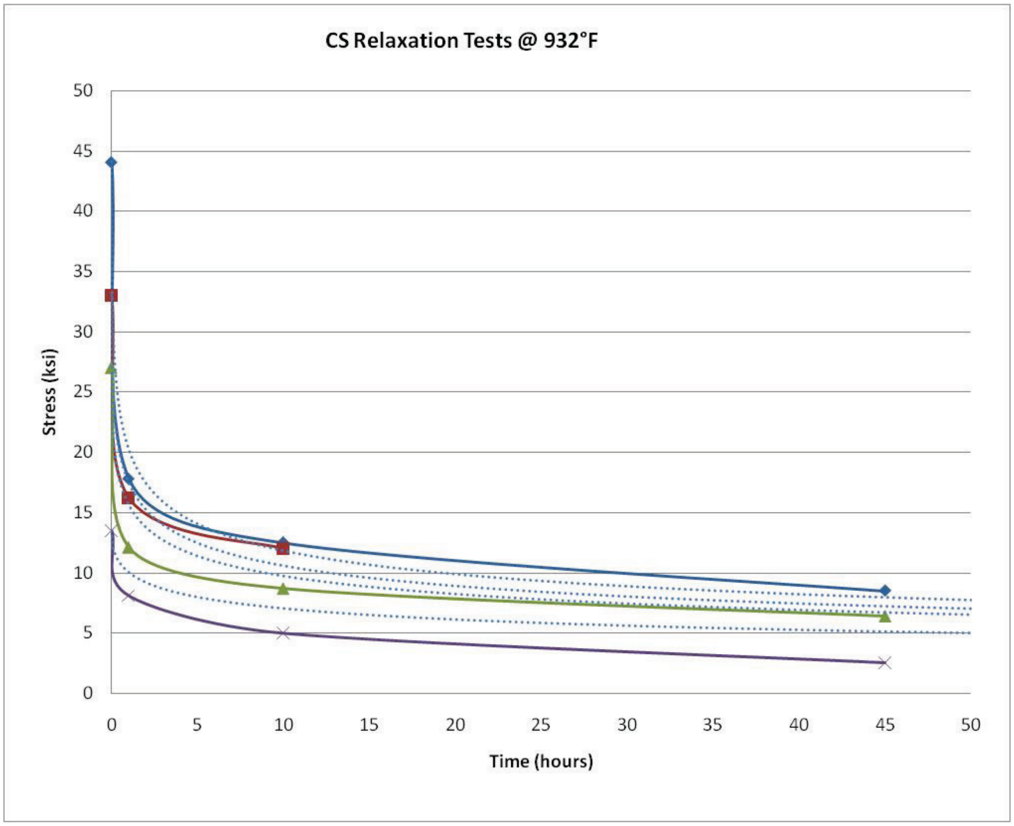

Figure 23 – Equation (1) Carbon Steel Relaxation Results at 932°F

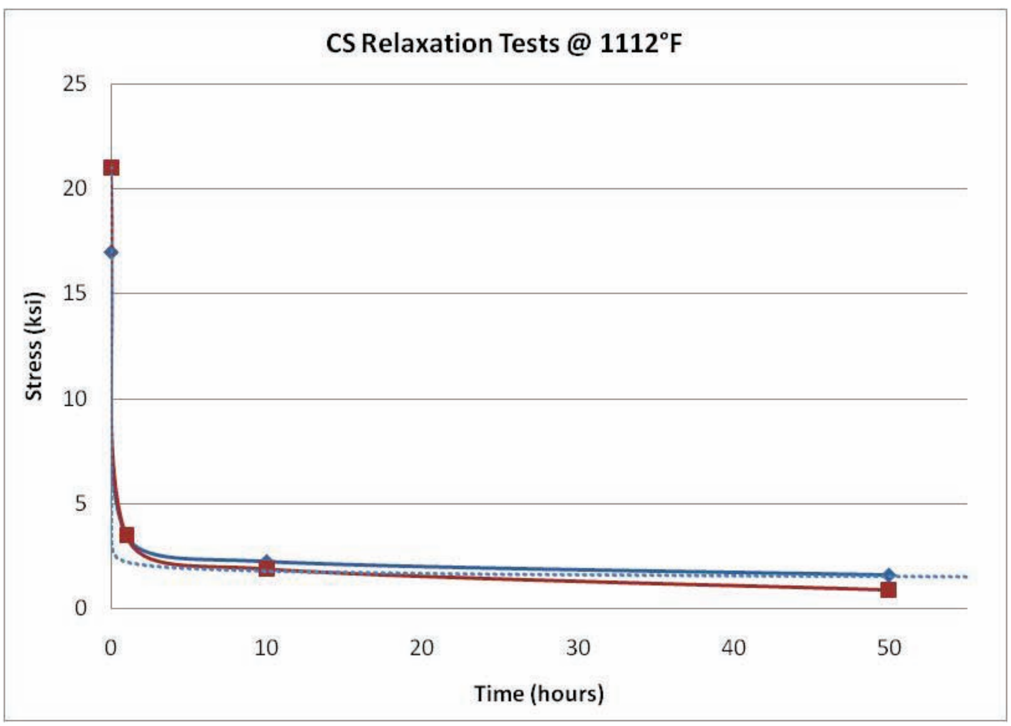

Figure 24 – Equation (1) Carbon Steel Relaxation Results at 1112°F

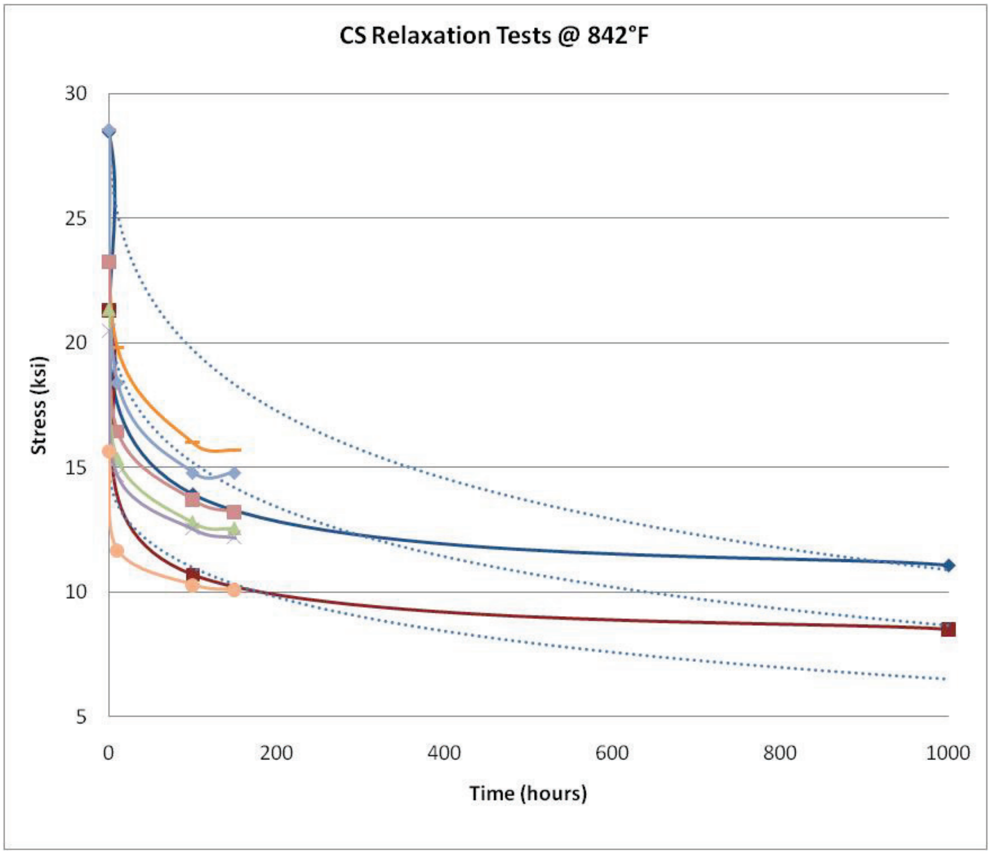

Figure 25 – Equation (1) Carbon Steel Relaxation Results at 842°F

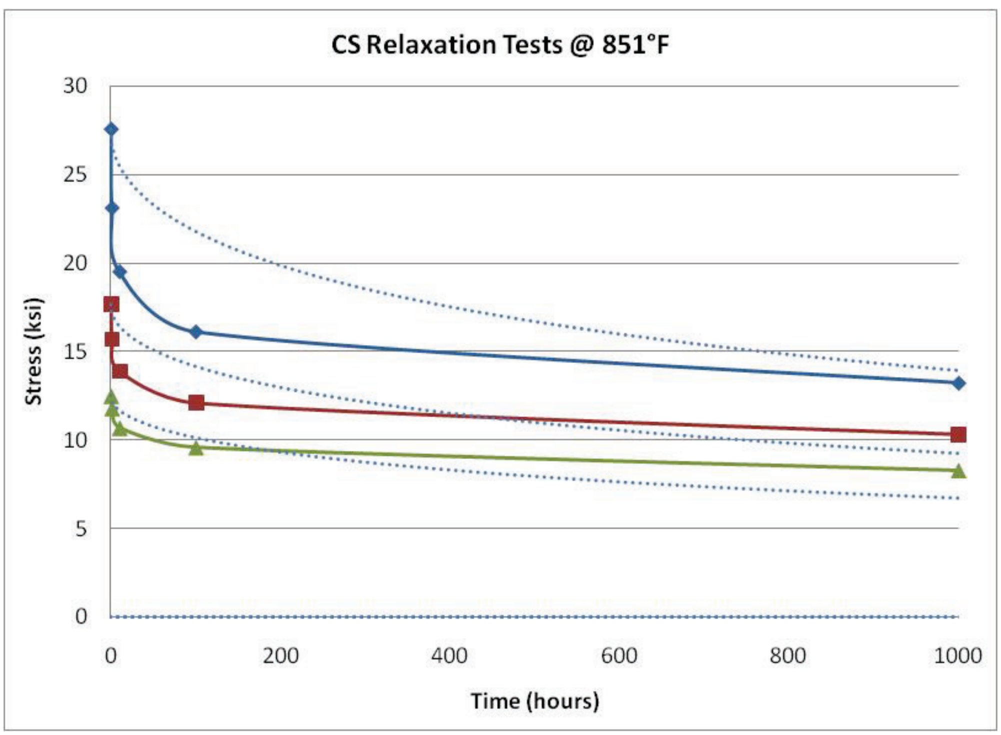

Figure 26 – Equation (1) Carbon Steel Relaxation Results at 851°F

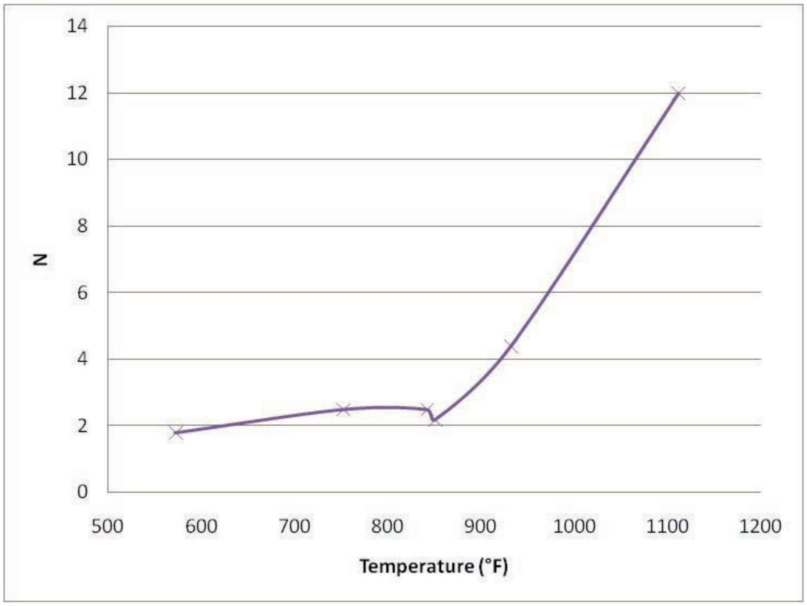

Figure 27 – Equation (1) Carbon Steel "N" vs. Temperature

If a similar approach is taken with Cr-Mo (of similar composition to ASTM A193-B7 bolting material) test results, then the data fits at different temperatures are again seen to have the correct form, while not being exact (Figure 28 to Figure 33). In these cases, the values of A and m were taken as 1.1×10^{-12} and -0.93, respectively, with units in ksi, in./in. and hours. It can be seen that the value of N versus temperature again appears bilinear (Figure 34), although once again there is a relatively small amount of data to support this determination. However, the combination of both of these relationships for carbon steel and CrMo materials allows, in principal, the determination of the creep/relaxation behavior of a very common flange configuration (SA105 with SA193-B7 bolts) at any temperature within the data range.

