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CRBRP STEAM-GENERATOR DESIGN EVOLUTION

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ABSTRACT

The overall design of the CRBRP Steam Generator is briefly discussed. Two areas of particular concern are highlighted and considerations leading to the final design are detailed:

Differential thermal expansion between the shell and the steam tubes is accomodated by the tubes flexing in the curved section of the shell. Support of the tubes by the internals structure is essential to permit free movement and minimize tube wear. Special spacer plate attachment and tube hole geometry promote unimpeded axial movement of the tubes by allowing individual tubes to rotate laterally and by providing lateral movement of the spacer plates relative to the adjacent support structure.

The water/steam heads of the CRBRP Steam Generator are spherical heads welded to the lower and upper tubesheets. They were chosen principally because they provide a positively sealed system and result in more favorable stresses in the tubesheets when compared to mechanically attached steamheads.

INTRODUCTION

Historically, steam generators have been critical components in power plants. They are exposed to thermal transients which produce high stresses in the affected parts or cause relative motion between parts which can result in wear. Internal parts are often susceptible to flow-induced or mechanical vibration which may result in fretting and wear. The effect of this is always undesirable and can shorten the life of the units. Design solutions to these problems have always been difficult, especially for components designed for the sodium environment in Liquid Metal Fast Breeder Reactor (LMFBR) systems.

The high operating temperatures and fast transients, coupled with the efficient thermal conductance of sodium create extremely severe design conditions for LMFBR components.

This paper highlights two such design challenges in the development of the Steam Generators for the Clinch River Breeder Reactor Plant (CRBRP).

DESCRIPTION OF THE CRBRP STEAM GENERATOR

The CRBRP includes a reactor core contained in a reactor vessel which is connected to three primary loops. Heat from the reactor is transported by sodium to each of the three Intermediate Heat Exchangers (IHX). From each of the IHXs an intermediate sodium loop connects to the steam generator system. There is no direct connection between the primary and the intermediate sodium loops (Figure 1).

Each loop is equipped with a steam generator system consisting of three identical modules: two evaporators working in parallel and one superheater which is in series with the two evaporators.

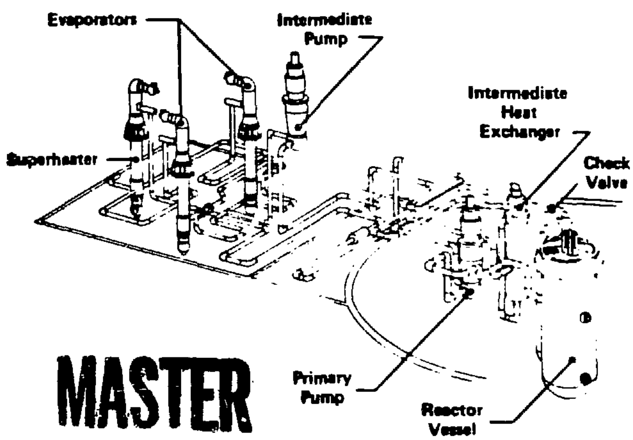

FIG. 1 CRBRP HEAT TRANSPORT SYSTEM

Design Description

The design of the CRBRP Steam Generators began in 1973 and was based on earlier development work conducted by Atomics International (AI), Division of Rockwell International Corporation. The resulting prototype steam generator, shown in Figure 2, was completed in 1981 and is presently being tested in the Sodium Components Test Installation (SCTI) at the Energy Technology Engineering Center (ETEC) in Canoga Park, California. The L-shaped component is particularly suited to accomodate the differential thermal expansion between the shell and tube bundle. A shroud around the tube bundle provides support for the tube support plates and is itself supported from the shell. A liner in the sodium inlet plenum protects the thick-walled pressure boundary from the effects of thermal transients. The liner also supports an elbow shroud which has a similar function for the pressure boundary in that area and in addition provides for tube supports in the elbow region, as will be discussed in more detail further-on in this paper. The steamheads in the prototype were also configured as spool pieces, bolted onto the face of the tubesheets and sealed by Flexitallic gaskets.

The Plant Units shown in Figure 3 are the result of further development by Westinghouse Electric Corporation, Advanced Energy Systems Division (AESD). Two of the changes made to the prototype design are the subject of this paper, namely:

- Replacement of the bolted spool pieces by all-welded steamheads, and
- Changes to the tube support structure to enhance the capability for relative movement of the steamtubes.

Operating Characteristics

For the purpose of these discussions, only the steady state operating conditions will be listed here:

100% Power/Loop	325	MWt		
Sodium Flow/Loop	13.5×10^6	lb/hr	$(6.12\times10^6$	kg/hr)
Water/Steam Flow/Evap.	1.1×10^6	lb/hr	$(.5\times10^6$	kg/hr)
Steam Flow Rate/S.H.	1.1×10^6	lb/hr	$(.5\times10^6$	kg/hr)
Recirculation Ratio	2	: 1		
Sodium Inlet Temp.	922	°F	(512	°C)
Sodium Inlet Pressure	193	psig	$(1.33\times10^6$	Pa)
Steam Outlet Temp.	906	°F	(485	°C)
Steam Outlet Pressure	1550	psig	$(10.6\times10^6$	Pa)
Feedwater Temperature	548	°F	(286	°C)
Feedwater Inlet Pres.	2024	psig	$(13.9\times10^6$	Pa)

In addition, a great number of normal, upset, emergency and faulted events consistent with the definition in the ASME B&PV Code must be considered including some with temperature transients of 120°F (66°C) per minute. It is with this background that the design modifications should be viewed.

