MECHANICAL DESIGN OF A SODIUM HEATED STEAM GENERATOR

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I. INTRODUCTION

In the nuclear energy field today, the sodium cooled fast breeder holds a unique position. Choice of sodium is dictated by nuclear and heat transfer considerations and hence one has to learn to live with certain technological difficulties. Chemical incompatibility between sodium; whose choice is dictated by the reactor; the source of energy and water steam required for the conversion of heat in electrical output through turbo alternator is indeed severe.

One of the major problems associated with sodium cooled fast reactor programme has therefore been the design of reliable and economical sodium heated steam generators.

This paper outlines the nature of the problems to be encountered in the design of sodium heated generators as studied by our group while working on the steam generators of "FBTR" the first fast reactor (15 MWe) now being built at Kalpakkam.

II. STEAM GENERATOR OF FBTR

Steam generators for FBTR have been designed to produce 74 tons/hr of steam at 480°C and 125 Bar. The total output of 50 MWth is obtained from 4 modules of 12.5 MWth each.

Some salient features of this steam generator are

- Once through type
- Absence of tube sheet
- Absence of baffles
- High value of pitch/tube outside diameter
- Use of Niobium stabilised /2¼Cr -1Mo ferritic steel as principal construction material

Sodium-water incompatibility strongly influences the choice of steam generator; i.e. (between re-circulating units of the drum type and once through type); material construction; spacing of tubes and the mechanical design of the unit.

Once through steam generators are more safe, economical and reliable than the drum type. The pitch of tube/tube outside diameter is much higher than the conventional heat exchanger so that in case of failure of one tube, the sodium water reaction products do not severely damage the adjoining tubes. Amongst the "Once-through type" units; modular approach leading to large number of small units is favoured from considerations of greater availability and safety.

Baffles are not normally required as sodium is an excellent heat transfer medium because of its high thermal conductivity.

Presence of sodium at relatively high temperature calls for the use of austenitic stainless steel, but the presence of chloride ions in water under conditions of mal-functioning of the purification system can lead stress corrosion of austenitic steel. $2\frac{1}{4}$Cr-1Mo ferritic steel used in high temperature conventional boilers on the other hand is susceptible to decarburisation at high temperature. Decarburisation has a dual disadvantage; loss of strength of the ferritic material and embrittlement of austenitic stainless steels. Hence, $2\frac{1}{4}$Cr-1Mo stabilised with niobium has been selected as the principal material of construction.

The general description and the operating condition of the steam generator for the full reactor power operation of 50MW(t) are as follows. (see also fig. 1)

Power exchanged per module MW(t)	12.5
Sodium inlet temperature (°C)	510
Sodium outlet temperature (°C)	295
Feedwater inlet temperature (°C)	200
Steam outlet temperature (°C)	480
Outlet steam pressure (bar)	125
Number of tubes	7
Tube outside diameter (mm)	33.7
Tube wall thickness (mm)	4
Pitch of tubes (mm)	60, triangular
Heat transfer length (m)	90.4

Fluid in the tube	water-steam
Fluid in the shell	sodium
Shell outside diameter (mm)	193.7
Shell wall thickness (mm)	8

The total heat transfer length of 90.4 meters is arranged in zig-zag form of "3S" to conserve space. Since there are only 7 tubes to be managed, tube to tube sheet joint has been avoided. The shell is enlarged at either ends and each of the 7 tubes come out from the shell through thermal baffles.

III. MECHANICAL DESIGN

General

The general design has been as per ASME Section VIII, combination of stresses as per ASME Section III and the fatigue analysis as per ASME Code case 1331-4.

(i) Tube Side Design

The design involves the stress calculations due to

- Internal pressure
- External pressure due to sodium water reaction
- Weight
- Differential thermal expansion between shell and tube
- Thermal stresses in the steady state and transient operation
- Thermal cycling of transition point
- Flow induced vibration
- Stratification

The tube thickness has been calculated for the design conditions of internal pressure of 140 Bars and temperatures of 520°C.

Tubes are checked for external pressure resulting from sodium water reaction of 78 Bars in the leaking module and 54 Bars in the adjacent module (The four modules are grouped in 2x2)

The stress due to weight is calculated by taking the tube to be simply supported between the supports placed at 1560mm apart and uniform load distribution.

The thermal stress due to differential expansion between shell and tube has been calculated on a conservative basis by neglecting the effect of bends and taking the shell and tube to be composite structure held at both the ends.

$$\text{Thermal stress in the tube} = \frac{\frac{1}{L}\, E\alpha}{1+\frac{\text{Area of tube}}{\text{Area of shell}}} \int_0^{\ell} (T\ \text{shell} - T\ \text{tube mean})\, d\ell$$

Where L is the heat transfer length of the steam generator.

