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1.0 INTRODUCTION

The results obtained during FY 1977 and 1978 in assessing the relative merits of different steam generator designs for a commercial LMFBR plant^(2,3,4,5) are presented in this report. The suppliers of the four steam generators studied and their principal design features are:

<u>Supplier</u>	<u>Tube</u>	<u>Shell</u>
Atomics International (AI)	Hockey, Single Wall (SWT)	Hockey Stick
Babcock & Wilcox (B-W)	Helical, Single Wall (SWT)	Straight
Westinghouse (W)	Straight, Double Wall (DWT)	Straight, Bellows
Foster-Wheeler (F-W)	Straight, Double Wall (DWT)	Straight, Bellows

These units were first resized to a consistent set of design and operating conditions for a commercial Benson, Sulzer, and saturated steam cycle plant (see Table 1.1). The attributes of these four types of designs were then reviewed with respect to the following considerations for each steam cycle:

- Design Features
- Thermal/Hydraulic Characteristics
- Structural Margins
- Scale-Up
- Manufacturing
- Maintenance/Inspection
- Reliability/Availability
- Cost

The major effort during FY 1977 was concentrated on a comparative evaluation of single-wall tube (SWT) versus double-wall tube (DWT) designs relative to their use in Benson, Sulzer, and saturated steam cycles. The effort during FY 1978 concentrated on a comparative evaluation of each steam generator relative to their design features, cost,

TABLE 1.1

SELECTED OPERATING PARAMETERS* FOR 1000 MWe LMFBR

	Benson	Sulzer	Saturated
Reactor Outlet Temperature ($^{\circ}\text{F}$)	930	930	875
Reactor Inlet Temperature ($^{\circ}\text{F}$)	650	650	595
Primary ΔT ($^{\circ}\text{F}$)	280	280	280
Primary Pump Head (ft)	500	500	500
Steam Generator Inlet Na ($^{\circ}\text{F}$)	900	900	815
Steam Generator Outlet Na ($^{\circ}\text{F}$)	600	600	550
Intermediate ΔT ($^{\circ}\text{F}$)	300	300	265
Feedwater Temperature ($^{\circ}\text{F}$)	470	470	424
Recirculation Ratio	None	1.05	6.0
Turbine Inlet Conditions			
Temperature ($^{\circ}\text{F}$)	850	850	546
Pressure (psia)	2,215	2,215	1,015
SG to Turbine Pressure Drop (psia)	75	75	25
Steam Cycle Efficiency (Gross)	0.391	0.391	0.344
Overall Thermal Efficiency (Net)	0.362	0.362	0.318
Core Power (MWt)	2,585	2,585	2,888
Generation (MWe)			
Gross	1,025	1,025	1,046
Net	949	949	949
Heat Rejection Load (MWt)	1,636	1,636	1,939

*Based on three loops per plant and three steam generator modules per loop

scale-up, potential failure modes, and their ability to accommodate transients that will be imposed on them when installed in the three steam cycle type plants. The more detailed comparison of design features was accomplished using the additional information provided in reports^(6,7) received since September 1977.

2.0 CONCLUSIONS AND RECOMMENDATIONS

Each of the following areas are covered in detail in the body of the report, and only a summary of each consideration is presented here.

2.1 DESIGN FEATURES

The comparison of design features shown in Figures 3.2 through 3.18 and Table 3.1 revealed the following observations:

- 1) Tubesheets - The B&W design is the simplest as it has the smallest size. The Westinghouse design is the most complex as it features two large upper tubesheets to provide a means for leak detection between the double-walled tubes. The F-W DWT lower tubesheet is also complex and the leak detection design appears impractical from a fabrication point of view.
- 2) Bellows - The AI and B&W designs do not require shell bellows to take care of tube to shell expansion differences. The shell bellows required in the Westinghouse and F-W designs are questionable both from structural and fabrication points of view.
- 3) Number of Welds - The number and length of welds is much less for the single-walled tube Babcock & Wilcox and AI designs than for the double-walled tube Westinghouse and Foster-Wheeler designs.
- 4) Active Surface Area -
 - a) Saturated Cycle. The active surface area of the B&W design is the smallest as it employs single-wall tubes with peak heat flux of up to 350,000 Btu/hr-ft² for the saturated cycle. Since the B&W design cannot readily employ a protected tube, the feasibility of employing this design for the saturated cycle is questionable. AI's design exhibits the next smaller

area as it also employs single-wall steam tubes covered by 304 stainless steel protector tubes over more than half of the active tube length to limit the peak heat flux to $<150,000 \text{ Btu/hr-ft}^2$. Westinghouse and F-W designs have larger surface areas due to their double-wall tubes. The F-W design is the largest of the four types since it has a short length of protector tubes over the double-wall tubes at the sodium inlet to limit heat flux. The W design require protector tubes if they use prestressed DWT's. Peak heat flux for both Westinghouse and F-W designs is comparable to that of AI.

- b) Benson Cycle. The active surface areas are much less for the AI and B&W SWT designs than for the Westinghouse and F-W DWT designs. Protector tubes are unnecessary as heat peak flux is low in the Benson Cycle.
- c) Sulzer Cycle. The evaporator active surface area is less for the B&W than for the AI design. The superheater active surface area is comparable for both the AI and the B&W design. The required active surface area is much greater for the Westinghouse and F-W DWT evaporator and superheaters than for the AI and B&W SWT designs. Protector tubes are unnecessary as peak heat flux is low in the Sulzer cycle.

Note: A feedwater temperature of 470°F was used for both the Benson and Sulzer cycles. This temperature was based on an optimization updated for this study. It appears that a lower feedwater temperature would significantly decrease the heat transfer area for these cycles for all concepts and would allow the areas of the DWT designs to be reduced to a practical size.

- 5) Sodium Flow Distribution and Tube Support - The B&W design has the most complicated sodium flow distribution system. However, the present design has been greatly improved over that presented in July 1977. The AI design is the least complicated since all tube support plates are identical. The W and F-W designs require a more complicated sodium flow distribution system because of their sensitivity to bundle flow maldistribution.

A comparison of the four designs, after resizing to a consistent set of operating conditions, is presented in Table 3.3. This was accomplished by normalizing to the same operating conditions. An examination of these data indicate each supplier used quite different assumptions in respect to selection of design factors, such as overall heat transfer coefficient, tube diameter, and active length of tube that affect size and cost. The funding level for this program did not permit optimization of each suppliers design to the same design margins.

2.2 THERMAL/STRUCTURAL

Thermal/structural analysis effort identified the W upper tubesheet as having the least capability to accept the temperature transients imposed during 40 years of plant operation. The large diameter ring surrounding the tubesheet increases the radial stress load both for steady state as well as reactor trip temperature transients. Analysis shows that the W upper tubesheet is satisfactory for the saturated cycle if thermal shielding is provided on the sodium side. Although an analysis was not performed, an assessment of its thermal stress resistance indicates its use in a Benson or Sulzer steam cycle plant is suspect even with thermal shielding provided on the sodium side. All the upper tubesheets in the other designs except possibly the helical tube design appear to be satisfactory for use in all three steam cycles. A cursory analysis of the helical upper tubesheet indicates it will undergo considerably higher stresses than the hockey stick design for the same Benson cycle reactor trip transient in spite of its much

smaller size. The reasons for this unexpected condition (see Section 4.1.1) are that the helical tubesheet has 1) a gas space (not a large heat capacity stagnant sodium region) to attenuate the rate of change of the steam temperature during a trip transient, and 2) a nonsymmetrical attachment to the steamhead and the shell. An extensive analysis effort and possibly redesign may be required to show the helical upper tube-sheet design will be satisfactory for superheated cycle plants.

An additional area of concern in the W design is the lack of thermal shielding for the sodium nozzles. It is the only design that does not show thermal shields for sodium nozzles.

The thermal/hydraulic analysis made on the Westinghouse and Foster-Wheeler double-wall, straight tube designs showed excessive tube-to-tube temperature variations can occur in all three cycles from both tube tolerances on the water side and maldistribution of flow on the sodium side.

The results show that designs not exhibiting individual tube-to-tube flexibility, i.e., the straight double-wall tube F-W and Westinghouse designs, have structural problems that are difficult to mitigate. These problems are tube buckling from excessive tube-to-tube expansion differences and excessive tube weld stress induced by flexure of the bellows. It is doubtful that design tolerances can be held sufficiently close to eliminate these effects. The following conclusions were drawn from the structural evaluation:

- 1) Bellows - Excessive bellows flexure under seismic and inertial loads induces high stresses locally on the tube bundle and tube welds. The external radial support for the bellows alleviates excessive flexure and reduces mechanical stresses in the steam tubes and tube welds, however, these will require tight clearances which could interfere with the bellows movement during operation or welding operations during fabrication.

- 2) Differential Steam Tube-to-Steam Tube Expansion - The DWT steam tubes in the active region are critical in the column buckling mode with excessive sodium flow maldistribution. In the W design, the single tubes between the two upper tubesheets are critical in the column buckling mode under steam blowdown conditions. The double-wall tube welds are also critical in shear with excessive sodium maldistribution and the single-tube welds at the upper tubesheet in the W design are critical in shear under steam blowdown conditions. The potential delta temperatures allowed are illustrated in Figure 4.5.
- 3) Seismic Loads - No items are critical for seismic loads acting alone, however, steam tubes and tube welds become critical under combined seismic and thermal loads. Seismic loads have only a minimal effect of tube welds.
- 4) DWT Prestress Relaxation - The mechanically bonded DWT required for sodium leak detection in the W and F-W design are subject to loss of prestress with time when used in Benson cycle and Sulzer cycle superheater temperature conditions.

2.3 SCALE-UP PROBLEMS

Assessment of scale-up problems for the four steam generator designs show the hockey stick design has the firmest U.S. development background. There is no significant experience in the U.S. with the helical tube design. The planned 70 MWt SCTI test of the B&W model will be the first significant test in the U.S.. The foreign experience is in the 10 to 53 MWt range. The applicability of the EBR-II experience to the scale-up of the DWT designs appears to be of limited value because of the great difference in size and many differences in design details. In addition, the EBR-II experience with relaxation of the prestress in double wall tubes indicate design changes are required for this concept.

In summary, it can be stated 1) the hockey stick design has the strongest U.S. development for scale-up to a commercial size design, 2) the helical design has the best test experience based on foreign development, however, exchange agreements are required to make this experience available, and 3) scale-up of the DWT designs must be based on future tests to be made on the planned 70-MWt test model.

2.4 MANUFACTURING

The AI hockey stick design concept uses current fabrication technology which does not change with scale-up of the steam generator. The flexibility of the AI tube-to-tubesheet assembly for scale-up gives it a big advantage compared to the fabrication of W and F-W tubesheets.

Because of the complex and massive tooling associated with the B&W helical tube design and hand fitup required for assembly of tube supports, this concept is penalized for all production rates. The use of smaller tubesheets reduces handling problems but requires additional fixturing problems to position the four individual tubesheets at each end. A large special machine must be engineered and developed to rotate the helical tube configurations as the tubes are wound onto the tube bundle.

The Westinghouse and Foster-Wheeler straight DWT design concepts hinge upon the development of the duplex tubes and the required large bellows. Neither is a new idea, as each has been used in much smaller units. The larger units require more complex fitup tooling and make the problem of obtaining crack-free bellows-to-shell welds much more difficult especially if dissimilar metals are required. The 0.50-in. thick bellows will be very difficult and expensive to fabricate. Single-ply bellows with thicknesses in excess of approximately 0.14 in. have yet to be formed by commercial vendors. The drilling of the leak detection holes in the lower tubesheet of the F-W design presents several problems that will be difficult to overcome.

2.5 INSPECTION/MAINTENANCE

Initial detection of a tube leak can be accomplished equally well with a single-wall or double-walled tube. However, subsequent identification, location, and repair of the tube leak will be much more difficult with a double-walled tube. Outer tube failures are especially hard to locate because there is no access for monitoring individual tubes. Repair of a leaking single-wall tube can be readily accomplished after location by the use of expansive plugs and welding at the upper and lower tubesheets to isolate the tube. The double-wall design cannot use a simple expansive plug because the annular grooves between tubes make sealing impossible. It also requires difficult to perform operations, e.g., plugging at the inner tubesheet as well as cutting out section of the double wall tube in the sodium region to prevent excessive tube buckling and tube weld stresses.

The shell bellows complicates shell inspection and introduces unknown inservice inspection (ISI) requirements. The bellows closure welds cannot be inspected radiographically. The replacement of a failed bellows cannot be accomplished without disassembling the shell. In-place repair of a leaking bellows does not appear possible because of requalification requirements.

2.6 RELIABILITY/AVAILABILITY

An availability analysis based on tube weld footage shows that the AI design has a distinct advantage over the F-W and W designs and a slight advantage over the B&W design. The results from ratings based on 21 failure modes show the hockey stick design has the least potential for failure when installed in plants employing any of the three steam cycles. The B&W design is next best for the Benson and Sulzer cycle but poor for the saturated cycle.

2.7 COST

The cost evaluation shows the price of the AI hockey stick design is significantly lower than any other design for all three steam cycles and the B&W helical design is the next best choice for Benson and Sulzer cycle. The evaluation also shows the shell wall thickness has a significant effect on cost. The major design features that contribute to high cost are weight, double wall tubes, bellows, and forgings (rings, dished headers, and dished shell caps). Low cost items are straight parts (shells, supports, and baffles) made from rolled or flat plates.

2.8 DESIGN SELECTION

A rating of the four designs was carried out based on a comparison of design simplicity, suitability for use with all three steam cycles, thermal/hydraulic/structural characteristics, scalability, manufacturing problems, ease of inspection and maintenance, reliability/available and cost. This rating listed in Table 2.1 indicated the order of preference to be hockey stick SWT first for all three steam cycles, the helical tube SWT second for the Benson and Sulzer cycles, and the F-W DWT second for the saturated cycle.

TABLE 2.1
ORDER OF DESIGN SELECTION

Choice	<u>Steam Cycle</u>		
	Benson	Sulzer	Saturated
1st	AI	AI	AI
2nd	B&W	B&W	F-W
3rd	F-W	<u>W</u>	<u>W</u>
4th	<u>W</u>	F-W	B&W

The hockey stick configuration is considered to provide the simplest and lowest cost design that is satisfactory for all three steam cycle plants. It is a test proven scalable design which has noncritical tube-to-tube thermal expansion characteristics, is easy to inspect and maintain, is estimated to have the highest reliability/availability and the least manufacturing cost impact from design change and scale-up considerations. The smaller size of the tube inlet orifices required for the Benson and Sulzer cycles, for this unit is not considered to be a problem, based on experience from previous testing.

The advantages of the helical tube design are that it (1) is similar to those selected for the Phenix and Monju plants and can, therefore, potentially exchange R&D information, (2) is satisfactory for the Benson and Sulzer steam cycles (but may require redesign of the upper tubesheet attachment to the shell nozzle), (3) has the largest tube orifices for Benson and Sulzer cycles, (4) has noncritical tube-to-tube thermal expansion characteristics, (5) is fairly easy to inspect and maintain, and (6) is estimated to have high reliability/availability. Its major disadvantages are (1) very complex design, (2) the inability to incorporate shock tubes make it appear unsatisfactory for use in the saturated steam cycle, (3) has the highest manufacturing cost impact from design change or scale-up considerations, (4) is 1.5 to 1.6 times the cost of the hockey stock design for the Benson and Sulzer cycles, and (5) the plugging of a single tube has the greatest effect on heat transfer area.

The F-W and W DWT designs are similar in design but employ many more and much longer tubes than the unit employed in EBR-II. The F-W design is theoretically satisfactory for all three steam cycles, however, the large outer ring on the sodium side of the W upper tubesheet tube places doubt on its suitability for the Benson and Sulzer cycles. The disadvantages of the two DWT designs is they (1) require very large scale-up from EBR-II (there are many design differences between the

commercial LMFBR, and EBR-II designs), (2) require close tolerances on baffle-to-shell, bellow-to-bellows support dimensions, (3) may require high bellows fabrication costs, (4) are estimated to have lower reliability/availability than single-wall tube designs, (5) require complex tube plugging procedures, (6) are 1.9 to 2.4 times more costly for the Benson and Sulzer steam cycles and 1.2 to 1.5 times more costly for the saturated steam cycle than the hockey stick design, and (7) have high manufacturing cost impact from design change or scale-up considerations. The problems associated with reinforced bellows fabrication and load transfer capability and the leak detection methods required for these designs make their feasibility for all three steam cycles questionable.

