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Calvert Cliffs
Nuclear Power Plant

A Member of the
Constellation Energy Group

December 11, 2001

U. S. Nuclear Regulatory Commission
Washington, DC 20555

ATTENTION: Document Control Desk

SUBJECT: Calvert Cliffs Nuclear Power Plant
Unit Nos. 1 & 2; Docket Nos. 50-317 & 50-318
Response to NRC Request for Additional Information Regarding ASME Code Relief Request to Authorize Use of Mechanical Nozzle Seal Assemblies at Calvert Cliffs Nuclear Power Plant as an Alternate Repair Method

REFERENCES:

(a) Letter from Mr. C. H. Cruse (CCNPP) to NRC Document Control Desk, dated November 17, 2000, Use of Mechanical Nozzle Seal Assemblies at Calvert Cliffs Nuclear Power Plant

(b) Letter from Mr. C. H. Cruse (CCNPP) to NRC Document Control Desk, dated February 27, 2001, Request for Relief from ASME Code Requirements to Authorize Use of Mechanical Nozzle Seal Assemblies at Calvert Cliffs Nuclear Power Plant as an Alternate Repair Method

(c) Letter from Mr. R. Clark (NRC) to Mr. C. H. Cruse (CCNPP) dated June 14, 2001, Request for Additional Information Regarding Calvert Cliffs Nuclear Power Plant, Unit Nos. 1 and 2, Relief Request (TAC Nos. MB0557 and MB0558)

By letter dated November 17, 2000 (Reference a), Calvert Cliffs Nuclear Power Plant, Inc. (CCNPP) submitted a request to install Mechanical Nozzle Seal Assemblies (MNSAs) at CCNPP. This request was later documented as a Relief Request from American Society of Mechanical Engineers (ASME) Code Requirements in Reference (b). Calvert Cliffs would like to be able to install MNSAs during the Unit 1 refueling outage in the spring of 2002 in the event that nozzle leakage is identified during the outage on a bottom, upper, or side pressurizer nozzle or hot leg nozzle. The expected benefit over traditional repair methods would be to prevent adverse impacts on outage duration and occupational radiation exposure. Attachment (1) provides CCNPP's response to Nuclear Regulatory Commission staff's request for additional information (Reference c).

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Should you have further questions regarding this matter, we will be pleased to discuss them with you.

Very truly yours,

STATE OF MARYLAND :
: **TO WIT:**
COUNTY OF CALVERT :

I, Charles H. Cruse, being duly sworn, state that I am Vice President - Nuclear Energy, Calvert Cliffs Nuclear Power Plant, Inc. (CCNPP), and that I am duly authorized to execute and file this License Amendment Request on behalf of CCNPP. To the best of my knowledge and belief, the statements contained in this document are true and correct. To the extent that these statements are not based on my personal knowledge, they are based upon information provided by other CCNPP employees and/or consultants. Such information has been reviewed in accordance with company practice and I believe it to be reliable.

Subscribed and sworn before me, a Notary Public in and for the State of Maryland and County of Calvert, this 11th day of December 2001.

WITNESS my Hand and Notarial Seal: Denise D. Snuki
Notary Public

My Commission Expires: 02/01/02
Date

CHC/TWG/dlm

cc: R. S. Fleishman, Esquire
J. E. Silberg, Esquire
Director, Project Directorate I-1, NRC
D. M. Skay, NRC
H. J. Miller, NRC
Resident Inspector, NRC
R. I. McLean, DNR

ATTACHMENT (1)

RESPONSE TO REQUEST FOR ADDITIONAL INFORMATION

MECHANICAL NOZZLE SEAL ASSEMBLY

ATTACHMENT (1)

RESPONSE TO REQUEST FOR ADDITIONAL INFORMATION MECHANICAL NOZZLE SEAL ASSEMBLY

The following additional information regarding various aspects of the Mechanical Nozzle Seal Assembly designs is provided to facilitate completion of Nuclear Regulatory Commission staff review.