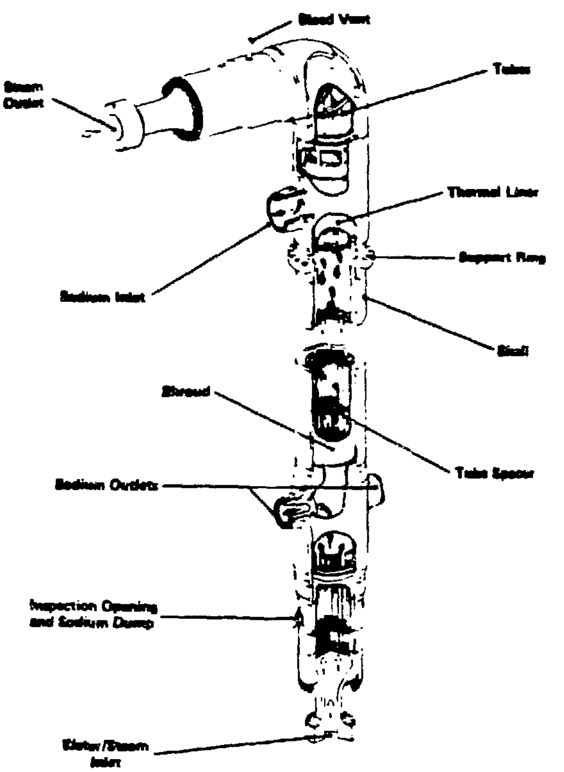

FIG. 2 CRBRP STEAM GENERATOR - PROTOTYPE

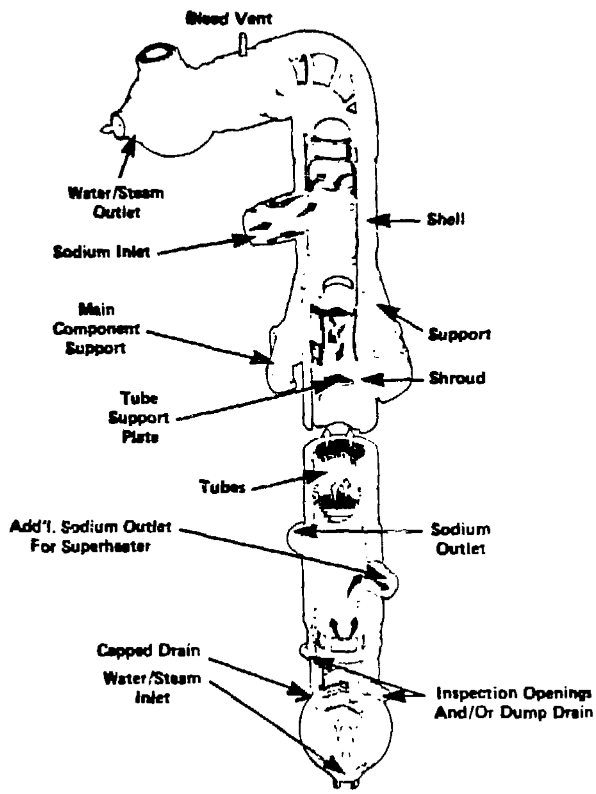

FIG. 3 CRBRP STEAM GENERATOR - PLANT UNIT

STEAMHEAD DESIGN

The design of the steamheads on the CRBRP Steam Generators evolved from bolted on spool pieces on the Prototype to integrally welded spherical heads with manways (Figure 4) on the Plant Units. The rationale for this change is presented in the following paragraphs.

