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Stress Analysis of the HFIR HB-2 and HB-3 Beam Tube Nozzles

P. T. Williams

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STRESS ANALYSIS OF THE HFIR HB-2 AND HB-3 BEAM TUBE NOZZLES

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ABSTRACT

The results of three-dimensional linear elastic stress analyses of the HFIR HB-2 and HB-3 nozzles are presented in this report. Finite element models were developed using the PATRAN pre-processing code and translated into ABAQUS input file format. A scoping analysis using simple geometries with internal pressure loading was carried out to assess the capabilities of the ABAQUS/Standard code to calculate maximum principal stress distributions within cylinders with and without holes. These scoping calculations were also used to provide estimates for the variation in tangential stress around the rim of a nozzle using the superposition of published closed-form solutions for the stress around a hole in an infinite flat plate under uniaxial tension. From the results of the detailed finite element models, peak stress concentration factors (based on the maximum principal stresses in tension) were calculated to be 3.0 for the HB-2 nozzle and 2.8 for the HB-3 nozzle. Submodels for each nozzle were built to calculate the maximum principal stress distribution in the weldment region around the nozzle, where displacement boundary conditions for the submodels were automatically calculated by ABAQUS using the results of the global nozzle models. Maximum principal stresses are plotted and tabulated for eight positions around each nozzle and nozzle weldment.

1. INTRODUCTION

The High Flux Isotope Reactor (HFIR) located at Oak Ridge National Laboratory (ORNL) is a versatile, 85-MW isotope production and test reactor with the capability and facilities for performing a wide variety of irradiation experiments. Among the primary purposes of the HFIR is the production of californium-252 and other transuranium isotopes for research, industrial, and medical applications. Additionally, the HFIR is used for neutron-scattering experiments to reveal the structure and dynamics of a wide range of materials. The neutron-scattering instruments installed on the horizontal beam tubes are applied in fundamental studies of materials of interest to solid-state physicists, chemists, biologists, polymer scientists, metallurgists, and colloid scientists. The HFIR is a beryllium-reflected, light-water-cooled and -moderated, flux-trap type reactor that uses highly enriched uranium-235 as the fuel.¹ Figures 1(a) and 1(b) are cutaway and elevation views of the reactor that show the pressure vessel, its location in the reactor pool, and some of the experiment facilities.

The HFIR is equipped with three horizontal beam tubes, extending outward from the reactor core midplane as shown in the plan view of Fig 2. The HB-2 beam tube extends radially from the reactor center line with its inner end penetrating the permanent reflector. Also penetrating the permanent reflector, the HB-3 beam tube extends tangentially from the core, offset approximately 10.5 in. from the reactor center line. The remaining beam tube is positioned on a tangential line with both ends, designated as HB-1 and HB-4, extending outward from the reactor. These beam tubes are supported by the reactor using bolted flanges connected to nozzles welded to the reactor vessel walls. In support of safety studies of the HFIR reactor vessel, it is the objective of this analysis to calculate the stresses within two of these nozzles under internal pressure loading. Specifically, the maximum principal stress distribution will be calculated as input for a probabilistic fracture mechanics analysis of the HFIR pressure vessel. Three-dimensional linear elastic stress analyses of the HB-2 and HB-3 horizontal beam tube nozzles were carried out using the ABAQUS/Standard² finite element computer program, a nuclear quality assurance certified (NQA-1) code.

Section 2 presents a description of the geometry, material properties, and assumptions that are used in the development of the finite-element models of the HB-2 and HB-3 nozzles. Scoping calculations, summarized in Sect. 3, provide estimates for the distribution of tangential stresses around a circular penetration in a cylinder and flat plate with loading conditions that approximate those applied in the more detailed finite-element analyses discussed in Sect. 4. Conclusions are given in Sect. 5.

Fig. 1. High Flux Isotope Reactor (HFIR) (a) reactor pressure vessel positioned in reactor pool, and (b) elevation view showing positions of beam tubes and fuel core.

Fig. 2. Plan view of HFIR core midplane showing four horizontal beam tubes.