1.0 Hydrostatic and Thermal Testing

1.1 Request

The hex head bolts attaching the pressurizer nozzle seal assembly to the fixture, as shown in Reference 2, are of unequal length. However, all bolts were torqued to produce the same axial preload. Explain the measures taken to ensure that the assembly was properly aligned with the axis of the nozzle, since the elongations of the bolts do not appear to have been the same.

Response

The position of the mechanical nozzle seal assembly (MNSA) with respect to the pressurizer is fixed by means of tightly-fitting shear pins, thus avoiding any sliding. This helps to maintain the proper alignment of the MNSA with respect to the axis of the nozzle. Next, during the application of the preload, calibrated torque wrenches are used and the final torques are applied in 4 to 5 increments. Using an alternating (cross) torquing pattern and gradually increasing torque values, proper seating/alignment is assured. Since the torque values rather than the bolt elongation are controlled, the torque ensures that equal forces are applied by each bolt. In addition, the installation process is inspected visually for any uneven gap situations. The combination of positioning the MNSA using tightly-fitting shear pins and carefully controlling the preload application ensures the MNSA is properly aligned with the axis of the nozzle.

1.2 Request

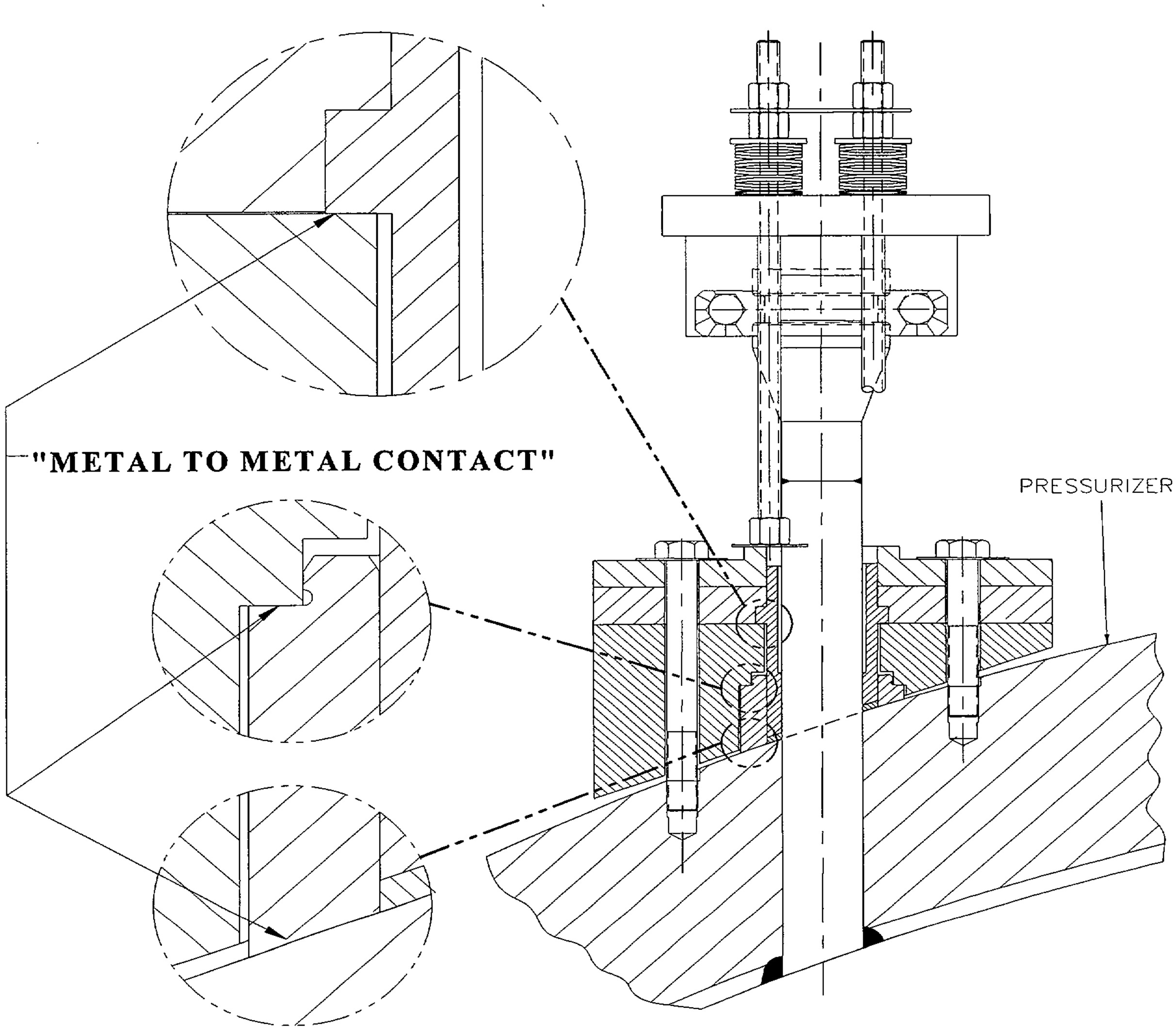
Reference (2) states that additional thermal test cycles are not considered necessary since the MNSA components are designed to go "metal-to-metal," confining the graphite seal in a fixed volume, and that thermal cycling does not "work" the seal; and, therefore, the MNSA design is not sensitive to thermal cycling effects.

