UCRL-

CR-105908

UCRL-CR--105908

DE91 008566

University of California
Department of Mechanical, Aeronautical and Materials Engineering
Davis, California 95616

HYLIFE-II POWER CONVERSION SYSTEM DESIGN AND COST STUDY

Final Report No. 1

September, 1990

Principal Investigator: Professor Myron A. Hoffman

Research Assistant: Ying Lee

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

Work performed on Intramural Purchase Order No. B108485 from the
Lawrence Livermore National Laboratory, Livermore, CA 94550

MP
DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MASTER

HYLIFE-II POWER CONVERSION SYSTEM DESIGN AND COST:

EXECUTIVE SUMMARY OF FIRST YEAR'S RESULTS

1. Definition of the Power Conversion System

The power conversion system for the HYLIFE-II fusion power plant has been defined to include the IHX's (intermediate heat exchangers) and everything to the right on Figure 1 of Attachment 1 (Hoffman, 1990). It is referred to simply as the BOP (balance of plant) in the rest of this report. The above is a convenient division between the reactor equipment and the rest of the fusion power plant since the BOP design and cost then depend only on the specification of the thermal power to the IHX's and the temperature of the primary Flibe coolant into and out of the IHX's, and is almost independent of the details of the reactor design. The main efforts during the first year have been on the definition and thermal-hydraulics of the IHX's, the steam generators and the steam power plant, leading to the definition of a reference BOP with the molten salt, Flibe, as the primary coolant. A summary of the key results in each of these areas is given in the following sections.

2. Intermediate Heat Exchangers

The first components of the reference BOP are the three cross-flow IHX's, each with a nominal rating of 1100 MWth, since the total thermal power from the fusion reactor is about 3300 MWth. (See Figure 7 of Attachment 1 for the power balance for the Base Case D reactor.) Each IHX supplies heat to a steam generator consisting of two evaporators with a nominal rating of 400 MWth each and one superheater with a nominal rating of 300 MWth.

The geometry and thermal-hydraulic performance of the IHX's have been

studied in detail for the case of low-viscosity FLiBe as the primary fluid and NaBF_4 (sodium fluoroborate) as the secondary fluid. A preliminary choice for the reference IHX geometry has been made on the basis of a simplified cost model. These results are reported in Attachment 1 (the paper for the 9th Topical Meeting on the Technology of Fusion Energy). The primary FLiBe exits the reactor and enters the IHX's at 923 K (650 C) and exits the IHX's at 873 K (600 C). About 81% of the FLiBe flow from the reactor bypasses the IHX's; hence only about 19% of the flow passes through the IHX's in order to produce the temperature drop of 50 C. This results in a reasonably compact IHX with a tube-bundle volume of only about 112 cu.m. and only 7880 tubes per IHX. This is comparable to a steam generator for a PWR (pressurized water reactor) of the same thermal rating.

One key issue is whether a modified Hastalloy N will have to be used for all surfaces in contact with the molten salts or whether a modified 316 SS (stainless steel) will have adequate corrosion resistance. It is estimated that the IHX will cost roughly \$55/kWth (in 1988 dollars) if it is made of Hastalloy N, while the cost may be as low as \$18/kWth if it can be made of some type of 316 SS. This issue affects not only the cost of the IHX's but also the cost of the reactor chamber, all the piping, the tritium removal equipment and the steam generator. Consequently, it must be addressed in FY91.

Although we are using our own simplified performance and cost equations for this BOP study, we have been reviewing the relevant equations in the SAFIRE code as well. When a serious overestimate of the IHX pumping power was discovered in the SAFIRE output, we examined the equation used in the SAFIRE code and found that it did not represent our IHX's using molten salts at all. The suggested equation for the pumping power (in MW) for the primary FLiBe coolant (subscript pc in the SAFIRE code) is:

$$\text{Primary } P_{P(PC)} = A_{1(FRIC)} \left(\frac{3000}{P_{th(TOT)}} \right)^{1.6} (\dot{m}_{PC(TOT)})^{2.8}$$

where $A_{1(FRIC)} = 5.12 \times 10^{-13}$ for a reference total thermal power of 3000 MW, and $P_{th(TOT)}$ is the actual total thermal power in megawatts. (The gravity contribution to the pumping power is included in with the bypass flow in the SAFIRE code.) The suggested equation for the secondary NaBF₄ coolant is of the same form with $A_{2(FRIC)} = 2.75 \times 10^{-14}$. These equations give results consistent with those found in the detailed IHX thermal-hydraulic design study.

3. Steam Generators

The steam generator design adopted for the HYLIFE-II BOP is a scaled-up version of the steam generator proposed for the CRBR (Clinch River Breeder Reactor). The CRBR steam generator consisted of two evaporator units and one superheater unit, all of an identical "hockey-stick" design. The steam generator had a nominal rating of about 325 MWth. Since each IHX supplies about 1100 MWth to the steam generator, our units must have about three times the number of tubes of the CRBR units. Each of the two evaporator units per IHX has a nominal rating of 400 MWth and supplies saturated steam at a quality of 0.50. The steam is separated from the water and sent to a single superheater unit per IHX with a nominal rating of 300 MWth. The steam exits the superheater at about 800 K (980 F) and about 16 MPa (about 2300 psia). There are three steam generators in all, one for each IHX, and hence a total of 9 hockey-stick units.

A detailed computer code for the steam generator has been written by Mr. Ying Lee and has been validated by comparison with the design and performance information available on the CRBR steam generator. The details of this work can be found in Attachment 2. This code will be used in FY91 to evaluate the

capital cost of the steam generators and to select the "best" steam generator geometry for minimum COE (cost of electricity).

The usual capital cost model for components like the steam generator units is simple to specify a cost per kilowatt thermal handled. This type of cost model is clearly inadequate for determining the best geometry of the steam generators for minimum COE. In order to determine the cost scaling of the steam generator units with the geometry, an improved cost model based on the number of tubes, the length of the tubes and the material selected has been developed (based on a cost model developed by Battelle Pacific Northwest Laboratories for dry cooling tower heat exchangers and condensers). The resulting equation for the capital cost (in millions of 1988 dollars) is:

$$C_{SG} = (32 N_c + 4.8 \times 10^5) (CF_L + CF_{ts}) + (800 N_c \frac{t_w}{0.001})] CF_{Material} CF_{NG}$$

where t_w is the tube wall thickness (in meters). The CF's (correction factors) for tube length, L, and tube-sheet cost (subscript ts) are given by:

$$CF_L = (0.050 L + 0.58)$$

$$CF_{ts} = (0.10 \ln N_c - 0.417)$$

The correction factor for the material selected has been estimated to be about 1.0 for modified 316 SS and about 3.0 for modified Hastalloy N. The correction factor for nuclear grade (NG) has been estimated at 2.5. However, better values for these correction factors must be obtained in FY91.

4. Reference Steam Power Plant and BOP

The reference SPP (steam power plant) cycle is very similar to the one selected for the HYLIFE-I power plant shown in Figure 8 of Attachment 1. The cycle is a modern subcritical, regenerative cycle without reheat. For our case, the steam exits the superheater at about 800 K (980 F) and at 16 MPa (about 2300 psia). The reference SPP is assumed to use conventional evaporative (wet) cooling towers. (The impact of using dry cooling towers will be assessed in the second year.) This results in a thermal efficiency of about 41%. For the Base Case D reactor power balance, the circulating power fraction is about 21%. This results in an overall fusion power plant efficiency of about 33% and a net electric power output of 1083 MWe (see Figure 7 of Attachment 1).

With the definition of the IHX's, the steam generators and the steam power plant, the reference BOP is essentially defined. However, much remains to be done to fill in the details, assess the costs and examine promising alternative design features. The work planned for the second year's effort will be described in the next section.

5. Plans for the 2nd Year's Effort

A basic goal of the second year's effort will be to develop cost models for each of the major components and subsystems which will allow us to optimize the BOP for minimum COE (cost of electricity). The BOP is presently estimated to represent at least 32% of the total direct capital cost of the HYLIFE-II fusion power plant. If efforts to reduce the cost of the drivers to one-third their present amount are successful, then the BOP direct capital cost will constitute about 48% of the total direct cost. Consequently, it is clearly worthwhile to examine various ways to minimize the BOP costs.

The second year's effort will include a study of many options such as those listed below; the exact list of options to be studied will be determined in consultation with Dr. Ralph Moir and the other researchers participating in the HYLIFE-II study.

1. The IHX has been modelled as a simple, single-pass cross-flow heat exchanger with flibe as the primary coolant and sodium fluoroborate as the secondary coolant. Since the molten salts may require special alloys such as a modified Hastalloy N or perhaps a modified stainless steel for good corrosion resistance at high temperature, the cost of the IHX and other components in contact with the molten salts can be significant. In order to minimize these costs, other heat exchanger geometries as well as heat transfer augmentation concepts will be examined. In addition, the design problems and cost of employing a double-walled IHX will be evaluated, just in case this turns out to be necessary for tritium containment.

2. It is presently proposed to use sodium fluoroborate as the secondary coolant which goes to the steam generator. If the environmental impact and safety studies now underway at INEL (Idaho National Engineering Laboratory) indicate that this molten salt is satisfactory from a tritium containment point of view, we will examine its impact on the cost of the steam generator. If this molten salt turns out to be unsatisfactory, then alternative fluids will be examined to see their impact on the design and cost of the steam generator.

3. It is very likely that dry cooling towers will have to be used in most future power plants including fusion power plants. The use of a dry cooling tower instead of a wet cooling tower reduces the amount of water required for

heat rejection, but it also reduces the thermal efficiency of the steam power plant. We propose to study the impact of the use of dry cooling towers on the steam power plant cycle as well as its effect on the COE for the HYLIFE-II fusion power plant.

4. Cost models for each of the components and subsystems of the BOP will be developed in order to assess the impact of each design change on the COE and to determine a final BOP configuration which minimizes the COE.

5. In support of the continuing effort to update and improve the SAFIRE ICF systems code, we will also compare our BOP design and cost models to those in the SAFIRE code. Wherever it is deemed necessary, revisions to the design and cost algorithms in the SAFIRE code will be proposed for the $\text{Flibe}/\text{NaBF}_4$ coolants of HYLIFE-II.

6. Finally, we will provide scale drawings of the plan and elevation views of the IHX's, the steam generators, the piping and the reactor in the reactor building, in order to give a clear idea of the relative size of the various components.

ATTACHMENT 1

**THE HEAT TRANSPORT SYSTEM AND PLANT DESIGN
FOR THE HYLIFE-II FUSION REACTOR**

Submitted to the 9th Topical Meeting
on the Technology of Fusion Energy,
October 7-11, 1990

THE HEAT TRANSPORT SYSTEM AND PLANT DESIGN FOR THE HYLIFE-II FUSION REACTOR

MYRON A. HOFFMAN

Department of Mechanical Engineering
University of California
Davis, CA 95616
(916) 752-2630

ABSTRACT

HYLIFE is the name given to a family of self-healing liquid-wall reactor concepts for ICF (inertial confinement fusion). This HYLIFE-II concept employs the molten salt, Flibe, for the liquid jets instead of liquid lithium used in the original HYLIFE-I study (Blink, et. al., 1985). A preliminary conceptual design study of the heat transport system and the BOP (balance of plant) of the HYLIFE-II fusion power plant is described in this paper with special emphasis on a scoping study to determine the best IHX (intermediate heat exchanger) geometry and flow conditions for minimum COE (cost of electricity).

REACTOR CHAMBER OPTIONS

The overall fusion power plant block diagram is shown in Fig. 1 for the case of one chamber. In this section we will briefly review some chamber concepts considered for the HYLIFE-II power plant for a nominal electric power production of about 1000 MWE.

The first HYLIFE-II reactor design considered consisted of 3 chambers producing about 1087 MWh per chamber; this case has been defined as Case A in the Project Report (Moir, et al, 1990). For the selected Flibe jet injection velocity, V_o , of 11.6 m/s and the fall distance between shots of 5.0 meters, this led to a repetition rate (i.e., the rate of microexplosions) of about 2.7 Hz, as shown on Fig. 2. For the specified driver energy, E_d , of 5 MJ and gain, G , of 70 (Bangerter, 1988) and a total fusion energy multiplication, M_f , of 1.15 (Tobin, 1990), the average thermal power produced in each of the three chambers of Case A is then:

$$F_{th} = E_d \cdot G \cdot F_{rep} \cdot M_f = 1087 \text{ MWth}$$

The injection velocity of 11.6 meters requires an effective static head above the nozzle exits of about 6.9 meters.

Cases B and C were other multi-chamber options discussed in Moir, et al (1990). These options are no longer of interest.

It is most desirable for low COE (cost of electricity) to obtain the total 3260 MWh from a single chamber. For the specified driver energy of 5 MJ and pellet gain of 70, this is possible for a compact chamber design with a liquid-jet fall distance between shots of 2.1 m and an injection velocity of about 16.2 m/s, leading to a rep rate of 8.1 Hz. This is referred to as Case D on Fig.2, and is our Base Case for this study; various concepts for achieving a single-chamber reactor are discussed in a companion paper by Moir (1990).

It should be noted that the design of the BOP (balance of plant) and the IHX's (intermediate heat exchangers) is essentially independent of the number of chambers and detailed configuration of the chambers once the total thermal power produced in the chambers and the temperature drop across the primary side of the IHX's are specified.

PRIMARY LOOP FLOWS AND PUMPING POWER

The jet array flow rate is determined by the injection velocity, the array geometry, and the requirement for an effective line thickness of Flibe of 0.5 meters at the plane of the microexplosion (subscript mp). For steady flows:

$$Q = V_{mp} A_{mp} \\ = \sqrt{V_o^2 + 2g\Delta H_{mp}} \pi (R_o^2 - R_i^2) PF_{mp}$$

For an annular array of jets with an inner radius of 0.5 m, an outer radius of 1.5 m and a packing fraction, PF_{mp} , of 0.50, the required jet flow for the single-chamber Base Case D is 53.6 cu.m/s, while each of the 3 chambers of Case A requires 44.8 cu.m/s, as shown in Table 1.

The mass flow rate to each IHX has been determined

Figure 1. The HYLIFE-II fusion power plant schematic showing one reactor chamber and one representative IHX and steam generator.

by specifying the temperature drop on the primary side of the IHX's and a nominal thermal power for each IHX of 1000 MW. The cooler flow exiting the IHX's is used to create a spray flow to enhance the condensation of the vapor after a shot (Bai and Schrock, 1990) as well as providing flow to cool the FSW (first structural wall). These flows then mix with the jet array flow in the bottom pool.

The primary FLiBe flow exiting the bottom of the reactor is divided into two main flows as indicated on Figure 1. Most of the flow bypasses the IHX's (intermediate heat exchangers); in most cases the bypass flow rate is equal to the jet array flow rate.

For the FLiBe pumps, a scaled up version of the molten salt pumps designed for the primary loop of the MSBR (molten salt breeder reactor) in Rosenthal, et.al. (1972) were adopted. This pump was designed to provide a head of up to 46 meters with a flow rate of about 1.2 m³/s. We assumed that these pumps could be scaled up to about 5 m³/s. For the total flow rate per chamber for Case A of 46.8 m³/s, we would require about 9 of these pumps for the bypass flow and 1 pump for the IHX flow for each chamber. For Base Case D with a total flow rate of about 66 m³/s, we would require 11 pumps for the bypass flow and 3 pumps for the IHX flow (see Table 1).

Figure 2. The required repetition rate versus injection velocity of the liquid jets for two representative fall distances of the jets between shots.

TABLE 1
Jet Array and Primary Loop Parameters

	Case A	Case D	
No. of Chambers	3	1	
Fall Distance between shots(m)	5.0	2.1	
Repetition Rate (Hz)	2.7	8.1	
Injection Velocity, V_o (m/s)	11.6	16.2	
Static Head Required to Produce V_o (m)	6.9	13.4	
<u>Vol. Flow Rates/Chamber (m^3/s):</u>			
1. Jet Array (Bypass Flow)	44.8	53.6	
2. Spray (Max.) } (IHx Flow)	3.2	9.7	
3. First Wall } (IHx Flow)	0.9	2.6	
Total Flow/Chamber	48.9	65.9	
<u>No. of Main Pumps/Chamber</u> (for Rated Flow per Pump = $5 m^3/s$):			
Bypass Flow	9	11	
IHX Flow	1	3	
<u>Method Used to Produce V_o</u>			
	Static Head	Static Head	Press. Pipes
<u>Bypass Pumping Power (MW_e):</u> (for $\eta_p = 80\%$)	3x24	~37	~37
<u>Bypass Pump Head</u>			
Gravity Head above Pool (m)	13.2	19.8	10.4
Friction + Minor Losses (m)	7.6	7.5	16.7
Total Pump Head (m)	20.8	27.3	27.1
<u>Bypass Pipes: Inner Diameter (m)</u>	1.0	1.6	1.0
<u>Estimated Total Flibe Inventory (m^3)</u>	600	960	750

A study was done to see the impact of the pipe diameter on the pumping power and the Flibe inventory. The results of this study indicate that a pipe inner diameter of about 1.0 meter is a good compromise choice and results in an acceptable Flibe velocity in the pipes of about 6.2 m/s (Moir et al., 1990). The values of pumping power and

Flibe inventory for this pipe diameter are given in Table 1 for Cases A and D.

The required pumping power has been estimated assuming that none of the dynamic head in the jet array is recovered in the pool at the bottom. The required net positive suction head was estimated using the same specific

speed, N_s , of 2725 and suction specific speed, S_s , of 11,000 (in units of $\text{RPM} \cdot \text{GPM}^{0.5} / \text{ft}^{0.75}$) chosen for the HYLIFE-I design (1978 Laser Program Annual Report, 1979). These mechanical pumps are assumed to have an isentropic efficiency of about 80% based on the more recent study of mechanical pumps for liquid metals for the Pulse*Star ICF reactor (McDowell and Murray, 1984).

