Proceedings of PVP2006-ICPVT-11
2006 ASME Pressure Vessels and Piping Division Conference
July 23-27, 2006, Vancouver, BC, Canada

PVP2006-ICPVT-11-93731

REINVESTIGATION OF A HEAT EXCHANGE FLANGE LEAK

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ABSTRACT

In a paper presented in 1994 [1], the authors examined a heat exchanger flange to ascertain the cause for a leak. This examination was conducted using Finite Element (FE) analysis procedures. At that time, it was not practical to accurately model the interaction between the flanges and gaskets as a function of time and the resultant temperature. In the ensuing time period, the available FE technology has improved dramatically. Faster computers and new parallel solver technology allow modeling of the flange components that was not practical 10 years ago. In this paper, the authors will re-examine the exchanger system using current technology and discuss the improved insight that this new technology provides to the problem solution.

INTRODUCTION

In the paper referenced above, the investigation and analysis of a leaking heat exchanger flange was reported. At the time that this work was done, the use of FE as a tool in much of the non-nuclear pressure vessel industry was just beginning. The personal computers in use at that time were orders of magnitude slower than those in use today. The workstations in use today have orders of magnitude more memory than the computers of 1992 had file storage space. Additionally, new computing algorithms have been developed that greatly reduce FE solution times.

As an example, a steady state heat transfer analysis of the subject model took 35 minutes to complete in 1992. That same analysis consumes 0.135 minutes of clock time today. The decrease in static stress analysis computer run time is even more dramatic. A run that took 330 minutes in 1992 completes in just 0.66 minute today!

The question to be answered by the current work was: Does implementation of this increased solution speed offer us ways to gain more insight into the problem or is it just bigger, faster and better? Perhaps more importantly, can the newer tools provide us with a better understanding of the nature of the problem?

DESCRIPTION OF PROBLEM

The earlier paper examined a leak in the bolted tube sheet joints of a stacked pair of type 321 stainless steel TEMA type BEU exchangers in 1200psi and 700°F in Hydrogen and Oil service. A sketch of the system is shown in Figure 1. The joints are not insulated, as is common industrial practice for bolted tube sheet joints in high temperature hydrocarbon heat exchanger services. The shell and channel insulation stops several inches from the back of the flanges. Therefore, the joints are analyzed without the effect of the shell and channel insulation.

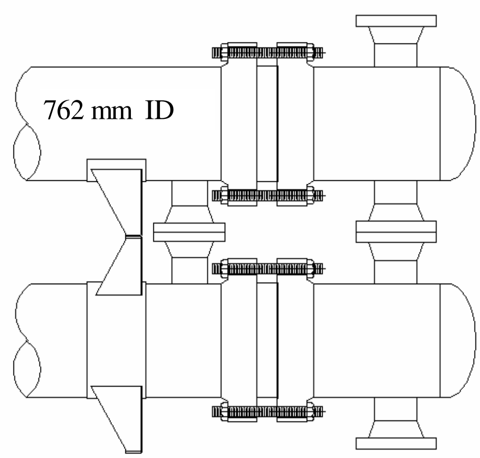

Figure 1 – Partial View of Exchangers

In 1992, it seemed clear that the stainless steel bolts were stretching and causing the joint to become loose upon shutdown. Furthermore, it was suspected that differential thermal growth between the tubesheet and the flanges was "scuffing" the spiral wound gasket, further exacerbating the leaking potential.

As stated in the 1994 paper the exchanger shell and channel flanges were found to be correctly designed to the 1992 to the then current ASME Section VIII Division 1 Appendix1 and had been successfully hydrotested in the fabricator's shop. Nonetheless, the flange leaked during initial operations and after three re-gasketing efforts. It became obvious that the stainless steel bolts had stretched in operation; each time the exchanger was cooled to ambient, the studs were extremely loose. The nuts were found to be only hand tight although the studs had been adequately torqued during assembly. The spiral wound gaskets displayed distress in the chevrons and radial scrapes and indentations in the gauge ring that were labeled as scuffing.

This scuffing was considered to be due to a differential movement of the mating gasket seating surfaces. However, it was not readily apparent if the stud stretching and the gasket scuffing were related or were due to unrelated drivers. The 1994 paper described the investigation to gain an understanding of the causes for these two conditions and reported the analysis results of steady state and snapshots of transient conditions during start-up and shut-down. The 1994 paper also included comparison of field data to analysis results and details of the retrofit of the closure that successfully eliminated the stretching of the studs and the gasket failure.