a. Provide a description, including a diagram, of what is meant by "metal-to-metal."

b. Provide test data or analysis to support the assertion that "thermal cycling does not work the seal" by showing that the graphite seal experiences zero thermal expansion or contraction under varying thermal conditions.

Response:

a. A detailed schematic of the area of concern is provided below.



b. The Grafoil seal is captured in a small volume that will not vary except microscopically. The bolt preload maintains the pressure and captures the seal to a nearly constant volume. Similar seals are being used in a variety of applications (valve stem seals) under significantly more thermal volume changes. Grafoil, therefore, can be considered to be a relatively stable seal material, especially when the volume changes are minor. Assuming an annular section of 1/4" (width) x 1/4" (height) x 1.1" (diameter), the volume is computed as 1/4 x 1/4 x 3.14 x 1.1 = 0.215875 [in^3]. Then, assuming an expansion coefficient of 6E-6 [in/in-Deg], and a temperature differential of 500°F, each dimension changes as follows:

0.250" expands to 0.25075" and
1.100" expands to 1.1033".

The total volume of the Grafoil seal is then 0.2178237 [in^3], which equals a percentage change of (0.2178237-0.215875)/0.215875 x 100 = 0.9 [%]. Thus, even neglecting the thermal expansion of the Grafoil seal, the total volume is well within the initial compression of the Grafoil seal (approximately 35%).

1.3 Request:

Provide justification why one hot leg seal assembly test and one pressurizer head seal assembly test represent an adequate sample to show MNSA structural integrity under operating conditions.

Response:

During the design process, tight geometrical tolerances are established to provide confidence regarding the structural integrity of all MNSAs manufactured for a specific application. The hydrostatic pressure testing performed on one MNSA for one specific application met the applicable American Society of Mechanical Engineers (ASME) Code testing requirements and provided confirmation that the MNSA design is acceptable to ensure structural integrity under all operating conditions.

During the manufacturing process, quality assurance measures are used to ensure that geometrical tolerances are controlled within acceptable limits. However, for defense-in-depth, all MNSAs delivered to Calvert Cliffs Nuclear Power Plant were subjected to a hydrostatic pressure test. These tests demonstrated seal tightness in every case thereby providing further assurance of MNSA structural integrity.

The main reason for proper performance of the MNSA seals is the tight dimensional controls over the hardware and the Grafoil seal. Each machined item, as well as the Grafoil seals, are subjected to rigid inspections by the suppliers. In addition, Westinghouse performs a series of receipt inspections prior to shipment of the equipment to the site. Many of these inspections are also witnessed by Calvert Cliffs Nuclear Power Plant inspectors.

1.4 Request:

Tests, ranging over years, of flexible graphite gaskets under high temperature conditions have been reported in the open literature (e.g., References 3 and 4). Clarify how the thermal cycling tests reported in Reference (1) provide a basis for extrapolating to two cycles of operation, measured in years, of as-installed MNSAs under transient operating conditions such as those found in the Reactor Coolant System.

