Structural Design of the IHX for PFBR

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INTRODUCTION

The intermediate heat exchanger (IHX) (Fig.1) transfers the heat from primary sodium to secondary sodium in Prototype Fast Breeder Reactor (PFBR). The hot and cold primary and secondary sodium temperatures are 803 K/653K and 778 K/623 K respectively. While the top tube sheet is exposed to high temperature which is in the creep range, the bottom tube sheet is subjected to severe thermal shocks in case of secondary sodium pump trip, reactor scram, etc. In order to check the design adequacy of the tubesheets under above situations, detailed heat transfer and mechanical analysis have been carried out.

ANALYSIS-METHODOLOGY

To perform elastic as well as inelastic analysis for the tube sheets of IHX, a finite element computer code 'NONLIN' has been developed. The code uses 8 noded isoparametric, axisymmetric elements for the thick tube sheet portions and two noded axisymmetric spring element for the tubes and the central shell welded to the tube sheets. The material modelling aspects like yield criteria, flow rule, monotonic, cyclic and creep hardening laws etc. are based on the interim guidelines of ORNL (Pugh, et al, 1972). The tubesheets along with the connecting tubes and the shell are modelled together as an axisymmetric structure. For the mechanical analysis of tubesheets the perforated portions are replaced by the homogeneous solid plates with the modified elastic constants. Further, they are assumed to always follow elastic behaviour which gives rise to higher strain accumulations in the solid rim portions due to elastic followup (Van de Graaf, et al, 1985). The equivalent solid portions as well as the surrounding solid rims are modelled using 8 noded isoparametric elements. Each row of tubes is modelled with 2 noded axisymmetric spring elements with the same axial stiffness value as that of the tubes in that row. The bending stiffness offered by the tubes on the perforated portions are considered negligible. The FEM model of IHX, used for the analysis is shown in the Fig.2.

In the heat conduction analysis of tube sheets, the temperature distributions in the perforated portions and solid portions have been determined separately. For the perforated portion, a unit cell model has been used. As per this model the radial temperature gradient is neglected and axial gradient is determined using one dimensional analysis with fin analogy. For the remaining solid portions, the conventional finite element analysis with 8 noded axisymmetric solid element, is performed. On the interface between the solid portion and the perforated portion, the heat transfer coefficients and temperatures corresponding to sodium flow in the last row of tubes are applied. For the analysis of bottom tube sheet with thermal baffles, the stagnant sodium in between the baffles is treated as the continuous solid with the heat transfer properties of sodium at appropriate temperature. The above model remains the same for both steady state as well as transient heat conduction analysis.

ANALYSIS-RESULTS

Design Conditions

The top tube sheet has to be designed for 0.85 MPa including creep effects under normal operation and 1.2 MPa excluding creep effects during closure of bottom isolation valve in one of the steam generators. The design life is 2×10^5 h for the design temperature of 811 K. The bottom tube sheet has to be designed for the pressure of 1.2 MPa and the temperature of 657 K. The design adequacy of the tube sheets is to be checked for the thermal transients like reactor scram, primary sodium pump trips, feed water pump trips, on-site or off-site power failure etc. The definitions for all the possible thermal transients have yet to be evolved. Presently, around 2000 normal operation to shut down cycles are considered for the tube sheets. However, for the bottom tube sheet an additional 100 secondary pump trips are included.

Determination of Tube Sheet Thickness

The stresses induced in the top and bottom tube sheets due to the mechanical loads are determined both for bent tube and straight tube concepts. The stresses in the top tube sheet for the differential pressure of 0.85 MPa are given in Table-I both for straight tube and bent tube cases. Fig.2 also shows the deformation pattern of the tube sheets for the straight tube concept. It is to be noted that the location of maximum stresses is different for the straight tube and bent tube cases. Even though the bending stresses near the junctions between tubesheet and connecting shells are considered as secondary in nature as per ASME, it has been treated as primary for the top tube sheet where creep effect is significant as suggested by Becht, et al, 1986. In order to meet the high temperature code case ASME N-47 for the primary stress limits, the minimum tubesheet thickness required for the top tube sheet is 80 mm in the case of straight tube and 190 mm for the bent tube case. Minimum tube sheet thickness required for the bottom tube sheet is 70 mm for straight tube case and 180 mm for the bent tube case. Considering the requirement of higher tube sheet thickness for the case of bent tube case and manufacturing cost involved, straight tube concept has been selected. One problem with the straight tube is that the differential expansion among the tubes gives rise to high compressive stress on the hotter tube. This is most severe for a tube which has to be plugged during service and always runs hot. However, the need for plugging is not foreseen as the tube ends would be eliminated in case of manufacturing defects of badly drilled tubesheet hole or defective tube to tube sheet weld and hence there is no risk of the thermal buckling.