Due to the lengthly runs times involved, the analyses conducted were either steady state thermal or linear static stress analysis. The system was evaluated under a series of loading conditions a number of assumptions were made concerning the likely transient conditions. In the end, it was concluded that both bolt stretching and gasket scuffing were ocuring and measures were taken to eliminate both.

In this paper, the authors will reexamine the old model and apply transient and nonlinear techniques that were not available at the time the original work was done. The current focus is on the transient temperatue distribution as it affects both the bolt load and the gasket scuffing.

FE MODEL

Figure 2 illustrates the FE model used for the analysis. By today's standards, it is relatively small. It has approximately 9,000 nodes defining 7,000 linear brick elements. Nonetheless, the mesh is of sufficient density to reflect accurate deflections and bolt stresses.

The outer components in Figure 2 are the flanges. The tubesheet is between the two flanges, separated by a pair of spiral wound gaskets. The gasket and incorporated gage ring are shown in Figure 3. Note that the field of the tubesheet was simulated with a material using a Young's Modulus and density reduced by 45% to simulate the holes in the tunsheet.

THEREMAL ANALYSIS

An initial steady state thermal analysis of the model was conducted using the same temperatures and heat transfer coefficients as were used in the original model. The temperatures computed were matched those from the original model. Since these temperatures also matched the field measured values, the analysis was then conducted as a transient thermal analysis. It was determined that the model reached steady state conditions in approximately 10 hours. A final thermal model was then run with the heating conditions on for 10 hours and then removed to allow the vessel to cool.

Figure 4 illustrates the temperature history at various points on the model. As may be seen, different portions of the model heat, and to a lessor extent, cool at different rates. This difference is especially evident between the bolts and the other components.

Figure 5 plots the temperature differentials between various components. The difference in the magnitude of the differential between the tubesheet and flange on the shell and channel sides was anticipated. The differing times at which they occur were not anticipated.

The overall temperature differential between the bolts and the other components had been measured in the field and thus came as no surprise. In the earlier paper, the authors had postulated that there might be a "thermal lag" that caused the bolts to relax their tension. This thermal lag can be seen as the difference between the peak and the steady state conditions of the bolt differential in Figure 5. At the time of the original analysis, the authors concluded that a lag on the order of 20 degrees (as shown) would result in a bolt stress reduction on the order of 8,700 psi.

STRESS ANALYSIS

The results of the thermal model were used as the input for a stress analysis of the model incorporating a non-linear thermoplastic materal model. A bi-linear material stress model was employed with the tangent modulus set to 10% of the original modulus. This is a reasonably accurate model of the actual material out to a strain of ~4% [2]. The material properties (E, Y and α) were all set as variables of the material temperature.

This analysis was run to model time of approximately 11 hours. This was sufficient to observe both the bolt elongation and the gasket scuffing. Figure 6 illustrates the axial stress history for the bolts. (Note that the time scale changes from the thermal to the stress results graph.) The indicated stress relaxation is approximately 13,000 psi. This is higher than was expected based on the earlier linear analysis. It should be noted that the bolts were initiated with a preload of 18,000 psi, as indicated on Figure 6.

Figure 7 illustrates the average strain in the bolts as a function of time. Here we can see that the maximum strain was approximately 0.003%. Both the stress and the strain in the bolts were well over the elastic limit, indicative of yielding. In Figure 8 shows a plot of the bi-linear stress-strain material model that was used. If we start on this curve at a value of ~0.003 in/in and travel back parallel to the initial modulus, we arrive at a residual strain of approximately 0.0018 in/in. Based on an initial length of 20", this represents an elongation in the bolts of approximately 0.036". The elongation due to the 18,000 psi preload was only 0.0124". Thus, it is clear that the model correctly predicted that the bolts would be loose upon cool-down.

SCUFFING

As was illustrated in Figures 4 and 5, the temperature of the tubesheet increases faster than that of the flanges. In the first few hours of heating, a difference in temperature on the order of 160-170 Deg. F is observed. Under steady state operating conditions, a temperature difference on the order of 50-60 Deg. F persists. Upon shut-down, the temperatures come together again.

Based on these temperature differentials, we would expect a differential in thermal growth between the tubesheet and flanges. This differential growth imposes a scuffing load on the gasket between the components.

In the model, the authors did not allow the gasket to slide on the tubesheet or flanges. Simulating this motion would have required the use of contact surfaces and resulted in a very significant increase in processing time. Instead, the shear modulus of the gage ring was set to an artificially low level to allow motion. Unfortunately, the shear modulus of the gasket itself was not altered and it dafaulted to a much higher value than was set for the gage ring. The result was that the gasket restrained the differential growth in the model.