The calculations indicate that the required total pump head for the bypass loop is then about 21 m for Case A and about 27 m for Case D when a static head is used to produce the injection velocity. The resulting pumping power required for Case A is about 73 MWe for three chambers as shown on Table 1. For the single-chamber Base Case D, the required pumping power is about 37 MWe. If pressurized piping can be used to produce the injection velocity, it is estimated that the pumping power will only be reduced slightly for Case D.

Recent neutronics studies of Tobin (1990) indicate that about 6.5% of the neutron energy passes through the jet array and is deposited in the chamber side-wall materials. In order to recover as much as possible of this energy and to keep the maximum FSW (first structural wall) below 700 C, the wall is cooled with Flibe. As can be seen from Table 1, only a modest total flow rate of about 2.6 cu.m/s is required to cool the FSW adequately for Base Case D.

BALANCE OF PLANT

For this discussion, the BOP (balance of plant) is defined to include the IHX's (intermediate heat exchangers), and everything to the right of them on Figure 1. In this section, the steam power plant (SPP) model will be described.

In the MSBR (molten salt breeder reactor) study (Rosenthal, et al., 1972), they proposed using a supercritical steam cycle. The boiling pressure was 24.8 MPa (3600 psia) and the thermal efficiency for a maximum superheated steam temperature of 811 K (1000 F) was 44.4%. While this performance is attractive, recent experience with supercritical steam power plants has indicated some unanticipated reliability and maintenance problems. As a result, a subcritical cycle such as proposed for the HYLIFE-I reactor (1978 Laser Program Annual Report, 1979) is considered to be the more conservative choice today.

While modern steam powerplants have very complex cycles designed to produce high thermal efficiency, it is possible to develop a very simple model of an entire class of subcritical cycles which has adequate accuracy for this scoping study. The thermal efficiency of the real steam powerplant (SPP) can be obtained by modifying the Carnot efficiency:

$$\eta_{sb} = \eta_{SPP} (1 - f_{loss}) \eta_{Carnot}$$

where f_{SPP} is the fraction of the Carnot efficiency achieved by present-day SPP's (typically about 70%), and f_{loss} is the fraction of the thermal power which is lost to the cold shield, etc. (typically 0.01 to 0.05). The Carnot efficiency is given by:

$$\eta_{Carnot} = 1 - \frac{T_{min}}{T_{max}}$$

where T_{min} is the condensation temperature of the steam and T_{max} is the maximum superheat temperature of the steam.

COST MODEL FOR THE IHX SCOPING STUDY

The capital cost of the HYLIFE-II power plant has been taken from a preliminary study by Meier (1988). The direct capital cost of the entire fusion power plant, excluding only the IHX's, was estimated as \$2632 millions for a single-chamber design. The direct capital cost of each IHX was assumed to scale as:

$$C_{1, IHX} = 10 + 0.009 A_{1, IHX}$$

where A is the heat transfer surface area (in sq.m) of each IHX and the cost is in millions of dollars. This cost equation includes an extra cost for using a modified Hastalloy N for improved corrosion resistance to the molten salts. These values are rather rough and are now in the process of being refined. However, this simple model allows us to examine the effect of various design choices for the IHX's on the thermal efficiency of the steam power plant and the COE for the purposes of this scoping study.

The simplified COE equation used in this study is:

$$COE = \frac{FCR (C_{Dir} + C_{IHX}) F_{Indir} + C_{O\&M}}{(P_{th} \eta_{sb} - P_{circ}) (8760) A}$$

where FCR is the fixed charge rate, C_{Dir} is the direct capital cost excluding only the IHX's, C_{IHX} is the direct capital cost of the three IHX's, F_{Indir} is a factor accounting for all the indirect costs, $C_{O\&M}$ is the annual operations and maintenance cost, P_{th} is the total thermal power to the steam power plant, P_{circ} is the total circulating power and A is the availability.

IHX SCOPING STUDY

As part of the initial HYLIFE-II study using Flibe for the liquid jets, it was important to know if we had the choice of using either the high viscosity or the low-viscosity Flibe. The high-viscosity Flibe has a viscosity of about 1.0 Pa-s at its melting point of 363 C (almost like

honey) and about 0.04 Pa-s at 600 C (like soy bean oil). The low-viscosity Flibe has a viscosity of about 0.02 Pa-s at its melting point of 460 C and about 0.008 Pa-s at 600 C; this is still almost an order of magnitude higher than water at room temperature.

The high-viscosity Flibe has the potential advantage of a lower melting point (363 C = 636 K) than the low viscosity Flibe (460 C = 733 K). However, the higher viscosity could lead to excessively large heat exchangers.

The geometry chosen for the IHX scoping study is the simple, single-pass, cross-flow heat exchanger sketched in Figure 3. The hot Flibe from the reactor is assumed to flow in the tubes, while the cold secondary molten salt flows across the tube bank (in order to have the cooler fluid in contact with the shell).

Figure 3. The simple cross-flow IHX geometry used for the scoping study.

RESULTS FOR HIGH-VISCOSITY FLIBE

For the first part of the scoping study, it was assumed that high-viscosity Flibe would be used on both the primary and secondary sides of the IHX. The results for one of the reference cases given in Moir, et al (1990), Case 2H for a 100° C temperature drop on the primary side, will be described next. The equations used to model the IHX are given in Moir, et al (1990).

The temperatures chosen for Case 2H are shown at the top of Table 2. These temperatures were chosen based on four principal constraints:

1. The minimum temperature of the cold, secondary Flibe, T_{ci} , must be well above the melting point to avoid freeze-up in the steam generator.

2. The maximum temperature of the cold,

secondary Flibe, T_{co} , must be as high as possible for good thermal efficiency of the steam power plant (SPP).

3. The temperature of the hot primary flibe entering the reactor (normally equal to T_{hi}) must not be too high to keep the vapor density within limits for ion or laser beam transmission.

4. The temperature differences between the hot, primary flibe and the cold, secondary flibe must be large enough to provide the required heat transfer in the specified tube lengths.

The fourth constraint is especially critical for the high-viscosity Flibe, since the tube flow is laminar and the Prandtl number is very high, resulting in poor heat transfer and very long thermal development lengths. The temperature values shown in Table 2 are the result of the many compromises needed to satisfy all of these constraints simultaneously.

The rest of the key results given in Table 2 are for IHX's with a nominal thermal rating of 1000 MW. Note that small 10 mm diameter tubes were chosen for the high-viscosity Flibe because larger diameters resulted in even larger volume IHX's. These numbers indicate that an IHX using high-viscosity Flibe is not out of the question. However, the size and cost will clearly be substantial.

It can be seen from the numbers at the bottom of Table 2 that the vapor number densities corresponding to the temperature of the Flibe entering the reactor are reasonably close to the limits presently estimated to be permissible for heavy ion beam transmission (Bangerter, 1988) and well below the density required for laser beam transmission. However, for heavy ion beam propagation, it may be necessary to use the cooler Flibe out of the IHX's to lower the vapor pressure in the regions near the beam paths.

The maximum permissible outlet temperature of the cold, secondary flibe, T_{co} was found to be 686 K for the IHX's of Case 2H, as shown in Table 2, which limits the efficiency of the steam power plant to about 34%.

RESULTS FOR LOW-VISCOSITY FLIBE

Next, an IHX using the low-viscosity Flibe as the primary coolant and sodium fluoroborate (NaBF_4) as the secondary coolant was studied, subject to the same four constraints listed above. The sodium fluoroborate was proposed for this purpose in the earlier molten salt breeder reactor program because it has some advantages for retaining the tritium (Briggs, 1971-2). Sodium fluoroborate has a melting point of 384 C = 657 K which lies between the melting points of the high and low viscosity Flibe.

The results of the parametric study are shown in Figs. 4, 5 and 6 for a temperature drop on the primary tube side of 50 C. The tube-side temperature drop had to be reduced from the original 100 C used for the high-viscosity flibe IHX's to 50 C to insure that the Reynolds number of the turbulent flow would be above 10,000 for a reasonably

TABLE 2
Intermediate Heat Exchangers

Key Parameters		High-Viscosity Flibe (Case 2H)	Low-Viscosity Flibe/NaBF ₄ (Case 20cc2)
Primary Fluid Inlet T_{hi}	(K)	863	923
Primary Fluid Outlet T_{ho}	(K)	763	873
Secondary Fluid Inlet T_{ci}	(K)	676	793
Secondary Fluid Outlet T_{co}	(K)	686	823
Thermal Rating, P_{th}	(MW _{th})	1000	1000
Tube Length, L_t	(m)	10	10
Tube Bundle Height, H	(m)	5.33	3.0
Tube Bundle Depth, D_{TB}	(m)	4.93	3.7
Vol. of Tube Bundle	(m ³)	265	111.7
Tube outer diam., D_o	(m)	0.01	0.02
Tube Side:			
Hot, Primary Flow Rate	(kg/s)	4250	8400
Pressure Drop	(MPa)	0.25	0.070
Pumping Power	(MW _e)	2.0	0.36
Shell Side:			
Cold, Secondary Flow Rate	(kg/s)	42,530	22,100
Pressure Drop	(MPa)	0.48	0.020
Pumping Power	(MW _e)	37.7	0.29
Overall Heat Transfer Coeff, U_o	(W/m ² -K)	335	2324
Heat Transfer Area $A_o(HT)$	(m ²)	23,500	4950
Number of Tubes		75,000	7880
$A_o(HT)/\text{Vol. of Tube Bundle}$	(m ² /m ³)	88.6	44.3
$P_{th}/\text{Vol. Tube Bundle}$	(MW/m ³)	3.77	9.0
Steam Powerplant Efficiency		34.0%	41.2%
$n_{\text{vapor}} (T_{hi})$	(cm ⁻³)	$\sim 7 \times 10^{13}$	$\sim 5 \times 10^{13}$
Acceptable n_{vapor} for Ion Beams		$\sim 5 \times 10^{13}$ (?)	$\sim 5 \times 10^{13}$ (?)
Acceptable n_{vapor} for Lasers		$\sim 10^{15}$ (?)	$\sim 10^{15}$ (?)

Figure 4. The COE (in 1988 dollars) as a function of the two cold secondary-side temperatures. (The hot tube-side Reynolds numbers are greater than 10,000 to the left of the dashed curve.)

wide range of IHX geometries (to avoid the uncertainties in the heat transfer associated with the transition range of Reynolds numbers from about 2000 to 10,000). The IHX designs to the left of the dashed line on Figs. 4 and 5 and below the dashed line on Fig. 6 have tube-flow Reynolds numbers greater than 10,000.

Fig. 4 shows that low COE's occur at high T_{co} 's (the cold secondary-side outlet temperature), because this produces the highest SPP thermal efficiency. However, the reference IHX design selected (shown by the circle) is a compromise to limit the cold-side mass flow rate of the sodium fluoroborate to 22,100 kg/s for each IHX, as shown on Fig. 5. It should be noted that this compromise choice is somewhat arbitrary, since the true cost of such a large flow rate has not been evaluated yet and is not in the simplified cost model used in this study. More refined cost models for this effect are now being developed.

Fig. 6 shows that for low COE's, we must choose IHX designs which are rather large and have relatively high capital costs, because these IHX's have higher cold-side outlet temperatures. The selected compromise IHX design costs about \$55/kWh (in 1988 dollars) due in part to the high cost estimates for a modified Hastalloy N which may have to be used for corrosion resistance to the molten salts. (If a modified stainless steel can be used, the cost is estimated to be closer to \$20/kWh.)

These IHX's have an outlet temperature, T_{co} , of 823 K which results in a relatively high thermal efficiency of 41.2% and a relatively low COE. We can conclude that the high thermal efficiency dominates the effect of the relatively high capital cost of the IHX's in the COE equation.

The results for this best compromise IHX design using low-viscosity Fluibe/NaBF₄ are summarized in Table 2 for comparison with the IHX using the high-viscosity Fluibe.

Figure 5. The cold secondary-side mass flow rate required for various cold side temperatures. (The hot tube-side Reynolds numbers are greater than 10,000 to the left of the dashed curve.)

The low-viscosity Fluibe IHX is much more compact; it has only about 42% of the volume and 11 % of the number of tubes of the high-viscosity Fluibe IHX shown in Table 2.

It should also be noted that a tube diameter of 0.02 m has been used for the low-viscosity Fluibe IHX rather than 0.01 m. The larger tube diameter causes the volume of the IHX to increase by about a factor of three, but it reduces the pumping power on the primary side of each IHX from 12.2 MWe to only 0.36 MWe. Even with this substantial volume penalty, the selected IHX's using low-viscosity Fluibe/NaBF₄ are now comparable in size to a steam generator for a pressurized water reactor with the same thermal rating (Moir, et al, 1990).

Figure 6. The direct capital cost of the three IHX's (in 1988 dollars) as a function of the cold secondary-side temperatures. (The hot tube-side Reynolds numbers are greater than 10,000 below the dashed curve.)

Figure 7. The detailed power balance for the single-chamber Base Case D.

All the temperatures can be higher than for the high-viscosity Flibe cases, because the vapor pressure of the low-viscosity Flibe is substantially lower than for the high-viscosity Flibe. This results in a higher cold-side outlet temperature, T_{co} , which in turn leads to the higher thermal efficiency of 41.2% shown in Table 2.

It was of interest to see the effect of replacing the secondary sodium fluoroborate coolant with low-viscosity Flibe. All four IHX temperatures and the thermal power were kept the same as for the case with sodium fluoroborate. The IHX dimensions came out to be almost exactly the same for the two different secondary fluids. The pressure drop using the low-viscosity Flibe on the secondary side was somewhat higher than for the NaBF_4 , but still negligibly small.

We see from these results that the choice of sodium fluoroborate versus low-viscosity Flibe for the secondary fluid can be based on other considerations than the size and cost of the IHX.

The detailed power flow diagram for the single-chamber Base Case D using three of the low-viscosity

Flibe/ NaBF_4 IHX's shown in Table 2 is given in Fig. 7. (Each IHX would have to be scaled up about 10% to handle the actual thermal power of 1104 MWth.) We see from Fig. 7 that the net electric power output is 1083 MWe with an overall circulating power fraction of 21% (including the 15% circulating power fraction to the drivers). Fig. 8 shows the type of sub-critical, non-reheat steam cycle which would be used to achieve this performance.

SUMMARY AND CONCLUSIONS

The results of this scoping study have shown that the use of high-viscosity Flibe as both the primary and secondary fluid results in an IHX which is much larger and costlier than a comparable IHX using low-viscosity Flibe for the primary coolant and either sodium fluoroborate or low-viscosity Flibe for the secondary coolant. Therefore, while the high-viscosity Flibe is not out of the question, the IHX's will clearly be large and costly components. For the low-viscosity flibe, on the other hand, the size and cost of the IHX's seem reasonable. Therefore HYLIFE-II has been

designed using the low-viscosity Fibre for the jet array.

The preliminary COE values obtained in this scoping study are much higher than we would like even for the single-chamber Base Case D. These high COE's occur even with a high thermal efficiency for the steam power plant. More refined cost studies are now in progress. However, it seems clear that major reductions in the capital costs, particularly of the drivers (which are now estimated to be almost 50% of the direct capital cost), are required to reduce the COE of these inertial confinement fusion concepts to more attractive levels.

ACKNOWLEDGEMENTS

The work reported here was performed under the auspices of the U.S. Department of Energy by the Lawrence Livermore National Laboratory under Contract No. W-7405-ENG-48.

REFERENCES

Bai, R.Y. and Schrock, V.E. (1990), "An Approximate Method for Analyzing Transient Condensation on Spray in HYLIFE-II", Ninth Topical Meeting on the Technology of Fusion Energy, Oak Brook, IL, October 7-11.

Bangerter, R. (1988), "Implications of ICF Target Requirements", Memo, Lawrence Livermore National Laboratory, Livermore, CA.

Blink, J.A., et al. (1985), "The High-Yield Lithium Injection Fusion-Energy (HYLIFE) Reactor", UCRL-53559, Lawrence Livermore National Laboratory Report.

Briggs, R.B. (1971-2), "Tritium in Molten Salt Reactors", *Reactor Technology*, Vol. 14, p.335.

Laser Program Annual Report for 1978, (1979), Vol. 3, UCRL-50021-78, Lawrence Livermore National Laboratory.

McDowell, M.W., and Murray, K.A. (1984), "Pulse*Star ICF Reactor - Heat Transfer Loop and Balance of Plant Considerations", UCRL-15598, Lawrence Livermore National Laboratory.

Meier, W.R. (1988), "Parametric Economic Studies for HYLIFE: Implications for a New Molten Salt Reactor", Lawrence Livermore National Laboratory Memo.

Moir, R.W., et al (1990), "HYLIFE-II Project Report", Lawrence Livermore National Laboratory (in preparation).

Moir, R.W. (1990), "HYLIFE-II ICF Reactor Design", Ninth Topical Meeting on the Technology of Fusion Energy, Oak Brook, IL, October 7-11.

Rosenthal, M.W., Haubenreich, P.N. and Briggs, R.B. (1972), "Molten Salt Reactor Program - The Development Status of Molten Salt Breeder Reactors", ORNL-4812, Oak Ridge National Laboratory.

Tobin, M.T. (1989), "Neutronics Analysis for HYLIFE II", Ninth Topical Meeting on the Technology of Fusion Energy, Oak Brook, IL, October 7-11.

Figure 8. The subcritical steam cycle proposed for HYLIFE-I.