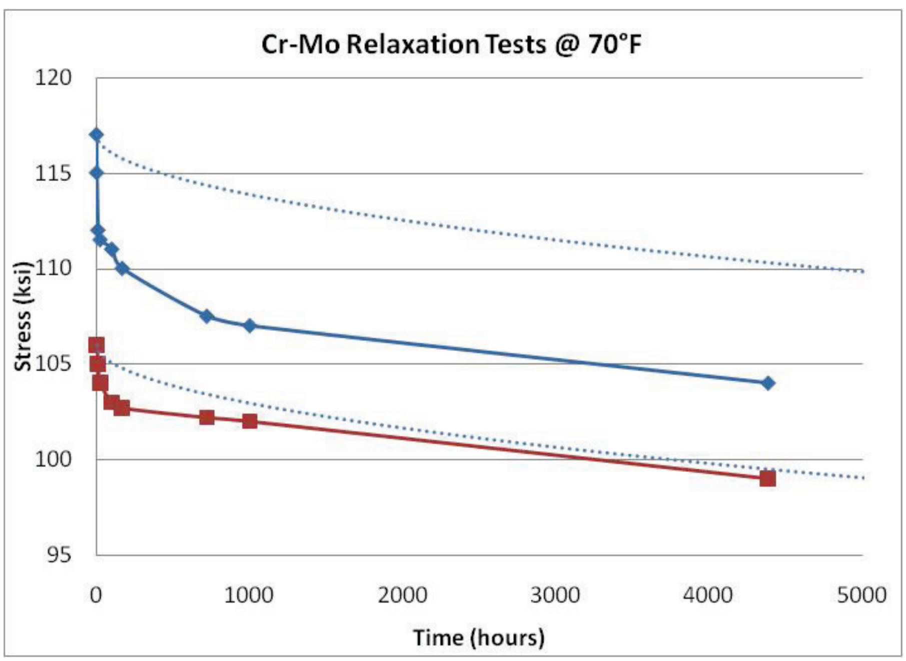

Figure 28 – Equation (1) CrMo Relaxation Results at 70°F

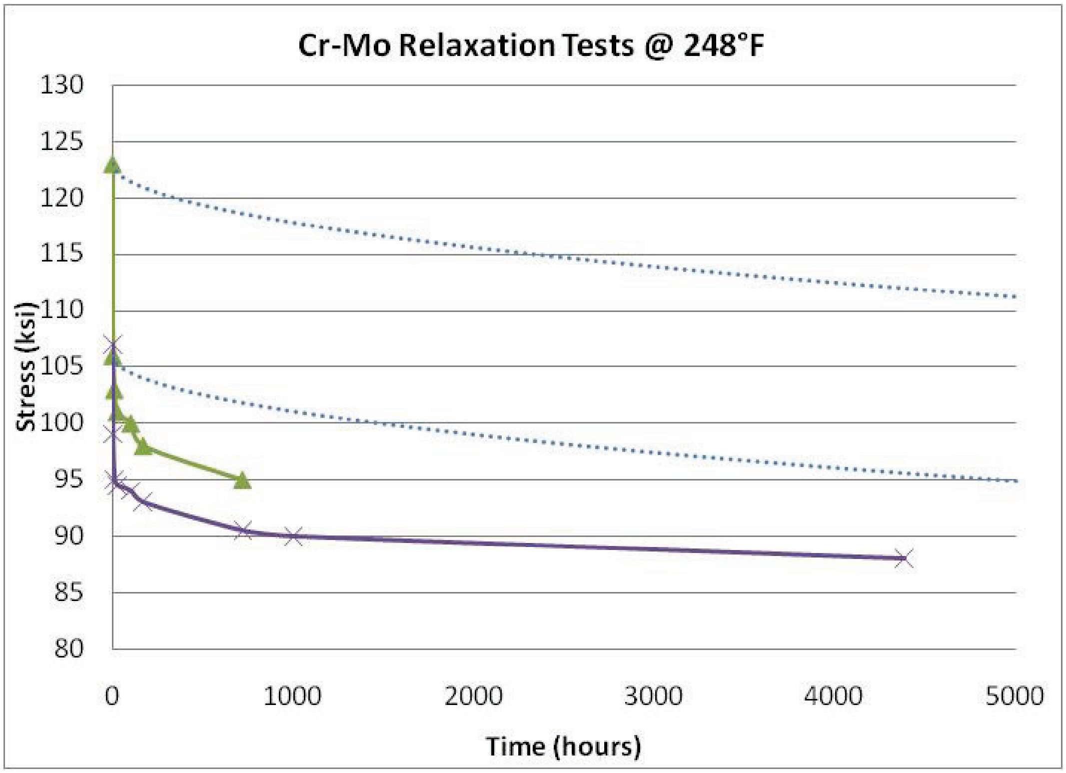

Figure 29 – Equation (1) CrMo Relaxation Results at 248°F

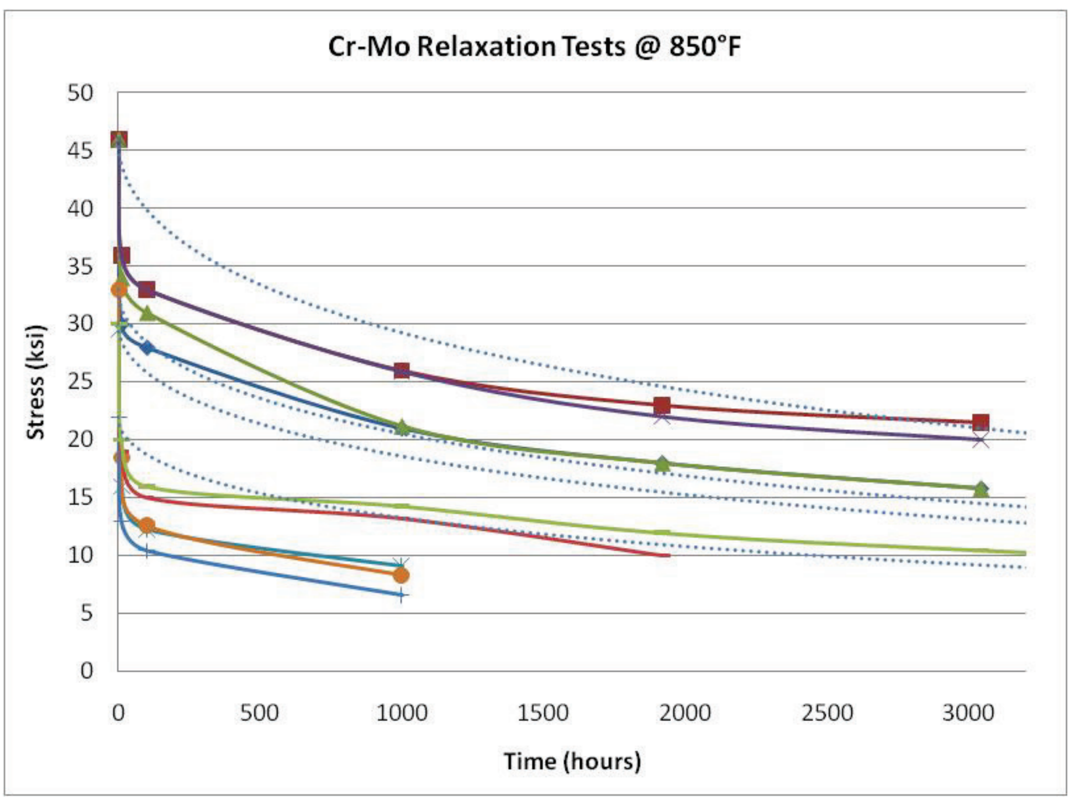

Figure 30 – Equation (1) CrMo Relaxation Results at 850°F

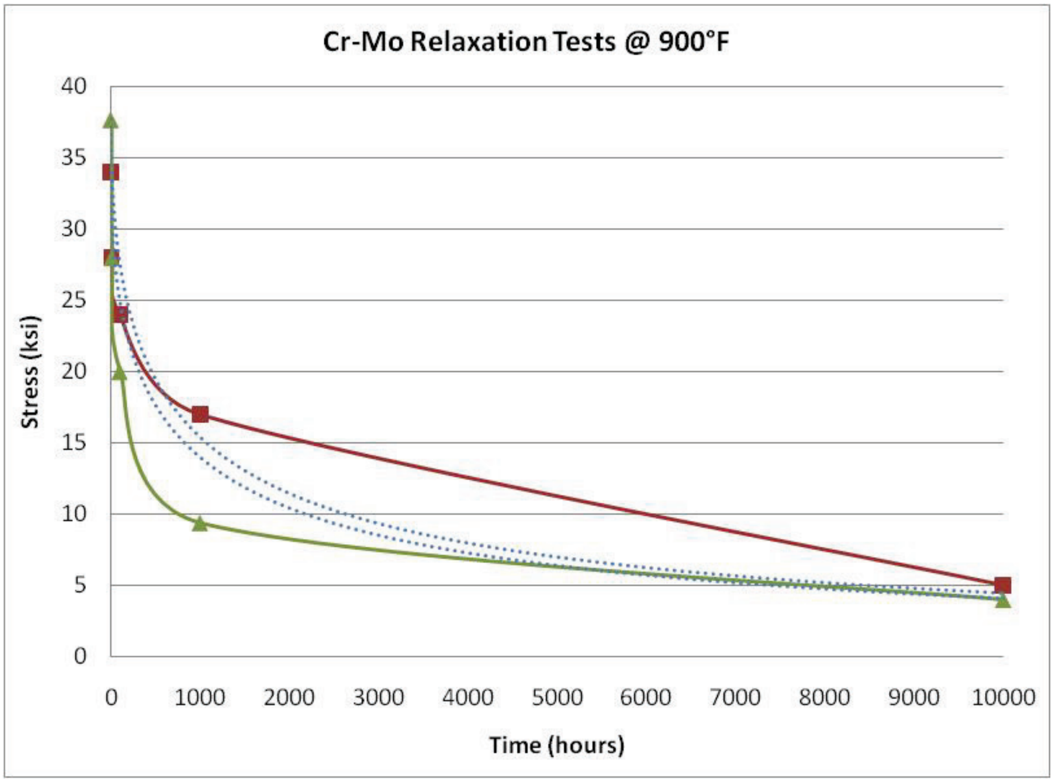

Figure 31 – Equation (1) CrMo Relaxation Results at 900°F

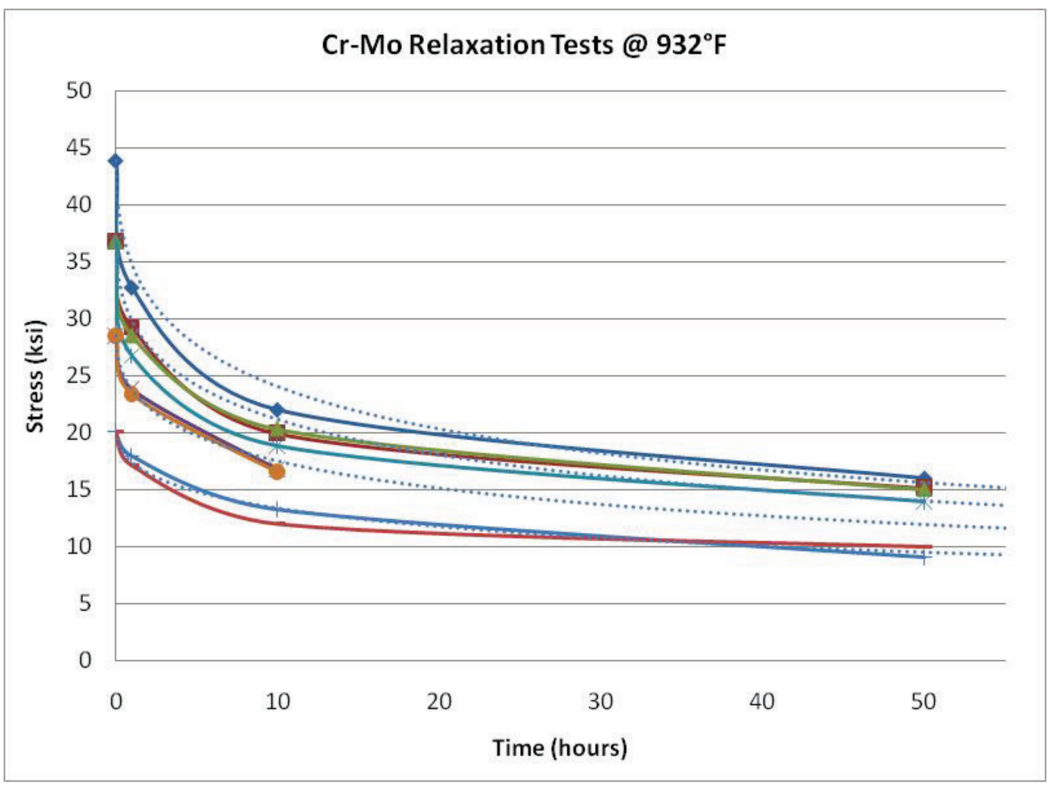

Figure 32 – Equation (1) CrMo Relaxation Results at 932°F

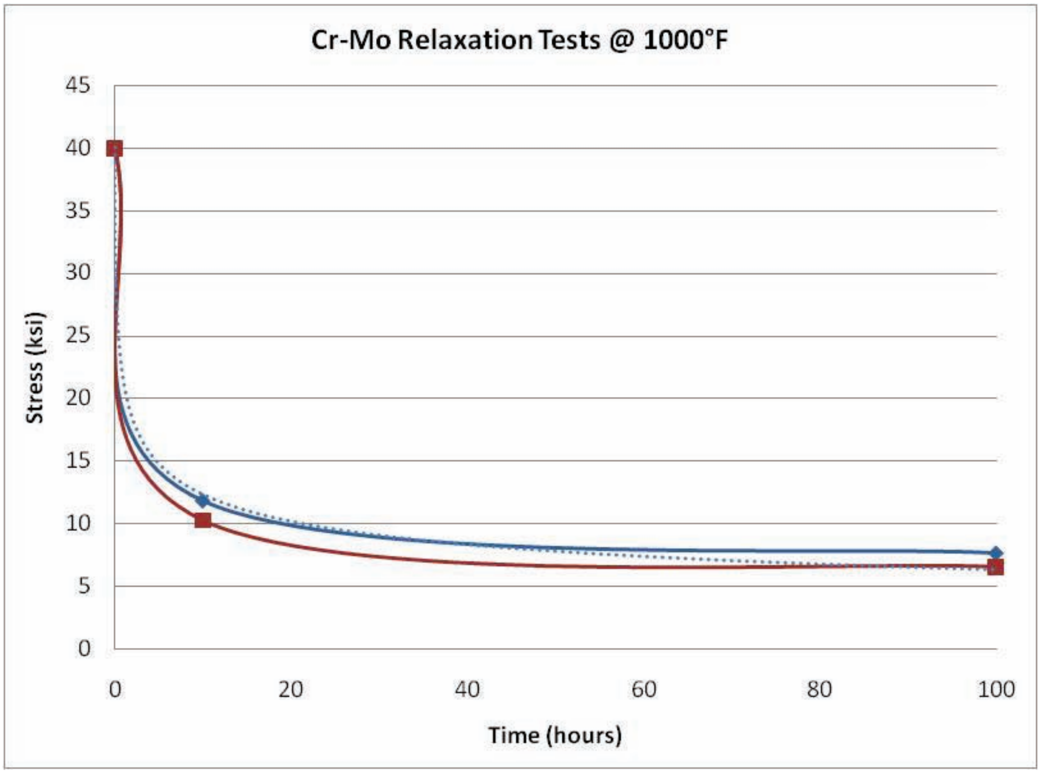

Figure 33 – Equation (1) CrMo Relaxation Results at 1000°F

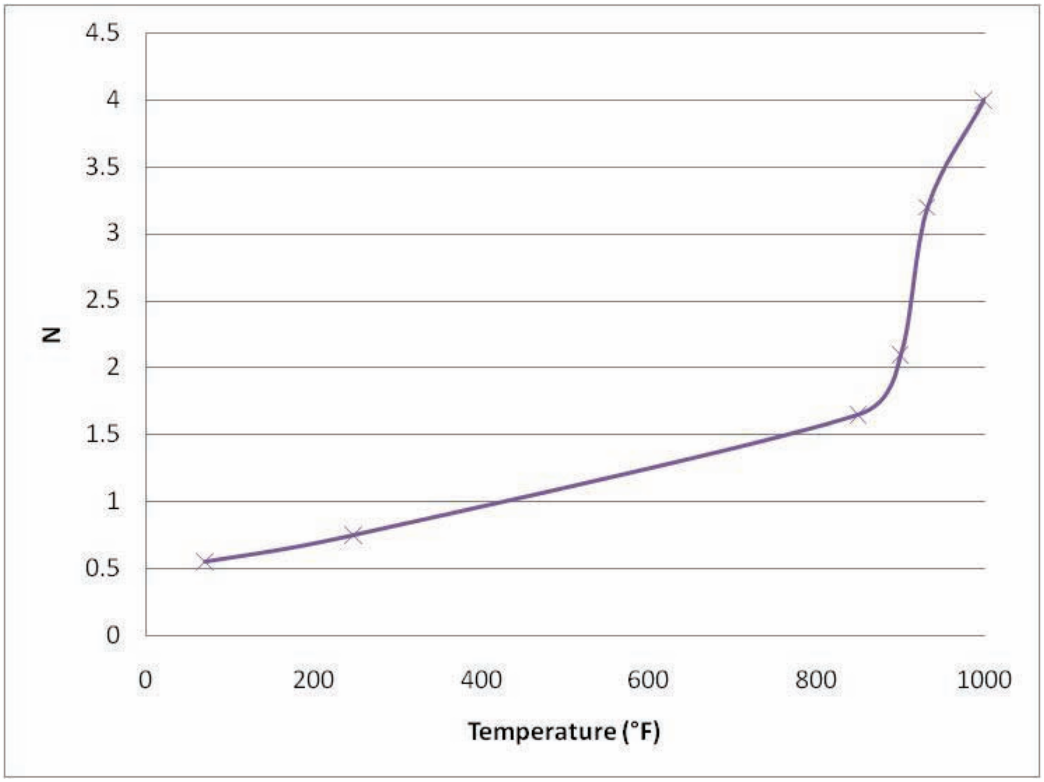

Figure 34 – Equation (1) CrMo "N" vs. Temperature

The flange joint components will see a combination of both creep and relaxation over their lifetime. Common creep relationships, such as the Omega method, are determined from constant load or constant strain rate testing (predominantly creep), whereas the data used to establish the relationships in this project are based on relaxation (constant deformation) testing. It is conceivable that the relationship used in the calculations may have needed to be modified to become closer to an "average" between creep and relaxation from a combination of pure creep and pure relaxation test data. It is expected that the form of the equation used was, however, still appropriate. In addition, the effect of multiaxial creep may be significant in the results, however since the flange stresses are predominantly due to bending of the flange ring due to the bolt load, it may also be appropriate to neglect the effects of multiaxiality or apply a simplified adjustment. For this project, the effects are neglected.

3.2 Finite Element Modeling

The established relationships from Equation (1) and the test data fits were programmed into a user subroutine in the ABAQUS commercial Finite Element Analysis (FEA) software and the results for comparison to the uniaxial relaxation test results were found to be identical to the previously presented finite difference generated data. However, when applied to the actual joint arrangement, the FEA allows for the reduction in bolt load due to the relaxation of each component, termed hereafter as the compound relaxation of the joint, rather than the independent relaxation of just the bolt or just the flange. The results of this FEA prediction can then be used for comparison with an approximate treatment using the flange code design equations (closed form solutions), followed by comparison with actual test results. Direct application of the code calculated flange stresses to determining flange creep strain is not appropriate, due to the fact that some flange stresses are local in nature (hub longitudinal stress S_H for example) and therefore creep/relaxation associated with that stress will not have as significant effect on the flange deformation (and, therefore, bolt load relaxation) as stresses that are closer to a membrane stress (such as the hub/shell junction tangential stress S_T).