Figure 9 illustrates the differential radial growth (gasket scuffing) observed in the transient analysis model. However, the model did not include an adequate approach to the frictional conditions that occur at the gasket-to-seating surface interface. The gasket differential sealing surface radial movement indicated in the model was 0.015" as the gasket, including gauge ring, was deforming. As discussed above, however, the modeling of the spiral wound gasket, with gage ring, did not include a contact surface element. The gasket cannot restrain the radial movement of the mating sealing surfaces of the flange and tube sheet seating surfaces. Therefore it was determined the sealing surfaces would have a maximum differential movement of 0.030" based on the differential temperature profile provided by the transient model. The authors concluded that additional refinements of the gasket properties in the model would provide gasket differential analysis results that would be on the order of that indicated by the differential temperature. The model indicated gasket differential scuffing movement and the seating surface differential temperature-evaluated scuffing are both included in Figure 9. The 1994 paper reported that the maximum gasket scuffing was on the order of magnitude of 0.048".

The authors expect that the additional analysis processing time required to include gasket surface contact elements would result in significantly less processing time than the individual analysis that the 1994 paper reported. However the authors do not consider it necessary to include contact elements in the transient model if appropriate material properties are given to the gasket. The use of contact elements would be more appropriately utilized in a secondary study of the gasket that did not include the total structure of the exchanger closure.

DISCUSSION

Since the model already existed, the time and effort required to set up the analyses described was minimal. The transient thermal solution required approximately 2.5 hours to complete on a dual processor workstation. Use of a larger timestep could have considerably reduced this time without any loss of accuracy. The transient stress solution required slightly over 13 hours on the same workstation. Both analyses were basically "set it and foget it" overnight runs. Post-processing the results required several hours of effort. Was the effort useful and did it shed any new light on the problem?

Thermal Results

The differential temperature information derived from the transient thermal analysis would have been extrordinarily useful to us at the time that the initial work was done. Conservatively, this analysis would have saved man days (if not man weeks!) of effort expended to understand the nature of the problem. Without question, the ability to conduct a transient thermal analysis of such a system in a reasonable amount of time is a plus and worth the effort.

In the original analysis, the authors estimated that the maximum differential temperature between the tubesheet and flanges was on the order of 250 Deg. F. Based on this value, the authors computed a maximum scuffing of 0.048". The current model indicates that the authors overestimated the temperature differential by approximately 80 Deg. F. However, the authors would still use a differential temperature value ~200 Deg F for the design today, so the differences in the analyses are not great.

STRESS RESULTS

The value of results from the transient non-linear stress analysis is not as evident. In the earlier analysis, the authors knew that the reported stresses based on the linear could not actually be achieved, because the bolts would yield and stretch. Nonetheless, the computed stretching of the bolts that was based on hand calculations in 1992 seem to be consistant with those calculated by the non-linear analysis.

Figure 6 illustrates the stress in the bolts. At bolt-up, the stress was 18,000 psi, resulting in a bolt load of approximately 117,000 lb per bolt. At the peak transient condition, the indicated stress was approximately 33,000 psi or 214,000 lb per bolt. As the temperatures equalize, the stress drops to 30,000 psi and a bolt load of 195,000 lb. As stated earlier, the closure was designed to the 1992 ASME Section VIII Division 1 Appendix1 and had been successfully hydrotested in the fabricator's shop but leaked during initial operations. The bolt stress calculations for seating the spiral wound gasket and restraining the hydrostatic end force was 12,000 psi and the specified bolt stress for assembly was set at 18,000 psi. The bolt loads, flange rotation and gasket contact stress data developed from the current transient model was in close agreement to that reported in the 1994 paper [1]. In the original paper the authors stated that bolt loads over 200,000 lb were expected to distort the flange. This is confirmed in the current analysis.

Figure 7 illustrates the strain history for the bolts. For the bolt material at 200 Deg. F, the yield strength is approximately 25,000 psi. The indicated strain of 0.003% and the peak stress of 33,000 psi both point to the fact that the bolts stretch. Figure 8 illustrates the stress-strain curve for the bolt materal. If we assume that the bolts will return along a path parallel to the initial modulus (indicated by the arrow), we can see that there is a residual strain of approximately 0.0018%. Over the 20" length of the bolts, this would represent a permanent strain of approximately 0.036". Since the stretch associated with the bolt-up stress of 18,000 psi was only about 0.0125", the bolts would be expected to be loose upon cool-down. This condition was in fact report from the field. This also correlates well with the hand calculations (.023-.035") in the original paper.

