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NON-LINEAR ANALYSIS OF A CLOSURE MANWAY USING SPIRAL WOUND GASKET WITH METAL-METAL CONTACT AND A NEW GEOMETRY APPROACH

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SUMMARY

This work presents the results of a PWR Pressurizer Closure Manway analysis. The manway geometry is slightly different from the conventional solution with the goal to reduce the bending stresses in the bolts when the system is pressurized. So the Salt stresses value will also be reduced. The viability of the proposed solution will be confirmed by: a) Verification of the stresses in the bolts connecting the blind flange to the nozzle by ASME III, subsection NB and b) Level of the Tightness reached in the Spiral Wound (type SG) gasket based in the criteria defined in the references.

SUMÁRIO

São apresentados os resultados da análise da boca de visita do pressurizador de um reator PWR adotando-se uma geometria ligeiramente diferente da solução convencional a fim de reduzir as tensões de flexão nos parafusos quando o sistema estiver sob pressão. Assim a tensão Salt também será reduzida. A viabilidade da solução proposta será confirmada pela verificação das tensões nos parafusos de conexão do flange cego com o bocal pela subseção NB da norma ASME III e pelo nível de estanqueidade obtido, baseado em critérios e propriedades definidos em referências.

1. Introduction

The scope of this work is to present the results of the analysis in the manway closure of a PWR pressurizer. The geometry is slightly different from the conventional solution to reduce the bending stresses in the bolts connecting the blind cover to the nozzle. So also the Salt stresses used in the fatigue evaluation will be reduced. A very good level of tightness is desired and Spiral Wound (type SG) is used. The design temperature is 360 ^{0}C.

Modelled and Actual Geometry - The actual geometry is formed by the following principal components: a)nozzle, b)gasket joint, c)seal membrane, d)bolts, e)flange, f)pressurizer body and g)tube segments. A sphere equivalent to the cylindric surface of the pressurizer, with a diameter of twice the cylinder diameter, was modelled up to 2.5*SQRT(Re*te), where Re and te are the internal radius and thickness, respectively, of the equivalent sphere, from the external surface of the nozzle (see fig. 1 and 4). The gasket joint is responsible by the tightness when compressed between the flange/ membrane and the nozzle. The tightness level must be high because the tighted fluid is primary water, so the tightness Class T3 [3] was selected.

Materials - All materials used in the pressurizer and the manway are considered SA.508.CL3 steel, except the bolts (SA.193.B7) and the gasket. The considered properties for these materials are presented in Table 1. The adopted bi-linear stress/deflection curve of the gasket (fig. 2) was obtained from [5] and the principal parameters are also shown in Table 1, where E is the Young's modulus, Sm is the ASME III allowable limit for the membrane stresses and u is the Poisson's modulus of the associated material.

Loads in the Structure - The loads in the manway are, basically: a)nominal design internal pressure (Pp) which is 16.55 MPa and b)pre-stress in the bolts. This pre-stress may be understood to have two components: b.1) Pre-stress "for tightness" - This is the value of pre-stress to reach the Class T3 tightness. At this point the stress in the gasket must be the Sya value and the gap between the membrane and nozzle must be near zero as justified later. b.2) Pre-stress "Extra" - Fex - This "extra" pre-stress applied in the bolts must be at least equal to

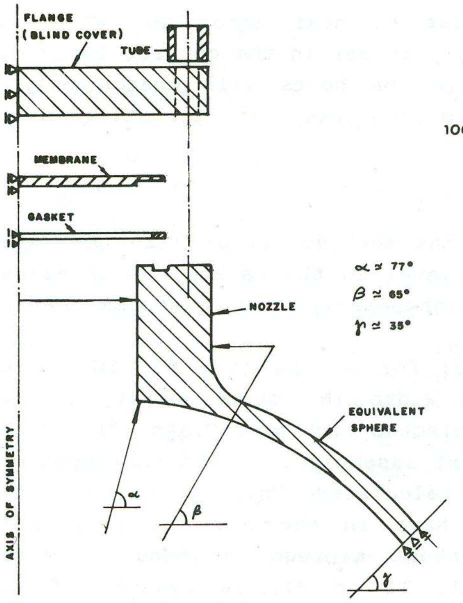

Figure 1: The Modelled Geometry
(with no scale)

