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**LMFBR STEAM GENERATOR DEVELOPMENT:
DUPLEX BAYONET TUBE STEAM GENERATOR**

Volume II

By
D. D. DeFur

NOTICE

Date Published—April 1975

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LMFBR STEAM
GENERATOR DEVELOPMENT

DUPLEX BAYONET TUBE
STEAM GENERATOR

ERDA CONTRACT AT (11-1) 2426

APRIL 1975

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ABSTRACT

This is Volume II of a two volume report prepared by Combustion Engineering, Inc. in response to ERDA Contract Number AT (11-1) 2426 for the development of advanced design steam generators.

This volume covers the work performed on the Duplex Bayonet Tube Steam Generator.

Volume I covers the work performed by C-E for the Matrix Steam Generator.

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1.0 INTRODUCTION

There are many considerations which can effect the design of a steam generating system, among these are safety, reliability and cost.

Operating experience with Liquid Metal Fast Breeder Reactor prototype steam generating models and plants indicates that a major cause of unreliability associated with such systems is the possibility of a sodium-water reaction. Should such reaction occur, as a result of leakage at the sodium/water interface, the unit could be damaged significantly and lead to the plant being shut down for repairs. Although a steam generator employing single walled heat transfer tubes represents the most economical capital cost design, the use of double walled or "duplex" tubes may prove to be more economical in terms of overall plant life cost since initial fabrication costs could be more than offset by less operational downtime due to leakage and removal of sodium/water reaction products and steam dump systems. In addition, a double walled unit is inherently safer than a single walled unit. These considerations have been strongly substantiated by the double walled Experimental Breeder Reactor-II (EBR-II) steam generators which have accumulated almost ten years of operating experience unhampered by shutdowns due to tube leakage. The duplex bayonet tube steam generators for the Hallam Nuclear Power Facility (HNPF) also operated completely reliably for several years during testing and over a year in power operation.

1.0 INTRODUCTION (continued)

This report represents the culmination of work performed in fulfillment of ERDA Contract AT(11-1)-2426, Task Agreement 2, in which alternate steam generator designs were developed and studied. The basic bayonet tube generator design previously developed by C-E under AEC Contract AT(11-1)-3031 was expanded by incorporating duplex heat transfer tubes to enhance the units overall safety and reliability. The effort consisted of providing and evaluating conceptual designs of the evaporator, superheater and reheater components for a large plant LMFBR steam generator (950 MWt per heat transport loop).

1.1 SUMMARY OF REPORT

In summary, the subsequent contents of this report are as follows:

- Section 1.3 provides the conclusions drawn from this study period.
- Section 2.0 contains a brief history of previous multiple walled steam generator designs and studies such as the EBR-I, EBR-II and HNPF steam generators and the Westinghouse duplex tube bench tests.
- Section 3.0 provides the detailed description of the C-E evaporator, superheater, and reheater units developed under this contract respectively and presents the conceptual design drawings which were generated during

1.1 SUMMARY OF REPORT (continued)

this study period. Figure 3.1 defines the characteristics of the heat transport loop for which the units were designed and shows the temperatures, pressures, flow rates, etc. for each of the units. This section also presents the construction materials which C-E feels are best suited for the design.

● Section 4.0 is entitled Design Considerations and represents the bulk of the work performed relative to the duplex bayonet tube unit under this contract. This section is broken down into four main sections (4.1, 4.2, 4.3 and 4.4) which report the detailed design effort, and one lesser section (4.5) which provides an index for assisting the reader in locating the discussion of points which were of specific concern during the February 1, 1975 Status Review Meeting between C-E and ERDA.

● Section 4.1 provides an in depth review of the assessments made relative to the thermal aspects of the steam generator. This was a dual fronted effort involving (1) the interfacial conductance between the duplex tubes which provide the sodium/water boundary, and (2) the thermal sizing per se of the units. Respectively these topics are reported in Sections 4.1.1 and 4.1.2. The two remaining sections (4.1.3 and 4.1.4) respectively address the reduction in heat transfer surface area obtained by using the high chromium Sandvik H-T9 steel for tubing material and the added heat

1.1 SUMMARY OF REPORT (continued)

transfer surface area (relative to a single walled unit of similar design) associated with using the double walled heat transfer tubes.

● Section 4.2 addresses the structural aspects of the steam generator from the points of view of Tube Sizing as reported in Section 4.2.1, the problem of relative movement or sliding between the duplex tubes as reported in Section 4.2.2, and the determination of contact pressures attainable both during manufacturing and operation as reported in Section 4.2.3.

● Section 4.3 reports two major manufacturing considerations which heavily influence the practicality of fabricating a duplex bayonet tube steam generator component of the design presented in this report. These were the process of physically producing duplex tubes of the desired quality, and providing adequate welds at all tube-to-tubesheet junctions. These subjects are addressed by Sections 4.3.1 and 4.3.2 respectively.

● Section 4.4 contains the safety and reliability considerations. This effort included an assessment of leak behavior and leak detection techniques as reported in Section 4.4.1 and a failure mode/fault tree analysis as reported in Section 4.4.2.

1.1 SUMMARY OF REPORT (continued)

- Section 5.0 presents a logical sequence for assembly of the steam generator components during manufacturing.

- Section 6.0, which entitled Development Program, outlines areas within the design where subsequent additional study and test effort is felt desirable. Those areas outlined in this section primarily include a more detailed analytical assessment of certain features from a structural integrity point of view; a "proof of principle" test involving physically producing samples of duplex tubing and assessing the resulting interfacial zone; a program to optimize the designs from a structural integrity/thermal performance point of view; and a fabrication cost analysis of the units for use as a comparative base against other steam generator designs.

1.2 CONCLUSIONS

It was generally concluded from this study that the fabrication of a duplex bayonet tube steam generator of the design shown in subsequent sections of this report is within present day state-of-the-art technology and that a unit of such design has high potential for becoming a reasonable contender for use in LMFBR Plants. The designs offer several advantages and disadvantages when compared with similar components of other designs. Some of the advantages and disadvantages are listed as follows:

1.2 CONCLUSIONS (continued)

● Advantages

- The units provide the feature of limiting the temperature of all high pressure tubesheets to (or near) the saturation temperature of steam at design pressure (690⁰F at 2600 psig) and thus considerably below the tubesheet material creep range.
- The units provide full penetration welds which can be readily inspected throughout; thus, all welds are of high integrity and are crevice free on both the water and sodium side.
- The units provide double tubesheets and duplex tubes with an intermediate leak detection fluid such that all sodium/water interfaces are separated by the intermediate fluid.
- The units share many features which were proven practical and reliable in the fabrication, testing and operation of the EBR-II and HNPF steam generators.

● Disadvantages

- A disadvantage which these bayonet tube units share is the lack of a convenient means of draining the water side. Since the gas space featured in C-E's previous steam generator design (Ref. AEC

1.2 CONCLUSIONS (continued)

Contract AT(11-1)-3031) has been deleted, these units could conceivably be redesigned slightly and operated in an inverted position to rectify the drainability consideration.

- The potential problems of relative tube sliding and strain concentration in the leak detection grooves could present structural reliability problems. These areas are recommended in Section 6.0 as areas requiring additional analytical study.
- The heat transfer annuli of any bayonet tube unit require a more difficult cleaning procedure than units of other designs such as straight through tubes attached to tubesheets at either end.
- The evaporator unit has a disadvantage in that boiling occurs in the relatively narrow heat transfer annuli.

Section 6.0 lists several areas where C-E feels additional work should be done to further verify the feasibility and functionability of duplex bayonet tube steam generators.

2.0 HISTORY OF MULTIPLE WALLED STEAM GENERATORS

A brief historical review of previous multiple tube walled steam generator designs should prove helpful in assessing the Duplex Bayonet Tube Steam Generator. The previous designs to which this section is addressed are the EBR-I, EBR-II and HNPF Steam Generators. In addition, certain studies pertinent to multiple walled heat transfer tubes are included.

2.1 EBR-I STEAM GENERATOR COMPONENTS

The EBR-I Steam Generator consisted of nine units of the design shown in Figure 2-1, connected in series (ref. 2-1). This arrangement provided the economizer, boiler, and superheater for the small experimental plant (approximately 4 MW(t) total rating). The heat transfer tube assembly of each component consisted of an inner and outer nickel tube and an intermediate copper tube. The assemblies were fabricated by mechanically drawing the tubes together and subsequently bonding the interfaces by thermal diffusion to enhance the heat transfer characteristics. The heat transfer tube assembly had an outside diameter of 2.625 inches and a total wall thickness of .313 inches. The individual tube wall thicknesses were .063, .125 and .125 inches respectively for the outer, intermediate,

EBR-I STEAM GENERATOR

FIGURE 2-1

2.1 EBR-I STEAM GENERATOR COMPONENTS (continued)

and inner tubes. Thus, the composite wall was 60% nickel and 40% copper.

The tubes were mechanically drawn together by passing them over a mandril to enlarge the inner tube and through a die to reduce the outer tube. The assemblies were then heated to form a copper-nickel alloy at the interfaces. This process yielded heat transfer coefficients of 204 BTU/hr-ft²-°F and 508 BTU/hr-ft²-°F for the film boiling and nucleate boiling sections respectively.

2.2 EBR-II STEAM GENERATOR COMPONENTS

Initially the EBR-II Steam Generator was to consist of eight evaporator units and four superheater units of the configuration shown in Figure 2-2 (ref. 2-1). Difficulties encountered in making consistently reliable tube to tubesheet welds in the superheater units caused fabrication of those tubes to be abandoned and two spare evaporator units were modified for use as replacements.

The double walled heat transfer tubes were manufactured from 2-1/4 Ch. 1 Mo. by drawing the assemblies simultaneously through a die and over a mandrel with final sizing being accomplished by drawing them over

EBR-II STEAM GENERATOR
FIGURE 2.2

2.2 EBR-II STEAM GENERATOR COMPONENTS (continued)

a second mandrel. Throughout the drawing operation the tube walls were seal welded together at either end to prevent contamination of the interface with lubricants.

The tubes were installed by welding the ends of the inner tubes to the steam tubesheets and the ends of the outer tubes to the sodium tubesheets. The space between the tubesheets and the interface between the double tube walls was open to the atmosphere. The welding problem which precluded fabrication of the superheater units was due to difficulties associated with welding the thin outer tube wall to the sodium tubesheet without damaging the inner tube. A procedure was developed for producing acceptable quality welds in the larger tubes of the evaporator unit. Tube sizes were 1.438 inch and .596 inch outside diameter with composite wall thicknesses of .187 inch and .096 inch respectively for the evaporator and superheater units.

Differential thermal expansion between the tube bundle and component shell was accommodated by prestressing the component shell during fabrication.

Overall heat transfer coefficient for the evaporator unit was 500 BTU/hr-ft²-°F as predicted indicating reasonably good interfacial conductance within the composite tube wall.

2.3 HNPF STEAM GENERATOR

The HNPF steam generators were sodium in tube horizontally mounted bayonet tube units as shown in Figure 2-3 (ref. 2-1). These units physically share many characteristics of C-E's Duplex Bayonet Tube Steam Generator in that the tubes were closed at one end; they were expanded one inside the other in situ in the steam generator; and, they provided an enclosed leakage monitoring gas system between the double tube walls and the dual tubesheets. The tubes were roller expanded from top to bottom after making both tube-to-tubesheet welds. The overall heat transfer coefficients noted during operation were high (800 BTU/hr-ft²-°F initially for the evaporator) indicating a good interfacial conductance.

2.4 OTHER MULTIPLE TUBE STEAM GENERATOR EFFORTS

Noteworthy at this point are the recent studies performed by Westinghouse Electric Corporation, Tampa Division under contract to the U. S. Atomic Energy Commission (ref. Contract no. AT (04-03) - 962). Within this study interface behavior between grooved duplex tubes were studied. The tube samples were produced by mechanically drawing the tubes together and partial penetration welds, similar to those used on EBR-II, were used to attach the tubes to end fittings. Several test sets were performed with both clean and water contaminated interfaces, with

HNP STEAM GENERATOR

FIGURE 2-3

2.4 OTHER MULTIPLE TUBE STEAM GENERATOR EFFORTS (continued)

various interface configurations, using both annealed and prestressed tubing. The study concluded that contact conductance varied between 2000 and 50,000 BTU/hr-ft²-°F depending on the interface contents, interface separation, and interface "preload".

REFERENCES FOR SECTION 2.0

- 2-1 John G. Yevick, "Fast Reactor Technology Plant Design",
The M.I. T. Press, Massachusetts Institute of Technology,
Cambridge, Massachusetts, 1966, pp. 208 - 217.

3.0 PRESENTATION OF DESIGN

The duplex bayonet tube steam generator design is based on the plant conditions shown in Figure 3-1. Each 950 MWt steam generator is comprised of two (2) 283 MWt evaporator units, one (1) 222 MWt superheater unit, and one (1) 162 MWt reheater unit. Thus, the power is split 60% for the evaporators, 23% for the superheater, and 17% for the reheater.

The components are similar in design and can be generally described as follows:

- Each component is a vertical, top supported, shell and tube type heat exchanger with sodium being the hotter shell side fluid and water/steam being the cooler tube side fluid.
- Each component employs the bayonet tube concept for accommodating differential thermal expansion between the tube bundle and shell. Typically a bayonet tube assembly is made up of four (4) tubes: an outer pressure tube (See Figure 3-2 for nomenclature), an inner pressure tube, an insulator tube, and a bayonet tube. Thus, since the sodium/water interface is comprised of two separate tubes tightly nested together, the tubes are "duplex" bayonet tubes.

SCHEMATIC OF HEAT TRANSPORT LOOP
FIGURE 3-1

NOMENCLATURE
FOR BAYONET TUBE ASSEMBLY
FIGURE 3-2

3.0 PRESENTATION OF DESIGN (continued)

- Each component provides dual tubesheets as shown in Figure 3-3. This feature is commensurate with the duplex tubes in that the outer pressure tube attaches to the lower or sodium/gas tubesheet, and the inner pressure tube attaches to the upper or water/gas tubesheet. Thus two separate and complete boundaries exist at all sodium/water interfaces. The space between these two boundaries defines a gas space which is used for detecting either sodium or water should leakage occur.
- Each component features full penetration high integrity welds at all tube-to-tubesheet and tube-to-end cap joints.

Specific details of each component are given in paragraphs 3.1, 3.2, and 3.3.

3.1 EVAPORATOR UNIT

The conceptual design for the evaporator unit is shown on Drawing E-63675022. The unit is approximately 67 feet long, has a tube bundle diameter of slightly over 51 inches, and weighs approximately 150 tons.

Feedwater enters the component through one 13.88 inch inlet nozzle and flows downward through the bayonet tubes to the lower extremity of the tube bundle. The water is then boiled in upflow as it re-traverses the length of the tube bundle through the heat transfer annulus

DOUBLE WALLED BAYONET TUBE
STEAM GENERATOR

FIGURE 3.3

3.1 EVAPORATOR UNIT (continued)

such that it reaches the high pressure tubesheet as saturated (or slightly superheated) steam. The steam passes upward around the outside of the inlet plenum and exits the unit through one 13 inch outlet nozzle. Sodium enters the unit through two 21 inch inlet nozzles and flows upward and radially through the sodium inlet plenum and into the tube bundle through eight flow distribution slots in the flow baffle. The sodium then flows downward giving up heat through the duplex pressure tubes to the counter flowing water and exits the component through the 29.25 inch outlet nozzle. Using this flow arrangement in conjunction with thermal shielding on the lower surface of the sodium/gas tubesheet, the entire upper portion of the component operates at saturation temperature. This is a noteworthy feature in that the high pressure steam/gas tubesheet functions at temperatures significantly below the creep range of the tubesheet material.

The tube bundle is made up of 613 duplex bayonet tube assemblies located on a triangular spacing of 1.95 inches. Individual tube sizes are summarized as follows:

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DUPLEX BAYONET TUBE
EVAPORATOR UNIT

FOR
FBR/LP HEAT TRANSPORT LOOP

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3.1 EVAPORATOR UNIT (continued)

	<u>Tube Outside Diameter (in.)</u>	<u>Tube Average Wall Thickness (in.)</u>
Outer pressure tube	1.450	0.080
Inner pressure tube	1.290	0.067
Insulator tube	0.781	0.040
Bayonet Tube	0.581	0.040

The tube bundle length is 48 feet and the total heat transfer area is 11,200 feet². Seventeen (17) eggcrate type tube supports are spaced along the length of the tube bundle to externally support the bayonet tube assemblies (See Section C-C of Drawing E-63675022). Spacers are provided on the bayonet and insulator tubes for supporting the respective tubes and maintaining concentricity of the annuli. The purpose of the insulator tube is to provide a stagnant fluid filled insulator annulus between the fluid flowing in the bayonet tube and the fluid flowing in the heat transfer annulus to serve as a thermal insulator between the two flow paths. This feature is highly desirable, if not necessary, in both the superheater and reheater units to limit the regenerative heat transfer across the bayonet tube wall; however, the effectiveness of this feature is dependent upon the contents of the insulator annulus. In the case of the superheater and reheater, the annulus will contain steam, which has a relatively low value

3.1 EVAPORATOR UNIT (continued)

of thermal conductivity, while in the evaporator the annulus will contain liquid water with a somewhat higher value of thermal conductivity.

Therefore, the effectiveness of the insulator tube principle in the evaporator will have to be weighed against the added material and fabrication cost for justification.

Thicknesses of shells, tubesheets, and other parts of the component are summarized as follows:

Steam chest shell	6.00 in.
Gas plenum shell	1.50 in.
Upper sodium shell	1.50 in.
Lower sodium shell	1.25 in.
Lower hemispherical head	1.25 in.
Sodium flow baffle	1.00 in.
Removable top head	18.31 in.
Removable feedwater inlet plenum head	5.00 in.
Feedwater lower head (low pressure tubesheet)	5.00 in.
Steam/gas tubesheet	16.50 in.
Sodium/gas tubesheet	9.00 in.

The materials selected for use in fabricating the component are as follows:

Inner and Outer pressure tubes	Sandvik H-T9
Bayonet and insulator tubes	SA-213 T22
Shells in contact with water/steam	SA-387 Gr. D.

3.1 EVAPORATOR UNIT (continued)

Shells in contact with sodium	SA-387 Gr. D with roll bond clad of SA-240 type 405 roll on sodium side
Sodium flow baffle, thermal liners, eggcrate bars, and thermal shields	SA-240 type 405
Steam/gas tubesheet and upper flange	SA-182 F22
Sodium/gas tubesheet	SA-182 F22 with weld overlay clad of SA-240 type 405 on sodium side
Bolting material	SA-193 Gr. B7
"O" rings for upper removable head	Inconel

3.2 SUPERHEATER DESIGN

The conceptual design for the superheater unit is depicted on Drawing E-63675023. The approximate overall height and weight of this component are 56 feet and 257 tons respectively. The tube bundle diameter is slightly under 73 inches.

Flow directions through the unit are such that saturated steam enters through one 24 inch inlet nozzle and flows downward through the heat transfer annuli to the lower extremity of the tube bundle. It is during this downward traversal of the tube bundle that sufficient heat is transferred from the counter flowing molten sodium to superheat the steam to slightly over 900°F by the time it reaches the closed ends of the bayonet tubes. The superheated steam then passes up through the bayonet tubes to exit the unit by way of the outlet plenum and 24 inch outlet nozzle. Sodium enters the unit through the 31.5 inch nozzle in the hemispherical lower head and flows upward through the tube bundle, out into the outlet plenum through eight flow distribution slots in the flow baffle and exits the component through two 22.5 inch outlet nozzles. Using this flow arrangement, in conjunction with thermal shielding on the lower surface of the sodium/gas tubesheet and thermal liners in the outlet nozzle, permits operation of the entire upper region of the unit at saturation temperature and thus well below the creep range of the tubesheet material.

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DUPLUX BAYONET TUBE	
SUPERHEATER UNIT	
FOR	
FABRIK HEAT TRANSPORT LOOP	
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3.2 SUPERHEATER DESIGN (continued)

The tube bundle contains 1345 bayonet tube assemblies located on a triangular spacing of 1.904 inches. Respective tube sizes are as shown below:

	<u>Tube outside Diameter (in.)</u>	<u>Tube Average Wall Thickness (in.)</u>
Outer pressure tube	1.404	0.082
Inner pressure tube	1.240	0.062
Insulator tube	0.916	0.040
Bayonet tube	0.716	0.040

The tube bundle heated length is 30 feet and has a total heat transfer area of 14,700 feet². Nine eggcrate type tube supports are spaced along the tube bundle length for external support (See drawing E-63675023) and internal spacers are provided for supporting the bayonet/insulator tubes and maintaining concentricity of the insulator and heat transfer annuli. As previously stated, the insulating effects provided by the insulator tubes is highly desirable, if not necessary, in this unit. These tubes are installed concentrically around the bayonet tubes such that annuli are formed between the respective tube walls. These annuli are open at the bottom and closed at the top such that they can become filled with steam but such that no flow will be introduced into them. Thus, a blanket of stagnant steam forms a thermal barrier around each bayonet tube.

3.2 SUPERHEATER DESIGN (continued)

This blanket will reduce the exchange of heat between the fluid being heated in the heat transfer annuli and the fluid flowing in the bayonet tubes.

(This feature was physically modeled and studied as a subtask of the previously mentioned AEC contract AT (11-1)-3031).

Thicknesses of shells, tubesheets, and other parts of the component are summarized as follows:

Steam chest shell	8.00 in.
Gas plenum shell	2.50 in.
Upper sodium shell	1.50 in.
Lower sodium shell	1.25 in.
Lower hemispherical head	1.25 in.
Sodium flow baffle	1.00 in.
Removable top head	23.00 in.
Removable steam inlet plenum head	9.00 in.
Steam lower head (low pressure tubesheet)	9.00 in.
Steam/gas tubesheet	23.00 in.
Sodium/gas tubesheet	12.00 in.

3.2 SUPERHEATER DESIGN (continued)

The materials selected for use in the component are as follows:

Inner and Outer pressure tubes	Sandvik H-T9
Bayonet and insulator tubes	SA-213 T22
Shells in contact with water/steam	SA-387 Gr. D.
Shells in contact with sodium	SA-387 Gr. D with roll bond clad of SA-240 type 405 roll on sodium side
Sodium flow baffle, thermal liners eggcrate bars, and thermal shields	SA-240 type 405
Steam/gas tubesheet and upper flange	SA-182 F22
Sodium/gas tubesheet	SA-182 F22 with weld overlay clad of SA-240 type 405 on sodium side
Bolting material	SA-193 Gr. B7
"O" rings for upper removable head	Inconel

3.3 REHEATER DESIGN

The conceptual design for the reheater component is shown on drawing E-63675033. The unit is approximately 50 feet tall, has a tube bundle diameter of 94 3/4 inches, and weights approximately 250 tons.

Fluid flow directions are the same as for the superheater unit. Saturated steam enters the heat transfer annuli side of the component through one 34 inch inlet nozzle and, after having been superheated to around 900°F during the downward passage through the tube bundle, exits through one 34 inch outlet nozzle. Again, sodium is passing through the shell side of the unit in counter flow to the steam. It enters through the 32 inch nozzle in the lower hemispherical head, flows upward through the shell side of the tube bundle, and exits through the 32 inch outlet nozzles. With this flow arrangement, like the superheater unit, the entire upper region of the component, including the high pressure tube-sheet, operates at saturation temperature and thus well below the tubesheet material creep range.

The tube bundle is made up of 1273 bayonet tube assemblies located on a 2.536 inch triangular spacing. Respective tube sizes are as summarized below:

3.3 REHEATER DESIGN (continued)

	<u>Tube outside Diameter (in.)</u>	<u>Tube Average Wall Thickness (in.)</u>
Outer pressure tube	2.036	0.063
Inner pressure tube	1.910	0.063
Insulator tube	1.405	0.040
Bayonet tube	1.205	0.040

The 23 feet heated length of the tube bundle provides a total heat transfer area of 15,500 feet². Seven eggcrate tube supports are used for shell side restraint of the bayonet tube assemblies and spacers are provided for maintaining annuli concentricity and bayonet/insulator tube support within the bayonet tubes.

As was the case with the superheater, the thermal insulation afforded by the stagnant steam blanket within the insulator annulus is highly desirable, if not necessary, for limiting the regenerative heat transfer within the reheater unit. This feature is provided by the insulator tubes which are concentrically installed around the bayonet tubes to form these annuli. They are open at the bottom and closed at the top such that they become filled with steam but such that no flow is induced into them.

3.3 REHEATER DESIGN (continued)

Thicknesses of shells tubesheets, and other parts of the reheater are summarized as follows:

Steam chest shell	3.00 in.
Gas plenum shell	2.50 in.
Upper sodium shell	1.75 in.
Lower sodium shell	1.50 in.
Lower hemispherical head	1.50 in.
Sodium flow baffle	1.00 in.
Removable top head	12.38 in.
Removable steam inlet plenum head	7.00 in.
Steam inlet plenum lower head	7.00 in.
Steam/gas tubesheet	20.00 in.
Sodium/gas tubesheet	20.00 in.

The materials selected for use in this component are as follows:

Inner and outer pressure tubes	SA-213 T22
Bayonet and insulator tubes	SA-213 T22
Shells in contact with water/steam	SA-387 Gr. D

3.3 REHEATER DESIGN (continued)

Shells in contact with sodium

SA-387 Gr. D with
roll bond clad
of SA-240 type 405
roll on sodium side

Sodium flow baffle, thermal iners

eggcrate bars, and thermal shields

SA-240 type 405

Steam/gas tubesheet and upper flange

SA-182 F22

Sodium/gas tubesheet

SA-182 F22 with weld
overlay clad of SA-240
type 405 on sodium
side

Bolting material

SA-193 Gr. B7

"O" rings for upper removable head

Inconel

4.0 DESIGN CONSIDERATIONS

Several in depth studies were performed relative to the duplex bayonet tube steam generator components presented in Section 3.0. These were directed categorically toward heat transfer characteristics, structural integrity, manufacturing techniques, and safety and reliability aspects of the components. These are addressed in detail in the following paragraphs.

4.1 THERMAL CONSIDERATIONS

The thermal aspects of the duplex bayonet tube steam generator which were considered include the conductance at the interface between the inner and outer pressure tubes, and the thermal sizing per se of the three components. In addition to these two major areas, studies were performed to assess the advantage of using the Sandvik H-T9 tubing material in lieu of the more common SA 213 T22 material and to assess the added heat transfer surface area required in the duplex units in comparison to comparable sized single walled units.

4.1.1 INTERFACE BETWEEN DUPLEX TUBES

An important heat transfer consideration that must be assessed in the duplex tube design is the thermal conductance of the interface region of the tubes, since it can be significant in determining the overall heat transfer characteristics of a steam generator. Although there are many factors involved in determining the contact resistance, the tube fabrication

4.1.1 INTERFACE BETWEEN DUPLEX TUBES (continued)

process is a critical factor which, for example, can be used to predict the upper and lower bounds on contact resistance. If the duplexing manufacturing technique is ideal, a metal-to-metal contact exists at both tube surfaces and an infinite contact conductance exists (i.e., the thermal resistance is zero). In the worst case, the tube surface does not come into contact and a third fluid, usually a gas like air or helium, fills the gap between the tubes and acts as the conducting medium. The interface conductance (h_{gap} , BTU/hr.ft.²°F) for this case depends on the gap fluid and is determined by the gas thermal conductivity divided by the distance or

$$h_{gap} = K_{gas}/d_{gap} \quad \text{EQ \#4.1-1}$$

where;

K_{gas} = Thermal conductivity of the gas (BTU/hr-ft-°F)

d_{gap} = Interfacial separation of two surfaces (ft)

Radiation effects are negligible because the temperature difference between both surfaces is small. The difference between these two cases represents a range of two orders of magnitude with the actual interface conductance probably somewhere in between. It was the purpose of this task to determine the contact conductance of the duplexed tube in order to provide a realistic assessment of the duplex tube heat transfer characteristics.

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER

An evaluation of the interface thermal conductance of the duplex tube, with grooves for leak detection application, is necessary to insure a realistic assessment of the duplex tube heat transfer characteristics. Depending upon circumstances, the thermal conductance of an interfacial gap can represent a large percentage of the overall heat transfer coefficient in a steam generator. There is no recognized formula that can be used in predicting the gap conductance, and at the present, a state-of-the-art situation exists. Factors which have been identified and which pertain to the thermal contact problem are:

(a) tube materials, (b) surface finish (asperity height and geometry, (c) surface cleanliness, (d) contact pressure, (e) interface temperature, and (f) interfacial fluid. Other factors which appear in some correlations include surface geometry (asperity wave length) and heat flux; however, these parameters will not be discussed as their impact can be included in the variables mentioned previously. Each of these parameters are discussed individually as follows:

a. Tube Material

The tube material is generally selected on the basis of factors other than heat transfer characteristics such as compatibility with coolants, strength, material cost, material availability, the fabrication process, etc. The resulting heat transfer across the gap, where microscopic asperities are in contact, will depend upon the thermal conductivity of

4. 1. 1. 1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

the material(s); and is usually defined in terms of an effective (harmonic mean) thermal conductivity:

$$K_{\text{gap}} = \frac{2 K_1 K_2}{K_1 + K_2} \quad \text{EQ \#4. 1-2}$$

where;

K_1 and K_2 are the respective thermal conductivities of the materials
(BTU/hr-ft²-°F)

In the case of two dissimilar materials, the equivalent metal conductivity across the interface gap can be significantly affected if one of the thermal conductivities is small relative to the other.

The yield strength of the tube material is also important as it represents a measure of the elastic and/or plastic deformation of the microscopic asperities which, in turn, can be related to the true metal-to-metal contact area. As noted earlier, if metal-to-metal contact exists (over the entire or a significant portion of the area), there is an infinite gap conductance. However, if the contact area becomes smaller, less heat transfer will occur at the metal-to-metal contact point with an increasing amount of heat being transferred across adjacent gas gaps via conduction and radiation phenomena. From a heat transfer viewpoint, a material with a low yield strength is desirable in that the material deforms and "spreads out" so more metal-to-metal area is available for heat transfer.

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

b. Surface Finish

The surface finish essentially determines the effective gap distance over which heat is to be transferred. Any relatively smooth surface, when examined on a microscopic level, will become non-uniform with the surface consisting of numerous peaks and valleys which will vary in height, shape, and frequency. The effective conduction length will be the gap separation distance, or equivalently, the height of two asperities which are touching. This is illustrated in Figure 4.1-1a with the corresponding actual surface profiles shown in Figure 4.1-1b and 4.1-1c (taken from Ref. 4.1-1). The surface roughness can be represented by the "average" (or root mean square) asperity height and the frequency (or wave length) of the asperities (See Figure 4.1-1). On a microscopic level, the asperity height represents the conductance length and the asperity wave length (for both surfaces) determines the number of thermal contact points. On the microscopic level, only the asperity height is generally needed for mathematical modeling since the contact point area for a "flat" surface is specified by an empirical constant or taken into account internally within the formulation itself. As inferred by the surface profiles in Figure 4.1-1, the surface characteristics can be appreciably affected by polishing, mechanical deformations, aging, and surface treatment or corrosion.

(a) SCHEMATIC OF SURFACES IN CONTACT

(b) TYPICAL SURFACE PROFILES

(c) SURFACES WHICH HAVE BEEN ATTACKED AND AGED

MICROSCOPIC SURFACE PROFILES
FIGURE 4. 1-1

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

c. Surface Cleanliness

The importance of a clean surface is obvious as it affects surface conditions. Changes in surface conditions will affect heat transfer as contaminants will (1) prevent metal-to-metal contact, (2) introduce low conductivity materials, (3) change the asperity heights (by etching or buildups), and (4) reduce radiation heat transfer. Corrosion or "crud" buildup appears to be the most significant aspect of surface conditions as a "thick" layer of contaminants can result in a large thermal resistance. For example, typical fouling factors for water-side contamination are in the 1000-2000 BTU/hr-ft²-°F range (Ref. 4.1-2). In the case of thin films, which could occur at the duplex interface, a .164 mil film results in thermal conductances of 345 and 4300 BTU/hr-ft²-°F for Al₂O₃ (Ref. 4.1-3) and caesium oxide* (Ref. 4.1-1), respectively. Although these materials will not occur at the duplex tube interface, they do indicate the order of magnitude effect that contaminants could have in assessing an effective conductance at the interfacial gap. The interesting characteristic of a film is that they are often porous and the presence of voids (in the film) would tend to reduce the thermal conductivity to a value of less than that of a solid film.

*A corrosion product identified to occur inside LMFBR uranium oxide fuel pins.

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

d. Contact Pressure

The contact pressure between two surfaces represents a controlling factor which represents the amount of "true" metal-to-metal contact area. As the contact pressure is increased, the distance between the surfaces close and more and more asperities come into contact. Those asperities which touch deform, either elastically or plastically, with a "spreading out" effect such that the net true contact area increases. The actual amount of metal contact will be non-linear with increasing pressure (along with a corresponding increase in interface conductance) and will depend upon a variety of parameters mentioned previously: asperity height, material hardness, etc.; however, the overall effect will be to increase heat transfer across the gap. A representative estimate of the true metal-to-ceramic area in contact in LMFBR fuel rods is shown in Figure 4.1-2 (See Ref. 4.1-1). In general, this area tends to be in the one percent range unless high contact pressures are present.

e. Interfacial Temperature

The interface temperature is important as it directly determines the thermal conductivities of the conducting interface metal and fluid. Since radiation heat transfer is a temperature dependent phenomenon, it may be a controlling factor if a large temperature gradient exists. In general, the local thermal gradients across the gap are not

TRUE METAL TO CERAMIC CONTACT
AREA BETWEEN CLAD AND FUEL SURFACES IN LMFBR
FUEL RODS

FIGURE 4.1-2

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

large enough to make the interface temperature a controlling parameter, although, on a gross level, the temperature variation may be important on a variety of other parameters, for example: yield strength, thermal conductivities, and contact pressure (resulting from creep and thermal expansion).

f. Interfacial Fluid

The presence of a heat conducting gas at the interfacial gap will markedly improve the heat transfer across the gap by providing another heat transfer path (along with asperity contact and radiation paths). As the thermal conductivity of gas varies inversely with their molecular weights, it is desirable to use a gas that is as light as possible. Hydrogen gas is the best conductor, with a conductivity of $0.192 \text{ BTU/hr-ft-}^{\circ}\text{F}$ at 800°F (Ref. 4.1-4); however, it has some disadvantages which are its explosive properties, it diffuses through metal as a result of its atomic form, and it causes surface embrittlement and cracking in metals, especially when used with steels. Helium is the next best gas as it lacks the disadvantages of hydrogen and still has a high conductivity of $0.138 \text{ BTU/hr-ft-}^{\circ}\text{F}$ at 800°F . Other gases, including steam, which are much heavier, have an order of magnitude decrease in conductivity and fall in the range of, taking nitrogen as an example at 800°F , $0.0293 \text{ BTU/hr-ft-}^{\circ}\text{F}$. The presence of a vacuum in the gap represents the worst case and illustrates the importance of an interfacial fluid. Here a contact pressure change from zero to 1000 psi

4.1.1.2 PREVIOUS DUPLEX TUBE EXPERIENCE

The design basis for the contact resistance of Hallam and EBR-II was obtained from the system description references in which the designers appeared not to be significantly concerned that contact resistance was a problem. In both cases, the gap resistance was apparently taken to be negligible. In the case of the Hallam Nuclear Power Facility (HNPF), the system description reference (Ref. 4.1-5 and -6) implied a negligible value but did not state so specifically. Further investigation showed that the contact resistances were not as negligible as originally indicated and that preliminary calculations for a small scale 3 Mwt HNPF model (Ref. 4.1-7) found that average gap conductances during operation depended upon gap fluid as shown in Table 4.1-I. No estimation of contact pressure was given. Among the conclusions made were: a gap conductance less than $1000 \text{ BTU/hr-ft}^2 \text{ } ^\circ\text{F}$ indicates a poor duplexing procedure, and a surface finish of 0.030 mils should be provided for the tubing.

An interesting point to note is that the gap thermal conductance, for helium and nitrogen gas, did affect the overall heat transfer coefficient in the 3 Mwt model. The corresponding clean overall heat transfer coefficients for the 3 Mwt model evaporator and superheater are will change the contact conductance from a near zero value to approximately $1000\text{-}2000 \text{ BTU/hr-ft}^2 \text{ } ^\circ\text{F}$ (see Equation 4.1-3 presented subsequently). The conductance will grow rapidly in a non-linear fashion as the contact pressure increases because more asperities come into contact so that true metal contact area grows.

TABLE 4.1-I
HALLAM 3 MWt MODEL GAP
CONDUCTANCES

Interfacial Gas	Gap Conductance (Btu/hr-ft ² -°F)
Vacuum	1136
Nitrogen	1745
Helium	2884

4.1.1.1 PARAMETERS WHICH INFLUENCE INTERFACE HEAT TRANSFER (continued)

740 and 220 BTU/hr-ft² - °F, respectively (fouled values are 540 and 200 BTU/hr-ft² - °F, respectively). The corresponding clean overall film coefficients when the gap resistance is excluded are approximately 995 and 240 BTU/hr-ft² - °F, respectively. As shown by the values above, the gap conductance affects the overall heat transfer coefficient in the evaporator because of a large boiling water film coefficient. In the superheater, the low steam film coefficient dominates the overall heat transfer coefficient. It was inferred that the Hallam plant used the same gap conductances that were obtained from the 3 Mwt model.

An infinite interface gap conductance was used in the EBR-II steam generator design (Ref. 4.1-8). An inspection of the EBR-II tubing indicated that metal-to-metal contact occurred over 80% of the duplexed gap interface area while the other 20% of the area served as a gas space with interface separations of 0.1 to 0.2 mils occurring. It was later found (Ref. 4.1-9) that ANL had developed a special duplexing procedure in which the outer tube was drawn through a die and the inner tube was pulled through a pin. This procedure produced a high interface contact pressure (the outer tube was in tension and the inner tube was in compression) of 3800 psi (with no loading), increasing to 4300 psi when the inner tube was pressurized to 1300 psi (no thermal loading). This data indicates the importance of the fabrication process so as to provide a high interface contact pressure which, in turn, assures a large percentage of true metal contact area. The knowledge of the measured contact pressure

4.1.1.2 PREVIOUS DUPLEX TUBE EXPERIENCE (continued)

value is an important variable because it will provide an estimate of the contact pressure ranges expected in this analysis.

The experimental data from Westinghouse (Ref. 4.1-10) indicates a high gap conductance which significantly depends upon the tube manufacturing technique: prestressed tubing - $50,000 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$ and annealed tubing (i.e., after duplexing) - $2000 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$. After interfacial gap contamination with oxygen and moisture, the prestressed tube showed no decrease in gap conductivity while that for the annealed tube decreased 15%. The reasons for this disparity were that, (1) the prestressed tube interfacial gap is smaller than that of the annealed tubing and correspondingly permits less contamination to occur, and (2) that any corrosive attack at the gap results in "swelling" of the metal interface, which in turn increases the contact pressure, and thus enhances or counterbalances any decrease in gap conductivity.

As a result of the large differences in heat transfer coefficients, a cautious approach should be taken until more information is available since this data was obtained from a bench test, which did not exactly mimic steam generator operating conditions, and there was a large amount of scattered data. However, the magnitude of the gap conductances

4.1.1.2 PREVIOUS DUPLEX TUBE EXPERIENCE (continued)

does provide a reasonable estimate for design parameters.