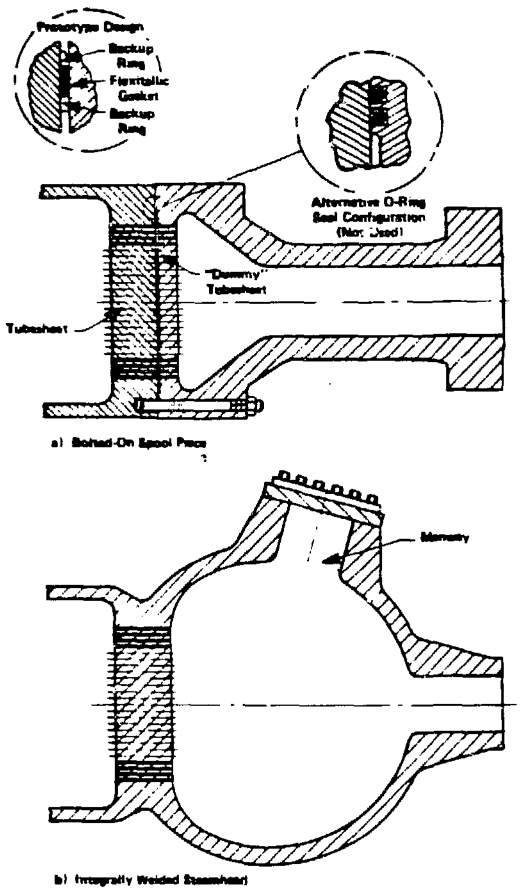

FIG. 4 STEAM HEAD CONFIGURATIONS

Removable Steamheads

Removable steamheads are advantageous for In-Service Inspection (ISI) and maintenance because they allow unlimited access to the surface of the tubesheet for all necessary operations. They also enhance handling, installation and removal of the steam generators since the unit weight is approximately 12 tons ($10.8x 10^3$ kg) less without the steamheads. However the bolted assembly requires mechanical seals and high bolt loads to seat the seals. The diameter of the tubesheets to which the steamheads are attached is limited by the need to pass the shell closure section over the tubesheet.

The tubesheets for the bolted-on steamheads are characterized by high ligament efficiency in the perforated region, and a relatively weak outer ring due to the O.D. constraint and the presence of 24 - 2.25 inch dia. (5.71 cm) threaded holes. During thermal transients, the perforated region responds rapidly to the temperature changes of the water flowing through the holes in the tubesheet, while the outer region follows much more slowly. The differential expansion/contraction between the rim and the perforated region of the tubesheet results in high stresses in the outer rim. Asymmetrical constraint of the movement of the rim by its connection to the shell results in rotation of the sealing face due to "cupping" or "bulging" of the tubesheet. A thermally matched tubesheet-like structure was provided in the spoolpiece to achieve thermally similar behavior between the two matching faces. However, the steamhead flange does not exhibit a rotation of the sealing face similar to the tubesheet rim because its relative stiffness is considerably greater than that of the perforated region of the dummy tubesheet. As a result, thermal transients will cause relative rotation of the two sealing faces and consequently change the seal compression. If the seal compression at any point during the transient becomes less than the elastic springback of the seal, a leak will occur.

The decision to change from a bolted to an integrally welded steamhead design was based on the inability to assure a leak-free system for all operating conditions, regardless of which type of seal was used. The principal candidate seals, judged to have best potential of satisfying the design requirements, were spiral wound gaskets and metal O-rings in varying applications.

Spiral wound gasket seals. The Prototype design included Flexitallic gaskets between the tubesheets and the steam heads. The use of these spiral wound, graphite impregnated seals for high temperature and pressure applications is widely accepted (1). These seals can tolerate relatively rough surface finishes and tolerate significant relative radial motion of the sealing surfaces. However, the large surface areas of these gaskets require correspondingly high flange loads to achieve the necessary seal pressures. The elastic springback capability of these seals is limited to a maximum of about .008 inch (.20 mm) at the normal operating temperature of 930°F (498°C), even when high strength nickel alloy (Inconel 718) (2) is used. Another characteristic of these seals is that leaks tend to be progressive, i.e. once started, the leak will quickly erode the filler material and the gasket will not reseal when the conditions that caused the leak have passed. The limited axial springback capability and the high load requirement rendered this type of seal unsuitable for CRBRP Steam Generator plant unit application.

Metal O-ring seals. To improve the tubesheet/steamhead seal function, replacement of the gasket with metal O-rings was evaluated. Metal O-rings are commonly used in nuclear and chemical plants for high pressure, high temperature seal applications. Typical nuclear applications are closure head seals for light water reactors operating at 600-700°F (315-371°C) and pressures of about 2500 psig ($17.2x 10^6$ Pa). Chemical plant applications include highly reactive and/or corrosive media at very high pressures (about 5000 psig) ($34.4x10^6$ Pa) and temperatures (about 1000°F) (537°C). Both single ring and double ring applications are commonly used.

O-rings provide good springback capability (up to 0.015"(.38 mm) for a 0.625"(15.8 mm) O.D. section ring) (3) to accomodate flange axial motion, but are less tolerant of radial motion and surface roughness. Springback capability can be enhanced by venting the I.D. of the ring to the pressurized medium, and by utilizing larger cross-sections. Rings may be coated with a soft metal (Silver in the CRBRP Steam Generator application) to improve sealing and resealing capability, however relatively fine seal surface finish (about 32 RMS) is required for adequate sealing. Moderately high seal loads (but significantly less than the gasket loads) are required for the O-rings in the double ring application considered for the CRBRP Steam Generators. The superior springback capability of the O-rings made them the preferred choice of seals for the CRBRP Steam Generator.