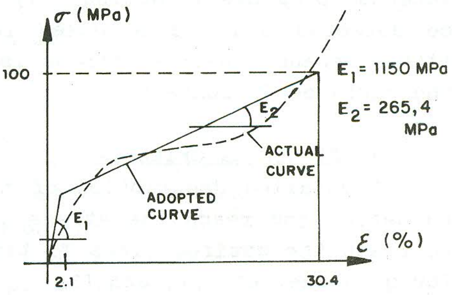

Figure 2: The adopted Stress-Strain Curve of the Gasket SG

Table 1 - Materials Properties (for 360 ^{0}C)

	SA-508-CL3	SA-193-B7	gasket
E (MPa)	182000.	182000.	E_1 = 1150. E_2 = 265.4
Sm (MPa)	184.1	187.5	---
u	0.3	0.	---

the force exerted by the pressure in the Flange. The scope is to avoid high stress fluctuations in the bolts during the pressurization and de-pressurization of the system. This fact, if it occurs, will produce high values of Salt stresses in the bolts and low allowable number of cycles in its fatigue analysis. The minimization of the bending stresses in the bolts, and therefore the Salt stress, is possible if all this "extra" pre-stress is acting between the membrane and nozzle surfaces. For this reason when the stress in the gasket reaches the Sya desired value the gap between these surfaces must be near zero. So, when the internal pressure is acting only a releaf in the contact force may be detected and the stresses in the bolts will change only a little because their stiffness is much lower then the stiffness of the surfaces in contact.

2. Tightness Criteria

A detailed description of the methodology used in this work to define and reach the stress level in the gasket, which allows to reach the desired Class T3 tightness-1/50000 mg/sec.mm, can be found in the ref. [1] and [3].

Basically the procedure is: for a type (CA, DJ, SG, SF or SW), medium diameter (Dt) and width (N) of a gasket, for an internal pressure (Pp), for a selected tightness Class (T1, T2 and T3), M value and efficiency joint assembly (e), the correspondent gasket stress value Sya may be calculated. This Sya value is the stress that the gasket shoud have in the pre-stressing phase without pressure. The selected minimum tightness level, associated with the Class T1, T2 or T3, is reached if, in operation, the condition $Smin > M * Pp$ is true, where Smin is the residual stress in the gasket after the application of the internal pressure.

Sya Values for the Used Gasket SG - Considering a) Pp=16.55 MPa, b) tightness Class T3, c) joint efficiency e=0.95, d) joint internal and external diameter (Oi=476 mm, Oe=506 mm) and e) gasket type SG, for the gasketed joint in this analysis, the following Sya values (table 2) were obtained based in the formulation of ref. [3]. (The selected Sya value can not be less then Sy).

Table 2 - Sya values for the analysed joint with e = 0.95

			Sya values for M = (in MPa)					
Class	TPMIN	Sy	3	4	5	6	7	8
T3	1649	105	135	120	111	105	99	95

3. Model

For the analysis an axi-symmetric model was used. The gasket was modelled with plane truss elements. The associated material has a bi-linear kinematic behavior. Between the surfaces of the flange and the membrane, out of the gasket region, some gap elements were defined with a null initial gap. Other initially open gap elements with 1.6mm were defined between the surfaces of the membrane and the nozzle.

The bolts were modelled with 2-D beams elements penetrating in the nozzle up to a depth equivalent to their diameter. Out of the flange the bolts are 100. mm long. To allow this construction there are tube segments, also modelled with 2-D beam elements, concentrical with the bolts, between the nut and the Flange.

In the model the nut-to-tube segment connection is continuous and in the base of the tube segment there are gap elements to allow the tube to rotate independently from the bolt (see fig. 3). The properties of the gap, truss and beam plane elements, presented in the table 3, were defined in a "per radian" base to allow their use in a axi-symmetric model.