Response:

For the Westinghouse Combustion Engineering plants, the controlling transients are heatup and cooldown. These envelop all other plant transients. The associated rates are reasonably slow and were conservatively simulated in testing using autoclave heatup rates of 100°F/hour. Cooldown is typically uncontrolled and achieved in much shorter times, thus aggravating any potential of leakage introduced by transient temperature/pressure profiles. Laboratory tests on a variety of Grafoil seals are limited to a few cycles. It has been Westinghouse's experience that, if there is a tendency of a seal to leak, it will occur during the initial cycle and not during later cycles. On that basis, and by testing to more extreme conditions, qualification of Grafoil seals by fewer cycles than those associated with a plant design condition is not only practical but also is justified.

2.0 <u>MNSA Analysis</u>

The MNSAs are characterized by simplified models consisting of linear-elastic uni-axial springs representing the MNSA metal member stiffnesses. References (5) and (6) provided detailed Asea

Brown Boveri, Inc. (ABB)/Combustion Engineering analyses of MNSA devices fastened to the hot leg and the pressurizer. The Nuclear Regulatory Commission staff has identified the following concerns that should be addressed regarding these analyses:

2.1 Request:

The loads and stresses in the MNSAs are calculated by excluding the Grafoil gasket from the models. Reference (7) indicates that the Grafoil material is highly compressible. Provide calculations of the loads and stresses in the MNSAs that include the non-linear load-deflection properties of the Grafoil (per Reference 7), and show that the results, as shown in References (5) and (6), bound the more refined analyses with the Grafoil gaskets.

Response:

The preload results in metal-to-metal contact within the MNSA, consequently the flexibility of the Grafoil seal does not affect any of the loads or stresses. The pressure loading in the seal itself is controlled by the volume compression of the seal. The volume compression of the seal is achieved as preload torque is applied and increased to a value at which further seal compression ceases. Any further increases in preload torque are directly reacted through the metal-to-metal contact region.

2.2 Request:

The analyses did not consider the compression of the gasket due to preloading of the hex head bolts.

a. Provide the basis for this assumption.

b. Provide an estimate of the compressed gasket thickness.

c. Provide an evaluation of the effect of gasket compression on the hex head bolt stiffnesses, since significant compression of the gasket could change the as-installed bolt lengths.

Response:

a. The preload exceeds the load necessary to compress the gasket. There is metal-to-metal contact and, once that contact has been established, any preload increase is being reacted by the metal parts, whereas the Grafoil seal load remains constant.

b. The compressed gasket thickness is calculated from the annular groove for the gasket and is 0.25 inch nominally.

c. The bolt preload controls the elongation in the bolt. The compression of the seal is 50 mils and is small compared with the bolt length of 4 inches. There is no direct effect from the gasket on the bolt stiffness. Once metal-to-metal contact is established, there is no effect from the Grafoil seal on the MNSA metal components.

2.3 Request:

The friction forces acting on the sleeves or nozzles in the MNSA analyses depend on the radial stress (compressive) resulting from the axial compression of the gasket. In the calculation of the friction forces, the radial stress is assumed to be the same as the maximum pressure determined in the hydrostatic test (Reference 2). Provide the basis for this assumption.

Response:

This value is assumed conservatively low since compression data on the gasket indicates higher compressive stress. The MNSA installation procedure requires torquing the hold-down bolts in increments until metal-to-metal contact is achieved. The MNSA compression collar and Grafoil gasket are designed such that the nominal compression of the gasket is 35%. The torque required to achieve this compression is in excess of 30 ft-lb. Once metal-to-metal contact is achieved, the torque is backed off to 30 ft-lb for operation. The following is a calculation for a MNSA designed for a nozzle on the side of the pressurizer.

Calculating the Radial Stress in the Grafoil Seal for the Side Pressurizer MNSA

The final torque value of the side pressurizer MNSA (1.315 Diameter Nozzle) is 30 ft-lbs. This equates to a bolt load of 3,600 lbs (for 4 ½-20 UNF bolts) using the following equation for bolt torque:

T=kLD, where k=0.2 and D=0.5

Calculating the surface area of the Grafoil seal (ID: 1.315 inches, OD: 1.815 inches) we get an area of 1.229 in^2. The compressive load (axial pressure) in the Grafoil seal thus equates to the total bolt load divided by the seal surface area.