GASKET SCUFFING

Figure 9 illustrates the differential radial movement or scuffing between the tubesheet and flanges. Unfortunately, an error in assigning the material properties limited the scuffing in the model. At low levels of scuffing (<.01"), the values look good. At higher levels, however, the movement was constained by the gasket material. Using the temperature differentials depicted in the thermal data and a straightforward differential thermal expansion model ($\delta = \alpha\ l\ \Delta T$), the scuffing has been computed and is shown on Figure 9.

The original paper predicted ~0.048" of scuffing. However, this was based on an estimated thermal differential of 250 Deg. F rather than the 170 Deg. F value computed in this analysis.

The gaskets on the actual unit exhibited significant damage attributed to scuffing and leaked. Based on this analysis, it would seem that the threshold for scuffing damage that resulted in leakage is lower than originally concluded.

CONCLUSIONS

As illustrated by the data from the transient thermal analysis, the current tools can provide information that is very useful and cost efficient. Having the ability to conduct this analysis in 1992 would have saved a considerable amount of analysis time. The temperature differential information indicates that the threshold for scuffing problems may well be lower than the authors had previously thought. This is an important piece of information that the newer technology allowed us to obtain.

On the other hand, the non-linear analysis did not provide a great deal of information that had not already developed with fairly straightforward hand calculations and linear analysis. Sometimes bigger, faster and better isn't.

REFERENCES

1. Porter, M.A., and Martens, D.H., 1994, *Investigation and Repair of Heat Exchanger Flange Leak*, PVP **278**, (Developments in Pressure Vessel and Piping), American Society of Mechanical Engineers, New York, NY.
2. Baddoo, N.R., 2003, "A Comparison of Structural Stainless Steel Design Standards", The Steel Construction Institute, pp.134.