ATTACHMENT 2

STEAM GENERATOR DESIGN AND PERFORMANCE ESTIMATES
FOR THE HYLIFE-II FUSION POWER PLANT

Progress Report
Ying Lee
September, 1990

STEAM GENERATOR REPORT

August 28, 1990

Written BY

Ying T. Lee

Revised By

**Dr. Myron A. Hoffman
University of California, Davis**

**Under Purchase Order No. B108485
from the Lawrence Livermore National Laboratory**

Table of Contents

Table of Contents	i
List of the TABLES	iv
List of the FIGURES	v
Nomenclature	vi
Chapter 1 <u>Introduction to the Steam Generator</u>	
1.1 <u>General Description of a Steam Generator</u>	1
1.2 <u>Steam Generator Design</u>	2
1.3 <u>Steam Generator Code (SG Code)</u>	3
Chapter 2 <u>SG Code and Tajbakhsh's Code</u>	
2.1 <u>Tajbakhsh's Helium/Water Steam Generator</u>	5
2.2 <u>Development of Steam Generator Code (SG Code)</u>	5
2.3 <u>Improvements of the SG Code</u>	6
2.4 <u>Other Changes on the SG Code</u>	7
2.5 <u>Some Limitations of the SG Code</u>	7
2.6 <u>Validation of the SG Code with Tajbakhsh's code</u>	8
Chapter 3 <u>SG Code and CRBR Steam Generator</u>	
3.1 <u>Sodium Heat Transfer Coefficient</u>	12
3.2 <u>Wall Thermal Conductivity</u>	13
3.3 <u>Heat Transfer Mechanism of Water</u>	13
3.4 <u>Pressure Drop of the Fluids</u>	14
3.5 <u>Minor Losses in the Steam Generator</u>	15
3.6 <u>Validation of the SG Code with CRBR data</u>	15

Chapter 4 HYLIFE-II Steam Generator

4.2	<u>Evaporator</u>	23
4.3	<u>Steam Separator</u>	23
4.4	<u>Superheater</u>	24
4.5	<u>Heat Transfer for the Sodium Fluoroborate</u>	25
4.6	<u>Wall Thermal Conductivity</u>	25
4.7	<u>Definition of the Regimes for Water Boiling</u>	26
4.8	<u>Heat Transfer to the Water</u>	26
4.8.1	Single Phase Liquid Boiling (SPL)	
4.8.3	Saturated Nucleate Boiling (SATNB)	
4.8.4	Film Boiling (FB)	
4.8.5	Single Phase Vapor (SPV)	
4.9	<u>Pressure Drop in the Steam Generator</u>	30
4.9.1	Primary Fluid Side	
4.9.2	Secondary Fluid Side	

Note ** Pages 44 to 55 omitted in this report **

Appendix A

A.5	<u>Check Case for the Sodium/Water Evaporator</u>	56
A.5.1	<u>Choose the Initial Parameters</u>	
A.5.2	<u>Tube Bundle and Tube Geometry</u>	
A.5.3	<u>Mass Flow Rate</u>	
A.5.4	<u>Curve Fit for the Sodium Properties</u>	
A.5.5	<u>Regime 1 & 2 (Single Phase Liquid and Subcooling Boiling)</u>	
A.5.6	<u>Pressure Drop in Regime 1 & 2</u>	
A.5.7	<u>Regime 3 (Saturated Nucleate Boiling)</u>	
A.5.8	<u>Pressure Drop on the Regime 3</u>	
A.5.9	<u>Regime 4 (Film Boiling)</u>	
A.5.10	<u>Pressure Drop on the Regime 4</u>	
A.5.12	<u>Summary for the Evaporator</u>	
A.6.1	<u>Choose the Initial Parameters</u>	
A.6.2	<u>Tube Bundle and Tube Geometry</u>	
A.6.3	<u>Mass Flow Rate</u>	
A.6.4	<u>Curve Fit for the Sodium Properties</u>	
A.6.5	<u>Regime 5 (Single Phase Vapor Boiling)</u>	
A.6.6	<u>Pressure Drop on the Regime 5</u>	
A.6.7	<u>Safety Factor for Superheater</u>	
A.6.8	<u>Summary of the Superheater</u>	
References		83

List of the TABLES

TABLE 1-1	Summary design data for a few steam generators .	4
TABLE 2-1	Comparison of Tajbakhsh's code and SG Code . . .	9
TABLE 2-2	Computer results for Tajbakhsh and SG Codes . . .	10
TABLE 3-1	Physical property data for sodium liquid metal .	17
TABLE 3-2	Results from CRBR, SG Code, and a Hand Calculation Using Average Properties	18
TABLE 3-3	Uncertainty results	19
TABLE 4-1	Physical properties of MSBR coolant salt	33
TABLE 4-2	Frictional multiplier as a function of quality and absolute pressure	34
TABLE 4-3	Void fraction α as a function of quality and absolute pressure	35

List of the FIGURES

FIGURE 2-1	THTR 300 MW Helium/Water steam generator	11
FIGURE 3-1	Hockey stick steam generator design for CRBR study	20
FIGURE 3-2	Graphical display of the uncertainty analysis results (TABLE 3-3)	21
FIGURE 4-1	Flow diagram of one of the three hockey stick steam generator systems for HYLIFE-II	36
FIGURE 4-2	Flow chart of one steam generator system	37
FIGURE 4-3	Definition of regimes inside an evaporator . . .	38
FIGURE 4-4	Definition of regime inside a superheater . . .	39
FIGURE 4-5	Critical heat flux in steam boiling	40
FIGURE 4-6	Value of ϕ_{lo}^2 as a function of pressure and mass quality	41
FIGURE 4-7	Void fraction α as a function of pressure and mass quality	41
FIGURE 4-8	Value of r_4 as a function of pressure and mass quality	42
FIGURE 4-9	Scaled-up of the CRBR steam generator	43

Nomenclature

Symbols

A	area	m^2
C	cost	M\$
C_p	heat capacity	$kJ/kg.K$
D	diameter	m
D_h	hydraulic diameter	m
f_o	fanning friction factor	
G	mass flow rate per unit area	$kg/m^2.s$
H	height	m
HR	heat rate	
h	heat transfer coefficient	$W/m^2.K$
i	enthalpy	kJ/kg
K	thermal conductivity	$W/m.K$
K_{minor}	minor loss	
L	tube length	m
$\dot{m}$	mass flow rate	kg/m^2
N_t	number of tube	
Nu	Nusselt number	
$P_{el(gross)}$	gross electrical power	MW
Pr	Prandtl number	
P_{th}	thermal power	MW

p	pressure	MPa
ΔP_f	frictional pressure drop	MPa
ΔP_a	acceleration pressure drop	MPa
ΔP_g	gravity pressure drop	MPa
q"	heat flux	MW/m ²
R	thermal resistance	K/W
Re	Reynolds number	
T	temperature	K
v	specific volume	m ³ /kg
X	steam quality	
Greek		
α	void fraction	
η_{th}	thermal efficiency of the cycle	
μ	viscosity coefficient	kg/m.s
π	pi value	
ϕ_{fo}	Martinelli-Nelson friction factor	

Subscripts

(number)	referring to number of regimes
DNB	departure from nucleate boiling
EV	evaporator
f	liquid
film	film
g, s	vapor
i	inlet condition
o	outlet condition
SH	superheater
sat	saturated
TB	tube bundle
ti	inner tube
to	outer tube
w	wall

Chapter 1 Introduction to the Steam Generator

Steam boilers have been used to produce steam for over 200 years, and constitute one of the earliest subjects for the application of engineering principles. A steam generator, a special design of steam boilers which uses the hot fluid flow rather than the products of combustion as the heat source, is an important unit in a fusion power plant. Since some of the materials in the fusion reactor become radioactive, in order to ensure the environmental and human safety, the reactor must be isolated inside a chamber. The way to transfer the heat generated in the reactor to the surroundings is by means of a heat exchanger system. A steam generator, a special kind of heat exchangers, functions not only as a bridge in the heat transfer process, but also as one of the barriers to the leakage of tritium, a highly dangerous radioactive material produced inside the reactor. The heat extracted in the steam generator is used to boil the water to create steam which in turn runs a traditional turbine to produce electricity.

1.1 General Description of a Steam Generator

A steam generator is one of the members of the heat exchanger family. Like any common heat exchangers, it consists of thousands of tubes running inside a cylindrical shell. The primary (hot) fluid flows outside the tubes while the secondary (cold) fluid flows inside the tubes or vice versa. The heat is transferred from the primary fluid to the secondary fluid through the thin tube wall. Although a heat exchanger can be parallel flow, counterflow, single cross flow or multi-pass cross flow, a steam generator is

usually a simple counterflow unit. Another difference between a steam generator and a common heat exchanger is that water is the secondary fluid in a steam generator and will have a phase change during its journey through the tubes. First, water enters a steam generator as a single phase compressed liquid. When it reaches the boiling point, a two phase water vapor mixture is formed inside the tubes. After all the liquid vaporizes, superheated steam is produced. Because of these different phases involved, the heat transfer mechanisms inside a steam generator are so complicated that it becomes a separate field of engineering.

1.2 Steam Generator Design

There are several steam generator design proposals suggested by previous studies on fast breeder reactor power plants as listed on TABLE 1-1. Based on the simple construction and the completeness of the information in the open literature, the hockey stick steam generator, which was studied and was designed by Atomic International (AI) and General Electric (GE) for the Clinch River Breeder Reactor (CRBR), has been selected for the HYLIFE-II fusion power plant.

The original design of the hockey stick steam generator for the CRBR consisted of three sets of steam generator systems with a total thermal load of 975 MW. Each set consisted of three identical hockey stick steam generators, two evaporators and one superheater, with a total thermal rating of 325 MW. The evaporator and superheater were identical in design to minimize cost and spare parts. They were shell and tube counterflow heat exchangers that

were vertical except for a 90 degrees bend at the top. The height of each hockey stick steam generator was 19.8 m with an outer shell diameter of 1.32 m. There were 757 tubes running inside the shell. The tubes were arranged in triangle pattern and were made of 2 1/4-Cr-1 Mo steel. Sodium was the primary fluid which flowed downward through the shell and the water/steam flowed upward through the tubes. This arrangement, as mentioned by GE study, provided excellent natural convection flow for decay heat removal purpose.

A scaled-up version of the CRBR hockey stick steam generator design will be adopted for the HYLIFE-II study. More information will be discussed in chapter 4.

1.3 Steam Generator Code (SG Code)

The SG Code is a computer program developed for the study of the HYLIFE-II's steam generator. However, when the program is completed, it can also be used for other similar steam generator designs in the future. Before the SG Code was used in the study of the steam generator for HYLIFE-II, data for two previous steam generator designs were used to validate the program. Detailed discussion can be found in next two chapters.

TABLE 1-1 Summary design data for a few steam generators

Reactor Type	PWR ^a	LMFBR ^a	LMFBR	Fusion Reactor
Plant	Oconee Unit 1	Cregs-Malville	CRBR Design	HYLIFE-II
Designer	B & W	French	AI	UCD & LLNL
Type	Once-through	Once-through	Separated Evaporator & Superheater units	Same as CRBR design
Tube shape	Straight	Helical	Straight plus bend (hockey-stick)	Same as CRBR design
Hot fluid	Pressurized water	Sodium	Sodium	Sodium Fluoroborate
Fluid in tube	Water/steam	Water/steam	Water/steam	Water/steam
Type of tube	Plain	Plain		
Tube material	Incoloy 800	Incoloy 800	2 ½ Cr-1-Mo	Hastelloy N or Modified 316 SS
Capacity (MWt)	1295	750	325	1100
Tube ID (mm)	14	20	10.36	10.36
Tube length (m)	15	91	19.2 in each unit	Scaled-up for minimum COE
Number of tubes	15530	1430	757 in each unit	

^a Data obtained from (Fraas, 1989)

Chapter 2 SG Code and Tajbakhsh's Code

Tajbakhsh's helium/water steam generator code (Tajbakhsh, 1984) results were used only for validation of our Steam Generator Code (SG Code), which is being developed mainly for the study of the molten salt steam generator of the HYLIFE-II study. Because of the different requirements of our SG Code, the solution method used in the two computer codes is quite different. A brief summary of the differences is listed in Table 2-1.

2.1 Tajbakhsh's Helium/Water Steam Generator

The basic design was a heat exchanger with helium flowing inside the shell, water flowing inside the tubes, and a return tube for helium in the center of the heat exchanger as shown in Figure 2-1. In the program, users can select between three options : variable heat flux and properties, constant heat flux and properties, or constant heat flux but variable properties of both the fluids.

2.2 Development of Steam Generator Code (SG Code)

The SG Code was developed under the same scheme as Tajbakhsh's program, and then was modified to match the sodium steam generator in the Clinch River Breeder Reactor (CRBR) study. Finally it was extended to suit the needs of the HYLIFE-II study. Instead of using only helium as the primary fluid, the SG Code can have sodium, sodium fluoroborate (NaBF_4) or low viscosity Flibe for the primary fluid. Since the SG Code started with the Tajbakhsh's program, the return tube design capacity was also included. However, it can be turn off if desired. Moreover, normally both the heat flux and properties of the fluids inside a steam generator are not constant,

so the SG Code adopts only the option for the variable heat flux and variable fluid properties inside its code.

2.3 Improvements of the SG Code

A large and more completed steam table (Steam, 1984) is used in the SG Code. The allowable pressure range of the water is 4.5 to 22 MPa, and the temperature range is 0°C to 1000°C. The pressure of the water in Tajbakhsh's study was about 7 MPa, the pressure of water in CRBR study was about 13 MPa, and the pressure of water in HYLLIFE-II study is expected to be 15 MPa. As a result, a larger and more completed steam table is appropriate in the SG Code for its ability to be used in the future design of other steam generators.

In the saturated nucleate boiling regime, Tajbakhsh used the Chisholm slip ratio correlation

$$S \equiv \frac{u_g}{u_f} = \left[X \frac{v_g}{v_f} + (1 - X) \right]^{\frac{1}{2}}$$

to estimate the void fraction which was required to calculate the pressure drop in this two-phase regime. In order to improve the accuracy of the calculation, the Martinelli-Nelson correlation of void fraction is used in the SG Code. Further discussion will be found in chapter 4.

Also, in the saturated nucleate boiling regime, the SG Code provides a choice of either Shah correlation (Shah, 1976), or the Jens and Lottes correlation to calculate the heat transfer coefficient of the two-phase water. Only Jens and Lottes correlation was available in Tajbakhsh's code.

2.4 Other Changes on the SG Code

During the calculation of the heat transfer coefficient in the saturated nucleate boiling, iteration is necessary. In Tajbakhsh's code, helium properties were used to carry out the iteration. However, in the SG Code, water properties are used to do the iteration. It is obvious that water is a better choice than helium in the SG Code because water is always used as the secondary fluid in a steam generator while there are several choices for the primary fluid. Moreover, there is limit knowledge of the sodium fluoroborate properties, so it is not wise to use its properties to do the iteration.

Also, during the calculation, Tajbakhsh's code used the average specific heat of single phase water to calculate the temperature change of the water in each increment. However, this is only an approximate method. The SG Code uses the enthalpy of water for the calculation of the temperature change which is a more accurate method.

2.5 Some Limitations of the SG Code

In order to simplify the computer program, the SG Code can only do calculations for variable heat flux and variable fluid properties as mentioned early. However, it is believed that this will not create a serious problem since real steam generator designs will have neither constant heat flux nor constant fluid properties.

2.6 Validation of the SG Code with Tajbakhsh's code

Despite the differences illustrated in previous sections, the results from both the computer programs agreed very well with each other. The numerical results are listed in TABLE 2-2. These results constitute the first validation of the SG Code.

TABLE 2-1 Comparison of Tajbakhsh's code and SG Code

Tajbakhsh's Code	SG Code
Characteristics 1. Primary fluid — Helium 2. Secondary fluid — Water 3. Steam generator with return tube	Helium, Sodium, Sodium Fluoroborate Water, Flibe Steam generator with or without return tube
Improvements 1. Small steam table 2. Void fraction is calculated by Chisholm slip ratio correlation 3. Uses Jens and Lottes correlation in the saturated nucleate boiling	Large steam table pressure from 4.5 to 22 MPa temperature from 0 to 1000°C Void fraction is based on the Martinelli and Nelson's graphical correlation Jens and Lottes or Shah correlations can be used
Changes 1. Uses the helium properties to do the iteration in the saturated nucleated boiling regime 2. Temperature of water is calculated from specific heat of water	Uses the water properties to do the iteration Temperature of water is calculated from the enthalpy of water
Limitation 1. For variable heat flux & properties, constant heat flux & properties, or constant heat flux & variable properties	Variable heat flux and variable properties only

TABLE 2-2 Computer results for Tajbakhsh and SG Codes

Input Parameter	Tajbakhsh	SG Code
Geometry		
Tube OD (mm)	0.01316	0.01316
Tube Wall (mm)	0.00150	0.00150
Pitch to Diameter Ratio	(1.565) ^a	1.565
Shell ID (m)	3.2	3.2
He Side		
Flow/Unit (kg/s)	512	512
Outlet Temperature (°C)	515	515
Water Side		
Flow/Unit (kg/s)	395.3	395.3
Inlet Temperature (°C)	447	447
Outlet Temperature (°C)	773	773
Inlet Pressure (MPa)	6.89	6.89
Output Parameter		
Geometry		
Duty/Unit (MW)	1009	1011
Number of Tube	18670	18670
Minimum Tube Length (m)	12.7	12.6
Heat Transfer Area OD (m ²)	9803	9726
He Side		
Inlet Temperature (°C)	895	895
Pressure Drop (MPa) ^b	0.062	0.061
Water Side		
Outlet Pressure (MPa)	6.819	6.820
Pressure Drop (MPa) ^c	0.071	0.068

^a P/D is the output parameter, and number of tube is the input parameter in Tajbakhsh's code

^b Pressure drop includes only frictional, acceleration, and gravity losses

^c Pressure drop includes only frictional and acceleration losses

FIGURE 2-1 THTR 300 MW Helium/Water steam generator (Fricker, 1976)

Chapter 3 SG Code and CRBR Steam Generator

As mentioned in Chapter 1, the CRBR steam generator design had three hockey stick steam generator systems, each of which consisted of 2 evaporators and 1 superheater. Detailed information about the design and testing of a small-scale version of the hockey stick steam generator is available in several reports published by Atomic International (AI) and General Electric (GE) : (Moody and Gabler, 1978), (Lillie and Bashford, 1978), and (Harty, 1974).