3.0 STEAM GENERATOR DESIGNS COMPARED

3.1 DESIGN FEATURES

There are four different steam generator designs currently being considered for use in commercial LMFBR's. They are the single-wall AI hockey stick and Babcock & Wilcox helical coil tube designs and the double wall (duplex), straight tube units proposed by Westinghouse and Foster-Wheeler. All four designs employ a shell and tube configuration with water-steam mixtures flowing upward through the tubes and sodium flowing downward on the shell side. Figure 3.1 presents schematic illustrations of these four basic configurations. Each concept has its own distinctive features. The Westinghouse and Foster-Wheeler designs are assumed to contain an integral steam drum on top of each steam generator for the saturated cycle operation.

Shell bellows are not required for the AI hockey stick or B&W helical SWT designs as they both provide adequate flexibility to take care of tube-to-shell and tube-to-tube differential expansion. The shell bellows are provided in the lower end of the shell of the Westinghouse and Foster-Wheeler straight DWT designs to accommodate the differential thermal movements between the shell and tubes. The bellows do not provide for tube-to-tube expansion differences; therefore, sodium and water side flow distribution is much more critical for DWT than for SWT designs. Because of the layup and tube-to-tubesheet welding sequence required, DWT steam generators can best be manufactured in a straight tube configuration. The flexible bellows employed must be externally reinforced on their outer surface and must have a substantial wall thickness of approximately 1/2-in. and generous, deep convolutions. The two DWT bellows designs appear to be questionable both from structural and fabrication points of view.

FOUR STEAM GENERATOR CONFIGURATIONS EVALUATED

*OTHER DESIGNS EMPLOY SEPARATE STEAM DRUM FOR THE SAT. CYCLE

Figure 3.1

3.1.1 Single-Wall Tube Designs

The offset leg at the top of the AI SWT design, Figure 3.2, provides tube-to-shell and individual tube-to-tube compliance. This allows large thermal transients and tube-to-tube differences to be accommodated while safe stress margins are maintained in the tubes. It features a very simple design that is easily scaled up with minimum impact on development costs. It employs easily inspected butt welded tube-to-tubesheet welds (two per tube), vibration suppressors, drilled tube spacers, a shroud to provide good flow distribution and thermal shields to protect the shell and sodium nozzles from thermal transients. A large stagnant sodium volume is also provided above the sodium inlet to reduce sodium/steam transients at the upper tubesheets. The structural design of the key areas (upper tubesheet and sodium nozzles) have been proven by structural analysis.

The Babcock & Wilcox SWT helical coil concept is shown in Figures 3.3 through 3.9. This is the design presented in Reference 7. Information on more recent design changes could not be obtained in time for inclusion in this report. It employs long, large-diameter tubes to reduce the required number of tubes and tube welds. The small number of tubes reduces the size and magnitude of structural discontinuities at the relatively small tubesheets. Many complex shaped helical tubes with pigtails at each end are required. The pigtails are necessary to mate the tubes to four water inlet and four steam outlet nozzles. The resulting tubes, up to 535 ft in length, dictate the need for up to 15 tube-to-tube welds along the length of each tube. Five different types of tube supports are required to support the tube pigtails, space the helical tubes, and support the tube bundle. The tube bundle is supported from crossbeams attached to both the inner and outer shrouds as shown in Figures 3.5 and 3.6. One sodium inlet and one sodium outlet nozzle are used. The sodium inlet flow distributor (6 radial arms as shown in Figures 3.3 and 3.7) is improved over that shown in their previous design, Figure 3.8. Differential thermal expansion throughout the tube

HOCKEY STICK SINGLE WALL TUBE — SIMPLEST AND ONLY SCALABLE, TEST PROVEN DESIGN TEST

Figure 3.2

AI Hockey Stick Steam Generator Design Features

SINGLE WALL HELICAL COIL DESIGN FEATURES

Figure 3.3

Babcock and Wilcox Helical Steam Generator Design Features

Figure 3.4. Helical Tube Design Pigtail Support Details

Figure 3.5. Main Tube Bundle Support
for the Helical Tube Design
(Upper Portion)

Figure 3.6. Main Tube Bundle Support for Helical Tube Design (Lower Portion)

2-10" ID SODIUM
INLET NOZZLES

1. THE STEAM GENERATOR TO BE DESIGNED AND FABRICATED TO MEET THE CODE REQUIREMENTS FOR CLASS 2 OF THE ASME BOILER AND PRESSURE VESSEL CODE.

2. DESIGN CONDITIONS

WATER-SIDE TEMPERATURE (S.M.)	225°F
WATER-SIDE TEMPERATURE (LEAP)	270°F
WATER-SIDE PRESSURE (S.M.)	2705 PSIG
WATER-SIDE PRESSURE (LEAP)	285 PSIG
SODIUM-SIDE TEMPERATURE (S.M.)	900°F
SODIUM-SIDE TEMPERATURE (LEAP)	925°F

3. OPERATING CONDITIONS (FULL LOAD)

SODIUM INLET TEMPERATURE	900°F
SODIUM INLET PRESSURE	125 PSIG
SODIUM OUTLET TEMPERATURE	800°F
STEAM OUTLET TEMPERATURE	835°F
STEAM OUTLET PRESSURE	2275 PSIG
WATER INLET TEMPERATURE	670°F

4. THE BASIC MATERIAL IS 2 1/4" CR AND

STEEL - SPECIFICATIONS AS FOLLOWS
PLATES - SA-387 CR 22 CLASS 1 ANNEALED
TUBES - SA-310 CR 22 CLASS 1 ANNEALED
PIPE - SA-310 CR 22 CLASS 1 ANNEALED
PIPE - SA-310 CR 22 CLASS 1 ANNEALED
NOZZLE - SA-310 CR 22 CLASS 1 ANNEALED
NOZZLE - SA-310 CR 22 CLASS 1 ANNEALED
NOZZLE - SA-310 CR 22 CLASS 1 ANNEALED
NOZZLE - SA-310 CR 22 CLASS 1 ANNEALED

5. ESTIMATED WEIGHT (GROSS) 1,000,000 LBS.

6. ESTIMATED WEIGHT (NET) 1,000,000 LBS.

7. ESTIMATED WEIGHT (NET) 1,000,000 LBS.

8. ESTIMATED WEIGHT (NET) 1,000,000 LBS.

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49. ESTIMATED WEIGHT (NET) 1,000,000 LBS.

50. ESTIMATED WEIGHT (NET) 1,000,000 LBS.

ARRGT OF
SODIUM HEATED
STEAM GENERATOR
INT. A
435 MWT

Figure 3.8. Previous Helical Tube Design Showing Different Sodium Distribution and Inlet Nozzle

bundle is easily accommodated, thereby reducing any effects of sodium flow maldistribution (streaming). It employs a cover gas space in the upper section to reduce sodium inventory in the shell and upper tubesheet transients. The gas space dictates the need for sodium level control. Because of the pigtailed and helical coil design, from 11 to 32% of the tube length is ineffective, depending on the specific cycle. The upper tubesheet (shown in Figure 3.9) features a nonsymmetrical configuration with a relatively large rim on the gas space side for joining the shell, tubesheet, and steamhead into an integral assembly. This type design amplifies the radial stresses that occur from reactor trip transients (see Section 4.1.1).

3.1.2 Double-Wall Tube Designs

The Westinghouse DWT design is shown in Figure 3.10. It contains helium leak detection fluid in the space between the double upper tubesheets. The outer tube of the double-wall tube assembly ends in this plenum such that the helium can enter from the four longitudinal grooves on the ID of each outer tube. The four grooves (.020-in. deep by .050-in. wide) provide a free helium gas flow path for quick detection of a sodium leak. The gas space between prestressed tubes impedes helium flow as the average gap is approximately 0.2 mils. For the saturated cycle, W has proposed two possible design options. One employs annealed DWT's with a gap between the two walls with a mixed gas for conduction control. The other uses prestressed mechanically bonded DWT's with shock tubes to reduce the heat flux. The feasibility of the gap approach seems questionable as the attainment of a uniform gas space between two concentric tubes without using a nonconducting wall gap spacer would be very difficult over the entire tube length (up to 70 ft). The W DWT designs for Benson and Sulzer cycles use prestressed mechanically bonded types. No shock tubes are required for these cycles. The inner tube passes through the leak detection region to the upper tubesheet. The use of three tubesheets and the helium space leak detection system

Figure 3.9 Upper Tubesheet for the Helical Tube Design

FIGURE 3-10

**CROSS SECTION VIEW
OF THE W DWT
STEAM GENERATOR
ASSEMBLY**

Rockwell International
Energy Systems Group

requires five fillet type welds per tube as shown in Figure 3.11. Fillet welds present a much more severe inspection problem for certifying welding quality than do the butt welds used in the single wall tube designs. The straight double-wall tubes are subject to buckling problems resulting from sodium maldistribution which causes tube-to-tube ΔT 's. An externally reinforced single-ply bellows, Figure 3.12, is required on the shell to accommodate differential thermal expansion between the shell and the tubes. The reinforcement is required to reduce the large pressure stresses in the bellows to an acceptable level. The reinforcement requires tight tolerances to adequately support the bellows. The use of thick large-diameter bellows requires a significant development effort for welding if dissimilar metals are used or for fabrication if formed rather than machined bellows are employed. Figure 3.13 shows that the upper sodium side tubesheet has a large 27-in. wide ring that is exposed to the same sodium inlet temperature as the center tubesheet section. This large ring amplifies the radial stresses that occur from reactor trip transients even if it were insulated from the sodium. The W design employs a hanger rod tube spacer support method without a shroud for prevention of sodium streaming. Of the four designs, this design has the greatest potential for tube buckling from tube-to-tube expansion differences in the active heat transfer region and single tubes between the two upper tubesheets (see Section 4.2.3).

The Foster-Wheeler DWT design, Figures 3.14 and 3.15, provides a single tubesheet at each end. The lower tubesheet is designed with a series of 3/8-in. diameter leak detection holes drilled through the neutral axis of each tubesheet as shown in Figure 3.16. They are drilled between each row of tubes in the tubesheet. At right angles to each of these holes, another series of holes are drilled connecting these holes to each of the tubesheet holes providing leak detection capabilities for the tube-to-tubesheet welds and tubes without employing a double tubesheet. This design appears very difficult from a fabrication point of view. Leak detection equipment will monitor this series of holes and provide

SWT DESIGN HAS FEWER TUBE-TO-TUBESHEET WELDS PER TUBE

Figure 3.11
Comparison of Tube-Tubesheet Weld Configurations

DWT DESIGN REQUIRES EXTERNALLY REINFORCED BELLOWS

LARGE DEVELOPMENT
TO OBTAIN CRACK
FREE WELDS

REINFORCEMENT

- REQUIRED TO REDUCE PRESSURE STRESS
- THERMAL LAGS CAN CAUSE GAPPING DISTORTIONS
- LOAD TRANSFER DIFFICULT TO PREDICT

REQUIRE CLOSE BELLOWS/SUPPORT TOLERANCES

- DIFFICULT FABRICATION TO REQUIRED TOLERANCES

THICKNESS

- LARGE FABRICATION DEVELOPMENT TO FORGE THICK WALL BELLOWS

DEAD WEIGHT
FORCE

Figure 3.12

Westinghouse Reinforced Shell Bellows Design Features

W STEAM GENERATOR DWT DESIGN UPPER SECTION

Figure 3.13

Westinghouse DWT Design, Upper Section

F-W PLBR STEAM GENERATOR HAS FUNDAMENTAL PROBLEMS SIMILAR TO W DESIGN

Figure 3.14

Foster Wheeler Steam Generator Design Features

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Figure 3.16

**F-W TUBESHEET
HOLES FOR LEAK
DETECTION
REQUIRE
UNREALISTIC
FABRICATION
TOLERANCES**

leak indications for the tube-to-tubesheet welds and tubes. The small gap (~0.2 mils) between the two walls of the prestress tube will greatly impede the flow of the leak detection fluid thereby delaying the detection of a sodium leak. In the W design, this flow impedance was reduced by the incorporation of four longitudinal grooves on the ID of each outer tube.

The F-W upper tubesheet design appears structurally superior to the W design in respect to diameter and transient performance. It has eliminated the need for a large ring surrounding the upper tubesheet and has provided a stagnant sodium region next to the tubesheet which is insulated from the inlet sodium inlet flow by thermal baffles as shown in Figure 3.17. The F-W design employs disc and donut type tube spacers, Figure 3.17, to impart a swirl to the sodium flow through the tube bundle. This design also features piston ring type seals around the OD of the tube spacers. These two features, swirl and seal, reduce the potential for tube-to-tube thermal expansion problems. A reinforced metal bellows, Figure 3.18, is provided in the lower end of the shell to absorb the differential thermal movements between the shell and tubes. Although its design is somewhat different from the W bellows, it still has the same fundamental problems. It uses split reinforcement rings to support the bellows.

3.2 DESIGN NORMALIZATION

3.2.1 As-Designed Unit Comparison

The thermal and hydraulic design conditions, physical dimensions, and performance characteristics of the steam generators as designed by AI, B&W, W, and F-W are tabulated in Table 3.1 for each steam cycle. Both the evaporators and the superheaters are included for the Sulzer cycle. Data were taken from Phase II reports submitted by each company to ERDA in May/June 1977, and from References 5, 6, and 7. The sizes given in Table 3.1 are quite different as they were designed for a

Figure 3.17. Thermal Baffle, Sodium Nozzle and Tube Spacer Details for FW DWT Steam Generator