Stress in removable steamheads. The high stud preloads required to obtain the necessary seal pressure and to carry the working load result in high bearing stresses on the seal stop surfaces and high stresses in both the male and the female threads. In addition, the threaded holes in the tubesheet reduce the strength of the outer portion of the tubesheet to the extent that the stress-critical ligament is located in the tubesheet rim area. The design to limit the flange bearing stresses has the most direct influence on seal performance, although the tubesheet stresses were also a significant concern. Bearing stresses are limited by the ASME B&PV-*Code* Case N-47 to the yield strength at temperature. Since the preload requirements are fixed by functional considerations, sufficient bearing area must be provided to limit the bearing stress to no greater than approximately 25 ksi (172.3×10^6 Pa), the tabulated yield strength (in N-47) of the tubesheet material at 940°F (504°C).

The design of the seal joint must balance the seal function and stress on the seal surfaces. Seal function, e.g. the amount of flange motion permissible to maintain a seal, can be controlled by providing a relief on one of the mating flange surfaces, so that relative rotation of the flanges occurs about the relief inside diameter. The flange contact area is then bounded by the relief diameter on the outside, and by the outside diameter of the outer O-ring groove (for double configuration), less the area of the bolt holes intersecting this annulus.

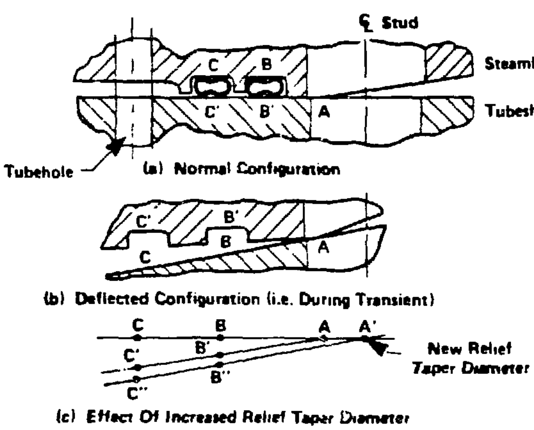

FIG. 5 SEAL ROTATION AS A FUNCTION OF RELIEF TAPER DIAMETER

For the CRBRP Steam Generator, the area inside the bolt circle was limited by the tubehole matrix in the tubesheet; consequently the only remaining parameter to adjust the contact area was the diameter of the relief. The effect of increasing the relief diameter is to increase the degree of relative axial motion of the flanges as shown in Figure 5.

The design tradeoff was to fix the relief diameter at a point where the maximum flange rotation was within seal capability and the bearing stress was less than the allowable. The minimum diameter to satisfy bearing stress allowables is determined in Figure 6. The reference relief diameter was a "best guess" number for which a series of transient seal deflection analyses were performed. Figure 6 shows that a significant increase in the diameter of the relief taper was required to satisfy the bearing stress allowable.

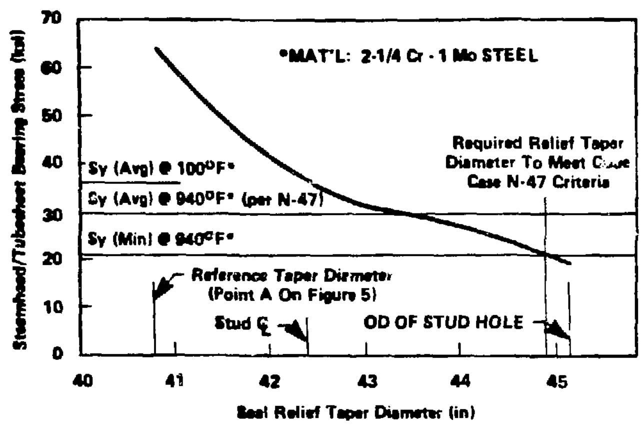

FIG. 6 SEAL SURFACE BEARING STRESS AS A FUNCTION OF RELIEF TAPER DIAMETER (200Klb/Bolt Preload Double O-Ring)

The results of the seal deflection analysis for the reference relief taper diameter (Figure 6) for several transients are shown as lines O, A and B on Figure 7. These results are based on detailed finite element models of the seal area. Line O is a duty cycle transient which causes the flanges to rotate inward, resulting in seal compression greater than the compression due to preload. Lines A and B are a normal and

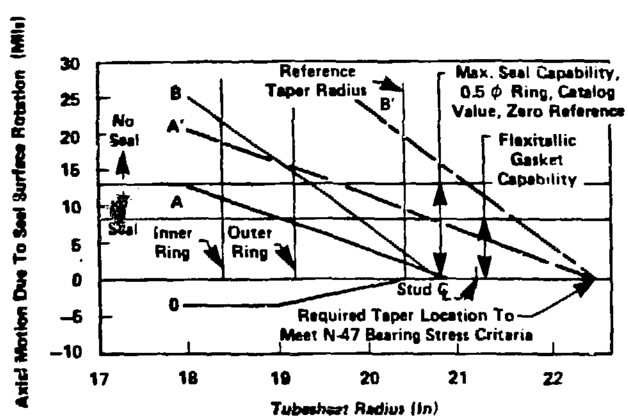

FIG. 7 SEAL MOTION AS A FUNCTION OF RELIEF TAPER LOCATION DOUBLE O-RING CONFIGURATION

emergency transient from the duty cycle of the CRBRP Steam Generator. The origins of Lines A and B are not located exactly at the relief radius due to local elastic and plastic deformations. Referenced to the maximum seal compression (line 0) and the springback capability, both the inner and outer rings are shown to leak for the emergency transient (Line B) and the inner ring is marginal for the normal transient (Line A). If the relief taper is located as required for bearing stresses, both inner and outer rings are predicted to leak for both the normal and emergency transients (Lines A' and B'). Note that the origin of A' and B' is the minimum zero bearing stress margin requirement. It is clear that under the particular geometric and duty cycle constraints on the CRBRP Steam Generators, adequate sealing could not be guaranteed for a design using mechanical seals.