The first application of the FEA method for confirmation against bolt relaxation test results was to model a 7/8 in. diameter UNC, ASTM A193-B7 bolt (CrMo) with ASTM A194-2H (carbon steel) nuts tightened on a carbon steel hollow cylinder to varying initial stress levels. Two FEA models were built for this comparison (Figure 35), one with the nut and threads explicitly modeled and one with a simplified approach where the nut and the bolt shank (with diameter equal to the thread root diameter) were modeled as solid cylinders tied at the nut-to-bolt interface. Both cases were run at 50ksi, 75ksi and 100ksi initial stress levels for 168 hours at 650°F, with the N factor linearly interpolated from the previously determined relationship versus temperature. The resulting reduction of bolt stress versus time for the fully detailed model and the simplified model due to component relaxation can be seen in Figure 36 and Figure 37, respectively. It can be seen, by comparison of these two figures, that the simplified model appears to adequately describe the relaxation behavior. The effect of strain hardening is apparent and it can be seen that after 150 hours, the stress level is still reducing, though at a slower rate. Most of the relaxation is predicted to have occurred within the first 100 hours.

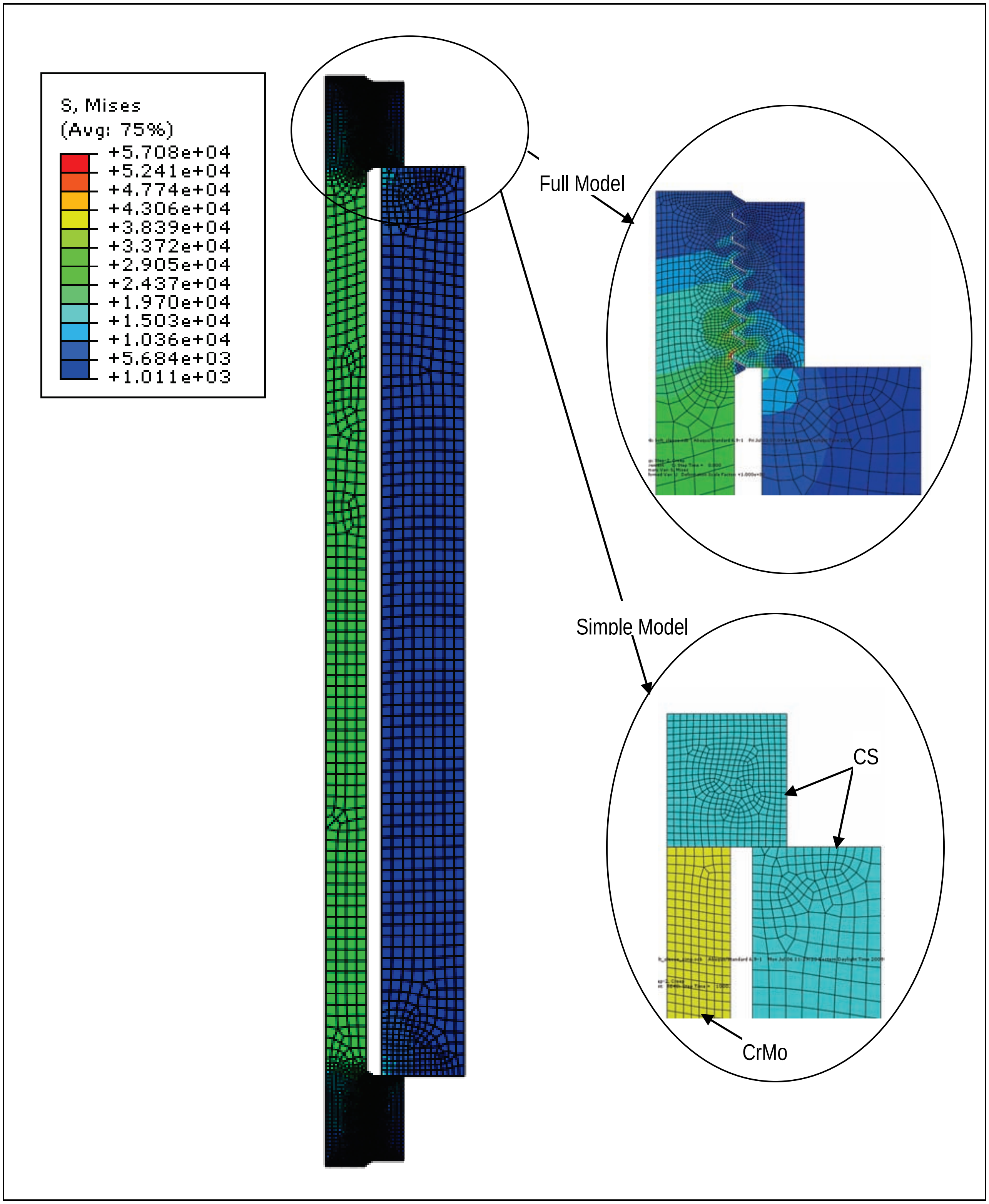

Figure 35 – Bolt/Cylinder FEA Model

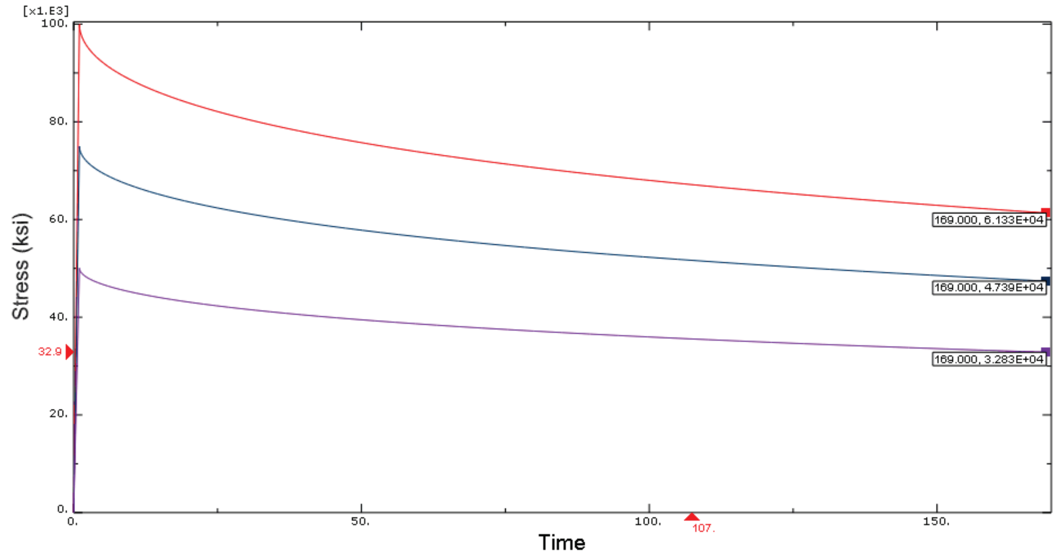

Figure 36 – Bolt/Cylinder Full FEA Model Bolt Stress vs. Time Results

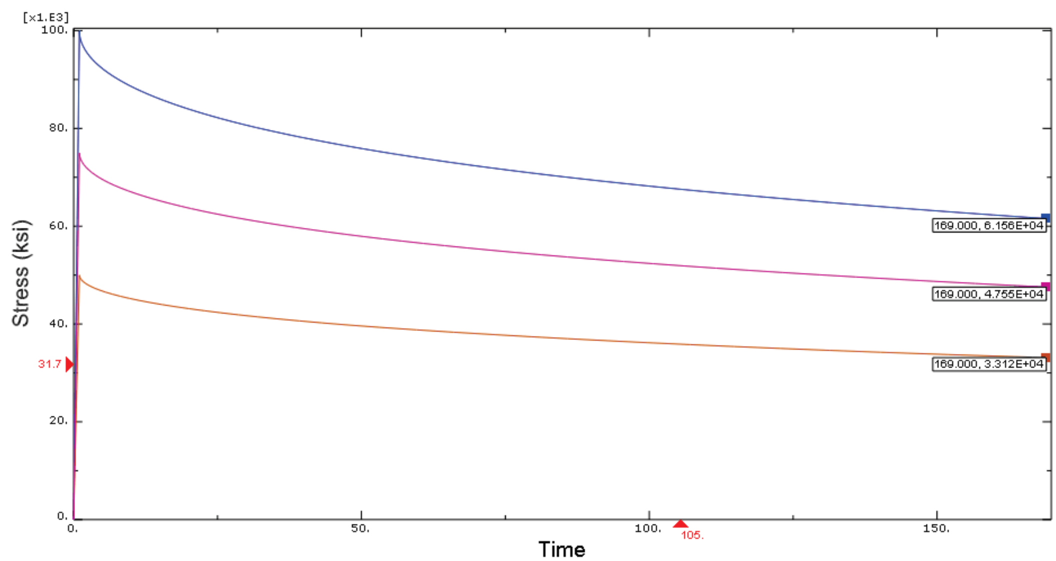

Figure 37 – Bolt/Cylinder Simple FEA Model Bolt Stress vs. Time Results

Subsequent FEA models of standard ASME B16.5 flange joints were constructed in order to examine the expected reduction in bolt stress for an initial assembly bolt stress of 52.5 ksi and a steady state uniform operating temperature of 850°F for 217 hours. The selection of the flange size and class for each analysis was made with consideration of the relative strength and failure mode (or dominant stress at gross plastic deformation) of the flange, established from previous work [62]. The importance of selecting flanges with different strength characteristics is to highlight the effect of the relative strength of the joint components and also the effect of the location of the dominant strength of the flange itself (flange ring, hub or hub/shell junction). Without consideration of these effects, it is likely that the methodology may be suitable only for a certain joint configuration and may not be applicable for other configurations.