4.1.1.3 ANALYTICAL MODELING AND FORMULATIONS

Many analytical formulations have been developed using experimental data taken from the aerospace, chemical, nuclear and other fields. Many of the papers are concerned with the fundamental theory of surfaces in contact (Reference 4.1-11 and -12), others have compiled data on specific conditions (Reference 4.1-13), and only recently have several models been developed for generalized usage. Shylkov (Reference 4.1-14) has correlated a large segment of data by combining the metal-to-metal contact with gas interface conduction with the equation:

$$h_{\text{gap}} = \frac{K_{\text{gas}}}{d_{\text{gap}}} \left\{ 1 + 0.32 \left(\frac{P}{Y} \frac{K}{3} \right)^{0.86} \right\} \frac{\bar{K}_{\text{metal}} d_{\text{gap}}}{K_{\text{gas}} a^*} \quad \text{EQ \#4.1-3}$$

where:

P = contact pressure (lb/in²)

Y = Material yield strength, taken as 47,000 lb/in² for HT-9 (lb/in²)

a^* = 0.04 mils

$K = 1$, when $(l_1 + l_2) \geq 0.03$ mils

l_1 and l_2 = asperity height at metal surface for surfaces 1 and 2 respectively (ft.)

$K = \left(\frac{30 \times 10^{-6}}{l_1 + l_2} \right)^{1/3}$, when $0.01 \leq (l_1 + l_2) \leq 0.03$ mils

$K = \frac{15 \times 10^{-6}}{(l_1 + l_2)}$, when $(l_1 + l_2) \leq 0.01$ mils

$\bar{K}_{\text{metal}}$ = harmonic mean thermal conductivity; see EQ #4.1-2 (BTU/hr-ft²-°F)

4.1.1.3 ANALYTICAL MODELING AND FORMULATIONS (continued)

It should be noted that the material properties of the softer metal (if there is one) should be used in calculating contact pressure parameters in this equation and the others which follow.

A more recent semi-empirical correlation has been developed by Hsieh and Touloukian (Ref. 4.1-15) which is in a form similar to that suggested by Shylkov:

$$h_{\text{gap}} = \frac{K_{\text{gas}}}{d_{\text{gap}}} \left\{ 1 + \frac{2}{3\pi} \left(\frac{P}{Y} \right) \frac{\bar{K}_{\text{metal}} d_{\text{gap}}}{K_{\text{gas}} \bar{a}} \right\} \quad \text{EQ \# 4.1-4}$$

where:

$$\bar{a} = 0.5 \text{ mils}$$

Ross and Stout (Ref. 4.1-14) have correlated the gap conductance by considering the contribution of heat transfer across the gap due to gas conduction and metal contact in the form:

$$h_{\text{gap}} = h_{\text{metal}} + h_{\text{gas}}$$

where:

$$h_{\text{metal}} = \text{equivalent metal heat transfer coefficient (BTU/hr-ft}^2\text{-F)}$$

$$h_{\text{gas}} = \text{equivalent gas gap heat transfer coefficient (BTU/hr-ft}^2\text{-F)}$$

or:

$$h_{\text{gap}} = \frac{\bar{K}_{\text{metal}} P}{0.0905 R^{1/2} H} + \frac{K_{\text{gas}}}{C \left\{ (l_1 + l_2) + (g) \right\}} \quad \text{EQ \#4.1-5}$$

4.1.1.3 ANALYTICAL MODELING AND FORMULATIONS (continued)

where:

$$R = \sqrt{(l_1^2 + l_2^2)/2}, \text{ root mean square asperity height}$$

$$C = 2.75 - .00176 P$$

H = material Meyer hardness, typically taken as 3.2 Y.

$$g = \left(\frac{2 - e}{c} \right) \frac{2}{Pr} \left(\frac{G}{G + 1} \right) l^*, \text{ temperature jump distance}$$

G = Ratio of specific heats

l^* = gas mean free path, taken as 3.28×10^{-7} ft.

e = accommodation coefficient, typical values are 0.1 for a polished surface and 0.9 for a very rough surface

Pr = Prandtl number

The g term represents the temperature jump distance which can be considered as a correction factor for the gas conductivity across very small distances.

That is, its purpose is to account for temperature discontinuities which occur at a gas/solid interface due to the imperfect energy equilibrium

between the gas and the solid. In general, the temperature jump distance is

not important unless the Knudsen number $Kn = l^*/C(l_1 + l_2)$ where

C is defined in equation 4.1-5 is smaller than one.

Bridges (Ref. 4.1-1) has developed the most recent correlation which, for surfaces in contact, can be represented, in a form similar to equation 4.1-5:

$$h_{\text{gap}} = 542 \left(\frac{P}{H} \right)^{1/2} \bar{K}_{\text{metal}} + \frac{K_{\text{gas}}}{C(l_1 + l_2) + g} \quad \text{EQ \#4.1-6}$$

4.1.1.3 ANALYTICAL MODELING AND FORMULATIONS (continued)

A graphical procedure for estimating the contact conductance is given by Veziroglu (Ref. 4.1-13). This method is tedious to use and simpler formulations are available.

4.1.1.4 DISCUSSION

The ability to predict the gap conductance is a state-of-the-art problem in which no existing formulation is recommended for general usage. Several of the parameters involved in the gap conductance problem will be summarized and discussed so that an assessment of their effects is possible.

The height of the surface asperities is very important as it determines the effective conductance length. That is, for the worst case, when the tubes are not in contact and are separated by a helium gap, the contact conductance is specified as:

$$h_{\text{gap}} = \frac{K_{\text{helium}}}{d_{\text{gap}}} \quad \text{EQ \#4.1-7}$$

where:

$$K_{\text{helium}} = \text{thermal conductivity at helium (BTU/hr-ft}^2\text{-}^{\circ}\text{F)}$$

4.1.1.4 DISCUSSION (continued)

Typical values have been substituted into Equation 4.1-7 and are plotted in Figure 4.1-3. Included in this figure is the gap conductance with a "temperature jump" correction, g , as given by Equation 4.1-5 in which the gap conductance formula becomes:

$$h_{\text{gap}} = \frac{K_{\text{helium}}}{d_{\text{gap}} + g} \quad \text{EQ \#4.1-8}$$

It is seen that temperature jump effect, which accounts for the difference in energy transfer between a gas-gas molecule collision and a gas-solid collision, becomes important only at very small distances and can be excluded here. No enhancement of the helium conductivity will occur as a result of natural convection effects since the Grashof number is less than 2000 due to the small interfacial distances involved (Ref. 4.1-16). Radiation effects are negligible as a result of a small temperature gradient at the gap.

It is expected that the tube surface finish will vary considerably depending upon fabrication technique, piping stock, etc., and will fall within a range of 1.0 to 0.03 mils with the latter being the recommended surface finish. The experimental data from the EBR-II and Westinghouse duplex tubing indicates the interfacial gaps between 0.1 and 0.5 mils have been actually obtained and measured. Using this information

FIGURE 4.1-3

4.1.1.4 DISCUSSION (continued)

to provide a conservative assumption, an interfacial helium gap of 0.5 mils is assumed to occur along with a surface finish equal to one-half of the gap distance (e.g., two asperities in contact) or 0.25 mils. This gap separation corresponds to a gap conductance of $3800 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$ due to the helium alone.

The all helium gap conductance represents a conservative number as no metal-to-metal contact conductance effects have been included. The problem now becomes one of providing a reasonable estimate of the metal-to-metal contact area, or conversely the contact pressure, so that the magnitude of the asperity-to-asperity heat transfer can be ascertained. A simple parallel path heat transfer model:

$$h_{\text{gap}} = (b_{\text{helium}} K_{\text{helium}} + b_{\text{metal}} K_{\text{metal}}) / d_{\text{gap}} \quad \text{EQ \#4.1-9}$$

where:

b_{helium} and b_{metal} = the respective percentages of helium path area and metal path area to total area.

K_{helium} = thermal conductivity of helium ($\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$)

K_{metal} = thermal conductivity of metal ($\text{BTU/hr-ft}^2\text{-}^\circ\text{F}$)

The behavior of equation 4.1-9 is tabulated in Table 4.1-II for different area ratios. The interesting point noticed in the table is that only a very small amount of metal-to-metal contact area is necessary to significantly alter the gap conductance. The data of Figure 4.1-2 indicates metal contact area in the order of 0.25%

TABLE 4.1-II
THE EFFECTS OF METAL-TO-METAL CONTACT AREA
ON
GAP HEAT TRANSFER COEFFICIENT

b_m % of Metal Area in Contact	b_{He} % of Helium, Area in Contact	h_{total}^* (Btu/hr-ft ² -°F)
0.0	100.0	3800
0.25	99.75	4800
0.50	99.50	5800
1.00	99.00	7800
2.5	97.5	13,900
5.0	95.0	24,000
10.0	90.0	44,000
50.0	50.0	206,000
100.0	0.0	408,000

*Based on the parameters: $K_{He} = 0.158 \text{ BTU/hr-ft-}^\circ\text{F}$

$K_m = 17.0 \text{ BTU/hr-ft-}^\circ\text{F}$

$l_{gap} = 0.5 \text{ mils}$

4.1.1.4 DISCUSSION (continued)

to 2.5%; however, most experimental studies do not measure the actual area in contact (as it is difficult to assess the amount of area in contact), but bypass the experimental uncertainties by correlating the data to contact pressure.

A review of the duplex tube data for contact areas reveals a lot of scatter with EBR-II having 80% metal-to-metal contact area, back calculations on the heat transfer coefficients indicate a 10% (50,000 BTU/hr-ft²-°F) and 0.10% (2000 BTU/hr-ft²-°F) for the Hallam data. This amount of scatter indicates the importance of the duplex fabrication process in providing a large amount of metal contact area and in reducing the uncertainties in the gap conductance calculations.

A majority of analytical models avoid the difficulty of determining the metal-to-metal contact area by utilizing the contact pressure. The semi-empirical Equations 4.1-3 and -5 and data by Wheeler (Ref. 4.1-14), are plotted in Figure 4.1-4 for vacuum conditions and provide a basis for evaluating only metal-to-metal conductance effects. When helium is included in the interfacial gap, these curves are shifted upward and are illustrated in Figure 4.1-5. Although the large

EFFECTS OF CONTACT PRESSURE ON
GAP CONDUCTANCE - VACUUM CONDITIONS
FIGURE 4.1-4

EFFECTS OF CONTACT PRESSURE ON
GAP CONDUCTANCE - HELIUM AT INTERFACE
FIGURE 4.1-5

4.1.1.4 DISCUSSION (continued)

uncertainty present in the various curves still exists, the formula of Hsieh and Touloukian, Equation 4.1-4 is recommended for usage as it is the most recent semi-empirical correlation, predicts reasonable trends, and is in a simple closed form solution. A conservative but realistic contact pressure of 1000 psi (See Section 4.2) was used in evaluating Equation 4.1-4 in which a gap conductance of $7,000 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$ is obtained and is recommended for the tube bundle sizing calculations (See Section 4.1.2).

The estimate of the contact pressure was made only on the basis of the applied internal pressure of 2600 psi in order to supply a lower bound. A contact pressure, due solely to duplex tube fabrication methods, of 3800 psi is indicated by Reference 4.1-9 (EBR-II data) and increases to 4300 psi when a 1000 psi internal pressure is applied. It is noted that the contact pressure increases by approximately one-half of the applied load. It is expected that the duplexing process will result in a contact pressure of 1000-3000 psi; however, since the fabrication process represents a major uncertainty in the contact pressure determination, this contribution to the interface gap conductance will be neglected. The interface gap pressure is therefore assumed to occur only from the internal tube pressure of 2600 psi. A contact pressure of 1000 psi, approximately 40% of the internal tube pressure, is assumed as inferred from the internal pressure loading effects found to occur in the EBR-II tubing. This value for contact pressure is a realistic but conservative, lower bound for

4.1.1.4 DISCUSSION (continued)

sizing of the interfacial gap conductance.

Another interesting consideration is that oxygen and moisture contamination of the gap may result in a higher contact pressure and therefore a better gap conductance. The Westinghouse data indicated, in at least one experiment, that gap contamination resulted in a "swelling" of the metal of the interface and that this swelling resulted in better heat transfer characteristics. That is, the improvement in heat transfer appeared to occur because a very tight fit was already obtained by the duplexing procedure, and that the contaminants built up or "grew" (due to surface attack) enough to close the helium gap and perhaps increase the interface contact pressure, such that there is more metal-to-metal contact after contamination than before it. Another Westinghouse tube had more slop at the interface and in this tube the heat transfer characteristic decreased 15% with gap contamination. Although both of these tests appear to contradict each other, they do seem reasonable if the gap separation distance is considered. In one case, the gap is large and contaminants can build up enough to decrease interface heat transfer and still maintain a helium gap. In the other case, the gap is small and the contaminant build-up can destroy the helium gap and provide a better heat conducting medium. Of course,

4.1.1.4 DISCUSSION (continued)

the duplexing procedure should be "clean" and contaminants avoided; however, should foreign material become lodged at the interface gap, they may improve heat transfer across the gap if a close fit is initially obtained.

The recommended gap coefficient is very large but will still have an effect on the heat transfer area. Typical values, as taken from the FBR/CP-I (Ref. 4.1-17) units,* of heat transfer coefficients are listed in Table 4.1-III along with the new overall heat transfer coefficients. The decrease in coefficients, due to the gap, is expected to be 7% for the evaporator, 6% for the superheater, and 4% for the reheater. The decrease is different for each unit as the heat transfer characteristics are also different in each case. It is interesting to note that a gap conductance of 50,000 BTU/hr-ft²-°F is needed if the overall heat transfer coefficient is to decrease by only one percent. This large gap conductance corresponds to a (P/Y) ratio (in equation 4.1-4) of 0.27 or a true metal-to-metal contact area (i.e., the b_m term in Equation 4.1-9) of 12%.

*The FBR/CP-I plant is a Combustion Engineering 1000 Mwe, 2400 Mwt 3-loop commercial prototype plant. The plant conditions for which the duplex tube steam generator has been sized is 1000 Mwe, 3800 Mwt, 4-loop large plant. The operating conditions in both plants are the same.

TABLE 4.1-III

EFFECTS OF GAP CONDUCTANCE ON THE
STEAM GENERATOR HEAT TRANSFER COEFFICIENTS

Unit	Estimated Overall Heat Transfer Coefficient (Btu/hr-ft ² -°F)	Gap Conductance (Btu/hr-ft ² -°F)	New Overall Heat Transfer Coefficient (Btu/hr-ft ² -°F)	Percent Change Due to Gap Conductance
Evaporator	500	7000	467	-7%
Superheater	450	7000	423	-6%
Reheater	300	7000	288	-4%

4.1.1.5 CONCLUSIONS

1. A literature survey of existing duplex tube data and models which analytically predict gap interface thermal conductance has been reviewed and compiled.

2. An interface gap conductance of 7000 BTU/hr-ft²-°F is recommended as a conservative lower bound. It is concluded that Equation 4.1-4 provides a reasonable closed-form method of analysis and is recommended for future design calculations.

3. This gap coefficient will decrease the overall heat transfer coefficient by 7% in the evaporator, 6% in the superheater, and 4% in the reheater. A gap conductance of 50,000 BTU/hr-ft²-°F is needed if the overall heat transfer coefficients are to decrease by only one percent.

4.1.2 THERMAL SIZING OF COMPONENTS

The purpose of this task was to size the duplex bayonet tube concept for the large plant size, 950 Mwt per heat transport loop, * and to provide the thermal-hydraulic characteristics and sizing for the evaporator, superheater, and reheater. Two important considerations involved in this analysis are the effects of (a) the increased thermal resistance in the duplexed wall, and (b) the gap contact conductance ** on overall unit size. It is of interest to determine the magnitude of these effects as the general trends in both these cases are known. The duplex tube steam generators will require more heat transfer area and will be correspondingly larger than a single-walled tube unit as a result of the additional thermal resistance of the thicker tube walls. The overall size of the steam generating system will increase or decrease as the gap conductance decreases or increases, respectively.

* One steam generating unit is comprised of two evaporators yielding 566 Mwt, one superheater yielding 222 Mwt and one reheater yielding 162 Mwt. There are four steam generating units per LMFBR plant.

**The gap contact conductance refers to the thermal transport across the interface of two surfaces in contact in which the major heat transfer mode is conduction through both a gaseous fluid and metal-to-metal contact at the interface. The term interface conductance refers to the combined effect of the gap contact conductance and the helium conductance present in the leak detection grooves.

4.1.2.1 DESIGN BASIS AND OPERATING CONDITIONS

The bayonet tube concept has been a reference concept at Combustion Engineering for some time. The adaptation of a duplex tube to a bayonet concept will result in a modification of previous designs; however, many of the design concepts will not be affected by the duplex tubing (i.e., flow shrouds, tube support structure, tubesheet size, leak detection technique, etc.). In the case of bayonet tubing values, this design basis has been established (Ref. 4.1-18 and 19) and these thicknesses and diameters are used in the analysis. Two different duplex tube designs were considered and are illustrated in Figure 4.1-6 as a reference design (the suggested design) and an alternate design (the back-up design). The reference design has a uniform inner tube thickness in the tubesheet section of the unit (See Region A in Figure 4.1-6) while the alternate design utilizes a non-uniform inner tube thickness over the same tubesheet-to-tubesheet length. The duplex pressure tube thicknesses used were designed against rupture and buckling failure modes with a complete analysis available in Reference 4.1-20. The tube pitch was selected by providing a 0.5-inch ligament between pressure tubes. The basis for this value is that it provides a reasonable flow area and provides enough distance so that egg crate tube supports can be used to support the tubing. A summary of the tube thicknesses and diameters for each unit is given in Table 4.1-IV for

DUPLEX TUBE SCHEMATIC
FIGURE 4.1-6

TABLE 4.1-IV
 DUPLEX BAYONET TUBE THICKNESSES AND DIAMETERS
 (All Dimensions in Inches)

Unit Dimensions	REFERENCE DESIGN			ALTERNATE DESIGN		
	Evaporator	Superheater	Reheater	Evaporator	Superheater	Reheater
Inner Bayonet Tube I.D. and O.D.	.501/.581	.636/.716	1.125/1.205	.501/.581	.636/.716	Identical to Reference Design Dimensions
Outer Bayonet Tube I.D. and O.D.	.701/.781	.836/.916	1.325/1.405	.701/.781	.836/.916	
Pressure Tube I.D. and O.D.	1.156/1.45	1.116/1.404	1.80/2.036	1.156/1.388	1.116/1.354	
Tube Pitch (60° Hexag- onal Layout)	1.95	1.904	2.536	1.888	1.854	
Pressure Tube Inner and Outer Bayonet Tube Thicknesses	.067/.080	.062/.082	.055/.063	.038/.078	.039/.080	

4.1.2.1 DESIGN BASIS AND OPERATING CONDITIONS (continued)

both the reference and alternate designs. The alternate design duplex pressure tube thicknesses are slightly smaller than those of the reference design as a result of a different pressure tube design criterion (Ref. 4.1-20).

The tubing material for the evaporator and superheater, for both reference and designs, is Sandvik HT-9. The use of the HT-9 material in the reheater is not necessary in that the added strength of this steel results in a wall thickness which is too thin to accommodate manufacturing. Since no operating, fabrication, or economic advantage is gained through the use of a high-strength steel, the more common material, 2.25 Cr-1 Mo is used for both the reference and alternate designs of the reheater.

The flow direction in the superheater and reheater units is down the heat transfer annulus (entering flow) and up the center riser. Since boiling stability must be maintained in the evaporator unit, and since this can only be done by boiling in upflow, flow direction is reversed in that unit such that flow enters the downcomer and exits from the heat transfer annulus. The direction of flow has no bearing on this preliminary analysis, as counterflow is maintained between the water and sodium; however, the flow directions described above are considered desirable since the high pressure tubesheets operate at relatively cool temperatures.

4.1.2.1 DESIGN BASIS AND OPERATING CONDITIONS (continued)

The operating conditions for the steam generating system are identical to those of the Large Plant FBR/LP-I design (3800 Mwt rating) of reference 4.1-21. These system conditions are listed in Table 4.1-V.

4.1.2.2 BASIC MODELING AND ASSUMPTIONS

The modeling of the steam generator system was performed using the energy equation:

$$q = UA \text{ (LMTD)} \quad \text{EQ \#4.1-10}$$

where:

q = energy transferred by the unit (Btu/hr)

U = overall heat transfer coefficient (Btu/hr-ft²-°F)

A = heat transfer area (ft²)

LMTD = log mean temperature difference (°F)

The components comprising the terms in Equation 4.1-10 will be further described in the following sections. The assumptions implicit in this equation include: (1) Steady state conditions, (2) material properties do not vary with temperature, (3) fully developed flow and corresponding constant convective heat transfer coefficients, and (4) no regenerative heating occurs across the inner and outer bayonet tubes.

TABLE 4.1-V

FBR/DUPLEX BAYONET TUBE STEAM GENERATOR
CONDITIONSTHERMAL RATING (Mwt):

Total Plant (Four 950 Mwt Units) 3800

POWER SPLIT (Mwt/percent of total per 950 Mwt unit)

(a) Evaporator (2 units)	566/ 60%
(b) Superheater	222/ 23%
(c) Reheater	162/ 17%

STEAM SIDE

	Temp. °F	Press. lb/in ²	Enth. BTU/lb
(a) Evaporator inlet	478	2650	463
(b) Evaporator outlet	690	2600	1130.2
(c) Superheater inlet	686.3	2550	1130.2
(d) Superheater outlet	905	2465	1391.5
(e) Reheater inlet	548	555	1260.6
(f) Reheater outlet	902	535	1465.5
(g) Turbine throttle conditions			
1. Superheater inlet	900	2415	1391.5
2. Reheater inlet	900	515	1465.5

(h) Evaporator flow (10^6 lb/hr)	1.449 (per unit)
(i) Superheater flow (10^6 lb/hr)	2.898
(j) Reheater flow (10^6 lb/hr)	2.695

SODIUM SIDE

(a) Superheater and Reheater inlet (°F)	975
(b) Superheater and Reheater outlet (°F)	854
(c) Evaporator inlet (°F)	854
(d) Evaporator outlet (°F)	675
(e) Evaporator flow (10^6 lb/hr)	17.8 (per unit)
(f) Superheater flow (10^6 lb/hr)	20.58
(g) Reheater flow (10^6 lb/hr)	15.02

4.1.2.2 BASIC MODELING AND ASSUMPTIONS (continued)

The first three assumptions have very little effect on the resulting analysis and can be neglected. The material properties used in the analysis do vary greatly for the temperature range considered and can be described by a mean value. For example, in the superheater, the sodium temperature varies from 686°F to 905°F with the sodium specific heat varying from 0.307 to 0.302 Btu/lb-°F (Ref. 4.1-22). The water-steam and wall material properties also show only a small variation (generally less than 5%) for the temperature ranges considered.

The third assumption is conservative as entrance effects, and the presence of "eggcrate" supports on the sodium side, would result in a developing flow condition (on both tube and shell sides) which would result in a slightly better convective heat transfer coefficient, h (Btu/hr-ft²-°F). This improvement in h would result in a slightly lower heat transfer area, A . An estimate of the increase in h can be ascertained by investigating the entrance heat transfer coefficient suggested by Kroll in Reference 4.1-23:

$$\frac{h_{\text{local}}}{h_{\text{infinite}}} = 1 + \left\{ \frac{D_H}{L} \right\}^{0.7} \quad \text{for } 2 \leq \frac{L}{D_H} < 60 \quad \text{EQ \#4.1-11}$$

where

D_H = Hydraulic diameter (ft)

L = Tube length (ft)

4.1.2.2 BASIC MODELING AND ASSUMPTIONS (continued)

Integration of this equation over the range $2 \leq L/D_H$ ≤ 60 and for a 0.5 inch value of D_H results in an average increase in h_{infinity} of 12% for the first 2.5 feet after which h_{local} is set equal to h_{infinity} . Considering that the effective heat transfer length is at least 30 feet, the entrance improvement in the tube side heat transfer coefficient can be considered in the overall design as a small (about 1%), built-in safety factor which renders the final overall unit sizing slightly conservative. The same reasoning applies to the shell side; however, eggcrate supports are placed approximately every 3 feet so the heat transfer enhancement occurs along the total tube length. This effect is not as great as it seems because the shell heat transfer coefficient is based on sodium flow and falls in the 5000 Btu/hr-ft²-°F range. The fact that the overall heat transfer coefficient, U , falls in the 200-600 Btu/hr-ft²-°F range, effectively reduces the improvement in the sodium coefficient. Based on the conditions quoted, this assumption results in a conservatism of about 8% in the heat transfer area.

The final assumption ignores the effects of regenerative heating which occurs across the inner and outer bayonet tubes. That is, as the cold fluid enters and flows down the downcomer, heat is transferred across the inner and outer bayonet tube from the hotter fluid in the heat transfer annulus, as shown in Figure 4.1-7. The net effect is that the bayonet tube length must now be longer (i.e., more heat transfer area),

REGENERATIVE HEATING
SCHEMATIC
FIGURE 4.1-7

4.1.2.2 BASIC MODELING AND ASSUMPTIONS (continued)

since the fluid in the annulus is cooler than if no heat were conducted across the inner bayonet tubes, in order to maintain the same exit temperature conditions. The amount of regenerative heating that occurs is a function of the insulating tubes, the size of the insulating gap, and the fluid in the gap. The scope of this task was limited and regenerative heating effects were not to be analyzed; however, previous reports (Ref. 4.1-19 and 23) have indicated that the tube length must increase by 6-10%, depending upon the unit design, in order to maintain the necessary steam generator exit temperatures.

In respect to the overall heat transfer coefficient, no fouling factors were considered. This effect, along with the calculational uncertainties associated with the individual film coefficients, was lumped into an overall safety factor. In keeping with previous parametric design studies (Ref. 4.1-18 and 19) a 10% safety factor was employed.

There are many uncertainties which affect the required size and performance of the steam generator. Among these are: (1) Water fouling factors in the range 1000 - 4000 Btu/hr-ft²-°F (Ref. 4.1-25) (sodium side fouling sometimes occur due to contamination with organics; a value of 4000 Btu/hr-ft²-°F represents a conservative estimate), (2) estimation of the number of plugged tubes over the unit's lifetime, (3) estimation of flow distribution effect on the unit performance*, and (4) regenerative heating effects in the case of the bayonet tube concept.

*However, tube corrosion tolerance of 0.003 inch for the water side, 0.008 inch for the sodium side, along with a 5% manufacturing tolerance (Ref. 4.1-26) were considered and included in the tube thicknesses.

4.1.2.2 BASIC MODELING AND ASSUMPTIONS (continued)

These effects are somewhat offset by the entrance effects discussed above. A more detailed design procedure should account for these contingencies more accurately.

As reflected in Section 4.1.1, a gap conductance of $7000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ was selected as recommended in Reference 4.1-27. This value was parametrically varied, from this reference point, in order to ascertain its effects on the overall unit size.

4.1.2.3 THERMAL-HYDRAULIC CALCULATIONS

The thermal sizing of all the units was accomplished through the use of the energy equation, Equation 4.1-10. A complete breakdown of the sizing procedure and the other equations necessary to evaluate Equation 4.1-10 are discussed in Appendix A. The hydraulic, or pressure drop, relationships used for this analysis are given in Appendix B. The set of equations in Appendix A and B were solved in order to obtain the preliminary thermal-hydraulic sizings for each of the units.

4.1.2.4. DISCUSSION

A. Thermal-Hydraulic Sizing

The thermal-hydraulic sizing of a heat exchanger can be considered more of a design problem than a mathematical exercise. While the analytical methods are tedious, they are generally straight forward. The design considerations, however, are more qualitative in nature and a "trade-off" study must generally be made.

In a trade-off study, once the ranges of the design variables are defined and the basic parametric trends are established, the optimal unit design may not be obvious because of the large number of variables or restraints involved. For example: (1) the heat transfer area should be as small as possible as the economic benefits generally increase as the unit size decreases, (2) a small pressure drop is desired so as to reduce pumping costs but not so small as to cause flow instabilities or significantly affect the heat transfer characteristics at low load conditions, and (3) a reasonable length to bundle diameter ratio should be maintained in order to obtain good flow distribution and to avoid unusual fabrication problems. The necessity of trading off these variable constraints is an integral part of the thermal-hydraulic sizing procedure.

First, the parametric trends of each unit must be established. The overall characteristics of the reference design

4.1.2.4. DISCUSSION (continued)

evaporator, superheater and reheater are shown in Figures 4.1-8, 9 and 10.

The alternate design unit characteristics are given in Figures 4.1-11 and 4.1-12 (both the reference and alternate design reheaters are identical).

In calculating the heat transfer performance characteristics of the duplex bayonet tube units, the estimated pressures listed in Table 4.1-V were utilized. Pressure drops between component exits and the turbine island were assumed at representative values so that conservative design conditions are employed. Pressure drops across and between components were assumed at typical values; their exact magnitudes have a secondary effect on tube bundle sizes.

The effects of fluid velocity on the overall heat transfer coefficient, U , which is directly related to the area, A , are seen in Figures 4.1-8 through 12. The area A decreases as the fluid velocity increases. After some point further increases in velocity result in only minor decreases in heat transfer area.

At the same time, however, the pressure drop through the steam side increases with velocity. These higher pressure drops are accepted as long as the associated incremental gain in performance justifies it.

There is no definite minimum pressure drop requirement, but a reasonable pressure drop must be maintained to

FIGURE 4.1-8

FIGURE 4.1-9

FIGURE 4. 1-10

FIGURE 4.1-11

FIGURE 4.1-12

4.1.2.4. DISCUSSION (continued)

insure good flow distribution and avoid thermal-hydraulic instabilities even at low load operation. If the unit operates at 20% load, the pressure drop will decrease as the square of the velocity and drop to approximately 4% of its full load pressure drop. Thus, the pressure drop is made as large as possible so as to decrease the heat transfer area and avoid low load flow instabilities.

These considerations were used in determining the final size of the reference and alternate designs. For these particular units, the pressure drop criteria governed the final unit selection as this (1) coincided with near minimum heat transfer area, and (2) the final tube length sizes were within reasonable limits. The selected design is denoted in Figures 4.1-8 thru 12 by the vertical line marked "reference velocity". These finalized design values, along with the unit design conditions, are tabulated in Table 4.1-VI.

B. Gap Conductance Effects

A major uncertainty in the duplex tube concept is the effect of the gap conductance on the steam generating component size. As mentioned previously, the advantage of the duplex tube design lies in its inherently high safety factor against a sodium-water reaction and a higher operating availability with less downtime due to leakage. The disadvantage of the design is that it needs a large heat transfer area as a result of the added thermal resistance present in double-walled tube (as opposed to a single-walled tube). This disadvantage is further augmented if a low gap conductance is present at the inner duplexed surface. It was part of this task to assess the effects of the gap conductance on the duplex tube design and determine its implication on the overall unit design.

4. 1. 2. 4 DISCUSSION (continued)

A parametric study of the effects of the gap conductance on the overall heat transfer coefficient was performed to answer this question. The results indicate that the gap conductance can significantly affect all the units, as shown in Figures 4.1-13, 14 and 15. An increase in the gap conductance, h_{gap} , from the reference value of 7000 Btu/hr-ft²-°F, by a factor of two will result in an increase of the coefficient, U, of approximately 8% for the evaporator and superheater (this corresponds to a heat transfer area decrease also of 8%). The reheater is not affected as the gap conductance is essentially at the limiting value and any increase in gap conductance will have little effect on the unit size. It is indicated in Reference 4.1-27 that the other two units will reach a limiting value when h_{gap} approaches 50,000 Btu/hr-ft²-°F. It is concluded that an increase in h_{gap} to higher values will result in further savings in heat transfer area; however, the corresponding savings will be small as the estimated h_{gap} value is near the optimal value.

A decrease in the gap conductance, from the reference value, is more significant and can substantially affect the unit size. As illustrated in Figures 4.1-13, 14, and 15, the effects of gap conductance on the overall heat transfer coefficient is non-linear. A decrease in the gap conductance to 4000 Btu/hr-ft²-°F, almost a factor

FIGURE 4.1-13

FIGURE 4.1-14

FIGURE 4.1-15

4.1.2.4 DISCUSSION (continued)

of two, will also approximately decrease the heat transfer coefficient by 8%. Any further decrease in the gap conductance will have a larger and more significant effect on the unit's design size. As an example, for the worst case of a poorly fabricated duplex tube, Reference 4.1-27 indicates that a gap conductance of $1000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ represents a lower bound. This corresponds to an increase in heat transfer area of about 95% for the evaporator (this curve is not shown in Figure 4.1-13), 50% for the superheater, and 25% for the reheater. Although this is the worst case, it illustrates the importance of maintaining a good gap conductance at the duplex interface. It is difficult to assess an uncertainty factor to the gap conductance; however, a survey of duplex tube operating conditions (Ref. 4.1-27) indicates that the $4000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ value for h_{gap} appears to be a reasonable lower bound. This corresponds to an increase of approximately 8% in heat transfer area with the exact values shown in Figures 4.1-13 thru 15.

The effects of the gap conductance become important as the gap conductance approaches the same order of magnitude as the other conductance terms. This can be illustrated through the thermal resistances contained in the overall heat transfer coefficient:

4. 1. 2. 4 DISCUSSION (continued)

$$U = \frac{1}{R_{\text{sodium}} + R_{\text{wall}} + R_{\text{interface}} + R_{\text{wall}} + R_{\text{steam}}} \quad \text{EQ \#4. 1-12}$$

where:

R = thermal resistances indicated by the subscripts (hr-ft²-°F/BTU)

The interface resistance is comprised of the gap conductance and the helium conductance of the leak detection grooves; however, the effect of the helium in the grooves is very small and the interface resistance can be approximately defined as the inverse of the gap conductance, or:

$$R_{\text{interface}} = \frac{1}{h_{\text{gap}}} \quad \text{EQ \#4. 1-13}$$

Typical resistance values* for the superheater are: (1) R_{sodium} is 0.0002 (hr-ft²-°F/Btu), (2) R_{wall} is 0.0008, (3) $R_{\text{interface}}$ is 0.00015, and (4) R_{steam} is 0.0008.

*Resistance values are factored by a diameter ratio so that the heat transfer area is defined in terms of the tube O.D. See Appendix A for the exact formulation.

4.1.2.4 DISCUSSION (continued)

The major thermal resistance is comprised of the wall resistance and steam film resistance. The effects of the interface resistance become more important as its value approaches that of the wall and steam resistances. The steam film coefficient is quite different in each unit with the evaporator having the largest film coefficient, the superheater next, and the reheater having the lowest film coefficient. Thus, the units with the lowest overall thermal resistance are most affected by the interface resistance. The effects of the gap resistance on unit size are shown in Figure 4.1-16 on an area normalized basis. As expected, the reheater is the least affected by a gap conductance change (a low steam film coefficient) while the evaporator is the most affected (a large water film coefficient).

It is of interest to compare the duplex tube design to that of a single-walled tube in order to provide a complete assessment of the duplex design. Single-wall tube thicknesses have been calculated in Reference 4.1-20 and, for the reference design, thicknesses are: evaporator 0.082 inches; superheater - 0.088 inches, and reheater - 0.085 inches. These tube thicknesses include corrosion and manufacturing tolerances. A comparison of the duplex tube units and the single-walled tube units are shown in Figure 4.1-17. At large gap conductances, the duplex evaporator needs about 26% more heat transfer area than a single-

FIGURE 4.1-16

FIGURE 4.1-17

4. 1. 2. 4 DISCUSSION (continued)

walled unit, the duplex superheater needs 22% more, and the reheater needs 5% more heat transfer area. It is noted that although the single-walled tubes are about 45% thinner than the duplex tubes, this entire percentage is not carried over directly as a reduction in heat transfer area (for the single-walled tube units) because the thermal resistances of the water and sodium films still comprise a large part of the overall heat transfer resistance. As illustrated in Figure 4.1-17, as the gap conductance in the duplex tube unit decreases, an increasing percentage of heat transfer area is needed. Thus, the necessity of maintaining a large gap conductance is essential if it is to become an economically viable concept.

4. 1. 2. 5 CONCLUSIONS

1. The thermal-hydraulic sizing of a duplex tube steam generating system comprised of an evaporator, superheater, and reheater has been performed. Overall dimensions are listed in Table 4.1-VI.

2. The water-side pressure drop was the major design parameter controlling the unit sizing, and this criterion gave the lowest value of heat transfer area.

TABLE 4.1-VI
OVERALL THERMAL-HYDRAULIC CHARACTERISTICS

Parameter \ Unit	REFERENCE DESIGN			ALTERNATE DESIGN		
	Evaporator	Superheater	Reheater	Evaporator	Superheater	Reheater
Tube Length* (feet)	48	30	23	42	27	Identical to Reference Design Dimensions
Bundle I.D. (feet)	4.3	6.4	7.8	4.3	6.4	
Tube Number (-)	610	1346	1273	610	1352	
Heat Transfer Area (ft ²)*	11,200	14,700	15,500	9,300	13,400	
Overall Film Coefficient (Btu/hr-ft ² -°F)	515	500	250	605	550	
LMTD	180	112	183	180	112	
Average Water or Steam Velocity (ft/sec)	3.5(inlet)	60	104	3.5(inlet)	60	
Average Sodium Velocity (ft/sec)	12	9	5	12	9	
Water Pressure Drop (psi)	22	70	22	18	70	
Sodium Pressure Drop (psi)	18.5	3	1.5	16	3	
Material	HT-9	HT-9	2.25 Cr-1 Mo	HT-9	HT-9	↓

*A ten percent (10%) factor of safety is included.

4.1.2.5 CONCLUSIONS (continued)

3. The gap conductance is a critical factor in the duplex tube design. For this system, the gap conductance is near its optimal upper level and an increase in gap conductance will have only a small effect on unit size. However, a decrease in gap conductance can have a significant effect on heat transfer area. An overall comparison between the heat transfer area needed in a duplex tube design and a single-walled design is illustrated in Figure 4.1-17.

4.1.3 EFFECTS OF TUBE MATERIAL ON GROSS EVAPORATOR SIZE

The high chromium content, Sandvik HT9 steel was assumed in sizing the duplex bayonet tube steam generator components. In order to evaluate the sizing advantage offered by this material, the evaporator unit was resized assuming the lower chromium SA 213 T22 tubing material. Tube wall thickness for both materials are summarized as follows:

	<u>Sandvik H-T9</u> (inch)	<u>SA 213 T22</u> (inch)
Inner pressure tube	0.062	0.131
Outer pressure tube (including .020 inch groove)	0.082	0.132
	_____	_____
Combined thickness	0.144	0.263

The added thickness associated with the SA 213 T22 causes a 56% increase in overall heat transfer resistance. Thus, using SA 213 T22 tubing in lieu of the Sandvik HT9 tubing would result in a 56% increase in heat transfer area for the evaporator unit.

4.1.4 ADDED SURFACE AREA ASSOCIATED WITH USING DUPLEX TUBES IN LIEU OF SINGLE WALLED TUBES

As reflected in the above summary of tube wall thicknesses, the combined thickness of the inner and outer evaporator pressure tubes is 0.144 inch assuming Sandvik HT9 material is used. If the unit were designed as a single walled unit using the same material, the pressure tube wall thickness would be 0.082 inch. Thus assuming the same heat transfer coefficients, etc., the evaporator heat transfer area would be 37.5% less for a single walled design than it is for the duplex tube design. Similar reductions in area would be noted in the superheater and reheater units if single walled tubes were considered.

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4.2 STRUCTURAL CONSIDERATIONS

There are several unique aspects of the duplex tube concept that influence their basic structural sizing.

The ASME Code does not specifically require that both tubes in a duplex design be sized for full steam side pressure. The first part of the sizing procedure involved the development of an acceptable criteria. This was followed by calculation of specific thickness recommendations for the preliminary design.