relieve these stresses, a crotch was provided at the transition between the tubesheet and the integral steamhead. The tubesheet ligament stresses were reduced. However, the critical stress was then located at the root of the crotch, regardless of the depth of the crotch. These results showed that removal of the "dummy"-tubesheet thermal mass of the bolted-on steamhead configuration and the contouring of the spherical head reduced the temperature difference between the perforated region and the rim of the tubesheet, leading to an overall improved stress condition in the tubesheet. An additional benefit resulting from the choice of a smooth transition (no crotch) between the tubesheet and the steamhead was the reduced stress concentration in the tubesheet rim as compared to the stress concentrations in the rim with the threaded stud holes.

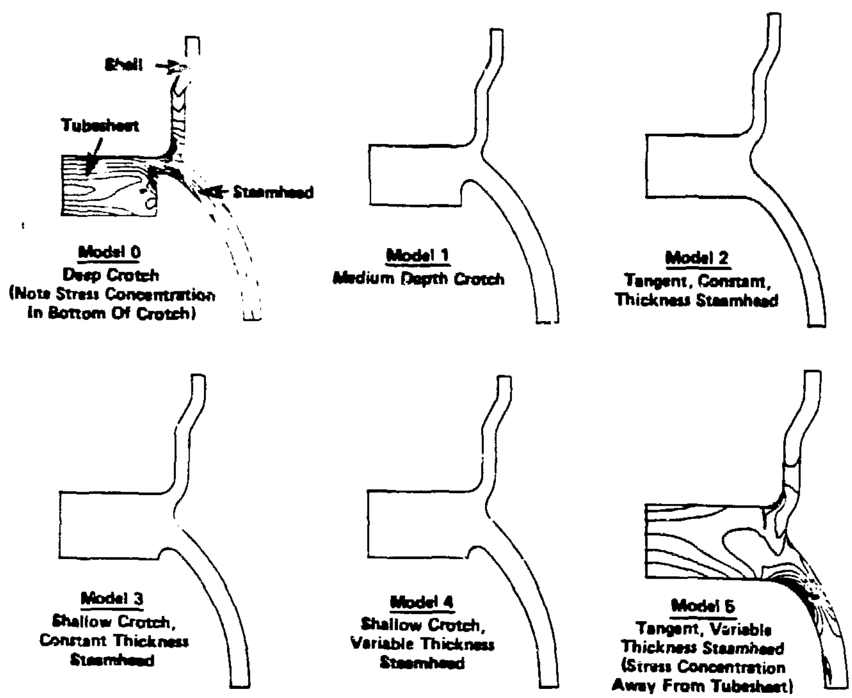

FIG. 8 INTEGRAL STEAMHEAD/TUBESHEET CONCEPTS

Integral Steamhead

Several concepts for integral steamheads were examined to solve the potential leakage problem between the tubesheet and the steamhead. Hemispherical heads were unsuitable because of headroom and access requirement for In-Service-Inspection (ISI). A parametric study on diameter, thickness and tubesheet transition configuration was performed to determine the optimum design for a spherical head (Figure 8). This study was based on elastically calculated primary and thermal stresses.

The transition between the steamhead and the tubesheet was the most critical parameter. Early stress analysis of the bolted tubesheet/steamhead joint had indicated high ligament stresses near the steamside surface of the tubesheet; consequently, to isolate and

The diameter and the thickness of the steamhead were further optimized to provide the smallest acceptable configuration. This was done to minimize the overall weight of the steam generator. Ultimately a 64" (1.62 m) outside diameter steamhead was chosen with a wall thickness of 3" (76.2 mm) for the inlet (cold) and 4" (101.5 mm) for the outlet (hot) steamhead. These thicknesses are principally dependent on the primary loading and bulk temperatures. Detailed inelastic analyses were performed to verify these design choices.

A 16" (406 mm) diameter manway is provided in the steamhead for access for ISI, cleaning and tube plugging, as required. The manway cover is sealed by a metal O-ring which in this application is acceptable due to the small thermal gradients in this joint.

Besides guaranteeing a leakfree Steam Generator, the use of an integral steamhead results in certain system related benefits. Since access is attained through the manway in the steamhead, there is no longer a need to remove a piping section (spool piece) to facilitate dismantling of the bolted-on steamheads. Co..equently, an all-welded system can be utilized ...h eliminates two additional bolted joints with ..chanical seals. The reduction in hardware requirements for one unit includes six piping flanges, 80 Inconel 718 studs and nuts, and four seals in addition to the steamhead hardware replaced. An all-welded system will result in greater plant reliability over the plant lifetime due to the elimination of seal leakage as a potential cause for downtime.