The steady state thermal stress due to radial temperature distribution is calculated as per Ref.2 and the value at inside surface is 0.76 E T for tube of 25.7/33.7mm diameter.

Code is under preparation to calculate the transient thermal stresses resulting from water pump stoppage and sodium pump stoppage.

Thermal shock at water inlet side. Rise of sodium temperation from 295 to 510°C in 200 seconds.

Thermal shock at steam outlet side: Decrease of steam temperature from 480°C to 325°C in 30 seconds.

The conventional Fritz's method of calculation is not valid as the temperature over the tube section is not uniform at start of shock and no side of the tube is insulated.

The thermal sleeves provided, where the tubes leave the shell on either side are checked for fatigue analysis by computing the alternating stress intensity to take into account the following stresses:

- Pressure
- Axial and radial temperature gradient
- Thermal expansion of the tube bends between module and water-steam subheader
- Stress concentration

As the boiling changes from nucleate to film boiling or from film to nucleate in the evaporator portion of the once through steam generator, there are significant changes in the heat flux and the temperature distribution across the wall (see fig.2 for temperature distribution along the length of steam generator). This is termed as thermal cycling of the transition point and fatigue analysis is called for as the frequency of the shift is reported to be 1 cycle/sec to 1 cycle per 10 seconds.

Fatigue analysis of the transition point has to be carried out of if $\delta\Delta t$, the difference between the Δt across wall under nuclear boiling conditions and the Δt across wall under film boiling exceeds the value of 18.4°C (Value permitted of $\delta \Delta t$ for which fatigue analysis is not required for our steam generator conditions as per ASME Section III). The stress intensity has been computed by taking into account the thermal stress due to radial temperature distribution and axial temperature. The transition point is assumed to oscillate between the mean temperature of the tube at position before critical point and just after critical point. The stress calculation due to axial temperature shift is done as per analysis of bimetallic joints (Ferritic steel to austenitic stainless steel) in Ref. 2.

Longitudinal thermal stress at inside surface of the tube at just before critical point.

$$= 0.715\, E \alpha \Delta T_1 \left(1 + \frac{\text{tube thickness}}{3 \text{ tube inside radius}}\right) + 0.23E \left(\alpha_2 T_2 - \alpha_1 T_1\right) \quad \text{.....(1)}$$

Longitudinal thermal stress at inside surface of the tube at just after critical point.

$$= 0.715\, E \alpha \Delta T_2 \left(1 + \frac{\text{tube thickness}}{3 \text{ tube inside radius}}\right) - 0.23E \left(\alpha_2 T_2 - \alpha_1 T_1\right) \quad \text{.....(2)}$$

$\Delta T1$ = Wall temperature difference at just before critical point

$\Delta T2$ = Wall temperature difference at just after critical point

T2 = Mean temperature of the tube at just after critical point

T1 = Mean temperature of the tube at just before critical point

The alternating stress intensity at the critical point calculated by half the algebric difference of the longitudinal stress at just before critical point and at just after critical point, is compared with the admissible fatigue strength Sa for 10^6 cycles.

The Δt, in general, can be reduced by increasing the water-steam side pressure, water flow rate and reducing the heat flux by lower LMTD.

Since this steam generator is a horizontal unit, there is danger that at certain flow conditions water-steam phase may separate out in the evaporator zone causing abrupt changes in the tube temperature. The problem is of hydro-thermal nature, firstly finding the conditions under which stratification can occur and then computing the thermal stresses.

The sodium flow in the shell is parallel flow. The amplitude of the vibration of tubes is calculated as per Ref.4 and the bending stress is calculated for this amplitude with both ends considered simply supported in one case and clamped in other case. The vibration of tubes and U bends is checked as per Ref. 5. The expansion bends between the module and subheader is checked against vibration to make sure that the exciting frequency does not approach the natural frequency of the bends.

(ii) Shell Side Design

The shell is designed as per the formulations given in ASME Section VIII for the following three cases.

i) Steady state pressure. The admissible stress is taken from the criteria of ASME Section VIII.

ii) Transient pressure in the accidental module; The admissible stress is the instanteous rupture strength of the material at design temperature.

iii) Transient pressure in the module adjacent to leaking module. The admissible stress is the lowest of the following three values.

i) 90% of the elastic limit at design temperature

ii) 1/3rd of the instanteous rupture strength at design temp.

iii) 2/3rd of elastic limit at ambient temperature.