Another aspect of the concept is the potential for relative sliding between the duplex tubes. This was studied to rule out the potential problems associated with gross fretting action.

It was necessary to study the contact pressure between the duplex tubes to justify the assumptions made in structural sizing, thermal sizing, relative sliding and fabrication studies.

4.2.1 TUBE SIZING

The work reported herein was performed in satisfaction of Task 3.1.3 of Alternate Steam Generator Study (Ref. 4.2-1).

"3.1.3 The ASME Code does not require that both tubes in a duplex design be sized for the full steam side pressure.

An evaluation of the design will be made and recommendations for the tube sizing will be developed."

Because the ASME Code (Ref. 4.2-2) does not specifically cover tube sizing for duplex tubes in contact, the first part of this task involved the development of an acceptable tube sizing procedure. This was followed by calculation of tube thicknesses and formulation of specific thickness recommendations for the preliminary design. These recommendations represent the minimum combined tube thickness to maintain structural reliability against ductile rupture and external pressure buckling. Also, these recommended minimum thicknesses reflect the desire to minimize cost of the steam generator components, whose unit size and hence, cost, are directly affected by duplex tube thickness.

4.2.1.1 DEFINITION OF THE STRUCTURAL PROBLEM

Generally, every structural design evaluation consists of two parts, a structural analysis and an evaluation (failure analysis). The structural analysis determines the response of the structure to

4.2.1.1 DEFINITION OF THE STRUCTURAL PROBLEM (continued)

the applied loads, and requires specification of the geometry, loads, and the material behavior model. The failure analysis is an evaluation of the structural response for specified failure modes. Although we have well developed theories for structural analysis, the only practical theories of failure that we have for engineering construction are those contained implicitly in the ASME Design Codes.

a. Geometry

The duplex tube geometry concept is shown in Figure 4.2-1. This figure was taken from Ref. 4.2-1 and was modified to indicate the structural connection between the upper and lower tubesheets. Two geometries were investigated - a reference design and an alternate design. The reference design (left side of Figure 4.2-1) has a uniform thickness inner tube throughout its length. The alternate design (right side of Figure 4.2-1) has a decreased thickness of the inner tube over the length of the sodium chamber. The reference design offers ease of fabrication. The alternate design offers the minimum combined thickness and consequent savings in heat transfer efficiency. A cross section of the duplex tube is shown in Figure 4.2-2. The depth of the helium leak detection grooves, e_G , was assumed equal to 0.020 inches for all the preliminary sizing calculations.

SCHEMATIC OF DUPLEX
TUBE
FIGURE 4.2-1

FIGURE 4.2-2
DUPLEX TUBE CROSS SECTION

4.2.1.1 DEFINITION OF THE STRUCTURAL PROBLEM (continued)

b. Loading

For this preliminary evaluation, the significant loadings on the duplex tube (after expansion) were judged to be:

1. Pressure of steam, sodium, and helium as shown in Figure 4.2-1.
2. Steady state axial and radial temperature distribution.
3. Axial and radial temperature distribution during a thermal transient.

c. Material

The material initially adopted for the duplex tubes of all steam generator components was HT-9, as agreed in Reference 4.2-3. However, interim results revealed that 2 1/4 Cr-1 Mo was more than adequate for the reheater and thus offered economic savings over HT-9 for the construction of this unit.

Material behavior models considered for structural analysis include elastic, plastic, and creep of various degrees of complexity. Generally, a simple elastic material behavior model was judged sufficient for the preliminary sizing analysis of the duplex tubes.

4.2.1.1 DEFINITION OF THE STRUCTURAL PROBLEM (continued)

d. Failure Modes

The application design code, ASME Code

Case 1592 (Ref. 4.1-4), addresses the following failure modes:

1. Ductile rupture (plastic or creep),
2. Ratcheting, or excessive deformation,
3. Creep-fatigue failure, and
4. Buckling.

Two additional failure modes are also acknowledged here for the case of the duplex tube:

1. Desire to maintain tube contact, and
2. Prevention of relative sliding between the tubes.

For the purpose of determining duplex tube thicknesses for preliminary design, only two of the above ASME Code Case failure modes were considered in the evaluation; these were ductile rupture and buckling. The evaluation of the two additional failure modes are reported in Sections 4.2.2 and 4.2.3.

4.2.1.2 THICKNESS SIZING PROCEDURE

4.2.1.2.1 RUPTURE CRITERIA

In respect to rupture, the pressure loads (P_{H_2O} , P_{He} , P_{Na}) and the tube expansion loads are considered most significant. The swaging loads are evaluated in Section 4.2.3.

4.2.1.2.1 RUPTURE CRITERIA (continued)

The duplex tube and header arrangement was analyzed in three regions (A, B, and C of Figure 4.2-1) for response to various load combinations. The rupture criteria assumed are the general primary membrane stress intensity limits (P_m) for design, normal, upset, and emergency conditions. These criteria, as prescribed by Reference 4.2-4, are as shown in Table 4.2-1.

The loading conditions assumed significant are described as follows:

1. Normal - normal operating conditions associated with the operating pressures and temperatures.
2. Upset - Sudden application or sudden loss of operating pressure on either side of tube in regions A and B. Maximum possible pressure differential across tube wall (Region C) which can occur accidentally by loss or application of any single operating pressure.
3. Emergency - Occurrence of a steam leak through the inner tube. Shutdown is expected as soon as possible after detection of a leak in order to plug the leaking tube. An uncertainty is the estimation of the pressure in the leak detection grooves of the outer tube. Upper and lower bounds on the leak pressure, P_{leak} , are steam pressure, P_{H_2O} , and helium pressure, P_{He} , respectively.

TABLE 4.2-I
RUPTURE CRITERIA

DESIGN CONDITIONS	$P_m = S_o$
NORMAL CONDITIONS	$P_m = \begin{cases} S_m \\ S_t \end{cases}$
UPSET CONDITIONS	$P_m = \begin{cases} S_m \\ S_t \end{cases}$
EMERGENCY CONDITIONS	$P_m = \begin{cases} 1.25 S_m \\ S_t \end{cases}$

Where P_m is obtained from the elastically calculated stresses, and S_o , S_m and S_t are code allowable stresses from Reference 4.2-4.

4.2.1.2.1 RUPTURE CRITERIA (continued)

4. Design - It is assumed that the ASME code intends that design conditions should represent an upper bound to operating loads and temperatures from 1 to 3 above, and also other more severe conditions (faulted) not considered here. However, in view of the possibility that design conditions may be either unrealistic or excessively conservative, thicknesses were calculated with and without design conditions for comparison.

The design and operating conditions (pressures and temperatures) and the inner tube inside radii are listed in Table 4.2-II. They are typical for a commercial LMFBR plant and were used for the preliminary sizing of the duplex tubes. Remarks pertaining to the selection of these values are as follows:

1. The operating pressures and temperatures were obtained from a nominal thermal-hydraulic energy balance for C-E's 1500 MWe commercial plant design (Ref. 4.2-5).
2. Design values were obtained by increasing the operating pressures and temperatures to account for:
 - (i) Normal operating fluctuations of pressure and temperature (20°F).
 - (ii) Pressure surges by estimating an equivalent static pressure and to account for maximum rise in temperature associated with expected up-transients.

TABLE 4.2-II

GEOMETRY RESTRICTIONS AND DESIGN CONDITIONS (PRESSURES AND
TEMPERATURES) FOR USE IN DUPLEX TUBE SIZING

UNIT	OPERATING PRESSURE/DESIGN PRESSURE (PSI/PSI)			DUPLEX TUBE ⁽¹⁾ REGION (REGIONS B & C)	BETWEEN (2) TUBE SHEETS (REGION A)	R ₁ (INCHES)
	$\frac{P_{Na}}{D_{PNa}}$	$\frac{P_{He}}{D_{PHe}}$	$\frac{P_{H_2O}}{D_{PH_2O}}$	TEMPERATURE (°F) OPER/DESIGN	TEMPERATURE (°F) OPER/DESIGN	
SUPERHEATER	120/250	300/350	2550/2600	975/1000	905/930	0.558 ⁽³⁾
EVAPORATOR	95/250	300/350	2650/2700	880/900 ⁽⁵⁾	850/875 ⁽⁵⁾	0.578 ⁽³⁾
REHEATER	120/250	300/350	555/600	975/1000	902/930	0.900 ⁽⁴⁾

(1) CORRESPONDS TO Na INLET TEMPERATURE

(2) CORRESPONDS TO H₂O OUTLET TEMPERATURE

(3) FROM CENC-1194, DEMO I DESIGN

(4) FBR CP-1 DESIGN

(5) INCLUDES CONSIDERATION OF LOW LOAD CONDITIONS

4. 2. 1. 2. 1 RUPTURE CRITERIA (continued)

3. It was recognized that the average temperature in region A could be significantly lower than in regions B and C, resulting in a corresponding effect on thickness.
4. Values for the inside radius (R_i) of the duplex tube for preliminary design are referenced by footnotes in Table 4.2-II.

The rupture criteria for the three regions are summarized next. Loads which have a short duration (relative to creep) are limited by S_m evaluated at the appropriate temperature. Loads of long duration (component design life) are limited by S_t evaluated at the appropriate temperature and for 2×10^5 hours. The superscripts A and BC refer to temperatures from Table 4.2-II. Since tube dimensions were expected to be more in the thick wall regime than thin wall, the calculations were based upon thick wall formulation.

A. Between Tubesheets (Region A), Inner Tube

For a thick cylinder with inside radius, R , and thickness, t , the primary membrane stress intensity due to internal pressure (P_i) and external pressure (P_o) is:

$$P_m = (P_i - P_o) \left(\frac{R}{t} + \frac{1}{2} \right) ; \text{ thus,}$$

4.2.1.2.1 RUPTURE CRITERIA (continued)

Normal Operating

$$(P_{H_2O} - P_{He}) \left(\frac{R_1}{t_1^{TS}} + \frac{1}{2} \right) \leq S_t^A \quad \text{EQ \#4.2-1}$$

Upset-Sudden Loss of P_{He}

$$P_{H_2O} \left(\frac{R_1}{t_1^{TS}} + \frac{1}{2} \right) \leq S_m^A \quad \text{EQ \#4.2-2}$$

Design

$$P_{H_2O}^D \left(\frac{R_1}{t_1^{TS}} + \frac{1}{2} \right) \leq S_o^A \quad \text{EQ \#4.2-3}$$

Condensing the above, one obtains:

$$t_1^{TS} \geq \frac{R_1}{S_A} \quad \text{EQ \#4.2-4}$$

$$S_A = \text{Min.} \left\{ \frac{S_t^A}{P_{H_2O} - P_{He}} - \frac{1}{2}, \frac{S_m^A}{P_{H_2O}} - \frac{1}{2}, \frac{S_o^A}{P_{H_2O}^D} - \frac{1}{2} \right\} \quad \text{EQ \#4.2-5}$$

B. Tube Ends (Region B), Spherical End Caps

For a thick sphere of inside

radius, R , and thickness, t , the primary membrane stress intensity due to internal pressure (P_i) and external pressure (P_o) is:

$$P_m = (P_i - P_o) \left(\frac{R}{2t} + \frac{1}{2} \right)$$

4.2.1.2.1 RUPTURE CRITERIA (continued)

Inner Tube

Normal Operating

$$(P_{H_2O} - P_{He}) \left(\frac{R_1}{2t_1} + \frac{1}{2} \right) \leq S_t^{BC} \quad \text{EQ \#4.2-6}$$

Upset-Sudden Loss of P_{He}

$$P_{H_2O} \left(\frac{R_1}{2t_1} + \frac{1}{2} \right) \leq S_m^{BC} \quad \text{EQ \#4.2-7}$$

Design

$$P_{H_2O}^D \left(\frac{R_1}{2t_1} + \frac{1}{2} \right) \leq S_o^{BC} \quad \text{EQ \#4.2-8}$$

Condensing the above, one obtains:

$$t_1 \geq \frac{R_1}{2S_{1B}} \quad \text{EQ \#4.2-9}$$

$$S_{1B} = \text{Min.} \left\{ \frac{S_t^{BC}}{P_{H_2O} - P_{He}} - \frac{1}{2}, \frac{S_m^{BC}}{P_{H_2O}} - \frac{1}{2}; \frac{S_o^{BC}}{P_{H_2O}^D} - \frac{1}{2} \right\} \quad \text{EQ \#4.2-10}$$

Outer Tube (for conservatism, assume tube end is also grooved and only t_G part of thickness resists rupture).

Normal Operating

$$(P_{He} - P_{Na}) \left(\frac{R_G}{2t_G} + \frac{1}{2} \right) \leq S_t^{BC} \quad \text{EQ \#4.2-11}$$

Upset-Sudden Loss of P_{Na}

$$P_{He} \left(\frac{R_G}{2t_G} + \frac{1}{2} \right) \leq S_m^{BC} \quad \text{EQ \#4.2-12}$$

4.2.1.2.1 RUPTURE CRITERIA (continued)

Emergency-Sudden Leak from Inner Tube

$$(P_{\text{leak}} - P_{\text{Na}}) \left(\frac{R_G}{2t_G} + \frac{1}{2} \right) \leq 1.2 S_m^{\text{BC}} \quad \text{EQ \#4.2-13}$$

Design

$$P_1^D \left(\frac{R_G}{2t_G} + \frac{1}{2} \right) \leq S_o^{\text{BC}} \quad \text{EQ \#4.2-14}$$

where:

$$P_1^D = \text{Max.} \left[P_{\text{He}}^D, (P_{\text{leak}}^D - P_{\text{Na}}^D), (P_{\text{leak}} - P_{\text{Na}}) \right] \quad \text{EQ \#4.2-15}$$

Condensing the above, one obtains:

$$t_2 \geq \frac{R_1 + t_1 + e_G}{2S_{2B}} + e_G \quad \text{EQ \#4.2-16}$$

where t_1 is the thickness of the lower part of the inner tube, and

$$S_{2B} = \text{Min.} \left\{ \frac{S_t^{\text{BC}}}{P_{\text{He}} - P_{\text{Na}}} - \frac{1}{2}, \frac{S_m^{\text{BC}}}{P_{\text{He}}} - \frac{1}{2}, \frac{1.2 S_m^{\text{BC}}}{P_{\text{leak}} - P_{\text{Na}}} - \frac{1}{2}; \frac{S_o^{\text{BC}}}{P_1^D} - \frac{1}{2} \right\} \quad \text{EQ \#4.2-17}$$

C. Duplex Tube (Region C)

The ASME Code does not provide specific procedures for the calculation of stress intensities in a duplex tube. Therefore, if ASME Code approval is ultimately required, consistency with the code philosophy should be maintained in the development of such procedures. Code treatment for the prevention of rupture is interpreted

4.2.1.2.1 RUPTURE CRITERIA (continued)

as follows. Rupture will not occur if all of the load controlled stress intensities (membrane and membrane plus linear bending) are below an allowable stress determined for the temperature and load duration. A load controlled stress intensity is one which is produced by an applied thermal or mechanical load which is not diminished by small deformation. The ASME Code identifies three types of load controlled quantities:

1. P_m (membrane), P_B (Linear bending) - derived from stresses which are required for equilibrium with external loads.
2. P_L (membrane) - includes P_m plus an enhanced load transmission caused by adjacent structural discontinuity.
3. Q (membrane plus linear bending) - secondary stress with a large amount of elastic follow-up.

From geometrical and loading considerations, it is assumed that only P_m is significant for the investigation of rupture of the duplex tube. The various pressure loadings which contribute to P_m are the internal (P_1) and external (P_2) pressures, the pressure in the grooves (P_{gap}), and the contact pressure (P_c) at the interface of the tubes. These pressures are illustrated in Figure 4.2-3.

FIGURE 4.2-3

DUPLEX TUBE CONTACT SURFACE AT PROGRESSIVE LEVELS OF MAGNIFICATION

4.2.1.2.1 RUPTURE CRITERIA (continued)

The contact pressure, P_c , is viewed as composed of two parts: (1) load controlled, $P_{c_{LC}}$, and (2) deformation controlled, $P_{c_{DC}}$. The determination of the total contact pressure would require a detailed inelastic analysis. However, the determination of $P_{c_{LC}}$ does not, since it is assumed to be obtainable from equilibrium alone. Therefore, $P_{c_{LC}}$ is independent of material behavior and is at most a function of the applied loads (P_1 , P_2 , P_{gap}) and purely geometric parameters. Furthermore, equilibrium with external loads imposes requirements only on the average and linear through thickness characteristics of the radial stress distribution in the tubes. Thereby, the contact pressure needed for rupture considerations is independent of the local geometric details at the interface. Thus, calculation of average stress components from which P_m is determined used an equivalent load controlled contact pressure, $\bar{P}_{c_{LC}}$. In general:

$$\bar{P}_{c_{LC}} = f(P_1, P_2, P_{gap}, t_1, t_2, \text{other gross parameters such as } R_1 \text{ and } e_G).$$

For cases in which it can be assumed that forces transmitted between the inner tube and outer tube by direct contact are unimportant, this is justified to be a valid assumption in Section 4.2.3, then

$$\bar{P}_{c_{LC}} = P_{gap}, \quad \text{EQ \#4.2-18}$$

where $P_{gap} = P_{He}$, and in case of a steam leak, $P_{gap} = P_{H_2O}$. An

4.2.1.2.1 RUPTURE CRITERIA (continued)

additional approximate formula for $\bar{P}_{cLC}$ was derived as described in Appendix C.

$$\bar{P}_{cLC} = \frac{P_1 t_2 + P_2 t_1}{t_1 + t_2} \quad \text{FQ \#4.2-19}$$

Calculations of thicknesses were carried out using both formulas. However, results from Equation 4.2-18 were chosen for recommended thicknesses.

While it was concluded that neither formula exactly determined $\bar{P}_{cLC}$, further refinement was considered beyond the work scope for the preliminary conceptual design evaluation. Because of some uncertainties associated with the $\bar{P}_{cLC}$ formula, it was attempted to select thicknesses in a manner which conservatively eliminated the dependence of thickness on the choice of the $\bar{P}_{cLC}$ formula. Further research might permit a reduction in these thicknesses through the use of the correct $\bar{P}_{cLC}$ formula.

The explicit rupture criteria for Region C were applied to both tubes combined, and to the outer tube considered separately with $\bar{P}_{cLC}$ acting on the inside. P_m was calculated from the pressures P_1 , P_2 and $\bar{P}_{cLC}$ using the formula for a thick cylinder. To be conservative, no credit was taken for the groove depth of the outer tube in resisting rupture.

4.2.1.2.1 RUPTURE CRITERIA (continued)

Both Tubes Combined

Normal Operating

$$(P_{H_2O} - P_{Na}) \left(\frac{R_1}{t_1 + t_G} + \frac{1}{2} \right) \leq S_t^{BC} \quad \text{EQ \#4.2-20}$$

Upset-Sudden Loss of P_{Na}

$$P_{H_2O} \left(\frac{R_1}{t_1 + t_G} + \frac{1}{2} \right) \leq S_m^{BC} \quad \text{EQ \#4.2-21}$$

Design

$$P_{H_2O}^D \left(\frac{R_1}{t_1 + t_G} + \frac{1}{2} \right) \leq S_o^{BC} \quad \text{EQ \#4.2-22}$$

Condensing the above, one obtains:

$$t_1 + t_2 \geq \frac{R_1}{S_{cc}} + e_G \quad \text{EQ \#4.2-23}$$

$$S_{cc} = \text{Min.} \left\{ \frac{S_t^{BC}}{P_{H_2O} - P_{Na}} - \frac{1}{2}, \frac{S_m^{BC}}{P_{H_2O}} - \frac{1}{2}, \frac{S_o^{BC}}{P_{H_2O}^D} - \frac{1}{2} \right\} \quad \text{EQ \#4.2-24}$$

Outer Tube

Normal Operating

$$(\bar{P}_{c_{LC}}^N - P_{Na}) \left(\frac{R_G}{t_G} + \frac{1}{2} \right) \leq S_t^{BC} \quad \text{EQ \#4.2-25}$$

where $\bar{P}_{c_{LC}}^N = \bar{P}_{c_{LC}}$ evaluated with $P_1 = P_{H_2O}$, $P_2 = P_{Na}$, $P_{gap} = P_{He}$.

4.2.1.2.1 RUPTURE CRITERIA (continued)

Upset-Sudden Maximum Pressure Differential by Loss of P_{Na}

$$\bar{P}_{cLC}^{Na} \left(\frac{R_G}{t_G} + \frac{1}{2} \right) \leq S_m^{BC} \quad \text{EQ \#4.2-26}$$

where

$$\bar{P}_{cLC}^{Na} = \bar{P}_{cLC} \text{ evaluated with } P_1 = P_{H_2O}, P_2 = P_{Na} = 0, P_{gap} = P_{He}.$$

Emergency-Sudden Leak from Steam Side

$$(\bar{P}_{cLC}^L - P_{Na}) \left(\frac{R_G}{t_G} + \frac{1}{2} \right) \leq 1.2 S_m^{BC} \quad \text{EQ \#4.2-27}$$

where

$$\bar{P}_{cLC}^L = \bar{P}_{cLC} \text{ evaluated with } P_1 = P_{H_2O}, P_2 = P_{Na}, P_{gap} = P_{leak}.$$

Design

$$P_2^D \left(\frac{R_G}{t_G} + \frac{1}{2} \right) \leq S_o^{BC} \quad \text{EQ \#4.2-28}$$

where

$$P_2^D = \text{Max. } \{ \bar{P}_{cLC}^N - P_{Na}, \bar{P}_{cLC}^{Na}, \bar{P}_{cLC}^L - P_{Na} \} \quad \text{EQ \#4.2-29}$$

evaluated with design or operating pressure values.

Condensing the above, one obtains:

$$t_2 \geq \frac{R_1 + t_1 + e_G}{S_{2c}} + e_G \quad \text{EQ \#4.2-30}$$

$$S_{2c} = \text{Min. } \left\{ \frac{S_t^{BC}}{\bar{P}_{cLC}^N - P_{Na}} - \frac{1}{2}, \frac{S_m^{BC}}{\bar{P}_{cLC}^{Na}} - \frac{1}{2}, \frac{1.2 S_m^{BC}}{\bar{P}_{cLC}^L - P_{Na}} - \frac{1}{2}, \frac{S_o^{BC}}{P_2^D} - \frac{1}{2} \right\} \quad \text{EQ \#4.2-31}$$

D. Summary of Rupture Criteria
for the Two Alternate Tube
Designs

Uniform Design

$$\text{Region A: } t_1^U \geq \frac{R_1}{S_A} \quad \text{EQ \# 4.2-4}$$

$$\text{Region B: } \left\{ \begin{array}{l} t_1^U \geq \frac{R_1}{2S_{1B}} \\ t_2^U \geq \frac{R_1 + t_1^U + e_G}{2S_{2B}} + e_G \end{array} \right. \quad \begin{array}{l} \text{EQ \# 4.2-9} \\ \text{EQ \# 4.2-16} \end{array}$$

$$\text{Region C: } \left\{ \begin{array}{l} t_1^U + t_2^U \geq \frac{R_1}{S_{CC}} + e_G \\ t_2^U \geq \frac{R_1 + t_1^U + e_G}{S_{2C}} + e_G \end{array} \right. \quad \begin{array}{l} \text{EQ \# 4.2-23} \\ \text{EQ \# 4.2-30} \end{array}$$

Non-Uniform Design

$$\text{Region A: } t_1^{TS} \geq \frac{R_1}{S_A} \quad \text{EQ \# 4.2-4}$$

$$\text{Region B: } \left\{ \begin{array}{l} t_1^N \geq \frac{R_1}{2S_{1B}} \\ t_2^N \geq \frac{R_1 + t_1^N + e_G}{2S_{2B}} + e_G \end{array} \right. \quad \begin{array}{l} \text{EQ \# 4.2-9} \\ \text{EQ \# 4.2-16} \end{array}$$

$$\text{Region C: } \left\{ \begin{array}{l} t_1^N + t_2^N \geq \frac{R_1}{S_{CC}} + e_G \\ t_2^N \geq \frac{R_1 + t_1^N + e_G}{S_{2C}} + e_G \end{array} \right. \quad \begin{array}{l} \text{EQ \# 4.2-23} \\ \text{EQ \# 4.2-30} \end{array}$$

4.2.1.2.1 RUPTURE CRITERIA (continued)

In both cases, the optimum

design is that which minimizes the combined thickness,

$t_1^U + t_2^U$ or $t_1^N + t_2^N$ below the tubesheets.

E. Thickness Bounds

From the following

criteria for tube sizing of a single walled tube, a lower bound on the

duplex tube combined thickness, t_{LB} , is obtained:

Normal Operating

$$(P_{H_2O} - P_{Na}) \left(\frac{R_1}{t_{LB}} + \frac{1}{2} \right) \leq S_t^{BC}$$

Upset-Sudden Loss of P_{Na}

$$P_{H_2O} \left(\frac{R_1}{t_{LB}} + \frac{1}{2} \right) \leq S_m^{BC}$$

Design

$$P_{H_2O}^D \left(\frac{R_1}{t_{LB}} + \frac{1}{2} \right) \leq S_o^{BC}$$

Lower Bound

$$t_{LB} = \frac{R_1}{S_{cc}}$$

EQ #4.2-32

Two single walled tubes plus allowance for the groove depth is assumed to provide a realistic upper bound on the duplex tube combined thickness.

Upper Bound

$$t_{UB} = 2t_{LB} + e_G$$

EQ #4.2-33

upper and lower bound thicknesses are for purposes of comparison only. The combined thickness for the duplex tube would be expected to be within these bounds.

4.2.1.2.2 BUCKLING CRITERIA

Possibilities for buckling

(elastic, plastic, creep) include:

1. External pressure in regions A, B and C
2. Axial load on inner tube, associated with thermal expansion in region A.

Elastic buckling under external pressure was evaluated and is reported here. Evaluation of thermal expansion buckling of the inner tube in region A was postponed until after the tube contact situation is further studied. Additionally, plastic buckling, or creep buckling effects on geometry changes, were considered beyond the scope of preliminary design. However, it should be noted that thickness reduction and diametrical growth during creep under internal pressure could lower buckling resistance. Possibly some additional thickness would be needed to compensate for creep geometry changes. Also, plastic buckling is related to the tangent modulus of the actual stress-strain curve at the appropriate stress level rather than the elastic modulus. The tangent modulus is always less than the elastic modulus. Thus, if plastic loadings occur, some additional thickness would be required.

The geometry of both tubes is simply a cylinder with a hemispherical end cap. Therefore, formulas for elastic buckling of a cylinder and a sphere are employed. The external

4.2.1.2.2 BUCKLING CRITERIA (continued)

pressure (P_c) for elastic buckling in the cylindrical region of the tube is assumed to be conservatively given by the following equation from reference 4.2-6:

$$P_{cr} = \frac{E}{4(1 - \nu^2)} \left(\frac{t}{R} \right)^3 \quad \text{EQ \#4.2-34}$$

E = Elastic Modulus

R = Inner Radius

ν = Poisson's Ratio

t = Thickness

The ASME Code (Ref. 4.2-4) requires a factor of safety of 3.0 on the critical buckling pressure, P_{cr} . Thus, the tube will not buckle under external pressure, $P_{ext.}$, provided:

$$P_{ext.} \leq \frac{E}{12(1 - \nu^2)} \left(\frac{t}{R} \right)^3 \quad \text{EQ \#4.2-35}$$

$$\text{or, } P_{ext.} \leq (0.0916)E \left(\frac{t}{R} \right)^3 \quad \text{EQ \#4.2-35a}$$

The critical elastic buckling pressure for a sphere is given by reference 4.2-6 as:

$$P_{cr} = \frac{2E}{\sqrt{3}(1 - \nu^2)} \left(\frac{t}{R} \right)^2 \quad \text{EQ \#4.2-36}$$

Again, incorporating a factor of safety of 3.0, the limiting external pressure for buckling of the spherical end cap is:

4.2.1.2.2 BUCKLING CRITERIA (continued)

$$P_{\text{ext}} \leq \frac{2E}{3\sqrt{3}(1-\nu^2)} \left(\frac{t}{R}\right)^2 \quad \text{or,} \quad \text{EQ \#4.2-37}$$

$$P_{\text{ext}} \leq (0.403) E \left(\frac{t}{R}\right)^2. \quad \text{EQ \#4.2-37a}$$

Upon comparison of equations 4.2-35a and -37a, it is apparent that the cylinder will always buckle before a sphere with the same $\frac{t}{R}$. This is illustrated in Figure 4.2-4. Buckling of the spherical end caps of the duplex tube is automatically prevented if buckling of the adjacent cylindrical sections is prevented. The data points shown in Figure 4.4-4 will be discussed in Section 4.2.1.3.2.

Practical considerations of the manufacture of duplex tubes implies that the inner tube would likely have a short unswaged cylindrical portion adjacent to its spherical end cap. This is shown in the Figure 4.2-1.

For reference and alternate designs, buckling possibilities for the inner tube are, (1) in the cylindrical tube between the tubesheets, and (2) in the unswaged cylindrical tube adjacent to the spherical end cap. The only significant difference between (1) and (2) for the inner tube is in the case of the non-uniform tube design. Because of the increased tube thickness in the region between the tube-sheets, condition (2) will always govern for the non-uniform design.

4.2.1.2.2 BUCKLING CRITERIA (continued)

On the other hand, since thickness and load are the same in both regions, conditions (1) and (2) are equivalent for the uniform design.

The outer tube is reinforced by the inner tube except in the region adjacent to its spherical end cap. This region is the only possibility for buckling of the outer tube in either of the uniform or non-uniform designs.

In summary, the elastic buckling criteria for the duplex tube are as follows:

Uniform Design/Inner Tube

$$\frac{P_{He}^D}{E_{max}} \leq \frac{1}{12(1-\nu^2)} \left\{ \frac{t_1^U}{R_1} \right\}^3 \quad \text{EQ \#4.2-38}$$

Uniform Design/Outer Tube

$$\frac{P_{Na}^D}{E_{max}} \leq \frac{1}{12(1-\nu^2)} \left\{ \frac{t_2^U - e_G}{R_1 + t_1^U - e_G} \right\}^3 \quad \text{EQ \#4.2-39}$$

Non-Uniform Design/Inner Tube

$$\frac{P_{He}^D}{E_{max}} \leq \frac{1}{12(1-\nu^2)} \left\{ \frac{t_1^N}{R_1} \right\}^3 \quad \text{EQ \#4.2-40}$$

and,

$$t_1^{TS} \geq t_1^N \quad \text{EQ \#4.2-41}$$

4. 2. 1. 2. 2 BUCKLING CRITERIA (continued)

Non-Uniform Design/Outer Tube

$$\frac{P_{Na}^D}{E_{max}} \leq \frac{1}{12 (1 - \nu)} \left\{ \frac{t_2^N - e_G}{R_1 + t_1^N + e_G} \right\}^3 \quad \text{EQ \#4.2-42}$$

4.2.1.3 PRELIMINARY SIZING CALCULATIONS

4.2.1.3.1 RUPTURE

To determine the thicknesses which satisfy the rupture criteria requires first calculating S_A , S_{1B} , S_{2B} , S_{CC} , and S_{2C} . These are ratios of a code allowable stress values (S_o , S_m , S_t) to the corresponding pressure quantities (P_{H_2O} , $P_{H_2O} - P_{He}$, etc.) as given in Equations 4.2-5, -10, -17, -24, and -31.

The allowable stress values (S_o , S_m , S_t) used for the rupture evaluation are presented in Table 4.2-III. Values for 2 1/4 Cr-1 Mo were obtained from Ref. 4.2-4. Values for HT-9 were derived from data obtained from Ref. 4.2-7 as discussed in Appendix D.

Initially, the reheater was sized for HT-9 material. However, the extremely thin wall thicknesses (0.009 inches) were considered significantly below the minimum thickness which can be economically fabricated. Since HT-9 is also a relatively expensive material, its use for the reheater would result in a double economic penalty. This was considered unjustifiable and the decision was made (Ref. 4.2-8) to adopt 2 1/4 Cr-1 Mo for the reheater.

TABLE 4.2-III
 DUPLEX TUBE RUPTURE SIZING ALLOWABLES (KSI)

UNIT	MATERIAL	REGION A			REGIONS B AND C		
		(1) S_o^A	(2) S_m^A	(3) S_t^A	(1) S_o^{BC}	(2) S_m^{BC}	(3) S_t^{BC}
SUPERHEATER	HT-9	26.7	29.4	23.4	24.9	27.3	19.8
EVAPORATOR	HT-9	26.7	31.1	23.4	26.7	30.1	23.4
REHEATER	2 1/4Cr-1Mo	11.8	16.8	8.8	7.8	15.9	5.8

- (1) EVALUATED AT THE DESIGN TEMPERATURE IN TABLE 4.2-II
 (2) EVALUATED AT THE OPERATING TEMPERATURE IN TABLE 4.2-II
 (3) EVALUATED FOR 200,000 HOURS AND AT THE OPERATING TEMPERATURE IN TABLE 4.2-II

4.2.1.3.1 RUPTURE (continued)

Table 4.2-IV contains values of the ratios S_A , S_{1B} , S_{2B} , S_{2C} , and S_{CC} computed for the component units and the applicable:

1. Temperatures and pressures of Table 4.2-II.
2. Code allowable stress values from Table 4.2-III.
3. Various loading conditions (normal operating, upset, emergency, and design).

Recall that values of S_A , S_{1B} , S_{2B} , S_{2C} , and S_{CC} to be used for thickness calculations are the minimum values calculated for the various loading conditions imposed upon each component (superheater, evaporator, and reheater), and each duplex tube region (A, B, C). The values listed in Table 4.2-IV also demonstrate the relative severity of each loading condition on the tube regions of each component.

Minimum values of S_A , S_{1B} , S_{2B} , S_{2C} , and S_{CC} were extracted from Table 4.2-IV and used in a graphical solution of the rupture criteria for tube thickness. To illustrate the graphical solution, recall the set of rupture criteria summarized in Section 4.2.1.2.1(d) for the reference duplex tube design (uniform thickness).

TABLE 4.2-IV

VALUES OF s_A , s_{1B} , s_{2B} , s_{2C} , AND s_{CC} COMPUTED FROM TEMPERATURES AND PRESSURES OF TABLE 4.2-II
AND ALLOWABLE STRESS VALUES FROM TABLE 2.

	LOADING CONDITION	s_{2C} (31)								
		s_A (5)	s_{1B} (10)	s_{2B} (17)		$\bar{P}_{CLC} = \text{EQ. (19)}$		$\bar{P}_{CLC} = P_{CAP} \text{ (18)}$		s_{CC} (24)
				$P_{LEAK} = P_{He}$	$P_{LEAK} = P_{H_2O}$	$P_{LEAK} = P_{He}$	$P_{LEAK} = P_{H_2O}$	$P_{LEAK} = P_{He}$	$P_{LEAK} = P_{H_2O}$	
SUPERHEATER (HT - 9)	DESIGN	9.77	9.08	70.7	9.75	15.47	15.47	70.7	9.75	9.08
	NORMAL	9.90	8.31	109.5	109.5	13.07	13.07	109.5	109.5	7.65
	UPSET	11.03	10.20	90.5	90.5	17.35	17.35	90.5	90.5	10.22
	EMERGENCY	—	—	181.2	12.93	22.0	22.0	181.2	12.93	—
EVAPORATOR (HT - 9)	DESIGN	9.39	9.39	75.79	9.95	15.98	15.98	75.79	9.95	9.39
	NORMAL	9.45	9.46	113.6	113.6	14.8	14.8	113.6	113.6	8.66
	UPSET	11.24	10.86	99.83	99.8	18.42	18.42	99.83	99.83	10.86
	EMERGENCY	—	—	175.7	13.64	23.1	23.1	175.7	13.64	—
REHEATER (2 1/4Cr-1Mo)	DESIGN	19.17	12.50	21.79	17.43	21.2	21.2	21.79	17.43	12.50
	NORMAL	34.01	22.24	31.72	31.72	21.7	21.7	31.72	31.72	12.83
	UPSET	29.77	28.15	52.50	52.50	47.3	47.3	52.50	52.50	28.15
	EMERGENCY	—	—	105.5	43.36	72.7	72.7	105.5	43.36	—

() FORMULA NUMBER

4. 2. 1. 3. 1 RUPTURE (continued)

$$t_1^U \geq \frac{R_1}{S_A} \quad \text{EQ \#4. 2-4}$$

$$t_1^U \geq \frac{R_1}{2S_{1B}} \quad \text{EQ \#4. 2-9}$$

$$t_2^U \geq \frac{R_1 + t_1^U + e_G}{2S_{2B}} + e_G \quad \text{EQ \#4. 2-16}$$

$$t_1^U + t_2^U \geq \frac{R_1}{S_{cc}} + e_G \quad \text{EQ \#4. 2-23}$$

$$t_2^U \geq \frac{R_1 + t_1^U + e_G}{S_{2C}} + e_G \quad \text{EQ \#4. 2-30}$$

Recall also, the equations for the upper and lower bounds on combined thickness of Section 4. 2. 1. 2. 1(e): The optimum thickness(es) should lie between these bounds.

$$t_{LB} = \frac{R_1}{S_{cc}} \quad \text{EQ \#4. 2-32}$$

$$t_{UB} = 2t_{LB} + e_G \quad \text{EQ \#4. 2-33}$$

4.2.1.3.1 RUPTURE (continued)

The graphical solution for the superheater with design conditions included, Figure 4.2-5, is discussed as follows:

First, minimum values of S_A , S_{1B} , S_{2B} , S_{2C} , and S_{CC} were taken from Table 4.2-IV from their respective superheater groupings of design, normal, upset, and emergency loading conditions. (For the case of design conditions excluded, only the latter three would be considered.) In this example, $\bar{P}_{c_{LC}}$ as given by Equation 4.2-19 and $P_{leak} = P_{H_2O}$ were also chosen. These minimum values of S_A , S_{1B} , S_{2B} , S_{2C} , and S_{CC} together with $R_1 = 0.558$ (Table 4.2-II) and $e_G = 0.020$ in. were substituted into rupture equations 4.2-4, -9, -16, -23, and -30. Each equation plots as a straight line as shown in Figure 4.2-5. The equations involving t_2 contain an allowance for the depth of the helium leak detection grooves. Any point (t_1, t_2) in the hatched region would satisfy all the rupture criteria simultaneously. The optimum solution is given by the point (t_1^U, t_2^U) which has the minimum combined thickness of all possible solutions. The point (t_1^U, t_2^U) is easily seen to be the one with minimum combined thickness by noting that any line of slope $\frac{dt_2}{dt_1} = -1$ is a line of constant combined thickness. Similarly, the

optimum solution for the non-uniform design is represented by (t_1^N, t_2^N) which is determined by ignoring equation (line) (Reference 4.2-4). Note that both solutions lie between the upper and lower bounds. Similarly, the various

4.2.1.3.1 RUPTURE (continued)

combinations of component, loading conditions, and leak assumptions are shown graphically in Figures 4.2-6 through 4.2-11. The optimum thicknesses obtained from these figures are summarized in Table 4.2-V. Also listed in Table 4.2-V is the combined thickness ($t_1 + t_2$) and a normalized combined thickness defined as,

$$\bar{t}_c = \frac{t_1 + t_2 - t_{LB}}{t_{UB} - t_{LB}}$$

or incorporating Equation 4.2-33 for the upper bound,

$$\bar{t}_c = \frac{t_1 + t_2 - t_{LB}}{t_{LB} + e_G} \quad \text{EQ \#4.2-43}$$

This normalized combined thickness provides a measure for the additional tube thickness of a duplex tube design compared to a single wall design. The normalized combined thicknesses for the three steam generator units are plotted versus the assumed leak and contact pressure parameters in Figures 4.2-12 through 4.2-14. It should be noted that the combined thickness, $t_1 + t_2$, must always include the groove depth, e_G . Therefore, the minimum value of t_c attainable, as shown in Figures 4.2-12 through 4.2-14, is greater than 0%. Results presented in these figures lead to the following observations and conclusions:

TABLE 4.2-V OPTIMUM THICKNESS FOR ALL CASES CONSIDERED

		REFERENCE DESIGN								ALTERNATE DESIGN				
		UNIFORM THICKNESS INNER TUBE								NON-UNIFORM THICKNESS INNER TUBE				
		$\bar{P}_{CLC}$	P_{LEAK}	t_1 (IN.)	t_2^* (IN.)	$t_1 + t_2^*$ (IN.)	$\bar{t}_C^*$ (%)		t_{1A} (IN.)	t_1 (IN.)	t_2^* (IN.)	$t_1 + t_2^*$ (IN.)	$\bar{t}_C^*$ (%)	
DESIGN INCLUDED	SUPER-HEATER	EQ. (19)**	= P_{H_2O} ○	.057	.068	.125	56	○	.057	.034	.067	.101	30	
			= P_{He} Δ	.057	.068	.125	56	Δ	.057	.034	.067	.101	30	
		= P_{GAP}	= P_{H_2O} □	.057	.085	.142	74	□	.057	.034	.083	.117	47	
			= P_{He} ▽	.057	.036	.093	22	▽	.057	.034	.059	.093	22	
	EVAP-ORATOR	EQ. (19)**	= P_{H_2O} ○	.062	.064	.126	68	○	.062	.031	.062	.093	30	
			= P_{He} Δ	.062	.064	.126	68	Δ	.062	.031	.062	.093	30	
		= P_{GAP}	= P_{H_2O} □	.062	.036	.148	93	□	.062	.031	.093	.114	54	
			= P_{He} ▽	.062	.030	.092	29	▽	.062	.031	.055	.087	23	
	REHEATER	EQ. (19)*	= P_{H_2O} ○	.047	.064	.111	42	○	.047	.036	.064	.100	30	
			= P_{He} Δ	.047	.064	.111	42	Δ	.047	.036	.064	.100	30	
		= P_{GAP}	= P_{H_2O} □	.047	.076	.123	55	□	.047	.036	.075	.111	42	
			= P_{He} ▽	.047	.064	.111	42	▽	.047	.036	.064	.100	30	
DESIGN EXCLUDED	SUPER-HEATER	EQ. (19)**	= P_{H_2O} ○	.056	.068	.124	55	○	.056	.034	.067	.101	30	
			= P_{He} Δ	.056	.068	.124	55	Δ	.056	.034	.067	.101	30	
		= P_{GAP}	= P_{H_2O} □	.056	.070	.126	57	□	.056	.034	.068	.102	31	
			= P_{He} ▽	.056	.033	.094	23	▽	.056	.034	.059	.093	22	
	EVAP-ORATOR	EQ. (19)**	= P_{H_2O} ○	.061	.064	.125	67	○	.061	.030	.062	.092	29	
			= P_{He} Δ	.061	.064	.125	67	Δ	.061	.030	.062	.092	29	
		= P_{GAP}	= P_{H_2O} □	.061	.068	.129	71	□	.061	.030	.066	.096	33	
			= P_{He} ▽	.061	.027	.088	24	▽	.061	.030	.057	.087	23	
	REHEATER	EQ. (19)**	= P_{H_2O} ○	.030	.064	.094	27	○	.030	.020	.070	.090	22	
			= P_{He} Δ	.030	.064	.094	27	Δ	.030	.020	.070	.090	22	
		= P_{GAP}	= P_{H_2O} □	.030	.050	.080	22	□	.030	.020	.070	.090	22	
			= P_{He} ▽	.030	.050	.090	22	▽	.030	.020	.070	.090	22	

* INCLUDES 0.020 IN. DEPTH OF HELIUM LEAK DETECTING GROOVE ** ASSUMED $\frac{t_1}{t_1 + t_2} = 0.6$

FIGURE 4.2-12

FIGURE 4.2-13

FIGURE 4.2-14

4. 2. 1. 3. 1 RUPTURE (continued)

1. Choice of Uniform Versus Non-Uniform Design

A penalty of up to 40% in additional normalized combined thickness is associated with a choice of the reference (uniform thickness) design. This indicates that, in general, a significant reduction in combined thickness can be realized by adopting the non-uniform thickness design. However, the final choice must be based on an evaluation of other factors (e.g., manufacturing) not considered in this structural evaluation.