METHOD OF SUPPORT OF INTERNALS AND STEAM TUBES

A steam generator must be designed to accomodate large thermal expansions resulting from operating transients. In case of the CRBRP Steam Generator, a typical upset event for the superheater, a trip from full power, results in a sodium inlet transient of 400°F (204°C) in 800 seconds with a maximum rate of 120°F (66.6°C) minimum. The temperature differences caused by such operating transients dictate that the attachments of the major internal structures and the steam tubes be free to expand relative to each other and to the shell. If free expansion were not permitted, large ..resses would develop. Although the units are designed to withstand faulted events such as large sodium/water reactions, the economic impact of repeated tube breaks makes the reduction of loads and resulting stresses caused by operating transients a primary design consideration.

These stresses can be eliminated by introducing clearances and gaps between mating parts. However, these gaps, if large enough, are very undesirable features when loads must be reacted between the mating parts. Loads from flow-induced vibrations or seismic excitation can cause impact between the parts resulting in fretting and wear. The design employed in the design of the CRBRP Steam Generator will be discussed.

Support Structures for Internals

The two major internal structures are the shroud around the major length of the tube bundle and the thermal liner/elbow shroud assembly. Both have to be supported from the shell such that thermal expansion

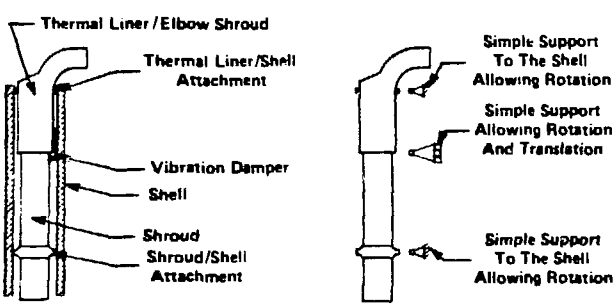

FIG. 9 BASIC CONCEPT OF INTERNAL STRUCTURE SUPPORT

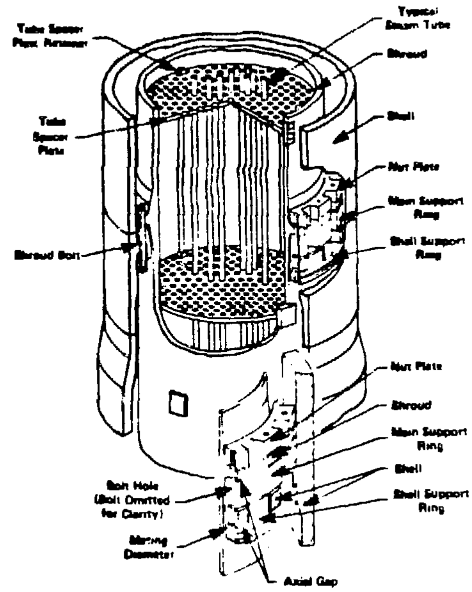

FIG. 10 SHROUD SUPPORT

is permitted and lateral loads to the shell can be reacted without impact. This is achieved by radial key designs.

Radial key designs. Both the thermal liner/elbow shroud and shroud are simply supported structural members to the shell. Figure 9 schematically shows the method of support. The design of the radial keys react lateral loads and allows for rotation of the internals relative to the shell. This prevents large moments from developing at the attachment points due to seismic loads and thermal distortion. Specifically, the shroud is supported by the shell 55" (1.4 m) above the sodium outlet nozzle. This attachment is comprised of two sets of radial keys which are machined into two separate rings. These rings are shown in Figure 10 as the main support ring and shell support ring. One set of radial keys, integral with the main support ring, keys into the shroud. The shell support ring keys into the shell. These two rings mate on a close fitting diameter which allows the two rings to be rotated relative to each other at installation. This rotation compensates for angular misalignment between the shell and shroud and permits close fits between the radial keys without match-machining at assembly. Axial gaps are provided above each radial key which permit rotation of the shroud relative to the shell. The design of this joint does not rely on bolts to react normal down loads from the shroud. The shroud rests directly on the shell and the bolts react up-loads which occur only at shipping and during faulted events. This approach eliminates fatigue loading of the bolts during operation.

The thermal liner is supported by radial keys 41" (1.0 m) above the sodium inlet. The keys which permit free radial expansion are mechanically mounted in the thermal liner as shown in Figure 11. The keys mate with keyways which are slotted in the top surface of a ring that is closely fitted into the inside diameter of the shell. This ring can be rotated at installation to compensate for rotational misalignment with the shroud.