Load = [3,600 lbs/bolt x (4 bolts)] / 1.229 in^2 = 11,715 psi

This equates to the following radial load in the Grafoil seal: 0.88 x Load = 10,310 psi

Since the MNSAs successfully completed hydrostatic testing, the lowest gasket pressure can be assumed to equal the hydrostatic pressure, which is a logical conclusion for the gasket to prevent leakage during the hydrostatic test. This value is thus assumed conservatively low.

2.4 Request:

For the upper and bottom pressurizer nozzle MNSA analyzed in Reference (6.2), show that this MNSA remains qualified to the ASME Section III Class 1 1989 stress criteria, if the analysis is based on the following circumstances:

a. The net ejection force is determined by considering the friction force based on the compressed gasket thickness.

b. The equivalent stiffness used for calculating the impact forces is based on the equivalent stiffness of the MNSA, including the stiffness of the Grafoil gasket.

c. The MNSA model is represented by an asymmetric geometry with the actual length hex head bolts, without the assumptions regarding hex head bolt length and compression collar length.

Response:

a. Installation of the MNSA will result in metal-to-metal contact between the bottom plates and the hot leg piping or pressurizer shell. The preload of the retention plate and the clamp (tie rods) during installation using Belville springs controls the ejection force. The preload will vary as a result of heatup and cooldown, and the Westinghouse analyses has captured these variations in a conservative manner and computed the impact forces based on the smallest preload conditions. The friction force will not significantly vary since the amount of contact pressure between seal and the nozzle shaft remains nearly constant throughout all plant operating conditions, and is basically controlled by the preload.

   The analysis in Reference 6.2 credits the friction force of the gasket for controlling the net ejection force. The friction force of the gasket is used to calculate a realistic value of the impact force to ensure the preload nominal value of stress is not excessive in the pre-faulted condition.

In order to determine the impact force, the net ejection force is determined by considering the friction force based on the compressed gasket thickness. The following evaluates the effects of the compressed gasket thickness (0.2 inch vs. 0.25 inch) on the impact force using the most limiting analysis of the pressurizer heater sleeve MNSA.

Force Due to Internal Pressure is:

$F_p = (2{,}500 \text{ psi})(1.050 \text{ in}^2) = 2{,}625$ lbs.

Friction Force is:

$F_f = P\,\mu\,A = (3{,}100 \text{ psi})(0.3)\,\pi\,(1.156 \text{ in})(0.2 \text{ in}) = 675$ lbs

where:

P	=	radial seal load (pressure) = 3,100 psi
μ	=	coefficient of friction = 0.3
A	=	surface area of the seal in contact with the sleeve surface = $\pi D h$
D	=	1.156 in
h	=	0.2 in

Net Ejection Force is:

$F_e = 2625 - 675 = 1{,}950$ lbs.

Total Deflection, Δx, is:

$\Delta x = [F_e \pm (F_e^2 + 2K_{eq} F_e \text{ Set})^{1/2}] / K_{eq} = 0.0986$ in

Given:

F_e	=	1950 lbs
K_{eq}	=	spring constant = 6.04×10^4 lb/in
Set	=	maximum soft Bellville washer set deflection plus the total deflection of the tie rods = 0.052 in

Impact Force is:

$F_{impact} = K_{eq}\Delta x = 6.04 \times 10^4$ [lb/in] (0.0986 [in]) = 5.96 kips

The value of F_{impact} = 6,000 lbs was used in subsequent analysis of the pressurizer heater sleeve MNSA components. This impact force value is greater than two times the ejection force of 2,625 lbs (neglecting frictional forces). The use of a non-conservative net ejection force ensures the dynamic load doubles the ejection load. This is a standard design practice and is not inconsistent with the ASME Code. Therefore, its use is justified.