Figure 3.18

F-W BELLOWS REINFORCEMENT DESIGN PROBLEMS SAME AS FOR W UNIT

Table 3.1

COMPARISON OF STEAM GENERATOR PARAMETERS AS DESIGNED BY SUPPLIER

	Saturated Cycle				Benson Cycle				Supercycle							
	AI	B&W	W	FW	AI	B&W	W	FW	AI		B&W		Westinghouse		GE/Foster Wheeler	
									Super-heater	Evap- orator	S	F	S	F	S	E
Performance																
Duty/Unit (MWt)	337	322	483	729	291.2	438	292	649	249.1	314.1	113	319	210	320	159	491
No. of Loops	3	3	4	4	3	2	3	4	3	3	3	3	3	3	4	4
Na Inlet T (°F)	815	815	815	815	900	900	900	913	900	814	900	817	900	818	913	820
Na Outlet T (°F)	550	550	550	550	600	600	604	600	614	600	620	581	618	583	620	592
FW Temp (°F)	529	530	532	530	470	470	466	380	662	470	660	380	660	380	660	380
Stm Outlet T (°F)	549	545	553	544	856	855	859	850	856	662	856	662	855	660	854	660
Peak Heat Fluxes (Btu/hr-ft ²)	150,000	150,000	155,000	159,000	159,000	118,000			119,000				125,000	125,000		
LMTD (Avg)	96.5	96.03	91.11	56.3	79.4	79.92	79.92	125.5	88.69	140.71	86.51	171.48	89.97	179.56	101	185
ΔT Na-H ₂ O, Hot End	266	270	262	271	44	41	41	63	46	152	45	145	45	158	59	160
ΔT Na-H ₂ O, Cold End	21	20	18	20	130	138	138	220	152	130	148	201	158	203	160	212
ΔP, Water/Stm (psi)		60	32	37	100	544	15				263	207	90	22		
ΔP, Na (psi)			20	25	10	14	15				11	12	20	15		
Na Velocities (ft/sec)	4				4	2.5			12	6						
Stm Velocities (ft/sec)	25.41				64	129			200	60						
Na Flow (lb/hr)	14.0x10 ⁶	13.4x10 ⁶	20.3x10 ⁶	30.5x10 ⁶	10.8x10 ⁶	16.3x10 ⁶	11.0x10 ⁶	23x10 ⁶	12.4x10 ⁶	16.2x10 ⁶	15.25x10 ⁶	15.25x10 ⁶	30.2x10 ⁶	15.1x10 ⁶	23x10 ⁶	23x10 ⁶
Water Flow (lb/hr)	8.7x10 ⁶	5.6x10 ⁶	12.5x10 ⁶	19.1x10 ⁶	1.1x10 ⁶	1.64x10 ⁶	1.1x10 ⁶		3.26x10 ⁶	1.72x10 ⁶	2.89x10 ⁶	3.04x10 ⁶	2.85x10 ⁶	1.496x10 ⁶	2.125x10 ⁶	2.35x10 ⁶
Water Inlet Pressure (psia)		1105	1100	1097			2305						2370	2421		
H ₂ O Exit Pressure (psia)	1040	1045		1060	2291	2290	2290	2275	2291				2285	2400	2275	
Circulation Ratio	6:1	4:1	6:1	6:1	-	-	-	-	-	3.05	-	-	-	-	-	1.1
Geometry																
Number/Loop	3	3	2	1	3	2	3	1	1	2	2	2	1	2	1	1
No. of Tubes	862	172	2580	2010	2224	240	3333	2605	1970	2434	196	176	2200	2600	2880	4815
Tube Length, Active/Total (ft)	77/98	159/	62/103	79/89	77/98	477/	63/81	110/	51.5/74	77/98	117/	201/	44.06/59	69.6/76	36/	62/
No. Tube Welds/Unit	1724		12506	8040	4448	3600	16665	10420	1940	4868	1960	2352	11000	13000	11520	19260
Tube Sheet Thickness, Na Inlet/Outlet (in.)			19/10/17	22/22	17/14		27/15/25						19/	22	18/	21.5
Tube OD, Outer Tube/Inner Tube	1.5	2.0	1.0/0.904	OD ID 1.5/1.35	5/8	1.25	0.8125/0.6605	OD ID 0.8125/0.405	5/8	5/8	1.25	1.25	0.8125/0.6565	0.75/0.628	OD ID 0.8125/0.405	OD ID 0.8125/0.405
Tube Wall Thickness, Outer/Inner	0.109	0.132	0.045/0.08	0.07/0.06	0.109	0.188	0.076/0.112		0.109	0.109	Min Nom 0.171/0.188	Min Nom 0.148/0.163	0.078/0.115	0.061/0.102		
Shell ID, ID/Thick (in.)	85/2	138/2.5	90.13/	127.5/2.0	72/2.0	142/	92.12/	97/3.5	82/2	77/2	120/	120/	83	83	OD 33/3.0	OD 125/4.25
Overall Unit Weight (ton)			430		200	500	350				460,000 lb	435,000 lb	270	300		
Na Nozzle, In/Out (in.)	22/	2-18/1-24	28/28	34.5/34.5	2-20/1-20	1-28/1-28	24/24		2-20/1-26	2-20/1-26	13/33	13/33	21/23			
H ₂ O Nozzle, In/Out (in.)	1	4-12/4-12	30/	34.5/23	1-16/1-16	4-8/4-8	5.6/14		4-8/4-8	4-8/4-8	14/17		16			
Max. Tube Wall, ΔT (°F)					63											
Shroud, Dia/Thick (in.)	75/1.0				58/1.0				60/1	60.5/1	112	112				
Tubing Material	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Pitch (in.)	2.34	2.375	1.625	2.5	1.125	1.75	1.4375		1.2	1.125			1.4375	1.375		
Tubewall Conductivity (Btu/hr-°F)																
HT Transfer Surface (ft ²)	26065	14121	39361	62357	28100	37777	44665	60950	30500	7509	18668	19566	62757			
Protective Tube OD/Thick/Length	1.975/0.2275/4		One between tubes	1.90/0.19/15			3/16 in. Inlet orifice									

*Too small, should be ~90

12-1/4 Cr - 1 Mo

124.25-0.00527

different number of loops per plant and different numbers of steam generator modules per loop for each steam cycle. In addition, each vendor used different cycle conditions and design factors for sizing the steam generator for each steam cycle. For example, for the saturated cycle, AI, B&W, W, and F-W have three 337 Mwt, three 322 Mwt, two 483 Mwt, and one 729 Mwt steam generators, respectively, per loop. All are based on three-loop plants except the F-W concept used in GE's design which employs four loops. AI, B&W, W, and F-W each use large tubes, having diameters of 1.5-in., 2.0-in., 1.0-in., and 1.5-in., respectively. The recirculation ratio is 6:1 in all cases except for the B&W helical design (4:1). Design thermal conditions are almost identical. For the saturated cycle protector tubes are required for all designs to reduce the heat flux to reasonable values. AI's design contains 304 stainless steel protector tubes over more than half of the active tube length. The W and F-W designs have a shorter length of protector tubes covering the duplex tubes at the sodium inlet side. Protector tubes do not appear to be feasible for the helical tube design.

For the Benson cycle, AI, B&W, Westinghouse, and F-W have 291, 435, 292, and 649 Mwt unit duty per loop, respectively. Each system consists of three loops except for the F-W design which has four loops. The tube OD is 0.625 for AI, 1.25 in. for B&W, and 0.8125 in. for Westinghouse and F-W. The PLBR Phase II design thermal conditions are almost identical for all designs except that the feedwater temperature for the F-W concept is $\sim 90^{\circ}\text{F}$ lower than others (380°F vs 470°F) and the sodium inlet temperature is 13°F higher.

For the Sulzer cycle designs, AI employs two evaporators with 314 Mwt each (0.625-in. tubes) and one superheater with 249 Mwt duty (0.625-in. tubes), B&W has two 319 Mwt evaporators and two 113 Mwt superheaters each with 1.25-in. tubes, Westinghouse has two 319 Mwt evaporators (0.75-in. tubes) and one 212 MW superheater (0.8125-in. tube), and finally, one 491 Mwt evaporator and one 159 Mwt superheater

each with 0.8125-in. tubes for the F-W design. Thermal design conditions for the superheaters are almost identical in all cases. However, feed-water temperature for the AI design is higher than those of the other three cases (470°F vs 380°F) in the evaporator designs. Again, AI, B&W, and Westinghouse designs are based on three loops and F-W on four loops. Also, as can be seen in Table 3.1, the shell diameter of the F-W superheater appears to be in error. Based on the shell diameter of 125 in. for the evaporator which contains 4815 tubes of the same size, the diameter for 2880 tubes should exceed 90 in.

3.2.2 Normalization of Units for Use in Comparison

To allow a meaningful comparison, it was necessary to put all of the steam generators on a similar basis. As discussed in the previous section, the data available at the start employed different numbers of heat transfer loops, design points, etc. The designs given in Table 3.1 were, therefore, adjusted to a consistent set of design conditions, namely, three heat transfer loops with three steam generator modules per loop and uniform cycle conditions. No attempt was made to reoptimize the number of tubes, tube diameters, or to correct for differences in heat transfer design margin differences. It was assumed each supplier had used the best design factors for their particular design and would be reluctant to change to design factors developed by another supplier.

The designs were all revised to the design operating conditions listed in Table 3.2. The results of this resizing effort is presented in Table 3.3. An examination of Table 3.3 shows comparative discrepancies in number of tubes and active surface area. These variances result from the normalization technique used and the design margins provided by the different manufacturers. It is believed that these discrepancies could only be eliminated by reoptimizing all designs to similar ground rules. The reoptimization of the cycle conditions and steam generator tube requirements for each design was beyond the funding available for this study.

TABLE 3.2
SELECTED OPERATING PARAMETERS FOR 1000 MWe LMFBR

	BENSON	SULZER	SATURATED
CORE POWER, MWt	2585	2585	2888
TURBINE INLET CONDITIONS			
PRESSURE, PSIA	2215	2215	1015
TEMPERATURE, °F	850	850	546
REACTOR OUTLET TEMPERATURE, °F	930	930	875
STEAM GENERATOR INLET NA, °F	900	900	815
STEAM GENERATOR OUTLET NA, °F	600	600	550
INTERMEDIATE ΔT , °F	300	300	265
FEEDWATER TEMPERATURE, °F	470	470	424
RECIRCULATION RATIO	NONE	1.05	6.0

MODULES PER LOOP/LOOPS PER PLANT

	BENSON		SULZER	SATURATED
	AI HOCKEY STICK	B&W HELICAL	W-DWT	FW-DWT
BY VENDOR	3/3	2/3	(2-1)/3	2/4
NORMALIZED	3/3	3/3	(2-1)/3	3/3

TABLE 3.3

COMPARISON OF STEAM GENERATOR PARAMETERS AFTER RESIZING TO COMPARABLE OPERATING CONDITIONS

	Saturated Cycle				Benson Cycle				Sulzer Cycle							
	AI	W	FW	B&W	AI	W	FW	B&W	AI		Westinghouse		FW		B&W	
									Super-heater	Evaporator	S	E	S	E	S	E
Core Power (MWt)	2988	2988	2988	2988	2585	2585	2585	2585	2585	2585	2585	2585	2585	2585	2585	2585
Duty/Unit (MWt)	337	337	337	337	291	291	291	291	249	312	249	312	249	312	249	312
No. per Loop/ No. Loops	3/3	3/3	3/3	3/3	3/3	3/3	3/3	3/3	1/3	2/3	1/3	2/3	1/3	2/3	1/3	2/3
Na Inlet T (°F)	815	815	815	815	900	900	900	900	900	814	900	814	900	814	900	814
Na Outlet T (°F)	550	550	550	550	600	600	600	600	814	600	814	600	814	600	814	600
FW Temp (°F)	529	529	529	529	470	470	470	470	662	470	662	470	662	470	662	470
Steam Outlet T (°F)	549	549	549	549	856	856	856	856	856	662	856	662	856	662	856	662
Steam Exit Press. (psia)	1040	1040	1040	1040	2291	2291	2291	2291	2291	-	2291	-	2291	-	2291	-
Na Flow (lb/hr)	13.9x10 ⁶	14.0x10 ⁶	14.3x10 ⁶	14.0x10 ⁶	10.8x10 ⁶	11.05x10 ⁶	16.4x10 ⁶	10x10 ⁶	32.2x10 ⁶	16.1x10 ⁶	36.3x10 ⁶	18.96x10 ⁶	42.4x10 ⁶	21x10 ⁶	33x10 ⁶	18.3x10 ⁶
Water/Steam Flow (lb/hr)	8.7x10 ⁶	8.66x10 ⁶		8.4x10 ⁶	1.1x10 ⁶	1.09x10 ⁶			3.25x10 ⁶	1.72x10 ⁶	3.42x10 ⁶	1.88x10 ⁶	3.92x10 ⁶	2.15x10 ⁶	6.25x10 ⁶	3.65x10 ⁶
Recirculation Ratio	6:1			6:1						1.05						
Na Vel. Max (ft/sec)	4				4				12	6						
No. of Tubes	862	1632	930	178	2224	3365	1858	163	1970	2434	2641	3265	5307	4410	424	236
Tube Length, Active/Total (ft)	77/98	62/103	79/89	159/71*	77/98	63/72	110/124	477/80*	51.5/74	77/98	44/59	63.6/79	36/	61/	187/55*	291/75*
Tube OD, Outer Tube/Inner Tube	1.5	1.0/0.904	1.5/1.35	2.0	5/8	0.8125/0.6605	0.8125/0.405 ID	1.25	5/8	5/8	0.8125/0.6565	0.75/0.628	0.8125/0.405 ID	0.8125/0.405 ID	1.25	1.25
Tube Wall Thickness, Outer/Inner	0.109	0.045/0.081	0.07/0.06	0.132	0.109	0.076/0.112		0.171	0.109	0.109	0.078/0.115	0.061/0.102			0.188	0.163
Tube Sheet Diam, Na In/Na Out	857/857	75/ /	897/	20/	727/	106/ /	82/	22/		777/	/ /	85/ /	130/	105/		15/
Tube Sheet Thick, Na In/Na Out		17/19	22/22	11 /	17/14	27/25										
Shell Diam, ID/Thickness (in.)	85/2.0	75/	89/1.75	140/2.5	72/2.0	92.5	82/3.5	117/	82/2	77/2	85/	87/	**44.8/3.0	105/4.25	177/	132/
Shroud Diam, OD/Thickness	75/1.0			121.5/	58/1.0				60/1	60.5/1						
No. Tube Welds	1724	8160	3720	1780	4448	16825	7432	2445	3940	4868	13205	16325	21228	17640	4240	2832
Length, Upper Stag. Region (ft)	17				17					17						
Length, Lower Stag. Region (ft)	4				4					4						
Orifice, orifice/ΔP	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
Tube Pitch	2.34	1.625	2.5	2.575	1.125	1.4375			1.2	1.125	1.4375	1.375				
Protective Tube, OD/Thick/Length	1.975/0.2275/41		1.90/0.19/15													
Na Nozzles, In/Out	1-24/1-24	1-24/1-24	1-24/1-24	2-18/1-26	1-20/1-20	1-24/1-24	2-16.5/				1-36/1-36	1-26/1-26			2- /	2- /2-
H ₂ O Nozzles, In/Out			1-24/1-	4-12/4-12	1-16/1-16	1-9.6/1-14					1-16/1-19	1- /1-18			4- /	4- /4-
Heat Transfer Surface (OD) (ft ²)	26065	26490	28851	14819	28000	45090	43478	25440	16600	30670	24718	40772	40639	57221	16230	22500
Overall Weight (ton)	200	171		370												
Steam Drum, ID/Length	12/65 Loop	7/30 Loop	8/40 Unit	12/65 Unit												

*Total length of the SG including nozzles and support skirt

**44.8 seems too small. It is estimated to be ~130.

Table 3.4 presents a summary of the different designs as a function of steam cycle. It can be seen that all designs employ large diameter tubes for the saturated cycle compared to the superheated cycles. Shock tubes are required to protect the tubes in the sodium inlet region of all concepts for the saturated cycle; however, these were not incorporated in the helical coil design because of the difficulty in fabrication of such an arrangement. It can be seen that the helical coil design employs long, large diameter tubes even for the superheat cycles. The helical coil tube length for the Benson cycle is almost 500 ft, going down to much shorter lengths for the other cycles. The AI hockey stick design employs small diameter tubes for the superheat cycles, and the DWT designs are about halfway in between the hockey stick and helical coil designs. Because of the W tube length limitation and because of the poorer heat transfer in DWT's, the W unit generally has a larger number of tubes in comparison to the other units for a given cycle (except in the Sulzer cycle where the F-W unit also employs fairly short tubes). The 110 ft length employed in the F-W Benson cycle design seems overly optimistic for a DWT design. Tables 3.5, 3.6, and 3.7 provide more detail for each cycle, giving the size of the major components in the different steam generators for each cycle.

3.3 DESIGN FEATURE COMPARISON

The following size and performance observations can be made from the steam generator supplier's drawings included in Tables 3.4 through 3.7.

- 1) Tubesheets - The AI design, Figure 3.2, employs thick tubesheets. The stagnant sodium volume in the tubesheet region helps to stabilize steam temperature and mitigate transient effects. The plant shutdown control system was optimized to take care of the characteristics of this tubesheet for either 4% or 7% pony motor speeds. With proper post scram flow control (easily achievable), tubesheet stresses are not a problem.