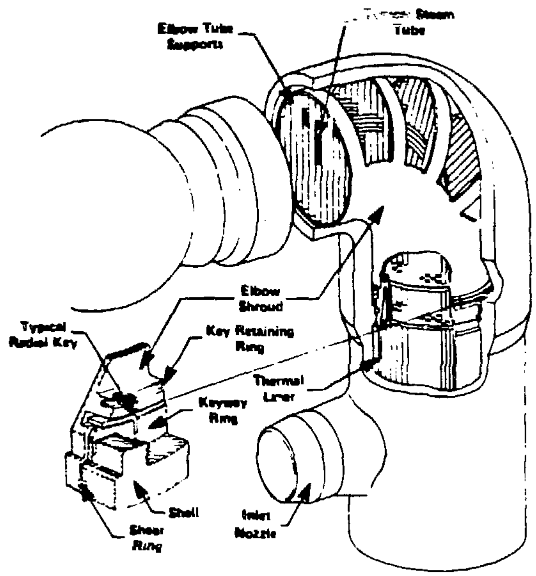

FIG. 11 ELBOW SECTION

The separate ring eliminates the need to place slots directly into the shell, via the main support ring, which would create stress risers in a critical area. The thermal liner rests on the top of this ring to react down-loads. Up-loads on the assembly are reacted by a shear ring located beneath the keys. An axial gap is provided between the shear ring and mating groove so that small rotations can occur between the shell and the thermal liner. Up-loads occur only during shipping and faulted events.

Vibration damper. The elbow shroud/thermal liner and shroud share a common point of support at the vibration damper. The vibration damper (Figures 12 and 13) which is attached to the thermal liner provides support between the thermal liner and the shroud with a set of radial keys. The keys are part of the lower portion of the vibration damper and the keyways are formed by a radially slotted ring that is integral to the shroud. These keys are different from the previously mentioned radial keys because each key is comprised of two cantilever springs that are preloaded into the keyways. The keys and preload are determined such that deflections due to normal operational lateral loads on the shroud will not exceed .010" (.25 mm). These loads are primarily from flow impingements on the shroud. For larger lateral loads that occur during shipping and seismic events, stops are provided to prevent excessive deflection of the keys. The combined lateral loads of the thermal liner and shroud are reacted via the upper portion of the damper which is comprised of 64 cantilever springs that press radially outward against the shell I.D. The stiffness of the individual springs and preload are selected such that normal operational loads result in less than .010" (.25 mm) deflection. Free radial expansion during transients is permitted between the thermal liner and the shell by compressing the individual springs.

The largest relative axial expansions occur at the damper. The shroud is free to expand relative to the shell from its lower support point and thermal liner from its attachment point above the sodium inlet nozzle. The relative motion can be 1.5" (38.1 mm). The compliance of the damper support is important because small changes in key/keyway configuration due to thermal distortion and misalignments will not result in large increases in the interfacing loads at the mating surfaces.