The modeled flanges were as listed below, with the FEA mesh plot and the creep strain after 217 hours plot figures referenced after each joint case. In all cases the modeled gasket was a solid rigid element, which creates a symmetry plane on the raised face, while allowing portions not in compression to come away from the plane.

Case 1: NPS 2in., cl.900 Flange – Figure 38

Case 2: NPS 3in., cl. 150 Flange – Figure 39 and Figure 40

Case 3: NPS 3in., cl. 300 Flange – Figure 41 and Figure 42

Case 4: NPS 6in., cl. 150 Flange – Figure 43 and Figure 44

In the cases shown, it can be seen that the effect of relaxation is distributed between the bolt and the flange, depending on the relative strength of the bolt to the flange. This is due to the fact that although the bolt is the most highly stressed component, the CrMo creep strength of the bolt is better than the flange.

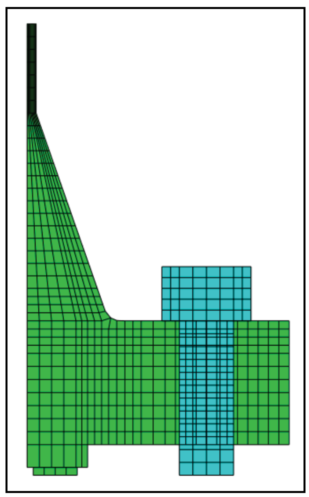

Figure 38 – NPS 2, cl. 900 FEA Model

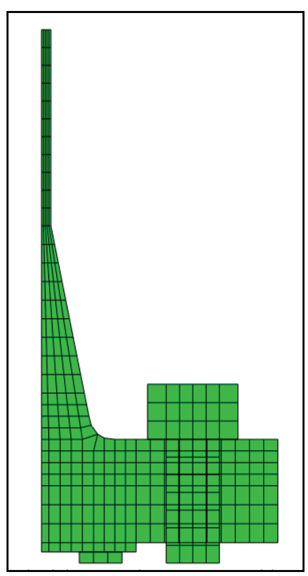

Figure 39 – NPS 3, cl. 150 FEA Model

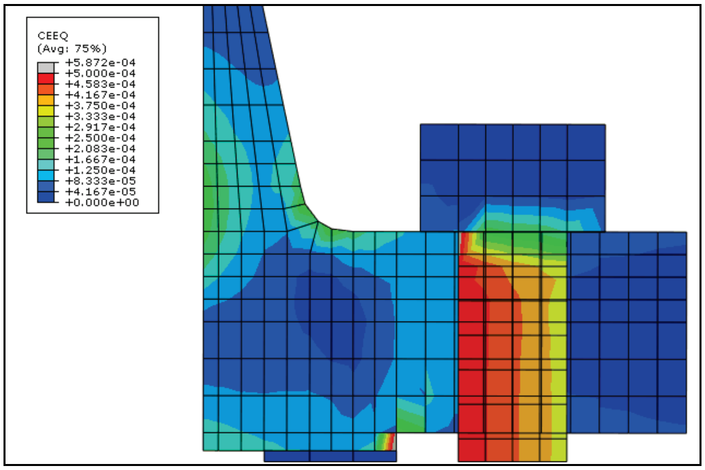

Figure 40 – NPS 3, cl. 150 Creep Strain @ 217 hrs

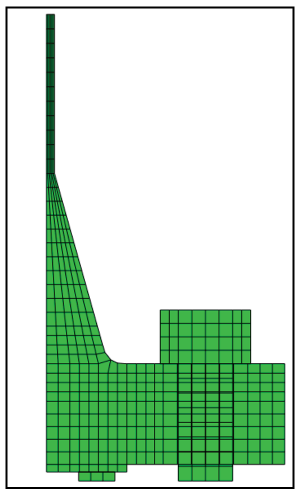

Figure 41 – NPS 3, cl. 300 FEA Model

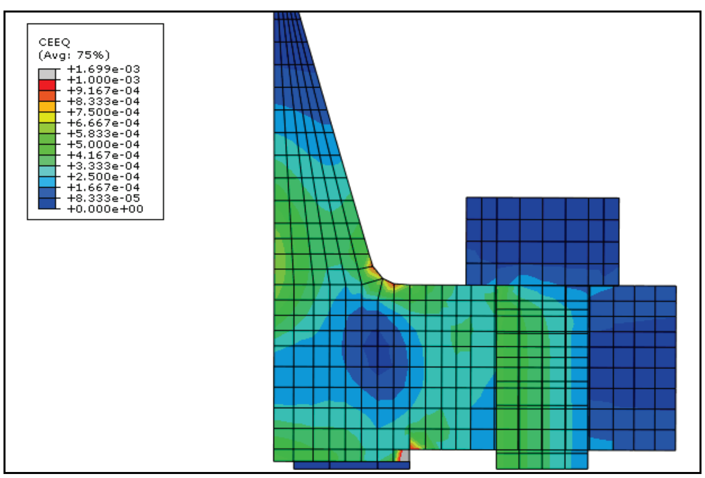

Figure 42 – NPS 3, cl. 300 Creep Strain @ 217hrs

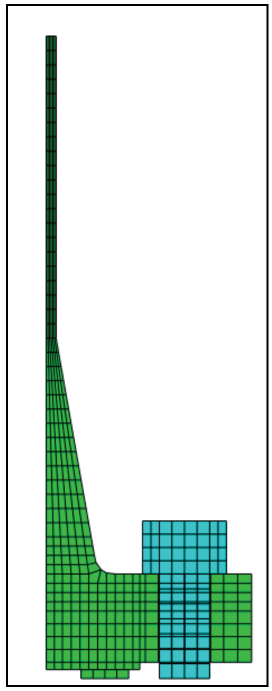

Figure 43 – NPS 6, cl. 150 FEA Model

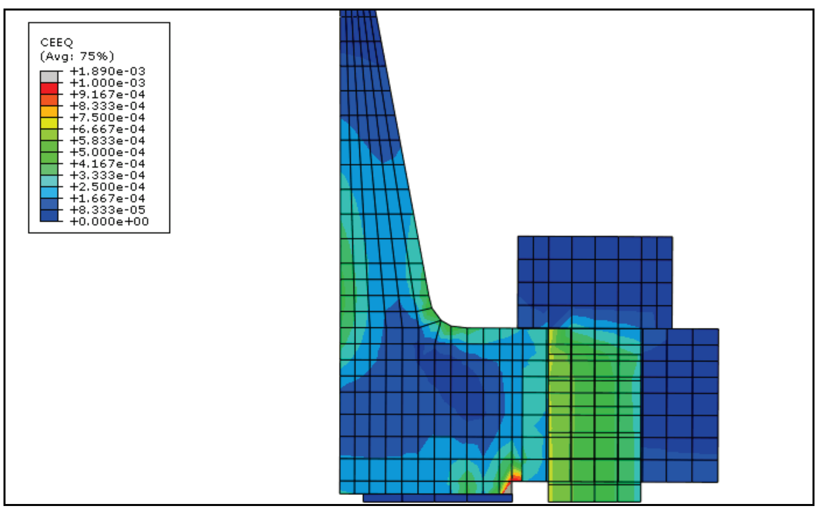

Figure 44 – NPS 6, cl. 150 Creep Strain @ 217 hrs

3.3 Approximation of Creep/Relaxation Behavior Using Code Stresses

The application of the code stresses to flange creep is complicated by two things; the fact that the code stresses are component directional stresses and also by the fact that the strength of the flange to resist rotation is primarily in the bending of the flange ring and distortion of the flange hub. The directional creep stress results may be used to calculate creep strain rates, but generally it is considered more accurate for multi-axial stress stats to use the Mises stress distribution. The strength of the flange being associated with multiple portions of the flange means that creep associated with, say, the longitudinal stresses on the outer fiber of the hub will not result in a proportional loss of strength (deformation) of the flange itself.

However, it is possible to make some simplifying assumptions that enable calculation of the effects of creep on the joint interaction, without needing new or overly complex sets of equations. This involves assuming that there is a predominant stress direction in the flange that can be treated as a uniaxial creep problem that will relate directly to the flange strength. For flange rings without a hub and not connected to a shell, the predominant stress direction (only one checked by code) is the tangential stress (hoop direction) in the flange ring. For other flanges, this stress direction is also associated with much larger portions of the flange ring and hub than other stresses. For example, the peak longitudinal stresses calculated for the hub are local to the outer surface and go from tension to compression across the thickness of the hub. In addition, the radial stress direction is local to the region only around the connection between the hub and flange ring, whereas the tangential stress acts as a bending stress across the entire flange ring and also as a tensile stress in both the hub/flange ring and hub/shell junction locations. The ASME code calculations only take into account the hub/flange ring location; however, it has been found that in some cases, for realistic assembly bolt loads, the hub/shell junction is the controlling location [62]. The method outlined in this document uses the current code calculated tangential stress (S_T) but it is recommended that the hub/shell location also be considered. This may be done by calculating it in accordance with the methods outlined in Brown [62] and substituting that stress into the equations if it is initially higher than S_T.

Using the above simplifications it is possible to determine relatively easily the bolt load relaxation with time by following an iterative (finite difference) approach with small increments of time. Such an approach may be achieved using a spreadsheet. The basis of the equations are that of the original Wesstrom work [63], which assume that the joint is deflection controlled once assembled, and the sum of any changes in deformation of components must be equal to zero. Therefore, the following equation can be found to apply.

$$\Delta l_{b_e} + \Delta l_{b_c} + \Delta l_{f_e} + \Delta l_{f_c} + \Delta l_{g_e} + \Delta l_{g_c} = \Delta l_{bi} + \Delta l_{fi} + \Delta l_{gi} \quad \textbf{(2)}$$

Where:

$$\Delta l_{bi} = \frac{\sigma_{bi} L_{bi}}{E_b} \quad \textbf{(3)}$$

$$\Delta l_{fi} = \frac{\boldsymbol{h}_g \sigma_{bi}}{q_f} \quad \text{and} \quad q_f = \frac{L g_0^2 \boldsymbol{h}_0 E_f}{0.91 V A_b n_b \boldsymbol{h}_g} \quad \text{for a hubbed flange} \quad \textbf{(4)}$$

$$\Delta l_{gi} = \frac{\sigma_{bi}}{q_g} \quad \text{and} \quad q_g = \frac{\pi A_g E_g}{4 t_g A_b n_b} \quad \textbf{(5)}$$

$$\epsilon_{f_c} = \epsilon_{f_{c0}} + \Delta\epsilon_{f_c} \quad \text{and} \quad \Delta\epsilon_{f_c} = A.\, S_T^N .\, \epsilon_{f_{c0}}^m .\, dt \quad \textbf{(6)}$$

$$\sigma_b = \frac{\Delta l_{bi} + 2\Delta l_{fi} + \Delta l_{gi} - t_g \epsilon_{g_c} - 2\epsilon_{f_c} \frac{E_f Q}{q_f} - \epsilon_{b_c} L_b}{\frac{L_b}{E_b} + 2\frac{h_g}{q_f} + \frac{1}{q_g}} \quad \textbf{(7)}$$

$$Q = \left[\left(\frac{Y}{t^2 B} + \frac{1.33\, te + 1}{L t^2 B}\right) A_b n_b\right]^{-1} \quad \textbf{(8)}$$

Using the above equations, the creep of the flange, bolts and gasket can be calculated for each increment of time and the residual bolt stress and creep strain at the end of that increment is then the input to determine the amount of creep strain that will occur in each of the components in the next increment of time. Care must be taken with this approach that the time steps are sufficiently small, particularly in the initially highly non-linear time period, such that the creep strain estimates are accurate.

Using this approach, the equations were input into a spreadsheet and the resulting bolt stress versus time at 850°C for the SA105/SA193-B7 materials and NPS 3, cl. 150, NPS 3, cl. 300 and NPS 6, cl.150 flanges outlined in the previous section were compared to FEA results for the same configurations. The comparative graphs of bolt stress versus time are shown in Figure 45 to Figure 48. It can be seen that the agreement is remarkably good for the flanges studied.

Overall, the approach of determining the loss in bolt load, and, therefore, gasket stress, using a simplified closed-form solution based on ASME code equations appears sound and is relatively easily employed. However, comparison with FEA results using the same material properties only proves that the method has merit from a calculation perspective. If the material properties are inaccurate, then neither one of the calculation approaches will predict accurate bolt load loss.

