FINITE ELEMENT INVESTIGATION OF A CBA REACTOR FOR THE EFFECTS OF THERMAL CYCLING

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Abstract
The Cold Bed Adsorption sulfur recovery process utilizes carbon steel reactor vessels that are subjected to thermal cycles. This paper presents the results of finite element investigation of the cyclic temperature profiles and operating stresses for the reactor vessels. The authors utilized a thermal model to establish temperature profiles resulting from the hot and cold processing conditions. These temperature results were placed in a stress investigation model. This model was utilized to investigate thermal, pressure and dead loading induced stresses. The stress results are compared to the 1995 (with addenda to 1996) ASME Section VIII Division 2 fatigue stress allowables utilizing the procedures included in the 1995 (with addenda to 1996) ASME Section VIII Division 2.

INTRODUCTION
The Cold Bed Adsorption (CBA) process utilizes catalyst containing vessels that remove sulfur from effluent gases before discharge to the atmosphere. Two catalyst vessels are utilized in the process. The vessels are cycled with one in sulfur removal service and one in regeneration service. The vessel in removal service is operated at the sulfur dew point temperature to adsorb the sulfur as the effluent gases pass through the vessel. The vessel is then removed from adsorption service and regenerated by heated to allow the accumulated sulfur to melt and be removed from the catalyst. This adsorption and regeneration cycle results in the vessel being subjected to substantial thermal gradients and corresponding thermal induced stresses.

ASME Section VIII Division 1 does not contain specific methodology for cyclic operating conditions but paragraph UG-20 does require that it must be considered in the design. The methodology in ASME Section VIII Division 2 is a generally accepted approach for complying with the UG-20 investigation requirement.

The CBA catalyst vessel is a horizontal, saddle supported A-516-70 carbon steel vessel with maximum coincidental operating conditions of 10 psig and 700 °F. The vessels' requirement for structural support ability results in a design that is suitable for a 50 psig operating condition. The vessel is designed per the 1995 (with addenda to 1996) ASME Section VIII Division 1 code for 50 psig and 750 °F conditions (further references to this Division and Division 2 refer to this code addition). The Vessel is specified as 100% radiograph with special magnetic particle testing (MT) inspection requirements. The vessel is 12 ft. Diameter and 82 ft. tangent to tangent. The nominal wall thickness is 5/8 inch except over the support saddles where the wall thickness is increased to 1 1/16 inch. The vessels have steam heated coils and 4" of thermal insulation for control of heat loss to atmosphere. The coil steam pressure is maintained to produce a 450 °F condensing temperature. The steam coils remain fully pressured during all steps of operation but the coil mitigation of thermal induced stresses is not considered in the analysis to produce a conservative design approach. During the adsorption cycle the vessel operates at 260 °F. During the regeneration the vessel is subjected to a cycle of heating with 650 °F gases until the total vessel achieves this temperature and then cooled by 260 °F gas until the vessel is ready for adsorption service again. This cycle is repeated on an approximate 26 to 28 hour cycle.

This heating and cooling cycle subjects the vessel to thermal induced stresses which result in significant thermal movements. The thermal stresses are induced when the inlet gas temperature is switched from 260 °F adsorption service gas flow to 650 °F regeneration gas flow nearly instantaneously. The reverse occurs on the cool down portion of the regeneration cycle. The catalyst bed acts as a heat sink which allows the vessel portion above the bed to become equal to the inlet temperature and the vessel portion below the bed to be at the previous cycle temperature. The regeneration gas changes from the 650 °F regeneration gas inlet temperature to the 260 °F previous process gas temperature in the catalyst bed. This condition causes the vessel to form a "banana" shape due to differential thermal expansion of the respective shell portions. The definition of this movement is important to the associated piping and platform design.

The design life of the vessel is 20 years resulting in approximately 6,200 complete cycles, based on a 27 hour average cycle life and 95% plant availability. The thermal cycle is composed of an adsorption cycle of approximately 13 hours and a regeneration cycle of approximately 13 hours with an allotted 1 hour of both vessels in adsorption service to prevent sulfur breakthrough.

The thermal gradient and stress investigation method selected was the use of finite element (FE) analysis. A FE thermal model was utilized to develop the temperature profiles and a FE stress model, utilizing the developed temperature profile, was utilized to analyze the associated stresses.

Thermal FE investigation

The COSMOS/M FEA program was used to model the temperature distribution in the vessel. The heat transfer model utilized a snap shot of the gas and catalyst bed temperatures and was analyzed as a steady state condition. The model included convection from the gas to the vessel wall and the conduction of the vessel metal wall to establish temperature distributions. The temperature profile across the catalyst bed was established based on prior calculation and operating temperature data. The gas temperature profile in the bed was conservatively modeled as changing from 260 °F to 650 °F in 12 inches of catalyst bed depth. The heat loss to the atmosphere was neglected as it was considered to be insignificant due to the heating coils and thermal insulation. This assumption was considered to be conservative as heat gain from the heating coils would tend to increase the minimum metal temperatures and heat loss to the heating coils/ambient would tend to decrease the maximum metal temperatures.

