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RATCHETING ASSESSMENT OF A FIXED TUBE SHEET HEAT EXCHANGER SUBJECT TO IN PHASE PRESSURE AND TEMPERATURE CYCLES

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ABSTRACT

An investigation of the cyclic elastic-plastic response of an Olefin plant heat exchanger subject to cyclic thermal and pressure loading is presented. Design by Analysis procedures for assessment of shakedown and ratcheting are considered, based on elastic and inelastic analysis methods. The heat exchanger tube sheet thickness is non-standard as it is considerably less than that required by conventional design by formula rules. Ratcheting assessment performed using elastic stress analysis and stress linearization indicates that shakedown occurs under the specified loading when the non-linear component of the through thickness stress is categorized as peak stress. In practice, the presence of the peak stress will cause local reverse plasticity or plastic shakedown in the component. In non-linear analysis with an elastic-perfectly plastic material model the vessel exhibits incremental plastic strain accumulation for 10 full load cycles, with no indication that the configuration will adapt to steady state elastic or plastic action; i.e. elastic shakedown or plastic shakedown. However, the strain increments are small and would not lead to the development of a global plastic collapse or gross plastic deformation during the specified life of the vessel. Cyclic analysis based on a strain hardening material model indicates that the vessel will adapt to plastic shakedown after 6 load cycles. This indicates that the stress categorization and linearization assumptions made in the elastic analysis are valid for this configuration.

Introduction

Conventional tube sheet design procedures are generally based on modified elastic plate bending analysis procedures, in which the perforated tube sheet is treated as a thin homogeneous plate with modified material properties used to simulate the structural effect of the perforations. This established methodology is safe and functionally effective but may lead to specification of a plate thickness greater than that actually required to safely contain the specified design pressurized. This conservatism can be reduced by basing the design on elastic-plastic design by analysis procedures specified in Codes such as ASME VIII Div 2 [1]. Behseta et al [13] showed that the primary design thickness of the tube-sheet in an Olefin plant heat exchanger can be considerably reduced if the inelastic design route is followed. This analysis considered the elastic-plastic response of the vessel under static thermal and mechanical loading only. It did not consider other modes such as ratcheting failure under cyclic loading. The object of this paper is to investigate the cyclic elastic-plastic response of the reduced thickness tube-sheet subject to cyclic loading up to the design pressure evaluated in the inelastic analysis.

Tube-sheets subject to pressure and temperature loads experience complex stress distributions in outer rows of the tube-sheet and at the junction between the tube-sheet and the shell. The stress in these regions exhibits characteristics of primary stress due to equilibrium requirements and secondary stress due to the through thickness temperature gradient and self-constraint at the stiff junction. The vessel is intended to operate under steady state condition, however, it is also subjected to repeated loading cycles due to deviation from standard operation modes, process upsets, etc. with largest stress range cycle ranging from start-up to full shut-down, which causes variation in stress magnitude. If the yield stress is exceeded in the first load cycle, this will result in a change in the nature of the stress due to establishment of a residual stress field. The subsequent cyclic behavior will then exhibit one of three responses: elastic shakedown, plastic shakedown or ratcheting (incremental collapse). Pressure vessel design by analysis procedures such as those contained in the ASME Boiler and Pressure Vessel Code Sec VIII, Div 2 require an assessment of the cyclic behavior in order to ensure that ratcheting does not occur under the prescribed mechanical and thermal loads. Both elastic and elastic-plastic analysis can be employed in ratcheting assessment of vessels.

Ratcheting Assessment

Closed form solutions of shakedown problems are very limited due to the complexity of the analysis [3]. More complex problems may be solved by applying plasticity bounding theorems to determine lower and upper bound shakedown loads [4]. These *direct methods* offer the advantage that the detailed load history is not required for the analysis. Instead, the various loads acting on the structure are specified and the shakedown theorems applied to establish a safety domain in load space. Several simplified finite element based methods for shakedown analysis based on bounding theorems have been proposed in the literature and applied to pressure vessel problems [5, 6, 7,8, 9, 10, 11]. These methods allow direct evaluation of shakedown loads without recourse to extensive inelastic finite element analysis. Modeling the full elastic-plastic response of a structure for a specified load history by incremental Finite Element Analysis gives the most complete simulation of the cyclic plastic response of the structure for any type of load cycle. However, this type of analysis may require significant computer effort for complex 3-D structures.