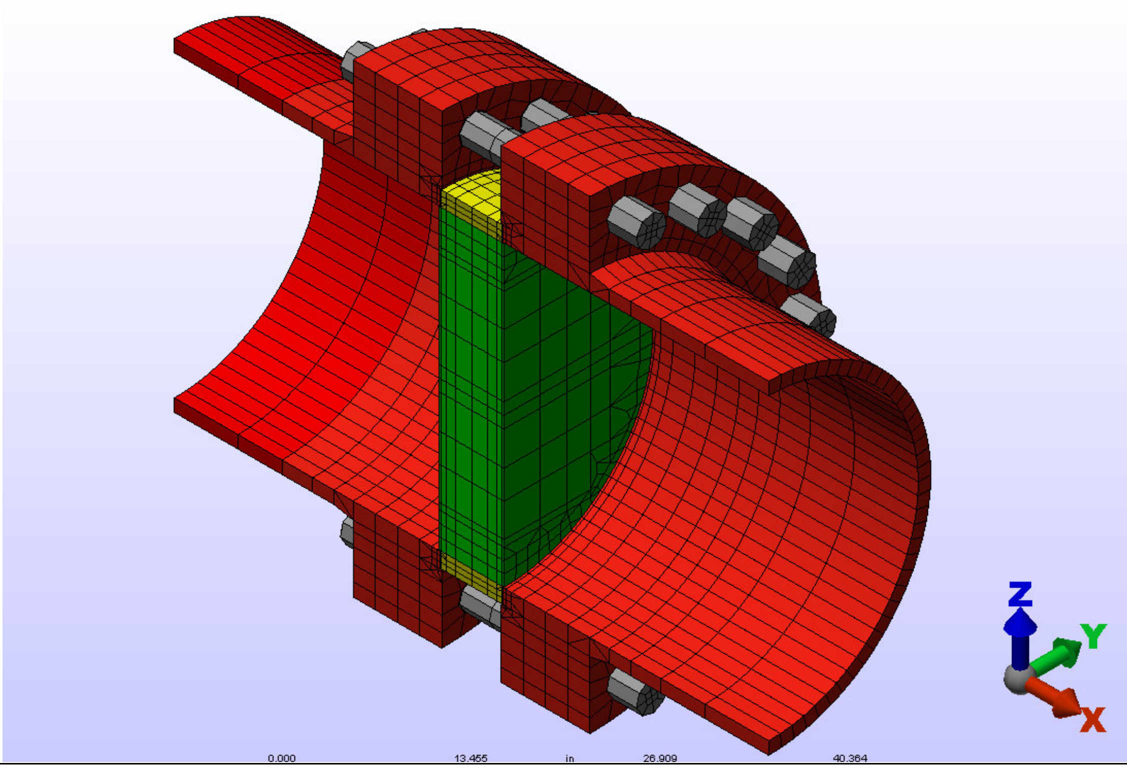

Figure 2 – Overall system configuration

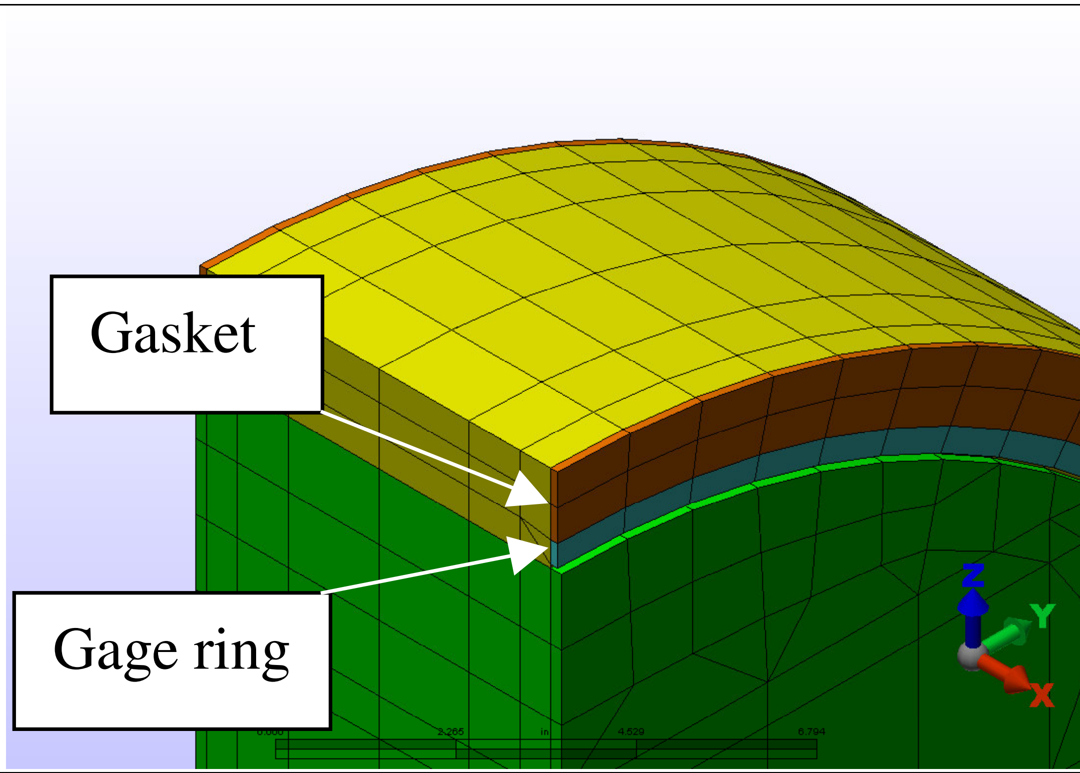

Figure 3 – Example of mesh