TABLE 3.4
STEAM CYCLE EFFECTS ON STEAM GENERATOR DESIGN

TUBE NO. DIAM/LENGTH (IN./FT)	SUPERHEATED			SATURATED	
	BENSON	SULZER		TUBES	SHOCK TUBE THICK/LENGTH
		EVAP.	SUPER.		
HOCKEY STICK $\frac{N}{OD/L}$	$\frac{2224}{.625/77}$	$\frac{2434}{.625/77}$	$\frac{1970}{.625/52}$	$\frac{862}{1.5/77}$	.23/41
HELICAL $\frac{N}{OD/L}$	$\frac{163}{1.25/477}$	$\frac{236}{1.25/291}$	$\frac{424}{1.25/117}$	$\frac{178}{2.0/159}$	**
W-DWT $\frac{N}{OD/L}$	$\frac{3365}{.813/63}$	$\frac{3265}{.75/64}$	$\frac{2641}{.813/44}$	$\frac{1632}{1.0/62}$	.25/11***
FW-DWT $\frac{N}{OD/L}$	$\frac{1858}{.813/110}$	$\frac{4410}{.813/61}$	$\frac{5307}{.813/36}$	$\frac{930}{1.5/79}$	.19/15

*INDIATES ORIFICES REQUIRED

**NO SHOCK TUBE INCLUDED - QUESTIONABLE DESIGN

***ALTERNATE DESIGN FEATURE GAS GAP BETWEEN TWO ANNEALED TUBES TO CONTROL HEAT FLUX

TABLE 3.5
COMPARISON OF STEAM GENERATOR SIZE
FOR BENSON CYCLE

	HOCKEY STICK	HELICAL TUBE	W (DWT)	FW (DWT)
TUBES:				
NUMBER	2224	163	3365	1858
OD, IN.	.625	1.25	.8125	.8125
THICKNESS, IN.	.109	.171	.188	.204
HEAT TRANSFER SURFACE, FT ²	28000	25440	45090	43480
ACTIVE LENGTH, FT	77	477	63	110
TOTAL LENGTH, FT	98	535	72	124
TUBE WELDS, NUMBER				
TOTAL	4448	2445	16825	7432
TUBESHEET DIA., IN.	72	22	160/106	82
SHELL ID, IN.	72	117	93	82

TABLE 3.6
COMPARISON OF STEAM GENERATOR PARAMETERS
FOR SULZER CYCLE

	HOCKEY STICK		HELICAL TUBE		W (DWT)		FW (DWT)	
	S	E	S	E	S	E	S	E
TUBES:								
NUMBER	1970	2434	424	236	2641	3265	5307	4410
OD, IN.	.625	.625	1.25	1.25	.8125	.75	.8125	.8125
THICKNESS, IN.	.109	.109	.188	.163	.193	.163	.204	.204
HEAT TRANSFER SURFACE, FT ²	16600	30670	16230	22500	24718	40772	40639	57221
ACTIVE LENGTH, FT	52	77	117	291	44	64	36	61
TOTAL LENGTH, FT	74	98	175	350	59	79	41	69
TUBE WELDS, NUMBER								
TOTAL	3940	4868	4240	2832	13205	16325	21228	17640
TUBESHEET DIA., IN.	82	77	16	15	130/85	130/85	130	105
SHELL ID, IN.	82	77	177	132	85	87	130	105

TABLE 3.7
COMPARISON OF STEAM GENERATOR SIZE
FOR SATURATED CYCLE

	HOCKEY STICK	HELICAL TUBE	W (DWT)	FW (DWT)
TUBES:				
NUMBER	862	178	1632	930
OD, IN.	1.5	2.0	1.0	1.5
THICKNESS, IN.	.109	.132	.126	.13
HEAT TRANSFER SURFACE, FT ²	26065	14819*	26490	28851
ACTIVE LENGTH, FT	77	159	62	79
SHOCK TUBE LENGTH, FT	41	0*	11	15
TOTAL LENGTH, FT	98	215	103	89
TUBE WELDS, NUMBER				
TOTAL	1724	1780	8160	3720
TUBESHEET DIA., IN.	85	20	120/75	89
SHELL ID, IN.	85	140	75	89

*NO SHROUD - RESULTS IN HIGH RISK Q/A

Although the B&W design, Figure 3.3, features four tubesheets at each end of their steam generator, they are of much smaller size compared to the others. This small diameter results in a thinner tubesheet, which in turn reduces thermal stress for the same type environment and attachment to steamhead and shell nozzle. As noted previously, this advantage has been negated to a considerable extent by the gas space and unsymmetrical attachments. They are also located outside the main shell boundary thereby facilitating the inspection of the tubes and tubesheets. Although the gas space eliminates direct sodium flow temperature transients, it increases steam temperature transient effects (see Section 4.0, pp 4-2).

The Westinghouse design, Figure 3.10, employs two large tubesheets at the upper section providing a helium leak detection chamber in between. The outer ring on the W sodium side upper tubesheet as well as the tube section are exposed to sodium inlet flow temperature transients. The outer ring makes the diameter of this tubesheet much larger than for any other design. The additional size required by this ring places a major drawback on this concept, especially for the Benson and Sulzer steam cycles (see Section 4.0).

The Foster-Wheeler design, Figure 3.15, has the thickest upper tubesheet (22 in. for the saturated cycle). However, the stagnant sodium between the tubesheet and thermal baffle reduces its thermal stress, compared to the W design.

- 2) Number of Tube Welds - The number of welds is much lower for the Babcock & Wilcox and AI designs than for the other two. The single-wall tube design only require a single tube-to-tubesheet weld at each tubesheet as shown in Figure 3.11, and can be manufactured in flexible bent tube configurations, i.e., the hockey stick or helical configurations. Since the B&W design employs very long tubes, the number of tube-to-tubesheet welds is smaller for the B&W than for the AI design; however, B&W requires a large number of tube-to-tube welds (~10 to 15 per tube - depends on the application).

Westinghouse and Foster-Wheeler designs require five and four welds, respectively, for each tube at the tubesheets alone because of their duplex tube configuration. This results in a great number of total tube welds for both duplex tube designs. In the Westinghouse design, the outer tube is welded to the inner tubesheet at each end, and the inner tube is welded to the outermost surface of the upper tubesheet as shown in Figure 3.11. This design leaves the sodium side of upper tubesheet exposed to the potential for crevice corrosion.

3) Active Heat Transfer Surface Area -

- a) Saturated Cycle - The B&W design cannot practically employ the protective tubes required to suppress the maximum heat flux in the saturated cycle. Their design indicates a peak heat flux of ~350,000 Btu/hr-ft² compared to ~150,000 Btu/hr-ft² or less for the other three designs. Of the practical designs, AI's design exhibits the smallest area and the 304 stainless steel protective tubes are used over 41 ft of

the total active tube length of 77 ft to reduce the maximum heat flux. The Foster-Wheeler design has the largest heat transfer area.

- b) Benson and Sulzer Cycles - Because the AI and Babcock & Wilcox design employs single-wall tubes, while the Westinghouse and Foster-Wheeler designs use double-wall tubes, the surface areas of the AI and B&W is much less than that of Westinghouse and F-W design. B&W designs require a little less area than the AI designs due to higher steam velocity employed.

Note: As noted in Section 2.1, a lower feedwater temperature would significantly decrease the heat transfer area for all three steam cycles and would allow the DWT designs to be reduced to a practical size.

- 4) Single Wall Versus Double Wall Tube - A summary of comparison of the design features of single and double wall tubes are summarized in Tables 3.8 and 3.9. Table 3.8 presents a comparison of design features. It shows the only advantage of the DWT is that it gives lower potential for a sodium-water leak, and if a leak occurs into the sodium side, an immediate reactor scram is not required, as the leak detection fluid will then bubble into the sodium with little or no effect on performance. Table 3.9 shows the single wall tubes have a superior experience foundation, both in number of units and failures.

The above design features are summarized in Table 3.10. It shows the hockey stick is the least complex and that the helical and DWT concepts are more complex and have several problem areas that must be resolved to show they are viable designs.

TABLE 3.8

SINGLE WALL (SWT) VERSUS DOUBLE WALL (DWT) TUBE DESIGN CHARACTERISTICS

CHARACTERISTIC	SWT	DWT
TUBE GEOMETRY	BENT/HELICAL	STRAIGHT
DIFFERENTIAL EXPANSION COMPENSATION		
• SHELL-TUBE BUNDLE	COMPLIANT TUBES	SHELL BELLOWS
• TUBE-TUBE	COMPLIANT TUBES	NONE
NUMBER OF TUBE-TUBESHEET WELDS	2/TUBE	4-5/TUBE
POTENTIAL FOR TUBE LEAKS	LOW	~TWICE SWT
LEAK DETECTION	H ₂ DETECTOR	THIRD FLUID
POTENTIAL FOR SODIUM-WATER LEAK	LOW	LOWER*
REPAIR UNAVAILABILITY		
• NUMBER OF REPAIRS REQUIRED	LOW	~TWICE SWT
• EASE OF REPAIR	SIMPLE	COMPLEX

*ADDED ADVANTAGE IN THAT IMMEDIATE SHUTDOWN IS NOT REQUIRED IF LEAK IS INTO SODIUM. LEAK DETECTION FLUID WILL THEN BUBBLE INTO SODIUM WITH LITTLE EFFECT ON PERFORMANCE.

TABLE 3.9

SINGLE VS DOUBLE WALL DESIGN SUCCESS RECORD

DOUBLE WALLEDSIR (11)
HNPF MODELAICO
EBR-I
HNPFEBR-II
SRE
DFRUNITS/FAILURES

19/14+

SINGLE WALLEDAPDA U TUBE
APDA BAYONET
LASL
W SERPENTINE
B&W 1 TUBE
GRAND QUEVILLY
KNK MODEL
MONJU
SGTRMSG
PHENIX
FIVES CAIL
STEIN
SNR (2)
SUPER PHENIX
MONJUFERMI
KNK
PHENIX
BOR60
BN350
PFR

21/11+

TABLE 3.10
GENERAL DESIGN FEATURES COMPARISON

- OVERALL FEATURES

- . HOCKEY STICK - SIMPLEST DESIGN
- . HELICAL TUBE MOST COMPLEX
 - MANY COMPLEX TUBE SHAPES AND TUBE BUNDLE SUPPORTS
 - MAY REQUIRE DIFFERENT SIZE ORIFICES FOR EACH TUBE
 - MANY TUBE-TO-TUBE WELDS
 - HELIUM/SODIUM INTERFACE REQUIRES LEVEL CONTROL/LEVEL CHANGES INDUCE TEMPERATURE CYCLING IN TUBE AT GAS LIQUID INTERFACE
 - SMALL TUBESHEETS DO NOT NECESSARILY PROVIDE BETTER TRANSIENT CAPABILITY (CAREFUL ATTENTION TO DESIGN FEATURES ARE STILL REQUIRED)
- . W AND F-W HAVE DWT COMPLEXITIES
 - DOUBLE OR COMPLEX TUBESHEETS (F-W LEAK DETECTION SCHEME QUESTIONABLE)
 - THICK SHELL BELLOWS
 - MOST INCHES OF WELDS

- ACTIVE SURFACE AREA

- . SAT. CYCLE - HELICAL TUBE SMALLEST, BUT SHOCK TUBES NOT FEASIBLE, OTHERS ABOUT EQUAL
- . BENSON AND SULZER CYCLES - HELICAL TUBE SLIGHTLY SMALLER, W AND F-W DOUBLE WALL TUBE DESIGNS MUCH LARGER THAN HOCKEY STICK

4.0 THERMAL/STRUCTURAL EVALUATION

Detailed thermal/structural evaluation of the four steam generator designs was confined to evaluating the problems with upper tubesheets, double wall tubes, and shell bellows when exposed to Benson, Sulzer, and saturated steam cycle conditions. A cursory comparative evaluation was made of the other regions such as sodium nozzles, tube supports, vessel supports, welds, and cover gas effects. The thermal hydraulics is presented first and the structural assessment then follows.

4.1 THERMAL TRANSIENT AND TUBESHEET EVALUATIONS

4.1.1 Upper Tubesheet

The typical sources of concern for the upper tubesheet result from high radial stresses occurring at the junction of the solid rim and the ligaments of the perforated region. Axial temperature gradients may increase the stress intensity at a particular location, but the radial temperature distribution is generally the primary driver. The perforation pattern during transients, tends to produce a sharp radial temperature gradient at the outer row of holes but nearly isothermal conditions in the perforated core of the tubesheet. The relative stiffness between the solid rim and the perforated core produces the maximum stress intensity at that interface. System transient mitigation control schemes and close attention to shell to tubesheet design configurations are required on all steam generator types to produce acceptable damage fractions for superheated cycle conditions. This problem is reduced in the saturated cycle because the steam temperature remains constant during most transients.

The perforated region of a tubesheet will respond rapidly to steam temperature changes whereas the outer solid rim will respond very slowly. The sodium inlet temperature is the primary driving force for changing

the temperature of the steam. The effect of sodium inlet temperature change on steam temperature at the tubesheet is a function of the design of the steam generator. A large stagnant sodium region next to the upper tubesheet will reduce the rate of change of steam outlet temperature whereas a stagnant gas region will not. A large solid rim area relative to the perforated region area will amplify the temperature gradient for a given steam temperature range.

4.1.1.1 Superheated Steam Generators

Transient temperatures were calculated for the AI hockey stick and BW helical tube upper tubesheet designs when exposed to a Benson cycle normal reactor trip. The results of the thermal analyses indicated that the critical delta temperature (rim minus perforated zone temperatures) for the B&W design will be approximately twice that for the AI design for the same reactor trip transient. The greater ΔT in the B&W tubesheet results primarily from the steam outlet temperature following the excursions of the sodium inlet temperature more rapidly than in the AI steam generator. This is due to the low thermal capacity of the gas space in the helical tube design and high thermal capacity provided by the hockey stick stagnant sodium region. It can be concluded from previous discussions of steam generator design features in Section 3.0 of this report, that for superheated cycle applications, the Westinghouse and FW steam generators will have steam outlet temperatures which will also follow the inlet sodium temperatures more rapidly than the hockey stick design as they employ either no stagnant sodium reservoir or a very small one. Based on these evaluations, the following conclusions about the superheated cycle tubesheets were drawn.

Atomics International (AI) - The AI configuration is based upon the technology and experience derived from the CRBR Steam Generator development program. A spherical steamhead is integral with the outlet tubesheet to avoid potential sealing concerns associated with bolted joints. The

tubesheet/shell configuration has been designed to minimize the effect of geometric discontinuities. The Normal Reactor Trip, which is the single most frequent upset transient, was found to produce acceptable creep/fatigue damage when evaluated by the ASME Elevated Service Temperature Criteria Code Case 1592-10.

Babcock & Wilcox (B&W) - The B&W design provides four tubesheets located in the spherical sodium vessel head similar to the AI design, but smaller. A buffer layer of gas is provided between the tubesheets and sodium interface. While the smaller tubesheets are desirable from a fabrication viewpoint, the thermal stresses are less sensitive to the tubesheet thickness than to radial stiffness (rim vs perforated region) and the radial ΔT that results from steam transients. The relatively large diameter of the nozzles on the shell increase the radial stiffness of the tubesheet and consequently the stress for a given thermal gradient. The unsymmetrical configuration for joining the tubesheet to the steam head and shell nozzle can cause large local bending stresses at the outer perforations. The effect of the stagnant gas region in the B&W design compared to stagnant sodium region in the AI design doubled radial ΔT 's as noted above. A structural analysis based on this increased ΔT indicated that the normal reactor trip may cause excessive stresses in this design. A redesign or extensive softening of temperature transients with additional mitigation controls is probably required for this design. The perfect hex of the perforated hole pattern will produce additional variations in circumferential temperature distribution not included here and may require extensive analysis to ascertain the maximum ligament stress intensity.

Westinghouse (W) - The Westinghouse double wall steam generator provides a double tubesheet configuration at the steam outlet end. The outer tube is welded to the helium side of the lower tubesheet while the inner tube extends through a helium chamber and is welded to the outer surface of the upper tubesheet. The inner tubesheet has a large rim

which extends out into the sodium inlet region. As a result, it may experience large radial and hoop stresses and potentially high ligament stress intensities. A rapid thermal response in the perforated region and a slow thermal response in the solid rim will produce an abrupt thermal gradient at the outer row of tube holes which, as noted before, when coupled with a high radial rim stiffness, will result in high stress intensities. Although this was not analyzed for the superheated cycle transients, it will be seen that it is a problem even at steady-state for the saturated case.

The tubesheet next to the steam header features a nonsymmetrical "Y"-configuration, Figure 3.13, for joining the helium plenum chamber, tubesheet, and steam head into an integral assembly. It is anticipated that this geometric discontinuity will cause large local bending stresses at the outer perforations.

Foster Wheeler (FW) - Again, while not analyzed for the superheated case, the Foster Wheeler design, Figure 3.15, utilizes a conical "Y"-joint configuration at the outer rim of the upper tubesheet. The tubesheet at the sodium side is undercut where it joins the cone. Nonuniform rim thickness with geometric discontinuities and the tubesheet undercut section could produce large ligament stresses on the steam surfaces with potentially extensive creep/fatigue damage. An undercut configuration, investigated by AI as part of the CRBR program, revealed that anticipated relief was not produced and, in fact, resulted in higher stresses than experienced in a nonundercut design.

4.1.1.2 Saturated Steam Generators

In the AI, B&W, and FW designs, the upper tubesheet and adjacent shell remain essentially isothermal during both steady-state and most transient conditions. As a result, significant thermal stresses are normally not present for these designs between the large rim and the

perforated region. In contrast, the Westinghouse design does not presently indicate a thermal baffle at the upper tubesheet and would thus operate at steady state with a temperature gradient across the tubesheet. This gradient can be adjusted, based upon the amount of thermal baffling, to provide a satisfactory stress level. However, because of the large rim in the Westinghouse design, it was found to have a problem even during steady-state operation. With no thermal baffles, the maximum ligament stress intensity is approximately twice the allowable value at 815°F. An additional analysis employing a 5-in. thermal baffle indicated an adequate design margin.