The criterion against ratcheting failure used in ASME VIII Division 2 elastic design by analysis, 5.5.6 *Ratcheting Assessment – Elastic Stress Analysis*, is derived from a simple prismatic bar model of an element of vessel wall [12]. The load is applied as a cyclic thermal axial strain, from $\varepsilon=0$ to $\varepsilon=\varepsilon_R$ and back to $\varepsilon=0$. Shakedown is assured if the strain range does not cause plastic strain during the unloading part of the load cycle. The maximum strain range meeting this condition, ε_R, is given by the equation:

$$\varepsilon_R = 2\sigma_Y / E \qquad (1)$$

where E is the material elastic modulus and σ_Y is the material yield strength. In pressure vessel design, $E\varepsilon_R$ is treated as an elastically calculated maximum stress range, σ_R. It can thus be stated that shakedown will occur if the elastic stress range σ_R is limited to twice the yield stress of the material:

$$\sigma_R \leq 2\sigma_Y \qquad (2)$$

The ASME allowable stress S tabulated in the Code for a given material and design or operating temperature has a value of around $S=2/3\sigma_y$ for most pressure vessel steels. The shakedown criterion, equation (2), can therefore be expressed as

$$\sigma_R \leq 3S \qquad (3)$$

The elastic stress range considered in the ratcheting assessment comprises of primary plus secondary stress. Peak stress does not affect the global failure of the vessel and is omitted in the assessment. In elastic design practice, peak stress is identified and removed from the ratcheting assessment by linearizing the through thickness stress distribution. Shakedown is demonstrated if it can be shown that the linearized primary membrane plus primary bending plus secondary stress range satisfies the 3S limit.

Guidelines for design against ratcheting based on elastic-plastic stress analysis are given in ASME VIII 5.5.7 *Ratcheting Assessment – Elastic-Plastic Stress Analysis*. This design procedure requires application, removal and re-application of the applied loadings during an elastic-plastic analysis of the vessel. An elastic-perfectly plastic material model based on the von Mises yield function and associated flow rule is specified and the effects of non-linear geometry must be included in the analysis. A minimum of three complete repetition of the load cycle is required. Design against ratcheting is demonstrated if one of three criteria is satisfied after 3 (or if needed, more) cycles: there is no plastic action, there is an elastic core in the primary-load-bearing boundary of the component or there is not a permanent change in the overall dimensions of the component.

Vessel and Finite Element Model

The reactor tube-sheet considered is the largest and heaviest heat exchanger in an Olefin plant, a chemical reactor with 3200 tubes. Dimensions, properties, temperatures and pressures, and basic material information, are given in equipment data sheet references [2] and [13] and summarized below:

Design fluid temperature on tube side= -4/190 °C

Design fluid temperature on shell side= -4/145 °C

Design pressure shell side = 1 MPa

Design pressure tube side = 4 MPa

Shell side mean wall temperature = 50 °C

Tube sheet mean wall temperature = 100 °C

The cyclic temperature variations from steady state steady flow condition is taken from the ambient, i.e. full shut down/ start up. A sketch of the area of interest local to the tube sheet/ channel connection with dimensions and un-corroded thickness after manufacture calculated using the classical ASME design by formula approach [14] is given in Figure 1. The 2.9 mm gap in figure 1 is required for installation and fit up of the tube to tubesheet, the attachment of tube to tube sheet is welded type with details according to App. A of Ref. [14] requirements.

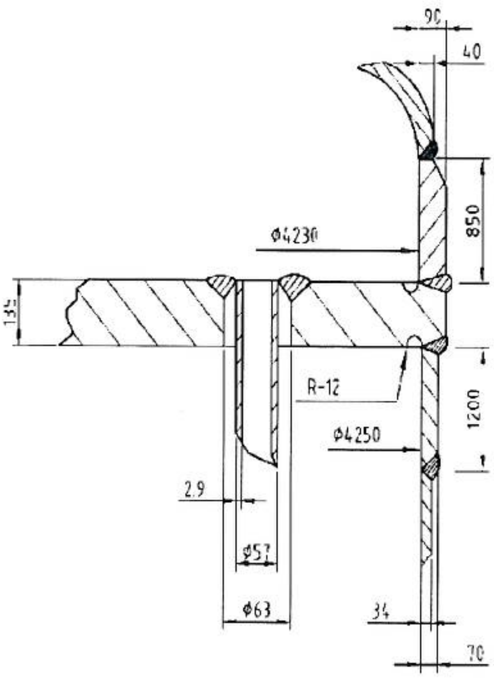


FIGURE 1. TUBE SHEET CONFIGURATION (DIMENSIONS (mm), NOT TO SCALE).