The shell side design for steam generator is governed by the transient pressure of 54 Bars in the module adjacent to leaking module with admissible stress as 1/3rd of instanteous rupture strength at design temperature of 520°C.

The thermal stress calculation is done in the same way as indicated under tube side design.

(iii) <u>Steam Generator Support</u>

The four modules are located inside a common structural casing and welded to the loop inlet and outlet headers. The steam generator is supported by hangers suspended from casing top in the cold conditions and on the twelve hot beams fabricated in a square section from special heat and corrosion resistance steel plate APS 10 M 4 (C=0.12%, Cr=2%, Al=0.35%). The sodium water-steam headers are fixed at one end and free at the other end. The sodium header fixation support is designed to take into sodium water reaction pressure in the header and thermal loadings at the anchor point. The steam-water header fixation support is designed for thermal loadings at the anchor point.

IV. <u>FUTURE STEAM GENERATORS</u>

FBTR concept of modular steam generator may not be economical for large commercial plants as the number of such modules required will be enormous and the trend is to go for large sized few units of vertical tube with closely spaced tubes. The design will get focussed on the tube sheet and tube to tube sheet joint. Detailed mathematical and experimental stress analysis will be required for tube sheet design specially with respect to thermal fatigue.

V. <u>REFERENCES</u>

1. ASME Boiler and Pressure Vessel Code, Section III, Section VIII and Code Intrepretation 1331.

2. Pressure Vessel Design by Harvey.

3. Design notes under Indo French Technical Collaboration for the project (classified).

4. Vibration of rods in parallel flow. Nucleonics (August 1959).

5. TEMA Code.

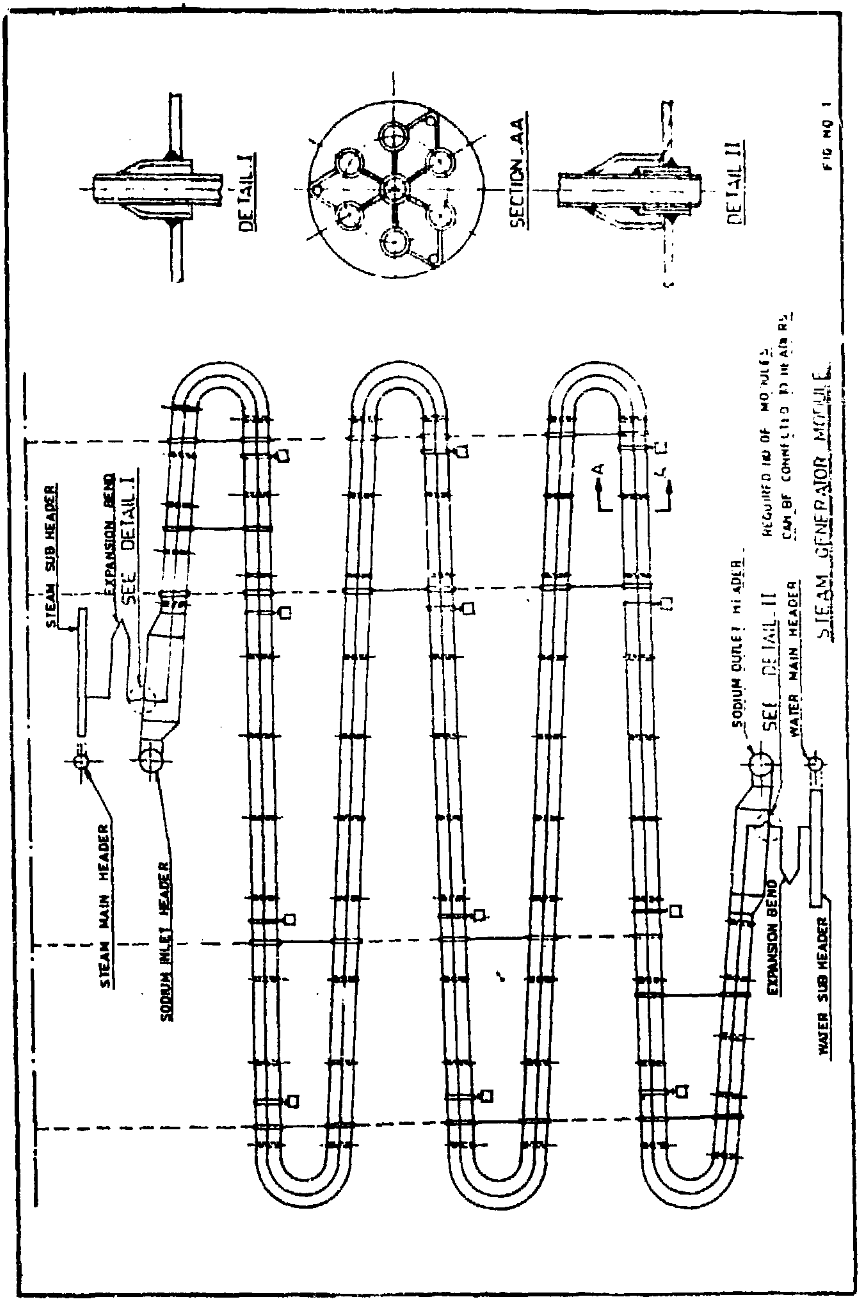
DETAIL I
SECTION AA
DETAIL II
STEAM SUB HEADER
EXPANSION BEND
SEE DETAIL I
STEAM MAIN HEADER
SODIUM INLET HEADER
A
A
REQUIRED NO OF MODULES
CAN BE CONNECTED TO HEADERS
SODIUM OUTLET HEADER
SEE DETAIL II
WATER MAIN HEADER
EXPANSION BEND
WATER SUB HEADER
STEAM GENERATOR MODULE