The transients imposed on the tubesheet normally cause only a change in steam quality but for a steam line break, a downshock will occur similar to that experienced by the upper tubesheets of the superheated steam generators. It would appear that in all but the Westinghouse design, there would not be much cyclic damage as a result of creep or fatigue. The Westinghouse design, which has a much larger rim around the perforated tubesheet, could be expected to exhibit a more complex behavior and subsequently, induce larger ligament stresses than the other designs.

4.1.2 Cover Gas Region Interface Thermal Transients

The sodium level fluctuations in the cover gas region of the helical design will give rise to thermal transients in the tubes and shell in this region. The magnitude of these temperature transients will vary with the steam cycle. The saturated cycle and the evaporator for the Sulzer cycle impose the most severe transients as they exhibit large temperature differences between the sodium and steam at the sodium inlet, i.e., 265°F for the saturated, 250°F for the Sulzer cycle evaporator vs 45°F for the Benon cycle and Sulzer cycle superheaters. Because of this, the tubes must be thermally shielded at the cover gas interface.

4.2 DWT PROBLEM AREAS

The straight DWT steam generator, Figure 4.1, basically consists of a shell with domed heads enclosing a bundle of steam tubes. A bellows is installed in the outer cylinder to alleviate stresses due to axial thermal expansion differential between the tube bundle and shell. The tube bundle is supported approximately every 30 in. by 0.75 in. thick tube spacers which are supported by tie rods. The outer cylinder is rigidly supported near one end and radially supported near its other end.

Preliminary examination of the Westinghouse and Foster Wheeler straight tube steam generator conceptual designs revealed several areas of the designs that could prove troublesome over the 40-year life of the plant. The following areas were identified for thermal and structural analysis:

- 1) The double-wall tubes (DWT) are prestressed during fabrication to assure high contact pressure between the two walls and good overall conductivity. Each DWT must be prestressed to obtain an average gap between the two wall (~0.2 mils average) to assure uniform heat transfer characteristics. Relaxation of the prestress with time or due to severe thermal transients could increase the gap thereby reducing contact pressure so as to degrade heat transfer. Such an effect could act in conjunction with (2) to impair structural integrity. The ASME Code does not currently consider cold-worked 2-1/4 Cr-1 Mo (required for forming prestressed DWT), so additional material properties may be required.
- 2) The straight tube design has no tube-to-tube compliance to accommodate differential expansion that can be caused by flow maldistribution on either the shell side or tube side. This can be especially critical during startup and shutdown.

STEAM GENERATOR

CROSS SECTION VIEW OF GENERATOR (TUBES NOT SHOWN)

TYP. TUBE CROSS SECTION

SECTION A-A (TUBE LOCATION)

SHELL MATERIAL

304 ANNEALED CREE STEEL
 $*E = 23.8 \times 10^6$ @ 700 °F (CONSERVATIVE)
 DENSITY = .3 LB/IN (SLIGHTLY CONSERVATIVE)
 $G = 9.154 \times 10^6$ @ 700 °F
 $\textcircled{D} S_M = 13400 \text{ PSI}$
 * CONSERVATIVE - $\textcircled{D}$ 1974 ASME CODE, SECTION III, DIV I, SUBSECT
 NA, APPENDIX I, TABLE 1-C.10 GIVES $E = 24.7 \times 10^6$

Figure 4.1

Typical Straight Double Wall Tube Steam Generator

- 3) A shell bellows is required for shell-tube bundle compliance. This bellows design appears to require an advancement in the state of the art for bellows analysis, fabrication, and qualification.
- 4) Both vertical and lateral seismic support appear difficult with the shell bellows with either the Foster Wheeler top support or the Westinghouse bottom support designs.
- 5) The Westinghouse design does not call for welding the outer tube to the tubesheet on the sodium side. This feature could lead to crevice corrosion.

A limited structural analysis of the straight DWT steam generator under various load combinations was performed. The analysis considered static, seismic, thermal, and pressure induced loads. An analysis was performed on only the most critical areas in the steam generator assembly. Static and seismic internal loads were obtained from the Atomics International "DRIPS" finite element computer program. The computer model, Figure 4.2, is composed of elements representing the outer steel shell, rigidly supported near one end. The steam tube bundle was represented by 3,334 tubes radially supported at approximately 30 in. spans by 0.75 in. thick tube spacers. The bellows were represented in the model as springs. The 0.75 in. thick tube spacers were input in the model as radial links. The weight of liquid sodium in the steam generator was considered.

4.2.1 Loss of Prestress

Excess tube wall temperature gradient can cause loss of prestress in a DWT and, under extreme conditions, could increase the gap excessively between the two tubes to reduce heat flux for a given hot/cold fluid temperature difference. One of the double-walled EBR-II superheaters

IDEALIZED STRUCTURE

CIRCLED NUMBERS INDICATE COMMON INTERFACE POINTS

A = L.H. HEAD DOME
 B = L.H. HEAD BLOCK
 C = L.H. HEAD CYLINDER SLOPE
 D = L.H. HEAD INNER CYLINDER
 E = L.H. HEAD OUTER CYLINDER
 F = BELLOWS
 G = BELLOWS INNER CYLINDER
 H = CYLINDER
 I = TUBE BUNDLE
 J = TUBE SUPPORT
 K = R.H. HEAD CYLINDER SLOPE

L = R.H. HEAD OUTER CYLINDER
 M = R.H. HEAD INNER CYLINDER
 N = R.H. HEAD SUPPORT
 O = R.H. HEAD BLOCK
 P = R.H. HEAD SHORT OUTER CYLINDER
 Q = R.H. HEAD DOME

Figure 4.2
 Idealized Structures Used in Computer Model

(No. 712) is showing anomalous performance after approximately 10-year's operation that can be explained by loss of prestress in the double-wall tube. At this time, it is not known for certain whether this unit is a metallurgically bonded or a mechanically bonded duplex tube structure. There is strong evidence that it is the latter as the exhibited erratic performance (i.e., steam temperature decreasing with increasing power near full power and the reverse with decreasing power; and steam temperature increasing with increased steam pressure) can be explained by change in the gap between the walls of mechanically bonded tubes. This could not occur in good quality, metallurgically bonded tubes since the metallurgically bonded double-wall tubes become essentially the same as thick, single-wall tubes.

The loss of prestress can occur as a result creep resulting from long duration exposure to high temperatures or to large delta temperature through the wall as illustrated by Figures 4.3 and 4.4. Figure 4.3 shows the effect of temperature on loss of prestress. At temperatures above 930⁰F, the loss of prestress would occur in less than 1 year, at 850⁰F in 10 plus years, and at temperatures below 800⁰F in 40 or more years. This would indicate mechanically bonded DWT would not lose prestress at saturated and Sulzer cycle evaporator temperature conditions but would at Sulzer and Benson cycle superheater temperature conditions. Figure 4.4 illustrates a mechanism for explaining the behavior of the EBR-II superheater. It gives strong support to a mechanism involving relaxation of preload between the tubes with operating time which, in turn, causes a reduction of the interfacial pressure between the tubes with consequent reduction in contact heat transfer coefficient giving rise to erratic thermal behavior.

4.2.2 Flow Maldistribution

Maldistribution of sodium flow can cause excessive temperature differences between perimeter and inner tubes of the bundle which can

**EFFECTS OF TIME AT TEMPERATURE
IN REDUCING COLD WORK
IN ANNEALED 2 1/4Cr-1Mo ***

* BASED ON ESG DATA ON ANNEALED 2 1/4Cr-1Mo
COLD WORKED TO 4-6%; TIME RELATED TO
THAT REQUIRED TO REDUCE COLD WORK TO NEAR ZERO,
BASED ON CORRELATING VARIABLE BEING HARDNESS.

Figure 4.3

* BASED ON 12/77 ORNL CREEP EQUATION
FOR ANNEALED 2 1/4Cr-1Mo

Figure 4.4

cause buckling of a DWT. Maldistribution of steam flow can occur from inner tube ID tolerance.

The results of the flow maldistribution study show that (1) most of the sodium pressure drop through the tube bundle is across the tube spacers, and (2) the excess sodium flowrate at the perimeter varies from 140 to 258% when the tube spacer gap varies from 1/8 to 1/2 in. The tube spacers have a 1/8 in. gap between the edge of the tube spacer and the steam generator shell when there is no shell eccentricity and the tube spacer is perfectly centered in the shell. The 1/2 in. gap corresponds to a shell with 1/4 in. eccentricity.

The potential maximum average tube temperature between an outer perimeter and the average tube bundle was calculated using the SETS computer program. It was assumed in this analysis that a tube of minimum flow area (within drawing tolerances) would be analyzed in conjunction with the excess sodium flow. The results of this analysis are summarized in Table 4.1 for the DWT-type steam generator:

TABLE 4.1
TUBE DELTA TEMPERATURE COMPARISON

Steam Cycle	Potential Tube-Tube ΔT ($^{\circ}\text{F}$) From Flow Maldistribution		
	Steam Flow/ Tube ID Tolerance	Sodium Flow/Tube Spacer to Shell Gap*	Total
Benson	61	57	118
Saturated	2	66	68
Sulzer (evaporator)	12	64	76

* Difference between tube bundle perimeter and average tube.

This table shows (1) excessive tube-tube delta temperatures can occur in all cycles, (2) a minimum tolerance cross sectional area tube increases the average temperature of a tube much more in the Benson steam cycle than in the saturated and Sulzer steam cycles (i.e., 60°F for the Benson steam cycle compared to 12°F for the Sulzer evaporator and 2°F for the saturated), and (3) the increased sodium flow increases the average temperature of a perimeter tube, but here the cycle-to-cycle differences are very small.

Structural analysis summarized in Figure 4.5 indicates the DWT stresses were critical when one tube was hotter or colder than the average tubes. Welds attaching the DWT to the end tubesheets were also critical for thermal induced stresses. The dual wall tubes and tube welds to the end tubesheets are critical for combined thermal and seismic loads. The DWT's are critical in the column buckling mode when one tube is hotter than the average tubes by more than 58°F . The possible ΔT 's are 118, 76, and 68°F , respectively, for the Benson, Sulzer, and saturated steam cycles indicating this type of failure could occur. Single-wall portion of the DWT's (as occurring between the two top tubesheets of the W design, Figure 4.5) are critical in column buckling when one tube is hotter than the average tubes by more than 40°F . The possible value is normally less than 20°F but can be greater than 200°F for a blowdown transient. The above buckling failure temperature differences are for thermal induced stresses only. The DWT's are critical in shear when one tube is hotter than the average tubes by more than 30°F or cooler by 30°F . These values are lower than possible in all cycles indicating this type of failure could occur. Single tubes are critical in shear when one tube is hotter or cooler than the average tubes by more than 70°F . The possible value is normally less than 20°F but can be up to 170 and 200°F , respectively, for rapid feedwater temperature change and blowdown transient. The above shear failure temperatures are for thermal induced stresses only.

DWT STRUCTURAL FAILURE CAN OCCUR IF TEMPERATURE DIFFERENCE EXCEED THESE LIMITS

Figure 4.5 DWT Tube-Tube and Tube-Shell Allowable and Potential Temperature Difference

4.2.3 Seismic Effects

Results from the finite element model revealed that the most severe seismic induced stresses combined with internal pressure induced stresses occur in the outer shell at the head-to-cylinder assembly transition and in the DWT near the bellows. It was determined that the tubes or their welds to the tubesheets are not critical for seismic and inertia loads acting alone. However, they are critical for these loads combined with thermal loads. The dual wall tubes will fail in the buckling mode under combined loading when the tube to average tube ΔT reaches more than 25°F . Single-wall tubes will fail in the buckling mode under combined loading when ΔT reaches more than 20°F . It is estimated this would only occur for a blowdown transient. Seismic loads have a minimal effect on the tube welds.

An analysis of the transition area between the steam generator outer cylinder assembly to the head assembly indicates excessive combined stresses due to an internal design pressure of 300 psi and seismic loading. Primary stress levels of approximately 40,000 psi exist in the transition radius. This situation may be relieved by local redesign.

4.2.4 Bellows

The Westinghouse and Foster Wheeler Steam Generators feature externally reinforced shell bellows. Reinforcement is required to provide acceptable design margin against bursting under design and faulted condition pressures. However, reinforcement reduces fatigue life and predictability. Preliminary analyses indicate the bellows without reinforcement do not satisfy Class 2 primary stress limits under design conditions (pressure = 300 psi, temperature = 700°F) nor Class 1 faulted event conditions (pressure = 600 psi, temperature = 700°F).

Gapping distortions between the bellows and the external reinforcement can result from thermal lags when the bellows is subjected to thermal transients. It is very difficult to predict load transfer under these conditions. Load transfer predictability problems also exist due to unknown gapping existing at installation. Also under axial extension, reinforcement will separate from the outer surface of the bellows and will not provide full constraint while under axial compression. In addition, the reinforcement will reduce the bellows flexibility and increases motion stresses thus reducing fatigue life.

The fabrication of the 0.5-in. thick bellows required presents difficulties since single ply bellows with thicknesses in excess of approximately 0.14 in. have yet to be formed by commercial vendors. Possible methods of fabrication exist, such as machining and explosive or spin forming, but these may have problems providing a well-controlled geometry which is critical to uniformity of convolution response and predictability of response. Also, making one convolution at a time and circumferential welding presents a long length of weld where fatigue cracking from a defect becomes a concern. In addition, because the bellows would be made of Incoloy (or similar material), obtaining crack-free dissimilar welds around the complete shell diameter may be a major problem.

4.3 SODIUM-WATER REACTION

Due to the limitations in scope of effort, sodium-water reactions were only estimated for the various designs, based on previous work. In general, the large units envisioned for future LMFBR's reduce the magnitude of sodium-water reactor (SWR) pressures relative to smaller units previously built. Larger diameter tubes and higher steam pressure both tend to create higher SWR pressures as they release more water faster into the sodium. Due to compensating effects, the low and high steam pressure designs should have about the same SWR pressures. However, the

much larger quantities of water released by larger diameter tubes (~10x) in the low-pressure designs (and the B&W high-pressure designs) will produce more reaction products and greater potential for secondary damage. The presence of a gas space in the B&W design may reduce the pressure pulse during a sodium-water reaction if it occurs in the upper region of the tube bundle. A cursory analysis indicates the gas space will not significantly attenuate a tube break accident if it occurs in the lower part of the steam generator. The duration of the pressure pulse (~1500 psia) is too short to be transmitted to the gas space. Therefore, the maximum design pressure within the steam generator may not be significantly affected by the gas ullage since a rupture in the lower section of the tube bundle is also a design condition. Estimated values for sodium-water reaction shell pressure are listed in Table 4.2:

TABLE 4.2
SODIUM-WATER REACTION SHELL PRESSURE (psi)
(Single tube failure)

	Benson	Sulzer (Evaporator)	Saturated
AI SWT Hockey Stick	450	490	520
B&W SWT Helical Coil*	840	1050	650
W Straight DWT	480	505	350
F-W Straight DWT	450	490	495

*At lower end of SG

4.4 OTHER CONSIDERATIONS

4.4.1 Sodium Nozzles

In the AI, B&W, and FW designs, the sodium nozzles are provided with thermal sleeves to protect against large temperature differences as

a result of sodium transients. The Westinghouse design, Figure 3.13, does not presently show comparable thermal protection and, thus, is susceptible to potential initiation and propagation of surface flaws and cracks.

4.4.2 Tube Supports

In the Foster Wheeler design, Figure 3.15, the tube support plates are hung by tie rods from the upper tubesheet. In addition to dead weight, the tie rods carry most of the sodium pressure drop through the steam generator. As a result, creep and creep ratchetting of the tie rod assembly may be a problem. Also, in the event of sodium-water reaction, reverse loading on these tie rods may cause tie rod column collapse and potential massive damage to tubes.

An assessment of the supports proposed for the helical tubes and tube bundle indicates they have taken care of all the structural problems (seismic and thermal expansion loads) anticipated for this type structure provided there are no gaps between the supports and tubes they clamp. Tests of a Monju plant steam generator model showed that excessive wear can occur due to thermal expansion if the tubes are not tightly clamped by the tube supports.