The material physical properties and material stress data are given in Table 1a and 1b Values are reported at the calculation temperature. S is the allowable stress based on Table 5A of reference [1].

TABLE 1a) MATERIAL PROPERTIES

Material	Elasticity Modulus E(MPa)	R_m/ $t_{calc.}$ (MPa)	Cold Yield Rp,0.2/ 20 °C (MPa)	Hot Yield Rp,0.2/ tcalc (MPa)
Upper Shell SA 537 Cl2[b]	193053	542.41	380	317.2
Lower Shell SA 516 Gr 70	195337	482.3	260	232
Tube Sheet SA 266 Cl 2	194173	482.3	250	217.5
Tubes SA 334 Gr 1	194173	379	205	181.5

TABLE 1b) DESIGN STRESS

Material	S (MPa)	1.5S (MPa)	t_{cal} [a] (°C)
Upper Shell SA 537 Cl2[b]	229.6	344.4	190
Lower Shell SA 516 Gr 70	154.7	232	145
Tube Sheet SA 266 Cl 2	144.7	217.12	167.5
Tubes SA 334 Gr 1	120.6	181	167.5

[a] Calculation temperatures are:

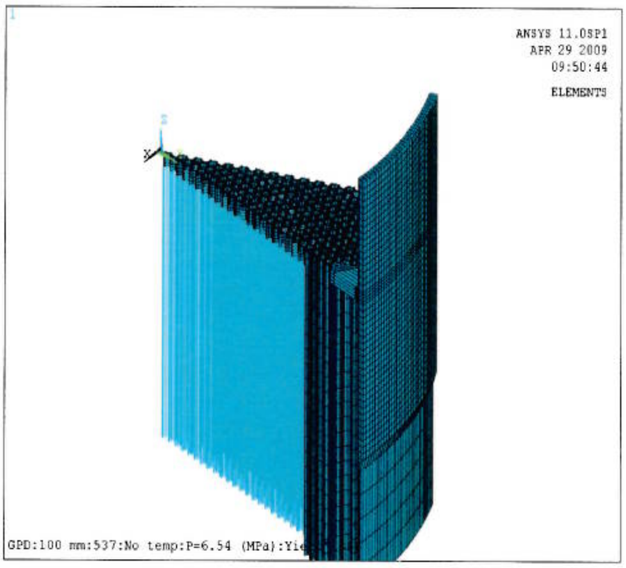
Fluid design temperature for shell material channel side.

Fluid design temperature for shell material shell side.

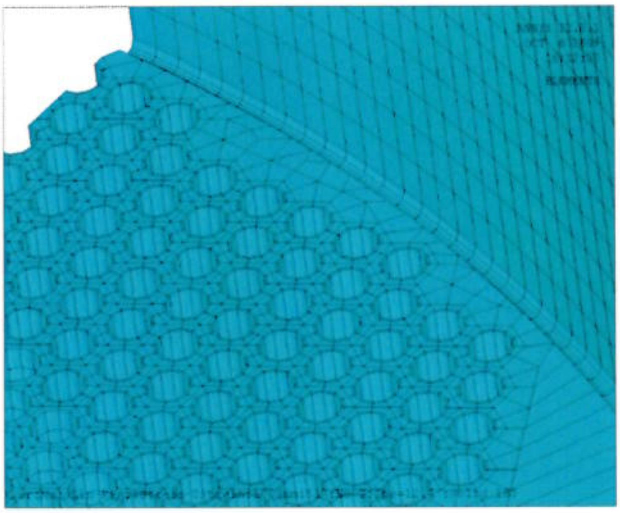
Average design temperature of shell and tube sides for tube sheet and tubes.

[b] Channel side shell, SA 537 CL-2 ($t \leq 63.5$ mm).