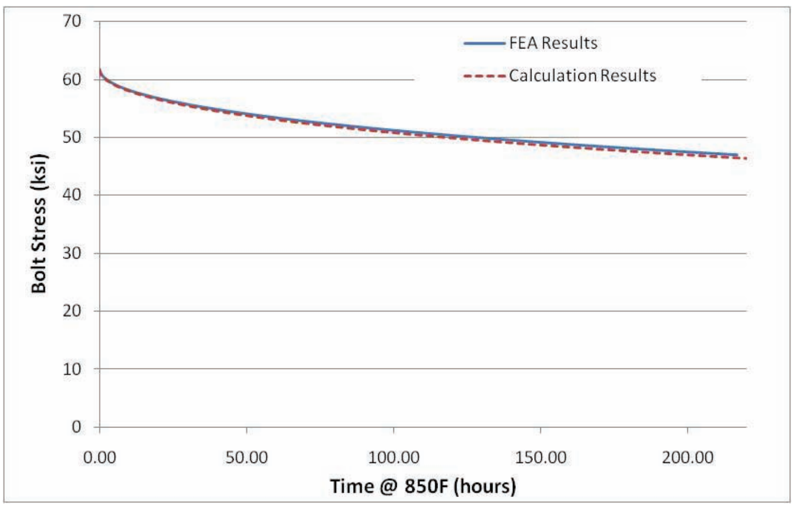

Figure 45 – NPS 3, cl. 150 Bolt Stress vs. Time

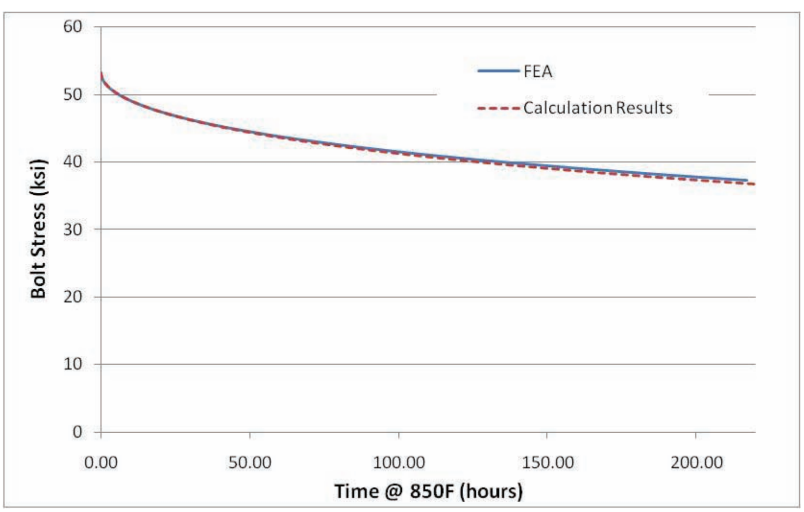

Figure 46 – NPS 3, cl. 300 Bolt Stress vs. Time

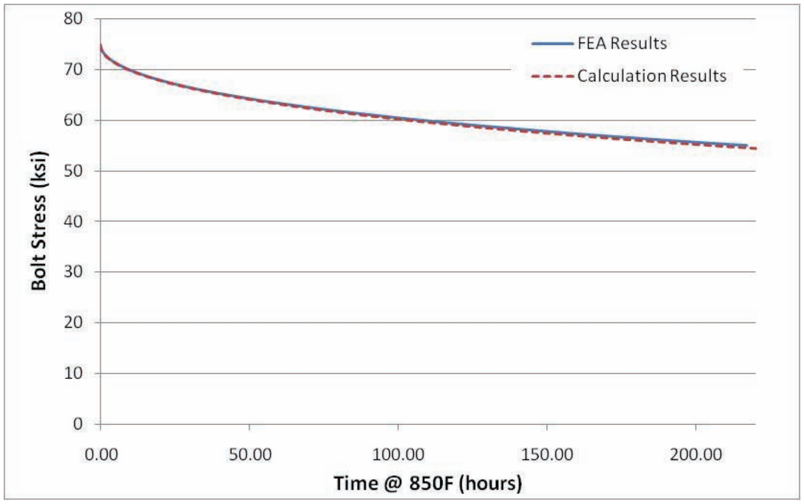

Figure 47 – NPS 6, cl. 150 Bolt Stress vs. Time

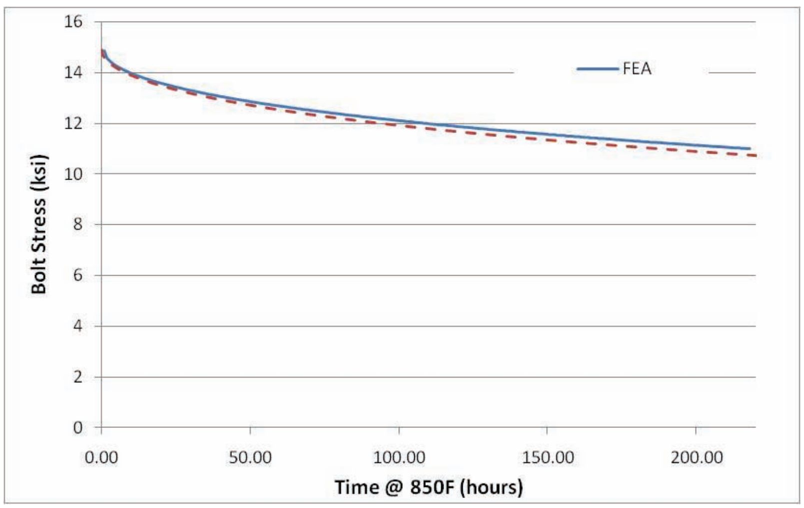

Figure 48 – NPS 2, cl. 900 Bolt Stress vs. Time

4 EXPERIMENTAL METHODS AND RESULTS

4.1 Experimental Methods

In order to examine joint relaxation, two series of tests were conducted. The first involved measurement of short term bolt load relaxation at relatively low temperatures when tightened on a hollow steel cylinder. The second tests involved relaxation measurements on the standard flange sizes listed in the previous section of this report. The purpose of both of these tests was to verify the effectiveness of both FEA and closed form calculations in predicting bolt load relaxation.

4.1.1 Bolt Relaxation

In these tests, a 7/8 inch SA193-B7 studbolt with SA194-2H nuts was assembled to varying preload values onto a hollow carbon steel (1020) cylinder. The studbolts were center-drilled at each end to enable ball-bearings to be used to ensure measurement alignment. A hardened washer was used under each nut and, in some cases, a single Key Belleville AFB-60 washer was also used under each nut (Figure 49). The arrangement was assembled at a controlled laboratory temperature, heated in an oven at 650°F for one week, removed, and then allowed to cool back to ambient. The bolt length was measured (Figure 50) in the initial unassembled, initial assembled, post-heating assembled and post-heating unassembled conditions. By subtracting the unloaded lengths from the loaded measurements, the bolt load relaxation could be determined. In each test run, several unassembled studs were also subjected to the same heat treatment and length measurement procedure in order to assess possible inaccuracies in the method, such as oxide build-up on the center-drill surfaces.

Figure 49 – Bolt Load Relaxation Arrangement

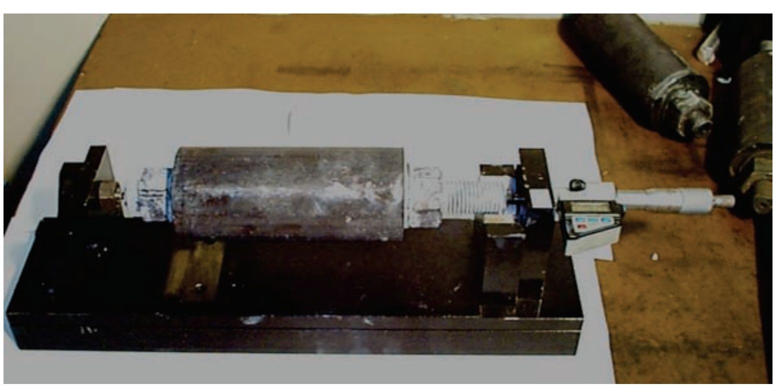

Figure 50 – Length Measurement Arrangement

4.1.2 Joint Relaxation

In order to examine the effect of joint component interaction on bolt load relaxation, several flanges of each flange size were subjected to the following testing sequence.

- Initial Machining (Figure 51):
  1) Both ends of each stud were skim cut and center drilled.
  2) The back of each flange ring in 8 locations was center-drilled.
- Initial Measurement and Assembly
  3) Flanges were assembled with studs to a torque of 10 ft.lb. Two of the 3in. cl.150 flanges had additional Bellville washers fitted (one with three washers under one nut [joint G-H] and the other with three washers under each nut, for a total of six washers [joint E-F]). Nuts and Studs were liberally lubricated with Loctite Nickel anti-seize prior to assembly.
  4) The metal temperatures of the studs and flanges were recorded
  5) The initial length of each stud and initial distance between each pair of flange center-drill holes was measured.
  6) The joints were assembled to a specified torque.
  7) The metal temperatures of the studs and flanges were recorded.
  8) The final length of each stud and as-assembled distance between each pair of flange center-drill holes were recorded.
- Initial Stage Creep/Relaxation
  9) The joints were stacked in a furnace and 2 thermocouples were attached.
  10) Furnace was heated rapidly and continued steady-state heating at for 12 hrs at 850 ± 5°F.
  11) Furnace was stopped and doors opened to allow rapid air cooling with continued thermocouple monitoring.
  12) Joints were removed from the furnace and, once at room temperature, the metal temperatures of the studs and flanges were recorded.
  13) Any oxide was lightly cleaned from center-drill locations.
  14) Length of each stud and distance between each pair of flange center-drill holes were recorded.
- Second Stage Creep/Relaxation
  15) Joints were re-stacked in furnace and 2 thermocouples attached.
  16) Furnace was heated rapidly and continued steady-state heating at for 48 hrs at 850 ± 5°F.
  17) Repeated activities 11 to 14.
- Final Stage Creep/Relaxation
  18) Joints were re-stacked in furnace and 2 thermocouples attached.
  19) Furnace was heated rapidly and continued steady-state heating at for 156 hrs at 850 ± 5°F.
  20) Repeated activities 11 to 14.
- Final Measurement

21) Loosened and then re-tightened to 10 ft.lb each nut in a circular pattern until the torque required to loosen is close to 10ft.lb. for each nut.

22) Length of each stud and distance between pairs of flange center-drill holes was recorded.

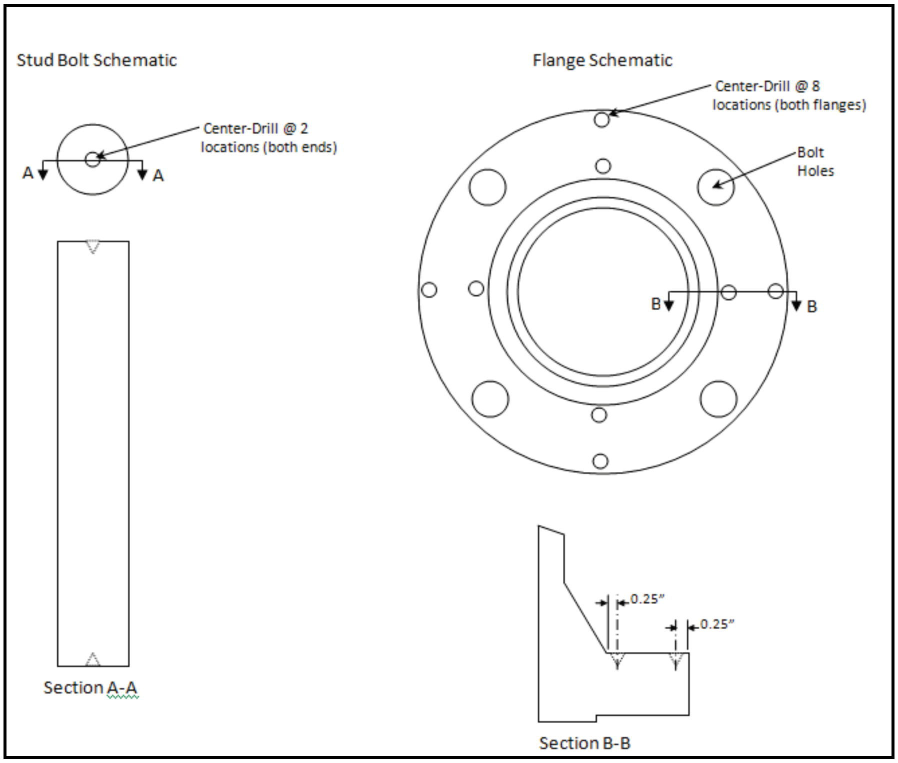

Figure 51 – Flange Joint Drilling Arrangement

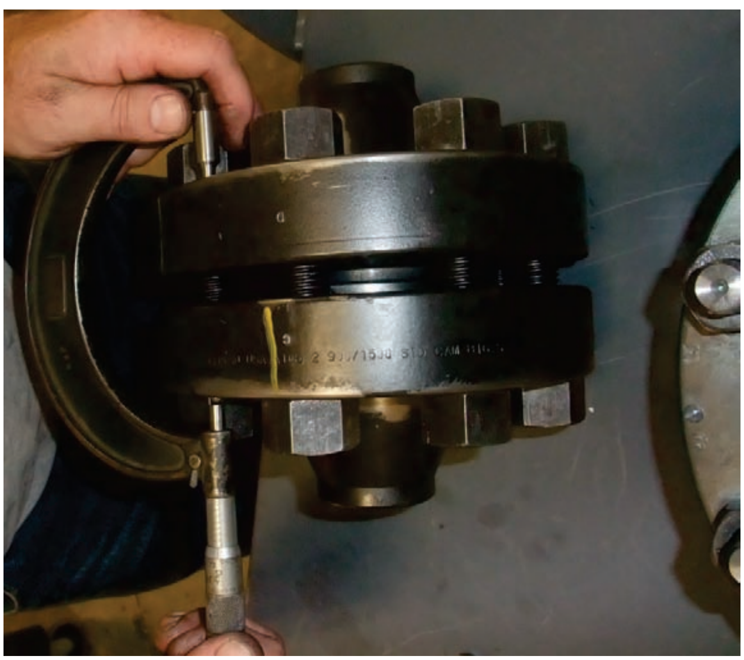

Figure 52 – Joint Measurement Arrangement

Figure 53 – Assembled Joint

Figure 54 – Assembled Joint

4.2 Experimental Results

4.2.1 Bolt Relaxation

The results of the bolt load relaxation tests versus initial bolt load as a percentage of ambient bolt yield are shown in Figure 55 as a percentage of the initial assembly load and in Figure 56 as a percentage of ambient nominal bolt yield (105ksi). It can be seen that although the higher loaded bolts lose a greater percentage of their initial load, in all cases the overall percentage of bolt load remaining is higher for the higher loaded bolts. This confirms the need to account for strain hardening in the assessment of bolt load relaxation. The Belleville washers did not appear to improve the bolt relaxation, as the results with Belleville washers are comparable to those without. In addition, it can be seen that the load relaxation predicted by the CrMo and CS properties determined in the previous section both considerably underestimate the bolt relaxation. In theory, the CrMo calculation results should be directly comparable to the test results, but they underestimate the relaxation that occurred by more than 20% of the initial bolt load. This is a significant non-conservative result that highlights the difficulties in taking relaxation data from one source and applying it to flanged joint relaxation. The test results were obtained at 650°F, which is 200°F below the closest ASTM DS-60 test results used to determine the material creep coefficients. The ASTM DS-60 test results are also at a stress level of around half that used in the bolt load relaxation tests. In addition, the results from the obtained coefficients and the DS-60 test results are not very accurate in the 168 hour timeframe and actually would indicate a significant underestimation of the relaxation is within this timeframe. Another interesting conclusion is that, as expected, when the bolt is re-used, its relaxation is significantly less than when tested in the as-new state.