TABLE-I: STRESSES IN THE TOP TUBE SHEET

Z-value mm	Radius = 235 mm*				Radius = 615 mm **			
	St.tube		Bend tube		St. tube		Bend tube	
	σ_r	σ_ϕ	σ_r	σ_ϕ	σ_r	σ_ϕ	σ_r	σ_ϕ
5	-1.0	-12.4	177.3	543.0	55.5	32.0	8.2	174.7
15	-1.7	-8.0	144.0	400.0	40.3	24.1	10.7	130.0
25	-0.4	-2.7	99.1	253.3	25.3	16.2	13.0	85.3
35	-1.0	-2.5	47.1	103.5	10.7	8.3	15.5	40.8
45	+2.3	+ 7.7	-7.6	47.5	-3.9	0.4	18.0	-3.7
55	3.8	+13.0	-63.1	-199.3	-18.6	-7.5	20.4	-48.2
65	5.1	+18.3	-117.1	-350.6	-33.5	-15.4	22.8	-92.9
75	3.1	+24.5	-188.6	-512.5	-48.8	-23.4	25.3	-137.7

Note: * Critical location for bend tube concept.
** Critical location for straight tube concept.

The minimum tube sheet thickness for the bottom tube sheet is slightly lower than that for top tube sheet, as a traditional approach, the thickness of 80 mm is selected for both the tube sheets. As far as bottom tube sheet is concerned, with the above thickness, the tube sheet can be protected against the severe thermal transients, by providing thermal baffles in radial/axial directions. On the other hand for the top tube sheet which is experiencing high temperature effects, the tube sheet thickness can be confirmed only after checking the design adequacy under operating conditions via detailed analysis.

Design Adequacy for the Top Tube Sheet Under Operating Condition

The temperature distributions (Fig.3) and in-turn thermal stresses are computed for the normal operation. The analysis indicated that the critical location is the junction between the top tube sheet and outer shell. Subsequently, the top tube sheet is checked for the design adequacy as per ASME Code Case N-47 as well as the French RCC-MR. As per the results of elastic analysis the top tube sheet does not meet the above code rules. Hence a simplified procedure which has been used for SNR-2 reactor (Vinzens, et al, 1986) and further detailed inelastic analysis with the code 'NONLIN' have been attempted and found that the tube sheet meets the design code requirement with a good margin. Fig.4 and 5 show the variation of effective stress vs time and effective strain range at each load cycle respectively at the junction. The following are some of the important results:

* The tube sheet meets the strain limits of ASME N-47 and RCC-MR.
* Total creep plus fatigue damage values are 3.63, 2.41 and 0.31 as per ASME N-47, RCC-MR and SNR-2 procedure respectively at the critical location.

Hence the top tube sheet does not meet the high temperature code rules via simple elastic analysis. By following the SNR-2 procedure, it is possible to meet the ASME N-47 and RCC-MR code rules. However to confirm this detailed analysis has been performed, as per this,

accumulated inelastic membrane strain at the end of life	= 0.3154%	(< 1%)
membrane + bending strain	= 1.58%	(< 2%)
local strain	= 2.2%	(< 5%)

The comparison of permissible cycles by various approaches are given in Table-II.