The vessel was modeled utilizing approximately 3,500 Quadrilateral and Triangular Thin Shell elements with both membrane and bending stiffness. The vessel was modeled as half of the vessel cut at the longitudinal center as the temperature profile can be considered to be symmetrical about this location. Symmetrical Boundary Conditions were specified at this plane of symmetry.

The temperature profiles for the transient portions of the heating cycle and cooling cycles were investigated and were found to be very similar (see FIGURE 1 & 2). The temperature iterations versus time displayed significant temperature differences between the inlet gas nozzle neck and the adjacent vessel wall. The higher velocity of the gas in the nozzle versus the vessel interior resulted in a significantly greater convection heat transfer coefficient within the nozzle neck than the rest of the vessel. This condition results in the nozzle changing temperature faster than the vessel shell which introduces significant thermal related stresses in the nozzle to shell junction area. It was necessary to design a thermal sleeve for use in the nozzles to develop a similar heat up and cool down rate for the nozzle and the vessel shell.

Stress FE Investigation

The same modeling characteristics were utilized as in the thermal FE investigation. The temperature profile developed during the thermal FE investigation was applied to the stress investigation model. A linear static analysis was utilized for the stress FE investigation. The area of maximum thermal stress was adjacent to the maximum thermal gradient in the catalyst bed area. The manways for catalyst bed servicing were on the on the side of the vessel slightly below the centerline of the vessel. The catalyst bed manways were installed in the 1 1/16 inch plate section of the vessel.

The FE indicated stresses for the heating and cooling steps are listed in TABLE 1 and displayed in FIGURE 4 & 5. These stresses were analyzed per procedures in ASME Section VIII Division 2 Mandatory APPENDIX 5 paragraph 5-110.3. The highest indicated thermal induced principle stress and stress differences were - 25,200 psi for the heating cycle and 25,200 for the cooling cycle, occurring at node 2535. These stresses occurred at the same node and are considered alternating stresses as the stresses in this area will reverse during the complete operating temperature cycle (see table 1). As the principle stress changes direction it is necessary to evaluate the six stress components to establish the alternating stress intensity (see table 2). The coincidence pressure of 10 psig that occurs during the heating and cooling cycles does not vary significantly during the cyclic operation. The sustained stresses due to pressure and dead loads at this node are essentially constant for the complete operating cycle. The piping forces and moments on the nozzles change during the complete cycle but the shell stress intensity in these areas is not the limiting stress.

Therefore the Section VIII Division Mandatory Apppendix 5 alternating stress (Salt) is essentially the result of the thermal induced stresses. From inspection of table 2 it is apparent that only one principle stress is significant as the others are an order of magnitude less. Therefore the alternating stress intensity is due to thermal stresses and approximately equal to: Note; sign convention is utilized for individual load case and the resulting stress is additive as the stresses are of opposite sign:

$$Salt = ((25{,}130 - 50) + (25150 + 50))/2 = 25{,}140 \text{ psi.}$$

This Salt value is suitable for accomplishing a preliminary fatigue cycle analysis. Unless the fatigue cycle life is near the actual cycle life requirement further rigorous stress analysis is not necessary. A more rigorous analysis of the TABLE 2 stresses can be accomplished by combining stress in accordance with procedures provided by Roark (1965). The results of the rigorous stress approach is presented for clarity in TABLE 3. Utilizing the Principle stress from TABLE 3 the alternating stress intensity is equal to:

$$Salt = ((28{,}520 + 130) + (21{,}940 + 20))/2 = 25{,}300 \text{ psi.}$$

Fatigue analysis exemptions determination is made utilizing ASME Section VIII Division 2 paragraph AD-160.2 criteria. If all of conditions of A or of B are achieved no further analysis is required.

Inspection of paragraph AD-160.2 Condition A yields the following tests;

(a) the expected (design) number of full range pressure cycles including startup and shutdown does not exceed 1000; this is met as the expected full design pressure cycles are less than 10.

(b) the expected number of operating pressure cycles in which the range of pressure variation exceeds 20% of the design pressure does not exceed 1000; this is met as the process operating pressure is less than 20% of design pressure and does not vary by more than 2% during operation.

(c) the effective number of changes in metal temperature between any two adjacent points in the pressure vessel does not exceed

1000. This sub-paragraph requires increasing the design number of cycles if the temperature difference across an element is greater than 50 °F. The differential temperature across the element at node 2535 is 119° F, therefore the design cycle requirement of 6,200 must be adjusted by a factor of 2 to 12,400 for comparing to the 1000 cycle exemption from further fatigue investigation. The adjusted design cycles exceed the allowable exemption of 1000 cycles per AD-160.2 Condition A (c) therefore exemption is not achieved.