FIG NO 1

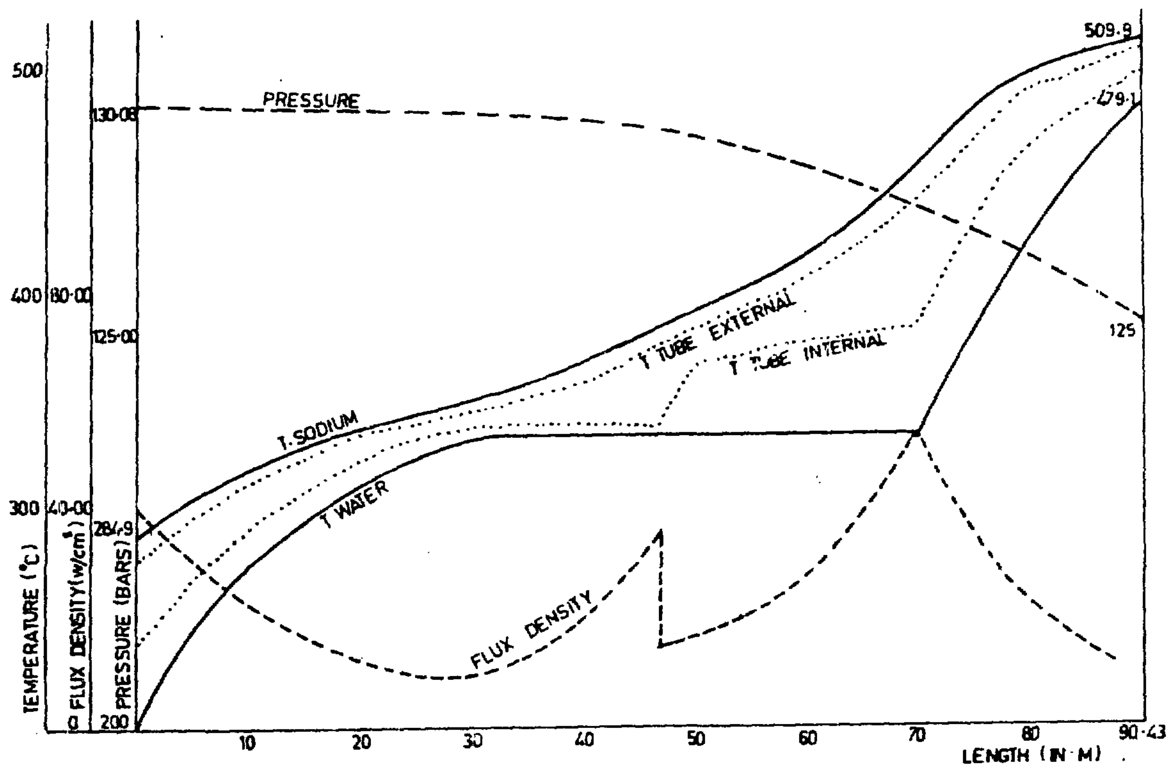
PRESSURE
T. SODIUM
T TUBE EXTERNAL
T TUBE INTERNAL
T WATER
FLUX DENSITY
509.9
479.1
125
TEMPERATURE (°C)
FLUX DENSITY (w/cm²)
PRESSURE (BARS)
500
400
300
200
80.00
40.00
0
130.08
125.00
28/.9
10
20
30
40
50
60
70
80
90.43
LENGTH (IN M)

FIG.- 2