Preliminary thickness recommendations satisfying rupture criteria only are provided for both designs in Table 4.2-VI.

2. Design Conditions - Include vs. Exclude

Figures 4.2-12 through 4.2-14 indicate that in most cases only a small additional penalty is incurred by including design conditions. Design conditions are conservative and can be overly conservative (maximum pressure is combined with maximum temperature regardless of whether or not this occurs in reality). The determination of duplex tube thicknesses by the procedures developed in this report includes many conservative assumptions of the various parameters involved. If the ASME Code recognized these conservatisms and accepted these tube sizing

*
TABLE 4.2-VI
RECOMMENDED MINIMUM THICKNESSES (INCHES)
SATISFYING RUPTURE CRITERIA ONLY

	REFERENCE DESIGN (UNIFORM THICKNESS)			ALTERNATE DESIGN (NON-UNIFORM THICKNESS)			
UNIT (MATERIAL)	U_{t_1}, t_1^{TS}	U_{t_2}	$U_{t_1 + t_2}$	t_1^{TS}	t_1^N	t_2^N	$t_1^N + t_2^N$
SUPERHEATER (HT -9)	0.056	0.070	0.126	0.056	0.034	0.068	0.102
EVAPORATOR (HT -9)	0.061	0.068	0.129	0.061	0.030	0.066	0.096
REHEATER (2 1/4 Cr - 1 Mo)	0.030	0.060	0.090	0.030	0.020	0.070	0.090

*
THESE RECOMMENDED THICKNESSES ARE BASED UPON:

(a) $\bar{P}_{CLC}$ = EQUATION (18)

(b) $P_{LEAK} = P_{H_2O}$

(c) DESIGN CONDITIONS EXCLUDED

4.2.1.3.1 RUPTURE (continued)

procedures, perhaps the question of omitting design conditions would also be considered. Such action could be taken later. In view of the preliminary nature of this sizing investigation, it was decided (Ref. 4.2-8) to exclude design conditions on final recommended thicknesses.

3. $\bar{P}_{c_{LC}}$ Formula and P_{leak} Bounds

Figures 4.2-12 through 4.2-14 show that the combined thickness is unaffected by the magnitude of the leak pressure when $\bar{P}_{c_{LC}}$ is given by Equation 4.2-19. This reflects the fact that Equation 4.2-19 does not contain p_{gap} .

Since Equation 4.2-19 involves t_1 and t_2 , the solution for the thicknesses is actually an iterative process when this formula is used for $\bar{P}_{c_{LC}}$. All solutions were obtained assuming $\frac{t_2}{t_1 + t_2} = 0.6$. The results in Table 4.2-V indicate that this value is representative and reasonable for preliminary sizing purposes. However, it would be expected that the iterative procedure would be used for final sizing to insure that the minimum combined thickness is determined.

In view of the preliminary nature of the determination of the $\bar{P}_{c_{LC}}$ formula, and the uncertainties associated with the determination of the actual leak pressure, it was judged that the best course at this stage was to adopt Equation 4.2-18 for $\bar{P}_{c_{LC}}$, and $P_{leak} = P_{H_2O}$.

4. 2. 1. 3. 1 RUPTURE (continued)

4. Duplex Tube Thicknesses

In view of the above observations, preliminary recommended minimum thicknesses satisfying the rupture criteria only were chosen from Table 4.2-V and are listed in Table 4.2-VI. They are subject to revision in the subsequent buckling evaluation. They were chosen for the set of parameters including:

- a. Design excluded
- b. $\bar{P}_{c_{LC}} = P_{gap}$
- c. $P_{leak} = P_{H_2O}$.

A review of the graphical solutions of Figures 4.2-6 through 4.2-11 shows that Equations 4.2-4, -9, -23 and -30 always controlled the optimum thicknesses. Assuming this is always true, the optimum thicknesses can be calculated directly from the following formulas:

Uniform Design

$$t_1^U = \frac{R_1}{S_A}$$

EQ #4.2-4

$$t_2^U = \text{Max} \left\{ \left[R_1 \left(\frac{1}{S_{CC}} - \frac{1}{S_A} \right) + e_G \right], \left[\frac{R_1 (1 + 1/S_A) + e_G}{S_{2C}} + e_G \right] \right\}$$

EQ #4.2-44

4.2.1.3.1 RUPTURE (continued)

Non-Uniform Design

$$t_1^{TS} = \frac{R_1}{S_A} \quad \text{EQ \#4.2-4}$$

$$t_1^N = \frac{R_1}{2S_{1B}} \quad \text{EQ \#4.2-9}$$

$$t_2^N = \text{Max} \left\{ \left[R_1 \left(\frac{1}{S_{CC}} - \frac{1}{2S_{1B}} \right) + e_G \right], \left[\frac{R_1 \left(1 + \frac{1}{2S_{1B}} \right) + e_G}{S_{2C}} + e_G \right] \right\}$$

EQ \#4.2-45

These simple formulas may be useful in the future as a quick duplex tube sizing procedure.

4.2.1.3.2 BUCKLING

Material, pressure, and geometrical

data for the buckling evaluation are given in Table 4.2-VII. Young's modulus (E) for HT-9 and 2 1/4 Cr-1 Mo (Ref. 4.2-4, -7) was evaluated at the design temperature for regions B and C (Table 4.2-II). The external pressures acting on the inner and outer tubes are the design pressure for helium and sodium, respectively (Table 4.2-II). Minimum thicknesses from rupture criteria (Table 4.2-VI) are also included for convenient reference.

TABLE 4.2-VII

DATA USED FOR BUCKLING CALCULATIONS

UNIT	MATERIAL	E @ DESIGN TEMPERATURE FOR REGIONS B & C (PSI)	P _{EXT} (PSI)		R ₁ (INCHES)	MINIMUM THICKNESS SATISFYING RUPTURE CRITERIA				
						REFERENCE DESIGN UNIFORM THICKNESS		ALTERNATE DESIGN NON-UNIFORM THICKNESS		
			INNER TUBE	OUTER TUBE		t ₁ ^U (IN.)	t ₂ ^U * (IN.)	t ₁ ^{TS} (IN.)	t ₁ ^N (IN.)	t ₂ ^N * (IN.)
SUPER- HEATER	HT - 9	19.0x10 ⁶	350	250	0.558	0.056	0.070	0.056	0.034	0.068
EVAP- ORATOR	HT - 9	21.2x10 ⁶	350	250	0.578	0.061	0.068	0.061	0.030	0.066
REHEATER	2 1/4Cr-1Mo	23.0x10 ⁶	350	250	0.900	0.030	0.060	0.030	0.020	0.070

* INCLUDES GROOVE DEPTH OF 0.020".

4.2.1.3.2 BUCKLING (continued)

The buckling calculation points corresponding to reference and alternate design tubes of each unit are shown in Figure 4.2-4. The data points were calculated from information in Table 4.2-VII. These results indicate that both the reference and alternate design inner tube of the reheater and the alternate design inner tube of the evaporator would buckle. The remaining tubes would be safe from buckling. Minimum tube thicknesses for the three inner tubes susceptible to buckling were recalculated from the buckling equation 4.2-35. However, since the rupture criteria provide a relation between the thickness of both inner and outer tubes, a further recalculation was necessary to establish a new minimum combined thickness (for rupture and buckling). This ascertained the shift in position of points R_2^U , R_2^N , and E_2^N originally plotted in Figure 4.2-4. The new data points, safe from buckling, are shown in Figure 4.2-15. Table 4.2-VIII summarizes the minimum thicknesses required to satisfy the buckling.

The evaporator inner tube thickness below the tubesheets increased by 0.003 in. with no practical increase in outer tube thickness. Also, the inner tube thickness between the tube-sheets remained unchanged because, in the alternate design, it was sufficient

4. 2. 1. 3. 2 BUCKLING (continued)

to prevent buckling (same point as E_1^U in Figure 4.2-4).

The buckling resistant thickness (0.049 in.) of the reheater inner tube is larger than the thickness required (0.030 in.) between the tubesheets by rupture criteria. Thus, there is no advantage for a non-uniform thickness inner tube for the reheater.

Recalculation of a new minimum thickness to optimize rupture and buckling resistance reduced the outer tube thickness by 0.008 in.

Recalculated tube thicknesses (reference and alternate design) for the three steam generator units are given in Table 4.2-IX. These final recommended tube thicknesses satisfy both rupture and buckling criteria. They also include a sodium side (t_2) and steam side (t_1 , t_1^{TS}) corrosion allowance of 0.008 in. and 0.003 in., respectively, (Ref. 4.2-9). An additional 5% is allowed for fabrication tolerances (Ref. 4.2-10).

TABLE 4.2-VIII

MINIMUM THICKNESS RECALCULATED TO PREVENT BUCKLING

	UNIFORM		NON-UNIFORM DESIGN		
	t_1^U	t_2^U	t_1^{TS}	t_1^N	t_2^N
EVAPORATOR	_____	_____	0.061	0.033	0.066
REHEATER	0.049	0.052	$\frac{*}{\text{_____}}$	$\frac{*}{\text{_____}}$	$\frac{*}{\text{_____}}$

* BUCKLING RESULTS ELIMINATED NON-UNIFORM DESIGN
ALTERNATE FOR REHEATER.

TABLE 4.2-IX
[†]
 RECOMMENDED AVERAGE THICKNESS FOR PRELIMINARY DESIGN
 SATISFYING BOTH RUPTURE AND BUCKLING CRITERIA

UNIT	MATERIAL	R ₁ (INCHES)	REFERENCE DESIGN (UNIFORM THICKNESS INNER TUBE)		ALTERNATE DESIGN (NON-UNIFORM THICKNESS INNER TUBE)		
			^U t ₁ (INCHES)	^U t ₂ (INCHES)	^{TS} t ₁ (INCHES)	^N t ₁ (INCHES)	^N t ₂ (INCHES)
SUPERHEATER	HT - 9	.558	0.062	0.082	0.062	0.039	0.080
EVAPORATOR	HT - 9	.578	0.067	0.080	0.067	0.038 *	0.078 *
REHEATER	2 1/4Cr-1Mo	.900	0.055 *	0.063 *	— *	— *	— *

* AFFECTED BY BUCKLING CRITERIA

30 YEAR CORROSION ALLOWANCE: STEAM SIDE (^Ut₁, ^{TS}t₁, ^Nt₁) = 0.003"

SODIUM SIDE (^Ut₂, ^Nt₂) = 0.008"

5% OF MINIMUM THICKNESS ADDED FOR DIMENSIONAL TOLERANCE.

[†] THESE RECOMMENDED THICKNESSES ARE BASED ON:

- (a) $\bar{P}_{CIC}$ = EQUATION (18)
- (b) P_{LEAK} = P_{H_2O}
- (c) DESIGN CONDITIONS EXCLUDED

4.2.1.4 CONCLUSIONS

In general, the analysis shows that the duplex tube concept requires a combined thickness approximately 50 to 75% greater than that required for a single walled tube. Table 4.2-IX contains thickness recommendations for two alternate designs of the duplex tube. The recommendations include dimensional allowances for sodium and steam corrosion and fabrication tolerances. The two alternate designs are (1) a uniform thickness inner tube, and (2) a decrease in thickness of the inner tube over the length of the sodium chamber. A choice of one design over the other would have to be based upon considerations not covered in this structural evaluation.

The thickness recommendations of Table 4.2-IX include a conservative assumption concerning a steam leak. That is, in the event of a leak on the steam side, full steam pressure is effective in the 0.020 in. leak detection grooves and the annulus of the duplex tube. Further analysis of steam pressure drop through postulated leaks would most likely show much lower pressures in this region. If so, tubing thicknesses could be reduced somewhat.

The sizing formulas developed for the prevention of rupture of the duplex tube are:

4.2.1.4 CONCLUSIONS (continued)

Inner tube:

Uniform thickness design, $t_1 = \frac{R_1}{S_A} \quad (t_1^{TS} = t_1)$

Non-uniform thickness design, $t_1 = \frac{R_1}{2S_{1B}} ; \quad t_1^{TS} = \frac{R_1}{S_A}$

Outer tube (0.020 in. grooves on inside):

$$t_2 = \max. \left\{ \left[\frac{R_1 + t_1 + e_G}{S_{2C}} + e_G \right], \left[\frac{R_1}{S_{CC}} + e_G - t_1 \right] \right\}$$

t_1 = thickness of inner tube below lower tubesheet

t_1^{TS} = thickness of inner tube between tubesheets

t_2 = thickness of outer tube

R_1 = inside radius of inner tube

e_G = depth of groove

$S_A, S_{1B}, S_{2C}, S_{CC}$ = stress to pressure ratios derived and discussed in the text of this report

Pertinent to the development of the duplex tube sizing procedure was the concept of an equivalent load-controlled contact pressure, $\bar{P}_{c, LC}$. This contact pressure acts on the interface between the tubes. It is related to the various fluid pressures and tube thicknesses. An approximate formula was developed and used in the calculations to obtain the thicknesses given in Table 4.2-IX. Further investigation and refinement in the formula might lead to a reduction in the thicknesses reported herein.

4.2.2 TUBE SLIDING AND LOSS OF CONTACT STUDY

Six modes of failure were identified for the duplex tube in the preliminary structural sizing reported in Section 4.2.1. The preliminary sizing was based on the prevention of two of the failure modes - rupture and buckling. In this Section, two more failure modes are investigated: relative sliding due to thermal expansion and pressure loading, and the possibility of loss of contact under operating pressures and temperatures.

Thermal and mechanical loadings were evaluated for their effect on relative sliding of the duplex tubes. Both local sliding (fretting) and gross sliding (of entire tube) were investigated. The formulation developed to evaluate sliding entailed several uncertainties as discussed in this report. Even though the formulation was qualitative in nature, the results for all loadings but one were sufficiently quantitative to rule out potential for either gross or local sliding.

The exception is differential thermal expansion in the tube-sheet region. This loading has the greatest potential to cause local sliding below the lower tubesheet where the inner tube first makes contact with the outer tube. With values of 0.3 for coefficient of friction and 1000 psi contact pressure, a frictional resistance of 300 psi must be overcome to cause local sliding. A 10.6°F temperature differential in the tubesheet region could do this. Thus, local sliding might be plausible during a thermal transient. Further analysis of this local sliding situation is

4.2.2 TUBE SLIDING AND LOSS OF CONTACT STUDY (Continued)

warranted. The nature of fretting involves small displacements and many cycles of application, on the order of a million or more. It would appear that cyclic thermal differentials in the tubesheet region would be much less than this over the operational life of the system.

The possibility of loss of contact during normal operating conditions was also investigated. The evaluation was limited to a comparison of the elastic displacements associated with the normal operating pressures and temperature distribution. The scope of this task did not include an analysis of the swaging process. Therefore, the associated plastic strain and contact pressure were treated qualitatively. Even so, it was determined that the pressure loadings would be sufficient to compensate for the thermal displacements which tend to separate the tubes in all cases except for the reheater. A swaging displacement of 0.0001 in. on the inner tube after initial contact would suffice to maintain contact for the reheater. Displacements much greater than this are expected in practice. Therefore, except for the reheater, these results indicated that total reliance on the plastic swaging strains to maintain contact is generally unnecessary.

4.2.2 TUBE SLIDING AND LOSS OF CONTACT STUDY (Continued)

A cursory evaluation of the effects of creep indicated that after sufficient creep deformation has occurred, only pressure loadings remain significant. This is beneficial because the squeezing effect of the pressure loading helps to maintain contact, even for the reheater.

4.2.2.1 RELATIVE SLIDING ANALYSIS

The geometry of the duplex tube concept was shown and discussed in Section 4.2.1. A view of the duplex tube below the tubesheets is shown in Figure 4.2-16. Also shown are thermal and mechanical loads considered significant to the sliding evaluation.

These loadings are:

1. Uniformly distributed pressures (p_1 , p_2 , and p_{gap}) and the axial force (F_p) required for axial equilibrium with these pressures.
2. A self-equilibrating axial force (F_T) acting on the tubesheet end of the tubes. F_T is due to the non-uniformity of temperature in the tubesheets, shell and that part of the inner tube in the helium leak detection space.
3. An axisymmetric temperature distribution, $T(x, r)$, varying both along the length and across the thickness of both tubes.

Figure 4.2-16
 SWAGED PORTION OF DUPLEX TUBE INDICATING LOADS
 CONSIDERED FOR SLIDING EVALUATION

4.2.2.1 RELATIVE SLIDING ANALYSIS

Two levels of sliding were distinguished:

1. Local Sliding - Sliding at a cross section (X).
2. Gross Sliding - Sliding at every cross section along the tube.

Both types of sliding were considered detrimental since they could result in material wear leading to blockage of the leak detection grooves. In the extreme, substantial thinning of the tube walls could also result.

It was assumed that local sliding is prevented provided,

$$\bar{\tau}_c (x) \leq f_s \bar{p}_c (x) \quad \text{EQ \#4.2-46(a)}$$

where

$\bar{\tau}_c (x)$ = Axial shear stress component averaged over the circumference of the contact area between the inner and outer tubes.

$\bar{p}_c (X)$ = Contact pressure equal to radial (normal) stress component averaged over the circumference of the contact area between the inner and outer tubes.

f_s = Coefficient of static friction.

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

It was further assumed that gross sliding is prevented provided,

$$\bar{\tau}_c \leq f_s \bar{p}_c \quad \text{EQ. 4.2-46(b)}$$

where

$\bar{\tau}_c$ and $\bar{p}_c$ represent $\tau_c(x)$ and $p_c(x)$

averaged over the tube length, respectively.

Reference 4.2-14 gives a range of 0.3 to 0.5 for f_s for general clean surfaces in contact. More specific data provided in Reference 4.2-15 indicates that a value of 0.4 is appropriate. The value of 0.3 was used for the lower bound calculations.

It was not within the workscope of this task to determine $\bar{p}_c(X)$ quantitatively. The magnitude of $\bar{p}_c(X)$ at any cross section was estimated in Reference 4.2-17 to range from 1000 to 3000 psi. Therefore, a conservative value of 1000 psi was used in the calculations. This estimate was verified by the work reported in Section 4.2.3.

The major part of this task was the determination of the shear stress produced by the various loads which have the potential to cause sliding. The total shear stress for both local and gross sliding was assumed to be composed of three parts:

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

$$\bar{\tau}_c(x) = \bar{\tau}_p(x) + \bar{\tau}_{p'}(x) + \bar{\tau}_{q'}(x) \quad \text{EQ #4.2-47a}$$

$$\tau_c = \tau_p + \tau_{p'} + \tau_{q'} \quad \text{EQ #4.2-47b}$$

$\bar{\tau}_p(\tau_p)$, $\bar{\tau}_{p'}(\tau_{p'})$ and $\bar{\tau}_{q'}(\tau_{q'})$ are due to the forementioned thermal and mechanical loads (1), (2) and (3) respectively. Each contribution to the total shear stress was determined separately in the following sections.

$$\text{a. } \bar{\tau}_p(x)$$

The pressure related loadings acting on the duplex tube and each tube separately are shown in Figure 4.2-17. Note that $\bar{p}_c(x)$ and $\bar{\tau}_p(x)$ are both considered to vary non-uniformly along the tubes. Axial equilibrium applied to the end of each tube beyond the swaged region gives,

$$\sigma_{1L} = \frac{p_1(R-t_1)^2 - p_{GAP} R^2}{t_1(2R-t_1)} \quad \text{EQ #4.2-48a}$$

and

$$\sigma_{2L} = \frac{p_{GAP} R^2 - p_2(R+t_2)^2}{t_2(2R+t_2)} \quad \text{EQ #4.2-48b}$$

applying axial equilibrium to various sections in Figure 4.2-17(a) gives,

Figure 4.2-17
PRESSURE RELATED LOADINGS ON DUPLEX TUBE

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

$$F_p = \sigma_{1L} A_1 + \sigma_{2L} A_2 \quad (x=L) \quad \text{EQ \#4.2-49a}$$

$$= \sigma_1(x) A_1 + \sigma_2(x) A_2 \quad (x=x) \quad \text{EQ \#4.2-49b}$$

$$= \sigma_{10} A_1 + \sigma_{20} A_2 \quad (x=0) \quad \text{EQ \#4.2-49c}$$

Here $\sigma_1(x)$, $\sigma_2(x)$ are the average axial stress in each tube at Section X and

$$A_1 = \pi[R^2 - (R-t_1)^2] \quad \text{EQ \#4.2-50a}$$

$$A_2 = \pi[(R+t_2)^2 - R^2] \quad \text{EQ \#4.2-50b}$$

The model adopted for determining the shear stress is shown in Figure 4.2-18. While the shear stress distribution, $\bar{\tau}_p(x)$ was considered to be non-uniform along the length of the tubes, for convenience it was assumed to be linear with a maximum value of $\bar{\tau}_{p_{\max}}$.

It is conservatively assumed that the total axial load, F_p , is transmitted across the contact surface through the shear stress, $\bar{\tau}_p(x)$. Therefore, by equilibrium,

FIGURE 4.2-18

SHEAR STRESS MODEL FOR PRESSURE LOADING

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

$$\bar{\tau}_{p_{\max}} = \frac{l' p}{\pi R L}$$

or

$$\bar{\tau}_{p_{\max}} = \frac{p_1(R-t_1)^2 - p_2(R+t_2)^2}{RL} \quad \text{EQ \#4.2-51}$$

Since $\bar{\tau}_p(x)$ has been assumed to be linear,

$$\bar{\tau}_p = \frac{\bar{\tau}_{p_{\max}}}{2} \quad \text{EQ \#4.2-52}$$

b. $\bar{\tau}_F(x)$

The loading, F_T , producing this shear stress is concentrated at the tubesheet end of the duplex tube as shown in Figure 4.2-19(a). Because of the nature of this loading, i.e. concentrated at one end, the shear stress is assumed to attenuate rapidly with distance from the tubesheet end. If no sliding occurs, this loading is similar to an axisymmetric moment applied to the end of a cylinder of thickness $t_1 + t_2$. Therefore, the shear stress distribution is assumed to be

$$\bar{\tau}_{F_{\max}}(x) = \tau_{F_{\max}} e^{-\lambda x} \quad \text{EQ \#4.2-53}$$

where

$$\lambda = 4 \sqrt{\frac{3(1-\nu^2)}{R^2(t_1+t_2)^2}} = \frac{1.286}{\sqrt{R(t_1+t_2)}} \quad \text{EQ \#4.2-54}$$

ν = Poisson's Ratio

FIGURE 4.2-19

TUBESHEET THERMAL EXPANSION LOADING (F_T) AND SHEAR STRESS MODEL

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

λ is the attenuation factor for an edge moment from Reference 4.2-18.

Applying axial equilibrium to either tube,

$$\int_0^L \bar{\tau}_F(x) 2\pi R dx = F_T \quad \text{EQ \#4.2-55}$$

from which,

$$\bar{\tau}_{F_{\max}} = \frac{\lambda F_T}{2\pi R} \quad \text{EQ \#4.2-56}$$

The associated uniform (along the tube)

shear stress is,

$$\bar{\tau}_F = \frac{1}{L} \int_0^L \bar{\tau}_F(x) dx \quad \text{EQ \#4.2-57}$$

or substituting Equation 4.2-53 into EQ 4.2-57,

$$\bar{\tau}_F = \frac{\bar{\tau}_{F_{\max}}}{L \lambda} \quad \text{EQ \#4.2-58}$$

F_T is determined from the gross temperature distribution characteristics in the tubesheet region. The simplified structural model is shown in Figure 4.2-20. Assuming sliding does not occur, differential thermal expansion in the tubesheet region will generate an internal force, F_T , acting on each tube as shown in Figure 4.2-20a. The structure is separated into the tubesheet-shell assembly (Figure 4.2-20b) and the inner tube (Figure 4.2-20c).

FIGURE 4.2-20
TEMPERATURE DISTRIBUTION AND STRUCTURAL MODEL USED
TO DETERMINE F_T

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

The internal force is determined by forcing compatibility of displacements, i.e. , $U_T = U_{TS}$. Assuming rigid tubesheets,

$$\frac{U_S}{L_S} = \frac{1}{E} \frac{N F_T}{A_S} + \alpha (T_S - T_O) \quad \text{Eq. \#4.2-59}$$

where

N = Total number of tubes

$A_S = 2\pi R_S t_S$

E = Elastic Modulus

α = Coefficient of thermal expansion

T_O = Initial stress free uniform temperature

Since the tubesheet is rigid, it contributes only a thermal displacement ($= \alpha(T_A - T_O) t_A$) so that,

$$U_{TS} = \frac{NL_S}{EA_S} F_T + \alpha (T_S - T_O) L_S + (T_A - T_O) t_A \quad \text{Eq. \#4.2-60}$$

The displacement of the inner tube is given by,

$$U_T = \left[- \frac{F_T}{EA} + \alpha (T_T - T_O) \right] (L_S + t_A) \quad \text{Eq. \#4.2-61}$$

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

F_T is determined by equating the displacements, $U_T = U_{TS}$,

$$F_T = \frac{E \alpha [L_s (T_T - T_s) + t_A (T_T - T_A)]}{\frac{NL_s}{A_s} + \frac{L_s + t_A}{A_1}}$$

or

$$F_T = \frac{E \alpha A_1 [(T_T - T_s) + \frac{t_A}{L_s} (T_T - T_A)]}{\frac{NA_1}{A_s} + 1 + \frac{t_A}{L_s}}$$

EQ #4.2-62

c. $\bar{\tau}_T (X)$

The general axisymmetric temperature distribution within a cylinder of mean radius, R_m , and thickness, t , is shown in Figure 4.2-21 and can be represented as

$$T(X, r) = \bar{T}(X) + \frac{\Delta T^L(X)}{t} [r - R_m] + \Delta T^N(X, r) \quad \text{EQ \#4.2-63}$$

where

$\bar{T}(X)$ = Temperature averaged through the thickness at section X.

$\Delta T^L(X)$ = Equivalent linear temperature difference between inside and outside surfaces at section X.

$\Delta T^N(X, r)$ = Difference between actual temperature distribution and equivalent linear temperature distribution.

FIGURE 4.2-21

CHARACTERISTICS OF A GENERAL TEMPERATURE DISTRIBUTION

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

For the purpose of investigating sliding, only $\bar{T}(X)$ is considered to vary with X . Thus, the temperature distribution in each tube at any time is assumed to be given by,

$$T_1(X, r) = \bar{T}_1(X) + \tilde{T}_1(r) \quad \text{EQ \#4.2-64a}$$

$$T_2(X, r) = \bar{T}_2(X) + \tilde{T}_2(r) \quad \text{EQ \#4.2-64b}$$

where $\tilde{T}_1(r)$ and $\tilde{T}_2(r)$ denote the non-linear radial temperature variation.

The classical elastic stress solution for an arbitrary radial temperature distribution (p.408-409, Ref. 4.2-19) which is uniform along a cylinder results in no axial shear stress. Therefore, $\bar{T}_1(X)$ and $\bar{T}_2(X)$ are assumed to be the only significant source of sliding shear stress. The shear stress $\bar{\tau}_T(X)$ was determined using the model shown in Figure 4.2-22

Axial equilibrium for both tubes combined requires,

$$\sigma_1(X) A_1 + \sigma_2(X) A_2 = 0 \quad \text{EQ \#4.2-65}$$

The total axial strain in each tube is given by,

FIGURE 4.2-22

MODEL AND GENERAL TEMPERATURE DISTRIBUTION FOR
DETERMINATION OF $\bar{\tau}_T(x)$

(TEMPERATURE PROFILE SHOWN IS SCHEMATIC, NOT
NECESSARILY ACTUAL)

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

$$\epsilon_1(X) = \frac{dU_1(X)}{dX} - \frac{\sigma_1(X)}{E} = \alpha(\bar{T}_1(X) - T_0) \quad \text{EQ \#4.2-66a}$$

$$\epsilon_2(X) = \frac{dU_2(X)}{dX} = \frac{\sigma_2(X)}{E} + \alpha(\bar{T}_2(X) - T_0) \quad \text{EQ \#4.2-66b}$$

Assuming contact is maintained. $U_1(X) = U_2(X)$ and $\epsilon_1(X) = \epsilon_2(X)$.

Equating EQ #4.2-66a and EQ #4.2-66b and using

EQ #4.2-65 results in,

$$\sigma_1(X) = -\frac{A_2}{A_1} \sigma_2(X) = \frac{E\alpha[\bar{T}_2(X) - \bar{T}_1(X)]}{(1 + \frac{A_1}{A_2})} \quad \text{EQ \#4.2-67}$$

Applying equilibrium to an axial element of each tube

(See Figure 4.2-23) yields,

$$\bar{\tau}_1(X) = \frac{A_1}{2\pi R} \frac{d\sigma_1}{dX} \quad \text{EQ \#4.2-68a}$$

$$\bar{\tau}_2(X) = -\frac{A_2}{2\pi R} \frac{d\sigma_2}{dX} \quad \text{EQ \#4.2-68b}$$

Making use of EQ #4.2-65 shows that $\bar{\tau}_1(X) = \bar{\tau}_2(X) = \bar{\tau}_T(X)$

Therefore, substituting EQ #4.2-67 into EQ #4.2-68a gives,

$$\bar{\tau}_T(X) = \frac{A_1}{2\pi R} \frac{E\alpha}{(1 + \frac{A_1}{A_2})} \frac{d}{dX} [\bar{T}_2(X) - \bar{T}_1(X)] \quad \text{EQ \#4.2-69}$$

FIGURE 4.2-23

AXIAL ELEMENT OF EACH TUBE FOR
EQUILIBRIUM CONSIDERATIONS

OUTER TUBE

INNER TUBE

4.2.2.1 RELATIVE SLIDING ANALYSIS (continued)

By integrating equation 4.2-69 we obtain,

$$\bar{\tau}_T = \frac{A_1}{2\pi R} \frac{E\alpha}{\left(1 + \frac{A_1}{A_2}\right)} \frac{1}{L} \{[\bar{T}_2(L) - \bar{T}_2(0)] - [\bar{T}_1(L) - \bar{T}_1(0)]\}$$

EQ #4.2-70

4.2.2.2 SLIDING EVALUATION

Combining equations 4.2-46 and 4.2-47, local

sliding is prevented provided,

$$\bar{\tau}_p(X) + \bar{\tau}_F(X) + \bar{\tau}_T(X) \leq f_s \bar{p}_c(X)$$

EQ #4.2-71a

and gross sliding is prevented if,

$$\bar{\tau}_p + \bar{\tau}_F + \bar{\tau}_T \leq f_s \bar{p}_c$$

EQ #4.2-71b

By the preceeding analyses, the maximum shear stress at any cross section for each loading is,

$$\bar{\tau}_{p_{\max}} = \frac{p_1 (R - t_1)^2 - p_2 (R + t_2)^2}{RL}$$

EQ #4.2-51

$$\bar{\tau}_{F_{\max}} = C_F \left[(T_T - T_S) + \frac{t_A}{L_S} (T_T - T_A) \right]$$

EQ #4.2-72

$$\bar{\tau}_{T_{\max}} = C_T \left\{ \frac{d}{dx} [\bar{T}_2(X) - \bar{T}_1(X)] \right\}_{\max}$$

EQ #4.2-73

where

$$C_F = \frac{\lambda E \alpha A_1}{2\pi R} \frac{1}{\left\{ \frac{NA_1}{A_S} + 1 + \frac{t_A}{L_S} \right\}}$$

EQ #4.2-74

$$C_T = \frac{A_1}{2\pi R} \frac{E\alpha}{\left\{ 1 + \frac{A_1}{A_2} \right\}}$$

EQ #4.2-75

4.2.2.2 SLIDING EVALUATION (continued)

The uniform (along the tube) shear stresses

for each loading are,

$$\bar{\tau}_p = \frac{\tau_{p_{max}}}{2} \quad \text{EQ \#4.2-52}$$

$$\bar{\tau}_F = \frac{\tau_{F_{max}}}{L\lambda} \quad \text{EQ \#4.2-53}$$

$$\bar{\tau}_T = C_T \frac{1}{L} \{ [\bar{T}_2(L) - \bar{T}_2(0)] - [\bar{T}_1(L) - \bar{T}_1(0)] \} \quad \text{EQ \#4.2-70}$$

Upon reviewing the preceeding equations, it was judged that various uncertainties precluded a really quantitative evaluation of sliding. The uncertainties were associated with:

1. $\bar{p}_c$ (contact pressure acting between tubes)
2. f_s (coefficient of static friction)
3. Thermal loadings (F_T and $T(X, r)$)
4. Analyses herein (solution to this problem is not in classical literature, must be derived).

Therefore, a conservative assessment of the potential for sliding was performed.

For this purpose, a representative value for the frictional resistance of

$f_s \bar{p}_c = (0.3)(1000) = 300$ psi was used.

4.2.2.2 SLIDING EVALUATION (continued)

Representative values for the geometric factors in the above equations are provided in Table 4.2-X for the duplex tube superheater, evaporator and reheater.

Since p_1 cannot exceed the steam pressure, p_{H_2O} , and $p_2 \geq 0$, a conservative approximation to $\bar{\tau}_{p_{max}}$ is,

$$\bar{\tau}_{p_{max}} = p_{H_2O} \frac{R}{L} \quad \text{EQ \#4.2-76}$$

Values for the operating steam pressure (Ref. 4.2-13) and nominal values for $E\alpha$ are listed in Table XI. Using values of R and L from Table X and p_{H_2O} from Table XI, the largest value calculated for $\bar{\tau}_{p_{max}}$ was 3.83 psi (for the superheater). Comparing 3.8 psi with the assumed lower bound frictional resistance of 300 psi, it was concluded that pressure loading has no significant potential to cause either local or gross sliding. This finding was considered to be sufficiently quantitative to eliminate pressure loading from further consideration.

The coefficients C_F , C_T , and t_A/L_S for evaluation of the remaining two loadings are given in Table XII. Substituting $C_F = 28.2$ and $C_T = 6.15$ from Table XII for the evaporator (worst case) into equations EQ #4.2-72 and 4.2-73 gave:

TABLE X
GEOMETRIC PARAMETERS FOR
SLIDING EVALUATION

UNIT	$R^{(1)}$ (in.)	$t_1^{(1)}$ (in.)	$t_2^{(1)}$ (in.)	$L^{(2)}$ (ft.)	$N^{(2)}$	$R_S^{(2)}$ (in.)	$t_S^{(3)}$ (in.)	$L_S^{(3)}$ (in.)	$t_A^{(3)}$ (in.)
SUPERHEATER	0.558	0.062	0.082	31.0	1346	48.4	2.5	18.0	12.0
EVAPORATOR	0.578	0.067	0.080	49.0	610	35.1	2.5*	18.0	9.0
REHEATER	0.900	0.055	0.063	24.0	1273	52.0*	2.5	18.0	20.0

(1) Obtained from Reference 4.2-13 for reference (uniform thickness) design.

(2) Obtained from Ref. 4.2-20.

(3) Obtained from drawings E-63675023-0/E-63675022-0/E-63675033-0 for the superheater/evaporator/reheater.

* Assumed representative.