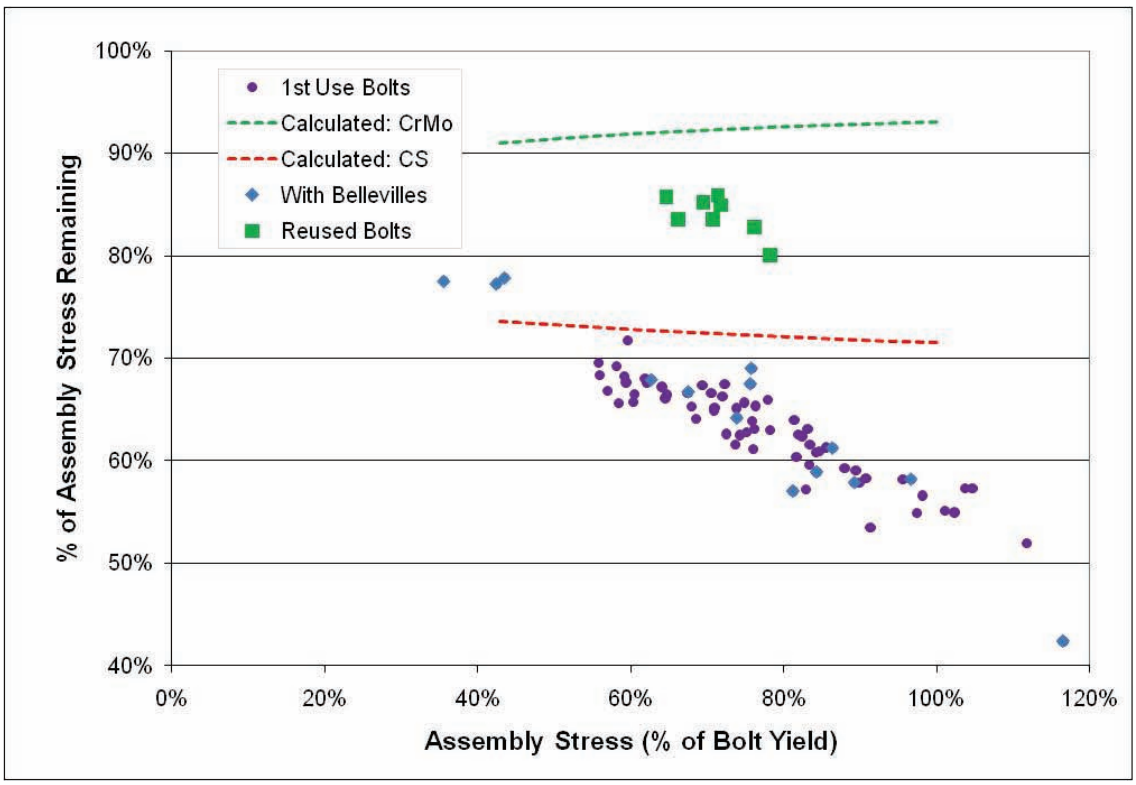

Figure 55 – Bolt Load Relaxation vs. Assembly Load (% of Assembly Load)

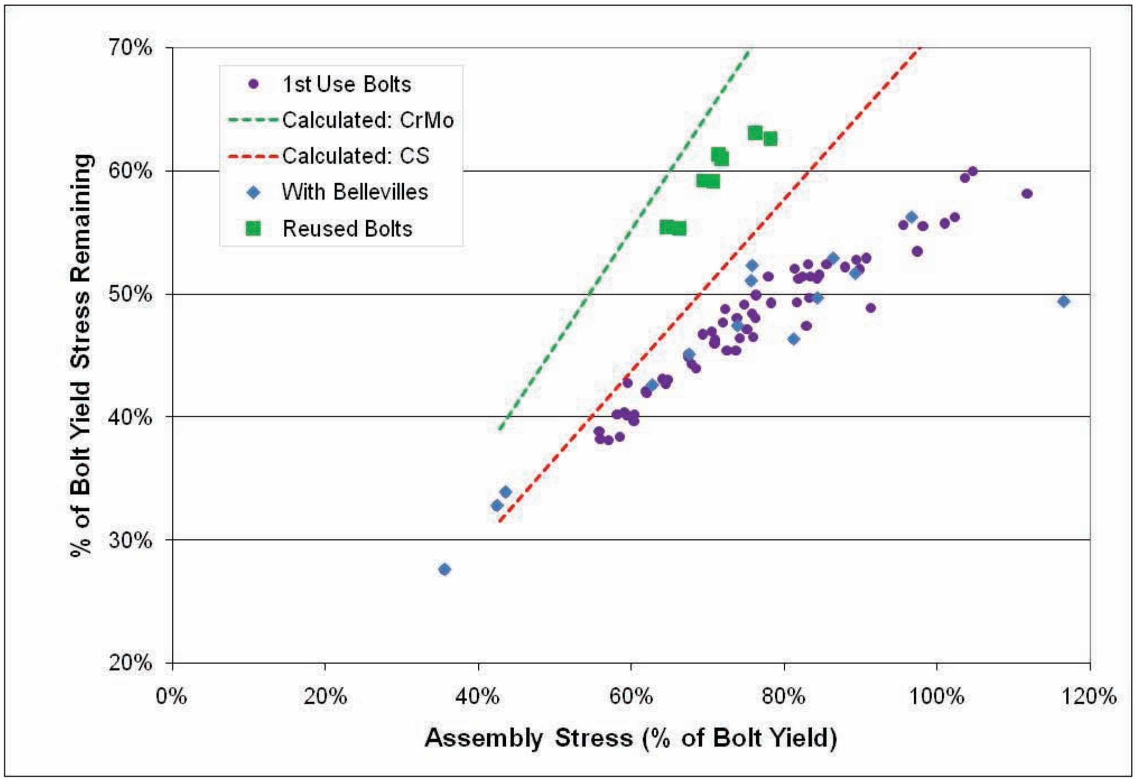

Figure 56 – Bolt Load Relaxation vs. Assembly Load (% of Ambient Bolt Yield Stress)

4.2.2 Joint Relaxation

The methods of bolt and flange deformation measurement were only accurate enough to offer an order of magnitude or directional estimate of the actual deformation occurring. In particular, measuring the flange deformation is rather complex due to the likelihood of relative movement between the flanges and across the different quadrants of the flanges. However, a comparison between the calculation method presented in the previous section and the test results will give an indication of whether the methods appear applicable. Given the poor relationship between the bolt load relaxation tests and the calculation method, it was expected that the bolt load relaxation would also be underestimated in these tests as well.

The NPS 2, cl. 900 results for relaxation tests on three joints are shown in Figure 57 and Figure 58. The bolt deformation, which indicates the sum of the bolt creep and elastic elongation during the tests and the final bolt creep by comparison between the final unloaded measurement and the initial measurement, are shown in Figure 57. It can be seen that the test measurements indicate very little bolt length change during the test and a final overall bolt elongation due to creep of 0.003 inches. In this case, the calculations show poor agreement, with the bolt length expected to decrease during operation and the final increase in bolt length due to creep to only be in the order of 0.001 inches, three times less than that measured in the test results. It is therefore evident that the analytical result would not have been a good predictor of the residual load in this case. The flange deformation results, which show the sum of the creep and elastic deformation of the flange during the test and the final creep deformation by comparison between the initial measurement and the final, are shown in Figure 58. It can be seen that test results are very erratic, indicating some sort of measurement error. In spite of this, it is still apparent that the analytical solution once again under-predicts the level of relaxation.

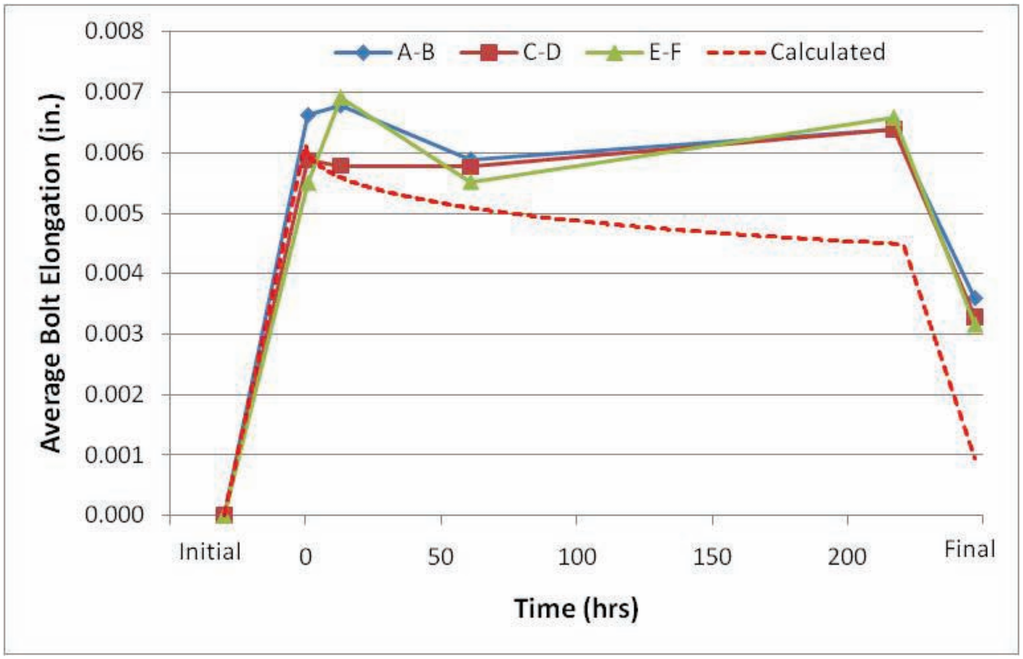

Figure 57 – NPS 2, cl.900 Bolt Deformation Results

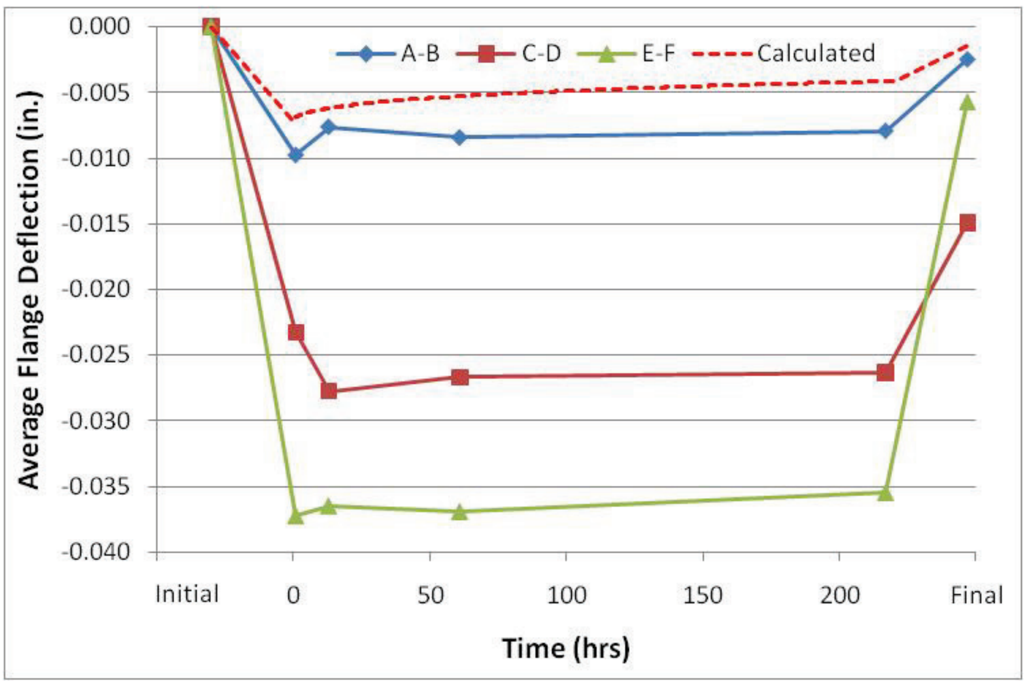

Figure 58 – NPS 2, cl.900 Flange Deformation Results

The NPS 3, cl.150 results are shown in Figure 59 and Figure 60. It appears that the bolt elongation is under-predicted by the analytical results by a factor of about half. In addition, the presence of Belleville washers has significantly increased the final bolt creep that occurs, by a factor of 2 to 3, which means that the additional rebound offered by the washers is offset in part by the additional increase in bolt length due to their presence. In addition, it can be seen that both of the flanges with Belleville washers had higher residual deformation than the flanges without Belleville washers, further indicating that the effectiveness of the Belleville washers will be reduced due to higher flange relaxation. There also appears to be a zero shift with the test results, as a positive flange deflection after the test is not considered possible. Therefore, a comparison between the analytical solution and the test results for the flange deformation is not practical, although the overall trends do seem to agree reasonably well.