Reinhardt *et al* have previously presented a ratcheting analysis and assessment of a tube-sheet subject to rapid transient thermal loading in which the perforated region of the tube-sheet was replaced by an equivalent solid plate with anisotropic yield properties [15]. Here, a full 3-D model of the tube-sheet and shell is used. The FEA model is illustrated in Figure 2. To minimize computing requirements, a symmetrical segment of vessel is modeled. The tube sheet,tubes, shell side shell, tube side shell are modeled using ANSYS 8 node solid 45 isoparametric elements, head effect has been considered as distributed axial load. The model consists of 42,482 elements and 82,238 nodes. Symmetry boundary conditions are applied on the cut surfaces of the modeled segment. Pressure loading is applied to the tube sheet, including the internal pressure in the tubes themselves.


ANSYS 11.0SP1
APR 29 2009
09:50:44
ELEMENTS
GPD:100 mm:537:No temp:P=6.54 (MPa):Yi

(a)



(b)

FIGURE 2. TUBE-SHEET FINITE ELEMENT MODEL.

In reference [2], the tube-sheet thickness of 136mm evaluated by the design by formula approach was reduced to 100 mm and the allowable design pressure was evaluated by ASME VIII Div 2, the limit pressure and plastic load pressure were reported according to the elastic- plastic design by analysis procedures. Material behavior post initial yielding was accounted by an ANSYS multi-linear stress-strain curves, these curves were derived from the true stress-strain curve procedure given in Annex 3.D of ASME VIII Div 2. It should be noted that material cyclic stress-strain curves should be used instead of the monotonic loading curve, in this work due to relatively low metal temperature (108 degree C maximum) and limited cycles due to dominantly steady-state operation of unit (start-up and shout downs only) the monotonic stress-strain curves were used. For purpose of ratcheting assessment, the elastic-plastic analysis design pressure was reduced by applying the service criteria to Plastic Work Curvature Criterion [2]. The design pressures evaluated by limit and elastic plastic analysis were 7.8MPa and 8.53MPa respectively. These values were treated as operating pressures and were applied in the ratcheting assessment of the vessel.

The steady state through thickness temperature distribution in the vessel was evaluated by a preliminary thermal analysis. The skin temperatures of 50°C and 108 °C were taken from the equipment data sheet. The analysis indicated an approximately linear temperature distribution through the thickness of the vessel components, applied as nodal temperatures in the ratcheting assessment.

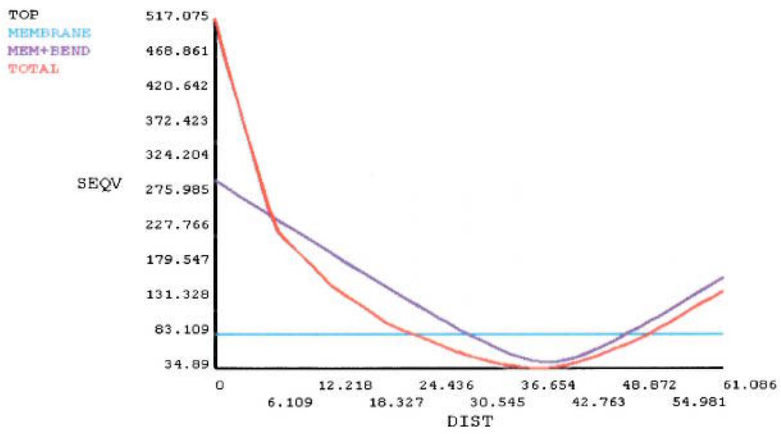
The anticipated number of start-up and shut-down operations during the life of the plant is expected to be in the region of 50.

Finite element Analysis

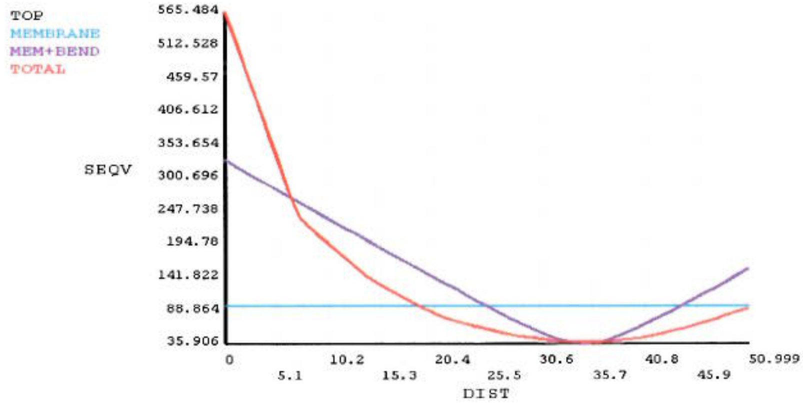
Elastic finite element with stress range calculation and stress linearization was performed for the applied temperature distribution with both the limit analysis based design pressure of 7.8MPa and elastic-plastic based design pressure of 8.53MPa. The worst case stress classification line for ratcheting assessment was identified as a path through the junction between the tube sheet and channel side shell of the exchanger. This region is the must highly stressed part of the reactor since tube sheet bending is additionally influenced by the upper shell action. The closeness of the outer tube limit zone to the radii location brings some local variation to this high stress region. The selected path is located in this region with starting node at the location of the largest stress reported by the software.