Table 3 - Properties of some of the 2-D elements

	Rigid beams	Bolts	Tubes	Units
Area/rd	1.E+6	3362.0	6656.	mm**2/rd
I/rd	1.E+12	3.53E+5	3.55E+6	mm**4/rd
u	0.	0.	0.	--
c	0.	1.E-5	0.	mm/mm.^{0}C

The boundary conditions defined in the model can be found in fig. 1. In the model the area ai (area/rd) of the truss elements (gasket) are given by ri * ti, where ri is the radial position and ti is the thickness associated with the truss element i. The total area of the gasket is 3682.5 mm**2/rd.

4. <u>Loads - Application Methodology</u>

The pre-stress in the bolts was applied to the beam elements modelling the bolts as a deformation E_0 defined by a temperature variation (DT). This deformation must act gradually, as actually occurs, due to non-linearities in the structure.

To take this fact into account, some load-steps were defined where the initial pre-stress "for tightness", the "extra" pre-stress and the pressure action (in this order) were gradually augmented.

<u>Pre-Stress "for Tightness" - Fpre</u> - Due to the fact that the flange is not rigid, it is necessary to calculate a deformation value to compensate the stress relief in the bolts due the rotation of the flange and the deformation of the gasket other then the bolt itself.

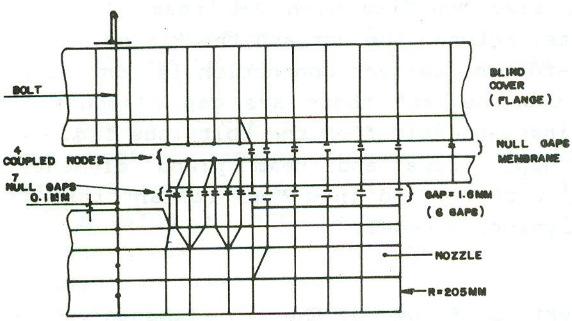

Figure 4: Detail of the Model in the Gasket Region

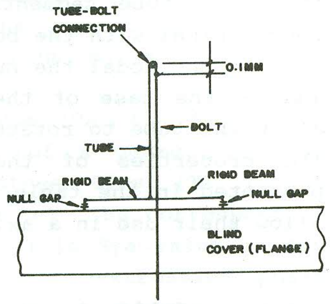

Figure 3: Tube - Bolt Connection Detail

The deformation E_0 is associated with a temperature variation in the bolts. If the E_0 value is correct the force/stress in the bolts, obtained from the model, will be near the desired value and the stress in the gasket will be the Sya value (associated to the corresponding M value). In the other hand, with the option to apply E_0 as a temperature variation, a new load step (RESTART) may be applied until the Sya value in the gasket is reached.

<u>Pre-Stress "Extra" - Fex</u> - This pre-stress will also be applied as DT in the bolts and it is calculated taking into account the stiffnesses of the bolts, flange and metal to metal contact. This pre-stress will act in the structure when:

a) the membrane-nozzle contact occurs and

b) the Sya value is reached in the gasket. The initial gap value (1.6 mm) between membrane and nozzle was obtained from the deformation of the gasket to the Sya stress level plus 10% in order to be conservative.

<u>Pressure</u> - The incremental pressure application will begin after the Fex value of the force in the contact between membrane and nozzle is reached. Without this contact the stresses

in the gasket would be reduced with the pressure application. With the contact, the stress in the gasket will not vary significantly during this step due to the great stiffness between surfaces, which pratically absorbs all variation of forces/ stresses caused by the applied pressure.

5 - Analysis and Results

With the described model and procedure the analysis was made in the steps described below, with several restarts to control the results.