Inspection of paragraph AD-160.2 Condition B results in the determination that subparagraph (d) would be limiting. The required calculation becomes (with information from appropriate reference table);

dt = (Sa/2)x(E)x(instantaneous coefficient of thermal expansion)
dt = (17,500/2)x(27.32*10^6)x(7.82*10^-6) = 41 °F

The differential temperature difference from the thermal FE investigation noted above exceeds this value therefore the exemption criteria is not achieved.

Based on the results of Condition A and B inspection the vessel must be further investigated for cyclic conditions.

Investigating the provisions of Division 2 paragraph 4-134 yields; indicated stresses (see TABLE 1-3) are to be compared to ASME Section VIII Division 1 stresses using Division 2 methodology. Per Section VIII Division 1 the allowable stress for the vessel is 17,500 psi at the 650 °F operating and 14,800 psi at the 750 °F vessel design temperature.

1) for conditions of combined $P_m + P_L + P_b$: from inspection of the thermal load case versus all load case stresses listed in the tables above it is obvious that that these stresses are significantly less than the allowable.
2) for conditions of combined $Pm + P_L + P_b + Q$: as cyclic investigation is required per above analysis the provisions of Mandatory Appendix 5 are to be utilized.

There is a difference of opinion as to the use of ASME Section VIII Division 2 methodology utilizing Division 1 stress allowables with respect to FE indicated stresses. The difference results from the concern that Division 1 vessels are not constructed to the same quality assurance levels as Division 2 vessels and the Division 2 Appendix 5 paragraph 5-111 requirement for consideration of fatigue strength reduction factor usage. The investigation of allowable ASME fabrication flaws is not provided by ordinary FE analysis. Therefore the methodology should include the use of a fatigue strength reduction factor to accommodate the concern for fabrication flaws. Typical FE analysis does investigate local structural discontinuities with an acceptable level of quantification. The authors' practice is to use a fatigue strength reduction factor of 1.2 for Radiographed and visually inspected butt welds for appropriately converged FE indicated stress intensities. The authors practice (based on quality assurance by MT of the root pass and completed weld and visual inspection of the completed weld) is to use a factor of 1.5 to 1.7 for full penetration nozzle welds and 1.7 (at toe) to 4 (at root) for non-full penetration (fillet type) nozzle welds. This is consistent with information presented by Hechmer and Kuhn (1997). The authors recommend a fatigue strength reduction factor of 1.5 for butt welds and 2.0 (at toe) to 4.0 (at root) for fillet welds (with similar quality control to above) for stress intensities developed by closed form calculations. It should be noted that in some applications this could change the area of investigation for cyclic stress. The fatigue strength reduction factor is utilized as a stress intensification factor. The calculated stress intensity is multiplied by the fatigue strength reduction factor before determining the corresponding cycle life. The authors recommend the use of these minimum fatigue strength reduction factors for appropriately converged FE developed stress intensities for vessels in non lethal service. Greater fatigue strength reduction factors may be appropriate for lethal services.

For this vessel the area of highest FEA indicated stress contains butt welds and with the utilization of a 1.2 fatigue strength reduction factor the alternating stress in butt weld will become 30,200 psi (25,150 times 1.2). The vessel area being investigated for thermal induced stress does contain fillet type nozzle welds for the access manways. The approximate thermal induced alternating stress intensity in the manway nozzle weld area was 10,100 psi. The use of the maximum 1.7 fatigue strength reduction factor results in a 17,200 psi alternating stress intensity. Therefore the butt weld area stresses govern the fatigue investigation.

Appendix 5 requires that the calculated alternating stresses, with fatigue strength reduction factor applied, to be suitable for the design cycle life. From inspection of Figure 5-110.1 it is apparent that approximately 40,000 psi alternating stress is allowable for the design life of 6,200 cycles. From inspection of Table 5-110.1 it is apparent that a Salt of 30,200 psi would provide a cycle life exceeding 20,000 cycles. Utilizing the fatigue strength reduction factor adjusted alternating stress of 30,200 psi it is necessary to interpolate the data for 20,000 and 50,000 cycles using the formula provided in the Table.

Cycles = 20,000x(50,000/20,000)^(log(31/30.2))/(log(31/23))
= 21,700

This exceeds the required 6,200 design cycle life of the vessel, therefore the design of the vessel is satisfactory for the cyclic service. The use of the rigorous developed stresses listed in TABLE 3 will not alter this conclusion as the Salt= 25,300x1.3 = 30,360 psi also provides in excess of 20,000 fatigue cycles.

It should be noted that a compressive buckling analysis was conducted to confirm the vessel stability. The authors recommend such an analysis for all shells with compressive loadings including temperature gradient effects.

Conclusions

The FE analysis of the cyclic conditions the CBA reactor vessel confirmed the design to be satisfactory for the design plant life. The thermal and stress FE investigation indicated that the greatest thermal stress occurred in the catalyst bed area. The investigation also indicated that the inlet and outlet gas nozzles would be exposed to thermal induced strains. Thermal sleeves were added to the design to reduce the strain in these areas.