A detailed hand calculation using the average properties of both the primary and secondary fluids is included in Appendix A. It provides information about the equations used in the SG Code. The model used in the SG Code is summarized in the next sections.

3.1 Sodium Heat Transfer Coefficient

From the study of the CRBR (Harty, 1974), an experimental correlation to estimate the Nusselt number of the sodium is given by

$$Nu = 0.25 + 6.20P/D + (-0.007 + 0.032P/D) Pe^{0.8-0.024P/D}$$

where P/D is pitch to tube diameter ratio,
 Pe is the Peclet number

Heat transfer through sodium, a liquid metal, differs from ordinary fluids. It has very high thermal conductivity, which gives rise to a much smaller Prandtl number. Because of this characteristic, the Dittus-Boelter correlation cannot be applied to sodium.

The other properties of sodium, eg. density, viscosity etc. are given in table form as shown in TABLE 3-1 (Singh, 1985). In order to input these data in the computer code, a third order curve

fit was used to fit the data. The results are

$$C_p = 1855 - 1.849T + 1.787 \times 10^{-3} T^2 - 4.902 \times 10^{-7} T^3$$

$$Pr = 4.658 \times 10^{-2} - 1.638 \times 10^{-4} T + 2.179 \times 10^{-7} T^2 \\ - 9.853 \times 10^{-11} T^3$$

$$\mu = 2.858 \times 10^{-3} - 9.767 \times 10^{-6} T + 1.257 \times 10^{-8} T^2 \\ - 5.588 \times 10^{-12} T^3$$

$$K = 100.1 - 2.836 \times 10^{-3} T - 1.095 \times 10^{-4} T^2 + 6.863 \times 10^{-8} T^3$$

$$1/v = 1246 - 1.507T + 2.251 \times 10^{-4} T^2 - 1.289 \times 10^{-6} T^3$$

3.2 Wall Thermal Conductivity

From the CRBR study, an equation for the thermal conductivity of the wall material (2 1/4 Cr-1 Mo) was found with the temperature dependence in it. As a result, the SG Code allows the input of thermal conductivity of the wall material to be either constant or a linear function of temperature.

$$K_w = A + B T$$

where A and B are constants for different wall materials,
T is the wall temperature in K

3.3 Heat Transfer Mechanism of Water

In the CRBR study, the water side was divided into the single-phase liquid, the subcooling boiling, the saturated nucleate boiling, and the film boiling regimes in the evaporator and superheated vapor phase regime in the superheater. The heat transfer correlations in all regimes are the same as those used in the comparison with the Tajbakhsh code except for the film boiling regime.

Since the SG Code was developed based on the Tajbakhsh's

helium-steam generator code, it assumed that the heat flux on the water side was low enough so that the water reached the dry-out (DO) condition. However, in the design of the CRBR hockey stick steam generator, the heat flux was high enough so that the water experienced the departure from nucleate boiling (DNB) instead of DO. As a result, a correlation to determine the DNB point has been adopted from the CRBR study in the SG Code. However, the final method of estimating the heat transfer coefficient of the water after DNB was not described in detail in the CRBR Final Report (Harty, 1974). As a result, a correlation given by Groeneveld for the post-dryout (PDO) regime has been tried in the SG Code. More detailed discussion can be found in next chapter.

3.4 Pressure Drop of the Fluids

In the CRBR reports, the pressure drops of both the fluids included friction, gravity, acceleration, entrance, exit, bending, orifice, and other minors loss contributions. However, it was not possible in the SG Code to estimate the entrance, exit, bending, orifice, and other minors losses simply because there was insufficient information about these minor losses in the CRBR reports. As a result, only friction, gravity, and acceleration losses are calculated on the water side and only frictional and acceleration losses on the sodium side. The treatment of the minor losses is discussed next.

3.5 Minor Losses in the Steam Generator

In the CRBR study, a detailed structural design of the hockey stick steam generator had been completed. As a result, the pressure

losses from the orifices, bending, entrance, and the exit were measured and were incorporated in their results. However, without such detailed information about these losses, the SG Code cannot calculate the same results as in the CRBR as shown in TABLE 3-2. Therefore, in order to match their pressure losses, an estimated total minor loss, K_{minor} , is used in the SG Code to make up the difference. More details about the calculation can be found in Appendix A.

3.6 Validation of the SG Code with CRBR data

As shown in TABLE 3-2, the result from the SG Code and the hand calculation using the average fluid properties agreed with the CRBR result very well. As a result, the validation of the SG Code with the CRBR data has been accomplished.

A few things should be pointed out before the end of this chapter. The tube bundle diameters of the CRBR study and the SG Code are the same as shown in TABLE 3-2. In the CRBR study, the tube bundle, with a protective shroud shell around it, was placed in the middle of the shell. With the seal rings placed between the top and the bottom of the shroud and the inner surface of the outer shell, it created a sealed annular gap between the shroud and the shell. It has been assumed that no sodium flowed in the gap between the tube bundle and the shell, and the SG Code assumed all the sodium flowed inside the tube bundle; ie. the small bypass sodium flow between the shroud and the tube bundle was neglected. This introduces an unknown, but negligible difference between the

Code results and the CRBR results.

Another area of concern is the estimation of the pressure drop in the steam generator. As discussed in the previous section, a K_{minor} is used to match the pressure drop between the CRBR result and SG Code result. This may create some difficulties in the scaled-up study of HYLIFE-II steam generator, since the minor losses dominate the pressure drops on both the primary and secondary sides (see Appendix A).

One final thing is the uncertainty analysis of the steam generator. As shown in TABLE 3-3 and FIGURE 3-2, the CRBR Project had included an uncertainty analysis in their study (Moody and Gabler, 1978). Then a safety factor was included in the final dimensions of the hockey stick steam generator. The combined uncertainty and the safety factors contributed about 42% and 30% of the total available heat transfer area of the evaporator and superheater respectively as shown in FIGURE 3-2 graphically. An uncertainty analysis has not been carried out in the HYLIFE-II study. However, in order to compensate for uncertainties, an overall safety factor will be used (see Appendix A for detailed calculation) and the result is shown on TABLE 3-3 and FIGURE 3-2.

TABLE 3-1 Physical property data for sodium liquid metal (Singh, 1985)

T (°C)	Density (kg/m ³)	Specific heat (J/kg.K)	Thermal conductivity (W/m.°C)	Dynamic viscosity (mNs/m ²)	Prandtl number
125	921	1370	86.0	0.61	0.0097
225	897	1320	80.0	0.42	0.0069
325	872	1280	74.0	0.32	0.0055
525	823	1270	63.0	0.23	0.0046
625	774	1280	59.0	0.18	0.0039

TABLE 3-2 Results from CRBR, SG Code, and a Hand Calculation Using Average Properties

Input Parameter	Evaporator			Superheater		
	CRBR	SG Code	Hand	CRBR	SG Code	Hand
Geometry						
Tube OD (mm)	15.9	15.9	15.9	15.9	15.9	15.9
Tube Wall (mm)	2.77	2.77	2.77	2.77	2.77	2.77
Tube Pitch (mm)	31			31		
Pitch to Diameter Ratio	1.9497	1.95	1.95	1.9497	1.95	1.95
Tube Bundle Diameter (m)	(0.896) ^a	0.896	0.896	(0.896) ^a	0.896	0.896
Na Side						
Flow/Unit (kg/h x E6)	3.05	3.05	3.05	6.1	6.1	6.1
Outlet Temperature (°C)	349	349	349	463	463	463
Water Side						
Flow/Unit (kg/h x E6)	0.5	0.5	0.5	0.5	0.5	0.5
Inlet Temperature (°C)	285	285	285	331	331	331
Outlet Temperature (°C)	331	334	331	485	488	485
Inlet Pressure (MPa)	13.2	13.2	13.2	12.1	12.1	12.1
Output Parameter						
Geometry						
Number of Tube	757	757	757	757	757	757
Minimum Tube Length (m)	8.1	8.2	8.1	9.9	9.6	10.3
Heat Transfer Area OD (m ²)	304.5	310.1	306.3	372.8	363.0	389.5
Na Side						
Duty Unit (MW)	120.8	118	116.3	83.4	84	81.4
Inlet Temperature (°C)	463	459	457	502	502	501
Pressure Drop (MPa) ^b	(0.108)	0.006	0.005	(0.42)	0.02	0.02
Water Side						
Outlet Pressure (MPa) ^c	12.4	13.0	13.06	10.2	11.4	11.46
Pressure Drop (MPa) ^c	(0.8)	0.2	0.14	(1.9)	0.7	0.64

^a Our estimate

^b Total pressure drop in the CRBR study. Only frictional, acceleration, and gravity losses are included in SG Code and hand calculation.

^c Total pressure drop in the CRBR study. Only frictional and acceleration losses are included in SG Code and hand calculation.

TABLE 3-3 Uncertainty results

Parameter	Heat Transfer Area OD (m ²)					
	Evaporator			Superheater		
	CRBR	SG Code	Hand	CRBR	SG Code	Hand
Minimum Heat Transfer Area	304.5	310.1	306.3	372.8	363.0	389.5
Plugging, Fouling & Correction	98.9			72.5		
Heat Transfer Uncertainty	63.3			48.3		
Excess Margin (m ²)	62.8	(219.4) a	(223.2) a	35.9	(166.5) a	(140.0) a
Available Heat Transfer Area	529.5	529.5	529.5	529.5	529.5	529.5

a Additional heat transfer areas are added in the SG Code and hand calculation to match the CRBR steam generator.

FIGURE 3-1 Hockey stick steam generator design for CRBR study

FIGURE 3-2 Graphical display of the uncertainty analysis results (TABLE 3-3)

Chapter 4 HYLIFE-II Steam Generator

After the careful study of the CRBR study, the hockey stick steam generator system was adopted for the HYLIFE-II study. The detailed setup is shown in FIGURES 4-1 and 4-2. Before the hockey stick steam generator could be adopted for the HYLIFE-II study, a few modifications of the CRBR hockey stick steam generator were necessary.

4.1 Modifications of the Steam Generator

The thermal capacity of the HYLIFE-II is 3312 MW, three times as much of the CRBR, so there will be either three times the number of steam generator systems used in the CRBR or three times the size for the same number (three) of steam generator systems in the CRBR. After careful study of the alternatives based on the size and cost of the steam generators, three sets of steam generator systems with the size upgraded from the CRBR will be used in the HYLIFE-II.

Another modification is the material used to construct the steam generator. Sodium fluoroborate, the primary fluid in the steam generator, is a molten salt which is corrosive to 2 1/4 Cr-1 Mo steel. As a result, an alloy which can resist the corrosion of the salt will be necessary. From the previous study of the Molten Salt Breeder Reactor (MSBR) report (Rosenthal, Haubenreich, and Briggs, 1972), Hastelloy N was found to be the best candidate for handling the sodium fluoroborate. However, as the MSBR pointed out, a more detailed study of the Hastelloy N was necessary, especially when water moisture was present in the salt. Based on the available information, Hastelloy N has been chosen to

construct the steam generator in the HYLIFE-II. However, if further research indicates that a modified 316 stainless steel is adequate, then this less expensive material will be adopted.

4.2 Evaporator

There are two evaporators in each set of steam generator systems for the HYLIFE-II. Similar to the design of the CRBR, the compressed water enters from the bottom of the steam generator and moves upward through tubes. The water exits at the top of the steam generator with a quality of 0.5. The vapor mixture then enters the steam separator which will be discussed in more detail in next section. The primary fluid, sodium fluoroborate coming from the superheater, enters at the top of the evaporator and flows downward around the tube bundles. FIGURE 4-1 shows the systematic flow chart of one steam generator system.

4.3 Steam Separator

When the water-vapor mixture exits from the two evaporators, it enters to a steam separator. The vapor is then separated and is pumped into the superheater which will be discussed in more detail in the next section. The saturated liquid, on the other hand, is mixed with the makeup water and is pumped back to the evaporators. The detailed flow path of the water is shown on FIGURE 4-1.

4.4 Superheater

One of the advantages of the CRBR hockey stick steam generator design is that the evaporator and superheater are identical. The main difference between an evaporator unit and a superheater unit is that the evaporator tubes have orifices at the inlet to stabilize the flow. Another minor difference is that one of the two bottom sodium flow outlets for the sodium flow in the evaporator unit is closed. After the steam exits from the steam separator, it enters at the bottom of the superheater and flows upward inside the tubes. The hot sodium fluoroborate from the intermediate heat exchangers (IHX's) flows downward outside the tubes. The superheated steam leaves at the top of the superheater and enters the turbine system to produce electricity. The sodium fluoroborate, on the other hand, exits the superheater and separates into two streams. Each of the streams then enters one evaporator to preheat and to boil the water as shown in FIGURE 4-1 and the returns to the pumps and IHX's. FIGURE 4-2 also shows the thermodynamic data for a steam generator system.

4.5 Heat Transfer for the Sodium Fluoroborate

From the study of the MSBR (Rosenthal, Haubenreich, and Briggs, 1972), the sodium fluoroborate salt behaves as a normal fluid, i.e. the Dittus-Boelter type correlation for the Nusselt number can be used. The correlation is

$$Nu = \frac{h D}{k} = a Re^b Pr^c$$

where Nu is the Nusselt number
Re is Reynolds number
Pr is Prandtl number
a, b, c are constants

where a, b, c were chosen as 0.023, 0.8, and 0.4 respectively. Pertinent physical properties for the salt were estimated in the MSBR study and are listed in TABLE 4-1.

4.6 Wall Thermal Conductivity

There are limit data on the variation of the thermal conductivity of Hastelloy N with temperature. However, since the thermal resistance of the tube wall is very small compared with the sodium fluoroborate and water, the average thermal conductivity of the alloy can be used without creating serious errors in the calculations. The average value of the thermal conductivity of the Hastelloy X for temperature around 600 °C is

$$K_w = 20 \text{ W/m.K}$$

which is the same value used in the IHX's calculation. (Hastelloy N has a somewhat higher K_w , about 28 W/m.K around 600 °C.)

4.7 Definition of the Regimes for Water Boiling

The regimes for the water boiling in the SG Code are the same as in the CRBR study. In the evaporator, there are single phase liquid heating (SPL), subcooling boiling (SCB), saturated nucleate boiling (SATNB), and film boiling (FB). In the superheater, there is only superheated vapor (SPV). Location of each regime can be found in FIGURES 4-3 and 4-4.

4.8 Heat Transfer to the Water

4.8.1 Single Phase Liquid (SPL)

In this regime, it can be considered to be a simple liquid to liquid heat exchange process. The temperature of the water is lower than the saturation temperature so there is no bubble formation occurring in the water.

An adequate approximation for the heat transfer coefficient for the liquid phase boiling of water is the Dittus-Boelter correlation which is

$$Nu = \frac{h D}{k} = a Re^b Pr^c$$

For the HYLIFE-II study, a, b, c were chosen as 0.023, 0.8, 0.4 respectively for the liquid water

4.8.2 Subcooling Boiling (SCB)

Even with the bulk temperature of the water still below the saturation temperature, the wall temperature can be above the saturation temperature of the water. As a result, bubbles can form on the surface of the wall in the subcooling boiling regime.

The Jens and Lottes correlation is used to determine the approximate beginning of the subcooled nucleate boiling regime (Point P on FIGURE 4-3). Point P is at the intersection of the extrapolated SPL curve and the Jens and Lottes T_{wall} line, as shown on FIGURE 4-3. The correlation is given as :

$$\Delta T_{sat(J\&L)} = T_{wall} - T_{sat} = 25 dq_i^{0.25} e^{-P/62}$$

where dq_i is heat flux in MW/m²
P is the pressure in bar

4.8.3 Saturated Nucleate Boiling (SATNB)

When the bulk temperature of the water reaches the saturation temperature, the saturated nucleate boiling regime begins. In this regime, large vapor bubbles are formed in the water channel. It is very complicated to estimate the amount of vapor present; ie. the vapor void fraction. We have chosen to use the Martinelli-Nelson correlation, as mentioned previously.

The Jens and Lottes correlation or Shah correlation can be used in this regime to get an estimate of the wall temperature. There are two possible cases for the end of the saturated nucleate boiling regime depending on the heat flux on tube wall as shown on FIGURE 4-5 (Collier, 1981). If the heat flux is relatively low, the water will reach the dryout point (DO). If the heat flux is above certain value, the water will reach the departure from nucleate boiling (DNB) instead. In the CRBR study, the heat flux was so high

that the water reached the DNB. The correlation to determine the DNB point suggested by CRBR study (Harty, 1974) is

For $dq'' > 6.3 \times 10^5 \text{ W/m}^2$

$$X_{\text{DNB}} = \frac{1614683}{i_{fg} G^{1/2}} \times \frac{v_g}{v_f}$$

For $dq'' < 6.3 \times 10^5 \text{ W/m}^2$

$$X_{\text{DNB}} = \frac{8.092 \times 10^{14}}{i_{fg} G^{1/2} q''^{1.5}} \times \frac{v_g}{v_f}$$

where i_{fg} is the latent heat of water in J/kg
 G is the mass flow rate in $\text{kg/m}^2 \cdot \text{s}$
 v_f is the specific volume of the liquid
 v_g is the specific volume of the vapor

However, these equations are only valid over the rather restricted range of the CRBR study (see FIGURE IV-9 of Harty, 1974). Further study is necessary to determine if this correlation is valid for the condition in the HYLIFE-II steam generator.