4.4.3 SG Support

The Foster Wheeler SG support ring appears to have insufficient ring bending stiffness for seismic loading. A thicker flange with additional reinforcement regions in the shell adjacent to the ring would carry the loading with lower primary stresses and would provide higher natural frequencies. For the B&W integral support skirt, the section change and intersection of the skirt with the vessel wall appears too abrupt and may present problems during seismic excitation.

4.4.4 Welds

Westinghouse and Foster Wheeler employ fillet welds for the tube-to-tubesheet connection. Fillet welds are typically sites for crack initiation, particularly in corrosive environments. In this application, radiographic inspection results are difficult to interpret. AI's experience suggests that these welds should be avoided for sodium service. The ASME Code requires full penetration welds for Class 1 pressure boundaries.

The Westinghouse design, Figure 3.13, features numerous welds located at structural discontinuities and section transition regions. Radiographic inspectability of these locations is difficult, especially the feedwater header to lower tubesheet weld. In contrast, the hockey stick and helical coil designs provide section uniformity surrounding a weld to assure convenient and meaningful radiography. The Westinghouse design employs the most structural discontinuities at weld joints, e.g., sodium inlet and outlet nozzles and annulus.

5.0 SCALE-UP POTENTIAL

5.1 HOCKEY STICK

Under the LMFBR Base Technology Program, an extensive study (Reference 1) was made which verified that the basic hockey stick steam generator concept adopted for the CRBR is readily scalable to sizes suitable for large (1000 MWe) plant application. This detailed review covered all important aspects: availability of material forms in the required sizes, structural characteristics, thermal/hydraulic characteristics, fabricability, inspectability, and ability to transport the units from shop to plant site.

In scaling up the hockey stick from the 157-tube, 35-MWt MSG to the 757-tube, CRBR units and then to larger commercial plant size units, the materials, tube sizes, tube pattern, and tube support characteristics remain essentially the same. Basically, only the number of tubes, shell diameter, and tubesheet and nozzle sizes increase as the component size increases. The hockey stick tube lengths may be different depending upon whether the unit operates in the once-through or recirculation mode. The concept is easily adaptable for either approach. This was illustrated when the MSG, which was designed for once-through operation, was also tested under evaporator mode conditions when the latter approach was adopted for CRBR. Present U.S. production capability can supply all materials needed for commercial-size hockey stick units.

It is projected that the operating conditions for the first generation large plants will place the steam generator in a more moderate environment than exists in CRBR, and key structural design margins will be larger. For example, detailed stress analyses indicate that maximum creep-fatigue effects in the larger unit at Benson cycle conditions will be less than half of those in CRBR. Also, peak heat fluxes in the departure from nucleate boiling (DNB) region will be lower under commercial

plant conditions, thus providing favorable corrosion and structural margins with respect to safe operating experience.

The fundamental thermal/hydraulic and tube structural characteristics of this design are independent of component size, and these features were fully validated in the highly successful 30-MWt modular steam generator (MSG) and other supporting development testing. The MSG tests, together with full-size CRBR hydraulic model tests, confirm that thermal/hydraulic design characteristics of the concept are favorable and predictable. Subsequent to its performance and endurance testing, the MSG underwent extensive post-test examination and was found to be in flawless condition. It was then refitted and installed in the large leak test rig where it recently completed a program of six tests in which large tube ruptures and Na-H₂O reactions were deliberately introduced. The MSG is now undergoing destructive examination. Findings show the unit accommodated the guillotine tube failures with minimal consequences. Prototype testing of the CRBR program will further verify the integrity and scalability of this design concept. AI is currently evaluating the possible use of the CRBRP prototype test to also serve as a proof test for a commercial size unit by adding additional testing at simulating Benson cycle operating conditions.

While there is no foreign experience on the hockey stick tube steam generator, much of the foreign and U.S. experience on other straight tube steam generators is applicable to the hockey stick concept, for example, vibration, fretting, and wear experience from other straight tube designs such as PFR U-tube, SNR straight tube prototypes, BN-350 U-tube superheater and AI ALCO-BLH sine wave tube test.

The basic fabrication, inspection, and quality assurance procedures are essentially the same for large plant hockey stick units as for the CRBRP units. Finally, shipment of plant-size units from shop to site is within the current rail and barge transport capability in the U.S.

Overall, the basic hockey stick concept is simple, reliable, and well substantiated by test. It can be readily scaled to sizes suitable for large commercial plants.

5.2 HELICAL COIL DESIGN

The U.S. lacks significant experience with helical coil steam generators designed for LMFBR application. To date, the U.S. has built and tested a 1/2-MWt, 2-tube unit and built, but not tested, a 25.2-MWt unit. The planned 70-MWt SCTI test of the B&W design model will be the first significant test in the U.S. However, there is much foreign experience with helical coil steam generators. The Japanese have tested two sets of 40-MWt evaporator and 10-MWt superheater modules. The French have tested a 49-MWt unit, and the Germans/Dutch have tested a 52.75-MWt unit. In addition, these countries have tested over 10 smaller units in the range of power from 0.5- to 6.0-MWt. Exchange agreements between the U.S. and these countries may make this extensive foreign experience available to support a U.S. helical coil development program.

The helical coil steam generator design is attractive for large-scale plant units and hence the strong foreign interest in the concept. One advantage of the concept is the coiled tube allows a more optimal L/D ratio for the heat transfer surface. In a helical coil unit, an L/D ratio of 3500 can be achieved, while in a straight tube design, the limit is ~2000. This allows higher heat transfer rates and lower heat transfer surface areas. In the helical coil design, the number of tubes is reduced, providing at least a potential for fewer tube welds and lower unavailability. The fewer tubes also allows tubes to be brought out through the shell to smaller tubesheets, eliminating the need for large tubesheets and tube-tubesheet welds in sodium. Another advantage to the helical coil design is the more efficient packaging that can potentially be attained over the straight tube unit. The major advantage of this is in shipping and building layout.

Scale-up problems for the helical coil concept relate primarily to the increased complexity with size. Manufacture of the large units requires massive tooling necessary to rotate the unit as the tubes are wound onto the unit. The pitch and number of tubes wound onto the unit changes with each additional row of tubes added. The problems for winding 7 rows onto the 70 MW must be sealed to a range of 18 to 26 row (depending on steam cycle) for the commercial units. Strict quality control must be maintained during this heavy machinery operation to insure a high integrity sodium-water boundary. Also, since very long tubes are required, intertube welds are necessary. Several other in-sodium tube welds are required for the "pigtailed" on either end of the helical portion of the tube. In effect, the helical coil unit has more inches of tube welds than the hockey stick tube unit. As the concept is scaled to large sizes, the complexity of the pigtailed increases, making a high-integrity design more difficult to maintain.

Direct scale-up of the helical coil design to large sizes may cause excessive risks. The Dutch/German team has selected small helical coil units nearly the same size as their 52.75-MWt test model for SNR-300, and for SNR-2 they propose scaling up only to 210-MWt. For Monju, the Japanese are scaling to 163 MWt evaporator units from their 40-MWt tests. On the other hand the French are taking a much bolder approach in scaling from a 49-MWt test unit to a 750-MWt plant unit for Super Phenix.

5.3 DOUBLE-WALL TUBE DESIGNS

The double-wall tube concept has been utilized in several small, early plant designs and has failed to gain significant attention for large-scale plants except fairly recently in the U.S. The most noted use of a double-wall design is for EBR-II, which employs ten identical steam generator modules, eight operating in a recirculation evaporator mode and two in a superheater mode. The units operate at an average power of 6.25 MWt. Two types of double-wall tubes are employed in these

units, mechanical bond and metallurgical bond. The mechanical bond is similar to that proposed for use in the W and F-W units, except no provision is made for interwall leak detection in EBR-II.

EBR-II steam generator experience has been generally good. There has been only one tube leak, which occurred at a steam weld early during the plant life. A potentially more serious problem has developed, however. The mechanically bonded tubes have apparently relaxed, and the tubes separate under certain heat transfer conditions causing fluctuations in outlet temperature and variation in overall performance by as much as 43%. The prestress applied to the tubing during manufacture has apparently relaxed after 10-15 years of operation at ~800°F. While the extent of the problem and the possible consequences of it have yet to be determined, it certainly flags a problem for a double-wall tube design in a superheat steam cycle and a potential one for a saturated cycle.

The most obvious reason for the lack of attention previously paid to the double-wall tube concept is the cost/benefit ratio of that design. The cost of a double-wall tube design is thought to be ~2 times that of a comparable single-wall unit. The major benefit of the double-tube concept is the prevention of steam-to-sodium leaks. However, it is anticipated that the double-wall tube unit will have more sodium-air and steam-air leaks resulting in a higher overall unavailability. In addition, advances in acoustic leak detection/location instrumentation are expected to provide early leak detection and greatly reduce the potential for major leaks in a single-wall tube unit.

Since there have been no large double-wall tube designs tested to date by any country, the scale-up to large size plant units will be based solely upon the EBR-II experience and on the planned 70-Mwt SCTI double-wall tube test. EBR-II experience is of limited value, since in a large unit the tube lengths are much longer and as a result flow distribution and tube-tube/tube-shell temperature differences will be of a more significant nature requiring closer attention. In addition, the

EBR-II experience with relaxation of the prestress in the tubes indicates that a different type of double-wall tube (or material) should be used. These factors significantly limit the applicability of the EBR-II as a basis for scale-up. The double-wall tube scale-up will, therefore, be based almost solely upon the planned 70-MWt double-wall tube test.

5.4 COMPARISON OF SCALE-UP POTENTIAL

The amount of worldwide experience with steam generators is presented in Table 5.1 and ranked by type of design and power level. Planned test or plant operation is indicated by single parenthesis and proposed plant unit sizes by double parentheses. Visual comparison of scale-up steps for the three concepts is provided in Figures 5.1 (hockey stick), 5.2 (helical), and 5.3 (double-wall tubes).

The double-wall tube concept shows no applicable experience to date other than EBR-II. EBR-II is very small in comparison to the planned and proposed tests for double-wall tubes and is not considered to be a proof test of the double-wall design. In addition, as discussed above, the EBR-II operating experience has shown problems with the mechanically bonded tubes.

The scale-up of the DWT designs to commercial sizes from EBR-II experience, requires too great a change of size and design to give credibility. The size difference is illustrated in Figure 5.3. Examination of Figures 5.4, 3.13, and 3.15 illustrate differences in design details. Table 5.2 lists major differences in design that significantly limits experience transfer. Experience with PWR units⁽⁹⁾ has shown that changes in seemingly unimportant details; for example, lattice type tube supports versus drilled plates, expansion of tubes versus rolling, or installation details of the inlet plenum divider plate can cause major problems. Therefore, there is no real starting point upon which a commercial DWT design could be based, and significantly greater experimental efforts are required to place the DWT design of the same technological level as the single tube hockey stick design.

TABLE 5.1

WORLD WIDE STEAM GENERATOR TESTING EXPERIENCE FOR HOCKEY STICK, HELICAL COIL AND DOUBLE WALL TUBE DESIGNS

NOTE:

() UNDER CONSTRUCTION
 (()) PLANNED OR PROPOSED

Rockwell International
 Energy Systems Group

FIGURE 5.1

HOCKEY STICK SCALE-UP

FIGURE 5.2
HELICAL COIL SCALE-UP

*Super Phenix operates at a higher log mean temperature difference and utilizes internal space more efficiently than the commercial design.

FIGURE 5.3

DOUBLE WALL TUBE SCALE-UP

Rockwell International
Energy Systems Group

EBR-II EVAPORATOR ASSEMBLY

Figure 5.4

TABLE 5.2

MAJOR DIFFERENCES BETWEEN EBR-11
AND COMMERCIAL SIZE DWT STEAM GENERATORS
SIGNIFICANTLY LIMITS EXPERIENCE TRANSFER

<u>MODIFIED ITEM</u>	<u>AFFECTED CHARACTERISTICS</u>
• NUMBER OF TUBES INCREASED FROM 73 TO 1014	FLOW MALDISTRIBUTION, THERMAL/ HYDRAULIC PERFORMANCE
• HEATER LENGTH INCREASED FROM 24.5 FT TO 79 FT	TUBE-TO-TUBE/TUBE-TO-SHELL EXPANSION*
• SHOCK TUBE INCREASED FROM 4 FT TO 15 FT	CORROSION, PRESSURE TUBE SUPPORT, FRETTING WEAR RISK*
• REVISE SHOCK TUBE SUPPORT	DESIGN UNCERTAINTIES
• REVISED TUBE SUPPORTS	TUBE VIBRATION AND WEAR*
• BELLOWS IN SHELL	DESIGN COMPLEXITY, RISK
• OUTER TUBE WELD SHIFTED TO SODIUM SIDE	LEAK DETECTION

*ALSO PERFORMANCE AND THERMAL/HYDRAULIC

The hockey stick concept shows significant testing experience with the AI-MSG and upcoming CRBRP prototype test. The individual tube-tube expansion capability of the hockey stick design gives more credibility to the adequacy of a subscale test. In addition to experience with the hockey stick design, a good deal of the design and operation experience with other straight tube concepts is applicable, notably PFR U-tube experience, SNR straight tube prototypes, BN-350 U-tube superheater experience, and the ALCO-BLH sine-wave tube test at SCTI. These tests and plant operating experiences have covered power levels of 17 to 30 Mwt.

The helical coil steam generator concept has been the most heavily tested concept up to 53 Mwt size. Scale-up to a large commercial size is underway in support of Super Phenix, although it is a very significant increase in the state of helical coil design. U.S. experience with helical coil designs is slight, and a sound commercial plant steam generator design would have to be based upon foreign experience in addition to the planned 70-Mwt helical coil test at SCTI.

In summary, the hockey stick and helical coil steam generator designs have superior test experience upon which to base a commercial plant design. The hockey stick has the strongest U.S. development background with the AI-MSG and CRBRP experience. The helical coil unit has little U.S. experience, but this is compensated by strong programs in foreign countries assuming exchange agreements could be reached making this experience available. The double-wall tube concept experience is limited to the EBR-II units and the planned 70-Mwt test. The scale-up to a commercial size unit will be based almost solely upon the 70-Mwt test, due to the limited value of the EBR-II experience relative to commercial sizes.

6.0 MANUFACTURING

An overall review was made of the four types of steam generators with the following conclusions being made based upon present knowlege of manufacturing technology. It must be noted that fabrication technology is constantly improving, and what appears difficult at this date may be manufactured with ease in the future. General comments on manufacturing problems for each design follows.

6.1 ATOMICS INTERNATIONAL HOCKEY STICK DESIGN

This configuration represents fabrication technology, which will not change with design scale-up. To enlarge the present model will require only a scale up of the tooling being used. Special tooling and processes have been developed to facilitate inspection and fabrication, including such items as in-place-radiography complete with its own shielding and advance state-of-the-art, out-of-position welding, for its shell. The computerized welding tool rotates 180 degrees both ways from the starting point around the shell. In addition, the flexibility of its tube-to-tubesheet design for scale up is significant.

The only problem area is the hockey stick shape. It presents special fixture problems during assembly and shipping that the other designs do not have.

6.2 BABCOCK & WILCOX HELICAL TUBE DESIGN

The Babcock & Wilcox Steam Generator was reviewed for fabrication producibility, and the following problem areas are mentioned for consideration.

- 1) The use of the helical tube concept requires the engineering, fabrication, development, and building of a large special

machine and tooling to coil the tubing into the desired configuration. The welding of tubes to the four tubesheets to the tube end configurations followed by assembly into the shell represents very difficult fabrication procedures as well as complex tooling.

- 2) Although this design allows the development of production techniques and tooling to roll the tubes into a spiral configuration, this advantage is negated by the large number of complex tube bundle and tube vibration supports. These supports require the tubes to be rolled and positioned adequately to accept the matching support rods. Long assembly times are envisioned from problems occurring in positioning and hand forcing the tubing to their required position. If the tooling problem to prevent springback of tubing is not resolved, it is anticipated that assembly problems will occur while installing tube spacers for the tube bundle supports. A bowed condition could develop with the force of the tubing acting against the long support bars (~40 ft). The machining of these long support bars will require special fixturing to machine and inspect to guarantee that design tolerances are met.
- 3) The tubesheets will have to be fixtured and held in the exact positional location. The individual coils and "pigtailed" (lead tubing) must be cut to length and prepared for tube-to-tubesheet welding after forming.
- 4) It is assumed that the two closure heads will be slipped over their respective four tubesheets with closure welds made at this time. It appears that design changes must be made to accomplish this in order to have closure welds which can be radiographed and repaired.