The generally accepted use of ASME Section VIII Division 2 methodology for cyclic life investigation includes the use of stress concentration factors or fatigue strength reduction factors for areas of local structural discontinuities. The authors recognize that the current FE analysis by use of elastic (linear) applications does not account for local plastic strain that may occur at areas of discontinuity) may not provide sufficient stress indication for local areas in which the stresses exceed the material yield. The authors have recommended typical factors for use at butt and fillet welds for appropriately converged FE indicated stress intensities for non-lethal services. For FE indicated stresses with adjusted values above yield, as maybe the case for many low cycle applications, the recommended factors may or may not be appropriate.

References

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TABLE 1 Principle, Intensity and Von Mises stresses for thermal load case (stress in psi)

Cycle step	Principle 1	Principle 2	Principle 3	Stress Intensity	Von Mises stress
Heating	- 2.9x10^-6	- 1,700	-25,200	25,200	24,400
Cooling	25,200	1,638	8.3x10^-5	25,200	24,400

TABLE 2 Six component stresses for thermal and load cases (stresses in psi)

Cycle step/load	Stress X	Stress Y	Stress Z	Shear XY	Shear XZ	Shear YZ
Heating/thermal	- 50	- 1,690	- 25,150	- 250	100	820
Heating/all loads	30	900	-21,930	150	-100	-540
Cooling/thermal	50	1,630	25,130	250	100	- 810
Cooling/all loads	120	4,200	28,350	640	- 310	-2170

TABLE 3 Principle, Intensity and Von Mises stresses for all load case based on TABLE 2 component stresses (psi)

Cycle step	Principle 1	Principle 2	Principle 3	Stress Intensity	Von Mises stress
Heating/all loads	-920	- 21,940	20	22,880	22,430
Cooling/all loads	28,520	- 130	4,280	28,650	26,720