For the other four analyzed MNSAs, a decreased value of friction force is bounded by the use of a conservative value of the impact force.

b. The equivalent stiffness of the MNSA is not determined by the Grafoil gasket stiffness (Reference 6.4, page F17 and Reference 6.2, page B18) because stiffness has no effect once the preload is being established and is maintained at a constant/stable value.

c. The most conservative lengths of components were used in calculations of the bolt and flange stiffnesses. The use of non-conservative numbers reduces the maximum bolt preload. For instance, if we use the longest bolt hole length and the height of the short compression collar in the pressurizer heater sleeve analysis, the following maximum bolt load will be calculated:

Stiffness of Hex Head Bolts

$$K_{bolts} = 4\frac{AE}{l} = 4\frac{(0.172\ \text{in}^2)(25.0\text{X}10^6\ \frac{\text{lbf}}{\text{in}^2})}{5.653\ \text{in}} = 3.04\text{E}6\ \frac{\text{lbf}}{\text{in}}$$

where:

l = effective length of bolt, assuming 0.5 inches of thread engagement = thread engagement + longest bolt hole length in the lower flange + upper flange (outboard) + upper flange (inboard) + retainer plate = 0.5 + 3.758 + 0.75 + 0.615+ 0.03 = 5.653 inches where the longest bolt hole length is in the pressurizer heater sleeve MNSA and is calculated as (7.057 inches - 2.56 inches tan 52.19°) = 3.758 inches

Stiffness of Overall Flange:

Stiffness of the upper flange, top:

$$K_{upper,top} = \frac{W}{y} = \frac{2\pi r_o}{\frac{a^3}{D}(\frac{C_1 L_9}{C_7} - L_3)} = 5.85\text{E}6\ \frac{\text{lbf}}{\text{in}}$$

(Reference to B-PENG-DR-006, pg. F28) (Reference 6)

Stiffness of the upper flange, bottom:

$$K_{upper,bot} = \frac{W}{y} = \frac{2\pi r_o}{\frac{a^3}{D}(\frac{C_1 L_9}{C_7} - L_3)} = 2.26\text{E}6\ \frac{\text{lbf}}{\text{in}}$$

(Reference to B-PENG-DR-006, pg. F29) (Reference 6)

Compression Collar:

$$K_{collars} = \frac{AE}{l} = \frac{(1.657\ \text{in}^2)(25.04\text{X}10^6\ \frac{\text{lbf}}{\text{in}^2})}{2.504\ \text{in}} = 1.65\text{E}7\ \frac{\text{lbf}}{\text{in}}$$

where:

l = average effective length,
= (4.689 inches – 2.00 inches* tan 52.19° + 1.00 inches* tan 21.44°) = 2.504 inches

Equivalent flange stiffness:

$$K_{flange} = \frac{1}{\frac{1}{(K_{upper,top} + K_{upper,bot})} + \frac{1}{K_{collar}}} = 5.45E6 \frac{lbf}{in}$$

Maximum bolt load is: $F_{max} = 3.6 + \left(\frac{3.04E6}{3.04E6 + 5.45E6}\right)\frac{6.00}{4} = 4.14 \text{ kips}$

Maximum bolt load is reduced by (4.43 - 4.14)/4.43 = 6.55%, when comparing to conservatively used values.

Therefore, the use of conservative lengths is justified.

2.5 Request:

The thermal stresses in the hex head bolts are determined by assuming that only the bolts take up the thermal expansion. Another assumption is that the length of these bolts is the same as the combined length of the collar and the thickness of the flange. For the MNSAs mounted on the pressurizer top and bottom heads, where pairs of hex head bolts may be of significantly different length, show that the thermal stresses in the hex head bolt are bounded by the current calculation, and that the preload is not lost in the longer bolts as a result of thermal expansion.

Response:

The controlling equation for thermal stresses is $\sigma_{t.e.} = E\,\alpha\,\Delta T$

where:

$\sigma_{t.e.}$ =	stress due to thermal expansion
E =	Modulus of Elasticity (psi)
α =	Coefficient of Linear Thermal Expansion
ΔT =	Change in Temperature

As is shown, thermal stresses in the bolts do not depend on the bolt length and, therefore, are equal in both short and long bolts. It is also assumed that additional load due to differential thermal expansion is completely taken by the deformation of the bolt and, therefore, does not reduce installation preload.

2.6 Request:

The stress calculations of the hex head bolts do not include an evaluation of bending stresses due to the rotation of the upper flange. Provide verification that these stresses are not significant.