Linearized equivalent stress plots for pressure 7.8MPa and 8.53MPa are shown in Figure 3a and Figure 3b respectively. The nonlinear component of stress through thickness is by default listed as a peak stress by the ANSYS linearization post-processor. Peak stress is associated with local stress concentration effects and is not included in ratcheting assessment.

Non-linear through thickness stress distributions may also arise in thick components and at global structural discontinuities due to structural equilibrium and compatibility requirements and in such cases are associated with primary and primary plus secondary stress. However, in this assessment since the solid elements (and not shell elements) have been employed the nonlinear component of stress is treated as wholly peak stress. The stress linearization plots of Figure 3 show that there is a high peak stress at the groove around the inside surface of the vessel. The effect of peak stress is observed through thickness to around 10% of the length of the stress classification line.


TOP
MEMBRANE
MEM+BEND
TOTAL
517.075
468.861
420.642
372.423
324.204
SEQV
275.985
227.766
179.547
131.328
83.109
34.89
0
12.218
24.436
36.654
48.872
61.086
6.109
18.327
30.545
42.763
54.981
DIST

(a)

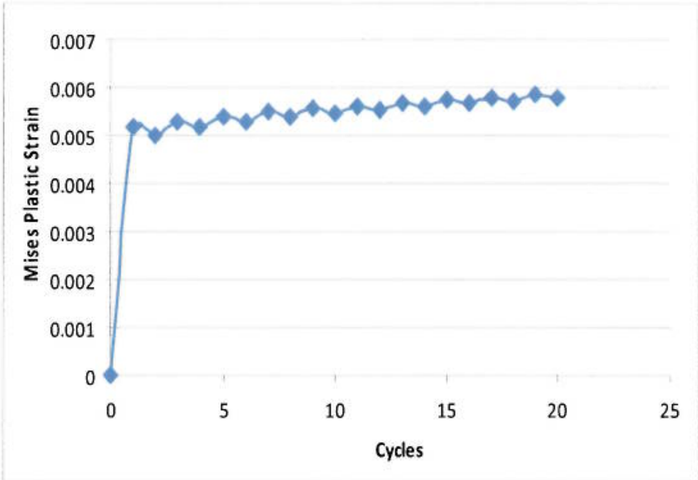


(b)

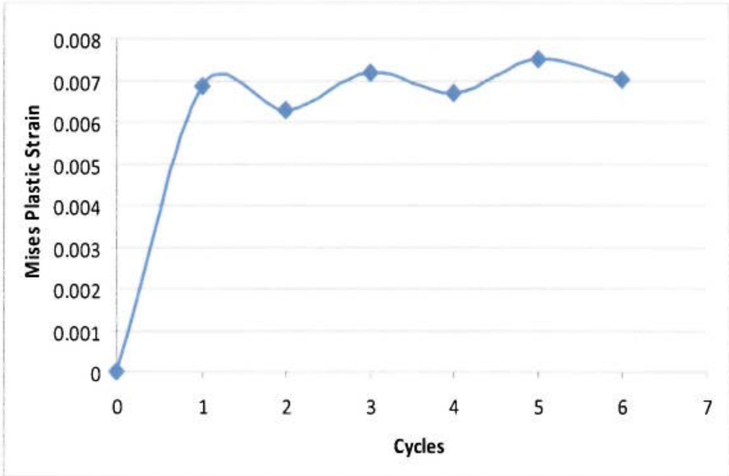
FIGURE 3. ACTUAL AND LINEARIZED EQUIVALENT STRESS DISTRIBUTION ALONG STRESS CLASSIFICATION LINE (a) P=7.8MPA (b) P=8.53MPA

The maximum linearized membrane plus bending equivalent stress was found to be 319MPa for P=7.8MPa and 330MPa for P=8.53MPa. The 3S limit for the tube-sheet material is 434MPa. Therefore, the elastic analysis ratcheting criterion is satisfied for both pressures considered. The presence of significant peak stress indicates that the vessel experiences plastic shakedown for the loads considered.