TABLE XI
MATERIAL PROPERTIES AND
PRESSURES FOR SLIDING EVALUATION

UNIT	MATERIAL	α_m^* (IN/IN-°F)	$E \alpha$ (PSI/°F) @Room Temp	P_{H_2O} (PSI)
SUPERHEATER	HT-9	6.85×10^{-6}	169	2550
EVAPORATOR	HT-9	6.75×10^{-6}	169	2650
REHEATER	2 1/4 Cr - 1 Mo	7.97×10^{-6}	182	555

*Mean (from room temperature) coefficient of thermal expansion.

TABLE XII
INTERIM RESULTS OF CALCULATIONS
FOR SLIDING EVALUATION

UNIT	C_F (PSI/°F)	C_T (IN-PSI/°F)	t_A/L_S	$\frac{NA_1}{A_S}$	λ (1/IN)
SUPERHEATER	23.4	5.95	0.66	0.384	4.55
EVAPORATOR	28.2	6.15	0.50	0.269	4.41
REHEATER	20.0	5.35	0.50	0.484	3.95

4.2.2.2 SLIDING EVALUATION (continued)

$$\bar{\tau}_{F_{\max}} = \left(\frac{28.2 \text{ psi}}{^{\circ}\text{F}} \right) \Delta T_{\text{eq}} (^{\circ}\text{F}) \quad \text{EQ \#4.2-77}$$

$$\bar{\tau}_{T_{\max}} = \left(\frac{6.15 \text{ psi}}{^{\circ}\text{F/in}} \right) \left\{ \frac{d}{dx} [\bar{T}_2 - \bar{T}_1] \right\}_{\max} \quad \text{EQ \#4.2-78}$$

where

$$\Delta T_{\text{eq}} = [(T_T - T_S) + \frac{t_A}{L_S} (T_T - T_A)] \quad \text{EQ \#4.2-79}$$

is an equivalent tubesheet region temperature differential. These results indicate that the shear stress created by either $\Delta T_{\text{eq}} = 10.6^{\circ}\text{F}$ or $\frac{d}{dx} \{\bar{T}_2 - \bar{T}_1\} = 50^{\circ} \text{ F/in}$ would equal the frictional resistance of 300 psi and would cause local sliding. Since it is the average wall temperature and not the surface temperature involved, it was judged that $\frac{d}{dx} \{\bar{T}_2 - \bar{T}_1\}$ is much less than 50°F/in . in practice. Thus, $\bar{\tau}_T$, like $\bar{\tau}_p$ was considered, quantitatively, to have no significant potential to cause local sliding.

However, in regard to $\bar{\tau}_F$, actual values for the equivalent tubesheet region temperature differential, ΔT_{eq} , have not been determined. This should be investigated further. It is judged that operating conditions such as startup, shutdown, scram, and swing loads might cause small temperature differentials of an unknown amount. If a temperature differential of 10.6°F were achieved, then differential thermal expansion in the tubesheet region would be the only potential source of local sliding of all the cases investigated here.

4.2.2.2 SLIDING EVALUATION (continued)

One further point that should be noted is that the tubesheets were considered to be rigid in this analysis. Further analysis should acknowledge possible dishing of the tubesheets.

Figure 4.2-24 shows a plot of $\bar{\tau}_F$ versus X several values of ΔT_{eq} and allows comparison of the severity of various temperature differentials for exceeding two values of the frictional resistance. With $\Delta T_{eq} = 10.6^\circ\text{F}$, the frictional resistance of 300 psi is just equaled at the point where the two tubes join in contact. At $\Delta T_{eq} = 31.8^\circ\text{F}$, the frictional resistance of 300 psi is exceeded over a length of 0.25 inches from where the two tubes join in contact.

In the case of gross sliding, $\bar{\tau}_p$ has already been found insignificant. Due to the presence of L in the denominator of equations 4.2-58 and 4.2-70, both $\bar{\tau}_F$ and $\bar{\tau}_T$ are also insignificant. Thus, in no case is gross sliding anticipated. For example, back calculation of the smallest thermal load required to produce 300 psi shear stress from equations 4.2-58 and 4.2-70 gave $\bar{\tau}_F = 300$ psi with $\Delta T_{eq} = 12100^\circ\text{F}$. Such a ΔT_{eq} is impossible and the gross sliding observations are considered to be sufficiently quantitative.

FIGURE 4.2-24

SHEARING STRESS DISTRIBUTION FOR VARIOUS TUBESHEET TEMPERATURE DIFFERENTIALS IN EVAPORATOR

4.2.2.3 LOSS OF CONTACT ANALYSIS

The investigation into the contact of two bodies involves careful accounting of the position of their surfaces relative to each other. Hence, an accurate analysis for surface displacements due to all applied loads is required. These displacements must be referenced to a stress-free, uniform temperature, initial surface configuration. The contact problem is complicated because the boundary condition changes when contact is made or broken. Thus, during contact, a structural interaction analysis is necessary; whereas, when there is no contact, each body may be analyzed independently. This nature of the boundary condition makes the contact problem non-linear, even if the material behavior is linear elastic.

In the case of the duplex tubes, the problem may be further complicated because the swaging process inherently involves (small) plastic deformation. This introduces a load history dependent material behavior before application of the very first service loading. This could complicate a comprehensive analysis for subsequent loadings.

The loadings considered significant in this investigation for the loss of contact evaluation were: (1) swaging loads, (2) pressures and (3) temperature loadings. The scope of this task did not include an (inelastic) analysis of the swaging loads. The non-linearity associated with load history dependent material behavior introduced by

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

swaging was avoided by assuming elastic response for the pressure and temperature loadings. The non-linearity associated with the contact boundary condition was also avoided by assuming that contact existed after swaging and for the analysis of subsequent pressure and temperature loadings. However, displacements due to swaging were included qualitatively to indicate the interrelation of all the significant loadings.

All loadings and deformations were assumed to be axisymmetric and to vary in the radial direction only. Variation along the tubes was ignored. Using this model, the radial displacement of either tube could be computed directly from the total hoop strain. For small strains, the total hoop strain is the sum of elastic, plastic and thermal expansion strains.

$$\frac{U}{r} = \epsilon_{\theta}^{TOT} = \epsilon_{\theta}^{EL} + \epsilon_{\theta}^{PL} + \epsilon_{\theta}^{TE} \quad \text{EQ \#4.2-80}$$

where

$$\epsilon_{\theta}^{TE} = \alpha_m [T(r) - T_0]$$

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

Plastic behavior was assumed to occur only during swaging. The material behavior during subsequent pressure and thermal loading was assumed elastic. The elastic strain was then the sum of strains due to pressure and temperature stresses. Therefore, the displacement of the contact surface of each tube was obtained from the sum of elastic (pressure and temperature), plastic, and thermal expansion strains:

$$\frac{U_{c1}}{R} = \epsilon_{\theta 1 p}^{EL} + \epsilon_{\theta 1 T}^{EL} + \epsilon_{\theta 1}^{PL} + \alpha_m [T_{c1} - T_o] \text{ EQ \#4.2-81a}$$

$$\frac{U_{c2}}{R} = \epsilon_{\theta 2 p}^{EL} + \epsilon_{\theta 2 T}^{EL} + \epsilon_{\theta 2}^{PL} + \alpha_m [T_{c2} - T_o] \text{ EQ \#4.2-81b}$$

A different temperature (T_{c1} , T_{c2}) was used for the contact surface of each tube to reflect the fact that the thermal contact resistance can be significant. Note that since the displacements are small ($\frac{U}{R} \ll 1.0$), the radius, R , of the contact surface after swaging may be used to calculate displacements. The elastic hoop strains associated with pressure and temperature stresses were determined as follows:

a. Pressure Displacements

The pressure loadings acting on the tubes separately and combined are shown in Figure 4.2-25 for the case where the tubes are in contact. The equivalent contact pressure, $\bar{p}_c$, includes both the load-controlled, $\bar{p}_{c_{Lc}}$, and the deformation-controlled part, $\bar{p}_{c_{DC}}$.

FIGURE 4.2-25

PRESSURE LOADINGS ACTING ON DUPLEX TUBES
IN CONTACT

BOTH TUBES COMBINED

INNER TUBE

OUTER TUBE

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

$$\bar{p}_c = \bar{p}_{c_{Lc}} + \bar{p}_{c_{Dc}} \quad \text{EQ \#4.2-82}$$

If the tubes are not in contact then $\bar{p}_c$ is replaced by p_{gap} , the applied pressure loading in the grooves.

The radial and hoop stresses in an open-ended thick cylinder due to internal and external pressures p_i , p_o are given by Reference 4.2-21,

$$\sigma_r = \frac{a^2 p_i - b^2 p_o - \frac{a^2 b^2}{r^2} (p_i - p_o)}{b^2 - a^2} \quad \text{EQ \#4.2-83a}$$

$$\sigma_\theta = \frac{a^2 p_i - b^2 p_o + \frac{a^2 b^2}{r^2} (p_i - p_o)}{b^2 - a^2} \quad \text{EQ \#4.2-83b}$$

where a, b are the internal and external radii, respectively. For a closed-ended cylinder, the following axial stresses must be included also,

$$\sigma_z = \frac{p_i a^2 - p_o b^2}{b^2 - a^2} \quad \text{EQ \#4.2-84}$$

The corresponding hoop strain from pressure induced stresses σ_r , σ_θ , σ_z is

$$\epsilon_\theta^{\text{EL}} = \frac{1}{E} [\sigma_\theta - \nu (\sigma_r + \sigma_z)] \quad \text{EQ \#4.2-85}$$

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

Substituting equations 4.2-83 and 84 into 85 gives,

$$\epsilon_{\theta p}^{EL} = \frac{1-2\nu}{E} \left[\frac{a^2 p_i - b^2 p_o}{b^2 - a^2} \right] + \frac{1+\nu}{E} \left[\frac{a^2 b^2}{r^2} \frac{(p_i - p_o)}{b^2 - a^2} \right] \quad \text{EQ \#4.2-86}$$

b. Thermal Displacements

The thermal stresses due to a radial temperature distribution, $T(r) = T(r) - T_o$, are given by Reference 4.2-19, p 412),

$$\sigma_r = \frac{E\alpha_m}{1-\nu} \frac{1}{r^2} \left[\frac{r^2 - a^2}{b^2 - a^2} \int_a^b T' r dr - \int_a^r T' r dr \right] \quad \text{EQ \#4.2-87a}$$

$$\sigma_{\theta} = \frac{E\alpha_m}{1-\nu} \frac{1}{r^2} \left[\frac{r^2 + a^2}{b^2 - a^2} \int_a^b T' r dr + \int_a^b T' r dr - T' r^2 \right] \quad \text{EQ \#4.2-87b}$$

$$\sigma_z = \frac{E\alpha_m}{1-\nu} \left[\frac{2}{b^2 - a^2} \int_a^r T' r dr - T' \right] \quad \text{EQ \#4.2-87c}$$

Substituting equations 4.2-87 into 85 gives,

$$\begin{aligned} \epsilon_{\theta T}^{EL} = & \frac{\alpha_m}{1-\nu} \frac{1}{b^2 - a^2} \left[1 - 3\nu + (1 + \nu) \frac{a^2}{r^2} \right] \int_a^b T' r dr \\ & + \frac{\alpha_m (1 + \nu)}{(1 - \nu)} \frac{1}{r^2} \int_a^r T' r dr - \alpha_m T' \end{aligned} \quad \text{EQ \#4.2-88}$$

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

c. Combined Displacements

Substituting Equation 4.2-86 and 88 into

4.2-81a using $r = R$, $a = R_1$, $b = R$, $p_i = p_1$, $p_o = \bar{p}_c$, the radial displacement of the contact surface of the inner tube is:

$$\begin{aligned} \frac{U_{1c}}{R} = & \left\{ \frac{1-2\nu}{E} (R_1^2 p_1 - R^2 \bar{p}_c) + \frac{1+\nu}{E} R_1^2 (\bar{p}_c - p_1) \right. \\ & \left. + 2\alpha_m \int_{R_1}^R (T - T_0) r dr \right\} / (R^2 - R_1^2) + \epsilon_{\theta 1}^{PL} \end{aligned} \quad \text{EQ \#4.2-89a}$$

Similarly, using $r = R$, $a = R$, $b = R_2$, $p_i = \bar{p}_c$, $p_o = p_2$, the radial displacement of the contact surface for the outer tube is:

$$\begin{aligned} \frac{U_{2c}}{R} = & \left\{ \frac{1-2\nu}{E} (R^2 \bar{p}_c - R_2^2 p_2) + \frac{1+\nu}{E} R_2^2 (\bar{p}_c - p_2) \right. \\ & \left. + 2\alpha_m \int_R^{R_2} (T - T_0) r dr \right\} / (R_2^2 - R^2) + \epsilon_{\theta 2}^{PL} \end{aligned} \quad \text{EQ \#4.2-89b}$$

An inelastic analysis of the swaging process would provide $\epsilon_{\sigma 1}^{PL}, \epsilon_{\sigma 2}^{PL}$ in terms of initial geometry, material properties and the swaging load (applied displacement together with an associated initial contact pressure, $\bar{p}_c$). Assuming elastic behavior for all loadings after swaging, $\epsilon_{\sigma 1}^{PL}, \epsilon_{\sigma 2}^{PL}$ remain constant under these conditions, the change in displacement may be obtained from Equation 4.2-89a and 89b.

$$\begin{aligned} \frac{\Delta U_{1c}}{R} = & \left\{ \frac{1-2\nu}{E} (R_1^2 \Delta p_1 - R^2 \Delta \bar{p}_c) + \frac{1+\nu}{E} R_1^2 (\Delta p_1 - \Delta \bar{p}_c) \right. \\ & \left. + 2\alpha_m \int_{R_1}^R \Delta T r dr \right\} / (R^2 - R_1^2) \end{aligned} \quad \text{EQ \#4.2-90a}$$

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

$$\frac{\Delta U_{2c}}{R} = \left\{ \frac{1-2\nu}{E} (R^2 \Delta \bar{p}_c - R_2^2 \Delta p_2) + \frac{1+\nu}{E} R_2^2 (\Delta \bar{p}_c - \Delta p_2) + 2\alpha_m \int_R^{R_2} \Delta T r dr \right\} / (R_2^2 - R^2) \quad \text{EQ \#4.2-90b}$$

For convenience, a load increment throughout which the tubes remain in contact is called a contact load increment. Also, a load increment throughout which the tubes are not in contact is referred to as a no-contact load increment.

During a contact load increment, the position of the contact surface of each tube is equal, $r_{1c} = r_{2c}$, and hence $\Delta U_{1c} = \Delta U_{2c}$. Equating Equations 4.2-90a and 90b provides a relation between $\Delta p_1, \Delta p_2, \Delta \bar{p}_c$ and $\Delta T(r)$. Δp_1 and Δp_2 are known applied external loadings. The one-dimensional (radial) heat transfer problem would provide $\Delta T(r)$. However, since the thermal resistance (Ref. 4.2-22) across the surfaces in contact varies significantly with $\bar{p}_c$, thermoelastic coupling may exist during a contact load increment. When coupling exists, the heat transfer problem then provides a second relation between $\Delta \bar{p}_c$ and $\Delta T(r)$.

During a no-contact load increment, $\bar{p}_c$ is equal to the known pressure loading, p_{gap} , applied in between the tubes. Since small displacements can have a significant effect on the thermal resistance (See Figure 3, Ref. 4.2-22) across the gap between the

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

tubes , there may be thermoelastic coupling in this case also. The one-dimensional (radial) heat transfer problem would provide a second relation between the tube separation distance (relative displacement) and $\Delta T(r)$.

Summarizing, during a no-contact load increment,

$\bar{p}_c = p_{\text{gap}}$ and $r_{1c} \leq r_{2c}$, whereas during a contact load increment, $\bar{p}_c \geq p_{\text{gap}}$ and $r_{1c} = r_{2c}$. If $\bar{p}_c = p_{\text{gap}}$ at the end of a contact load increment, loss of contact occurs and is followed by a no-contact load increment.

On the other hand, obtaining contact requires that $r_{1c} = r_{2c}$ at the end of a no-contact load increment, and is followed by a contact load increment.

In general, for both kinds of load increment thermoelastic coupling may exist.

For this investigation, no thermoelastic coupling analyses were considered necessary for the following reasons. First of all, contact is assumed to exist after swaging and during subsequent loadings. If, under these assumptions, loss of contact is found to occur then the conditions necessary to maintain contact are thereby determined. Therefore, only contact load increments were analyzed. Also, reference 4.2-22 showed that (relative to the other thermal resistances involved) essentially infinite gap conductance ($7000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$) exists if $\bar{p}_c \geq 1000 \text{ psi}$, approximately. Therefore, assuming $\bar{p}_c \geq 1000 \text{ psi}$, thermoelastic coupling occurs only if $\bar{p}_c > 1000 \text{ psi}$. Since the steam generator units have been sized assuming $\bar{p}_c \geq 1000 \text{ psi}$ during normal operating conditions, the temperature distributions $T(r)$ corresponding to normal

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

operating conditions were calculated assuming no thermal elastic coupling.

The displacement of the contact surface of each tube is given by equations 4.2-89a and 89b. To illustrate the tendency of the tubes to separate due to the various causes it is convenient to consider the displacement of the outer tube relative to the inner tube, $\delta = U_{2c} - U_{1c}$.

$$\text{Or, } \frac{\delta}{R} = \frac{(2-\nu)}{E} \left\{ \frac{R^2(R_2^2 - R_1^2)}{(R^2 - R_1^2)(R_2^2 - R^2)} \bar{p}_c - \frac{R_1^2}{(R^2 - R_1^2)} p_1 - \frac{R_2^2}{(R_2^2 - R^2)} p_2 \right\} \\ + \frac{2\alpha_m}{(R_2^2 - R^2)} \int_0^{R_2} T r dr - \frac{2\alpha_m}{(R^2 - R_1^2)} \int_0^R T r dr + \frac{\delta_{swag}}{R} \quad \text{EQ \#4.2-91}$$

where because the inner tube is plastically deformed more than the outer tube.

$$\frac{\delta_{swag}}{R} = \frac{U_{2c}^{swag}}{R} - \frac{U_{1c}^{swag}}{R} = \epsilon_{\theta 1}^{PL} - \epsilon_{\theta 1}^{PL} < 0 \quad \text{EQ \#4.2-92}$$

Since $\bar{p}_c \geq p_{gap}$, let

$$\bar{p}_c^* = \bar{p}_c - p_{gap} \geq 0 \quad \text{EQ \#4.2-93}$$

Now Equation 4.2-91 can be separated as,

$$\frac{\delta}{R} = \frac{\delta^*}{R} + \frac{\delta_p}{R} + \frac{\delta_T}{R} + \frac{\delta_{swag}}{R} \quad \text{EQ \#4.2-94}$$

where displacement due to contact pressure in excess of p_{gap} is:

$$\frac{\delta^*}{R} = \frac{(2-\nu)}{E} \left\{ \frac{R^2(R_2^2 - R_1^2)}{(R^2 - R_1^2)(R_2^2 - R^2)} \bar{p}_c^* \right\} \quad \text{EQ \#4.2-95}$$

4.2.2.3 LOSS OF CONTACT ANALYSIS (Continued)

and displacement from known applied pressures is:

$$\frac{\delta}{R} = \frac{(2-\nu)}{E} \left[\frac{R^2(R_2^2 - R_1^2)}{(R^2 - R_1^2)(R_2^2 - R_1^2)} p_a - \left(\frac{R_1^2}{R^2 - R_1^2} \right) p_1 - \left(\frac{R_2^2}{R^2 - R_2^2} \right) p_2 \right] \quad \text{EQ \#4.2-96}$$

and displacement from temperature distribution is:

$$\frac{\delta_T}{R} = 2\alpha_m \left[\frac{1}{R^2 - R_1^2} \int_{R_1}^R r^2 T r dr - \frac{1}{R^2 - R_1^2} \int_{R_1}^R T r dr \right] \quad \text{EQ \#4.2-97}$$

δ_T depends only on the equivalent linear characteristics of the temperature distribution. The equivalent linear temperature distribution for each tube is,

Inner Tube

$$T_1(r) = \bar{T}_1 + \frac{\Delta T_1^L}{(R - R_1)} \left(r - \frac{R_1 + R}{2} \right) \quad \text{EQ \#4.2-98a}$$

Outer Tube

$$T_2(r) = \bar{T}_2 + \frac{\Delta T_2^L}{(R_2 - R)} \left(r - \frac{R + R_2}{2} \right) \quad \text{EQ \#4.2-98b}$$

Substituting Equations 4.2-98a and 98b into 97 gives,

$$\frac{\delta_T}{R} = \alpha_m (\bar{T}_2 - \bar{T}_1) + \frac{\alpha_m}{6} \left[\frac{t_2 \Delta T_2^L}{R + R_2} - \frac{t_1 \Delta T_1^L}{R + R_1} \right] \quad \text{EQ \#4.2-99}$$

4.2.2.4 LOSS OF CONTACT EVALUATION

For computation purposes, practical approximations to equations 4.2-95, 96 and 99 are provided by assuming $R_1 + R_2 \cong 2R$,

$$R + R_1 \cong 2R, \quad R + R_2 \cong 2R, \quad \frac{R_1^2}{R + R_2} \cong \frac{R}{2}, \quad \frac{R_2^2}{R + R_1} \cong \frac{R}{2},$$

$$\frac{\delta^*}{R} = \frac{(2-\nu)}{E} \left[\frac{(t_1 + t_2) R}{2 t_1 t_2} \bar{p}_c^* \right] \quad \text{EQ \#4.2-100}$$

$$\frac{\delta_p}{R} = \frac{(2-\nu)}{E} \left[\frac{(t_1 + t_2) R}{2 t_1 t_2} p_{\text{gap}} - \frac{R}{2 t_1} p_1 - \frac{R}{2 t_2} p_2 \right] \quad \text{EQ \#4.2-101}$$

$$\frac{\delta_T}{R} = \alpha_m (\bar{T}_2 - \bar{T}_1) + \frac{\alpha_m}{12} \left[\frac{t_2}{R} \Delta T_2^L - \frac{t_1}{R} \Delta T_1^L \right] \quad \text{EQ \#4.2-102}$$

For convenient reference recall,

$$\frac{\delta_{\text{swag}}}{R} = \frac{U_{2C}^{\text{Swag}}}{R} - \frac{U_{1C}^{\text{Swag}}}{R} = \epsilon_{\theta 2}^{\text{PL}} - \epsilon_{\theta 1}^{\text{PL}} \quad \text{EQ \#4.2-92}$$

and

$$\frac{\delta}{R} = \frac{\delta^*}{R} + \frac{\delta_p}{R} + \frac{\delta_T}{R} + \frac{\delta_{\text{swag}}}{R} \quad \text{EQ \#4.2-94}$$

(+) (-) (+) (-)

4.2.2.4 LOSS OF CONTACT EVALUATION (Continued)

The algebraic sign of each term in equation 4.2-94 is indicated below each term. δ^* is positive from equation 4.2-100 since $\bar{p}_c^*$ is positive from equation 4.2-93. δ_{swag} is negative (by definition) as shown in equation 4.2-92. The signs of δ_p and δ_T depend on the loadings. However, it turned out (See Table 4.2-XVI) that δ_p was always negative and δ_T was always positive for the loading cases considered.

If contact is maintained, then $\delta = 0$, and equation 4.2-94 gives,

$$\frac{\delta^*}{R} = - \left(\frac{\delta_p}{R} + \frac{\delta_T}{R} + \frac{\delta_{\text{swag}}}{R} \right) \geq 0 \quad \text{EQ \#4.2-103}$$

because δ^* must be positive for physically real (i. e., $\bar{p}_c^* > 0$) contact pressure. Since δ_p , δ_T , δ_{swag} can all be varied independently through the applied pressures, temperatures and the extent of swaging, equation EQ #4.2-103 shows that all combinations do not necessarily result in maintaining contact. Only those combinations of pressures, temperatures and swaging which result in $\delta^* > 0$ maintain contact. Since δ_{swag} is unknown, an absolute evaluation of loss of contact cannot be made. However, from equation 4.2-103 we can determine the smallest value of δ_{swag} required to maintain contact,

$$\frac{\delta_{\text{swag}}}{R} = - \left(\frac{\delta_p}{R} + \frac{\delta_T}{R} \right) \quad \text{EQ \#4.2-104}$$

4.2.2.4 LOSS OF CONTACT EVALUATION (Continued)

The pressures p_1 , p_2 , p_{gap} corresponding to normal operating conditions are listed in Table 4.2-XIII. The equivalent linear temperature distribution characteristics for steady state normal operating conditions are provided in Table 4.2-XIV. Tube thicknesses and radii for both the uniform thickness (reference) and the bi-thickness (alternate) inner tube designs are provided in Table 4.2-XV. δ_p and δ_T , computed from the information in Tables 4.2-XIII, XIV, and XV, are presented in Table 4.2-XVI. Note that in the equation for $\delta T/R$ (equation 4.2-102) the last term is negligible in comparison to the first term. The values of δ_p and δ_T in Table XVI show that

$$- \left(\frac{\delta p}{R} + \frac{\delta T}{R} \right) > 0$$

for the superheater and evaporator but not the reheater. This means that contact would be maintained in the superheater and evaporator even without swaging. This is because the pressures tend to squeeze the tubes together more than thermal expansion tends to separate them. The values of $|\delta_p|/\delta_T$ in Table 4.2-XVI indicate that the pressure displacements are more than twice that which is sufficient to compensate for the thermal displacements in the superheater and evaporator. In the case of the reheater the pressure displacements are about one-third (0.361) that necessary to compensate for the thermal displacements. Assuming

TABLE 4.2-XIII
PRESSURES CORRESPONDING TO NORMAL
OPERATING CONDITIONS

UNIT	P_1 (P_{H_2O}) lb/in ²	P_{gap} (P_{He}) lb/in ²	P_2 (P_{Na}) lb/in ²
SUPERHEATER	2550	300	120
EVAPORATOR	2650	300	95
REHEATER	555	300	120

Values obtained from Table 1 of Reference 4.2-13

TABLE 4.2-XIV

EQUIVALENT LINEAR CHARACTERISTICS OF TEMPERATURE DISTRIBUTION
FOR STEADY STATE NORMAL OPERATING CONDITIONS

	Uniform Thickness (Reference) Design					Non-Uniform Thickness (Alternate) Design				
UNIT	$\bar{T}_1$	$\bar{T}_2$	ΔT_1^L	ΔT_2^L	ΔT_c	$\bar{T}_1$	$\bar{T}_2$	ΔT_1^L	ΔT_2^L	ΔT_c
SUPERHEATER	865	893	-19	-22	-8	864	891	-13	-24	-9
EVAPORATOR	681	729	-34	-36	-14	631	722	-23	-43	-17
REHEATER	879	901	-11	-12	-6	-	-	-	-	-

All temperatures in $^{\circ}\text{F}$.

TABLE 4.2-XV
 DUPLEX TUBE THICKNESSES AND RADII (INCHES)

UNIT	MATERIAL	Reference Design Uniform Thickness			Alternate Design Non-Uniform Thickness		
		R ₁	U t ₁	U t ₂	TS t ₁	N t ₁	t ₂ ^N
SUPERHEATER	HT - 9	.558	0.062	0.082	0.062	0.039	0.080
EVAPORATOR	HT - 9	.578	0.067	0.080	0.067	0.038	0.078
REHEATER	2 1/4 Cr-1 Mo	.900	0.055	0.063	- *	- *	- *

*There is no alternate design for the reheater.

TABLE 4.2-XVI

STRAINS/DISPLACEMENTS ASSOCIATED WITH SWAGING AND
NORMAL OPERATING PRESSURES AND TEMPERATURES FOR
LOSS OF CONTACT EVALUATION

UNIT	Reference Design				Alternate Design			
	ϵ_p/R (%)	ϵ_T/R (%)	$ \frac{\epsilon_p}{\epsilon_T} $	Min. U_{lc}^{Swag} (inches)	ϵ_p/R (%)	ϵ_T/R (%)	$ \frac{\epsilon_p}{\epsilon_T} $	Min. U_{lc}^{Swag} (inches)
SUPERHEATER	-0.0946	+0.0192	4.93	0.0*	-0.148	+0.0185	8.0	0.0*
EVAPORATOR	-0.0842	+0.0314	2.68	0.0*	-0.147	+0.0614	2.4	0.0*
REHEATER	-0.00632	+0.0175	0.361	0.0001	-	-	-	-

*Pressures alone are sufficient to compensate for thermal displacements and maintain contact.

4.2.2.4 LOSS OF CONTACT EVALUATION (Continued)

$U_{2c}^{swag} = 0$ for simplicity, U_{1c}^{swag} was computed from equations 4.2-92 and 104 and is included in Table 4.2-XVI also. As computed, U_{1c}^{swag} is the minimum (plastic) swaging displacement of the contact surface of the inner tube necessary to maintain contact. These results indicate that very little (if any) swaging is required to maintain contact. Since considerably more than 0.0001 inches of relative swaging displacement is anticipated, it was concluded that loss of contact would not occur under normal operating pressures and temperatures in any of the steam generator units.

Thermal transients induced by reactor scrams would lower sodium temperature, resulting in elastic contraction of the outer tube. Thus, thermal transients were not considered as potential sources of loss of contact. Furthermore, loss of contact during a thermal transient is not as significant as during normal operation.

A. Consideration of Creep Deformation

It was agreed (Ref. 4.2-23) at the start of the duplex tube structural evaluation that a brief consideration should be given to the maximum possible effect of creep deformation on tube contact. This is discussed next.

4.2.2.4 LOSS OF CONTACT EVALUATION (Continued)

The duplex tube sizing procedure included, where applicable, allowable stress limits which would limit creep deformation to 1% in 200,000 hours. Thus, at most, a maximum of 1% creep deformation would be anticipated. Intuitively, it is expected that creep deformation would interact with the swaging, pressure, and temperature loadings in the following manner:

Swaging - The contact pressure, $\bar{p}_c^*$ would eventually relax to zero with sufficient creep deformation. In the absence of knowledge of actual swaging strain, the magnitude of creep necessary to relax $\bar{p}_c^*$ to zero is unknown.

Pressure - During operation there is a net internal pressure applied to both tubes combined. Creep deformation would expand both tubes radially. It was previously shown that δ_p , pressure induced displacement, tends to squeeze both tubes together. Thus, this aspect of the pressure loadings would assist in maintaining contact and means that creep would only tend to reduce the combined wall thickness (a maximum of 1%).

Thermal - Creep strain accumulation due to the pressure loadings could eventually nullify the effect of the relative thermal displacement of the tubes.

4.2.2.4 LOSS OF CONTACT EVALUATION (Continued)

General Creep Considerations- It is noted that appreciable creep behavior is not anticipated with either HT-9 or 2 1/4 Cr - 1 Mo because of the low temperatures involved. By appreciable creep behavior is meant creep strains greater than approximately 0.5%. However, creep strains less than 0.5% could still have a significant effect on the tube contact conditions. As seen previously, a small strain can significantly modify the compatibility conditions at the tube contact surface. In fact, judging from the strains in Table 4.2-XVI, a creep strain of only 0.06% is sufficient to eliminate the effect of thermal loading.

B. Conclusions Regarding Creep

In the extreme, creep deformation eventually would nullify the effects of the swaging and thermal loads. Thereafter, creep deformation would be controlled by pressure loadings. Pressure induced creep deformation would tend to expand both tubes (with a corresponding thickness reduction). Most significant is that the squeezing effect of the pressure would maintain contact, even in the reheater, after creep deformation had occurred.

4.2.3 DETERMINATION OF CONTACT PRESSURES

Since the contact pressure which can be established and maintained between the pressure tubes is an important factor in both the thermal and the structural response of the duplex tube steam generator, a brief study was performed to determine the attainable contact pressures. The primary objective of the study was the establishment of the functional relationship which exists between the expanding displacement applied to the inside surface of the inner pressure tube and the residual contact pressure after completion of the expanding process. The sensitivity of the residual contact pressure to both initial tube clearance and the elasto-plastic response of the tube materials was also determined by varying these parameters over their likely limits.

The secondary objective of the contact pressure study was to investigate the effect of the nominal steady state loading on the tubes. The change in the contact pressure between tubes resulting from the nominal steady state condition was calculated. The implication of the calculated results with regard to failure of the tubes through violation of the functional operating requirements of no tube separation or relative sliding was also considered.

4.2.3 DETERMINATION OF CONTACT PRESSURES (continued)

4.2.3.1 SCOPE LIMITATIONS

The scope of this study was restricted in several areas to ensure the development of meaningful results in the available time without compromising the accuracy of the effort. These scope limitations and their justification are discussed in the balance of this section.

(1) Duplex Tube Geometry - Only the tube geometry corresponding to the C-E reference design evaporator unit was considered in the study. Since both the evaporator and the superheater are constructed of SANDVIK HT9 and have quite similar pressure tube dimensions, relationships which are developed for the residual contact pressure in the evaporator unit duplex tube module should be good estimates for the comparable relationships in the superheater unit. Further, from the nominal steady state loadings in the three units it appears that failure by violation of the functional operating requirements will be most critical in the evaporator unit. On consideration of these factors, a detailed analysis of a duplex tube module from the evaporator unit was concluded to offer the most meaningful results for this study; however, the reheater unit, because of its unique material and geometry, should be considered in a future effort.

4.2.3.1 SCOPE LIMITATIONS (continued)

(2) Duplex Tube Loading - Since reasonably accurate structural temperature estimates were available for only the nominal steady state operating condition, this was the only condition considered. Axial temperature variation along the length of the duplex tube is expected to be quite small away from the tube ends; consequently, the nominal steady state temperatures should apply over a large percentage of the tube length. Therefore, it can be concluded that during steady state operation the gross response of the tube is controlled by the average or nominal steady state loads. Since the failure modes of interest, i.e., violations of the functional operating requirements, are concerned with gross tube separation or sliding, consideration of only the nominal steady state loading condition is justified.

(3) Model Considerations - In order to simplify the required analysis several assumptions were made. First, the tubes were considered to be infinitely long; axial variations in both tube geometry and loading were neglected and end effects at the tubesheet and end caps were ignored. This assumption can be made with the same logic which was discussed in Item (2) above. Second, circumferential

4.2.3.1 SCOPE LIMITATIONS (continued)

variation in the tube geometry, in particular the presence of the leak detection grooves on the outer pressure tube, was neglected. This assumption is difficult to support and is made solely for analytical expedience. Although no supporting literature can be cited, it is believed that the consideration of these grooves would result in some reduction (perhaps 25%) in the maximum residual contact pressure and would also lead to significant strain accumulation at one or more of the grooves during fabrication.

These two assumptions allowed the evaporator duplex tube module to be modelled as a short length of two axisymmetric tubes in interacting, but distinct, conditions of generalized plane strain. The details of the modeling technique employed are discussed in the following section.

4.2.3.2 ANALYTICAL MODEL

Although a number of analytical solutions for the response of an elasto-plastic tube to internal pressure or expanding displacement have been presented in the literature (see Reference 4.2-24 for a bibliography), an independent solution by finite element analysis was used here, without

4.2.3.2 ANALYTICAL MODEL (continued)

reference to previous results, for several reasons. First, most of the previous solutions assume conditions of plane stress or plane strain in the tube; while this can be shown to have an insignificant effect on the radial stress (and residual contact pressure), it leads to widely varying results for axial stresses and could significantly influence the predicted tube sliding response. Second, and most importantly, the more theoretically sound formulations of the problem require solution by numerical techniques. The effort required to develop and certify a computer program to solve the proposed problem and perform the desired parameter study, when finite element programs certified for inelastic analysis of axisymmetric structures are available, cannot be justified.

The ANSYS finite element program, as described in Reference 4.2-25, was employed to model the evaporator duplex tube module. The model geometry, including pertinent dimensions, node numbers, and element numbers, is shown in Figure 4.2-26. All dimensions used were given previously in Section 3.1 of this report. Both the inner and outer pressure tubes are modelled with ten ANSYS STIF42 axisymmetric, isoparametric quadrilateral elements through the

Model Dimensions:

$$r_{1i} = 0.578''$$

$$r_{2i} = 0.645''$$

$$h = 0.010''$$

$$r_{1o} = 0.645''$$

$$r_{2o} = 0.725''$$

$$t_1 = 0.067''$$

$$t_2 = 0.080''$$

ANSYS FINITE ELEMENT MODEL

FIGURE 4.2-26

4.2.3.2 ANALYTICAL MODEL (continued)

thickness. Although the assumptions made in Section 4.2.3.1 eliminated dimensions along the tube axes from affecting the problem, a 0.01" tube length was used to meet the recommended aspect ratio limit for the STIF42 elements.

The interface between the inner and outer pressure tubes is represented by the two ANSYS STIF12 gap interface elements between nodes 11 and 12 and nodes 33 and 34. This element allows for the specification of an initial clearance or gap, δ , between the incident nodes so that until this gap is closed each node (in this application the two tubes) can deform independently. However, after closing the gap the STIF 12 element allows a normal force (F_N) and a tangential or shear force (F_S) to be transmitted between the incident nodes. Further, the shear force must satisfy an inequality of the form

$$F_S \leq \mu F_N \qquad \text{EQ \#4.2-105}$$

where μ is a positive real number less than 1.0. If Equation 4.2-105 cannot be satisfied at any point in the loading history, relative sliding is allowed until the shear force reaches an acceptable value. In this analysis the parameter μ was assigned a value of 0.4; the initial clearance or gap, δ , was varied

4.2.3.2 ANALYTICAL MODEL (continued)

from 0.000" to 0.040" to bound the anticipated range required to assemble the tubes.

The boundary conditions used in the finite element model can be described in terms of the degrees of freedom u_r and u_z as shown on Figure 4.2-26. The appropriate boundary conditions are

$$u_{z_i} = 0 \qquad i = 23 - 44 \qquad \text{EQ \#4.2-106}$$

$$u_{z_i} = u_{z_{11}} \qquad i = 1 - 10 \qquad \text{EQ \#4.2-107}$$

$$u_{z_i} = u_{z_{12}} \qquad i = 13 - 22 \qquad \text{EQ \#4.2-108}$$

where the notation u_{r_i} and u_{z_i} refers to the translations of the i^{th} node in the radial (r) and axial (z) directions, respectively. Equations 4.2-107 and 4.2-108 express the previously stated requirement that each tube be in a distinct condition of generalized plane strain, related by the action of the interface element connecting nodes 11 and 12.

The material properties required to perform the proposed analysis are Young's Modulus (E), the coefficient of thermal expansion (α), Poisson's Ratio (ν) and an elasto-plastic constitutive equation, all of which vary with temperature for most structural materials. Additionally, if the material exhibits creep phenomena at the stresses

4.2.3.2 ANALYTICAL MODEL (continued)

and temperature characteristic of the nominal steady state condition, a creep law is also necessary. Data on SANDVIK HT-9 was available from two sources: the SANDVIK HT-9 data sheet, Reference 4.2-26, and a general data tabulation for Class II 12% chrome steels which includes HT-9 in Reference 4.2-27. From Reference 4.2-26 the values of Young's Modulus, E , and the coefficient of thermal expansion, α , tabulated in Table 4.2-XVII were obtained. The value of Poisson's Ratio, ν , in Table 4.2-XVII was estimated from data in Reference 4.2-27.

Although adequate data was not available from both sources to develop a simple creep law for HT-9, the creep strength data presented in Reference 4.2-26 indicates that creep response would be insignificant at the temperatures and stresses anticipated in the evaporator duplex tubes. Consequently, creep response need not be considered in this analysis.

The elasto-plastic constitutive equation for HT-9 also is not readily available in the literature. Data for several other Class II 12% chromes from Reference 4.2-27 indicates a stress-strain curve with a marked knee from the yield strain to approximately 3% strain followed by continually increasing strain without increase in stress.