4.8.4 Film Boiling (FB)

When the heat flux of the wall exceeds the critical DNB value, the film boiling regime is reached. In this regime, the water near the wall surface is vaporized and a film of vapor is formed surrounding the core flow which may be a mixture of water-vapor and droplets, or it may be liquid water with bubbles and voids in it, as shown in FIGURE 4-3. The heat transfer coefficient of the water drops dramatically in this regime; as a result, there is a rapid increase of the wall temperature, as indicated on FIGURE 4-3.

Since in the CRBR study, it was found that the water undergoes DNB, a Nusselt number for the post-DNB film boiling is necessary.

In the final report of the CRBR (Harty 1974), the Dispersed Flow Film Boiling Model, which took the non-equilibrium flow of the steam/water droplets mixture into consideration, was used. However, they did not clearly explain the details of this method. In the latest CRBR study (Moody and Gabler, 1978), a modified Bishop correlation was proposed :

$$\text{Nu}_f = 0.0193 \text{ Re}_f^{0.8} \text{ Pr}_f^{1.23} \\ * \left[X + \left(1 + X \right) \frac{v_f}{v_g} \right]^{0.68} \left(\frac{v_f}{v_g} \right)^{0.68}$$

where all Re_f and Pr_f are evaluated at T_{film}

$$T_{\text{film}} = \frac{1}{2} (T_{\text{wall}} + T_{\text{bulk}})$$

This correlation, as the CRBR claimed, should be good at high water mass flows rate. After considering these suggestions from the CRBR reports carefully, we chose to try an alternative correlation for the post-dryout (PDO) regime proposed by Groeneveld (Groeneveld, 1973), the same correlation used in the Tajbakhsh code. This correlation appears to give good results in our calculations. So the Groeneveld correlation has been adopted:

$$\text{Nu}_g = 0.0109 \left(\text{Re}_g \left[X + \frac{v_f}{v_g} (1-X) \right] \right)^{0.8} \text{Pr}_{g,w}^{1.23} Y^{1.15} \\ Y = 1 - 0.1 \left(\frac{v_g}{v_f} - 1 \right)^{0.4} (1 - X)^{0.4}$$

where Re_g is evaluated at vapor properties
 $\text{Pr}_{g,w}$ is evaluated at wall temperature

4.8.5 Single Phase Vapor (SPV)

In the CRBR study (Harty, 1974), the Bishop correlation was found to give good agreement with their test data. The correlation

is

$$Nu_f = 0.0073 Re_f^{0.886} Pr_f^{0.61}$$

where all Re_f and Pr_f are evaluated at T_{film}

$$T_{film} = \frac{1}{2} (T_{wall} + T_{bulk})$$

4.9 Pressure Drop in the Steam Generator

4.9.1 Primary Fluid Side

There are limited data on the sodium fluoroborate molten salt pressure drops. It has been assumed that the pressure drop can be approximated by the standard pressure drop model as

$$\Delta P = \Delta P_f + \Delta P_a$$

and

$$\Delta P_f = \frac{2 f_o v G^2 L}{D_h}$$

$$\Delta P_a \approx 0 \quad (\text{ for single-phase liquids })$$

$$\Delta P_g = 0 \quad (\text{ for a closed loop system })$$

where f_o is the Fanning friction factor
 v is the specific volume of the fluid
 G is the mass flow rate
 L is the length of the tube
 D_h is the hydraulic diameter
 g is the gravity

4.9.2 Secondary Fluid Side

The pressure drops for the single phase liquid or vapor can be estimated by

$$\Delta P = \Delta P_f + \Delta P_a + \Delta P_g$$

and

$$\Delta P_f = \frac{2 f_o v_m G^2 L}{D_{ii}}$$

$$\Delta P_a = (v_o - v_i) G^2$$

$$\Delta P_g = \frac{g L}{v_m}$$

where v_o is the outlet specific volume of water
 v_i is the inlet specific volume of water
 v_m is the average specific volume of water
 D_{ii} is the inner diameter of the tube

There is greater difficulty in estimating the pressure drop in the water-vapor mixture accurately because of the complexities of the various two-phase flow regimes. However, a lot of study has been done to estimate the pressure drop of the two-phase water/steam flow inside tubes. The pressure drop is mainly caused by three factors : friction on the tube wall, acceleration of the fluid, and the gravity loss. One way to estimate the pressure drop is to use a correlation of experiment data such as the Martinelli-Nelson correlation (Martinelli, Nelson, 1947). The equations are

$$\Delta P = \Delta P_f + \Delta P_a + \Delta P_g$$

and

$$\Delta P_f = \frac{2 f_o v_f G^2 L}{D_{ii}} \quad r3$$

$$\Delta P_a = v_f G^2 \quad r2$$

$$\Delta P_g = \frac{g L}{v_f} \quad r4$$

$$\text{where } r3 = \frac{1}{X_{ex}} \int_0^{X_{ex}} \phi_{fo}^2 dx$$

$$r2 = \frac{X_{\alpha}^2 v_g}{\alpha v_f} + \frac{(1 - X_{\alpha})^2}{1 - \alpha} - 1$$

$$r4 = \frac{1}{X_{ex}} \int_0^{X_{ex}} \left[\frac{v_g}{v_f} \alpha + (1 - \alpha) \right] dx$$

These terms are usually given in graphical form (Collier, 1981) as shown in FIGURES 4-6,7,8 and must be input as a table of values or as fitted equations. In the HYLIFE-II study, digitized tables are used as shown in TABLES 4-2 and 4-3. Double linear interpolation is used to obtain values at intermediate conditions.

4.10 Conclusions and Future Work

After the SG Code validation by comparison with both the Tajbakhsh calculations and the CRBR data, it is now considered to be usable in the HYLIFE-II steam generator design study. As mentioned earlier in the chapter, three sets of hockey stick steam generator systems which are scaled up from the CRBR study will be used for the HYLIFE-II.

The scaled up steam generator proposed for the commercial LMFBR is shown in FIGURE 4-9. It can be seen that both the length and the tube bundle diameter have been increased. For the HYLIFE-II steam generators, we plan to investigate different combinations of length and tube bundle diameter with the objective of finding the best dimensions for the minimum cost of electricity (COE).

TABLE 4-1 Physical properties of MSBR coolant salt

Composition	NaBF ₄ .NaF (92-8 mole %)	
Melting-Point temperature	384°C	
Properties at	454°C	621°C
Density, g/cm ³	1.93	1.82
Heat capacity, cal/g-°C (kJ/kg.K)	0.36 (1.506)	0.36 (1.506)
Viscosity, centipoise (Ns/m ²)	1.9 (0.0019)	1.1 (0.0011)
Thermal conductivity, W/°C-cm	0.0045	0.0040
Vapor pressure, torr (kPa)	8 (1.1)	252 (33.3)

TABLE 4-2 Friction multiplier as a function of quality and absolute pressure
(Martinelli-Nelson)

Steam Quality	Pressure (Pa)								
	101353	689476	3447380	6894760	10342140	13789520	17236900	20684280	22104601
0.00	1	1	1	1	1	1	1	1	1.00
0.01	5.6	3.5	1.8	1.6	1.35	1.2	1.1	1.05	1.00
0.05	30	15	5.3	3.6	2.4	1.75	1.43	1.17	1.00
0.10	69	28	8.9	5.4	3.4	2.45	1.75	1.30	1.00
0.20	150	56	16.2	8.6	5.1	3.25	2.19	1.51	1.00
0.30	245	83	23.0	11.6	6.8	4.04	2.62	1.68	1.00
0.40	350	115	29.2	14.4	8.4	4.82	3.02	1.83	1.00
0.50	450	145	34.9	17.0	9.9	5.59	3.38	1.97	1.00
0.60	545	174	40.0	19.4	11.1	6.34	3.70	2.10	1.00
0.70	625	199	44.6	21.4	12.1	7.05	3.96	2.23	1.00
0.80	685	216	48.6	22.9	12.8	7.70	4.15	2.35	1.00
0.90	720	210	48.0	22.3	13.0	7.95	4.20	2.38	1.00
1.00	525	130	30.0	15.0	8.6	5.90	3.70	2.15	1.00

TABLE 4-3 Void fraction α as a function of quality and absolute pressure
(Martinelli-Nelson)

Steam Quality	Pressure (Pa)								
	101353	689476	3447380	6894760	10342140	13789520	17236900	20684280	22104601
0.00	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
0.01	0.683	0.550	0.317	0.167	0.110	0.067	0.050	0.033	0.014
0.02	0.750	0.643	0.443	0.286	0.200	0.129	0.093	0.050	0.017
0.03	0.786	0.700	0.517	0.371	0.257	0.171	0.129	0.071	0.021
0.04	0.814	0.729	0.586	0.429	0.321	0.214	0.157	0.100	0.029
0.05	0.836	0.757	0.621	0.474	0.357	0.257	0.186	0.114	0.036
0.06	0.843	0.771	0.643	0.514	0.400	0.300	0.211	0.134	0.047
0.07	0.846	0.786	0.657	0.543	0.429	0.329	0.231	0.150	0.057
0.08	0.864	0.800	0.693	0.571	0.460	0.357	0.257	0.171	0.071
0.09	0.871	0.807	0.707	0.600	0.493	0.379	0.279	0.186	0.093
0.10	0.899	0.814	0.721	0.614	0.514	0.400	0.307	0.200	0.100
0.20	0.929	0.871	0.807	0.736	0.650	0.557	0.443	0.343	0.200
0.30	0.964	0.911	0.857	0.803	0.736	0.671	0.557	0.443	0.300
0.40	0.971	0.929	0.893	0.843	0.793	0.743	0.664	0.540	0.400
0.50	0.986	0.943	0.921	0.893	0.843	0.800	0.729	0.629	0.500
0.60	0.994	0.957	0.936	0.914	0.879	0.843	0.800	0.707	0.600
0.70	0.996	0.979	0.957	0.943	0.914	0.900	0.843	0.793	0.700
0.80	0.997	0.993	0.974	0.957	0.943	0.936	0.900	0.850	0.800
0.90	0.999	0.997	0.994	0.990	0.986	0.971	0.943	0.921	0.900
1.00	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000

FIGURE 4-2 Flow chart of one steam generator system rated at 1000 MWt for HYLIFE-II

W = kg/s
T = K

H = KJ/kg
P = MPa

X = steam quality

* pressure drop is neglected in primary and secondary fluids

FIGURE 4-3 Definition of regimes inside an evaporator

FIGURE 4-4 Definition of regime inside a superheater

FIGURE 4-5 Critical heat flux in steam boiling (Collier, 1981)

FIGURE 4-6 Value of ϕ_{fo}^2 as a function of pressure and mass quality (Collier, 1981)

FIGURE 4-7 Void fraction α as a function of pressure and mass quality (Collier, 1981)

FIGURE 4-8 Value of r_4 as a function of pressure and mass quality (Thom, 1964)

AI-MSG
35-MWt

X3
➔

CRBRP
90/118-MWt

X3
➔

COMMERCIAL 350-MWt

78-AU18-38-39A

FIGURE 4-9 Scale up of the CRBR steam generator

APPENDIX A

HAND CALCULATION OF THE THERMAL-HYDRAULIC PERFORMANCE OF THE CLINCH RIVER BREEDER REACTOR SODIUM/WATER EVAPORATOR AND SUPERHEATER USING AVERAGE FLUID PROPERTIES

Appendix A

A.1 Check Case for the Sodium/Water Evaporator

A.1.1 Choose the Initial Parameters

Water side

$$\dot{m}_{w(EV)} = 0.5 \times 10^6 \text{ kg/hr}$$

Sodium side

$$\dot{m}_{p(EV)} = 3.05 \times 10^6 \text{ kg/hr}$$

Evaporator geometry

$$\text{Pitch} = 0.031 \text{ m}, N_t = 757, D_{to} = 0.0159 \text{ m}$$

$$D_{ti} = 0.01036 \text{ m}$$

A.1.2 Tube Bundle and Tube Geometry

$$\frac{1}{4} \pi D_{TB}^2 = 0.866 (\text{Pitch})^2 N_t$$

$$D_{TB} = [4(0.866)(0.031)^2(757)/\pi]^{1/2} = 0.896 \text{ m}$$

$$\begin{aligned} P/D &= \frac{\text{Pitch}}{D_{to}} \\ &= \frac{0.031}{0.0159} = 1.95 \end{aligned}$$

$$\begin{aligned} D_h &= \frac{4 \text{ Area}}{P_{wet}} \\ &= \frac{4 [\frac{1}{2}(0.866)(31)^2 - \frac{1}{2}(\frac{1}{4}\pi)(15.9)^2]}{\frac{1}{4}\pi(15.9)} = 0.0507 \text{ m} \end{aligned}$$

A.1.3 Mass Flow Rate

Water side

$$\begin{aligned}
 G_{s(EV)} &= \frac{\dot{m}_{s(EV)}}{\frac{1}{4} \pi D_o^2 N_t} \\
 &= \frac{0.5 \times 10^6}{3600} \times \frac{1}{\frac{1}{4} \pi (0.01036)^2 (757)} = 2176.5 \text{ kg.m}^2.\text{s}
 \end{aligned}$$

Sodium side

$$\begin{aligned}
 G_{p(EV)} &= \frac{\dot{m}_{p(EV)}}{\frac{1}{4} \pi [D_{TB}^2 - N_t D_o^2]} \\
 &= \frac{3.05 \times 10^6}{3600} \times \frac{1}{\frac{1}{4} \pi [(0.896)^2 - 757 (0.0159)^2]} \\
 &= 1766.2 \text{ kg/m}^2.\text{s}
 \end{aligned}$$

A.1.4 Curve Fit for the Sodium Properties

Use the third order curve fit for the sodium properties

$$Cp_p = 1855 - 1.849T + 1.787 \times 10^{-3}T^2 - 4.902 \times 10^{-7}T^3$$

$$\begin{aligned}
 Pr_p &= 4.658 \times 10^{-2} - 1.638 \times 10^{-4}T + 2.179 \times 10^{-7}T^2 \\
 &\quad - 9.853 \times 10^{-11}T^3
 \end{aligned}$$

$$\begin{aligned}
 \mu_p &= 2.858 \times 10^{-3} - 9.767 \times 10^{-6}T + 1.257 \times 10^{-8}T^2 \\
 &\quad - 5.588 \times 10^{-12}T^3
 \end{aligned}$$

$$K_p = 100.1 - 2.836 \times 10^{-3}T - 1.095 \times 10^{-4}T^2 + 6.863 \times 10^{-8}T^3$$

$$1/v_p = 1246 - 1.507T + 2.251 \times 10^{-3}T^2 - 1.289 \times 10^{-6}T^3$$

A.1.5 Regime 1 & 2 (Single Phase Liquid and Subcooling Boiling) to Point P

A.1.5.1 Input Temperature and Pressure

Water side

$$T_{sl(1\&2)} = 558.15 \text{ K}, T_{so(1\&2)} = 605.2 \text{ K}, P_{sl(1\&2)} = 13.2 \text{ MPa}$$

Sodium side

$$T_{po(1\&2)} = 622.15 \text{ K}$$

A.1.5.2 From the steam table

$$i_{si(1\&2)} = 1259.72 \text{ kJ/kg}$$

$$i_{so(1\&2)} = 1538 \text{ kJ/kg}$$

A.1.5.3 First Law of Thermodynamics

$$\begin{aligned} P_{th(1\&2)} &= m_{s(EV)} (i_{si(1\&2)} - i_{so(1\&2)}) \\ &= \frac{0.5 \times 10^6}{3600} (1538 - 1259.72) \times 10^3 = 38.7 \text{ MW} \end{aligned}$$

$$\begin{aligned} \bar{T}_{s(1\&2)} &= \frac{1}{2} (T_{so(1\&2)} + T_{si(1\&2)}) \\ &= \frac{1}{2} (605.2 + 558.15) = 581.68 \text{ K} \end{aligned}$$

A.1.5.4 First Law of Thermodynamics

$$C_{pP(1\&2)}(T_{po(1\&2)}) = 1278.3 \text{ J/kg.K}$$

$$P_{th(1\&2)} = m_{p(EV)} C_{pP(1\&2)} (T_{pi(1\&2)} - T_{po(1\&2)})$$

$$38.7 \times 10^6 = \frac{3.05 \times 10^6}{3600} (T_{pi(1\&2)} - 622.15) \times 10^3$$

$$T_{pi(1\&2)} = 657.84 \text{ K}$$

$$\begin{aligned} \bar{T}_{p(1\&2)} &= \frac{1}{2} (T_{pi(1\&2)} + T_{po(1\&2)}) \\ &= \frac{1}{2} (657.84 + 622.15) = 639.99 \text{ K} \end{aligned}$$

A.1.5.5 Water properties

$$Pr_{s(1\&2)}(\bar{T}_{s(1\&2)}) = 0.9287$$

$$\mu_{s(1\&2)}(\bar{T}_{s(1\&2)}) = 0.00008406$$

$$K_{s(1\&2)}(\bar{T}_{s(1\&2)}) = 0.5364$$

$$Re_{s(1\&2)} = \frac{G_{s(EV)} D_{ii}}{\mu_{s(1\&2)}} = \frac{2176.5 (0.01036)}{0.00008406} = 2.68 \times 10^5$$

$$Nu_{s(1\&2)} = 0.023 Re_{s(1\&2)}^{0.8} Pr_{s(1\&2)}^{0.4}$$

(From CRBR Final Reoprt, 1974 Harty)

$$= 0.023 (2.68 \times 10^5)^{0.8} (0.9287)^{0.4} = 492$$

$$h_{s(1\&2)} = \frac{Nu_{s(1\&2)} K_{s(1\&2)}}{D_{li}} = \frac{492 (0.5364)}{0.01036} = 25459$$