Small deformation theory elastic perfectly plastic analysis was applied for a large number of cycles (typically 100). For 10 full cycles (20 half cycles) of the applied temperature distribution and pressure of 7.8MPa, the von Mises equivalent plastic strain accumulation is shown in Figure 4a. The analysis was repeated for 3 full cycles of the applied temperature distribution and P=8.53MPa: the equivalent plastic strain accumulation is shown in Figure 4b.


0.007
0.006
0.005
0.004
0.003
0.002
0.001
0
Mises Plastic Strain
0
5
10
15
20
25
Cycles

(a)



(b)

FIGURE 4. SMALL DEFORMATION THEORY ELASTIC-PERFECTLY PLASTIC RESPONSE UNDER APPLIED TEMPERATURE DISTRIBUTION AND (a) P=7.8MPa (b) P=8.53MPa.

Figure 4 shows that the vessel experiences incremental growth in plastic strain, i.e. ratcheting, for up to the 10 cycles modeled. The strain increments are very small, as would be the total plastic strain accumulation over the 50 full cycle life of the component and the design can be accepted as fit for purpose on that basis. However, the underlying ratcheting failure mechanism differs from the plastic shakedown mechanism implicit in the elastic analysis and stress categorization analysis. This suggests that the nonlinearity stress identified in the stress linearization procedure is not wholly peak stress.

The effect of material strain hardening on the cyclic response was investigated by performing elastic-plastic analysis using the ASME strain hardening material model. The equivalent strain accumulation for the applied temperature and P=7.8MPa is shown in Figure 5. In this analysis, the vessel exhibits plastic strain accumulation during the first 5 full cycles, after which the behavior shakes down to alternating plastic strain without further strain accumulation. A similar response was observed for a cyclic pressure of 8.53MPa. Figures 4 and 5 are presented for a node at the junction of radii to the upper shell.

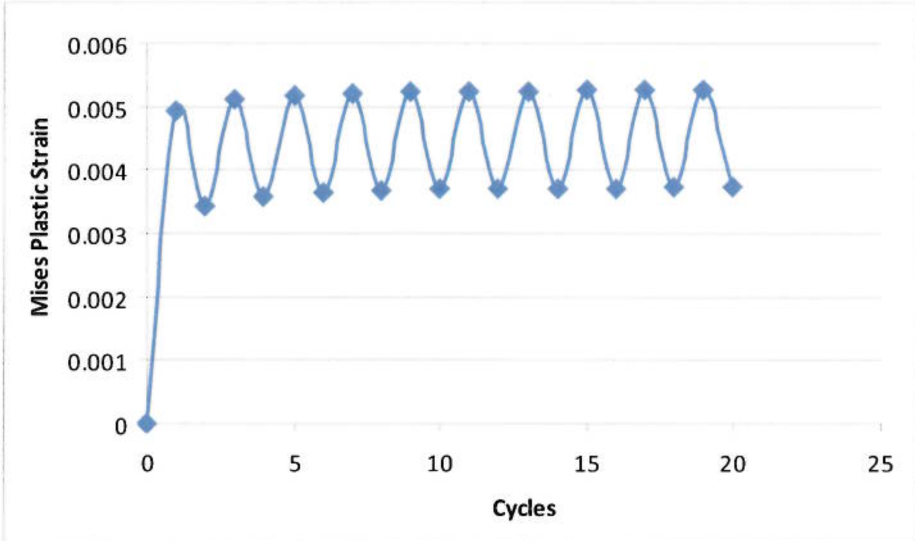
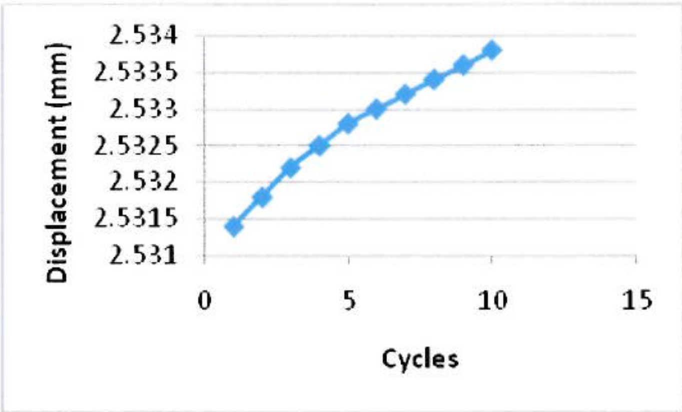
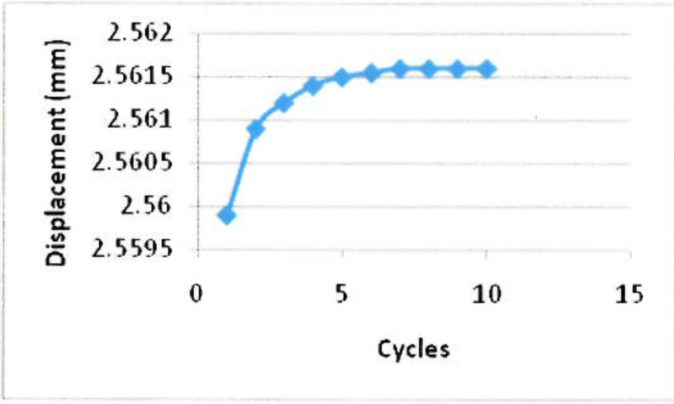