TABLE 4.2- XVII
MECHANICAL PROPERTIES FOR SANDVIK HT-9

Temperature, T(°F)	Young's Modulus E(ksi)	Coefficient of Thermal Expansion α (in/in/°F)	Poisson Ratio ν
68.	30.00	$5.32 \cdot 10^{-6}$	0.3
100.	29.79	$5.46 \cdot 10^{-6}$
200.	29.10	$5.77 \cdot 10^{-6}$	
300.	28.40	$6.01 \cdot 10^{-6}$	
400.	27.68	$6.19 \cdot 10^{-6}$	
500.	26.94	$6.34 \cdot 10^{-6}$	
600.	26.18	$6.48 \cdot 10^{-6}$	
700.	25.40	$6.61 \cdot 10^{-6}$	
800.	24.60	$6.74 \cdot 10^{-6}$	
900.	23.78	$6.85 \cdot 10^{-6}$	
1000.	22.94	$6.96 \cdot 10^{-6}$	0.3

4.2.3.2 ANALYTICAL MODEL (Continued)

Assuming HT-9 would demonstrate a similar response, the two bounding constitutive relations sketched in Figure 4.2-27 with the "exact" curve were employed. In Figure 4.2-27 Curve A demonstrates a perfectly plastic material response and Curve B is a strain hardening response. Curve A is defined by the average material ultimate strength σ_1 , while Curve B is defined by the average yield strength, σ_0 , and the elasto-plastic modulus, E_p . Values of σ_0 were estimated from the minimum yield stresses given in Reference 4.2-27 and the characteristic yield stress scatter for Class II 12% chromes given in Reference 4.2-27. The average ultimate strength, σ_1 , was estimated by assuming the ratio σ_1/σ_0 to be constant for all temperatures. Finally, the plastic slope at room temperature, E_p , was estimated from the elongation data in Reference 4.2-26 and extrapolated to higher temperatures by assuming the ratio E_p/E to be independent of temperature. The values of E and σ_0 and their relationship to E_p and σ_1 are also included on Figure 4.2-27.

The finite element model described above was subjected to a loading history which represented the tube expanding process followed by a loading to the nominal steady

Curve A: Perfect Plasticity

Curve B: Strain Hardening

T(°F)	E(ksi)	σ_0 (ksi)
68	30000.	78.9
212	-	69.4
392	-	66.4
572	-	63.3
752	25000.	56.7
932	-	47.2
1032	-	41.1
1112	22000.	-

$$\sigma_1 = 1.5 \sigma_0$$

$$E_p = 0.6 E$$

CONSTITUTIVE EQUATIONS

FIGURE 4.2-27

4.2.3.2 ANALYTICAL MODEL (continued)

state condition. The tube expansion process was modeled as an imposed radial displacement u_0 on nodes 1 and 23, so that

$$u_{r1} = u_{r23} = u_0$$

Once the desired value of u_0 was attained the specified radial displacements on nodes 1 and 23 were deleted and the tubes unloaded to a residual stress field. The residual normal forces in the two interface elements then defined the residual contact pressures.

Once the residual contact pressure developed by the tube expanding process was established, the residual steady state contact pressure was computed by applying the nominal steady state loading shown in Figure 4.2-28. During the application of these loads the relative sliding response of the two tubes was also predicted.

Since both unloading from the expanded displacement u_0 to the residual stress field and loading to the nominal steady state condition occur elastically, it is possible to calculate the change in residual contact pressure during these phases of the load history from the equations of elasticity. However, the contact pressure at the fully expanded position must be calculated inelastically. A large

Pressures:

$$P_{H_2O} = 2650 \text{ psi}$$

$$P_{He} = 300 \text{ psi}$$

$$P_{Na} = 95 \text{ psi}$$

Temperatures:

$$T_{1i} = 664.0^{\circ}\text{F}$$

$$T_{1o} = 697.6^{\circ}\text{F}$$

$$T_{2i} = 711.1^{\circ}\text{F}$$

$$T_{2o} = 746.9^{\circ}\text{F}$$

NOMINAL STEADY STATE LOADING

FIGURE 4.2-28

4.2.3.2 ANALYTICAL MODEL (continued)

portion of the results presented in the following section were obtained by superimposing the contact pressure at the fully expanded position as computed with the finite element model and hand calculated estimates of the elastic unloading and loading to steady state. The hand calculated results are always clearly identified.

4.2.3.3 RESULTS, CONCLUSIONS AND RECOMMENDATIONS

Results for tube contact pressures were obtained for four distinct cases as defined below:

- 1) Both tubes behave perfectly plastically as shown by Curve A on Figure 4.2-27; no initial radial gap between tubes, $\delta = 0.000$ ";
- 2) Same as Case (1) except $\delta = 0.020$ ";
- 3) Same as Case (1) except $\delta = 0.040$ ";
- 4) The inner pressure tube responds perfectly plastically as shown by Curve A on Figure 4.2-27 and the outer pressure tube strain hardens as shown by Curve B on Figure 4.2-27; the initial radial gap is $\delta = 0.020$ ".

Cases 1), 2) and 3) were investigated to determine the effect of the gap size, δ , on the residual contact pressure. The selected gap sizes should bound the actual

value; a twenty mil gap was considered to be the most probable required value. From the results of these cases it was concluded that the development of a positive residual pressure with any reasonable gap size would require large strains in the inner pressure tube and moderate strains (.4%) in the outer pressure tube. While the perfectly plastic material response assumed in these cases gives a reasonable approximation of the material behavior for large strains, it is a poor representation for strains of the magnitude expected in the outer pressure tube. Consequently, Case 4) was formulated to improve the estimate of the stress-strain response of the outer tube with the most likely gap size. These results are displayed graphically in Figures 4.2-29 and 4.2-30. For each case the residual and steady state contact pressures are plotted against the expanding displacement u_0 . A second abscissa, the equivalent strain at the inside surface of the outer tube, is also presented in each figure.

As these figures indicate, residual contact pressures of 1.25 ksi and steady state residual contact pressures of 1.76 ksi can be attained in the evaporator unit irrespective of material behavior or initial gap size. However,

4.2.3.3 RESULTS, CONCLUSIONS AND RECOMMENDATIONS (continued)

the presence of strain hardening behavior in the outer tube and reasonably large gap sizes, such as Case (4), requires large amounts of strain at the inside surface of the outer tube to form a joint. This could be a significant problem, since the leak detection grooves on the outer tube will cause severe strain concentrations. Proper preworking of the tubes before assembly of the duplex tube unit can be used to minimize this problem.

A second result, not covered by the graphical data, was that the expansion duplex tube forming process set up a condition of incipient sliding between tubes. That is, after removal of the fabrication loads the tube interface is loaded such that relative upward sliding of the inner tube is imminent. Sliding was then observed to occur throughout the heatup. Further sliding would not be expected upon unloading or reloading to this condition but could occur during transients. Because of the obvious inadequacy of the model used here to treat this problem no effort was made to continue the investigation of tube sliding; however, further consideration of the problem is recommended.

EFFECT OF EXPANDING DISPLACEMENT, U_o
ON THE CONTACT PRESSURE FOR CASES 1, 2, AND 3

FIGURE 4.2-29

EFFECT OF EXPANDING DISPLACEMENT, U_o
ON THE CONTACT PRESSURE FOR CASE 4

FIGURE 4.2-30

4.2.3.3 RESULTS, CONCLUSIONS AND RECOMMENDATIONS (continued)

In summary, the following conclusions can be drawn regarding the residual contact pressures in a duplex tube for the C-E reference design evaporator unit:

1. With proper definition of the tube expansion process residual contact pressures of 1.25 ksi can be achieved and residual steady state contact pressures of 1.76 ksi are possible; consequently, tube separation during operation can be avoided.
2. Care must be taken to properly pre-work both tubes before duplex tube assembly to minimize the amount of plastic straining required in either tube to form a positive joint. Generally, the outer tube should be fully hardened and the inner tube should be annealed.
3. Relative tube sliding is indicated during heatup to steady state. Sliding during cooldown and reheatup is not expected; sliding in transients from steady state cannot be predicted with this analysis.

Because of the scope limitations and model simplifications which were made to expedite this study and

4.2.3.3 RESULTS, CONCLUSIONS AND RECOMMENDATIONS (continued)

because of the data development problems with HT-9 several recommendations for future study into this problem have been made; namely,

1. A more extensive parameter study, including geometric details and material behaviors characteristic of the reheater unit, is required.
2. An investigation of the strain concentration problem at the leak detection grooves on the outer pressure tube, including its effect on the residual contact pressure, is needed.
3. Although it is not considered to be serious, the tube sliding problem indicated in this analysis should be examined with a more sophisticated model to eliminate any uncertainty.

REFERENCES FOR SECTION 4.2

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- 4.2-4 ASME Boiler and Pressure Vessel Code, 1974 Code Cases 1592-1596.
- 4.2-5 Technical Work Request, "FBR Large Plant - Heat Exchanger Tube Bundle Sizes", FBR-74-218.
- 4.2-6 "Formulas for Stress and Strain", by R. J. Roark, p. 318, third edition, 1954.
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REFERENCES FOR SECTION 4.2 (continued)

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- 4.2-18 "Formulas for Stress and Strain", by R. J. Roark, Third Edition, 1954, pp. 268.
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4.3 MANUFACTURING CONSIDERATIONS

There are several major manufacturing considerations which greatly influence the practicality of building a duplex bayonet tube steam generator component. Among these are the physical process of "duplexing" the tubes, and producing the required high integrity tube to tubesheet welds. The following paragraphs are addressed to providing in-depth discussions of these areas.

4.3.1 DUPLEXING METHODS

As pointed out in previous paragraphs, the thermal performance of the steam generator will be heavily influenced by the effectiveness of the method selected for fabricating the duplex tubes. Subsequent discussion (Section 4.4) shows that the duplexing method is also important in establishing effective leak detection. In the ideal case, good metal-to-metal contact should be achieved over a portion of the interfacial area (as a result of high interface pressure) for the heat transfer capability with a small interconnected gap to provide for leak detection. The duplexing method employed should provide fairly high contact pressure at the interface and the interface should have a controlled degree of roughness.

In addition, in order to meet the design objective of providing full penetration, high integrity welds throughout the component, the welds at the outer tube to end cap joint and at the outer tube to

4.3.1 DUPLEXING METHODS (continued)

tubesheet joint must be made and inspected prior to installing the inner tube. Pre-assembly of the two tubes will not permit this objective to be met; therefore, the duplexing method employed should lend itself to in situ assembly of the tubes within the component.

The methods available for fabricating double walled tubes categorically fall into two approaches or a combination of the two. These are, mechanically reducing the outer tube onto the inner tube and mechanically expanding the inner tube into the outer tube. The techniques available within the category of reducing the outer tube onto the inner tube generally do not lend themselves to in situ applications as desired because of the close proximity of tubes within the tube bundle and the "eggcrate" tube supports. Thus, those methods will be considered subsequently as possible alternatives to in situ duplexing methods. Among the methods potentially available for expanding the inner tube into the outer tube are ball sizing, thermally shrinking, hydraulic forming, explosive forming and continuous rolling.

Ball sizing entails pushing or pulling a shaped tool of the required size through the entire length of the inner tube. The size of the tool would be such that a predetermined interference would exist between

4.3.1 DUPLEXING METHODS (continued)

the tool and the inner tube wall so that metal would be physically displaced as the tool is moved through the tube. Inherent to this procedure are two phenomena which make it unsuitable for this application. These are as follows: because of the magnitude of the forces involved, the tool would have to be pulled through the tube requiring either a "collapsible" tool or an open ended tube; and, the continuously increasing "wave" of metal which would be pushed axially ahead of the tool would preclude moving the tool through such lengths of tubing.

Thermally shrinking the tubes together would involve, of course, assembling final sized tubes by inserting a chilled inner tube into a heated outer tube and allowing the respective tubes to reach an isothermal state where interfacial contact would be achieved. Assuming the inner tube was chilled to 0°F and the outer tube was heated to 1000°F, the tube diameter differentials would be an order of magnitude less than the best diametrical tolerances achievable and two orders of magnitude less than the clearance required to practically insert the inner tube into the outer tube. This approach is certainly not feasible for this application.

Hydraulic pressure could be used to expand the inner tube into contact with the outer tube; however, tube wall thickness tolerances

4.3.1 DUPLEXING METHODS (continued)

would permit variations in interfacial contact pressure, and final sizes (both inside diameter and outside diameter) would not necessarily be consistent or repeatable.

Explosive forming has been successfully used in numerous heat exchangers for expanding tubes into tubesheets. The process consists of fitting a semi-rigid plastic case containing an explosive charge (Primacord for example) into the tube and detonating the explosive charge. A high degree of exactness in sizing the explosive charge is not needed for expanding tubes into tubesheet using this technique; however, a tubesheet can be considered as providing an infinitely rigid backup for the tube whereas the relative thin outer wall in this application cannot. Thus, explosive forming would produce somewhat similar results to hydraulic forming in that tube diameters and contact pressures would be inconsistent.

Continuous rolling (as opposed to step rolling) appears to be the most likely candidate for performing the desired in situ tube duplexing operation. This technique differs from hydraulic and explosive forming in that the finished dimensions are controlled by the tool size. It also differs from the ball sizing process in that the axial "wave" of material will constitute much less volume and result in only a slight "axial

4.3.1 DUPLEXING METHODS (continued)

extrusion" in lieu of a possible tool seizure. The process would employ a self feeding torque shaft driven tool, similar to those currently in use within the industry for continuous rolling tubes into tubesheets, to roll the inner tube between the tube to tubesheet weld to an elevation just above the tube/end cap weld (See Detail E of Drawing E-63675022 for graphic reference). The duplexing operation would be done after all tube to tubesheet and tube to end cap welds had been made and inspected. The direction of rolling would be from top to bottom so that the axial extrusion could be accommodated in the end cap region. This is the same procedure used to successfully fabricate the HNPF steam generators. It is the general consensus among the foremost manufacturers of such rolling tools* that this procedure is within the present day state-of-the-art and that post rolled inside diameter tolerances and surface finishes would be superior to those for "as purchased" tubing. This is important to assure correct support of the bayonet/insulator tube assembly which is subsequently installed inside the duplex tube. The outer diameter of the outer pressure tube could be slightly enlarged, depending upon tolerance buildup, as a result of the rolling operation. This would, however, have no adverse effects on the component.

*Airetool Division of Dresser Industries and Wiedeke Plant of Elliot Company

4.3.1 DUPLEXING METHODS (continued)

To assure unhampered insertion of the inner tube into the outer tube, a positive installation clearance of .020 to .040 inch on the diameter must be provided. (Such factors as tube out of roundness, tube diameter tolerances on the inside diameter of the inner tube and on the outside diameter of the outer tube, tube straightness, etc. will determine the required installation clearance; however, 0.040 inch is felt to be a conservative upper limit. This point is pending verification. (See Section 6.0). The installation clearance would be plastically closed and the outer tube will be elastically enlarged during the rolling operation (See paragraph 4.2.1).

Even though the full penetration weld design objective mentioned above cannot be met using duplex tubes which are assembled external to the component, the methods associated with pre-assembled tubes were reviewed as an alternate approach to in situ expansion (continuous rolling) of the inner tube into the outer tube. The fabrication of double walled tubes using a tubing draw bench is relatively easy to accomplish using present day state-of-the-art techniques and equipment. A sufficiently oversized outer tube is placed over the inner tube and the assembly is hot drawn through an externally fitting die. (This procedure might also include either a previous,

4.3.1 DUPLEXING METHODS (continued)

simultaneous, or subsequent pass over an internal mandril). This is the procedure employed in fabricating tubes for the EBR-II steam generator and in fabricating samples for the Westinghouse duplex tube bench tests. Figure 4.3-1 depicts a typical tube-to-tubesheet joint which would be considered for use with pre-assembled tubes. As pointed out in this Figure partial penetration tube to tubesheet welds are unavoidable at the lower tubesheet. A full penetration weld could conceivably be provided at the tube to end cap weld; however, it would have to be made prior to the duplexing operation and would preclude the use of an internally fitting mandril and possibly not be compatible with the drawing operation.

FIGURE 4.3-1
TYPICAL TUBE-TO-TUBESHEET JOINT COMBINATION FOR USE
WITH PRE-ASSEMBLED BAYONET TUBES

4.3.2 TUBE-TO-TUBESHEET WELD DESIGNS

The configuration of the tube-to-tubesheet joint within any shell and tube heat exchanger must be carefully considered because of the undesirable thermal and structural reactions which are prone to occur between the relatively thin tube wall and the relatively thick tubesheet. A major design objective set down for the duplex bayonet tube steam generator is to provide full penetration (crevice free) welds, the configurations of which must lend themselves to detailed nondestructive examination, throughout each component. Drawings E-63675022, E-63675023 and E-63675033 (See Section 3.0) reflect such full penetration welds at both tubesheets for the evaporator, superheater and reheater units respectively. These welds along with others of both the full and partial penetration type, are discussed in detail in the following paragraphs.

4.3.2.1 FULL PENETRATION WELDS AT BOTH THE SODIUM/GAS TUBESHEET AND AT THE STEAM/GAS TUBESHEET

Figure 4.3-2 depicts the full penetration welds incorporated into the designs presented in Section 3.0. The lowermost or outer tube-to-sodium/gas tubesheet weld is a joint developed previously by Combustion Engineering under AEC Contract (AT 11-1)-3031 (Ref. 4.3-1). The weld is completely free of crevices and provides a smooth transition contour from the thin tube wall to the thick tubesheet. It is a multipass weld

FIGURE 4.3-2
FULL PENETRATION WELDS AS SHOWN
ON DUPLEX BAYONET TUBE STEAM
GENERATOR DESIGN

4.3.2.1 FULL PENETRATION WELDS AT BOTH THE SODIUM/ GAS TUBESHEET AND AT THE STEAM/GAS TUBESHEET (continued)

accomplished from inside the tube using a specially developed gas tungsten arc torch with a continuous filler wire feed. Figure 4.3-3 shows the tube end and tubesheet configuration prior to welding and the finished tube-to-tubesheet weld. The tube end is mechanically upset to provide a reinforced section at the weld joint.

The uppermost or inner tube-to-steam/gas tubesheet weld is the DeSchelde type corner weld featured in numerous European units (the SNR steam generators as an example). Figure 4.3-4 shows the tube and tubesheet prior to welding and the finished weld. This process uses no filler metal since the extended end of the tube provides the necessary material to accomplish the weld. This is one of the four basic joint designs which were considered during the previously mentioned AEC contracted tube-to-tubesheet joint development program.

As an alternate to the corner weld, a weld of the type shown in Figure 4.3-3 could be employed at the inner tube-to-steam/gas tubesheet; however, this would complicate assembly of the unit since it would require installing the steam/gas tubesheet after positioning the inner tubes in the steam generator. Should the corner weld be shown to be undesirable due to structural consideration of axial tube loading or for some other reason, this alternate would provide an excellent "fall-back" even at the expense of a more complicated assembly sequence. Figure 4.3-5 shows this alternate combination of tube-to-tubesheet welds.

FIGURE 4.3-3
FULL PENETRATION OUTER TUBE-TO-SODIUM/GAS TUBESHEET
(THE TYPE DEVELOPED BY C-E UNDER
AEC CONTRACT AT (11-1)-3031)

FIGURE 4.3-4
FULL PENETRATION INNER TUBE-TO-STEAM/GAS TUBESHEET
(DeSCHelde CORNER WELD TYPE)

Note: This approach requires installation of the inner pressure tube prior to installing the steam/gas tubesheet. Thus, the manufacturing sequence would be more complicated.

FIGURE 4.3-5
ALTERNATE FULL PENETRATION WELD COMBINATION
FOR DUPLEX BAYONET TUBE
STEAM GENERATOR

4.3.2.1 FULL PENETRATION WELDS AT BOTH THE SODIUM/ GAS TUBESHEET AND AT THE STEAM/GAS TUBESHEET (continued)

All of these approaches are compatible with present day state-of-the-art technology for nondestructive examination techniques using radiography, ultrasonic examination, penetrant testing, magnetic particle examination, and eddy current testing.

4.3.2.2 PARTIAL PENETRATION WELDS AT ONE OR BOTH TUBESHEETS

Should unforeseen problems of an insurmountable magnitude arise relative to the in situ rolling of the inner pressure tube into the outer pressure tube (See Section 4.3.1), the achievement of full penetration welds at both tubesheets would not be possible. Figure 4.3-6a, 6b and 6c depict typical combinations of partial and full penetration welds which would be compatible with pre-assembled pressure tubes. The configuration shown in Figure 4.3-6a, which has partial penetration welds at both tubesheets is of the type utilized in the Westinghouse duplex tube bench tests. A full penetration (DeSchelde corner weld type) weld could be used with the same partial penetration weld arrangement as shown in Figure 4.3-6b. In addition, the reinforced internal bore weld (as developed by C-E under the previously mentioned AEC contract) might possibly be used in conjunction with the partial penetration sodium/gas tube-to-tubesheet weld as shown in Figure 4.3-6c.

FIGURE 4.3-6a
PARTIAL PENETRATION WELDS AT BOTH TUBESHEETS

FIGURE 4.3-6b
PARTIAL PENETRATION WELD AT SODIUM/GAS
TUBESHEET

Note: This approach requires installation of the inner pressure tube prior to installing the steam/gas tubesheet. Thus, the manufacturing sequence would be more complicated.

FIGURE 4.3-6c
PARTIAL PENETRATION WELD AT SODIUM/GAS
TUBESHEET

4.3.2.2 PARTIAL PENETRATION WELDS AT ONE OR BOTH TUBESHEETS (continued)

All configurations which feature partial penetration tube-to-tubesheet welds compromise the design objective of providing full penetration welds throughout the unit.

REFERENCES FOR SECTION 4.3

- 4.3-1 Interim Report - "Tube-to-Tubesheet Welding Development Program for LMFBR Steam Generator Development", Combustion Engineering, Inc., AEC Contract Number AT (11-1) - 3031, December 20, 1972.

4.4 SAFETY AND RELIABILITY CONSIDERATIONS

The safety and reliability aspects of the Duplex Bayonet Tube steam generator components were studied by: (1) evaluating the behavior and detection of both water and sodium side leaks; and (2) performing a failure mode and equalitative fault tree analysis. Subsequent paragraphs outline the respective analyses performed and the conclusions drawn from them.

4.4.1 LEAK DETECTION/BEHAVIOR EVALUATION

The duplex tube consists of two tubes in contact with each other with grooves placed between the tubes to facilitate leak detection. Leaks at various locations in the duplex tube were considered using both laminar flow and diffusion. It is desirable to detect a single leak in a reasonable time so that corrective actions can be taken and a sodium-water reaction prevented. This could lead to simplification of the plant by eliminating sodium water reaction equipment.

4.4.1.1 LEAK DETECTION MODELS AND LOCATIONS

The duplex tube being considered consists of two tubes in contact with each other. Four grooves are placed between them for leak detection (See Figures 4.4-1 and 4.4-2). Dimensions are summarized in Table 4.4-I. The two tubes are pressed together to ensure good thermal conductivity. Because of irregularities in the tube surface, the tubes are assumed to contact each other at 5% of the area. For the remaining 95% of the area, a gap of 0.25 mil is assumed to exist. Four grooves are cut into the inside of the outer tube. These grooves are 90° apart, have a 20 mil radius, and run the length of the tube. The grooves are filled with helium and connected to a helium plenum. The tubes are 45 feet long and are connected to tubesheets 9.84 inches thick. The region between the tubesheets creates a helium plenum to which the grooves are connected. Sodium surrounds the duplex tube while steam is in the center. The properties of the fluids in the three

FIGURE 4.4-1
CROSS SECTIONAL VIEW OF DUPLEX TUBE

FIGURE 4.4-2

TABLE 4.4-1
DIMENSIONS USED IN LEAKAGE STUDY

Tubes

Length	576 in.
Tubesheet Thickness	9.84 in.
Temperature	788 ⁰ F

Inner Tube

I. D.	1.0551 in.
O. D.	1.2913 in.

Outer Tube

I. D.	1.2918 in.
O. D.	1.4488 in.

Groove

Location	90 ⁰ Apart
Radius	.020 in.
Area	6.28 x 10 ⁻⁴ in ²
Equi. Dia.	.0283 in.
Volume	.362 in. ³
Endfitting Volume	.653 in. ³

TABLE 4.4-I (continued)

Gap

Thickness	.25 mil
Contact (%)	5%
Circumference	3.795 in.

Leak Between Grooves

Height	.394 in.
Effective Height	.374 in.
Width	2.48×10^{-4} in.
Length	.992 in.

4.4.1.1 LEAK DETECTION MODELS AND LOCATIONS (continued)

regions are given in Table 4.4-II. These conditions apply to the evaporator but results will in general apply to the reheater and superheater as well.

Leaks are considered in both the inner tube (steam side) and outer tube (sodium side). Two axial locations are considered. The leak was assumed to occur at the weld at the bottom of the lower tubesheet which is 9.84 inches from the helium plenum (Location B, Figure 4.4-2). Leaks at the bottom of the tube (45 feet from plenum) were also considered (Location A, Figure 4.4-2). The leaks were circumferentially located at a groove or midway between two grooves. These assumptions lead to upper and lower limits on the location of the leaks. Two different helium pressures were considered and the leak directions considered were: (1) steam into helium; (2) sodium into helium (low He pressure) and (3) helium into sodium (high He pressure).

The two types of leak flow regimes considered were laminar and diffusion. Molecular flow was neglected since the assumed gap between the two tubes was significantly larger than the mean free path length of the molecules involved. Helium has a mean free path length of 2.75×10^{-6} inches while the duplex tube gap width is 2.50×10^{-4} inches. This distance is sufficiently large to ensure laminar flow and diffusion.

TABLE 4.4-II
FLUID PROPERTIES USED IN LEAKAGE STUDY

	Regions		
	Sodium	He	Steam
Temperature, °F**	878	878	878
Viscosity, lb-sec/in ²	3.68×10^{-8}	4.93×10^{-9}	3.96×10^{-9}
Molecular Wt., lbm/mole	5.07×10^{-2}	8.82×10^{-3}	3.97×10^{-2}
Pressure, atm	6.5	4.4* 21.4	181.3
Inventory, lb-moles	1.014×10^4	3.47×10^{-2} 4.12×10^{-1}	-
Diffusion Coefficient, in ² /sec (4.4 atm He pressure)	0.0794	-	0.1679
Diffusion Coefficient, in ² /sec (21.4 atm He pressure)	0.0132	-	0.0288

* Low pressure operation

**Na entrance temperature used

4.4.1.1 LEAK DETECTION MODELS AND LOCATIONS (continued)

The leaks located at the grooves were assumed sufficiently large so that they did not interfere with the flow. This means that the leakage rate was determined by the size of the groove and the distance from the leak to the helium plenum and not the size of the hole in the tube. This represents upper limit leak rates. The leaks between the grooves, however, were assumed to be cracks .394 inch high with a width the same as the duplex tube gap width (.25 mils). Here the leak rates are determined by the hydraulic resistance of the gap between the two tubes. The crack length of .394 inch was selected for convenience and the leak rates are approximately proportional to the assumed length of the crack.

4.4.1.2 LEAK FLOW BY DIFFUSION

Diffusion of steam and sodium vapor into the helium plenum was considered. The diffusion process is the movement of a chemical species from a region of high concentration to a region of low concentration. It is assumed that liquid sodium or steam leaks into a groove at a flow rate so small that it only wets the groove at the leak but does not flow up the groove. Diffusion of the material from this point source to the helium plenum is considered.

4.4.1.2 LEAK FLOW BY DIFFUSION (continued)

Diffusion is described by Fick's law which states that a chemical species A diffuses in the direction of decreasing concentration of A in material B. Fick's law can be written for a one-dimensional system, as

$$J_{AY} = -D_{fAB} \frac{d(\rho_A)}{dY} \quad \text{EQ \#4.4-1}$$

where

J_{AY} = mass flux of A in the Y direction (lbm/in²-sec)

D_{fAB} = diffusivity of chemical specie A in chemical specie B (in²/sec)

ρ_A = density or concentration of A (lbm/in³)

Y = length along groove (in)

The density gradient was approximated by the density of the leaking material (Material A) divided by the distance from the leak to the helium plenum. The leakage rate is calculated from the equation:

$$LR = \frac{A_r D_{fAHe} \rho_A}{D M_n A_{He}} \quad \text{EQ \#4.4-2}$$

where

LR = leakage rate, ppm/s

A_r = groove area, in²

L = distance from leak to plenum, in

ρ_A = density of species A at leak, lbm/in³

n_{He} = molecules inventory of helium, lbm mole

M_A = molecular weight of Material A, lbm/lb mole

4.4.1.2 LEAK FLOW BY DIFFUSION (continued)

$D_{A\text{ He}}$ = diffusion coefficient of A in Helium, in^2/sec

subscript

A = sodium or steam

He = helium

This equation was solved for various conditions.

The lengths (L) were selected to represent the maximum and minimum lengths possible for a groove. The material properties are given in Table 4.4-II. The diffusion coefficients were calculated, i.e., from theory (reference 4.4-1) and reported in Section 4.3, Appendix A - Aluminum Matrix Leak Behavior. The coefficients are similar in size to those given in Reference 4.4-1 for other materials. The density of the sodium used in Equation 4.4-2 is not that of liquid sodium but that of saturated sodium vapor at the liquid surface. The saturation pressure used was 2.73×10^{-3} atm for sodium at 890°F . This is the maximum value for the system and gives the largest diffusion rates. The vapor pressure at the evaporator outlet is only 1.34×10^{-4} atm, 5% of the pressure and therefore density used. The sodium vapor density used was $3.72 \times 10^{-8} \text{ lbm/in}^3$.

The leakage rates from diffusion are shown in Table 4.4-III. This table shows that detection of such leaks would be difficult if it is assumed that concentrations of 1PPM would be required for

TABLE 4.4-III
LEAK RATES DUE TO DIFFUSION (PPM/S)

	He at 21.4 atm pressure		He at 4.4 atm pressure	
	Bottom Leak	Top Leak	Bottom Leak	Top Leak
Sodium	5.58×10^{-11}	3.21×10^{-9}	1.65×10^{-9}	9.51×10^{-8}
Steam	4.68×10^{-8}	2.70×10^{-6}	1.33×10^{-6}	7.65×10^{-5}

4.4.1.2 LEAK FLOW BY DIFFUSION (continued)

detection (an easily satisfied detection requirement). The fastest leak shown on this table would require 15 days before concentrations were detectable. This means that the material must be carried by other means to the upper plenum to be detected, if very small leaks are to be detected. By varying the pressure in the helium plenum, the material might be pumped to the plenum (See Paragraph 4.4.1.4). Detection of sodium would still depend on the vapor pressure, liquid surface area created and diffusion from the saturated layer of gas at the surface. Equation 4.4-2 can still be used.

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW

When a leak in one of the tubes becomes sufficiently large, material will start to move up or down the grooves by laminar flow. This type of flow depends on the total pressure gradient. Leaks located both in a groove and midway between two grooves were considered. These are discussed respectively in subsequent paragraphs.

(A) Leaks occurring at a groove location

For an upper limit calculation, a hole was assumed to exist at one of the grooves and the hole was large in area compared to the cross sectional area of the groove. Flow was limited by the hydraulic resistance of the groove. The groove was approximated by a round hole of equivalent area and the volumetric flow rate was calculated by the following equation as taken from Reference 4.4-1:

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW (continued)

$$q_2 = \frac{\pi(P_1 - P_2) \bar{P} D^4}{128 \mu L_1 P_2} \quad \text{EQ \#4.4-3}$$

where

q = volumetric flow rate, in³/sec

P = pressure, psia

$\bar{P}$ = average pressure, psia

D = diameter, in

μ = viscosity, lb-sec/in²

L_1 = distance from leak to groove, in

Subscripts

1 = high pressure end

2 = low pressure end

For leaks at the bottom of the duplex tube, it is assumed that the hole is sufficiently large to immediately fill the region in the endfitting between the two tubes so that fluid flow is permitted in all 4 grooves. For this case, the volumetric flow rates are calculated by multiplying Equation 4.4-3 by a factor of four. When the leak is at the top of the duplex tube, the hydraulic resistance of the flow path to the endfitting and back up is sufficiently large that this flow can be neglected.

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW (continued)

Leakage rate in PPM/S is calculated from

$$LR = \frac{q P_2}{RT n_2} \times 10^6$$

EQ #4.4-4

where

LR = concentration rise at pt. 2, PPM/S

R = general gas constant = 1.8544×10^4 psia - in³/lbmole - °R

T = temperature, °R

n_2 = inventory of material at pt. 2, lb-mole

P_2 = pressure at pt. 2, lb/in²

Equation 4.4-4 can be applied to the steam flowing into the helium or the helium flowing into the sodium but not the liquid sodium flowing into the helium region (low pressure He conditions).

The results are given in Table 4.4-IV (labeled at groove) and show that the steam is easily detectable. When the helium is under high pressure, detection of sodium side leaks will take from 16 seconds to 4.2 minutes. These times are longer than the times to detect steam (approximately 1 second) because of the smaller pressure differences between the two regions and the larger sodium inventory (1.014×10^4 lb-moles). Detection times are based on a 1PPM concentration and neglect detector delay times.

TABLE 4.4-IV
 DUPLEX BAYONET TUBE STEAM GENERATOR
 LEAK RATES AND DETECTION TIMES

Detection Parameter	Leak Distance to Plenum	Leak Location A = At Groove B = Between Grooves	High Pressure Helium PPM/S	Low Pressure Helium	Detection ** Time Sec.
Steam in Helium	10 in. 45 ft. 10 in. 45 ft.	A A B B	135,000 9,200 204 176	670,000 PPM/S 44,000 PPM/S 1,975 PPM/S 186 PPM/S	less than 1 less than 1 less than 1 less than 1
Sodium in Helium	10 in. 45 ft. 10 in. 45 ft.	A A B B	(Diffusion) + 10^{-9} 10^{-11} 10^{-9} 10^{-11}	(1,586 in ³ /S) (0.109 in ³ /S) (5.18 x 10 ⁻⁴ in ³ /S) (5.06 x 10 ⁻⁴ in ³ /S)	less than 1 19 4050 4150
Helium in Sodium	10 in. 45 ft. 10 in. 45 ft.	A A B B	0.06 4×10^{-3} 1.18×10^{-5} 1.17×10^{-5}	- - - -	16 250 10^5 10^5

*Time to Fill Groove

**Excluding Detector Delay Time and Assuming 1PPM is Detectable

+Detection Times are Long, I. E., $\geq$ Months

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW (continued)

When the helium pressure is less than the sodium pressure, the liquid sodium flows into the grooves. If the leak is at the bottom of the tube, the sodium flows into all four grooves (and end fitting) and fills the grooves in 19 seconds. At this point, detection of the liquid sodium in helium plenum should be possible because of the increased liquid sodium surface area and short distance to the sodium vapor detector. If and when the helium plenum becomes saturated with sodium vapor, the concentrations would vary from 616 ppm at 890°F to possibly 30 ppm at 680°F. The time it takes to reach saturation (or to reach detectable levels) would depend on the amount of He turbulence, surface area of liquid and location of the detector, and would be of the order of minutes.

(B) Leaks Occurring Between Groove Locations

Leakage caused by a crack in a tube between two of the grooves was analyzed. The crack is assumed to be 1 cm long and has a width equal to that of the helium gap (.25 mil). The analysis assumed that the leak flow through the crack was associated with a single groove only (See Figure 4.4-3). The appropriate equations were written for flow between the helium plenum (Point 1) and the coolant (either Na or H₂O, Point 3). Equation 4.4-3 represents the flow between Point 1 and Point 2. The flow between Point 2 and 3 can be written (Reference 4.4-1):

FIGURE 4.4-3

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW (continued)

$$q_2 = \frac{HG^3 (P_2 - P_3) \bar{P}}{12\mu L_2 P_2} \quad \text{EQ \#4.4-5}$$

where

H = slot height, in.

G = slot thickness, in.

L_2 = slot length, in.

P = pressure, psia

μ = viscosity, lb-sec/in²

$\bar{P}$ = average pressure, psia

This combined with Equation 4.4-3 to solve for q_2

thus eliminating P_2 :

$$q_2 = \frac{k_1 k_2 (P_1 - P_3)}{k_2 P_3 + k_1 P_1} \quad \text{in}^3/\text{sec} \quad \text{EQ \#4.4-6}$$

where

$$k_1 = \frac{D^4 \bar{P}}{128\mu L_1} \quad \text{in}^3/\text{sec}$$

$$k_2 = \frac{HG^3 \bar{P}}{12\mu L_2} \quad \text{in}^3/\text{sec}$$

The leakage rates in ppm/s can be calculated

from Equation 4.4-4. This equation applies whether P_1 is greater than or less than P_3 .

4.4.1.3 LEAKS PRODUCING LAMINAR FLOW (continued)

As in a previous case, leakage rates were calculated for leaks at the bottom and top of the tubes. The flow rates and detection times are given in Table 4.4-IV under B, "Leak Location Between Grooves". The results show that the leakage rate is determined by the hydraulic resistance of the gap and crack. The distance from the leak to the plenum and the effects of the groove on the leakage rates are less important. The times necessary for the liquid sodium to fill the grooves are given for the low pressure helium system. These times are large (approximately 1 hour) because it was assumed that the sodium had to fill the end fitting and other 3 grooves. The time to detect helium in sodium is about 30 hours because of the large sodium inventory. As in the earlier cases, detection of steam leaks appears to be easy.

The gap width strongly effects the leakage rate and ability for detection. This gap depends on the surface finish of the tubes and how hard the outer tube is swaged onto the inner one. If the gap was twice as large as used (.50 mil), liquid sodium flow would increase by a factor of 8. Smaller cracks would likewise decrease flow.

4.4.1.4 LEAK DETECTION BY PLENUM PRESSURE CYCLING

It was shown that detection from diffusion (Paragraph 4.4.1.2) is very difficult. Detection could be greatly enhanced by cycling the helium pressure and forcing helium exchange between the grooves and plenum. This would increase diffusion by increasing the

4.4.1.4 LEAK DETECTION BY PLENUM PRESSURE CYCLING (continued)

concentration gradients ($\delta\rho/\delta r$, Equation 4.4-1) or by moving droplets of liquid sodium and sodium vapor to the helium plenum. This exchange of helium is based on forcing helium into the space in the duplex endfitting. Since the volume of the grooves is 1.46 in.³, the volume of the endfitting should be designed to ensure full exchanges. This volume is determined by:

$$V_{\text{endfitting, in}^3} = 1.46 (P_{\text{high}}/P_{\text{low}} - 1) \quad \text{EQ \#4.4-7}$$

This is derived simply from the ideal gas law.

If the pressure was varied from 4.4 atm to 5.65 atm ($\Delta P = 1.25$ atm), an endfitting volume of 5.19 in.³ would be required. The outer tube would have to be 25.2 in. longer than the inner one.

4.4.1.5 COMPARISON TO EXPERIMENTAL RESULTS

Westinghouse (Reference 4.4-2) ran experiments on duplex tubes with grooves for detection. They used a helium pressure 1 atm above the sodium pressure. They reported that detection of steam was relatively easy (8 seconds to 13 minutes) but detection of helium in sodium was difficult. They considered 4 to 20 mil holes located both at the 20 mil

4.4.1.5 COMPARISON TO EXPERIMENTAL RESULTS (continued)

grooves and between them. They tried to detect leaks by sodium vapor detection from back diffusion and variations in the helium pressure. In all cases, the detection of sodium vapor failed. Helium pressure variations were detected for only half the tests. Some plugging of the holes was observed. They also tested the detection of helium in flowing sodium and felt they could detect amounts as small as 0.1 ppb. This assumes no "background" helium is present in the sodium. This equipment was not used in the above tests. Their bundle test results were qualitatively consistent with the results presented here.