- 5) This configuration will require the longest tubing (up to 535 ft) for fabrication. It can be assumed that additional problems will occur in obtaining and maintaining high quality tubing as these lengths are made by tube-to-tube welding of up to 15 shorter length tubes.

6.3 WESTINGHOUSE STRAIGHT DOUBLE-WALL TUBE DESIGN

The Westinghouse Steam Generator concept (Typical Drawing L-LM-1024-AI) has been reviewed for fabrication producibility with the following problem areas mentioned for consideration.

- 1) The prestressed tube concept represents a difficult fabrication technique as the inner and outer tubes are coextruded with about a 0.2 mil interference fit to give good heat transfer characteristics. Considerable research and development must be done to develop methods to produce this assembly on a production basis. The slightest imperfection or variation in tolerancing could cause binding during assembly. The process appears extremely expensive and it is hard to visualize what methods could be used to insert and pull or guide the tubing during assembly.
- 2) The requirements to use four 0.020-in. radius grooves at 90 degrees on the ID of the outer tube for water-to-helium or helium-to-sodium leak detection is a questionable application. A development program would be required to attain and maintain the required grooves on a production basis. The configuration can be attained by an extrusion or broaching process. In either case, the process must be controlled to tight tolerances. The present indicated tolerances appear too tight to be feasible.

- 3) Westinghouse states that cladding of the tubesheets is required to obtain acceptable quality welding of the tube assemblies to the tubesheet. The cladding buildup and subsequent machining and inspections is an expensive process. It will be difficult to guarantee a leak-free path area in the cladding region.
- 4) The fillet welding of the OD tubes to the tubesheet represents a hand process. This creates a control problem in getting a uniform weld completely around the tube in the small area between tubes. The small volume between tubes may cause a temperature and pressure rise causing the closing fillet weld to actually blow out the weld. Inspection and hand repair will be difficult.
- 5) The welding of the tubesheet must be done with the flanged and dished head in position while providing insufficient room to do the welding. It is difficult to perceive how this part could be assembled to the unit after welding.
- 6) The use of a bellows configurations requires the development and test of a unit before manufacture. It can be assumed that this is in the realm of present technology with fabrication process development, but the design as shown will require the bellows portion to be slipped over the tubing bundle before the tube-to-tubesheet welding is accomplished.

6.4 FOSTER-WHEELER STRAIGHT TUBE DESIGN

The Foster-Wheeler/GE concept has the following potential problem areas:

- 1) The drilling of 3/8 diameter holes between tubesheet holes required special techniques to accomplish for the size, depth of hole, and degree of accuracy. It is a gun barrel drilling operation which is a specialty in itself

and requires special tooling. It is anticipated that for this size hole, the drill could drift approximately .002 in. per in. Sulphur chlorinated oil is required. The use of this oil is prohibited for most nuclear applications. If used, its removal by cleaning is very difficult and expensive. The right angle hole required for each tubesheet hole will require special tooling and this may be impossible to accomplish at this time. The intersection of the holes will create a burr which will be very difficult to remove and to inspect for the same reason. It is possible that the metal removed during deburring will lodge in the system necessitating an elaborate inspection and cleaning procedure to guarantee complete cleanness.

- 2) The use of a duplex tube concept requires a special procedure, fabrication, and inspection to join the inner and outer tube as noted above for the W design. It mentioned that the tube assembly is to be prestressed. This will require special procedures and possibly development tests to guarantee that the prestress will not relax during the life of the plant.
- 3) The establishment of a weld overlay on the internal diameter of the shell to provide a seal surface for each tube spacer piston ring could become a fabrication and assembly problem. The length of the rolled and welded shell will be determined by available sub-arc welding equipment and available machine tool capability to machine the inconel overlay to mate with proposed support plates. The number of shell welds will be determined by machine capability.
- 4) The joint configuration as proposed for the orbital welding of the outer tube with subsequent NDE inspection by magnetic particle of the tube back chip and magnetic

particle and radiograph inspection of the final weld is impossible. The back chip operation cannot be done and radiograph will not provide sufficient definition to make an analysis. The joint definition as shown does not lend itself to orbital welding and there is always the risk of actually welding the inner and outer tubes together at this point.

- 5) The 58% flow hole pattern represents a very expensive fabrication process to attain the configuration as described. Considering the number of flow hole areas, thickness of spacer plates, and configuration, it represents a very high cost item.

A comparison of the four designs shows that the AI hockey stick design is best for all production rates. The B&W helical tube design is penalized for all production rates by the large size special machine required to coil the tubing and by the required hand fitup of the complex tube vibration and bundle tube spacer supports.

7.0 INSPECTION/MAINTENANCE

7.1 TUBING

7.1.1 Inspection

Inspection comprises two categories: 1) inservice inspection and 2) inspection to determine leak location. In a single-walled tube, visual and/or ultrasonic (or eddy-current - depending on material) techniques can be used to detect developing failures. For double-walled tubes, visual inspection is useful for the inner tube, but useless for the outer tube. The usefulness of eddy current or ultrasonic techniques for double-walled tubes is a development area with questionable chance of success.

Initial detection of leaks in both approaches is about equal but location of leaks is substantially superior for single-walled tubes. However, in operation the DWT design has an advantage if the leak is into the sodium side. Then the leak detection fluid bubbles into the sodium with essentially no harm to performance. Acoustic emission detection techniques can be used to detect the presence of leaks in either approach, although the signature for a double-walled tube leak may be less defined. Pressure decay techniques, which monitor the helium annulus pressure, are the primary leak detection scheme for the double-walled tube, whereas acoustic leak detectors and reaction products monitors (hydrogen detectors), located as shown in Figure 7.1, are the primary single-walled tube leak detection scheme. Several options are available to locate the leaking tube or tubes in a single-walled unit. Ultrasonic monitoring or fiber optic viewing within tubes after localizing the failure by acoustic emission triangulation, is the primary technique. Figures 7.2 through 7.4 show examples of the equipment used for in-tube monitoring. British experience with visual examination indicates this

Steam Generator Leak Detection System

Figure 7.1

9153-1283

FIGURE 7.2 Ultrasonic Leak Detection
and Drive Cable

FIGURE 7.3 Tube Monitor Drive Machine

No. N254T1330003
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Rockwell International
Aircraft International Division

6537-40104

FIGURE 7.4 Ultrasonic Monitor Instrument

6537-40105

technique can very quickly localize the failure. Pressure cycling the shell side while personnel monitor the tube side for visual and/or auditory effects also may quickly locate leaks. The latter technique may also be augmented by local and/or hand held acoustic monitors. The double-walled tube eliminates many of the above options. Outer tube failures are especially hard to locate because there is no access for monitoring individual tubes. Both personnel observation and in-tube inspections are eliminated, unless some ultrasonic technique to detect outer tube failures with a sound inner tube can be developed. The tightly surrounding outer tube also masks the effects of varying pressure to detect inner tube leaks. Acoustic monitoring to locate leaks appears the most likely solution for double-walled tubes. Double-walled construction also introduces fillet welds which are difficult to inspect during fabrication and also produce built-in cracks for trapping impurities and may also lead to failure due to crack propagation or corrosion.

7.1.2 Repair

Tube repair in either configuration will involve plugging the leaking tube. For single-walled tubes, plugging is a three-step operation, as depicted in Figure 7.5. Access to the AI hockey stick design is as depicted in Figure 7.6. Access would be equivalent for double-walled construction, but both outer and inner tubesheets must be plugged. Grooves are also difficult to clean before plugging and a high annulus pressure is required to prevent Na/NaO contamination in the tube-to-tube interface. Once installed, the inner plug is difficult to inspect.

FIGURE 7.5 Single Wall Tube Plugging
Devices

HOCKEY STICK DESIGN PROVIDES EASY ACCESS FOR MAINTENANCE

REFERENCE ACCESS CONCEPT

ALTERNATE ACCESS CONCEPT

Figure 7.6 Hockey Stick Design Provides Easy Access for Maintenance

Rockwell International

Atomics International Division

Double-walled tube plugging is depicted in Figure 7.7. It shows seven steps are required to plug a double wall tube. Several of these operations are very difficult especially operation Number 4. This operation requires the removal of length of DWT in the sodium region to prevent excessive tube buckling stresses.

The double-walled construction also introduces a significant maintenance problem if sodium ever enters the helium annulus (helium pressure normally prevents such entry in the event of a leak). The tube-to-tube interface and grooves provide a trap for reaction products with no natural cleanout. Progressive caustic corrosion may eventually destroy the entire tube bundle and just the presence of the sodium provides an unknown and variable background for any reaction product type leak detectors.

DOUBLE TUBE - TUBE PLUGGING

- ①. ② TUBE CUTTING
- ③ PULL TUBE
- ④ INSTALL EXPANSIVE PLUG
- ⑤ SEAL WELD GROOVE OPENINGS (INDEX ON PLUG RECESS)
- ⑥ PLUG OUTER TUBESHEET

Figure 7.7

7.1.3 Conclusions

Failure of double-walled tubing is much harder to detect and locate than single-walled tubing. Locating the failure is much easier for single-walled tubing. Ultrasonic monitoring or fiber optic viewing, as well as pressure cycling the shell side, are successful techniques for inspecting single-walled tubing, but are not as easily applied to examining and locating failures in double-walled tubing.

Repair of single-walled tubing, once a failure is located, is comparatively straightforward and easy to perform, while cleaning and plugging of double-walled tubing presents more complications. If sodium enters the grooves or the helium annulus, corrosion products will be difficult to remove and progressive caustic corrosion may propagate to the entire tube bundle.

7.2 BELLOWS

The double-walled tube configuration introduces a very large diameter bellows in the steam generator shell which is not required in the hockey stick configured single-walled tube design. This bellows introduces several maintenance and inspection problems which are nonexistent in the single-walled unit.

7.2.1 Inspection

The bellows introduces a dissimilar metal and dissimilar metal weld. This will complicate any shell inspection and introduce new and unknown ISI requirements. Inspection of the closure weld is difficult, if not impossible.

7.2.2 Repair

Code acceptance of bellows appears possible, but fabrication to the code is difficult and expensive. This would be amplified in any repair

operation. In the bellows installation presented, the bellows is not readily replaceable. Replacement requires using a segmented bellows or disassembling the shell. To requalify a unit to code and NRC satisfaction is both a licensing and development item of highly problematic nature.

8.0 RELIABILITY/AVAILABILITY EVALUATION

A cursory reliability/availability comparison of the four steam generator designs was made based on 22 potential failure mechanisms. A reliability comparison based on leaks in tube-to-tubesheet welds was made in considerable detail. These analyses indicated the W and F-W DWT steam generators will have a greater unavailability penalty than the hockey stick and helical SWT designs.

8.1 FAILURE MODES

The relative effects of the following 13 of the 22 failure mechanisms were assessed to be approximately equal for all units based on present information concerning features of each design:

- 1) Corrosion
- 2) Creep Deformation
- 3) Creep Rupture
- 4) Damage from Internal Component Failure
- 5) Decarburization
- 6) Erosion
- 7) Flow Restriction
- 8) Fouling
- 9) Fretting/Wear
- 10) Galling/Friction
- 11) Mechanical Overload
- 12) Thermal Fatigue
- 13) Thermal Overload

The failure mechanisms which might be expected to result in different reliability/availability for the different units are presented in Tables 8.1, 8.2, and 8.3. The parameters which led to these ratings are listed in an abbreviated form on the tables. The results of these ratings are

TABLE 8.1
BENSON CYCLE RELIABILITY CONSIDERATIONS

POTENTIAL FAILURE MODES/RISKS	AI HOCKEY STICK SWT	B&W HELICAL SWT	W STRAIGHT DWT	F-W STRAIGHT DWT
T/T'S WELDS NO/MOD/FT TOT.	+ 4,448/6,552	++ 330/972	-- 6,730/32,211	-- 3,716/14,229
T/T WELDS NO/MOD/FT TOT.	+ NONE	-- 2,119/6,241	+ NONE	+ NONE
BELLOWS WELDS NO/MOD/FT TOT.	+ NONE	+ NONE	-- 2/438	-- 2/386
DWT PRESTRESS RELAXATION	+ NONE	+ NONE	-	-
UNDETECTED MFG. FLAWS	+ EASY TO INSPECT	+ EASY TO INSPECT	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS
FASTENER FAILURE/LOOSE PARTS	+	- MANY FASTENERS	+	+
VIBRATION	++ PARALLEL FLOW CONFIGURATION	- CROSS FLOW	+ SOME CROSS FLOW (DISC-DONUT)	+ SOME CROSS FLOW (DISC-DONUT)
ORIFICE PLUGGING	- SMALLER DIA.	+	+	+
H ₂ O NOZZLES (NO.)	+	- NUMBER(8) & LOCATION	+	+
TOTAL	+9	+2	-1	-1

TABLE 8.2
SULZER CYCLE RELIABILITY CONSIDERATIONS

POTENTIAL FAILURE MODES/RISKS	AI HOCKEY STICK SWT	B&W HELICAL SWT	W STRAIGHT DWT	F-W STRAIGHT DWT
T/TS WELDS NO/MOD(E-S)/FT TOT.	+ (4,868-3,940)/6,531	++ (472-848)/1,757	-- (6,530-5,281)/27,657	-- (8,810-6,866)/31,275
T/T WELDS NO/MOD(E-S)/FT TOT.	+ NONE	-- (3,392-2,360)/7,964	+ NONE	+ NONE
BELLOWS WELDS NO/MOD/FT TOT.	+ NONE	+ NONE	-- 2/410	-- 2/495
DWT PRESTRESS RELAXATION	+ NONE	+ NONE	-	-
UNDETECTED MFG. FLAWS (MORE OPERATIONS)	+ EASY TO INSPECT	+ EASY TO INSPECT	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS
FASTENER FAILURE/LOOSE PARTS	+	- MANY FASTENERS	+	+
VIBRATION	++ PARALLEL FLOW CONFIGURATION TESTED	- CROSS FLOW	+ SOME CROSS FLOW (DISC-DONUT)	+ SOME CROSS FLOW (DISC-DONUT)
ORIFICE PLUGGING	- SMALLER DIA.	+	+	+
H ₂ O NOZZLES (NO.)	+	- NUMBER(8) & LOCATION	+	+
TOTAL	+9	+2	-1	-1

TABLE 8.3
SATURATED CYCLE RELIABILITY CONSIDERATIONS

POTENTIAL FAILURE MODES/RISKS	AI HOCKEY STICK SWT	B&W HELICAL SWT	W STRAIGHT DWT	F-W STRAIGHT DWT
T/TS WELDS NO/MOD/FT TOT.	+ 1,724/6,093	++ 356/1,678	-- 3,264/19,224	-- 1,860/13,149
T/T WELDS NO/MOD/FT TOT.	+ NONE	-- 1,424/6,710	+ NONE	+ NONE
BELLOWS WELDS NO/MOD/FT TOT.	+ NONE	+ NONE	-- 2/353	-- 2/419
DWT PRESTRESS RELAXATION	+ NONE	+ NONE	-	-
UNDETECTED MFG. FLAWS	+ EASY TO INSPECT	+ EASY TO INSPECT	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS	- DIFFICULT TO INSPECT DW, NO. & TYPE WELDS BELLOWS
SHOCK TUBES	+ INCORPORATED	-- FEASIBILITY UNCERTAIN	+ INCORPORATED	+ INCORPORATED
FASTENER FAILURE/LOOSE PARTS	+	- MANY FASTENERS	+	+
VIBRATION	++ PARALLEL FLOW CONFIGURATION TESTED	- CROSS FLOW	+ SOME CROSS FLOW (DISC-DONUT)	+ SOME CROSS FLOW (DISC-DONUT)
H ₂ O NOZZLES (NO.)	+	- NUMBER(8) & LOCATION	+	+
TOTAL	+11	-1	-1	-1

summarized in Table 8.4. It shows the hockey stick design has the least potential for failure when installed in all three steam cycle type plants. The B&W design is the next best for the Benson and Sulzer cycles but poor for the saturated cycle.

8.2 TUBE LEAKS

The expected major source of leakage in the steam generators would be failures in tube-to-tubesheet welds. The frequency of these failures has been estimated to be about 4×10^{-10} occurrences per hour per inch of weld and is considered to depend on the total length of the weld joints. The estimated mean time to repair a tube by plugging is 360 hours. The basis for this evaluation is that if a tube leak is detected, either in a single-wall or double-wall design, the plant will be shut down and the tube repaired.