TABLE-II: COMPARISON OF PERMISSIBLE CYCLES

Detail	Elastic		Simplified Analysis (SNR-2)	Inelastic N-47/RCC-MR
	N-47	RCC-MR		
Creep damage/ cycle (Nc)	0.385×10^{-3}	1.2×10^{-3}	0.150×10^{-3}	0.1×10^{-3}
Fatigue damage/cycle (Nf)	1.43×10^{-3}	0.01×10^{-3}	0.005×10^{-3}	0.3×10^{-6}
Total damage/cycle (D=Nc+Nf)	1.815×10^{-3}	1.21×10^{-3}	0.155×10^{-3}	0.1×10^{-3}
Admissible damage(Dmax)	1	0.9945	0.959	1.0
No. of permissible cycle N = Dmax/D	550	825	6185	1×10^{4}

CONCLUSION

* straight tube design has significant advantages over bent tube design
* minimium tube sheet thickness of 80 mm has been recommended for both the tube sheets
* top tube sheet with the thickness of 80 mm does not meet the high temperature design requirements by simple elastic analysis. However, with the same thickness a simplified elastic-plastic analysis and further detailed inelastic analysis indicated that the tube sheet meets the code requirements
* the methodology has been established to analyse the bottom tube sheet with thermal baffles

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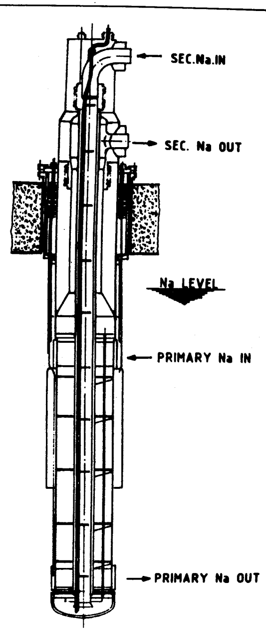

FIG. 1 PFBR IHX

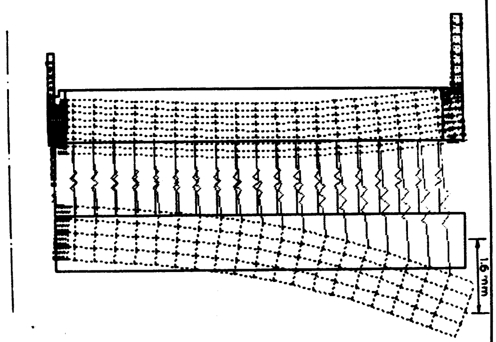

FIG. 2 DEFLECTION UNDER PRESSURE LOADS

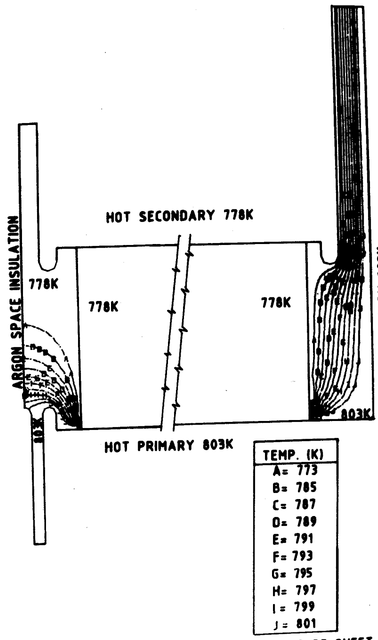

FIG.3 TEMPERATURE DISTRIBUTION IN THE TOP TUBE SHEET

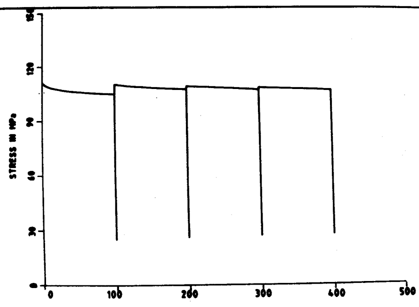

FIG.4 EFFECTIVE STRESS WITH TIME AT HOTTER FACE

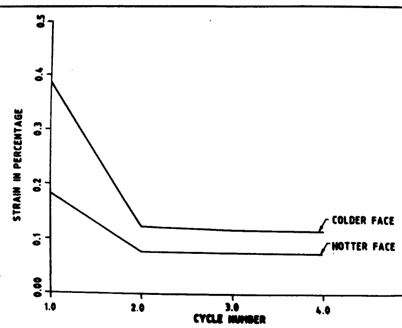

FIG.5 EFFECTIVE STRAIN RANGE WITH CYCLE