4.4.1.6 RESULTS AND CONCLUSIONS

Two helium pressures were considered, one greater than and one less than the sodium pressure. The following was concluded independent of which pressure is used:

1. Steam side leaks would easily be detected using water vapor detectors.
2. Detection of material moving through the grooves by diffusion will be difficult because of the geometry and small mass fluxes.
3. Detection of leaks from changes in the helium pressure would only work for very major leaks. Leaks considered in this work would not easily be detected.

4.4.1.6 RESULTS AND CONCLUSIONS (continued)

For the operating condition with helium pressure less than the sodium pressure, it was concluded that sodium vapor detection from the evaporation of sodium in the grooves is difficult. Liquid sodium would have to fill the grooves or be displaced from the grooves by pressure pulses.

For helium pressure greater than the sodium pressure, detection of helium in the sodium is possible. For an intermediate sized leak located at a groove, detection might take 20 - 500 seconds. For small leaks between the grooves, detection might take 10 to 30 hours. These long detection times are due to the large sodium inventory and because we assume a concentration of 1 PPM is required for detection. If detection at lower concentrations is possible, as Westinghouse suggests, detection times would be reduced proportionately.

Operation with the helium pressure exceeding the sodium pressure (intermediate helium pressure) is recommended. This mode of operation eliminates the need of helium pressure cycling and prevents sodium from flowing into the helium region. Problems associated with cleaning the sodium from the leaking tube or neighboring tubes are eliminated.

4.4.1.6 RESULTS AND CONCLUSIONS (continued)

In general, it is concluded that separate sodium and water/steam leaks can be detected within sufficient time to insure the elimination of a Na-H₂O reaction. That is, the leak will be detected, the unit shutdown and the leaking tube plugged. Furthermore, the probability of inducing a leak in the second pressure tube as a result of a leak in the other pressure tube is extremely low. This is due to the clean inert helium atmosphere and the tight contact between the duplex pressure tubes and the relatively short time between detection and tube plugging.

4.4.2 FAILURE MODE/FAULT TREE ANALYSIS

The failure mode analysis for the duplex bayonet tube consists of a standard failure mode analysis combined with a qualitative fault tree analysis. The undesired or "Top" event for the fault tree was chosen to be a sodium-water reaction. Sodium-water reactions have been identified as the major contributor to the unavailability of the steam generator (Ref. 4.4-3), and lowering the probability of their occurrence is the motivation behind the duplex bayonet tube design.

4.4.2.1 FAILURE MODE ANALYSIS (FMA)

FMA, as a basic tool of reliability assessment, has been used and described elsewhere (Ref. 4.4-4, 5, 6, 7, 8). A distinctive feature of this type of analysis is the examination of each type of component failure independently of all other components. Everything except the component under consideration is assumed to operate properly. A description of each component and its functions, based on Drawing E-63675023 was made prior to the analysis; the list is included in this report as Table 4.4-V. Potential failure modes were identified with the help of this list and possible causes or circumstances promoting these events are included. The FMA charts (Table 4.4-VI) also list the immediate consequences of each failure mode; where appropriate, methods of detection and inherent compensating provisions are noted as well as potential effects on the system. Probabilities have not been included in this analysis.

TABLE 4.4-V
COMPONENT DESCRIPTION AND FUNCTION FOR DUPLEX
TUBE STEAM GENERATOR FAILURE MODE ANALYSIS

1. REMOVABLE TUBE ASSEMBLY

<u>Number</u>	<u>Component</u>	<u>Function / Description</u>
1.1	Bayonet Tube Assembly	
1.1.1	Bayonet Tube	Forms downward/upward flow path for incoming steam/outgoing steam in superheater/evaporator.
1.1.2	Bayonet Tube Spacer	Shouldered pins preventing lateral motion of bayonet assemblies.
1.1.3	Bayonet Tube Spacer Key Welds	Attaches the tube spacers to the insulator tube.
1.1.4	Insulator Tube	Prevents regenerative heating in superheater.
1.1.5	Steam Chest Tubesheet	Supports bayonet tube assembly and forms base of steam outlet/water inlet plenum in superheater/evaporator - low pressure tubesheet - not a sodium/water interface; welded to channel shell.
1.1.6	Steam Chest Tube to Tubesheet Weld	Seal welds attaching bayonet tube assembly to steam chest tubesheet.
1.1.7	Threaded Fittings	Threaded and tapered fittings which mechanically attach bayonet assemblies to bayonet tubesheet.
1.2	Outer Wall Assembly	
1.2.1	Outer Wall High Pressure Tubes	Transfers heat from sodium to inner tube; barrier to sodium entry into third fluid region and a secondary barrier to steam/water entering sodium.
1.2.2	Outer Wall High Pressure Caps	Functions as 1.2.1.
1.2.3	Outer Wall High Pressure Cap Welds	Attaches outer high pressure tube end cap to outer high pressure tube.
1.2.4	Sodium Side Tubesheet	Supports the outer wall high pressure tubes and forms the base for the helium gas plenum
1.2.5	Sodium Side Tube-to-Tubesheet Welds	Attaches outer high pressure tubes to high pressure tubesheet.
1.2.6	Sodium Side Tubesheet Baffles	Lessens effect of thermal transients on high pressure tubesheet by absorbing thermal shock.

TABLE 4.4-V (continued)

1. REMOVABLE TUBE ASSEMBLY (Continued)

<u>Number</u>	<u>Component</u>	<u>Function / Description</u>
1.2.7	Tubesheet Baffle Supports	
1.3	Inner Wall Assembly	
1.3.1	Inner Wall High Pressure Tubes	Transfers heat from outer tube to water/steam and prevents steam from entering helium region; forms a secondary barrier to steam/water entering sodium.
1.3.2	Inner Wall High Pressure Caps	Reverses flow of water/steam in inner wall high pressure tubes and other functions of 1.3.1.
1.3.3	Inner Wall High Pressure Cap Welds	Attaches inner wall tube end caps to inner wall tubes.
1.3.4	Steam Side Tubesheet	Supports inner wall high pressure tubes; it forms a boundary between helium plenum and water/steam inlet/outlet plenum.
1.3.5	Steam Side Tube-to-Tube-sheet Weld	Attaches inner wall high pressure tubes to low pressure tubesheet.
1.3.6	Leak Detection Grooves (on inner wall of outer tube)	(3) Provides a path for the helium gas; the grooves runs the length of outer tube allowing any sodium or water vapor to enter helium plenum.
1.4	Tube Bundle Assembly	
1.4.1	Tube Bundle Thermal Liner	Supports "eggcrates" and prevents rotation of bundle within shell, enables sodium submersion of tube bundle.
1.4.2	Tube Bundle Lateral Support Keys	Allows the support keys to be attached to the bundle liner; these are slots in rings which are integral to the liner.
1.4.3	Eggcrate Supports	Primary supports for the outer wall high pressure tubing; friction contact acts to maintain spacing.
1.4.4	Eggcrate Support Keys and Welds	Attaches the assemblies to the bundle liner; "platform" arrangement accommodates thermal expansion.

TABLE 4.4-V (continued)

1. REMOVABLE TUBE ASSEMBLY (Continued)

<u>Number</u>	<u>Component</u>	<u>Function / Description</u>
1.4.5	Windows	Evaporator: distributes flow of incoming sodium around all high pressure tubes uniformly. Superheater: acts as a thermal baffle to shield sodium shell against severe transients.
1.4.6	Tube Bundle Liner Flange and Flange Bolts	Attaches thermal liner to tubesheet; flange is welded to liner and bolted to sodium side tubesheet.

TABLE 4.4-V (continued)

2. SODIUM SHELL ASSEMBLY

<u>Number</u>	<u>Component</u>	<u>Function / Description</u>
2.1.0	Sodium Shell Welds	Forms a pressure boundary between sodium and atmosphere.
2.1.1	Sodium Inlet Nozzle	Transports sodium from IHX/superheater to superheater /evaporator.
2.1.2	Sodium Inlet Nozzle Liner	Lessens thermal shock on inlet nozzle.
2.2.1	Sodium Bypass Seal	Spring - loaded seal which seals the lower part of shell from sodium flow; directs sodium flow upward in the annulus.
2.2.2	Sodium Bypass Seal Inner Support Ring	Welded onto the tube bundle liner and forms one point of attachment for bypass seal.
2.2.3	Sodium Bypass Seal Outer Support Ring	Welded onto sodium shell.
2.3.1	Sodium Outlet Nozzle	Carries sodium from superheater/evaporator; boundary between sodium and atmosphere.
2.3.2	Sodium Outlet Nozzle Thermal Liner	Protects sodium inlet/outlet nozzle against thermal shock.
2.3.3	Sodium Outlet Nozzle Thermal Liner Support	Supports the liner and centers it in the nozzle.
2.4.1	Sodium Shell Upper Liner	Protects the upper shell against thermal shock.
2.5.1	Channel Shell	Forms the upper outer housing for the steam generator; forms pressure boundary between steam/water inlet and outlet nozzles and atmosphere.
2.6.1	Diverter	Carries steam into/out of the superheater/evaporator, acts as a thermal liner.
2.6.2	Diverter Base	Supports diverter.
2.7.1	Helium Inlet/Outlet Nozzles	Carries helium to/from steam generator; forms boundary between helium and atmosphere.
2.8.1	Superheater/Evaporator Inlet/Outlet Nozzle	Carries water/steam from/to superheater/evaporator pressure boundary between water/steam and atmosphere.
2.8.2	Superheater/Evaporator Thermal Liner	Absorbs thermal shock.

TABLE 4.4-V (continued)

2. SODIUM SHELL ASSEMBLY (Continued)

<u>Number</u>	<u>Component</u>	<u>Function / Description</u>
2.8.3	Superheater/Evaporator Thermal Liner Bolts	Attaches thermal liner to inlet/outlet nozzle.
2.8.4	W/S Plenum Boundary	Forms a pressure barrier between inlet/ outlet water/steam.
2.8.5	Lateral Support Flange	Provides a point of attachment for channel shell bolts.
2.8.6	Lateral Support Bolts	Fastens the channel cover to channel shell walls.
2.9.1	Top Cover	Seals the upper shell and forms a pressure boundary against atmosphere.
2.9.2	Top Cover Bolts	Fastens top cover to channel shell.
2.9.3	Top Cover Seal	Double "O" ring which allows a leaktight seal to be formed by top cover.
2.9.4	Vessel Welds	
3.0	Vessel Support	None.

TABLE 4.4-VI FAILURE MODE CHARTS FOR DOUBLE WALLED BAYONET TUBE

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.1.1	bayonet tube	cracking	cyclic loading (thermal) corrosion defective tube	mixing of incoming water/steam with insulating fluid	indirectly, by reduced thermal performance	a design margin of 10% has been included in sizing calculations	small reduction in thermal performance	adequate water chemistry to be maintained
1.1.2	bayonet tube spacers	vibration	weld failure fabrication or design error wear from flow vibration	rubbing of bayonet tube against insulator tube improper flow in single tube		1 spacer per foot, with a 60 degree separation	same as above	contacting surfaces non abrasive
1.1.3	bayonet tube spacer welds	separation	weld porosity inclusions entrapped flux corrosion excess load human error	spacer may be detached	same as 1.1.1	spacers are press fit into bayonet tubes	same as above	
1.1.4	insulator tube	cracking	same as 1.1.1	direct mixing of insulating fluid and outgoing water/steam	same as 1.1.1	same as 1.1	reduction in thermal performance regenerative heating	same as 1.1.1
1.1.5	steam chest tube sheet	material separation	thermal cycling stress corrosion fatigue cracking	leakage of inlet steam/water into outlet steam/water	same as 1.1.1	this tubesheet is not a sodium-water interface	reduction in thermal performance	5 inch, 2-1/4-Cr-1 Mo tubesheet
		excessive deformation	excessive load excess temperature bending of shell	bayonet and/or insulator tube distortion		shell is 7 inches thick in this region		

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.1.6	steam chest tube-to- tubesheet welds	leakage	same as 1.1.3 stress corrosion thermal fatigue	leakage of in- let steam/ water into outlet steam/ water	same as 1.1.1	this is a low pressure tubesheet, well separated from sodium tubes are also fastened by fittings	reduction in thermal performance	seal weld penetration tested visually inspected
1.1.7	threaded fitting and bolt	cracking	excess load design or fabrication error stress corrosion overtighten- ing	distortion or excess load on bayonet or insulator tube increased load on remaining bolt	periodic inspection	tubes are also supported by "eggcrates"		
4-249	1.2.1 outer wall high pressure tubes	leakage at support points	fretting corrosion stress corrosion corrosion fatigue excessive thermal or mechanical loading flaw in tubing during manufacture	helium enters sodium side	helium detected by gas chromatograph slow helium pressure decrease	"eggcrates" are rounded bars with polished and deburred edges duplex tube design minimizes consequences fastener minimizes effects of differential thermal expansion	system shutdown after detection	tubes will be x-rayed, penetrant tested, hydro - statically tested
		leakage in other region	stress corrosion corrosion excessive thermal or mechanical loading flaw in tubing during manufacture	same as above		duplex tube design minimizes consequences	same as above	same as above
		fouling of heat transfer tube surface	impurities in sodium	potential corrosion site		10% design margin plus a large fouling factor	reduced thermal performance	adequate sodium purity to be maintained

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.2.2	outer wall high pressure caps	leakage	stress corro- sion corrosion excess thermal or mechanical loads	helium enters sodium side	helium detected by g.c. system and slow helium pressure decrease	duplex tube minimizes consequences	system shutdown after detec- tion	same as 1.2.1
1.2.3	outer wall high pressure cap welds	same as 1.1.3, plus stress corrosion thermal fatigue	same as above	same as above	same as above	duplex tube minimizes consequences	same as above	full penetration weld x-rayed penetrant test
1.2.4	sodium side tubesheet	material separation	cracking from thermal or mechanical fatigue stress corro- sion corrosion excessive loads	helium enters sodium system improper tube loading	same as above	baffles lessen thermal transient effects tubesheet is not a sodium-water interface	system shut down after detection	10 inch 2-1/4-Cr-1 Mo tubesheet
		excessive deforma- tion	warping of shell improper loading improper temperature distribution	tube vibration stress on eggcrates sliding between inner and outer tubes		high pressure tubing supported by two tubesheets		
1.2.5	sodium side tube-to- tubesheet welds	leakage	same as 1.1.3, plus stress corro- sion thermal fatigue	helium enters sodium system	helium detected by gas chromatograph helium pressure decrease	the tubesheet is not a sodium-water interface	system shutdown after detection	IBW, x-rayed penetration tested

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.2.6	sodium side tubesheet baffles	cracking	corrosion corrosion fatigue	tubesheet thermal shielding decreased increased possibility of tubesheet damage from from inlet sodium transient possible damage to tube bundle	thermocouples monitor stagnant sodium region	6 separate baffles	system shutdown for repair	1-inch thick baffles
1.2.7	tubesheet baffle supports	failure	corrosion corrosion fatigue	loosening of baffle, possible damage to tube bundle, plugging of tube	possibly, indirectly by reduced thermal performance		same as above	
1.3.1	inner wall high pressure tubes	leakage at: tube spacer region	stress corrosion corrosion fretting corrosion excessive thermal or mechanical loading flaw in tube during manufacture	water/steam enters helium region	water vapor detect- ed by hygrometer helium pressure increase hydrogen vapor increase	duplex tube design minimizes consequences	system shutdown after detection	spacers are rounded and polished
		leakage in other regions	stress corrosion corrosion excessive thermal or mechanical loading flaw in tube during manufacture	same as above	same as above	same as above	same as above	

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.3.1 (Cont)		fouling	impurities in water/steam	potential formation of corrosion deposits	reduced thermal performance	10% design margin includes this failure mode		
		high gap resistance	decrease in steam pressure	temperature maldistribution	indirectly, by reduced thermal performance	conservative estimate of gap conductance high conductivity fluid in gap	reduced thermal performance	
1.3.2	inner wall high pressure caps	leakage	stress corrosion excessive thermal or mechanical loading	water vapor enters helium region	water vapor detected by hygrometer	small volume will pressurize quickly, preventing prolonged damage from water/steam jet	system shutdown after detection	
1.3.3	inner wall high pressure cap welds	leakage	same as 1.1.3, plus stress corrosion thermal fatigue shock (hydraulic)	water vapor enters helium region increased stress on remaining intact part of weld potential weld failure	same as above	duplex tube minimizes consequences and insures separation of sodium and water/steam	system shutdown after detection	circumferential full penetration, butt weld x-rayed penetrant tested
1.3.4	steam side tubesheet	material separation	thermal or mechanical fatigue stress corrosion creep warping of support shell	leakage of water/steam into helium plenum excess load on high pressure tubing	water vapor detected by hygrometer	duplex tube design with separate tubesheets reduces consequences of this fault helium gas shields this tubesheet from severe sodium thermal transients	system shutdown on leak detection	

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.3.4 (Cont)		excessive deformation	excess mechanical load warping of support shell temperature maldistri- bution	excess load on high pressure tubing sliding between inner and outer wall tubes			possible tube vibration stress on eggcrate supports reduced thermal performance	
1.3.5	Steam Side Tube-to-Tube- sheet Welds	leakage	same as 1.1.3, plus stress corro- sion thermal fatigue	leakage of water/ steam into helium	water vapor detected by hygrometer helium pressure increase	double tubesheet design	system shutdown on leak detection	IBW, x-rayed penetration tested
1.3.6	leak detection groove	plugged	foreign material, possibly corrosion products from water/ steam or sodium in helium region	reduction in flow area available to sodium steam vapor, or helium		three grooved paths to helium plenum provides conservative flow area		
		sealed	design/ fabrication error	same as above				
1.4.1	tube bundle thermal liner	deformation	excessive loading from eggcrates creep	possible tube vibration excess load on tubes possible tube wear maldistribution of sodium flow		design projects quiescent sodium flow		method of attachment to shell allows for differential thermal expansion

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
1.4.2	liner support keys	cracking	liner vibration corrosion stress corrosion	liner may not remain centered in shell		liner is keyed in several locations	sodium flow maldistribution	method of attachments minimizes stress due to differential expansion
1.4.3	eggcrate supports	cracking	mechanical fatigue	tube vibration		tubes are constrained by nine levels of eggcrate supports		same as above
1.4.4	eggcrate support keys and welds	weld failure	weld porosity inclusions entrapped flux thermal fatigue excess load human error corrosion	vibration of duplex tube leading to fretting corrosion		intersections of eggcrates welded eggcrates are rounded and polished nine layers of supports maintain rigidity of tube bundle in shell		design of support keys accomodates differential thermal expansion full penetration weld design deletes crevices
1.4.5	windows	enlargening	corrosion/erosion	nonuniform distribution of flow to duplex tubes	indirectly, by reduced thermal performance	windows located below sodium level to ensure low flow rate	reduced thermal performances	
1.4.6	tube bundle liner flange and bolts	bolt failure	stress corrosion overtightening defective bolt	increased strain on remaining fasteners, including possible flange cracking		liner is flexibly supported at shell midpoint and inlet nozzle region		
		weld failure or flange rupture	same as 1.1.3 thermal stress	increased strain on remaining fasteners liner distortion maldistribution of flow				

TABLE 4.4-VI (continued)

<u>No.</u>	<u>Component</u>	<u>Failure Mode</u>	<u>Cause</u>	<u>Local Effect(s)</u>	<u>Method of Detection</u>	<u>Inherent Compensating Provision</u>	<u>Effect on System</u>	<u>Remarks</u>
2.1.0	sodium shell	leakage at regions other than welds	excessive load missiles corrosion	sodium escapes steam generator boundary	building instrumen- tation		system shutdown after detection personnel hazard	regions of high temperature or severe temperature transients are shielded hydrostatically tested
2.1.1	sodium inlet nozzle	leakage	corrosion stress corro- sion thermal/ mechanical fatigue	sodium escapes steam generator boundary	same as above		system shutdown personnel hazard	nozzle shielded by thermal liner
		material separation	excess temperature excess loading	sodium fire possible excess load on inlet piping	same as above	high pressure tube- sheet well baffled	system shutdown personnel hazard loss of coolant to system may bring thermal shock	
2.1.2	inlet nozzle thermal liner	distortion	corrosion excess temperature	minor flow redistribution		superheater/evaporator has 41.5"/20.0" I.D. inlet nozzle	insignificant unless severe distortion leads to cracking, loss of thermal protection for nozzle, shell	attachment accommodates thermal expansion
2.2.1	sodium bypass seal	material separation	thermal/ mechanical fatigue excess load creep	superheater/ evaporator outlet/inlet sodium contact with unlined shell	indirectly, by flowmeters	tubesheets well- baffled	overheat of sodium shell loss of coolant to evaporator	

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
2.2.2, 2.2.3	sodium by-pass seal support rings	material separation	thermal/mechanical fatigue excess temperature	inlet/outlet sodium contact with shell	indirectly, by flowmeters	seals are spring loaded to cushion effect of flow interruption	same as above	attachment accommodates thermal expansion
2.3.1	sodium outlet nozzle	leakage	same as 2.1.1	sodium escapes steam generation to boundary leak growth	building instrumentation	nozzle is protected by thermal liner critical components are shielded thermally	system shutdown upon detection	
		material separation	same as 2.1.1	same as above sodium fire	building instrumentation indirectly, by flowmeter		system shutdown upon detection possible thermal shock to system coolant loss to IHX/SHTR.	
2.3.2	sodium outlet nozzle thermal liner	distortion	corrosion excess temperature	minor flow redistribution		superheater/evaporator has 20.0"/41.5" I.D. outlet nozzle	same as 2.1.2	attachment accommodates thermal expansion
2.3.3	thermal liner support pins	failure of welds	same as 1.1.3, plus corrosion stress corrosion	increased stress on remaining pins		multiple fasteners fix liner to shell	insignificant	design accommodates thermal expansion
2.4.1	sodium shell upper liner	distortion	corrosion excess temperature	possible overheating of sodium shell potential for damage from sodium inlet transient			insignificant unless severe distortion causes material separation of liner	attachment accommodates differential thermal expansion between shell and liner

TABLE 4.4-VI (continued)

<u>No.</u>	<u>Component</u>	<u>Failure Mode</u>	<u>Cause</u>	<u>Local Effect(s)</u>	<u>Method of Detection</u>	<u>Inherent Compensating Provision</u>	<u>Effect on System</u>	<u>Remarks</u>
2.5	channel shell	leakage (welds)	same as 1.1.3 thermal fatigue	w/s escapes boundary of steam genera- tor	building instrumen- tation		system shutdown upon detection	7 inch channel shell thickness
		bending	creep excess temperature	excess load on inlet/outlet nozzle				
2.6.1	diverter	leakage	corrosion	w/s escapes to atmosphere w/s contacts diverter base	building instru- ments	plenum boundary shielded from thermal shock by diverter base	system shutdown upon detection	
		distortion	excess temperature creep	minor flow redistribution				
2.6.2	diverter base	cracking	cyclic mechanical/ thermal loading	w/s contact with upper plenum boundary			insignificant unless severe inlet w/s transient	
2.7.1	helium inlet/ outlet nozzle	leakage	excess load	ability to monitor leaks compromised	low helium pressure building instru- ments	duplex tube lessens probability of sodium water reaction, loss of detection is not critical	system shutdown	
		warpage	thermal fatigue	helium pressure decrease				

TABLE 4.4-VI (continued)

No.	Component	Failure Mode	Cause	Local Effect(s)	Method of Detection	Inherent Compensating Provision	Effect on System	Remarks
2.7.2	helium plenum handholes	leakage	improper load supplied by bolts	helium escapes steam generator boundary	low helium pressure building instrumen- tation	helium pressure is not prime leak detection method	system shutdown on low helium pressure	
2.8.1	superheater/ evaporator inlet/ outlet nozzle	leakage	same as 2.1.1	w/s escapes steam generator boundary	building instru- ments		system shutdown	
		material separation	same as 2.1.1	same as above			system shutdown large leak may result in system overheat	
2.8.2	thermal liner	distortion	corrosion excess temperature	minor redistribution of w/s flow pattern		local overheating of shell	insignificant	
		cracking	prolonged and excessive distortion	mixing of inlet/ outlet w/s	indirectly, by reduced thermal performance		reduced thermal efficiency	
2.8.3	thermal liner bolts	failure	stress corrosion corrosion overtighten- ing defective bolt	possible crack- ing of thermal liner increased stress on remaining bolts possible vibration of liner possible leak between inlet/ outlet w/s	indirectly, by reduced thermal performance		reduced thermal efficiency	

TABLE 4.4-VI (continued)

<u>No.</u>	<u>Component</u>	<u>Failure Mode</u>	<u>Cause</u>	<u>Local Effect(s)</u>	<u>Method of Detection</u>	<u>Inherent Compensating Provision</u>	<u>Effect on System</u>	<u>Remarks</u>
2.8.4	w/s plenum boundary	leakage	corrosion	inlet w/s mix- ing with out- let w/s	indirectly, by reduced thermal performance		reduced thermal efficiency	
		material separation	thermal stress	same as above				
2.8.5	lateral support flanges	cracking	thermal/ mechanical fatigue improper load applied by support bolts	vibration of plenum boundary, diverter		flange is multiple bolted to channel shell		
		weld failure	same as 1.1.3, plus excessive load	same as above				
2.8.6	lateral support bolts	failure	corrosion overtighten- ing defective bolt	same as above				
2.9.1	top cover	material separation	excess pressure excess temperature	w/s escapes steam generator	building instrumen- tation		large leak may produce system overheat system shutdown	
2.9.2	top cover bolts	failure	same as 2.8.6	same as above	building instrumen- tation periodic inspection		system shutdown	

TABLE 4.4-VI (continued)

<u>No.</u>	<u>Component</u>	<u>Failure Mode</u>	<u>Cause</u>	<u>Local Effect(s)</u>	<u>Method of Detection</u>	<u>Inherent Compensating Provision</u>	<u>Effect on System</u>	<u>Remarks</u>
2.9.3	top cover seal	leakage	cracking from improper tightening of cover bolts improper fit	same as above	same as above	both "O" rings must fail for leakage	 system shutdown	
2.9.4	vessel welds	leakage in lower shell welds	same as 1.1.3, plus stress corrosion excess mechanical load	sodium escapes steam generator boundary loss of sodium to IHX	building instru- mentation		system shutdown personnel hazard	standard butt weld, subject to periodic visual inspection
		leakage at sodium side tube sheet-to- shell welds	same as above, plus thermal fatigue excessive tube- sheet defor- mation	same as above	same as above		same as above	
		leakage at steam side tubesheet- to-shell welds	same as 1.1.3, plus	w/s escapes steam generator boundary	building instru- mentation		system shutdown on large leak	

4.4.2.1 FAILURE MODE ANALYSIS (FMA) (continued)

No single failure resulting in a sodium-water reaction could be identified from the FMA. Preliminary analysis of the design indicates a high degree of redundancy. The FMA is of limited value in analyzing system interaction because it is component oriented, and it is not suited to demonstrating that a single event (such as a sodium-water reaction) requires multiple component failures and has many possible system failure modes (minimum combinations of component failures which produce the undesired event).

4.4.2.2 FAULT TREE ANALYSIS (FTA)

Awareness of the FMA limitations motivated the construction of a preliminary fault tree, included in this report as Drawing SE-FBR-651-007. Symbols used on the fault tree have their usual meaning, but a key has been included. System failure modes may be discovered by starting at the bottom and following a path to TOP. The tree itself was deduced from TOP to the basic faults.

Any analysis of this type is based on certain assumptions or boundary conditions; the assumptions used in conducting this analysis are listed below:

1. The analysis was performed for a single unit operating at full power steadystate conditions.
2. The primary faults are taken to be leaks of a size for which the probability of detection is essentially one. Leaking fluids

4.4.2.2 FAULT TREE ANALYSIS (FTA) (continued)

2. (continued)

are assumed to reach the detector.

3. The system behaves in the same way regardless of leak location, i.e., groove versus interface. Leaks have not been categorized according to location because the fault sequence is essentially independent of leak location. Research on leak detection for a similar design (Ref. 4.4-10) justifies this assumption on a preliminary basis, as it appears that the only difference will be in the time required to detect the leak.
4. Leak detection capability in helium for hydrogen and sodium vapor is provided by gas chromatographs; water vapor is monitored in helium with an aluminum hygrometer. Sodium side leaks are detected by monitoring sodium for helium with a gas chromatograph. Backup detection is provided by helium pressure transducers.
5. Detection of a leak initiates prompt and proper corrective action; detector parameters (Ref. 4.4-11) have in no way been modeled in this analysis.
6. The ability of sodium and water leaks to "communicate" within the steam generator depends on various conditional connections; some of the hexagonal symbols on Drawing SE-FBR-651-007 indicate these connections. The probability of sodium combining

FAULT TREE FOR DUPLEX HAYONET TUBE STEAM GENERATOR

KEY

AND GATE
EXISTENCE OF ALL INPUTS REQUIRED TO PRODUCE OUTPUT

OR GATE
OUTPUT WILL EXIST IF AT LEAST ONE INPUT IS PRESENT

INHIBIT GATE
INPUT PRODUCES OUTPUT DIRECTLY WHEN CONDITIONAL INPUT IS SATISFIED.

RECTANGLE
A FAULT EVENT USUALLY RESULTING FROM THE COMBINATION OF MORE BASIC FAULTS ACTING THROUGH LOGIC GATES

DIAMOND
A FAULT EVENT NOT DEVELOPED TO ITS CAUSE

CIRCLE
A BASIC COMPONENT FAULT AN INDEPENDENT EVENT

TRIANGLE
A CONNECTING OR TRANSFER SYMBOL.

A - FAILURE TO DETECT HELIUM IN SODIUM
B - FAILURE TO DETECT SODIUM VAPOR
C - FAILURE TO DETECT WATER VAPOR
D - FAILURE TO DETECT HYDROGEN
E - FAILURE TO DETECT HELIUM OVER PRESSURE
F - FAILURE TO DETECT HELIUM UNDER PRESSURE
G - FAILURE OF HELIUM SYSTEM (OVER AND UNDER PRESSURE)
H - CAUSTIC CORROSION OF WP BOUNDARY IN SAME TUBE
I - CAUSTIC CORROSION OF WP BOUNDARY IN PLENUM OR ADJACENT GROOVE REGION

4.4.2.2 FAULT TREE ANALYSIS (FTA) (continued)

6. (continued)

with water in the same tube region is assumed to be one.

The potential for fluid re-entry from a detection region through the helium plenum is also indicated on the fault tree. A leak in the water/steam tube combined with the environment in the steam generator allows no appreciable condensation of the water/steam mixture; it is expected that the water/steam mixture will remain in the plenum region, or escape via the low pressure path offered by the helium outlet. Leaking sodium, however, may re-enter adjacent helium grooves from the plenum.

7. The possibility of induced leakage has been included, although it is expected to be very small. High pressure steam flowing parallel to the sodium tube for a short time and caustic corrosion by sodium are the projected mechanisms.
8. A proper helium pressure (between sodium and water/steam) will allow only trace quantities of sodium to enter the helium region, if a sodium leak occurs. The effects of this backflow sodium serve, in this analysis, to aid leak detection.
9. External helium depressurization mechanisms, such as rupture or leakage, are expected to lower the helium pressure rapidly.
10. The location of sodium-water reactions has been limited to the sodium side and the helium region. A fault sequence involving a pressure reversal on the water/steam side has not been

4.4.2.2 FAULT TREE ANALYSIS (FTA) (continued)

10. (continued)

included in the analysis.

11. Human failures of any type have not been included explicitly in the fault tree; faults such as failure to respond to detector signals, failure to pressurize helium, and various maintenance failures constitute only a partial list of typical human failures. Modeling of such behavior could not be attempted in a preliminary reliability assessment, so exclusion of these faults is not a drawback.

An inspection of drawings SE-FBR-651-007 and SE-FBR-651-006 (See Section 4.4 of Volume I , Matrix Steam Generator) shows two types of differences only partially linked to differences in design. Duplex bayonet tube leaks were not separated according to location because no essential difference appears to occur in the response of the system when leak location is varied. Potential fluid reentry and induced leakage have not been modeled as leak location dependent, and boundary condition (Ref. 4.4-5) makes the leak independent of position. The second difference in the two designs is more formal: the occurrence of a fluid leak may contribute several possible paths to TOP, and on Drawing SE-FBR-651-007 multiple path contributions are symbolized by the transfer symbol, whereas on drawing SE-FBR-651-006 multiple path contributions are illustrated explicitly by including the primary fault each time it occurs. Remaining fault tree differences are design dependent.

4.4.2.2 FAULT TREE ANALYSIS (FTA) (continued)

The fault tree shows only a simple failure of the helium pressure system for each pair of sodium and water leaks, and thus avoids the logical pitfall of displaying mutually exclusive faults in the same system failure mode.

Visual inspection of the fault tree will show that, for any given primary fault, another primary fault together with multiple detector failures is required to reach the TOP. Induced leaks and reactions produced by re-entry through the helium plenum (albeit very unlikely) require multiple detector failures. Within the boundaries established for the analysis, a sodium-water reaction requires at least two independent faults.

4.4.2.3 CONCLUSIONS

1. No single failure can be found whose immediate consequence is a sodium-water reaction.
2. Under all fault conditions considered, at least two independent failures are required to produce a sodium-water reaction.
3. Failure sequences involving an induced leak always require multiple detector failures in order to produce a sodium-water reaction.

REFERENCES FOR SECTION 4.4

- 4.4-1 R. B. Bird, W. E. Stewart, E. N. Lighfoot, Transient Phenomena, John Wiley and Sons, Inc., 1960.
- 4.4-2 A. Lohmeire, D. J. Bongaards, "Development and Testing Results of W Double Wall Tube Steam Generator Including Thermal Model Tests, "Joint USA/USSR Seminar on Development of LMFBR Steam Generators, Los Angeles, California, December 2 - 4, 1974.
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- 4.4-7 G. Das Guptas, A. C. Gangadharan, "Failure Mode and Effect Analysis of the Foster Wheeler Protected Bayonet Tube Steam Generator Concept", SM-74-3, Rev. 0, Foster Wheeler Corporation, Livingston, New Jersey, June 1974.
- 4.4-8 LMFBR Engineering Staff, "Steam Generator Module Preliminary Failure Mode and Effects Analysis", TI-0 36-610-008, Atomics International, Canoga Park, CA, September 1974.

REFERENCES FOR SECTION 4.4 (continued)

- 4.4-9 D. F. Haasl, "Advanced Concepts in Fault Tree Analysis", System Safety Symposium, June 8-9, 1965, Seattle: The Boeing Company.
- 4.4-10 A. Lohmeier, D. J. Bongaards, "Development and Test Results of Westinghouse Double Wall Tube Steam Generator", p. 221, Joint US-USSR Seminar on Development of Sodium Fast Breeder Reactors", December 1974.
- 4.4-11 J. A. Ford, et al., "Criteria for Detection of Small Sodium-Water Reactions in LMFBR Steam Generators", APDA-255, Atomic Power Development Associates, June 1970.

4.5 OTHER CONSIDERATIONS

There were some areas within the duplex bayonet tube steam generator component designs which were of specific concern during the February 1, 1975 status review meeting between C-E and ERDA. These areas prompted the following work items to be performed (Reference 4.5-1):

- A. An estimate of the gross effects of using SA 213 T22 tubing material in the evaporator unit in lieu of the Sandvik H-T9 material.
- B. An investigation of stress concentration at the leak detection grooves.
- C. A discussion of using a "no leak detection" approach in lieu of the monitored gas leak detection system.
- D. A discussion of the differences in residual stresses associated with internal expansion versus external reduction in regard to the duplexing operation.
- E. A discussion of alternate tube-to-tubesheet welding approaches.

This section is provided as an aid to the reader in finding these particular points of concern.

4. 5. 1 EFFECTS OF USING SA 213 T22 TUBING MATERIAL
 IN LIEU OF SANDVIK H-T9 (ITEM 4. 5(A))

The structural impact of using the more common SA 213 T22 tubing material in lieu of the higher strength Sandvik H-T9 material is reported in Appendix "E" and the thermal sizing effects of using these thicker tubes were previously reported in Section 4. 1. 3 of this report.

4. 5. 2 STRESS CONCENTRATION OF THE LEAK DETECTION
 GROOVES (ITEM 4. 5(B))

Appendix "E" addresses the problem of stress concentration at the leak detection grooves.

4. 5. 3 NO LEAK DETECTION MONITORING SYSTEM (ITEM
 4. 5(C))

Appendix "F" provides a discussion of duplex tube designs without intermediate fluid leak detection systems.

4. 5. 4 DIFFERENCES IN RESIDUAL STRESSES ASSOCIATED WITH
 THE TWO APPROACHES TO DUPLEXING THE TUBES
 (ITEM 4. 5(D))

This subject is dealt with in Section 4. 2.

4. 5. 5 ALTERNATE TUBE-TO-TUBESHEET WELD APPROACHES
 (ITEM 4. 5(E))

Section 4. 3. 2 deals with this subject in detail.

REFERENCES FOR SECTION 4. 5

4. 5. 1 "Commitment and Agreements" made at meeting at Chattanooga
 between ERDA and C-E, February 1, 1975.

5.0 MANUFACTURING SEQUENCE

The assembly sequence for each of the respective duplex bayonet tube steam generator components (evaporator, superheater and reheater) is the same. Figures 5 - 1a, b, and c delineate a logical sequence for producing the units.

As shown in Figure 5 - 1a, the lower sodium shell and the thermal liner (with the eggcrate tube supports installed) would be fabricated and rested on the shop floor in an upright position. The thermal baffle plates would be "stowed" in the thermal liner with tube holes properly aligned. The outer pressure tubes would then be inserted through the thermal baffle plates and eggcrate tube supports and allowed to rest on the shop floor as shown. Concurrently, an upper shell/gas plenum subassembly would be fabricated from the water side inlet and outlet nozzles, the high pressure shell plates, the flange for the removable top head, the two tubesheets, and two gas plenum shell plate.

The upper shell/gas plenum subassembly would then be installed on the uprighted lower sodium shell by installing segments of the upper sodium shell as shown at

MANUFACTURING SEQUENCE
FIGURE 5 - 1a

MANUFACTURING SEQUENCE (CONT)
FIGURE 5 - 1b

CONTINUED

MANUFACTURING SEQUENCE (CONT'D)
FIGURE 5 - 1c

5.0 MANUFACTURING SEQUENCE (continued)

the extreme right of Figure 5 - 1a . Thus the outer pressure tubes and the thermal baffle plates are temporarily stored inside the shell with "working space" provided below the sodium/gas tubesheet and "work access" provided between the upper sodium shell segments.

As shown in Figure 5 - 1b, the outer pressure tubes would be raised and welded to the sodium/gas tubesheet using the internal multipass tube-to-tubesheet welding technique developed by Combustion Engineering under AEC contract (AT 11-1)-3031 (See Section 4.3.2). Welding would be done in rows across the tube bundle to facilitate placement and retrieval of X-ray film during the tube-to-tubesheet joint inspection process.

After completing the process of welding and inspecting the joints, the thermal baffle plates would be lifted into final position and supported from the sodium/gas tubesheet and the remaining upper sodium shell segments would be installed.

As shown in Figure 5 - 1c the inner pressure tubes would be installed down through the steam/gas tubesheet and inside the outer pressure tubes. After "duplexing" the inner tubes into intimate contact with the outer tubes as described in Section 4.3.1, the inner tube-to-tubesheet welds would be

5.0 MANUFACTURING SEQUENCE (continued)

made using a corner weld technique (See Section 4.3.2). Access for X-ray film placement and retrieval is provided by hand holes located in the gas plenum shell.