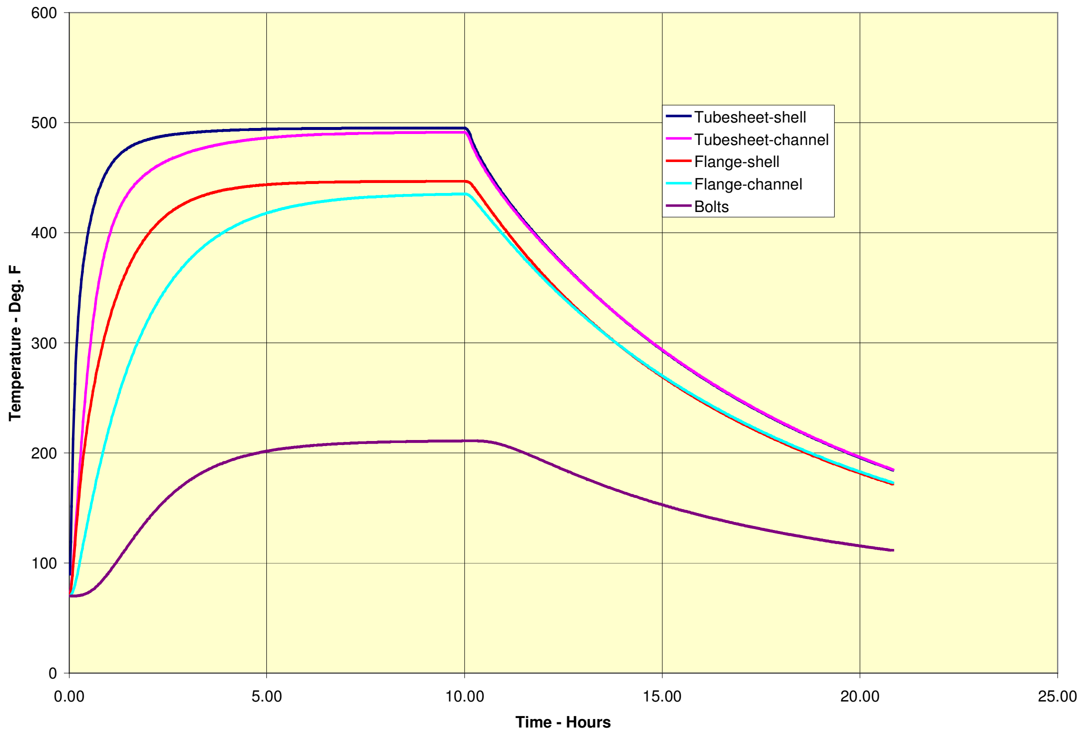

Figure 4 – Temperature history

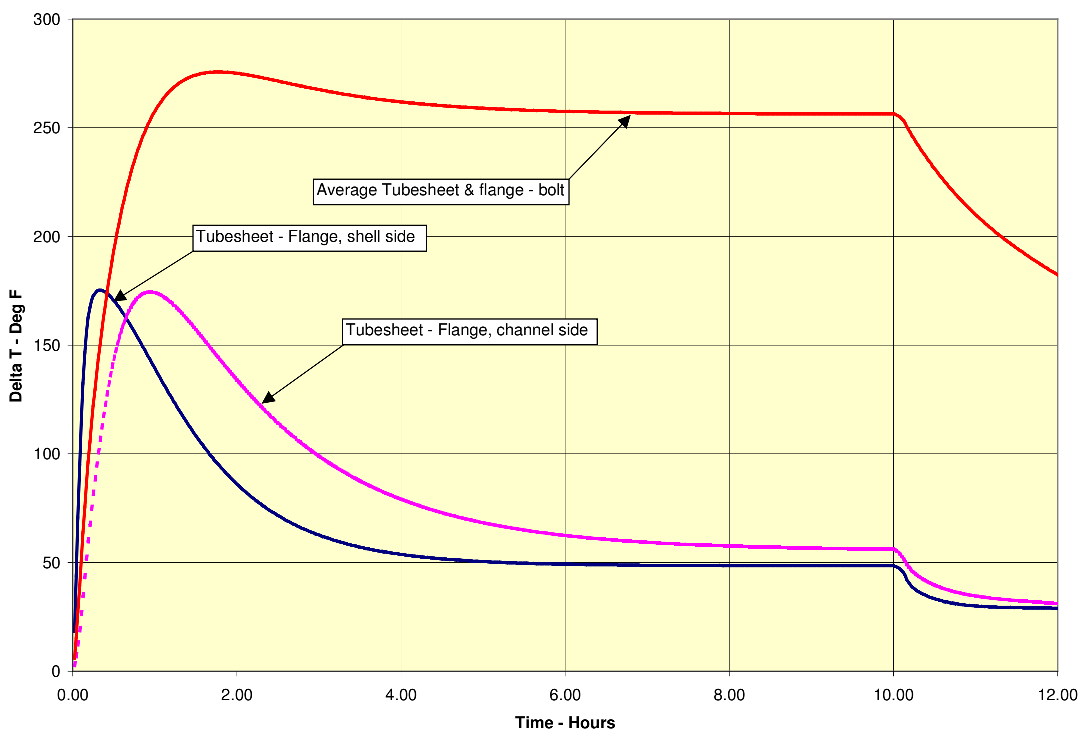

Figure 5 – Temperature differentials