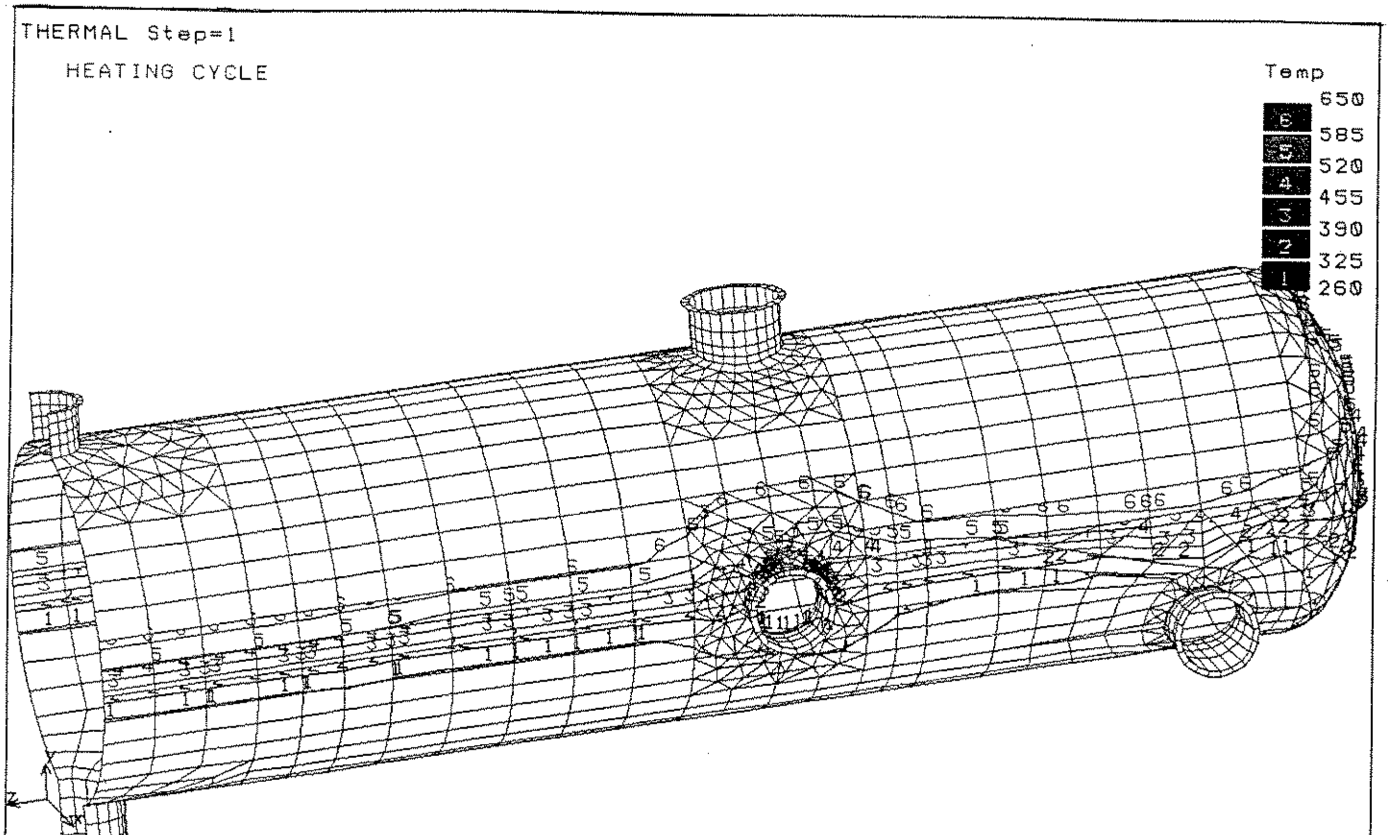


Figure 1