The results of this analysis are tabulated in Table 8.5. It indicates the Foster-Wheeler and Westinghouse steam generator designs will result in a greater unavailability penalty. The Babcock & Wilcox design is slightly greater in unavailability than the AI design. The backup for these results is presented in Table 8.6 and discussed below.

Atomics International Design

For the Benson cycle steam generators, there are 2,224 - 5/8-in. tubes. For nine generators this requires 20,016 tubes and for 2 welds per tube, there are 78,600 in. of weld. This gives an unavailability of 0.0181, and the mean-time between failures (MTBF) of 4.27 years.

For the Sulzer cycle there are 2,434 - 5/8-in. tubes in each evaporator and 1,970 - 5/8-in. tubes in the superheater, 20,574 tubes in all. In 2 welds per tube, this gives 80,560 in. of weld. This gives an unavailability of 0.0182 and a MTBF of 4.16 years.

TABLE 8.4
POTENTIAL FAILURE MODE COMPARISON*
SHOWS HOCKEY STICK DESIGN BEST

STEAM CYCLE	AI HOCKEY STICK	B&W HELICAL	W DOUBLE WALL TUBE	F-W DOUBLE WALL TUBE
BENSON	+9	+2	-1	-1
SULZER	+9	+1	-1	-1
SATURATED	+10	-1	-1	-1

+ LESS POTENTIAL FOR FAILURE

- GREATER POTENTIAL FOR FAILURE

TABLE 8.5
UNAVAILABILITY COMPARISON AMONG
FOUR CONCEPTUAL STEAM GENERATORS

Designer	Unavailability Difference, %		
	Benson	Sulzer	Saturated
Atomics International (SWT)	Base	Base	Base
Babcock & Wilcox (SWT)	-0.37	+0.16	+0.03
Foster-Wheeler* (DWT)	+6.77	+5.88	+3.65
Westinghouse* (DWT)	+2.26	+6.90	+2.12

*including bellows

TABLE 8.6
STEAM GENERATOR UNAVAILABILITY

	BENSON		SULZER		SATURATED	
	WELD FT	% UA	WELD FT	% UA	WELD FT	% UA
HOCKEY STICK AI						
T/TS	6,552	1.81	6,531	1.82	6,093	1.69
HELICAL TUBE B&W						
T/TS	972	.27	1,756	.50	1,678	.47
T/T	6,241	1.17	7,965	1.48	6,710	1.25
TOTAL	7,213	1.44	9,721	1.98	8,388	1.72
DOUBLE WALL W						
T/TS	32,211	8.33	27,657	7.47	19,224	5.15
BELLOWS	438	.25	407	.23	353	.20
TOTAL	32,649	8.58	28,064	7.70	19,577	5.35
DOUBLE WALL F-W						
T/TS	14,229	3.86	31,275	8.39	13,149	3.58
BELLOWS	386	.22	427	.24	419	.24
TOTAL	14,615	4.08	31,702	8.63	13,568	3.82

FAILURE RATE: 4×10^{-10} FAILURES PER HOUR PER INCH OF T/TS AND BELLOWS WELD

FAILURE RATE: $(.67)(4 \times 10^{-10})$ FAILURES PER HOUR PER INCH OF T/T WELD

MEAN TIMES TO RESTORE: 588 HR FOR TUBES (PLUGGING)
(2)x588 HR FOR BELLOWS

For the saturated cycle, the nine steam generators have 7,758 - 1-1/2-in. tubes or 73,117 in. of weld. The unavailability is 0.0169 and the MTBF is 4.59 years.

Babcock & Wilcox Design

For the Benson cycle, there are 163 - 1-1/4-in. tubes per generator. The tubes are 477-ft long and are constructed of short lengths welded together. The total number of welds per tube are 15, including the tube-to-tubesheet welds. For nine generators this gives 86,413 in. of weld. The unavailability is 0.0144 and the MTBF is 3.89 years.

For the Sulzer cycle there are 236 - 1-1/4-in. tubes in each evaporator with 12 welds per tube and 424 - 1-1/4-in. tubes in the superheater with 10 welds per tube, giving 29,712 tube welds or 116,678 in. of weld. The unavailability is 0.0198 and the MTBF is 2.88 years.

For the saturated cycle, the nine generators have 178 - 2-in. tubes. At 10 welds per tube, there are 100,658 in. of weld. The unavailability is 0.0172. The MTBF is 3.34 years.

Foster-Wheeler Design

For the Benson cycle, the generators have 1,858 duplex tubes. The diameter of the outer tube is 0.8125 in. and the inner tube is 0.6605 in. The outer tubes are welded in two places (two per tubesheet) and the inner tube in two places (one per tubesheet).

It is assumed that any leak will be detected and will thereby result in an outage for repairs, thus, the weld subject to failure amounts to 170,735 in. The unavailability is 0.0408. The MTBF is 1.65 years.

For the Sulzer cycle, the evaporators have 4,410 duplex tubes each and the superheater 5,307 duplex tubes. Tube diameters are 0.8125 and 0.6605 in., with four welds on outer tubes and two on the inner. This produces 608,600 in. of weld. The unavailability is 0.0863. The MTBF is about 0.66 year.

For the saturated cycle, each generator has 930 tubes with the outer tube of 1.5-in. diameter and the inner tube 1.23 in. Welds are the same. For nine generators, this yields 222,456 in. of weld. The unavailability is 0.0382. The MTBF is 1.81 years.

Westinghouse Design

For the Benson cycle, each generator has 3,365 duplex tubes of 0.8125-in. diameter for the outer tube and 0.6605 for the inner. Welds are three on outer tube and two on inner. For nine steam generators, there are 357,595 in. of weld. The unavailability is 0.0858. The MTBF is about 0.94 years.

For the Sulzer cycle, there are 3,265 duplex tubes in each evaporator of diameters 0.75 and 0.628 and 2,641 in the superheater of diameters 0.8125 and 0.6565. There are 309,125 in. of weld. The unavailability is 0.0770. The MTBF is 1.09 years.

For the saturated cycle, nine steam generators have 1,632 duplex tubes each of diameters 1.0 in. and 0.904 in. Welds are the same. The length of welds is 221,858 in. The unavailability is 0.0535. The MTBF is 1.51 years.

9.0 COST COMPARISON

A fabrication cost assessment was made of the four steam generator designs as listed in Table 3.3 for each steam cycle. The assessment included cost for the material, labor, tooling, handling and warehousing required to produce nine steam generator modules.

The relative total cost obtained from this assessment is summarized below in Table 9.1.

TABLE 9.1
RELATIVE TOTAL COST SUMMARY

Steam Generator Design		Steam Cycle		
		Benson	Sulzer	Saturated
AI	Hockey Stick	1.06	1.11	1.00
BW	Helical	1.48	1.71	N/A
FW	DWT	1.93	2.66	1.17
<u>W</u>	DWT	2.49	2.07	1.54

This table shows that the order of selection based on cost only is first, AI SWT hockey stick design for all three steam cycles; second, BW SWT helical design for Benson and Sulzer cycles and FW DWT design for the saturated cycle; third, FW DWT design for Benson, and W for the Sulzer cycle; and fourth, W DWT design for Benson and saturated cycles and FW for Sulzer cycle. It is estimated the FW design would have been the third selection for the Sulzer cycle if the tube lengths had been increased to give a smaller diameter shell. The cost of the AI design was nearly the

same for all three steam cycles with the lowest being for the saturated cycle. As a result, the AI saturated cycle design was used as the basis for comparing the cost of all other designs. Table 9.1 also shows that except for the W Sulzer cycle design, the cost of the steam generators is lowest for the saturated cycle and highest for the Sulzer cycle. This results from the fact that surface areas are lowest for the saturated cycle and highest for the Sulzer cycle, as shown in Table 3.3.

Table 9.2 presents a breakdown summary of the relative cost values plus critical dimensions which affect cost. This summary comparison of the fabrication costs shows the major cost items are shell, headers, and tubes. The cost of tooling, tubesheets, and internals is relatively small compared to these major cost items. In general, the cost of the major items increases with weight, even though the cost/lb varies significantly due to differences in types of construction; e.g., dished headers and nozzle ring are made from forgings that cost two to four times that for straight parts (rolled plate, supports, and flanges) on a weight basis, and the bellows required for the two DWT designs cost two to three times that for straight parts. The most notable exception to this generalization is the cost of the tubes. The helical design had the lowest cost tubes for all three steam cycle plants even though they were heavier than the AI tubes for Benson and Sulzer cycles. This results from its featuring the largest diameter tube for all designs. The cost of tubing on a weight basis increases with length and varies inversely with diameter.

The weight of the shells and headers are proportional to shell length, diameter, and wall thickness and has a significant effect on cost. Examination of Tables 3.1 and 3.3 shows the data obtained on wall thickness was incomplete and inconsistent with respect to the diameter of the shell and expected sodium-water reaction pressure. Therefore, the wall thicknesses were recalculated to the nearest 1/4-in. based on shell diameter only to obtain comparable relative costs. Corrections were not made for differences in sodium-water reaction pressure as this was beyond the scope of this effort.

TABLE 9.2

COST COMPARISON FOR 9 STEAM GENERATOR MODULES

	Benson Cycle				Sulzer Cycle				Saturated Cycle			
	AI	BW	W	FW	AI	BW	W	FW	AI	BW	W	FW
MATERIAL COST												
Shell & Headers	0.146	0.293	0.237	0.207	0.166	0.381	0.213	0.299	0.165	0.299	0.166	0.159
Shell ID/Length/Thickness (in.) (ft) (in.)	72/98/2.0	117/80/3.25	93/72/2.5	82/124/2.25	77/98/2.0	132/75/3.5	87/79/2.5	105/69/3.0	85/98/1.5	140/71/2.5	75/103/1.5	89/89/1.75
Bel lows	No	No	Yes	Yes	82/74/2.25 No	177/55/4.75 No	85/59/2.25 Yes	130/41/3.5 Yes	No	No	Yes	Yes
Tubes	0.147	0.123	0.346	0.325	0.142	0.103	0.312	0.383	0.092	0.061	0.231	0.139
Type	SWT	SWT	DWT	DWT	SWT	SWT	DWT	DWT	SWT	SWT	DWT	DWT
OD/ID (in.-in.)	0.625/0.407	1.25/0.908	0.813/0.437	0.813/0.405	0.625/0.407	1.25/0.924 0.875	0.75/0.424* 0.813/0.427†	0.813/0.405	1.5/1.282	2.0/1.736	1.0/0.748	1.5/1.24
Length/No. (ft)	98/2224	533/163	72/3365	124/1858	98/2434* 74/1970†	347/236* 173/424†	79/3265* 59/2641†	71/4410* 46/5307†	98/862	215/178	103/1632	85/930
Tubesheets	0.021	0.006	0.155	0.040	0.027	0.002	0.089	0.106	0.041	0.005	0.056	0.049
No.	2	8	3	2	2	8	3	2	2	8	3	2
Dia/Thickness (Total) (in.) (in.)	72/31	22/24	160/15 106/52	82/44	77/31* 82/38†	15/16 16/16	130/24 85/80	105/60* 130/60†	85/42	20/22	120/10 75/36	89/44
Internals	0.017	0.037	0.041	0.031	0.010	0.043	0.033	0.043	0.016	0.032	0.025	0.019
% of Total Mat. Cost	5.5	5.5	5.5	8.8	5.5	5.5	5.5	8.8	5.5	5.5	5.5	8.8
Tooling	0.021	0.044	0.057	0.044	0.022	0.050	0.047	0.061	0.020	0.038	0.035	0.027
% of Total Mat. Cost	6	9	7	7	6	9	7	7	6	9	7	7
Handling/Warehousing (3.9% of Total Mat. Cost)	0.012	0.016	0.028	0.022	0.007	0.019	0.024	0.030	0.011	0.014	0.018	0.013
LABOR COST												
Labor (2 x Total Mat. Cost)	0.693	0.967	1.630	1.261	0.739	1.115	1.354	1.739	0.655	0.835	1.000	0.765
TOTAL COST												
Labor & Material	1.06	1.48	2.49	1.93	1.11	1.71	2.07	2.66	1.00	1.29	1.54	1.17

*Evaporator
†Superheater

Reference Cost Value

Table 9.2 shows the cost of shell and headers is primarily influenced by diameter. The AI design has the lowest cost for shell and headers in the Benson and Sulzer cycles, and second, for the saturated cycle. BW was fourth in all three cycles; the W design was second in the Sulzer, third in the saturated cycle and Benson cycles; and the FW design was first in the saturated design, second for the Benson and third for the Sulzer cycle.

The BW design has the lowest cost tubes for all three cycles because it has the largest diameter and fewest number of tubes in all cases. The AI tube cost was second in all three cycles. The W double-wall tube cost was third for the Sulzer cycle and fourth for the Benson and saturated cycles. The FW double-wall tube cost was third for the Benson and saturated cycle and fourth for the Sulzer cycles design. The switchover in cost for the W and FW double-wall tubes is the result of differences in relative tube length and diameter for the three cycles.

The BW design has the lowest cost tubesheet for all three cycles because of its small size. The AI tubesheet was second in cost for all steam cycles. The FW tubesheet was third for the Benson and saturated cycles and fourth for the Sulzer cycle. It is estimated their tubesheet would also have been third in cost for the Sulzer cycle if they had chosen longer tubes for this design to reduce shell diameter as noted above. The W tubesheet was third for the Sulzer cycle and fourth for the Benson and saturated cycles because of its large diameter and use of three tubesheets per module.

The AI design has the lowest cost for tooling, handling/warehousing, and labor as these items were taken as a percentage of total material cost, and the total cost of AI design was lowest in all three cycles as noted above.

10.0 PREFERRED DESIGN SELECTION

The four steam generator designs were rated for each of the three steam cycles to determine the preferred design for each cycle. The rating parameters and results are given in Table 10.2. The ratings were based on information given in the previous sections of this report. In this table, the lowest number is the preferred design. These results indicate the preferred design for each steam cycle is as summarized in Table 10.1. These two tables indicate the AI hockey stick design is first choice for all three cycles and the B&W helical design is second choice for the superheated cycles. The FW DWT design is second choice for the saturated cycle and third choice for the Benson and fourth choice for the Sulzer cycle. It is estimated it would also have been third choice for the Sulzer cycle if they had chosen longer length tubes for that design.

TABLE 10.1
ORDER OF DESIGN SELECTION

Choice	Steam Cycle		
	Benson	Sulzer	Saturated
1st	AI	AI	AI
2nd	B&W	B&W	FW
3rd	FW	<u>W</u>	<u>W</u>
4th	<u>W</u>	FW	B&W

TABLE 10.2
COMPARATIVE EVALUATION
(lowest number best)

RATING PARAMETER	Design											
	Benson				Sulzer				Saturated			
	AI	B&W	W	FW	AI	B&W	W	FW	AI	B&W	W	FW
Design Features	1	2	4	3	1	2	4	3	1		4	3
Simplicity	1	4	3	2	1	4	3	2	1		3	2
Sodium Maldistribution	2	1	4	3	2	1	4	3	2		4	3
Tubesheet Insulation	1	2	4	3	1	2	4	3	1		4	3
Shell Bellows	0	0	2	1	0	0	1	2	0		1	2
Tube Bundle Support	<u>3</u>	<u>4</u>	<u>2</u>	<u>1</u>	<u>3</u>	<u>4</u>	<u>2</u>	<u>1</u>	<u>3</u>		<u>2</u>	<u>1</u>
	7	11	15	10	7	11	14	11	7		14	11
Thermal/Hydraulic	1	2	4	3	1	2	4	3	1		4	3
Structural	1	2	4	3	1	2	4	3	1		4	3
Scale-up	1	2	4	3	1	2	3	4	1		3	4
Manufacturing	1	4	2	3	1	4	2	3	1		2	3
Cost	1	2	4	3	1	2	3	4	1		4	2
Inspection/Maintenance	2	1	4	3	2	1	3	4	2		4	3
Reliability	<u>1</u>	<u>2</u>	<u>4</u>	<u>3</u>	<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>	<u>1</u>		<u>3</u>	<u>2</u>
TOTAL	9	17	30	24	9	17	26	28	9	N/A	28	23

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