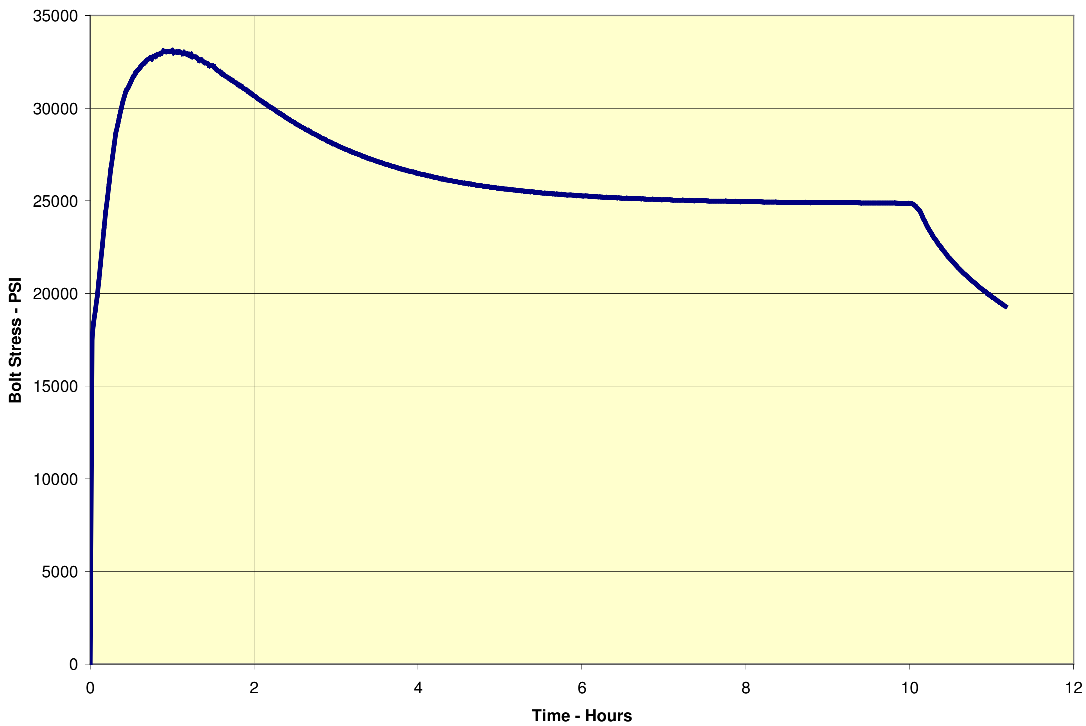

Figure 6 – Bolt stress history

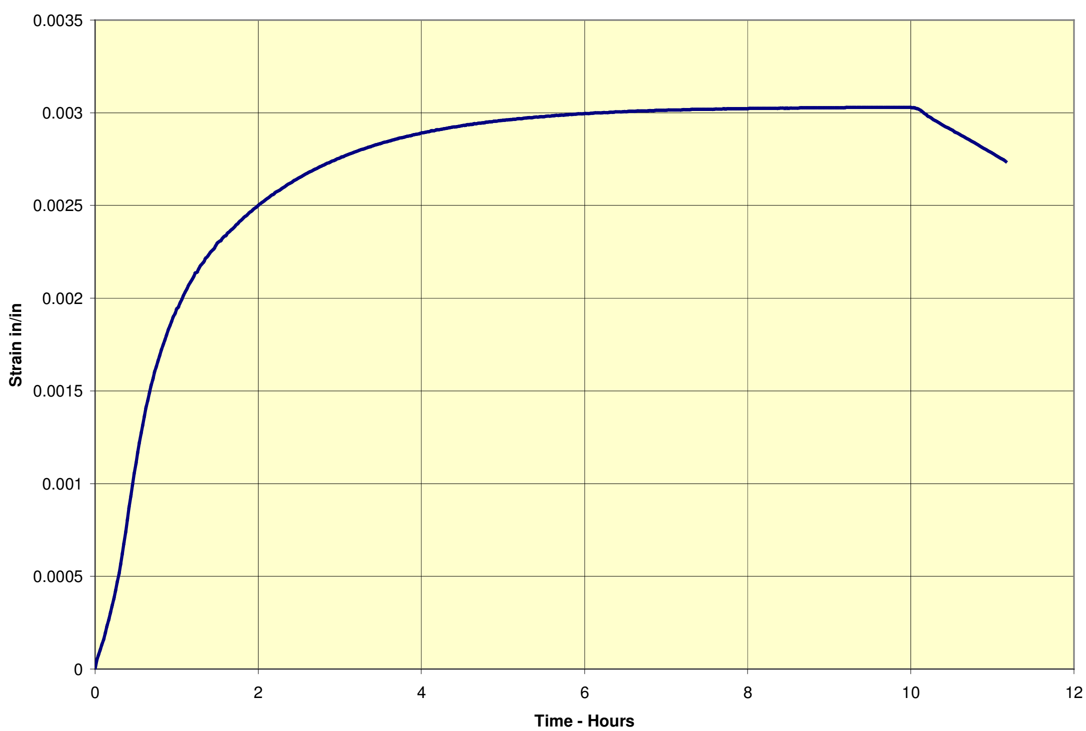

Figure 7 – Bolt strain history