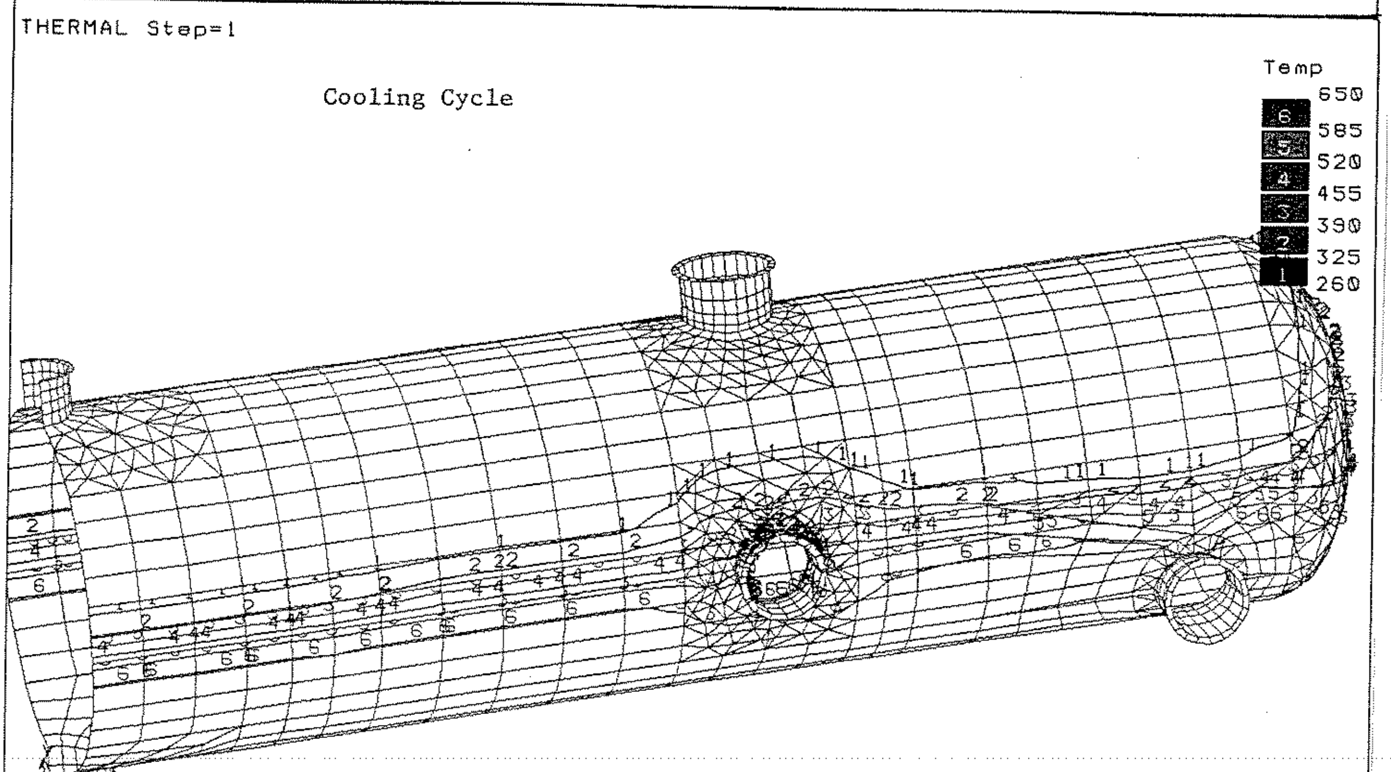
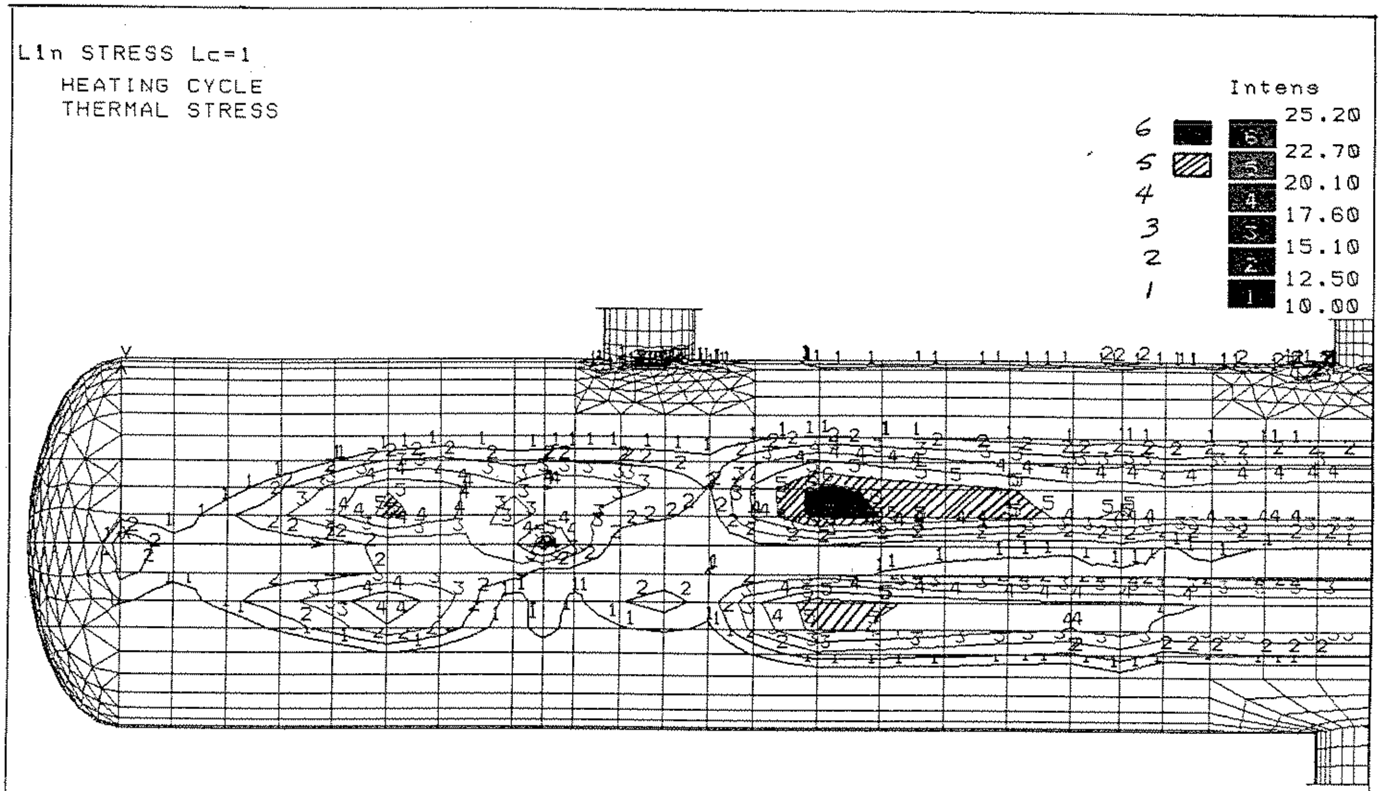