A.1.5.6 Wall properties

$$\bar{T}_{w(1\&2)} \approx \frac{1}{2} (\bar{T}_{p(1\&2)} + \bar{T}_{s(1\&2)}) = \frac{1}{2} (639.99 + 581.68) = 610.84 \text{ K}$$

$$K_{w(1\&2)} (\bar{T}_{w(1\&2)}) = 49.37 - 0.01713 \bar{T}_{w(1\&2)} \quad (\text{CRBR Final Report})$$

$$= 38.91 \text{ W/m.K}$$

(5.7) Sodium properties

$$Pr_{p(1\&2)} (\bar{T}_{p(1\&2)}) = 0.005171$$

$$\mu_{p(1\&2)} (\bar{T}_{p(1\&2)}) = 0.000291$$

$$K_{p(1\&2)} (\bar{T}_{p(1\&2)}) = 71.4251$$

$$Re_{p(1\&2)} = \frac{G_{p(EV)} D_h}{\mu_{p(1\&2)}} = \frac{1766.2 (0.0507)}{0.000291} = 308048$$

$$Nu = 0.25 + 6.20 P/D + (-0.007 + 0.032 P/D) Pe^{0.8-0.024 P/D}$$

(CRBR Final Report)

with $P/D = 1.95$

$$Nu_{p(1\&2)} = 12.34 + 0.0554 (Re_{p(1\&2)} Pr_{p(1\&2)})^{0.7532}$$

$$= 12.34 + 0.0554 [(308048) (0.005171)]^{0.7532}$$

$$= 26.54$$

$$h_{p(1\&2)} = \frac{Nu_{p(1\&2)} K_{p(1\&2)}}{D_h}$$

$$= \frac{26.54 (71.4251)}{0.0507} = 37495 \text{ W/m}^2 \cdot \text{K}$$

A.1.5.7 Thermal resistance

$$R_{s(1\&2)} = \frac{1}{\pi D_{ii} N_i L_{(1\&2)} h_{s(1\&2)}}$$

$$= \frac{1}{\pi (0.01036) (757) (24459) L_{(1\&2)}} = \frac{1.59 \times 10^{-6}}{L_{(1\&2)}} \text{ K/W}$$

$$R_{p(1\&2)} = \frac{1}{\pi D_{to} N_i L_{(1\&2)} h_{p(1\&2)}}$$

$$= \frac{1}{\pi (0.0159) (757) (37495) L_{(1\&2)}} = \frac{7.05 \times 10^{-7}}{L_{(1\&2)}} \text{ K/W}$$

$$R_{w(1\&2)} = \frac{\ln(D_{to} / D_{ii})}{2 \pi N_i L K_{w(1\&2)}}$$

$$= \frac{\ln(0.0159/0.01036)}{2\pi (757) (38.97) L_{(1\&2)}} = \frac{2.31 \times 10^{-6}}{L_{(1\&2)}} \text{ K/W}$$

$$R_{(1\&2)} = R_{s(1\&2)} + R_{p(1\&2)} + R_{w(1\&2)}$$

$$= \frac{(1.67 + 0.705 + 2.31) \times 10^{-6}}{L_{(1\&2)}} = \frac{4.69 \times 10^{-5}}{L_{(1\&2)}} \text{ K/W}$$

where $L_{(1\&2)}$ - length of tube in the regimes 1 & 2

A.1.5.8 Log Mean Temperature Difference Method

$$\Delta T_{LM(1\&2)} = \frac{(T_{pi(1\&2)} - T_{so(1\&2)}) - (T_{po(1\&2)} - T_{si(1\&2)})}{\ln[(T_{pi(1\&2)} - T_{so(1\&2)}) / (T_{po(1\&2)} - T_{si(1\&2)})]}$$

$$= \frac{(657.84 - 605.2) - (622.15 - 558.15)}{\ln[(657.84 - 605.2) / (622.15 - 558.15)]} = 58.14 \text{ K}$$

$$P_{th(1\&2)} = \frac{\Delta T_{LM(1\&2)}}{R} = 58.14 \times \frac{L_{(1\&2)}}{4.69 \times 10^{-6}} = 38.7 \times 10^6$$

$$L_{(1\&2)} = 3.1 \text{ m}$$

A.1.6 Pressure Drop in Regime 1 & 2

A.1.6.1 Pressure drop on the water side

$$\begin{aligned} f_{os(1\&2)} &= 0.046 \operatorname{Re}_{s(1\&2)}^{-0.2} \\ &= 0.046 (2.68 \times 10^5)^{-0.2} = 0.003776 \end{aligned}$$

$$v_{si(1\&2)} = 0.0013337 \text{ m}^3/\text{kg}, \quad v_{so(1\&2)} = 0.0015662 \text{ m}^3/\text{kg}$$

$$\begin{aligned} v_{sm(1\&2)} &= \frac{1}{2} (v_{si(1\&2)} + v_{so(1\&2)}) \\ &= \frac{1}{2} (0.0013337 + 0.0015662) = 0.001450 \text{ m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} \Delta P_{sf(1\&2)} &= \frac{2 f_{os(1\&2)} v_{sm(1\&2)} G_s(\text{EV})^2 L_{(1\&2)}}{D_{ii}} \\ &= \frac{2 (0.003776) (0.00145) (2176.5)^2 (3.1)}{0.01036} = 0.015 \text{ MPa} \end{aligned}$$

$$\begin{aligned} \Delta P_{ss(1\&2)} &= (v_{so(1\&2)} - v_{si(1\&2)}) G_s(\text{EV})^2 \\ &= (0.0015662 - 0.0013337) (2176.5)^2 = 0.001 \text{ MPa} \end{aligned}$$

$$\Delta P_{sg(1\&2)} = \frac{g L_{(1\&2)}}{v_{sm(1\&2)}} = \frac{9.81 (3.1)}{0.00145} = 0.021 \text{ MPa}$$

$$\begin{aligned} \Delta P_s(1\&2) &= \Delta P_{sf(1\&2)} + \Delta P_{ss(1\&2)} + \Delta P_{sg(1\&2)} \\ &= 0.015 + 0.001 + 0.021 = 0.037 \text{ MPa} \end{aligned}$$

A.1.6.2 Pressure drop on the sodium side

$$\begin{aligned} f_{op(1\&2)} &= 0.046 \operatorname{Re}_{p(1\&2)}^{-0.2} \\ &= 0.046 (308048)^{-0.2} = 0.003673 \end{aligned}$$

$$1/v_{(1\&2)}(\bar{T}_{p(1\&2)}) = 865.8 \text{ kg/m}^3$$

$$v_{(1\&2)}(\bar{T}_{p(1\&2)}) = 0.001155 \text{ m}^3/\text{kg}$$

$$\begin{aligned} \Delta P_{pf(1\&2)} &= \frac{2 f_{op(1\&2)} v_{p(1\&2)} G_p(\text{EV})^2 L_{(1\&2)}}{D_h} \\ &= \frac{2 (0.003673) (0.001155) (1766.2)^2 (3.1)}{0.0507} = 0.002 \text{ MPa} \end{aligned}$$

$$\Delta P_{pa(1\&2)} = (v_{po(1\&2)} - v_{pi(1\&2)}) G_{p(EV)}^2 \approx 0 \text{ MPa}$$

$$\Delta P_{pg(1\&2)} = 0 \quad (\text{because sodium is in a closed loop})$$

$$\begin{aligned} \Delta P_{p(1\&2)} &= \Delta P_{pi(1\&2)} + \Delta P_{pa(1\&2)} + \Delta P_{pg(1\&2)} \\ &= 0.002 + 0 + 0 = 0.002 \text{ MPa} \end{aligned}$$

A.1.7 Regime 3 (Saturation Nucleating Boiling)

A.1.7.1 Input temperature and pressure

Water side

$$T_{w(3)} = 605.2 \text{ K}, T_{bo(3)} = 605.2 \text{ K}, P_{w(3)} = 13.2 \text{ MPa},$$

$$X_{w(3)} = 0.0$$

Sodium side

$$T_{po(3)} = 657.84 \text{ K}$$

A.1.7.2 From the steam table

$$i_f = 1538 \text{ kJ/kg}$$

$$i_g = 2656.2 \text{ kJ/kg}$$

$$i_{fg} = 2656.2 - 1538 = 1118.2 \text{ kJ/kg}$$

$$v_f = 0.001575 \text{ m}^3/\text{kg}$$

$$v_g = 0.012522 \text{ m}^3/\text{kg}$$

A.1.7.3 Exit steam quality

for $q'' > 6.3 \times 10^5 \text{ W/m}^2$

$$X_{DNB} = \frac{1614683}{i_{fg} G^{1/2}} \times \frac{v_g}{v_f} \quad (\text{CRBR Final Report})$$

where $i_{fg} - \text{J/kg}$, $G - \text{kg/m}^2 \cdot \text{s}$

$$X_{DNB} = \frac{1614683}{1131 \times 10^3 (2176.5)^{1/2}} \times \frac{0.012522}{0.001575} = 0.246$$

A.1.7.4 First Law of Thermodynamics

$$i_{si(3)} = i_f + i_{fg} X_{i(3)} = 1538 + 1118.2(0) = 1538 \text{ kJ/kg}$$

$$i_{so(3)} = i_f + i_{fg} X_{DNB} = 1538 + 1118.2(0.246) = 1813 \text{ kJ/kg}$$

$$\begin{aligned} P_{th(3)} &= \dot{m}_{(EV)} (i_{si(3)} - i_{so(3)}) \\ &= \frac{0.5 \times 10^6}{3600} (1813 - 1538) \times 10^3 = 38.2 \text{ MW} \end{aligned}$$

$$\bar{T}_{s(3)} = \frac{1}{2} (T_{so(3)} + T_{si(3)}) = \frac{1}{2} (605.2 + 605.2) = 605.2 \text{ K}$$

A.1.7.5 First Law of Thermodynamics

$$C_{pP(3)}(T_{po(3)}) = 1272.4 \text{ J/kg.K}$$

$$P_{th(3)} = \dot{m}_{P(EV)} C_{pP(3)} (T_{pi(3)} - T_{po(3)})$$

$$38.2 \times 10^6 = \frac{3.05 \times 10^6}{3600} (1272.4) (T_{pi(3)} - 657.84) \times 10^3$$

$$T_{pi(3)} = 675.56 \text{ K}$$

$$\bar{T}_{P(3)} = \frac{1}{2} (T_{pi(3)} + T_{po(3)}) = \frac{1}{2} (675.56 + 657.84) = 675.56 \text{ K}$$

A.1.7.6 Water properties

$$\overline{q_{li(3)}}^n = \frac{P_{th(3)}}{N_i \pi D_{li} L_{(3)}}$$

where $L_{(3)}$ - length of the tube in the regime 3

$$\Delta \bar{T}_{J\&L} = 25 \overline{q_{li(3)}}^{0.25} e^{-P/62} \quad (\text{Jens and Lottes})$$

where $\overline{q_{li(3)}}^n$ - MW/m², P - bar

$$\overline{q_{li(3)}}^n = \frac{38.2 \times 10^6}{\pi (757) (0.01036) L_{(3)}} = \frac{1.6}{L_{(3)}} \text{ MW/m}^2$$

$$\Delta \bar{T}_{J\&L} = 25 \left(\frac{1.6}{L_{(3)}} \right)^{0.25} e^{-130/62} = 3.32 L_{(3)}^{-0.25}$$

$$\bar{T}_{w(3)} = T_{mi} + \Delta \bar{T}_{J\&L} = 605.2 + 3.32 L_{(3)}^{-0.25}$$

$$\begin{aligned}
 h_{w(3)} &= \frac{P_{th(3)}}{\pi D_{li} N_t (\bar{T}_{w(3)} - T_{sat}) L_{(3)}} \\
 &= \frac{38.2 \times 10^6}{\pi (0.01036) (757)} \times L_{(3)}^{-0.75} = 4.67 \times 10^5 L_{(3)}^{-0.75}
 \end{aligned}$$

A.1.7.7 Wall properties

$$\bar{T}_{w(3)} \approx \frac{1}{2} (\bar{T}_{p(3)} + \bar{T}_{s(3)}) = \frac{1}{2} (675.56 + 605.2) = 640.38 \text{ K}$$

$$\begin{aligned}
 K_{w(3)}(\bar{T}_{w(3)}) &= 49.37 - 0.01713 \bar{T}_{w(3)} \quad (\text{CRBR Final Report}) \\
 &= 38.40 \text{ W/m}
 \end{aligned}$$

A.1.7.8 Sodium properties

$$Pr_{p(3)}(\bar{T}_{p(3)}) = 0.004991$$

$$\mu_{p(3)}(\bar{T}_{p(3)}) = 0.000274 \text{ kg/m.s}$$

$$K_{p(3)}(\bar{T}_{p(3)}) = 69.370$$

$$Re_{p(3)} = \frac{G_{p(EV)} D_h}{\mu_{p(3)}} = \frac{1766.2 (0.0507)}{0.000274} = 327489$$

with $P/D = 1.95$

$$\begin{aligned}
 Nu_{p(3)} &= 12.34 + 0.0554 (Re_{p(3)} Pr_{p(3)})^{0.7532} \\
 &= 12.34 + 0.0554 [(327489) (0.004991)]^{0.7532} = 26.92
 \end{aligned}$$

$$h_{p(3)} = \frac{Nu_{p(3)} K_{p(3)}}{D_h} = \frac{26.92 (69.37)}{0.0507} = 36799 \text{ W/m}^2 \cdot \text{K}$$

A.1.7.9 Thermal resistance

$$\begin{aligned}
 R_{u(3)} &= \frac{1}{\pi D_{li} N_t L_{(3)} h_{u(3)}} \\
 &= \frac{L_{(3)}^{0.75}}{\pi (0.01036) (757) (L_{(3)}) (4.67 \times 10^5)} \\
 &= 8.68 \times 10^{-8} L_{(3)}^{-0.25}
 \end{aligned}$$

$$\begin{aligned}
 R_{p(3)} &= \frac{1}{\pi D_{to} N_t L_{(3)} h_{p(3)}} \\
 &= \frac{1}{\pi (0.0159) (757) (36799) L_{(3)}} = \frac{7.19 \times 10^{-7}}{L_{(3)}} \text{ K/W} \\
 R_{w(3)} &= \frac{\ln(D_{to} / D_{ti})}{2 \pi N_t L_{(3)} K_{w(3)}} \\
 &= \frac{\ln(0.0159/0.01036)}{2 \pi (757) (38.40) L_{(3)}} = \frac{2.35 \times 10^{-6}}{L_{(3)}} \\
 R_{(3)} &= R_{s(3)} + R_{p(3)} + R_{w(3)} \\
 &= 8.68 \times 10^{-8} L_{(3)}^{-0.25} + \frac{(0.719 + 2.345 \times 10^{-6})}{L_{(3)}}
 \end{aligned}$$

A.1.7.10 Log Mean Temperature Difference Method

$$\begin{aligned}
 T_{LM(3)} &= \frac{(T_{pi(3)} - T_{so(3)}) - (T_{po(3)} - T_{si(3)})}{\ln[(T_{pi(3)} - T_{so(3)}) / (T_{po(3)} - T_{si(3)})]} \\
 &= \frac{(693.29 - 605.2) - (657.84 - 605.2)}{\ln[(693.29 - 605.2) / (657.84 - 605.2)]} = 68.85 \text{ K} \\
 P_{th(3)} &= \frac{T_{LM(3)}}{R_{(3)}} = \frac{68.85}{R_{(3)}} = 38.2 \times 10^6
 \end{aligned}$$

By trial and error

$$L_{(3)} = 1.3 \text{ m}$$

A.1.7.11 Check on the heat flux inside the tube

$$\overline{q_{i(3)}} = \frac{1.6}{L_{(3)}} = \frac{1.6}{1.3} = 1.23 \text{ MW/m}^2 > 0.65 \text{ MW/m}^2$$

The choice of the X_{DNB} is justified

A.1.8 Pressure Drop on the Regime 3

A.1.8.1 Pressure drop on the water side

$$\mu_{w(3)} = 0.00007361 \text{ kg/m.s}$$

$$Re_{w(3)} = \frac{G_{w(EV)} D_{ii}}{\mu_{w(3)}} = \frac{2176.5(0.01036)}{0.00007361} = 3.06 \times 10^5$$

$$f_{w(3)} = 0.046 Re_{w(3)}^{-0.2}$$

$$= 0.046 (3.06 \times 10^5)^{-0.2} = 0.003677$$

$$\Delta P_{w(3)} = \frac{2 f_{w(3)} v_{wf} G_{w(EV)}^2 L_{(3)}}{D_{ii}} r_3$$

$$\text{where } r_3 = \frac{1}{X_{DNB}} \int_0^{X_{DNB}} \phi_{fo}^2 dx$$

ϕ_{fo}^2 - Martinelli-Nelson two phase frictional multiplier

$$\Delta P_{w(3)} = \frac{2(0.003677)(0.001575)(2176.5)^2(1.3)}{0.01036} (2.8)$$

$$= 0.019 \text{ MPa}$$

$$\Delta P_{m(3)} = v_{mf} G_{w(EV)}^2 r_2$$

$$\text{where } r_2 = \frac{X_{DNB}^2 v_g}{\alpha v_f} + \frac{(1 - X_{DNB})^2}{1 - \alpha} - 1$$

α - Martinelli-Nelson void fraction

$$\Delta P_{m(3)} = (0.001575)(2176.5)^2(1.3) = 0.010 \text{ MPa}$$

$$\Delta P_{g(3)} = \frac{g L_{(3)}}{v_{gf}} r_4$$

$$\text{where } r_4 = \frac{1}{X_{DNB}} \int_0^{X_{DNB}} \left[\frac{v_f}{v_g} \alpha + (1 - \alpha) \right] dx$$

$$\Delta P_{g(3)} = \frac{9.81(1.3)}{0.001575}(0.6) = 0.004 \text{ MPa}$$

$$\begin{aligned}\Delta P_{s(3)} &= \Delta P_{sl(3)} + \Delta P_{sm(3)} + \Delta P_{sg(3)} \\ &= 0.019 + 0.010 + 0.004 = 0.033 \text{ MPa}\end{aligned}$$