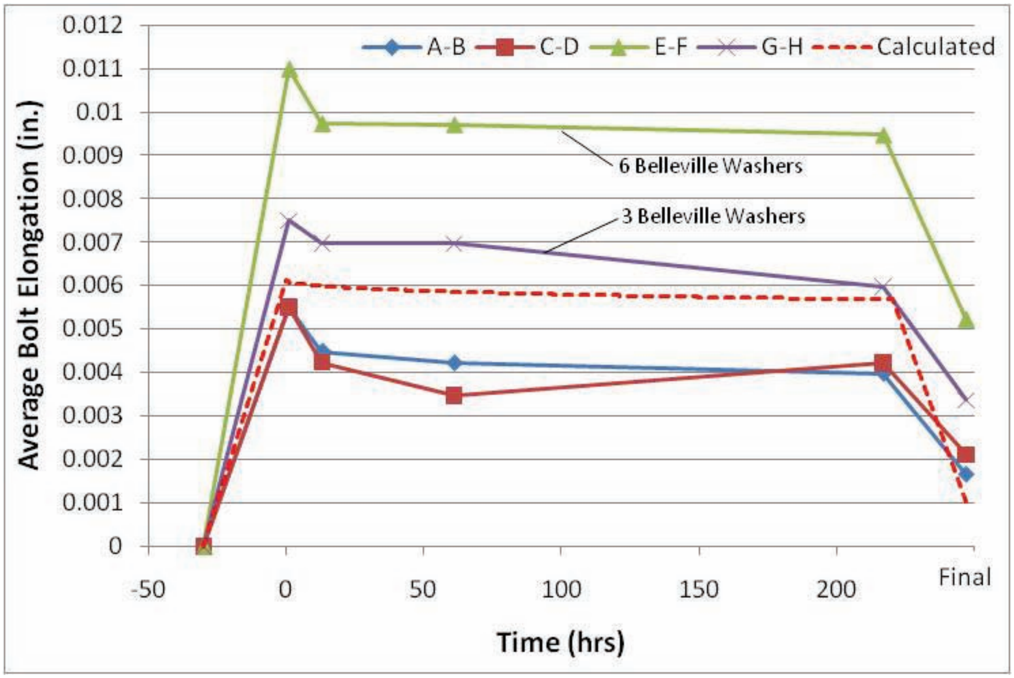

Figure 59 – NPS 3, cl.150 Bolt Deformation Results

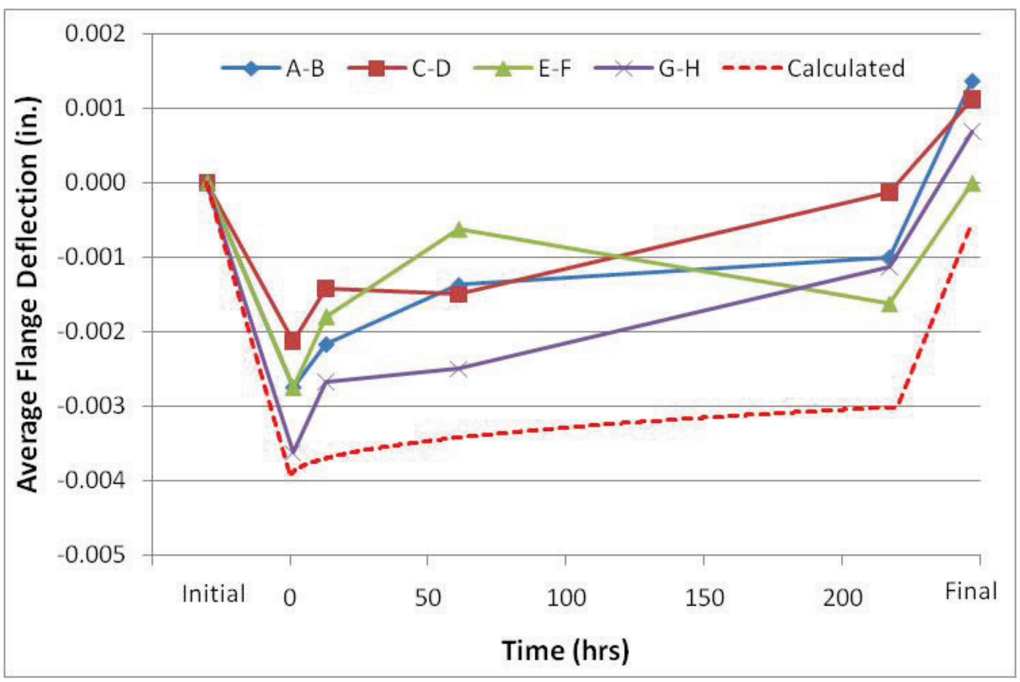

Figure 60 – NPS 3, cl.150 Flange Deformation Results

If the initial assembly and final bolt load are examined for each of the flanges (Figure 61), it can be seen that all cases lost about 60% of the initial bolt load, irrespective of whether Belleville washers were installed or not. This is due not only to the additional bolt and flange deformation mentioned previously, but to creep/relaxation and changes in stiffness of the washers themselves. This is evident in load-deflection tests run on the 3-stack and 6-stack washers (Figure 62 and Figure 63, respectively). It can be seen that the as-received unloading deflection curve has a slope of about 82% of theoretical (based on the manufacturer specification for load and deflection) when tested new for 3 washers in series. After the creep tests, the unloading slope is reduced to only about 63% of the theoretical value. For the 6-washer stack, the percentage of theoretical for before and after the creep test are 70% and 42%, respectively. In both cases, it is evident that the washers themselves have been affected by the time at temperature, but still remain functional. Therefore, the fact that they had little effect on the remaining bolt load is primarily attributable to the higher levels of bolt and flange deformation when Belleville Washers are used.

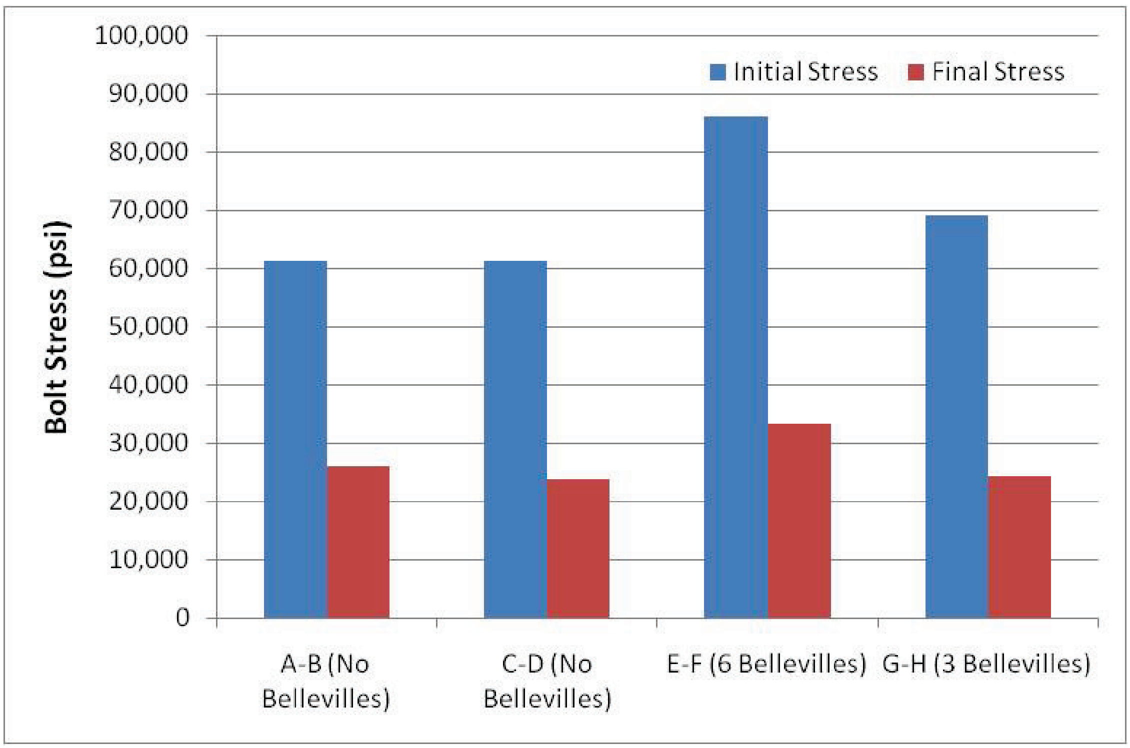

Figure 61 – NPS 3, cl.150 Remaining Bolt Stress Results

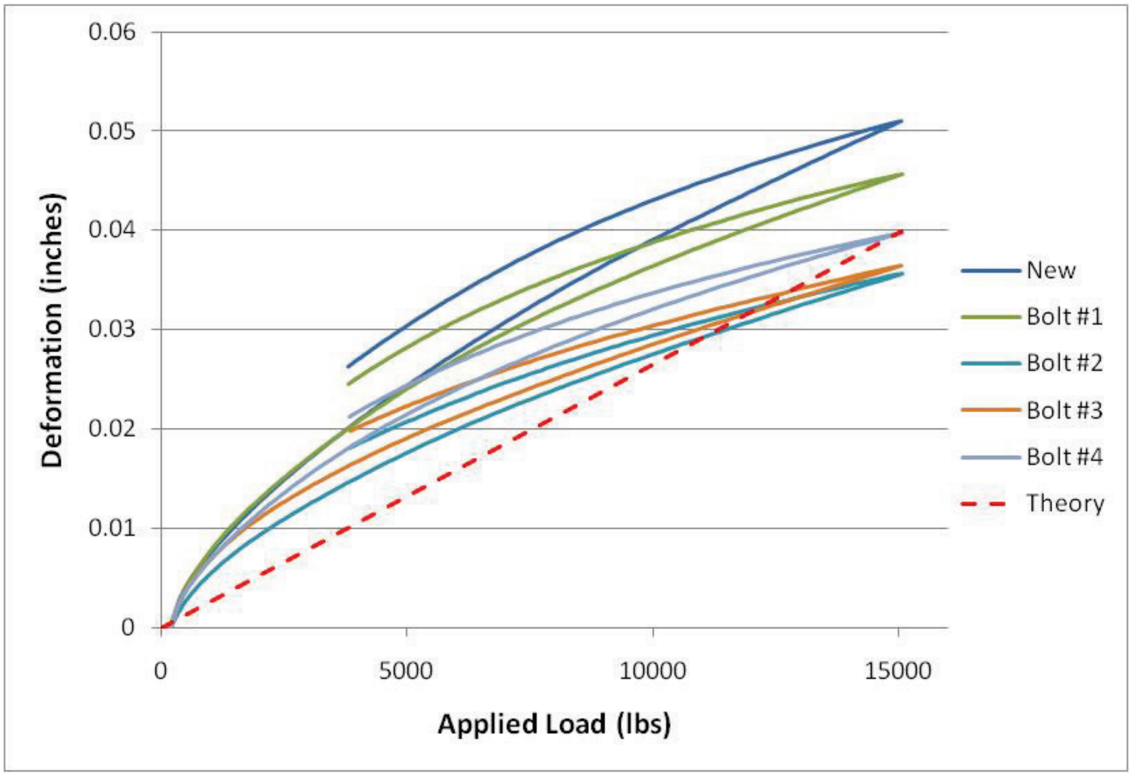

Figure 62 – 3 Belleville Washer Stack Load-Deformation Results

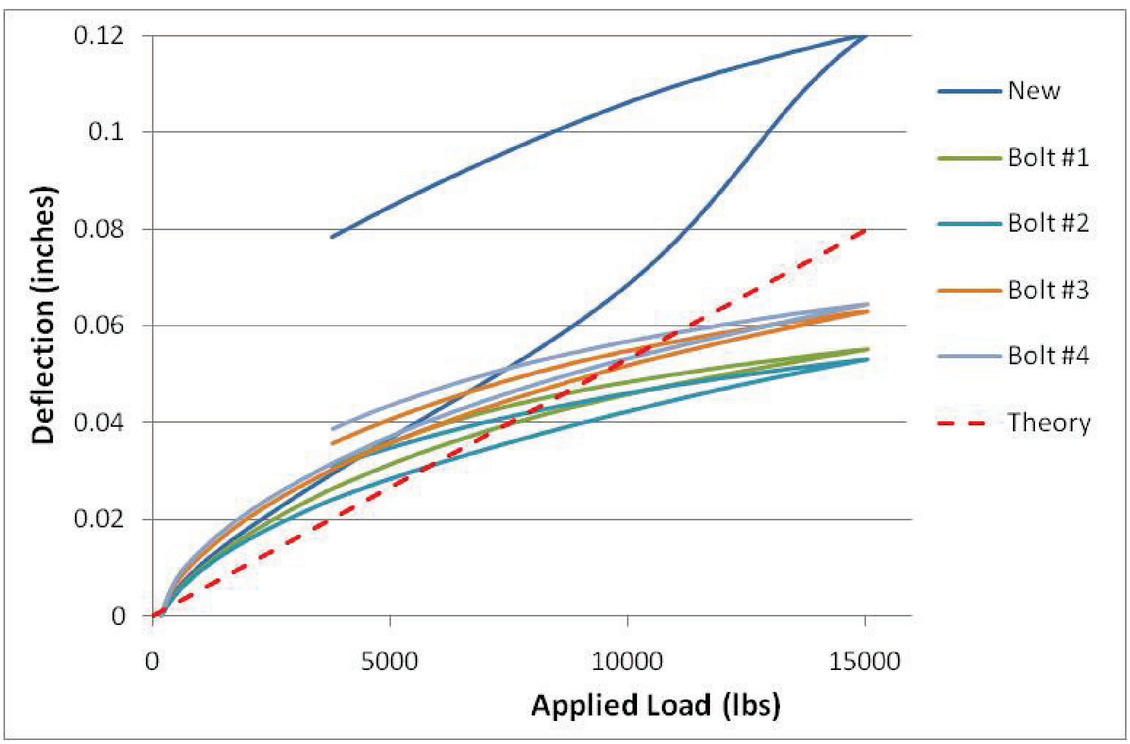

Figure 63 – 6 Belleville Washer Stack Load-Deformation Results

The NPS 3, cl.300 results are shown in Figure 64 and Figure 65, it can be seen that the bolt creep is under-predicted by a factor of slightly more than 3 in this case. There is sufficient spread in the flange deformation results that it is possible that the flange creep prediction is reasonable, however given the inaccuracy of the bolt load prediction, this is probably more due to chance than anything else. The NPS 6, cl.150 results are shown in Figure 66 and Figure 67. In this case, the analytical solution once again appears to be of the order of 2 to 3 times less than the test results indicate.