Response:

In order to evaluate bending stresses in the hex head bolts, the radial slope of the upper flange under preload must be determined first. The most limiting analysis of the pressurizer heater sleeve MNSA is used as an example.

According to Reference (7), Table 24, Case 1a, the radial slope of the outer edge is:

$$\theta_A = K_{\theta A} \frac{wa^2}{D} = 6.33\text{ E-4 radians}$$

where:

a = outer radius, 1.875 in
w = unit line load = Preload/$2\pi r_o$ = 1,818.9 lbs/in
$K_{\theta A}$ = 0.5763, since b/a=0.672

$$D = \frac{Et^3}{12(1-\gamma^2)} = 9.66E5 \text{in} - \text{lb}$$

where:

t = thickness, 0.75 in
γ = Poisson's ratio, 0.3
E = elastic modulus, 25.0E6 psi

According to Reference (7), Table 3, Case 3c, the angular displacement at fixed end/simply supported beam is calculated as:

$$\theta_A = \frac{ML}{4EI}$$

Therefore,

$M = 4EI\theta_a / L$ = 62.98 in-lbs

where:

I = $\pi d^4/64$ = 2.345E-3 in^4 - moment of inertia of hex head bolt, based on pitch diameter
L = the shortest bolt, conservatively = 0.992 inches (shortest bolt hole length in the lower flange) + 1.365 inch (thickness of the upper flange) = 2.357 inches
d = 0.4675 max pitch diameter

This bending moment causes the bending stress in the hex head bolt of:

$\sigma_b = Mc/I$ = (62.98 x 0.234) / 2.345 x 10-3 = 6,285 psi

The maximum stress intensity (σ_{max}) is then determined:

$$\sigma_{max} = 2\sqrt{\left(\frac{\sigma_t + \sigma_b}{2}\right)^2 + (\tau_T)^2} = 51.54 \text{ ksi} \qquad < 2.7\, S_m = 72.4\text{ksi} \; (S_m \text{ at } 700°F)$$

$< 3.0\, S_m = 80.4$ksi (S_m at 700°F)

where the stresses from the analysis:

σ_t = 40.17 ksi
σ_b = 6.29 ksi
τ_T = 11.16 ksi

It can be seen that even using this very conservative approach, the maximum stress intensity has increased by only 12% (51.54 vs. 45.95) and remains well within allowables.

2.7 Request:

The MNSA assembly drawings (References 9-12) require Grafoil gaskets to have a final density of 90 lb/ft^3. Provide information showing how this requirement will be met.

Response:

The volume for each seal configuration is calculated for the dimensions at the upper end of the tolerance band and for the dimensions at the lower end of the tolerance band. The weight in grams is then calculated at each of these volumes based on the 90 lb/ft^3 density. Each seal is weighed on a beam balance scale during receipt inspection and must fall within those maximum and minimum weights to be acceptable.

2.8 Request:

The design bolt stresses for the hex head bolts, per ASME Section III, NB-3231 and Appendix E, as required by ASME Section XI, IWB-7320, used a value of "m" of 1.3 to calculate the compression load to ensure a tight joint. Reference (8) identifies this value as "m"=2.0, obtained from the same source. Provide an explanation of this inconsistency.

Response:

The inconsistency between "m" values in Reference (8) and the Westinghouse reports is explained by differences in the "minimum design seating stress" (factor "y" per Appendix E) that is necessary to seat a gasket and to maintain a tight joint. Reference (8) identifies "y" as 900 psi, while values required to achieve the sealing are significantly higher. According to Reference (13)(also referenced in Westinghouse reports), the minimum design seating stress is reported in a range of 6,100 – 10,300 psi. The "m" factor corresponding to such stresses is 1.3. Therefore, the "m" value of 1.3 is more appropriate.

2.9 Request:

The stress calculations of clamp components, such as the compression collars, did not include shear stresses due to differential thermal expansion. Provide justification showing these stresses meet the design shear allowable.

Response:

The preload in the bolts controls the load in the compression collar. Since the bolts are relatively long, they supply a virtually constant load onto the bottom plate assemblies and compression collar. Therefore, the cyclic loading is negligible.