A.1.8.2 Pressure drop on the sodium side

$$\begin{aligned}f_{op(3)} &= 0.064 \text{ Re}_{p(3)}^{-0.2} \\ &= 0.064 (327489)^{-0.2} = 0.003628\end{aligned}$$

$$1/v_{(3)}(\bar{T}_{p(3)}) = 857.6 \text{ kg/m}^3$$

$$v_{(3)}(\bar{T}_{p(3)}) = 0.001166 \text{ m}^3/\text{kg}$$

$$\begin{aligned}\Delta P_{p(3)} &= \frac{2 f_{op(3)} v_{p(3)} G_{p(EV)}^2 L_{(3)}}{D_h} \\ &= \frac{2 (0.003628) (0.001166) (1766.2)^2 (1.3)}{0.0507} = 0.001 \text{ MPa}\end{aligned}$$

$$\Delta P_{ps(3)} = (v_{po(3)} - v_{pi(3)}) G_{p(EV)}^2 \approx 0 \text{ MPa}$$

$$\Delta P_{pg(3)} = 0 \quad (\text{because sodium is in a closed loop})$$

$$\begin{aligned}\Delta P_{p(3)} &= \Delta P_{p(3)} + \Delta P_{ps(3)} + \Delta P_{pg(3)} \\ &= 0.001 + 0 + 0 = 0.001 \text{ MPa}\end{aligned}$$

A.1.9 Regime 4 (Film Boiling)

A.1.9.1 Input temperature and pressure

Water side

$$T_{si(4)} = 605.2 \text{ K}, T_{so(4)} = 605.2 \text{ K}, P_{si(4)} = 13.2 \text{ MPa},$$

$$X_{i(4)} = 0.246, X_{o(4)} = 0.50$$

Sodium side

$$T_{po(4)} = 693.29 \text{ K}$$

A.1.9.2 From the steam table

$$i_{si(4)} = i_f + i_{fg} X_{i(4)} = 1538 + 1118.2(0) = 1813 \text{ kJ/kg}$$

$$i_{so(4)} = i_f + i_{fg} X_{o(4)} = 1538 + 1118.2(0.246) = 2097 \text{ kJ/kg}$$

A.1.9.3 First Law of Thermodynamics

$$P_{th(4)} = m_{s(EV)} (i_{si(4)} - i_{so(4)})$$

$$= \frac{0.5 \times 10^6}{3600} (2097-1813) \times 10^3 = 39.4 \text{ MW}$$

$$\bar{T}_{s(4)} = \frac{1}{2} (T_{so(4)} + T_{si(4)}) = \frac{1}{2} (605.2+605.2) = 605.2 \text{ K}$$

A.1.9.4 First Law of Thermodynamics

$$C_{pp(4)}(T_{po(4)}) = 1268.7 \text{ kJ/kg.K}$$

$$P_{th(4)} = m_{p(EV)} C_{p(4)} (T_{pi(4)} - T_{po(4)})$$

$$39.4 \times 10^6 = \frac{3.05 \times 10^6}{3600} (1268.7) (T_{pi(4)} - 693.29) \times 10^3$$

$$T_{pi(4)} = 729.98 \text{ K}$$

$$\bar{T}_{p(4)} = \frac{1}{2} (T_{pi(4)} + T_{po(4)}) = \frac{1}{2} (729.98+693.29) = 711.63 \text{ K}$$

A.1.9.5 Wall properties

$$\bar{T}_{w(4)} \approx \frac{1}{2} (\bar{T}_{p(4)} + \bar{T}_{s(4)}) = \frac{1}{2} (711.63+605.2) = 658.42 \text{ K}$$

$$K_{w(4)}(\bar{T}_{w(4)}) = 49.37 - 0.01713 \bar{T}_{w(4)} \quad (\text{CRBR Final Report})$$

$$= 38.09 \text{ W/m.K}$$

A.1.9.6 Water properties

$$Nu_g = 0.0109 \left(Re_g \left[X + \frac{v_f}{v_g} (1 - X) \right] \right)^{0.8} Pr_{g,w}^{1.23} Y^{-1.15}$$

$$Y = 1 - 0.1 \left(\frac{v_g}{v_f} - 1 \right)^{0.4} (1 - X)^{0.4}$$

where Re_g is evaluated at vapor properties

$Pr_{g,w}$ is evaluated at vapor properties at $\bar{T}_{w(4)}$

$$\mu_{sg(4)}(\bar{T}_{s(4)}) = 0.0000218 \text{ kg/m.s}$$

$$Pr_{sg,w(4)}(T_{w(4)}) = 1.3185$$

$$K_{sg(4)}(\bar{T}_{s(4)}) = 0.09972 \text{ W/m.K}$$

$$v_{sf(4)} = v_f = 0.001566 \text{ m}^3/\text{kg}$$

$$v_{sg(4)} = v_g = 0.012781 \text{ m}^3/\text{kg}$$

$$Re_{sg(4)} = \frac{G_{s(EV)} D_{ii}}{\mu_{sg(4)}} = \frac{2176.5(0.01036)}{0.0000218} = 1.03 \times 10^6$$

$$\frac{v_{af(4)}}{v_{sg(4)}} = \frac{0.001575}{0.012522} = 0.1258$$

$$Y_i = 1 - 0.1 \left(\frac{1}{0.1258} - 1 \right)^{0.4} (1 - 0.246)^{0.4} = 0.7828$$

$$Nu_{sgi(4)} = 0.00109 \{ 1.03 \times 10^6 [0.246 + 0.1258(1 - 0.246)] \}^{0.989} \\ * 1.3185^{1.41} 0.7828^{-1.15} \\ = 244$$

$$Y_o = 1 - 0.1 \left(\frac{1}{0.1258} - 1 \right)^{0.4} (1 - 0.50)^{0.4} = 0.8354$$

$$Nu_{sgo(4)} = 0.00109 \{ 1.03 \times 10^6 [0.50 + 0.1258(1 - 0.50)] \}^{0.989} \\ * 1.3185^{1.41} 0.8254^{-1.15} \\ = 996$$

$$Nu_{sg(4)} = \frac{1}{2} (Nu_{sgi(4)} + Nu_{sgo(4)}) = \frac{1}{2} (244 + 996) = 620$$

$$h_{s(4)} = \frac{Nu_{sg(4)} K_{sg(4)}}{D_{ii}} = \frac{620(0.09972)}{0.01036} = 5969$$

A.1.9.7 Sodium properties

$$Pr_{p(4)}(\bar{T}_{p(4)}) = 0.004855$$

$$\mu_{p(4)}(\bar{T}_{p(4)}) = 0.000259 \text{ kg/m.s}$$

$$K_{p(4)}(\bar{T}_{p(4)}) = 67.36$$

$$Re_{p(4)} = \frac{G_{p(EV)} D_h}{\mu_{p(4)}} = \frac{1766.2(0.0507)}{0.000259} = 345550$$

with $P/D = 1.95$

$$\begin{aligned} Nu_{P(4)} &= 12.34 + 0.0554 (Re_{P(4)} Pr_{P(4)})^{0.7532} \\ &= 12.34 + 0.0554 [(34550)(0.004855)]^{0.7532} = 27.21 \end{aligned}$$

$$h_{P(4)} = \frac{Nu_{P(4)} K_{P(4)}}{D_h} = \frac{27.21(67.3.68)}{0.0507} = 36118 \text{ W/m}^2.\text{K}$$

A.1.1.9.8 Thermal resistance

$$\begin{aligned} R_{s(4)} &= \frac{1}{\pi D_{ti} N_t L_{(4)} h_{s(4)}} \\ &= \frac{1}{\pi (0.01036) (757) (5969) L_{(4)}} = \frac{6.80 \times 10^{-6}}{L_{(4)}} \text{ K/W} \end{aligned}$$

$$\begin{aligned} R_{P(4)} &= \frac{1}{\pi D_{to} N_t L_{(4)} h_{P(4)}} \\ &= \frac{1}{\pi (0.0159) (757) (36118) L_{(4)}} = \frac{7.32 \times 10^{-7}}{L_{(4)}} \text{ K/W} \end{aligned}$$

$$\begin{aligned} R_{w(4)} &= \frac{\ln(D_{to} / D_{ti})}{2 \pi N_t L_{(4)} K_{w(4)}} \\ &= \frac{\ln(0.0159/0.01036)}{2 \pi (757) (38.09) L_{(4)}} = \frac{2.36 \times 10^{-6}}{L_{(4)}} \text{ K/W} \end{aligned}$$

$$\begin{aligned} R_{(4)} &= R_{s(4)} + R_{P(4)} + R_{w(4)} \\ &= \frac{(6.80 + 0.732 + 2.36) \times 10^{-6}}{L_{(4)}} = \frac{9.89 \times 10^{-5}}{L_{(4)}} \end{aligned}$$

where $L_{(4)}$ - length of the tube in regime 4

A.1.9.9 Log Mean Temperature Difference Method

$$\Delta T_{LM(4)} = \frac{(T_{pi(4)} - T_{so(4)}) - (T_{po(4)} - T_{si(4)})}{\ln[(T_{pi(4)} - T_{so(4)}) / (T_{po(4)} - T_{si(4)})]} \\ = \frac{(729.98 - 605.2) - (693.29 - 605.2)}{\ln[(729.98 - 605.2) / (693.29 - 605.2)]} = 105.37 \text{ K}$$

$$P_{th(4)} = \frac{\Delta T_{LM(4)}}{R_{(4)}} = 105.37 \times \frac{L_{(4)}}{9.89 \times 10^{-5}} = 39.4 \times 10^6$$

$$L_{(4)} = 3.7 \text{ m}$$

A.1.10 Pressure Drop on the Regime 4

A.1.10.1 Pressure drop on the water side

$$\mu_{sf(4)} = 0.00007361$$

$$Re_{sf(4)} = \frac{G_{s(EV)} D_{ii}}{\mu_{sf(4)}} = \frac{2176.5(0.01036)}{0.00007361} = 3.06 \times 10^5$$

$$f_{os(4)} = 0.046 Re_{sf(4)}^{-0.2} \\ = 0.046 (3.06 \times 10^5)^{-0.2} = 0.003677$$

$$\Delta P_{sf(4)} = \frac{2 f_{os(4)} v_{sf} G_{s(EV)}^2 L_{(3+4)}}{D_{ii}} r_3 - P_{sf(3)}$$

$$\text{where } r_3 = \frac{1}{X_{ex}} \int_0^{X_{ex}} \phi_{fo}^2 dx$$

ϕ_{fo}^2 - Martinelli-Nelson two phase frictional multiplier

$$\Delta P_{sf(4)} = \frac{2(0.003677)(0.001575)(2176.5)^2(1.3+3.7)}{0.01036} (4.0) - 0.019 \\ = 0.058 \text{ MPa}$$

$$\Delta P_{ss(4)} = v_{sf} G_{s(EV)}^2 r_2 - \Delta P_{ss(3)}$$

$$\text{where } r_2 = \frac{X_{ex}^2 v_g}{\alpha v_f} + \frac{(1 - X_{ex})^2}{1 - \alpha} - 1$$

α - Martinelli-Nelson void fraction

$$\Delta P_{sa(4)} = (0.001575) (2176.5)^2 (2.4) - 0.010 = 0.008 \text{ MPa}$$

$$\Delta P_{sg(4)} = \frac{g L_{(3&4)}}{v_{sf}} r_4 - \Delta P_{sg(3)}$$

$$\text{where } r_4 = \frac{1}{X_{ex}} \int_0^{X_{ex}} \left[\frac{v_f}{v_g} - \alpha + (1 - \alpha) \right] dx$$

$$\Delta P_{sg(4)} = \frac{9.81(1.3+3.7)}{0.001575} (0.5) - 0.004 = 0.007 \text{ MPa}$$

$$\begin{aligned} \Delta P_{s(4)} &= \Delta P_{sa(4)} + \Delta P_{ss(4)} + \Delta P_{sg(4)} \\ &= 0.058 + 0.008 + 0.007 = 0.074 \text{ MPa} \end{aligned}$$

$$\begin{aligned} \Delta P_{s(EV)} &= \Delta P_{s(1&2)} + \Delta P_{s(3)} + \Delta P_{s(4)} \\ &= 0.037 + 0.033 + 0.074 = 0.144 \text{ MPa} \end{aligned}$$

$$\Delta P_{sCRBR(EV)} = 0.8 \text{ MPa}$$

$$\begin{aligned} \Delta P_{sminor(EV)} &= \Delta P_{sCRBR(EV)} - \Delta P_{s(EV)} \\ &= 0.8 - 0.144 = 0.656 \text{ MPa} \end{aligned}$$

$$\begin{aligned} v_{sm} &= \frac{1}{2} (v_{s(1)} + v_{s(5)}) = \frac{1}{2} (0.0013337 + 0.0070485) \\ &= 0.004191 \text{ m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} \Delta P_{sminor(EV)} &= \frac{1}{2} G_{s(EV)}^2 v_{sm} K_{sminor(EV)} \\ 6.56 \times 10^5 &= \frac{1}{2} (2176.5)^2 (0.004191) K_{sminor(EV)} \\ K_{sminor(EV)} &= 66.1 \end{aligned}$$

A.1.10.2 Pressure drop on the sodium side

$$\begin{aligned} f_{op(4)} &= 0.046 \text{ Re}_{p(4)}^{-0.2} \\ &= 0.046 (345550)^{-0.2} = 0.003590 \end{aligned}$$

$$1/v_{p(4)} (\bar{T}_{p(4)}) = 848.9 \text{ kg/m}^3$$

$$v_{p(4)} \bar{T}_{p(4)} = 0.001178 \text{ m}^3/\text{kg}$$

$$\Delta P_{p(4)} = \frac{2 f_{op(4)} v_{p(4)} G_{p(EV)}^2 L_{(4)}}{D_b}$$

$$= \frac{2(0.003590)(0.001178)(1766.2)^2(3.7)}{0.0507} = 0.002 \text{ MPa}$$

$$\Delta P_{pa(4)} = (v_{po(4)} - v_{pi(4)}) G_{p(EV)}^2 \approx 0 \text{ MPa}$$

$$\Delta P_{pg(4)} = 0 \quad (\text{because sodium is in a closed loop})$$

$$\Delta P_{p(4)} = \Delta P_{p(4)} + \Delta P_{pa(4)} + \Delta P_{pg(4)}$$

$$= 0.002 + 0 + 0 = 0.002 \text{ MPa}$$

$$\Delta P_{p(EV)} = \Delta P_{p(1\&2)} + \Delta P_{p(3)} + \Delta P_{p(4)}$$

$$= 0.002 + 0.001 + 0.002 = 0.005 \text{ MPa}$$

$$\Delta P_{pCRBR(EV)} = 0.108 \text{ MPa}$$

$$\Delta P_{pminor(EV)} = \Delta P_{pCRBR(EV)} - \Delta P_{p(EV)}$$

$$= 0.108 - 0.05 = 0.103 \text{ MPa}$$

$$v_{pm} = \frac{1}{2} (v_{po(1)} + v_{pi(5)}) = \frac{1}{2} (0.001155 + 0.001178)$$

$$= 0.001167 \text{ m}^3/\text{kg}$$

$$\Delta P_{pminor(EV)} = \frac{1}{2} G_{p(EV)}^2 v_{pm} K_{pminor(EV)}$$

$$1.03 \times 10^5 = \frac{1}{2} (1766.2)^2 (0.001167) K_{pminor(EV)}$$

$$K_{pminor(EV)} = 56.6$$

A.1.11 Safety Factor for Evaporator

$$P_{th(EV)} = P_{th(1\&2)} + P_{th(3)} + P_{th(4)}$$

$$= 38.7 + 38.2 + 39.4 = 116.3 \text{ MW}$$

$$A_{tot} = \text{Total heat transfer area} = 529.5 \text{ m}^2 \quad (\text{TABLE 4-3})$$

$$L_{(EV)} = L_{(1\&2)} + L_{(3)} + L_{(4)}$$

$$= 3.1 + 1.3 + 3.7 = 8.1 \text{ m}$$

$$A_{(EV)} = \pi D_{to} L_{(EV)} N_i$$

$$= \pi (0.0159) (8.1) (757) = 306.3 \text{ m}^2$$

$$SF_{(EV)} = \frac{529.5}{306.3} = 1.73$$

A.1.12 Summary for the Evaporator

$$P_{th(EV)} = 116.3 \text{ MW}$$

$$L_{(EV)} = 8.1 \text{ m}$$

$$\Delta P_{p(EV)} = 0.005 \text{ MPa}$$

$$K_{pminor(EV)} = 56.6$$

$$\Delta P_{u(EV)} = 0.144 \text{ MPa}$$

$$K_{uminor(EV)} = 66.1$$

$$SF_{(EV)} = 1.73$$

A.2 Check Case for the Sodium/Water Superheater

A.2.1 Choose the Initial Parameters

Water side

$$\dot{m}_{w(SH)} = 0.5 \times 10^6 \text{ kg/hr}$$

Sodium side

$$\dot{m}_{p(SH)} = 6.1 \times 10^6 \text{ kg/hr}$$

Superheater geometry

$$\text{Pitch} = 0.031 \text{ m}, N_t = 757, D_{to} = 0.0159 \text{ m}$$

$$D_{ti} = 0.01036 \text{ m}$$

A.2.2 Tube Bundle and Tube Geometry

$$\frac{1}{4} \pi D_{TB}^2 = 0.866 (\text{Pitch})^2 N_t$$

$$D_{TB} = [4(0.866)(0.031)^2(757)/\pi]^{1/2} = 0.896 \text{ m}$$

$$\begin{aligned} P/D &= \frac{\text{Pitch}}{D_{to}} \\ &= \frac{0.031}{0.0159} = 1.95 \end{aligned}$$

$$\begin{aligned} D_h &= \frac{4 \text{ Area}}{P_{wet}} \\ &= \frac{4 \left[\frac{1}{2} (0.866) (31)^2 - \frac{1}{2} (\frac{1}{2} \pi) (15.9)^2 \right]}{\frac{1}{2} \pi (15.9)} = 0.0507 \text{ m} \end{aligned}$$