Figure 2


Lin STRESS Lc=1
HEATING CYCLE
THERMAL STRESS
Intens
25.20
22.70
20.10
17.60
15.10
12.50
10.00

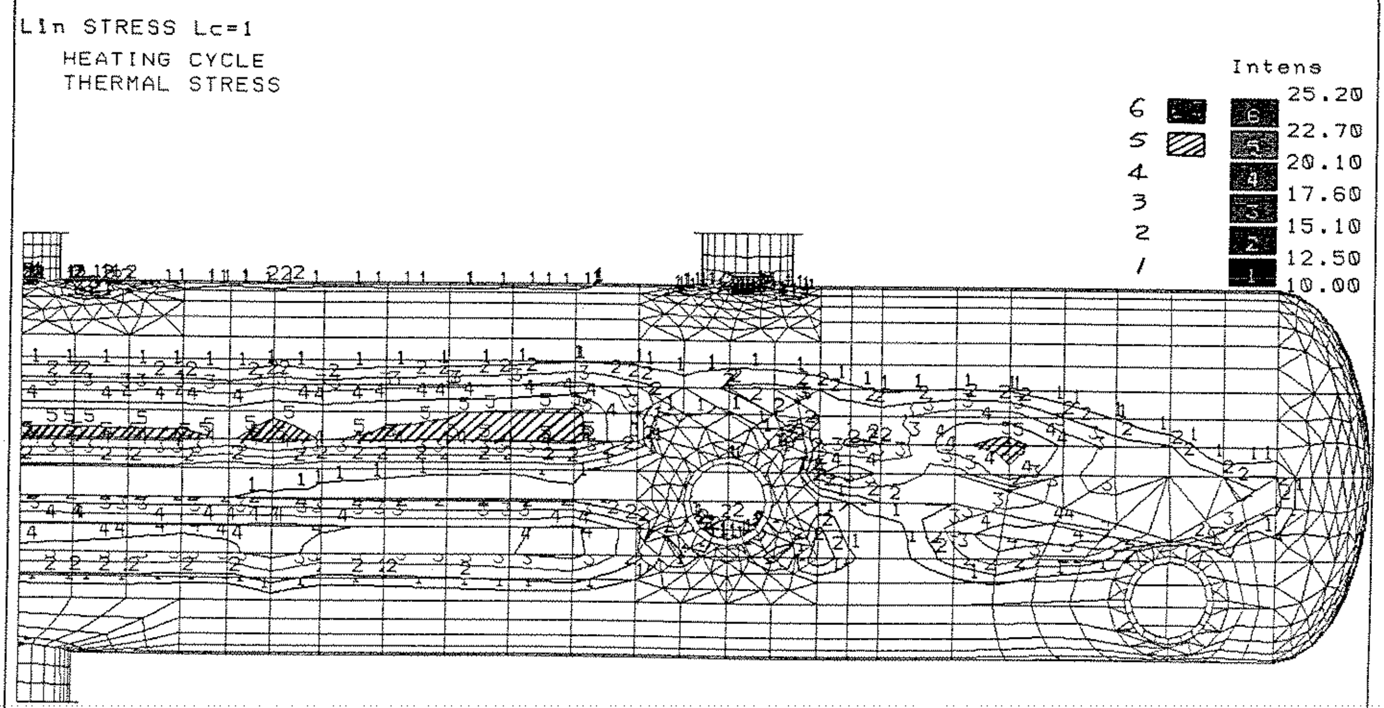
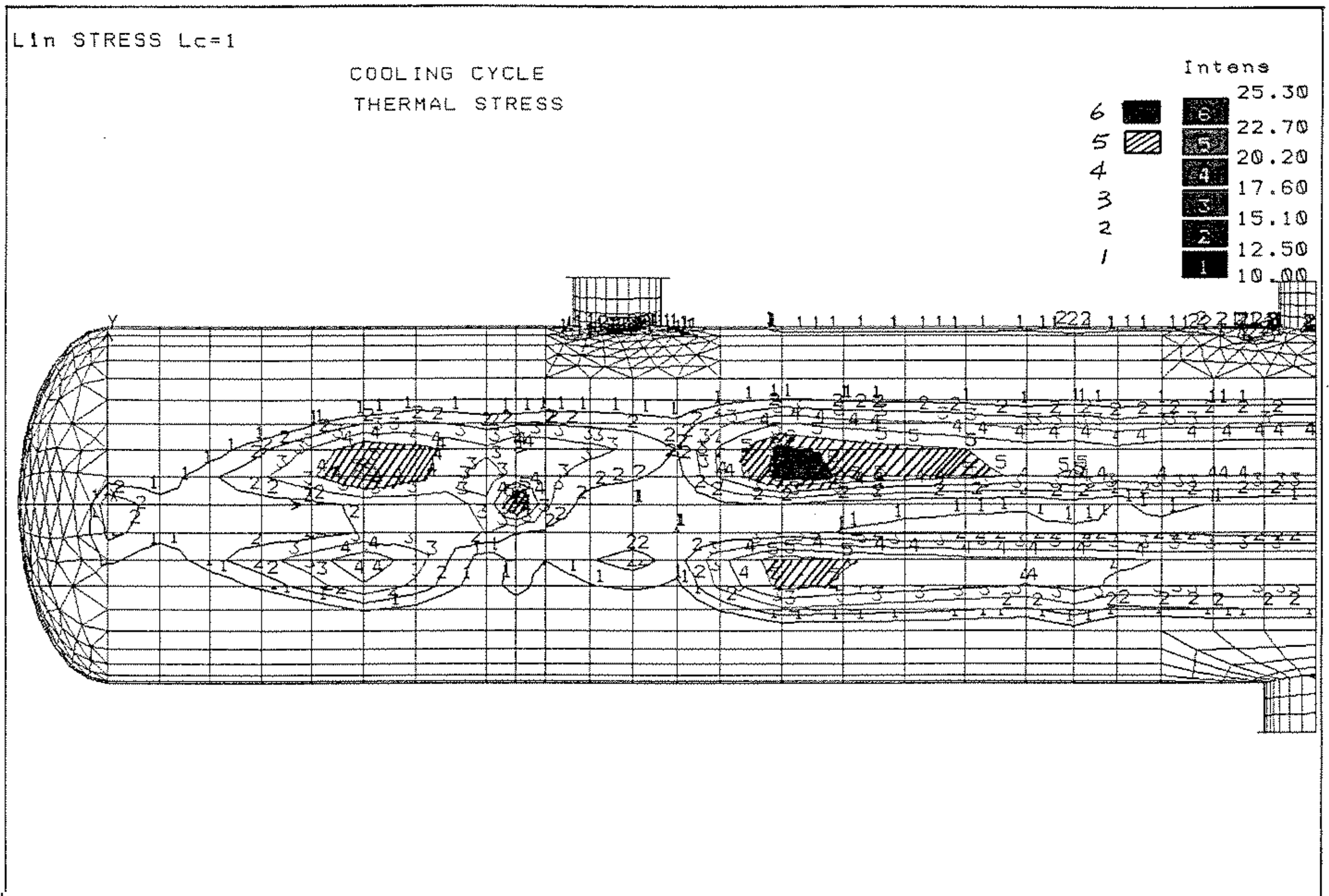
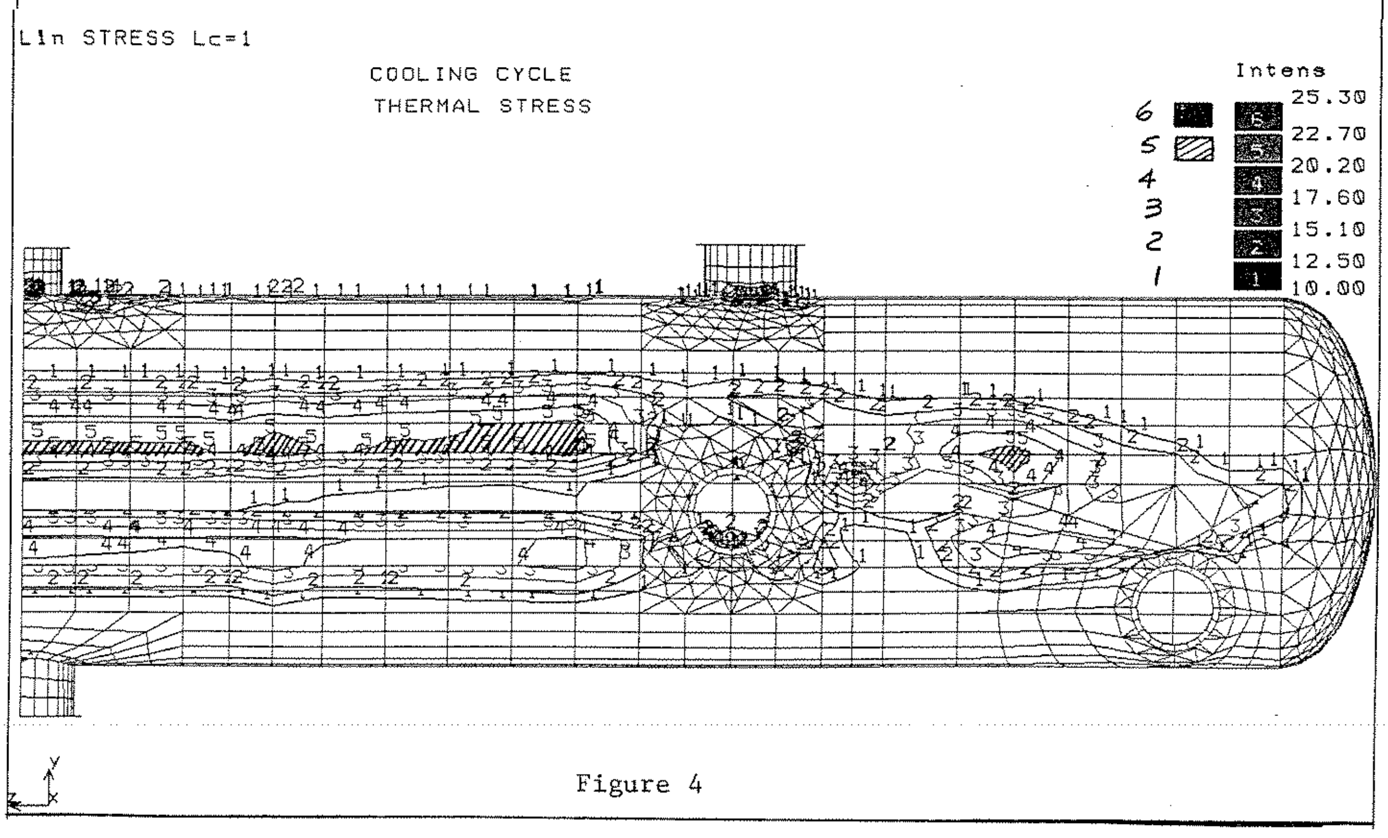

Lin STRESS Lc=1
HEATING CYCLE
THERMAL STRESS
Intens
25.20
22.70
20.10
17.60
15.10
12.50
10.00

Figure 3



Lin STRESS Lc=1
COOLING CYCLE
THERMAL STRESS
Intens
25.30
22.70
20.20
17.60
15.10
12.50
10.00
6
5
4
3
2
1



Lin STRESS Lc=1
COOLING CYCLE
THERMAL STRESS
Intens
25.30
22.70
20.20
17.60
15.10
12.50
10.00
6
5
4
3
2
1

Figure 4