According to ASME Section III, NB-3213.9 B&PV Code, a thermal stress is classified as a secondary stress (due to added conservatism, analyses of the tie rods and hex head bolts considered thermal stresses as primary). Since those stresses are minimal (for instance, 8.19 ksi at heater sleeve MNSA) and are limited to high allowables of 3Sm (for instance, 3Sm = 48ksi at 700°F for compression collar), they were not considered in analysis.

The shear stress was considered for computations of the primary stresses.

REFERENCES

(1) Enclosure (9) to Reference (1) Proprietary Vendor Report TR-PENG-042, "Test Report of MNSA Hydrostatic and Thermal Tests," dated July 1997

(2) Enclosure (4) to Reference (1) Proprietary Calculation No. CA04893, Rev. 0, "Design Evaluation of MNSA for Various Applications at Calvert Cliffs Units 1 and 2," dated November 17, 2000

(3) Derenne, M., et.al., "Elevated Temperature Characterization of Flexible Graphite Sheet Materials for Bolted Flanged Joints," WRC Bulletin 419, February 1997

(4) Hahn, R., et.al., "Long-term Behavior of Graphite-Based Gaskets at Elevated Temperature," PVP-Vol. 382, Analysis of Bolted Joints, 1999

(5) Enclosure (12) to Reference (1) Proprietary ABB Design Report No. B-PENG-DR-005, Rev. 01, "Addendum to CENC-1179 Analytical Report for BGE CCNPP Units 1 and 2 - Piping," November 17, 2000, with the following proprietary attachments:

 (5.1) Attachment A, "Analysis of BG&E Units I and II Hot Leg RTD Nozzle MNSA"

 (5.2) Attachment B, "Analysis of BG&E Units I and II Hot Leg PDT/Sampling Nozzle MNSA"

 (5.3) Attachment D, "Calculation No. A-CCNPP-9449-1230, Rev. 00, Evaluation of Attachment Locations for MNSA Devices on BG&E CCNPP Units I and II Hot Leg Piping RTD and PDT/Sampling Instrument Nozzles"

(6) Enclosure (13) to Reference (1) Proprietary ABB Design Report No. B-PENG-DR-006, Rev. 02, "Addendum to CENC-1179 Analytical Report for BGE CCNPP Units 1 and 2 - Pressurizers," November 17, 2000, with the following proprietary attachments:

 (6.1) Attachment A, "Analysis of BG&E Units I and II Side Pressurizer RTD Nozzle MNSA"

 (6.2) Attachment B, "Analysis of BG&E Units I and II Upper and Bottom Pressurizer Nozzle MNSA"

 (6.3) Attachment D, "Calculation No. A-CCNPP-9449-1229, Rev. 01, Evaluation of Attachment Locations for MNSA Devices on BG&E CCNPP Units I and II Pressurizer Top Head, Shell and Bottom Head Instrument Nozzles and Heater Sleeves"

 (6.4) Attachment F, "Analysis of BG&E Units I and II Heater Sleeve MNSA"

(7) Young, W. C., "Roark's Formulas for Stress and Strain," Sixth Edition

(8) Attachment 3 to Enclosure (1) to Reference (1): Raub, H. S., Optimum Gasketing with Grafoil Flexible Graphite. Union Carbide Corporation publication, undated

(9) Enclosure (14) to Reference (1) ABB Drawing No. E-MNSABGE-228-001, Rev. 02, "Hot Leg RTD MNSA," November 17, 2000

(10) Enclosure (15) to Reference (1) ABB Drawing No. E-MNSABGE-228-002, Rev. 02, "Hot Leg PDT/sampling MNSA," November 17, 2000

(11) Enclosure (18) to Reference (1) ABB Drawing No. E-MNSABGE-228-007, Rev. 02, "Bottom Pressurizer MNSA," November 17, 2000

(12) Enclosure (21) to Reference (1) BGE Drawing No. 12019-0079, Rev. OD, "Side Pressurizer RTD MNSA," November 17, 2000

(13) Howard, R. A., "Engineering Design Manual," Union Carbide Grafoil, Volume One, Sheet and Laminated Products, 1987