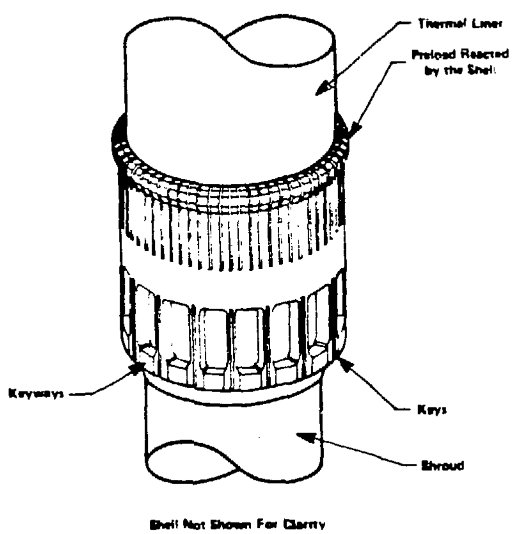

FIG. 12 VIBRATION DAMPER

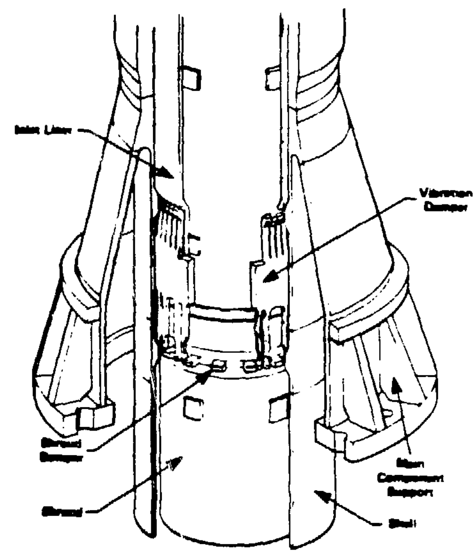

FIG. 13 SHELL WELDMENT AND COMPONENT SUPPORT

Tube spacer plate support. Steam tube to shell temperature differences as high as 350°F (176°C) can occur during some plant transients. The resulting change in length between the shell and the tube is compensated by the sliding of the steam tubes along the axial length of the bundle in the spacer plates and the resulting flexing of the steam tubes in the elbow. The steam tubes are laterally supported along the active length of the steam generator by 19 spacer plates attached to the shroud. The individual spacer plates can move .125" (3.2 mm) laterally in their supports to relieve portions of the tube side loads caused by misalignment of the spacer plates. Minimizing tube side loads is necessary to reduce tube wear and to permit free axial sliding of the tubes relative to the support plate. The benefit of allowing the spacer plates to move laterally is most pronounced at the top and bottom spacer plates. Allowing the bottom plate to float nearly eliminates tube side loads caused by misalignment between the bottom of the shroud and the tubesheet during fabrication. The top plate is the most critical location for assuring proper sliding of the steam tubes. The tube side loads at this plate are highest because the tubes flexing in the elbow pivot at this location. Allowing this plate to move .125" (3.2 mm) reduces the side loads by nearly 40 percent by directly subtracting from the thermal expansion in the horizontal leg of the steam generator and distributing the load downward to adjacent spacer plates. However, lateral spacer plate displacement must be limited to values that can be accomodated without overstressing the tubes during seismic and shipping loads.

Tube spacer plate hole design. Permitting the individual spacer plates to move laterally increases the relative angle between the tube and the spacer plate from that of aligned and rigidly supported spacer plates. Sufficient relative rotation must be allowed in the design of tube hole to preclude tube jamming in the spacer plate. Jamming could otherwise occur when the tube contacts both at the top and bottom of the tube hole. If this were to occur, a large force couple could develop as the tube attempted to further rotate. This could cause large drag forces to develop, inhibiting free expansion. At the top spacer plate, relative rotations can be as high as 2.5 degrees which includes mechanical misalignment, pressure loads, and worst case tube/shell temperature difference. A spacer plate thickness of 1.25" (31.15 mm) is required for structural adequacy of the spacer plate to react pressure loads caused by a sodium/water reaction and tube drag loads. If a reasonably small gap between the tube and the spacer plate is used, for example .018" (.45 mm), only a .8 degree rotation is possible with a straight bored hole. These design considerations result in the tube hole geometry shown in Figure 14. The cylindrical portion of the hole provides a close gap to properly support the tube and the conical portion allows the tube to rotate without jamming when the spacer plate thickness is 1.25" (31.75 mm). The required rotation angle varies between spacer plate locations due to varying loading conditions and mechanical misalignments. During the design of the CRBRP Steam Generator, it was found that two plates were required above the bundle inlet, the first to maintain sufficient contact area between the tube and the spacer plate and a small enough gap to minimize tube wear due to crossflow in the inlet, and the top plate to accommodate the tube rotation angle. For this reason, the top spacer plate has a larger gap (.030" vs. .014") (.762 mm vs. .355 mm) between the tube and the spacer plate than the spacers in the active flow region.

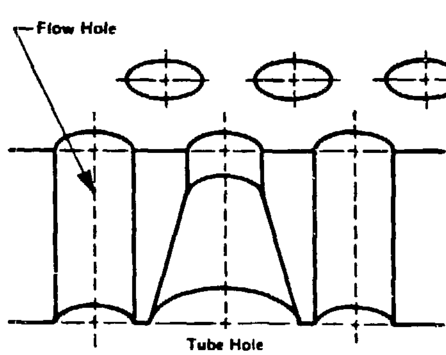

FIG. 14 TUBE HOLE GEOMETRY

Elbow Tube Support

In the elbow region, the steam tubes are laterally supported in planes that lie parallel to the bend in the elbow. The supports are required to react shipping and seismic loads which act perpendicular to the bend plane of the steam tubes. The supports are comprised of plates placed between the tubes and are attached to the elbow shroud. The steam tubes are free to flex and slide in the plane of the bend to compensate for tube to shell thermal expansion. The supports are unique in that for a particular tube the supports alternate from one side of the tube to the other side, as shown schematically in Figure 15. This feature allows the tube to laterally flex if an obstruction gets wedged between the tube and the support. If the supports were on both sides of the tube at one angular position, the tube cannot deflect sufficiently to dislodge the obstruction. Also, the uneven support reduces the potential of a resonant condition for tube vibration due to seismic or shipping loading. This is because for the lowest lateral natural frequency of vibration, the location of the nodes and anti-nodes reverse, depending on the direction of the lateral displacement of the tube.

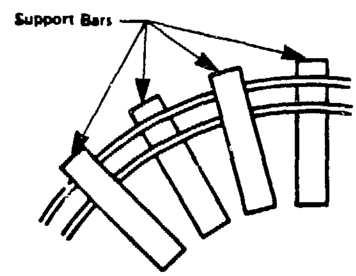

FIG. 15 ELBOW TUBE SUPPORT CONCEPT

CONCLUSION

The advantages with respect to function, maintenance and reduced complexity of the system were the basis for the choice of the integrally welded steamheads for the CRBRP Steam Generator plant units.

Tubes and major internal structures have been designed to accomodate differential thermal expansion. The methods used involve the implementation of radial key designs at major structural supports and a tube hole configuration that permits the tubes to slide and flex in the elbow region. Loads on the steam tubes have been minimized, resulting in reduction of potential wear and corresponding higher reliability.

REFERENCES

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(3) UAP Components, Inc., Bulletin 101B.