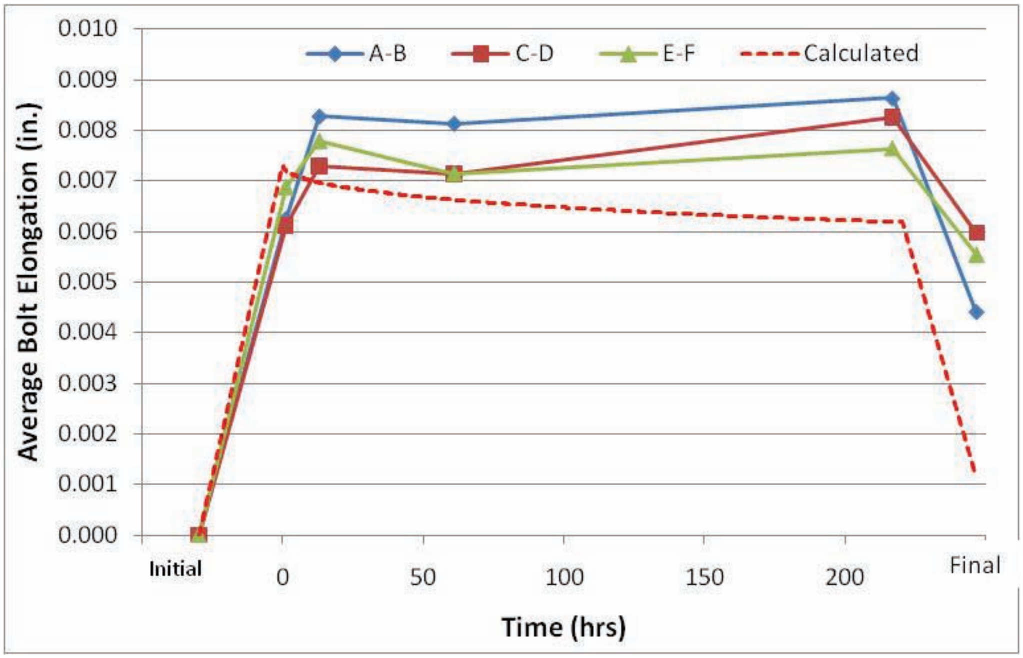

Figure 64 – NPS 3, cl.300 Bolt Deformation Results

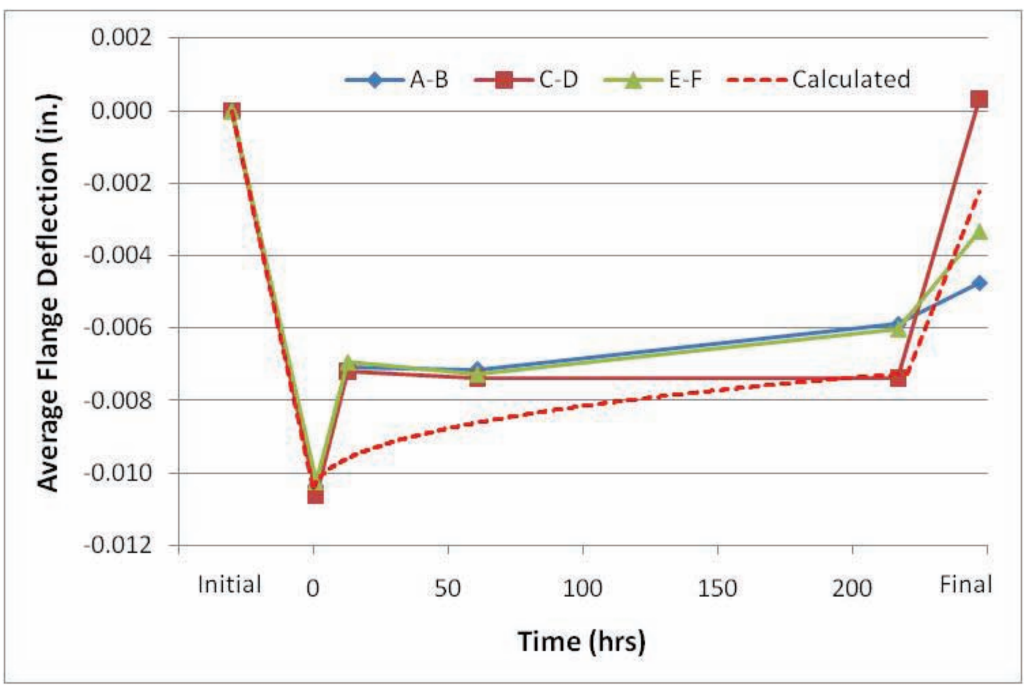

Figure 65 – NPS 3, cl.300 Flange Deformation Results

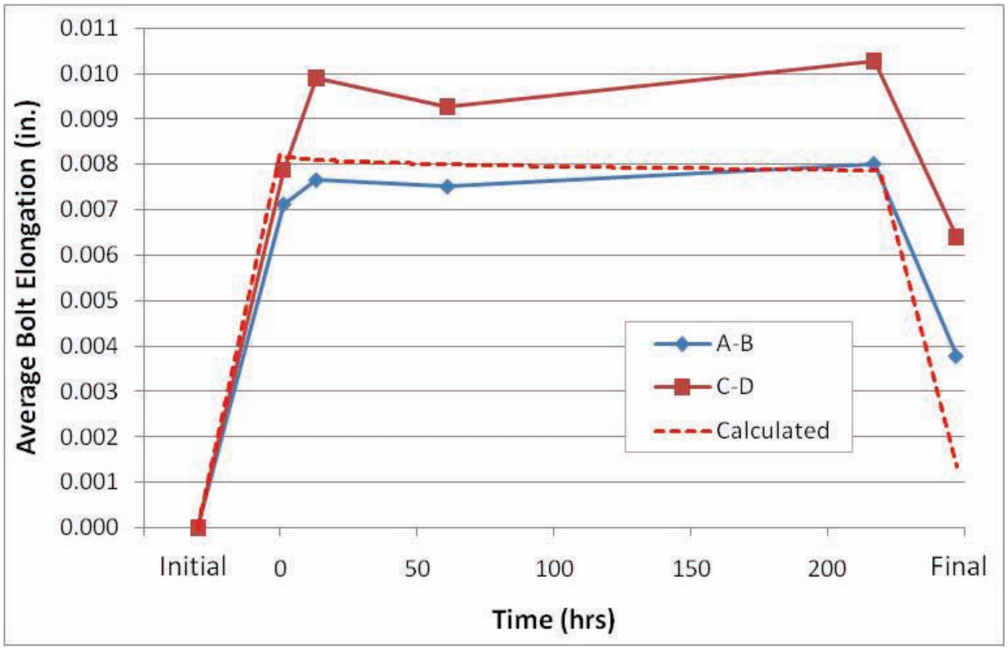

Figure 66 – NPS 6, cl.150 Bolt Deformation Results

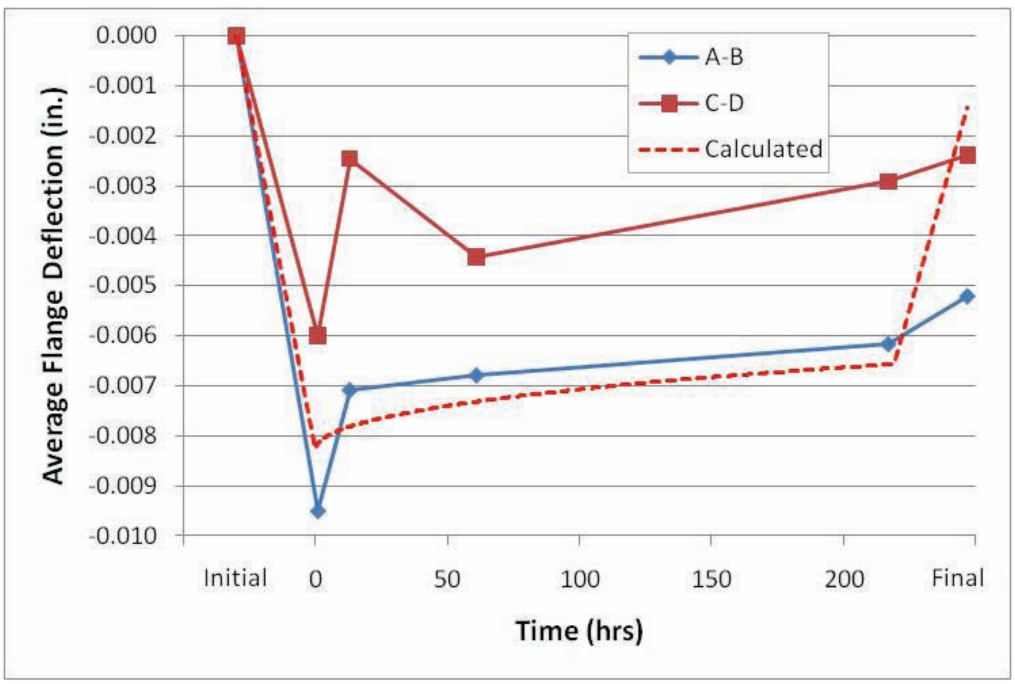

Figure 67 – NPS 6, cl.150 Flange Deformation Results

In all of the cases, however, it appears that the trends predicted by the analytical solution match reasonably well with the test results and therefore it is reasonable to assume that the differences between results are mostly due to the inaccurate material properties used. It is entirely possible that the material properties could be fine-tuned to better match the test results, however, this proves little, as what is required is a consistent method to get from uniaxial type relaxation test results for a given material to the calculation of flanged-joint behavior.

5 CONCLUSIONS

This project has demonstrated that it is possible to predict the creep/relaxation behavior of bolted flanged joints, however the major limitation in doing so remains the lack of suitable material data in the required stress and temperature ranges. It also appears that while the use of FEA for predicting flange behavior is becoming simpler and more rapid, it is possible to accurately predict the joint behavior using current code equations. Since it is easier to codify closed form solutions, it will be less likely that mistakes will be made in the implementation when compared to FEA. Since the analytical results are comparable, certainly within the accuracy of creep material properties and the accuracy required for prediction of flange leakage (life), it is concluded that including creep into code flange joint design is possible.

This study did not examine the effects of temperature variation in a flange (an un-insulated flange, for example) on the creep/relaxation behavior. However, given the variability of the results due to inaccurate material properties, the effect of the thermal gradients in the joint are unlikely to be as significant and, once material properties are improved, could easily be included by consideration of the actual operating temperatures of the components using methods such as that outlined in Brown [32].

The study also found, from experimental results, that Belleville washers seem to provide no benefit when used on flanges in a creep environment.

It should be noted that unless there is a material mismatch or poor design detail, creep material failure of a flange or bolt is not the most likely failure mode. The deformation associated with creep of the components will generally result in flange joint leakage prior to creep failure of the steel components.

6 RECOMMENDATIONS

It is recommended that further work be focused on determining the appropriate creep/relaxation behavior for commonly used flange and bolt materials in the full operating range of temperatures. This effort should focus on establishing relationships for new (first-use) materials, as calculations using the obtained relationships will be conservative for re-used materials.

A simple assessment of flange creep can be written into the pressure vessel and piping design codes and should be based on assessment of only the primary membrane stresses (bolts and flange hoop stresses). The creep strain should be applied as a component deformation in the joint elastic interaction calculations using a finite difference approach in order to determine how long the bolt load remains above a suitable limit to ensure joint sealing. However, before any such assessment method is included in the code, two significant hurdles must be overcome. The first is that a source of applicable creep relaxation data for common joint materials and also a standardized method for establishing similar data for other materials must be available. In addition, the current pressure vessel and piping design code approach of using a much lower design bolt load than commonly applied in the field during assembly must be addressed. Ideally, the design code would be changed to use more realistic bolt loads, thus facilitating the inclusion of creep relaxation assessment directly into the method without the need for additional factors to account for the fact that the design bolt load is of the order of half that which will be applied in the field.

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