Step of Pre-Stress "for Tightness" - From table 2 Sya = 111. MPa (M=5) was choosed. From the geometry defined in fig. 1 we have: a)Kpl=87.1 MN/mm; b)Kga=1.7 MN/mm and c)Kpa=17.8 MN/mm, repectively the flange, gasket and bolts stiffnesses. The initial pre-stress force applied to the bolts Fpre, without membrane-nozzle contact, is the force developed in the gasket. So Fpre=2.568 MN. Taking into account the related stiffnesses, the pre-deformation E_0 which must be applied in the model to obtain Fpre is (in a first approximation):

$$E_0 = Fpre/lop * (1/Kpa + 1/Kpl + 1/Kga) = 0.7799\% \quad (1)$$

In equation (1) lop is the "free" bolt length (216. mm) with a ficticious thermal coefficient value of 1.0E-5. The corresponding temperature variation is DT=779.9 ^{0}C. The value of DT=800 ^{0}C was adopted. The reference temperature was defined as 0.0 ^{0}C so, in the following, temperature and temperature variation will have the same significance. To reach the DT value 9 load steps were initially defined, respectively, with -50., -100., -200., -300., -400., -500, -600., -700. and -800. ^{0}C. It was necessary to perform two other load steps, with -900. and -1040. ^{0}C to reach Sya in the gasket. The results are presented in fig. 5 and table 4 where it can be seen that there is a small contact force between membrane and nozzle as desired. The force in the bolts in this point is 2.75 MN.

Step of Pre-Stress "Extra" - In this step of the analysis the stiffness of the gasket will not be considered, and the stiffness of the flange (Kpl) will be re-calculated. An initial estimate of the nozzle-membrane contact stiffness (Kcon) gives 480.3 MN/mm. For Kpl the re-calculation gives 58.8 MN/mm. The stiffness of the bolts are the same and the contact force between nozzle and membrane is Fcon = Pp * Area = 2.95 MN. With this figures the "extra" deformation to be applied in the bolts, in a first approximation, to reproduce Fcon, is E_0 = 0.103% which corresponds to 103 ^{0}C of temperature variation. This value was imposed in the model in 3 load steps but the contact force did not reach the desired value (see fig. 5). The pratically linear behavior of the stresses in the bolts and contact force allows an extrapolation to the temperature value of -1215 ^{0}C to obtain the contact force Fcon. So this value was imposed in the model in one more load step and the Fcon value was obtained (see fig. 5). After this point the structure/model was ready to apply the pressure.

<u>Step of Pressure</u> - With the bolts temperature fixed in -1215^{0}C the pressure was applied in 7 load steps (0.1, 0.3, 0.4, 0.5, 0.7, 0.9 and 1.0*Pp). The results are shown in fig. 5 and table 4. From the results it can be seen that, for 0.9*Pp only one gap between nozzle and membrane was closed, and for 1.0*Pp this gap was open. Due to this fact the stresses in the bolts, which were lowering, began to grow and the stresss in the gasket showed a small reduction.

Table 4 - Results of the analysis

LOADS		(1)	TUBE		BOLTS		(2)	(3)	(2)	(2)	
pressure	temp.	gap force	mem-brane	membrane+ bending	membrane+ bending	mem-brane	gap force	gasket stress	contact force	closed gaps	analysis phase
-	-50.	-27.	-8.3	8.7	18.6	16.5	-28.0	-15.1	-	0	
-	-100.	-48.	-14.8	15.5	33.0	29.0	-51.0	-26.6	-	0	pre-
-	-200.	-64.	-20.0	21.0	44.0	39.0	-68.0	-36.0	-	0	stress
-	-500.	-115.	-35.0	37.0	79.0	70.0	-121.	-64.0	-	0	
-	-800.	-165.	-51.0	53.0	144.	101.	-174.	-92.0	-	0	"for
-	-990.	-198.	-61.0	64.0	136.	121.	-208.	-109.5	-	0	tightness"
-	-1040.	-214.	-66.0	69.0	147.	131.	-225.	-113.0	-23.0	2	
-	-1100.	-299.	-93.0	97.0	208.	183.	-316.	-114.0	-195.	4	"extra"
-	-1180.	-414.	-128.	135.	289.	254.	-438.	-114.5	-380.	4	pre-
-	-1215.	-464.	-144.	151.	325.	284.	-492.	-116.0	-528.	4	stress
1.655	-1215.	-457.	-142.	150.	323.	281.	-486.	-116.0	-470.	4	
4.965	-1215.	-443.	-138.	147.	321.	273.	-475.	-116.0	-354.	3	
8.275	-1215.	-430.	-134.	144.	319.	266.	-465.	-115.5	-238.	2	pressure
14.585	-1215.	-409.	-129.	141.	319.	256.	-452.	-114.0	-50.0	1	
16.550	-1215.	-412.	-131.	143.	326.	259.	-457.	-113.0	---	0	
MPa	^{0}C	KN/rd	MPa		MPa		KN/rd	MPa	KN/rd	Units	