The high temperature steam chest would be aligned and installed as shown at the extreme right of Figure 5 - 1c and the bayonet/insulator tube assemblies would be installed through the high temperature tubesheet into the duplex bayonet tubes. The unit would be completed by installing the steam chest top plate, the removable upper head, and the lower hemispherical head.

This assembly sequence is intended to show only the practicability of manufacturing a duplex bayonet tube steam generator component and does not reflect such variables as best facility usage, shop burdening, schedular aspects, etc. which would naturally influence the sequence of production. Such an in-depth study of the manufacturing sequence was not within the scope of this effort.

6.0 DEVELOPMENT PROGRAM

There are several areas in which additional study effort relative to the duplex bayonet tube steam generator would be desirable. These areas include additional structural and thermal analytical study, cost evaluation, study of tube fouling in the evaporator heat transfer annulus, and some limited physical testing for both manufacturing "proof of principle", and heat transfer assessment. These desirable "follow-on" study tasks are outlined in the following paragraphs.

6.1 ANALYTICAL FOLLOW-ON EFFORT

6.1.1 STRUCTURAL

In accordance with the recommendations in Section 4.2.3.3, the structural follow-on design effort for the duplex bayonet tube steam generator would include the following analytical tasks:

A. The parameter study reported in Section 4.2.3 would be extended to include the configuration of the reheater unit. Additionally, more precise estimates of the material properties, particularly the constitutive equations, would be developed and used in this task.

6.1.1 STRUCTURAL

B. The effect of the leak detection grooves on the response of the pressure tubes to the duplexing process would be established with a three-dimensional, inelastic finite element model of a short slice of the tubes. Ideally, the geometry of both the evaporator and reheater should be analyzed considering the nominal gap size required for assembly of the tubes. Tolerance build up, resulting in one deep groove and three shallow grooves, would be considered to develop the most severe strain accumulation possible.

C. A long cylinder length of the duplexed tubes would be analyzed to determine the effect of the incipient sliding condition on the response of the tubes during cyclic operation. An elastic, axisymmetric finite element model would be employed. The tubes would be preloaded to an attainable residual contact pressure with relative upward sliding of the inner pressure tube imminent. Then, the unit would be subjected to a nominal operation sequence including a number of estimated transients. The response of the tubes to this operational loading history will determine whether sliding can occur in only a few isolated instances or must be expected continually throughout the life of the unit.

D. An investigation would be performed of the development of axial loads in the pressure tubes due to relative deformations of the two tubesheets.

6.1.1 STRUCTURAL (continued)

These loads could result in buckling of the pressure tubes or in failure of the tube-to-tubesheet welds; they also contribute to the previously discussed sliding problem. Relative deformations of the tubesheets are caused by tubesheet bowing or doming due to transverse pressure loads and temperature gradients and by differential thermal growth between the vessel shell and tube bundle. An elastic, axisymmetric finite element analysis of a portion of the vessel shell, both tubesheets, and the length of the inner pressure tubes between the tubesheets would be performed. The tubesheets would be modeled as continuous, homogenous plates; however, their material properties will be modified in the perforated region to account for reduced stiffness. The inner pressure tube would be modeled as simple spar or beam elements between the tubesheets.

6.1.2 THERMAL

Further investigations would be performed to optimize the configuration from a heat transfer/structural integrity point of view. This study would include the sizing of the heat transfer annuli, the tube diameters, and the tube center-to-center locations.

The option of using a spiral flow path in the heat transfer annuli in lieu of a straight through flow path would be investigated as a means of improving the heat transfer characteristics of the evaporator unit.

6.1.2 THERMAL (continued)

This would be physically accomplished by using a spirally wrapped wire (or some similar device) in place of the intermittently located spacers as shown on the drawings in Section 3.0.

6.2 PHYSICAL ASSESSMENT OF THE DUPLEXING OPERATION

A "proof of principle" test of the duplexing operation is felt to be highly desirable as a follow-on study effort. This would entail purchasing several representative sizes of tubing and physically performing the continuous rolling operation described in Section 4.3.1. The following tests and/or examinations would be made to confirm or modify some of the assumptions made in this study effort:

- The tubes would have grid lines etched or photographically applied to their respective outer surfaces. The location of these grid lines would be established before and after the duplexing operation to determine gross plastic changes undergone by the material during the rolling process. In addition, strain gages would be employed to determine outer tube wall strain during the duplexing operation.
- Post fabrication microscopic examination of the interfacial zone would be made to determine the resulting average "gap" and the amount of metal-to-metal contact achieved by the duplexing process.

6.2 PHYSICAL ASSESSMENT OF THE DUPLEXING OPERATION (continued)

- A bench test would be performed to determine helium leak rates through "fabricated" leak sites both at leak detection grooves and between leak detection grooves.
- A bench test would be performed to determine the thermal resistance (or conductance) of the interfacial zone between the duplex tubes. (This test would be accomplished using an internal heater and external thermocouples to determine the overall wall resistance and back calculating to determine the "gap" resistance).
- A "push/pull" bench test would be performed to assess the magnitude of loads required to induce slippage between the respective tubes.
- Short sections of the duplex tubing would be used to determine the contact pressure attained during fabrication.

6.3 ONE TUBE SODIUM HEATED INTEGRAL TEST

A single-tube sodium heated integral test would be planned and the test model would be designed as an immediate follow-on to this study effort. Actual performance of the test would be accomplished in a subsequent time period. The intent of this test would be to obtain the following:

- A. Two-phase heat transfer data.
- B. Hydrodynamic stability data for the evaporator unit.

6.3 ONE TUBE SODIUM HEATED INTEGRAL TEST (continued)

C. Short term and induced fouling in the heat transfer annulus
and the study of "crud buildup" removal by chemical cleaning.

6.4 THERMAL TRANSIENT STUDY

A study would be performed to better define and study the structural effects of certain of the most severe system thermal transients. This study would include but not necessarily be limited to the following transients:

- Normal startup and shutdown
- Emergency shutdown
- Sudden loss of sodium flow

6.5 COST EVALUATION

For comparative purposes, a cost evaluation of the duplex bayonet tube components would be performed. This would necessitate analyzing the cost of a comparable sized single walled bayonet tube unit to permit a direct comparison to be made. Fabrication costs associated with making production tube-to-tubesheet welds, which to date have only been made under laboratory conditions, and the costs associated with the tube rolling operation would also have to be considered.

6.6 OTHER EFFORTS

There are two other items which would be considered under a follow-on study. These are aimed at rectification of problems associated with water side drainability and with stress/strain concentration in the leak detection grooves. These two subjects are delineated in A and B as follows:

A. Water Side Drainability

An inherent disadvantage of any bayonet tube component orientated as shown in the drawings of Section 3.0, is that the water side of the unit is not drainable. This disadvantage could be rectified in the case of the duplex bayonet tube units, since the units do not feature a cover gas, by redesigning the unit to operate in a position inverted from that shown in the Section 3.0 drawings. This alternative would be studied to assess the ramifications associated with operating and maintaining such an "inverted" unit.

B. Stress/Strain Concentration in Leak Detection Grooves

An alternate design would be studies in which a third or intermediate tube would be incorporated between the pressure tubes. The intent of this additional tube would be to provide a non-structural member in which the leak detection grooves could be located without concern for stress/strain concentration. This effort would assess the thermal performance effects caused by the addition of such a structurally dormant member.

APPENDIX "A"

THERMAL SIZING EQUATIONS

APPENDIX A

Thermal Sizing Equations

The thermal sizing of the unit is performed by solving the energy equation (see EQ #4.1-10) through the appropriate selection of the terms which comprise it. In order to avoid redundancy, the sizing procedure for the evaporator will be described as it is the most tedious calculation. The equations which were used to size the superheater and reheater are those described in the evaporator's "superheater" section.

The evaporator was divided into four water-side heat transfer regions, as illustrated in Figure A.1, which include: single phase forced convection, saturated or nucleate boiling ($x \leq 0.3$), mist flow (i.e., film boiling) ($0.3 < x < 1.0$), and single phase vapor ($x = 1.0$). As this is only a preliminary sizing, a detailed investigation of DNB, dryout, axial heat transfer coefficient variation, wall temperatures, etc., is not warranted. Each heat transfer region is evaluated by assuming an average value for temperatures, material properties, and heat transfer coefficients for the length of that region. Once the overall heat transfer coefficient and the log-mean temperature difference are known, the heat transfer area and a corresponding tube length are easily calculated. The total tube length is then found by adding the tube length for each section. A safety factor of ten percent is then added to the heat transfer area (either as a longer tube or a larger tube number, depending upon design considerations) to account for possible contingencies.

Heat Transfer Length

The determination of the heat transfer length of the unit is the object of the thermal sizing. This is accomplished through the use of the energy equation:

$$q_{\text{region}} = UA (\text{LMTD}) \quad (\text{A-1})$$

where:

q_{region} = heat transferred in the region (Btu/hr)

U = overall heat transfer coefficient (Btu/hr-ft²-°F)

A = heat transfer area (ft²)

LMTD = log mean-temperature difference (°F)

An energy balance is performed on the evaporator unit to find the heat transferred per region. That is (see Figure A.1),

$$\text{GENERAL: } q_{\text{total}} = M_{\text{fluid}} C_p (\Delta T_{\text{preheat}} + \Delta T_{\text{boiling}} + \Delta T_{\text{superheat}}) \quad (\text{A-2})$$

$$\text{WATER: } q_{\text{total}} = M_{\text{water}} \left[(h_{\text{sat liquid}} - h_{\text{inlet}}) + h_{\text{fg}} + (h_{\text{out}} - h_{\text{sat vapor}}) \right] \quad (\text{A-3})$$

$$\text{SODIUM: } q_{\text{total}} = M_{\text{sodium}} C_p [(T_2 - T_{\text{inlet}}) + (T_4 - T_2) + (T_{\text{outlet}} - T_4)] \quad (\text{A-4})$$

The overall heat transfer coefficient is specified by the summation of the thermal resistances as follows: water side convection (R_{water}), tube wall conduction (R_{wall}), interfacial contact conductance and helium grooves ($R_{\text{interface}}$), sodium side conduction (R_{sodium}) and a fouling factor (R_{fouling}), or:

$$U = \frac{1}{R_{\text{sodium}} + R_{\text{wall}} + R_{\text{water}}} \quad (\text{A-5})$$

FIGURE A.1

IDEALIZED EVAPORATOR HEAT TRANSFER REGION

SCHEMATIC

where

$$R_{\text{sodium}} = \frac{1}{h_{\text{sodium}}} + \frac{1}{h_{\text{fouling sodium}}}$$

$$R_{\text{wall}} = \frac{D_o \ln(D_o/D_j)}{2K_{\text{wall}}} + \frac{D_o}{D_j} \frac{1}{h_{\text{interface}}} + \frac{D_o \ln(D_j/D_i)}{2K_{\text{wall}}}$$

$$R_{\text{water}} = \frac{D_o}{D_i} \frac{1}{h_{\text{water}}} + \frac{D_o}{D_i} \frac{1}{h_{\text{fouling water}}}$$

The convection coefficients, h_{water} and h_{sodium} , and the interface conduction, $h_{\text{interface}}$, are discussed in more detail later. No fouling factors were assumed in this analysis.

The heat transfer area, A , is:

$$A = n \pi D_o L \quad (\text{A-6})$$

where

n = tube number (-)

D_o = outer tube diameter (ft)

L = tube length (ft)

The log-mean temperature difference per region is defined as:

$$(\text{LMTD}) = \frac{(T_{\text{sodium out}} - T_{\text{water in}}) - (T_{\text{sodium in}} - T_{\text{water out}})}{\log \left[\frac{(T_{\text{sodium in}} - T_{\text{water out}})}{(T_{\text{sodium out}} - T_{\text{water in}})} \right]} \quad (\text{A-7})$$

The tube length can then be determined by combining equations (A-1) through (A-7) to yield:

$$L_{\text{region}} = \frac{[M_{\text{water}} (h_2 - h_1)]_{\text{region}}}{\pi D_o n (U_{\text{region}}) (\text{LMTD})_{\text{region}}} \quad (\text{A-8})$$

with

$$L_{\text{total}} = \sum_{i=1}^{i=n} L_i \quad (\text{A-9})$$

Tube Number

The tube number can be arbitrarily selected although usually it is based on the maximum velocity of the steam, $V_{\max}$, occurring at the unit's outlet. The tube number can then be evaluated by using Equation (A-4) in combination with the mass balance:

$$M_{\text{water}} = n \rho A_{\text{flow}} V_{\max} \quad (\text{A-10})$$

to yield:

$$n = \frac{q_{\text{total}}}{(\rho A_{\text{flow}} V_{\max}) C_p (T_{\text{sodium out}} - T_{\text{sodium in}})} \quad (\text{A-11})$$

The maximum steam velocity is checked both in the downcomer and annulus using formula (A-10) with the respective flow areas:

$$A_{\text{downcomer}} = \frac{\pi}{4} (D_i)^2 \quad (\text{A-12})$$

$$A_{\text{annulus}} = \frac{\pi}{4} (D_o^2 - D_i^2) \quad (\text{A-13})$$

Heat Transfer Coefficients(a) Sodium

Nusselt number relationships for sodium were taken from Reference A-1 and the following shell side relationship was used:

$$Nu = 6.66 + 3.126 \left(\frac{P}{D}\right) + 1.184 \left(\frac{P}{D}\right)^2 + 0.0155 (\bar{\Psi} Pe)^{0.86} \quad (\text{A-14})$$

where

$$\bar{\Psi} = 1 - \frac{1.82}{Pr(\epsilon_M/\nu)_{\max}^{1.4}}$$

$(\epsilon_M/\nu)_{\max} = f(\text{Re, channel geometry})$, and was evaluated from an equation curve-fitted to Fig. 5.2 in Ref. A-1

$$(\epsilon_M/\nu)_{\max} = \frac{1.7}{(P/D)} \exp(-2.95 + .311 \ln(\text{Re}) + .0257 (\ln \text{Re})^2)$$

$$Pe = \text{Peclet number} = \text{RePr}$$

(b) Interfacial Conductance

The interfacial conductance is the effective conductance for both the helium grooves, h_{groove} , and the contact conductance at the interfacial gap, h_{gap} . The value of h_{gap} is taken from Reference (A-2) and h_{groove} is based on conduction across the helium only as:

$$h_{\text{groove}} = \frac{K_{\text{helium}}}{l_{\text{groove}}} \quad (\text{A-15})$$

where

K_{helium} = helium conductance (Btu/hr-ft-°F)

l_{groove} = height of groove (ft)

For parallel heat transfer paths, the overall conduction across the gap is:

$$h_{\text{interface}} = b_1 h_{\text{gap}} + b_2 h_{\text{groove}} \quad (\text{A-16})$$

where

b_1 = percentage of area governing h_{gap} phenomena

b_2 = percentage of area governing h_{groove} phenomena

(c) Water

The correlation recommended by McAdams, (A-3) was used when all liquid or all vapor phase was present (i.e., preheater, superheater, or reheater).

$$\text{Nu} = 0.023 \text{ Re}^{.8} \text{ Pr}^{.4} \quad (\text{A-17})$$

For saturated boiling, the Jens-Lottes correlation^(A-4) was used for a quality, X , less than 0.3.

$$h_{\text{sat}} = (q/A) / \Delta T_{\text{sat}} \quad (\text{A-18})$$

with

$$\Delta T_{\text{sat}} = \frac{60 (q/A \times 10^{-6})^{1/4}}{e^{(P/900)}}$$

where

q/A = heat flux (Btu/hr-ft²)

P = pressure (psia)

The Tong correlation(A-4)for film boiling was used when the quality, x , was between 0.3 and 1.0.

$$Nu_v = 0.005 Re_v Pr_v^{0.5} \quad (A-19)$$

where the vapor material properties are used in this equation and are evaluated at the wall temperature.

REFERENCES FOR APPENDIX "A"

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- A-3 W. H. McAdams, "Heat Transmission", McGraw-Hill Book Company, New York, Third Edition, 1954, p. 219.
- A-4 L. S. Tong, "Boiling Heat Transfer and Two-Phase Flow", John Wiley and Sons, Inc., New York, 1966, pp. 111-135.

APPENDIX "B"

HYDRAULIC SIZING EQUATIONS

APPENDIX B

Hydraulic Sizing Equations

The heat losses or pressure drop losses include fluid friction, tube entrance and exit conditions (tube side only), turning (tube side only) and egg-crate losses (shell side only). Inlet and exit nozzle losses are not considered. These losses are summarized in Table B.1

The total pressure drop, ΔP_{total} , is defined as:

$$\Delta P_{\text{total}} = \Delta P_{\text{friction}} + \Delta P_{\text{minor}} \quad (\text{B.1})$$

(a) Single Phase Fluids

The single phase (all liquid or vapor) friction pressure loss, $\Delta P_{\text{friction}}$, is determined by the formulation:

$$\Delta P_{\text{friction}} = f \left(\frac{L}{D_H} \right) \frac{\rho V^2}{2g_c} \quad (\text{B.2})$$

Table B.1

Pressure Drop Losses

Tube Side

1. Contraction loss from upper plenum into inner bayonet tube.
2. Frictional flow loss in the inner bayonet tube.
3. Turnaround loss at bottom of main tube.
4. Frictional flow loss in annulus between main tube and outer bayonet tube.
5. Expansion loss from annulus to lower plenum.

Shell Side

1. Frictional flow loss along primary tubes.
2. Expansion and contraction losses due to tube supports (egg crates).

The friction factor, f , is based on the Moody chart and given as:

$$f = 0.0055 \left[1 + \left(20,000 \left(\frac{e}{D_H} \right) + \frac{10^6}{Re} \right)^{1/3} \right] \quad (B.3)$$

where

Re = Reynolds number and with an assumed relative roughness, e/D ,
for smooth tubes of:

$$\frac{e}{D_H} = 1 \times 10^{-4}$$

(b) Two Phase Liquids

In the boiling section of the evaporator, a homogeneous model (B-1)
was used in calculating the pressure drops in the bubbly flow region,
 $x \leq 0.3$,

$$\Delta P = f \left(\frac{L}{D} \right) \frac{\rho_{av} V_{av}^2}{2g_c} \quad (B.4)$$

where

f = Moody friction factor

$$\rho_{av} = \rho_{liquid} (1 - \alpha) + \rho_{vapor} \alpha$$

α = void fraction

$$V_{av} = \frac{M_{water}}{\rho_{av} \Delta_{flow}}$$

When the quality is larger than 0.3 and an annular flow regime is dominant, (B-1)
the friction loss is based on the vapor flow rate with a correction for void
fraction (a separated flow model):

$$\Delta P = C_{void} \left(\frac{L}{D} \right) \frac{\rho_{vapor} V_{vapor}^2}{2g_c} \quad (B.5)$$

with

$$C_{void} = 0.005 \left[\frac{1 + 75(1 - \alpha)}{\alpha^{5/2}} \right]$$

(c) Minor Losses

The minor losses are based on the formulation:

$$\Delta P_{\text{minor}} = C_1 \frac{\rho_1 V_1^2}{2g_c} + C_2 \frac{\rho_2 V_2^2}{2g_c} + \dots + C_n \frac{\rho_n V_n^2}{2g_c} = \sum_{i=1}^n C_i \frac{\rho_i V_i^2}{2g_c} \quad (\text{B.4})$$

where the minor losses coefficients, C_i , for the various situations are listed in Table B.2.^(B-2-3) For the case of the egg crates placed on the shell side, the pressure drop was evaluated as a sudden contraction and then expansion. The area change through the egg crates was evaluated through geometric parameters as:

$$A_{\text{egg crate}} = D_o \left(P - \frac{\pi D_o}{4} \right) \quad (\text{B.5})$$

where

P = tube pitch (ft)

D_o = outer diameter of the tube (ft)

Egg crates were assumed to be placed every three feet.

Table B.2

Hydraulic Loss Coefficients

$$\beta = \frac{\text{smaller cross-sectional area}}{\text{larger cross-sectional area}}$$

<u>Flow Condition</u>	<u>C_i</u>
Sudden contraction ⁽¹⁶⁾	0.45 (1- β)
Sudden expansion ⁽¹⁶⁾	$\left(\frac{1}{\beta} - 1 \right)^2$
180° turnaround	2.0
Pipe entrance ⁽¹⁷⁾	0.8
Pipe exit ⁽¹⁷⁾	0.3

REFERENCES FOR APPENDIX "B"

- B-1 G. B. Wallis, "One Dimensional Two-Phase Flow", McGraw-Hill Book Company, New York, 1969, pp. 17-38, pp. 315-372.
- B-2 R. B. Bird, W. E. Steward, E. N. Lightfoot, Transport Phenomena, John Wiley & Sons, Inc., New York, 1960.
- B-3 Interim Report, LMFBR Steam Generator Development, AEC Contract AT(30-1)-4230, C-E Contract No. 7670, Task 1.3, "Preliminary Design and Analysis", Appendix, p. 39, March 1971.

APPENDIX "C"

LOAD CONTROL CONTACT PRESSURE

APPENDIX C

Figure C-1 illustrates the duplex tubes in contact and as individual components. Pressures required for consideration of rupture are also shown. The average radial stress in each tube (C-1 b,c) is:

$$(b) \quad \bar{\sigma}_{r_1} = -\frac{(p_1 + \bar{p}_{c_{LC}})}{2} \quad (C-1)$$

$$(c) \quad \bar{\sigma}_{r_2} = -\frac{(\bar{p}_{c_{LC}} + p_2)}{2} \quad (C-2)$$

Average radial stress for both tubes combined (assumed a single tube of thickness $t_1 + t_2$) is:

$$(a) \quad \bar{\sigma}_{r_c} = -\frac{(p_1 + p_2)}{2} \quad (C-3)$$

From Figure C-2, $\bar{\sigma}_{r_c}$ is given by,

$$\bar{\sigma}_{r_c} = \frac{\bar{\sigma}_{r_1} t_1 + \bar{\sigma}_{r_2} t_2}{t_1 + t_2} \quad (C-4)$$

Substituting (C-1, C-2, and C-3) into (C-4) and solving for $\bar{p}_{c_{LC}}$ gives:

$$\bar{p}_{c_{LC}} = \frac{p_1 t_2 + p_2 t_1}{t_1 + t_2} \quad (C-5)$$

FIGURE C-1. DETERMINATION OF $\bar{p}_{CLC}$

BOTH TUBES IN CONTACT

INNER TUBE

OUTER TUBE

FIGURE C-2

RELATION BETWEEN RADIAL STRESS COMPONENT AVERAGED
OVER BOTH TUBES COMBINED ($\bar{\sigma}_{rc}$) TO THAT OVER
EACH TUBE CONSIDERED SEPERATELY ($\bar{\sigma}_{r1}, \bar{\sigma}_{r2}$)

APPENDIX "D"

HT-9 STRESS ALLOWABLES

APPENDIX D

Since HT-9 is not an ASME Code approved material, allowable stresses were derived from available material properties given in Table D-1 and Figure D-1 (Ref. D-1). The allowable stresses are listed in Table D-2. S_o and S_m were taken directly from Table D-1. Values of S_t were needed for 200,000 hours but Table D-2 contained S_t values only for 100,000 hours. Therefore, S_t for 200,000 hours was calculated as 0.67 times the stress to rupture in 200,000 hours from Figure D-1. This calculation assumed that the rupture stress controlled instead of the stress for 1% creep strain. This assumption seemed justified since the S_t values in Table D-1 corresponded with 0.67 times the stress to rupture in 100,000 hours as given in Figure D-1.

TABLE D-1

ALLOWABLE STRESS HT-9 (TUBING)

TEMP.		S_u	S_y	S_o	S_m	S_t	S_{mt}	S_g
100		107.0	71.0	26.7	35.7		35.7	71.0
500			59.3	↓	↓		↓	59.3
600			56.0	↓	↓		↓	56.0
650			54.3	↓	35.7		35.7	52.8
700			52.7	↓	35.1		35.1	48.8
750			51.0	↓	34.0		34.0	44.8
800			48.8	↓	32.5		32.5	40.8
850			46.6	↓	31.1		31.1	36.8
900			44.3	↓	29.5		29.5	32.8
930			43.0	26.7	28.7	24.3	24.3	30.4
950			42.2	26.4	28.1	23.0	23.0	28.8
970			41.2	25.8	27.5	21.6	21.6	27.0
985			40.6	25.4	27.1	20.0	20.0	25.0
1005			39.6	24.8	26.4	18.1	18.1	22.6
1020			39.0	24.4	26.0	16.1	16.1	20.1
1040			38.0	23.8	25.3	14.3	14.3	17.9
1060			37.2	23.5	24.8	12.4	12.4	15.5
				$\frac{5}{8} S_y$	$\frac{2}{3} S_y$	Δ for 1% CREEP in 100,000 hrs $\cdot 0.7 \Delta$ for rupture in 100,000 hrs		$S_{y_{AVG T}}$
				OR	OR			OR
				$\frac{1}{4} S_u$	$\frac{1}{3} S_u$			$1.25 S_{t_{MCH}}$

FIGURE D-1

TABLE D-2

STRESS ALLOWABLES FOR HT - 9 (KSI)

TEMPERATURE (°F)	S _o	S _m	S _t @ 2x10 ⁵ HOURS
100	26.7	35.7	23.4
500	26.7	35.7	23.4
850	26.7	31.1	23.4
900	26.7	29.5	23.4
930	26.7	28.7	23.4
1005	24.8	26.4	17.4
1060	23.5	24.8	11.4

REFERENCES FOR APPENDIX D

- D-1 HT-9 Material Property Data Received from J. W. Bass of C-E,
NCD at Chattanooga.

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APPENDIX "E"

RESPONSE TO ERDA STRUCTURAL COMMENTS

Appendix E

This appendix addresses two questions posed by ERDA ^(E-1) concerning:

- (1) The duplex tube thicknesses for the evaporator if 2-1/4 Cr-1 Mo material is used instead of HT-9.
- (2) Stress concentration effects of helium leak detection grooves.

2-1/4 Cr-1 Mo Evaporator Tubes

Minimum thicknesses were calculated for the evaporator using Equations 4.2-4, -44, -9 and -45 as summarized in Section 4.2. The combination of parameters (see Section 4.2) was that used for the original recommended thicknesses. In changing material from HT-9 to 2-1/4 Cr-1 Mo, different ASME code stress allowances are required. Table E-1 contains an outline of the calculations. Comparing these thicknesses with those in Table for evaporator, it is concluded that 2-1/4 Cr-1 Mo requires thicknesses approximately double those required for HT-9.

Stress Concentration at Grooves

Even though the rupture criteria involve only the average (P_m , P_L) or linear (P_b , Q) stress distribution characteristics, the stress concentration effect of the grooves should be considered. Since the grooves introduce a local reduction in the wall thickness from t_2 to t_G , the stress intensities for the rupture criteria were computed from stresses based on the smaller thickness, t_G . Thus, the stress concentration effect of the grooves was included in the rupture evaluation in this report. However, in respect to other failure criteria, the effect of the grooves is believed to be most significant in regard to fatigue.

In fatigue, the effect of the grooves is to create a local peak stress concentration. The major effort in a fatigue evaluation lies in calculating the peak stress ranges for each variable loading. Because of the amount of analysis required, and the secondary effect of fatigue on tube sizing, a detailed fatigue evaluation was not performed. It was considered beyond the scope of a preliminary structural evaluation of the duplex tube concept. How-

Table E-1

Calculation Summary for 2-1/4 Cr-1 Mo
Evaporator Duplex Tube Thicknesses
Satisfying Rupture Criteria

Case Considered: (a) Design Conditions Excluded

(b) $\bar{p}_{c_{lc}} = p_{GAP}$ EQ #4.2-18

(c) $p_{LEAK} = p_{H_2O}$ EQ #4.2-19

ASME Code Stress Allowables (ksi) from Reference (4):

$$S_t^A = 12.3, \quad S_m^A = 17.6, \quad S_t^{Bc} = 10.7, \quad S_m^{Bc} = 17.4$$

Intermediate Results:

$$S_A = 4.73, \quad S_{18} = 4.06, \quad S_{cc} = 3.69, \quad S_{2c} = 7.37$$

Thicknesses (inches):

Reference (uniform) Design

Alternate (non-uniform) Design

$$\left. \begin{array}{l} t_1^u = t_1^{TS} = 0.122 \\ t_2^u = 0.118 \end{array} \right\} \text{MINIMUM THICKNESSES} \left\{ \begin{array}{l} t_1^N = 0.071, \quad t_1^{TS} = 0.122 \\ t_2^N = 0.111 \end{array} \right.$$

$$\left. \begin{array}{l} t_1^u = t_1^{TS} = \underline{0.131} \\ t_2^u = \underline{0.132} \end{array} \right\} \text{AVERAGE THICKNESSES (INCLUDING CORROSION AND TOLERANCE)} \left\{ \begin{array}{l} t_1^N = \underline{0.078}, \quad t_1^{TS} = \underline{0.131} \\ t_2^N = \underline{0.125} \end{array} \right.$$

ever, the following assessment was made to indicate the potential for fatigue problems and the significance of the grooves.

Therefore, thermal transients are potentially the most significant source of fatigue damage. The set of thermal transients in Table E-2 is considered representative of the large plant conditions. Based on experience in thermal transient fatigue analysis, it is judged that the total fatigue damage due to the combinations of severity and number of occurrences shown in this table would not be significant. Therefore, regardless of the stress concentration at the grooves, fatigue will probably not be a problem because either the loadings are not severe enough or their damage is minimized by minor design changes. This should be varified in future development of the Duplex Tube.

TABLE E-2 (Ref. E-2)

DEMONSTRATION PLANT STEAM GENERATOR DESIGN TRANSIENTS

	<u>Event</u>	<u>Location</u>	<u>Input Temps., °F</u>		<u>No. of Occur- ences</u>	<u>Max. Rate of Tempera- ture Change</u>
			<u>Start</u>	<u>End</u>		
	<u>Normal Conditions</u>					
(1)	Heatup & Cooldown	Shell	Ambient	400°F	10	+ 5°F/hr.
			400°F	Ambient	10	- 5°F/hr.
(2)	Startup	Sodium Inlet Nozzle @ SH	400°F	924°F	600	+100°F/hr.
		FW Inlet Nozzle @ Drum	100°F	370°F		
		Evap. Water Inlet Nozzle	400°F	655°F		
(3)	Shutdown	Sodium Inlet Nozzle @ SH	924°F	400°F	250	-100°F/hr.
		FW Inlet Nozzle @ Drum	370°F	100°F		
		Evap. Water Inlet Nozzle	655°F	400°F		
(4)	Load Changes					
	40 - 100% Power					
	Normal	Sodium Inlet Nozzle @ SH	924°F	960°F	9000	1.8°F/min.
	Power	FW Inlet Nozzle @ Drum	370°F	460°F		4.5°F/min.
	Increase					
	Normal	Sodium Inlet Nozzle @ SH	960°F	924°F	9000	-1.8°F/min.
	Power	FW Inlet Nozzle @ Drum	460°F	370°F		-4.5°F/min.
	Decrease					
	<u>Upset Conditions</u>					
(5)	Normal Scram	Sodium Inlet Nozzle @ SH	960°F	1020°F	350	+ .2°F/sec.
		FW Inlet Nozzle @ Drum	460°F	460°F		No change
(6)	Loss of Recirc.	Sodium Inlet Nozzle @ SH	960°F	960°F	2/pump	{ No change }
	Pump Flow	FW Inlet Nozzle @ Drum	460°F	460°F		
	<u>Emergency Conditions</u>					
(7)	Sec. Na Pump	Sodium Inlet Nozzle @ SH	960°F	960°F	3/loop	{ No change }
	Seizure With	FW Inlet Nozzle @ Drum	460°F	460°F		
	Scram					
(8)	Pri. Sodium	Sodium Inlet Nozzle @ SH	960°F	660°F	3/loop	-3°F/sec.
	Pump Seizure	FW Inlet Nozzle @ Drum	460°F	460°F		No change
	With Scram					
(9)	Activation of	Sodium Inlet Nozzle @ SH	960°F	1020°F	2/module	+ .2°F/sec.
	Na/H ₂ O reaction	FW Inlet Nozzle @ Drum	460°F	460°F		No change
	protection system					

REFERENCES FOR APPENDIX E

- E-1 "Commitments and Agreements" Made at Meeting at Chattanooga
between AEC (ERDA) and C-E, February 1, 1975.

APPENDIX "F"

RESPONSE TO ERDA "NO LEAK DETECTION" COMMENT

DUPLEX TUBE DESIGNS WITHOUT INTERMEDIATE FLUID LEAK DETECTION

Duplex tube steam generators with minimum or no intermediate fluid leak detection have been proposed and operated successfully. The EBR-II system requires only periodic inspection between tubesheets to look for gross evidence of leaks. Recently, it has been proposed that preformed duplex tubes be used with only one tubesheet at each end and that the outer and inner tubes be welded, respectively, to the inside and the outside of the tubesheet. In this arrangement, no attempt is made to detect single leaks into the gap between tubes. Since this gap is sealed off and since each tube is designed to sustain full pressure, it is claimed that a single such leak will not need any attention for the remaining life of the unit.

Two principal advantages are claimed for this design: (1) the duplex tube will stop crack propagation and thus greatly reduce the probability of a single fault leading to a major sodium-water reaction; (2) with no gap leak detection and with a double barrier, the down-time for tube plugging due to small leaks will be much less than for either a single wall design or for a duplex design with gap leak detection. The following general reasoning is applied in defending the second advantage:

1. Based on judgement and limited experience with well designed and fabricated sodium steam generators, the expected failure rate (small leak in a single tube) is about 0.1/year/module for CRBR sized modules (~ 760 tubes/module).
2. A duplex tube steam generator will have about three times as many tubes as a single wall generator; thus, assuming the same failure rate per tube as a single wall steam generator, the failure rate will be 0.3/year/module or 2.7/year for a nine-module reactor system.
3. If intermediate fluid leak detection is employed, the reactor down-time for the duplex design would be about 800 hr/yr, assuming 300 hours to plug a tube and no reactor operation during tube plugging. This is predicted on the requirement that tubes be plugged whenever a leak is detected because of the effects of intercommunication of the leaking fluid within detection spaces.
4. If no intermediate fluid leak detection is employed and if each intermediate fluid space is sealed to prevent intercommunication of leaks, then the down-time for repair of small leaks is negligible. This very low down-time prediction is based on the assumption that each tube has the same failure probability as a single wall design and that there are no common failure mechanisms for duplex tubes. Thus, the probability of failure of a duplex tube is the product of the failure

probability of each tube, i.e., $1.3 \times 10^{-4} \times 1.3 \times 10^{-4} = 1.7 \times 10^{-8}$ /duplex tube/year. CRBR would have about 10^4 duplex tubes giving an estimated down-time of $1.7 \times 10^{-4} \times 300 = .05$ /hours/year.

This line of thought leads to the following conclusion for down-time of an LMFBR plant like CRBR:

<u>Steam Generator Design</u>	<u>Down-time (hr/yr)</u>
1. Single wall	270
2. Duplex with intermediate leak detection	800
3. Sealed duplex without intermediate leak detection	0.05

The down-times for the first two designs are high enough to represent a sufficient penalty to make the third design look attractive relative to the others.

There are two major faults in this line of reasoning: (1) the assumption of no common failure mechanism for duplex tubes, and (2) the assumption that each tube in a duplex tube design has the same failure probability as a tube in a single wall design.

Several common failure mechanisms exist for duplex tubes. A steam side leak will allow water into the gap between tubes and, given time, this could lead to corrosion failure of the outer tube by several mechanisms, including: retention of tube cleaning solutions in the gap, interaction with contaminants in the gap, and concentration of feed water contaminants in the gap by evaporation after each start-up. Also, although the duplex tube would stop fatigue crack propagation from one tube to the other, a second fatigue crack could occur in the same area of the second tube with a relatively high probability if the two fatigue cracks were initiated by a common failure such as the local failure of a tube support leading to excessive tube vibration.

These common failure mechanisms require considerable time between the leak in the first tube and the leak in the second tube. Thus, an efficient leak detection system should prevent most common failures of duplex tubes.

The individual failure probability of single tubes in duplex tube design is not necessarily the same as the failure probability of the corresponding single wall design. Furthermore, important differences exist among the failure probabilities of various single wall steam generator designs.

The most common cause of a steam generator leak is a crack in a weld, especially a crack in a tube-to-tubesheet weld. The reliability of these welds depends upon the design of the joint, the quality of the welding technique, the type and quality of weld inspection and the loadings on the weld. For a preformed duplex tube welded to a single tubesheet, the

the weld joint design and the type and quality of weld inspection are, of necessity, inferior to those of a single wall full penetration butt weld for a corresponding steam generator design. Further, as compared to a hockey stick or J-tube design, the tube arrangement and tube-to-tubesheet joint of the C-E bayonet tube design is superior in that there are no bending loads on the joint or anywhere on the tubes, in that the joint is reenforced and in that a multi-pass welding technique employing filler material is used.

It is not now possible to accurately quantify the above reliability considerations; however, we judge that the following relative estimates are not unreasonable:

<u>Steam Generator Design</u>	<u>Relative Failure Probability per Tube</u>
1. Single wall hockey stick or J-tube, butt welded to tubesheet	1.0
2. Preformed duplex hockey stick or J-tube double tubesheets at each end for leak detection, outer tube fillet weld, inner tube seal weld	1.5 (outer tube) 1.5 (inner tube)
3. Preformed duplex hockey stick or J-tube, single tubesheet at each end, outer tube fillet weld, inner tube seal weld	1.5 (outer tube) 1.5 (inner tube)
4. Duplex bayonet tube, double upper tubesheet, outer tube-to-tubesheet joint, reenforced, multi-pass weld, inner tube full penetration corner weld, butt welded end caps	0.5 (outer tube) 0.7 (inner tube)

Because of the common failure mechanisms described above for duplex tubes without leak detection, it is our view that a reliability improvement over a single wall design of no more than a factor of 10 can be achieved. Also, the sealed duplex tube design must have thicker tube walls than a duplex tube with leak detection because each tube must be sized to take the full steam pressure for essentially the full life of the unit. Using this correction and the above revised estimates for reliability, the following values of down-time were calculated for a plant similar to CRBRP with five types of steam generators. (A factor of 1.2 was used to adjust for the slightly larger surface area requirement of a bayonet tube design relative to a J-tube or hockey stick design.)

<u>Steam Generator Design</u>	<u>Down-time (hr/yr)</u>
1. Single wall hockey stick or J-tube	270
2. Duplex hockey stick or J-tube with intermediate leak detection	1100
3. Sealed duplex hockey stick or J-tube without leak detection	130
4. Duplex bayonet tube with intermediate leak detection	540
5. Sealed duplex bayonet tube without leak detection	60

Of course, the absolute magnitude of the above estimates are dependent on basic reliability estimates which are quite uncertain; thus, one should not place much emphasis on the absolute number of hours predicted. It is the relative comparisons that are significant in this analysis.' The main point is that the down-time improvement that should be associated with the sealed, no-leak detection, duplex tube concept is less than a factor of 10 rather than a factor of 10,000 as claimed by some of its proponents.

This improvement in down-time for the sealed duplex tube steam generator units must be weighed against their increased probability of a major sodium-water reaction compared to a duplex tube arrangement with intermediate fluid leak detection. Also, the added costs and decreased system reliability associated with the sodium-water reaction products systems and steam dump systems required for the sealed duplex concept must be accounted for in comparing it with the alternate system using intermediate fluid leak detection. At this time it is not possible to make a clear-cut economic distinction among the several systems proposed. It is our view, however, that there is a very good possibility that the overall economics of the duplex bayonet tube with leak detection will compare quite favorably with other proposed designs. This, together with virtual elimination of the possibility of a major sodium-water reaction, makes this a very attractive design.