A.2.3 Mass Flow Rate

Water side

$$\begin{aligned}
 G_{w(SH)} &= \frac{\dot{m}_{w(SH)}}{\frac{1}{4} \pi D_u^2 N_t} \\
 &= \frac{0.5 \times 10^6}{3600} \times \frac{1}{\frac{1}{4} \pi (0.01036)^2 (757)} \\
 &= 2176.5 \text{ kg/m}^2 \cdot \text{s}
 \end{aligned}$$

Sodium side

$$\begin{aligned}
 G_{p(SH)} &= \frac{\dot{m}_{p(SH)}}{\frac{1}{4} \pi [D_{IB}^2 - N_t D_{IO}^2]} \\
 &= \frac{6.1 \times 10^6}{3600} \times \frac{1}{\frac{1}{4} \pi [(0.896)^2 - 757(0.0159)^2]} \\
 &= 3532.4 \text{ kg/m}^2 \cdot \text{s}
 \end{aligned}$$

A.2.4 Curve Fit for the Sodium Properties

Use the third order curve fit for the sodium properties

$$C_{p_p} = 1855 - 1.849T + 1.787 \times 10^{-3} T^2 - 4.902 \times 10^{-7} T^3$$

$$\begin{aligned}
 Pr_p &= 4.658 \times 10^{-2} - 1.638 \times 10^{-4} T + 2.179 \times 10^{-7} T^2 \\
 &\quad - 9.853 \times 10^{-11} T^3
 \end{aligned}$$

$$\begin{aligned}
 \mu_p &= 2.858 \times 10^{-3} - 9.767 \times 10^{-6} T + 1.257 \times 10^{-8} T^2 \\
 &\quad - 5.588 \times 10^{-12} T^3
 \end{aligned}$$

$$K_p = 100.1 - 2.836 \times 10^{-3} T - 1.095 \times 10^{-4} T^2 + 6.863 \times 10^{-8} T^3$$

$$1/\nu_p = 1246 - 1.507T + 2.251 \times 10^{-3} T^2 - 1.283 \times 10^{-6} T^3$$

A.2.5 Regime 5 (Single Phase Vapor Boiling)

A.2.5.1 Input temperature and pressure

Water side

$$T_{si(5)} = 604.15 \text{ K}, T_{so(5)} = 758.15 \text{ K}, P_{si(5)} = 12.1 \text{ MPa}$$

Sodium side

$$T_{po(5)} = 736.15 \text{ K}$$

A.2.5.2 From the steam table

$$i_{si(5)} = 2718.44 \text{ kJ/kg}$$

$$i_{so(5)} = 3304.58 \text{ kJ/kg}$$

A.2.5.3 First Law of Thermodynamics

$$\begin{aligned} P_{th(5)} &= m_{s(SH)} (i_{si(5)} - i_{so(5)}) \\ &= \frac{0.5 \times 10^6}{3600} (3304.58 - 2718.44) \times 10^3 = 81.4 \text{ MW} \end{aligned}$$

$$\begin{aligned} \bar{T}_{s(5)} &= \frac{1}{2} (T_{so(5)} + T_{si(5)}) \\ &= \frac{1}{2} (604.15 + 758.15) = 681.15 \text{ K} \end{aligned}$$

A.2.5.4 First Law of Thermodynamics

$$C_{pP(5)}(T_{pi(5)}) = 1266.7 \text{ J/kg.K}$$

$$P_{th(5)} = m_{p(SH)} C_{pP(5)} (T_{pi(5)} - T_{po(5)})$$

$$81.4 \times 10^6 = \frac{6.1 \times 10^6}{3600} (1266.7) (T_{pi(5)} - 736)$$

$$T_{pi(5)} = 774.08 \text{ K}$$

$$\begin{aligned} \bar{T}_{p(5)} &= \frac{1}{2} (T_{pi(5)} + T_{po(5)}) \\ &= \frac{1}{2} (774.08 + 736.15) = 755.11 \text{ K} \end{aligned}$$

A.2.5.5 Water properties

$$Nu_{film} = 0.0073 Re_{film}^{0.886} Pr_{film}^{0.61} \quad (CRBR \text{ Final Report})$$

where all properties are evaluated at T_{film}

$$\begin{aligned} T_{film} &= \frac{1}{2} (T_{wall} + T_{bulk}) \\ &\approx \frac{1}{2} (\frac{1}{2} (\bar{T}_{p(s)} + \bar{T}_{u(s)}) + \bar{T}_{u(s)}) \\ &= \frac{1}{2} (\frac{1}{2} (755.11 + 681.15) + 681.15) = 699.64 \text{ K} \end{aligned}$$

$$Re_{s(film(s))} = \frac{G_{u(SH)} D_{li}}{\mu_{u(s)}} = \frac{2176.5(0.01036)}{0.0000258} = 8.74 \times 10^5$$

$$Pr_{s(film(s))} = 1.1333$$

$$Nu_{s(film(s))} = 0.0073 (8.74 \times 10^5)^{0.886} (1.1333)^{0.61} = 1448$$

$$h_{u(s)} = \frac{Nu_{s(film(s))} K_{s(film(s))}}{D_{li}} = \frac{1448(0.07275)}{0.01036} = 10165$$

A.2.5.6 Wall properties

$$\bar{T}_{w(s)} \approx \frac{1}{2} (\bar{T}_{p(s)} + \bar{T}_{u(s)}) = \frac{1}{2} (755.11 + 681.15) = 718.13 \text{ K}$$

$$\begin{aligned} K_{w(s)}(\bar{T}_{w(s)}) &= 49.37 - 0.01713 \bar{T}_{w(s)} \quad (CRBRP \text{ Final Report}) \\ &= 37.07 \text{ W/m.K} \end{aligned}$$

A.2.5.7 Sodium properties

$$Pr_{p(s)}(\bar{T}_{p(s)}) = 0.004715$$

$$\mu_{p(s)}(\bar{T}_{p(s)}) = 0.000244 \text{ kg/m.s}$$

$$K_{p(s)}(\bar{T}_{p(s)}) = 65.0715$$

$$Re_{p(s)} = \frac{G_{p(SH)} D_h}{\mu_{p(s)}} = \frac{3528.5(0.0507)}{0.00244} = 734038$$

with $P/D = 1.95$

$$\begin{aligned} Nu_{p(s)} &= 12.34 + 0.0554 (Re_{p(s)} Pr_{p(s)})^{0.7532} \\ &= 12.34 + 0.0554 [(733884) (0.004715)]^{0.7532} = 37.99 \end{aligned}$$

$$h_{p(5)} = \frac{Nu_{p(5)} K_{p(5)}}{D_h} = \frac{37.99(65.0715)}{0.0507} = 48721 \text{ W/m}^2 \cdot \text{K}$$

A.2.5.8 Thermal resistance

$$R_{u(5)} = \frac{1}{\pi D_{ii} N_t L_{(5)} h_{u(5)}} \\ = \frac{1}{\pi (0.01036) (757) (10165) L_{(5)}} = \frac{3.99 \times 10^{-6}}{L_{(5)}} \text{ K/W}$$

$$R_{p(5)} = \frac{1}{\pi D_{to} N_t L_{(5)} h_{p(5)}} \\ = \frac{1}{\pi (0.0159) (757) (48721) L_{(5)}} = \frac{5.43 \times 10^{-7}}{L_{(5)}} \text{ K/W}$$

$$R_{w(5)} = \frac{\ln(D_{to} / D_{ii})}{2 \pi N_t L_{(5)} K_{w(5)}} \\ = \frac{\ln(0.0159/0.01036)}{2\pi (757) (37.07) L_{(5)}} = \frac{2.43 \times 10^{-6}}{L_{(5)}} \text{ K/W}$$

$$R_{(5)} = R_{u(5)} + R_{p(5)} + R_{w(5)} \\ = \frac{(3.99 + 0.543 + 2.43) \times 10^{-6}}{L_{(5)}} = \frac{6.97 \times 10^{-6}}{L_{(5)}} \text{ K/W}$$

where $L_{(5)}$ - length of the tube in regime 5

A.2.5 Log Mean Temperature Difference Method

$$\Delta T_{LM(5)} = \frac{(T_{pi(5)} - T_{ao(5)}) - (T_{po(5)} - T_{si(5)})}{\ln[(T_{pi(5)} - T_{ao(5)}) / (T_{po(5)} - T_{si(5)})]} \\ = \frac{(773.93 - 758.15) - (736 - 604.15)}{\ln[(773.93 - 758.15) / (736 - 604.15)]} = 54.67 \text{ K}$$

$$P_{th(5)} = \frac{\Delta T_{LM(5)}}{R_{(5)}} = 54.67 \times \frac{L_{(5)}}{6.97 \times 10^{-6}} = 81.4 \times 10^6$$

$$L_{(5)} = 10.3 \text{ m}$$

A.2.6 Pressure Drop on the Regime 5

A.2.6.1 Pressure drop on the water side

$$\begin{aligned} f_{ow(5)} &= 0.046 Re_{w(5)}^{-0.2} \\ &= 0.046 (8.74 \times 10^5)^{-0.2} = 0.002982 \end{aligned}$$

$$v_{wi(5)} = 0.015001 \text{ m}^3/\text{kg}, \quad v_{wo(5)} = 0.025996 \text{ m}^3/\text{kg}$$

$$\begin{aligned} v_{wm(5)} &= \frac{1}{2} (v_{wi(5)} + v_{wo(5)}) \\ &= \frac{1}{2} (0.015001 + 0.025996) = 0.020499 \text{ m}^3/\text{kg} \end{aligned}$$

$$\begin{aligned} \Delta P_{f(5)} &= \frac{2 f_{ow(5)} v_{wm(5)} G_{w(SH)}^2 L_{(5)}}{D_{ii}} \\ &= \frac{2 (0.002982) (0.020499) (2176.5)^2 (10.3)}{0.01036} \\ &= 0.58 \text{ MPa} \end{aligned}$$

$$\begin{aligned} \Delta P_{sa(5)} &= (v_{wo(5)} - v_{wi(5)}) G_{w(SH)}^2 \\ &= (0.025996 - 0.015001) (2176.5)^2 = 0.052 \text{ MPa} \end{aligned}$$

$$\Delta P_{sg(5)} = \frac{g L_{(5)}}{v_{wm(5)}} = \frac{9.81 (10.4)}{0.020449} = 0.005 \text{ MPa}$$

$$\begin{aligned} \Delta P_{s(5)} &= \Delta P_{f(5)} + \Delta P_{sa(5)} + \Delta P_{sg(5)} \\ &= 0.58 + 0.052 + 0.005 = 0.64 \text{ MPa} \end{aligned}$$

$$\Delta P_{w(SH)} = \Delta P_{s(5)} = 0.64 \text{ MPa}$$

$$\Delta P_{sCRBR(SH)} = 1.9 \text{ MPa}$$

$$\begin{aligned} \Delta P_{sminor(SH)} &= \Delta P_{sCRBR(SH)} - \Delta P_{w(SH)} \\ &= 1.9 - 0.64 = 1.26 \text{ MPa} \end{aligned}$$

$$v_{wm} = v_{wm(5)} = 0.020499 \text{ m}^3/\text{kg}$$

$$\Delta P_{\text{minor(SH)}} = \frac{1}{2} G_{\text{p(SH)}}^2 v_{\text{um}} K_{\text{minor(SH)}}$$

$$1.26 \times 10^6 = \frac{1}{2} (2176.5)^2 (0.020499) K_{\text{minor(SH)}}$$

$$K_{\text{minor(SH)}} = 26.0$$

A.2.6.2 Pressure drop on the sodium side

$$f_{\text{op(5)}} = 0.046 \text{ Re}_{\text{p(5)}}^{-0.2}$$

$$= 0.046 (734038)^{-0.2} = 0.003088$$

$$1/v_{\text{p(5)}} \bar{T}_{\text{p(5)}} = 836.7 \text{ kg/m}^3$$

$$v_{\text{p(5)}} \bar{T}_{\text{p(5)}} = 0.001195 \text{ m}^3/\text{kg}$$

$$\begin{aligned} \Delta P_{\text{p(5)}} &= \frac{2 f_{\text{op(5)}} v_{\text{p(5)}} G_{\text{p(SH)}}^2 L_{\text{(5)}}}{D_h} \\ &= \frac{2 (0.003088) (0.001195) (3532.4)^2 (10.3)}{0.0507} \\ &= 0.019 \text{ MPa} \end{aligned}$$

$$\Delta P_{\text{ps(5)}} = (v_{\text{ps(5)}} - v_{\text{p(5)}}) G_{\text{p(SH)}}^2 \approx 0 \text{ MPa}$$

$$\Delta P_{\text{pg(5)}} = 0 \quad (\text{because sodium is in a closed loop})$$

$$\begin{aligned} \Delta P_{\text{p(5)}} &= \Delta P_{\text{pl(5)}} + \Delta P_{\text{ps(5)}} + \Delta P_{\text{pg(5)}} \\ &= 0.019 + 0 + 0 = 0.019 \text{ MPa} \end{aligned}$$

$$\Delta P_{\text{p(SH)}} = \Delta P_{\text{p(5)}} = 0.019 \text{ MPa}$$

$$\Delta P_{\text{pCRBR(SH)}} = 0.42 \text{ MPa}$$

$$\begin{aligned} \Delta P_{\text{pminor(SH)}} &= \Delta P_{\text{pCRBR(SH)}} - \Delta P_{\text{p(SH)}} \\ &= 0.42 - 0.019 = 0.401 \text{ MPa} \end{aligned}$$

$$v_{\text{pm}} = v_{\text{p(5)}} = 0.001195 \text{ m}^3/\text{kg}$$

$$\Delta P_{\text{pminor(SH)}} = \frac{1}{2} G_{\text{p(SH)}}^2 v_{\text{pm}} K_{\text{pminor(SH)}}$$

$$4.01 \times 10^5 = \frac{1}{2} (3532.4)^2 (0.001195) K_{\text{pminor(SH)}}$$

$$K_{\text{pminor(SH)}} = 53.7$$

A.2.7 Safety Factor for Superheater

$$P_{th(SH)} = P_{th(5)} = 81.4 \text{ MW}$$

$$A_{tot} = \text{Total heat transfer area} = 529.5 \text{ m}^2$$

(TABLE 4-3)

$$L_{(SH)} = L_{(5)} = 10.3 \text{ m}$$

$$\begin{aligned} A_{(SH)} &= \pi D_{to} L_{(SH)} N_{tube} \\ &= \pi (0.0159) (10.3) (757) = 389.5 \text{ m}^2 \end{aligned}$$

$$SF_{(SH)} = \frac{529.5}{389.5} = 1.36$$

A.2.8 Summary of the Superheater

$$P_{th(SH)} = 81.4 \text{ MW}$$

$$L_{(SH)} = 10.3 \text{ m}$$

$$\Delta P_{p(SH)} = 0.019 \text{ MPa}$$

$$K_{minor} = 53.7$$

$$\Delta P_{s(SH)} = 0.64 \text{ MPa}$$

$$K_{minor} = 26.0$$

$$SF_{(SH)} = 1.36$$

References

Collier, J.G. (1981), Convective Boiling and Condensation, 2nd Edition, McGraw-Hill press, UK, pp. 42-44, 221-230.

Fraas, A.P. (1989), Heat Exchanger Design, Wiley-Interscience Publication, New York, pp. 303.

Fricker, H.W. (1976), "Design and Manufacturing Experience for the German Thorium High-Temperature Reactor 300-MW(e) Steam Generator," Nuclear Technology, Vol. 28.

Groeneveld, D.C. (1973), "Post-Dryout Heat Transfer at Reactor Operating Condition," AECL-4513, National Topical Meeting on Water Reactor Safety, ANS, Salt Lake City, Utah.

Harty, R.B. (1974), "Modular Steam Generator Final Project Report," TR-097-330-010, Atomics International Division, Rockwell International.

Lillie, A.F. and Bashford, A.L. (1978), "The Design, Manufacturing, and Test Status of the CRBRP Hockey Stick Steam Generator," GEFR-SP-042, Joint U.S./Japan LMFBR Steam Generator Seminar Conference.

Martinelli, R.C. and Nelson, D.B., (1947), "Predication of Pressure Drop During Forced-Circulation Boiling of Water," Paper presented at the Annual Meeting, Atlantic City, N.J. of ASME.

Moody, E. and Gabler, M.J. (1978), "Heat Transfer with Hockey-Stick Steam Generator," GEFR-SP-070, Joint U.S./Japan LMFBR Steam Generator Conference.

Rosenthal, M.W., Haubenreich, P.N., and Briggs, R.B. (1972) "The Development Status of MOLTEN-SALT BREEDER REACTORS," ORNL-4812, Oak Ridge National Laboratory.

Shah, M. M. (1976), "A New Correlation for Heat Transfer During Boiling Flow Through Pipes", ASHRAE Trans, Vol 82 Part 2, pp. 66-86, Paper presented at the Annual Meeting, Seattle, Washington.

Singh, J. (1985), Heat Transfer Fluids and Systems for Process and Energy Applications, Marcel Dekker, Inc., New York, pp. 224.

Steam Table in SI-Units (1984), Springer-Verlag, New York, Table 1.

Tajbakhsh, A.E. (1984), "Thermal Hydraulic Design Study of Helium-Helium Heat Exchangers and Once-Through Steam Generator," M.S. Thesis, University of California, Davis.

Thom, J.R.S. (1964), "Predication of Pressure Drop During Forced Circulation Boiling of Water," Int. J. Heat Mass Transfer, Vol 7, pp. 709-724.