Notes: (1) gaps between tubes and blind flange; (2) gaps between blind flange and nozzle; (3) medium value of the stresses in the elements

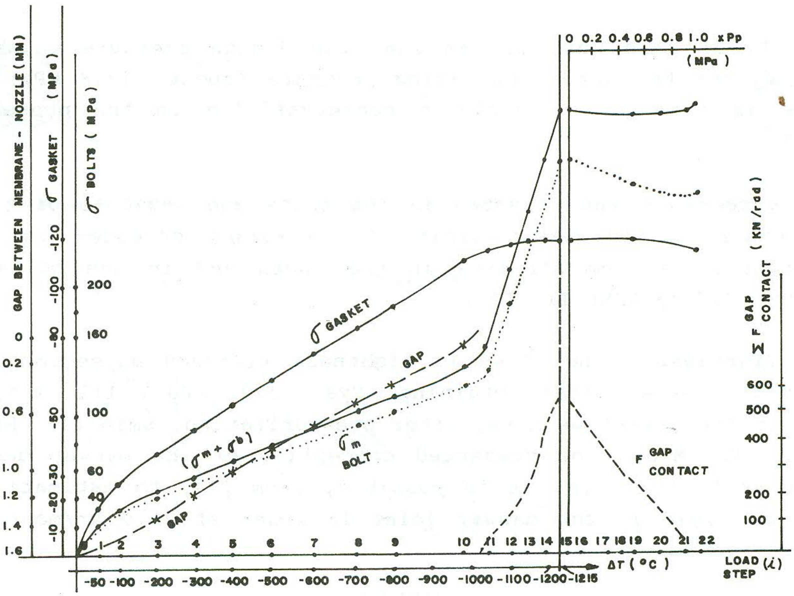

Figure 5: Results of the Analysis

6. <u>Final Remarks and Conclusions</u>

The prediction of a gap=1.6 mm between nozzle and membrane showed to be correct. The solution with the concentrical tubes to allow bolts longer than the flange thickness and the metal to metal contact betwen nozzle and membrane showed a good agreement with the expected reduction in the alternate stress in the bolts during the pressurization and de-pressurization of the system. (A previous analysis was performed without the tube segments and with a consequent shorter "free" bolt length. The resulting membrane+ bending bolt stress was near the ASME limit).

<u>Bolts</u> - During the pre-stress "for tightness" the force in the bolts reaches 2.75 MN and in the final it reaches 6.0 MN. This last value must be applied in the assembly to allow the good behavior of the joint as expected. To maintain the metal to metal contact after the pressurization the force in the bolts can be estimated in more 10% for instance, but it can be seen that: a)the differential thermal expansion between the stainless steel (nozzle, membrane, flange) and the ferritic steel (bolts) will produce, when in service, a higher pre-stress force in the bolts, and b) in this work it was used the design pressure which is higher than the normal operating pressure (about 13.8 MPa). So there is already an intrinsic conservativism in the presented results.

Stresses - The stresses in the bolts and segments of tubes are under the allowable limits for membrane stresses and for membrane + bending stresses in the tubes and in the bolts as recommended by ASME III [2].

Tightness - The Class T3 tightness (1/50000 mg/sec.mm) was reached because, after obtaining Sya (=113. MPa > 111. MPa, to M=5) in the gasket we have, after pressurization, Smin=113. MPa > M*Pp (=83. MPa). The presented conception of the manway design conducts to Smin=Sya. It is possible, from [3], to estimate the loss of fluid in the manway joint in order of 25.5 gr/month as maximum value.

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