FIGURE 5. SMALL DEFORMATION THEORY STRAIN HARDENING RESPONSE UNDER APPLIED TEMPERATURE DISTRIBUTION AND P=7.8MPa.

Permanent deformation of the component is, obviously, correspondingly small. Figure 6a shows the deformation of the highest loaded point in the vessel, at the junction between the tube-sheet and the channel side shell, given by elastic-perfectly plastic analysis. The permanent deformation is seen to increase for the first 10 full cycles and continues to increase in subsequent cycles. Figure 6b shows the deformation of the same point given by strain hardening analysis. Deformation increases for the first 6 cycles but thereafter no further permanent deformation occurs.


Displacement (mm)
2.534
2.5335
2.533
2.5325
2.532
2.5315
2.531
0
5
10
15
Cycles

(a)



(b)

FIGURE 6. DEFORMATION OF HIGH STRESS POINT AT INTERSECTION BETWEEN TUBE-SHEET AND CHANNEL SIDE SHELL (a) ELASTIC PERFECTLY PLASTIC (b) STRAIN HARDENING.

CONCLUSION

The elastic stress analysis, stress linearization and assumed stress categorization, in which the non-linear component of through thickness stress distribution is assumed to be peak stress, indicates that the tube-sheet configuration exhibits shakedown under the cyclic temperature distribution and pressures considered. The presence of significant peak stress indicates that the likely failure mechanism is low cycle fatigue associated with the steady state cyclic plastic strain range. Inelastic analysis based on an elastic-perfectly plastic material model indicates that the configuration experiences a different failure mechanism, incremental plastic straining or ratcheting. The cyclic plastic strain increments are small and would not give rise to a global plastic failure mechanism over the required 50 cycle life of the vessel by a considerable margin, thus the vessel can be deemed to be acceptable for service. This result indicates that the stress categorization procedure used in the elastic analysis may not be appropriate. In this case it is not conservative to assume that the non-linear component of through thickness stress is wholly peak stress and therefore can be disregarded in elastic shakedown analysis. The result of the inelastic analysis indicates that in fact what is reported as the peak stress in elastic analysis is in part secondary in nature and should not be wholly excluded from the elastic ratcheting assessment.

ASME VII Div. 2 does not include a ratchet checking procedure based on non-linear analysis assuming a strain hardening material model. However, this advanced type of analysis can be more representative of the actual structural behavior of the component (although highly dependent on the plasticity model used). The results given by the strain hardening analysis indicate that in practice the structure would shake down to steady state alternating plasticity after 6 load cycles, with no global plastic deformation thereafter. This finding, although not appropriate for Code design, indicates that whilst the elastic design route presented here may not be strictly conservative, it is likely to be safe in practice with respect to the shakedown criterion due to the constraining effect of the material strain hardening on the growth of plastic zones.

Acknowledgement
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