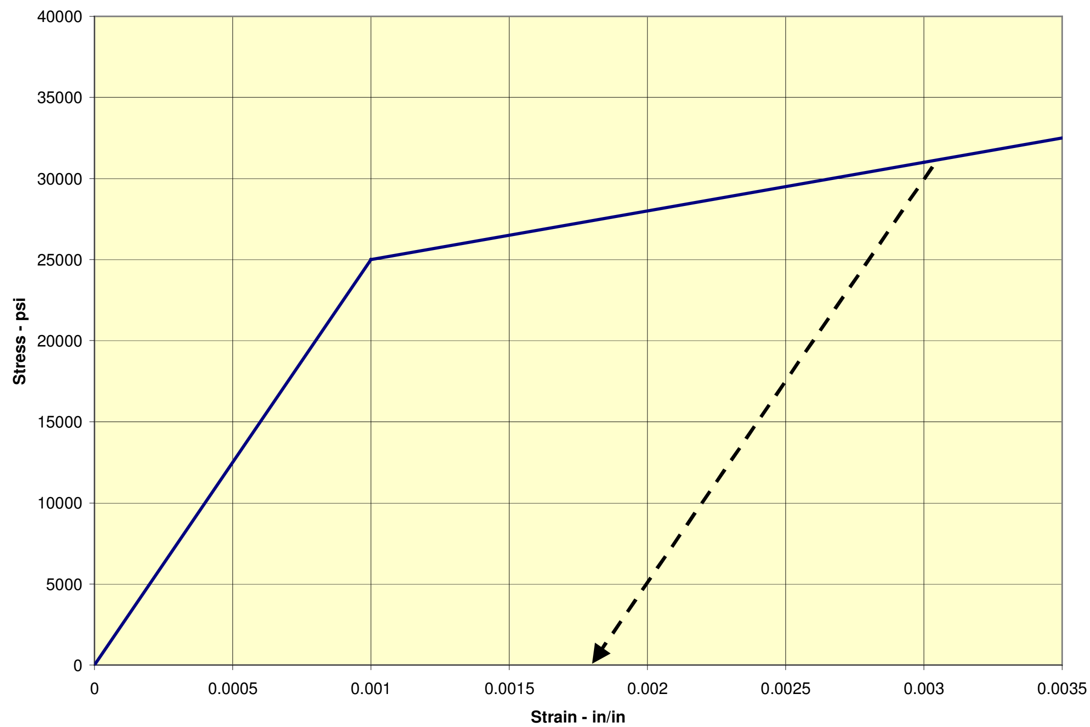

Figure 8 - Stress-Strain

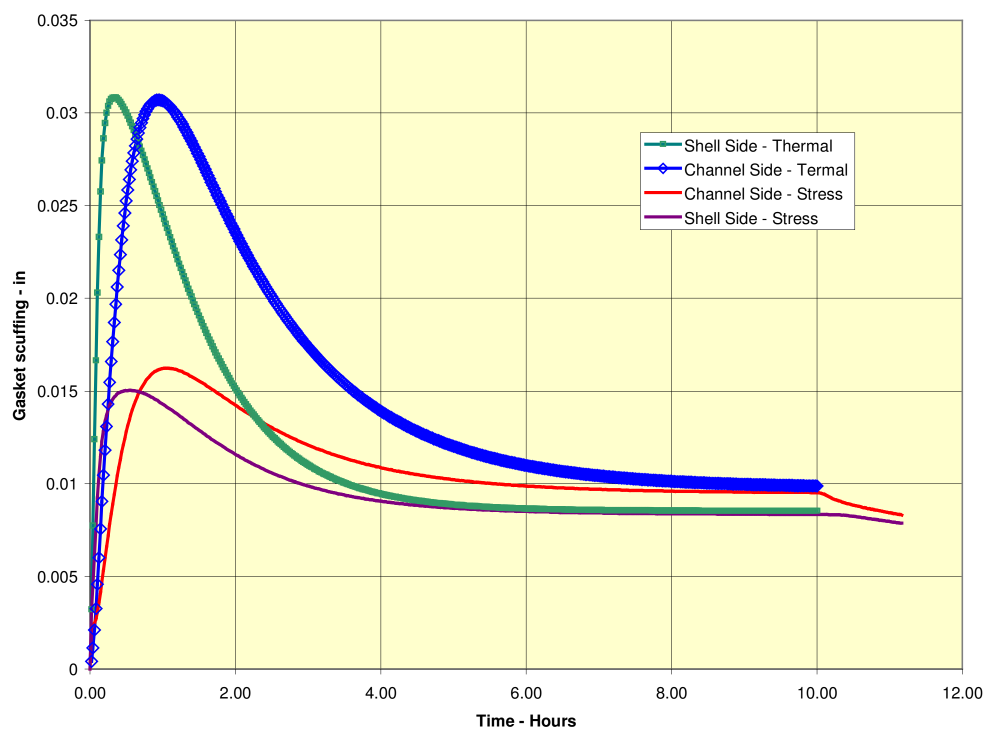

Figure 9 – Gasket scuffing