2. HFIR HB-2 AND HB-3 NOZZLES: GEOMETRY AND MATERIAL PROPERTIES

The HFIR core is contained in a 94 in. internal diameter (ID) pressure vessel constructed of carbon steel with austenitic stainless steel cladding on the inner and outer surfaces of the vessel. Including the cladding, the vessel walls are nominally 3 1/8 in. thick. A stainless steel extension is attached to the hemispherical lower end of the vessel to permit control rod access through the pool floor. As indicated in Fig. 3, the centerline of both beam tube nozzles are located approximately 111 in. below the top head and 17 in. above the lower girth weld. Detailed dimensions for both nozzles are presented in Fig 4. For both nozzles, the interior reentrant corners at the vessel ID are rounded off with a 1 in. radius. It is assumed that the steel in both the vessel and nozzle is homogeneous and isotropic with a constant Young's modulus of elasticity, E , of 28,000 ksi and a Poisson's ratio, ν , equal to 0.3. The nozzles are assumed to be a homogeneous material that is integral with the vessel. The HB-2 and HB-3 nozzles are welded to the pressure vessel with circumferential welds on radii of 15 in. and 10.5 in., respectively. Variations in material properties that may induce stress gradients between the vessel and nozzle base metal, cladding, cladding heat affected zone (HAZ), weld filler material, and weld HAZ are assumed to be secondary and are neglected.

Fig. 3. Elevation view of HFIR pressure vessel showing location of the horizontal beam tube nozzle.

Fig. 4. Plan view of (a) HB-2, and (b) HB-3 nozzle sections.

3. SCOPING CALCULATIONS

Scoping calculations were carried out to assess the adequacy of the specific finite elements, boundary conditions, and approximate mesh discretization selected for the HB-2 and HB-3 nozzle models. These calculations involved calculating the state of stress in a circular cylinder with the same internal diameter, wall thickness, and material properties employed in the nozzle analysis. Figure 5 shows a quadrant of a cylinder with a 94 in. ID and wall thickness of 3.125 in. The longitudinal edges are constrained by reflective symmetry boundary conditions where displacements normal to the surface of the edges are fixed. These surfaces are aligned with the coordinate planes. The left and right edges are free.

Lamé's solution³ to the differential equation of equilibrium for a thick-walled cylinder under internal pressure describes the distribution of the radial, σ_{rr} and circumferential (hoop), $\sigma_{\theta\theta}$ stresses as a function of the radius, r , within the cylinder,

$$\begin{aligned}\sigma_{rr} &= \frac{p_i r_i^2}{r_o^2 - r_i^2} \left(1 - \frac{r_o^2}{r^2} \right) \text{ for } r_i \leq r \leq r_o, \\ \sigma_{\theta\theta} &= \frac{p_i r_i^2}{r_o^2 - r_i^2} \left(1 + \frac{r_o^2}{r^2} \right) \text{ for } r_i \leq r \leq r_o,\end{aligned}\quad (1)$$

where r_i is the internal radius, r_o is the outer radius, and p_i is the internal pressure. Since the ends of the cylinder are free, the axial (meridional) stress, σ_{zz} is equal to zero. The radial stresses are compressive (negative) and small in absolute value compared to the tensile circumferential stresses. The peak radial stress is $-p_i$ at the inner radius and decreases to 0.0 at the outer radius. For the test problem with a 1.0 ksi pressure loading, the circumferential stresses vary from a maximum of 15.556 ksi at the inner radius down to 14.556 ksi at the outer radius. The axial strain, ϵ_{zz} , is given by

$$\epsilon_{zz} = - \frac{2\nu}{E} \frac{r_i^2 p_i}{r_o^2 - r_i^2} = - \frac{2 \times 0.3}{2.8 \times 10^7} \frac{47^2 \times 1000}{(50.125^2 - 47^2)} = -1.5596 \times 10^{-4} \quad (2)$$

The results of the ABAQUS solution for this problem are presented in Fig. 5. A model of the cylinder consisting of 1237 nodes and 192 elements was constructed using the PATRAN preprocessing program. The elements are 20-node hexahedra with incomplete quadratic basis functions. Two elements spanned the thickness of the cylinder. Since the circumferential and radial stresses are the maximum, σ_3 , and minimum, σ_1 , principal stresses, respectively, the corresponding ABAQUS solution for the principal stresses are reported in Fig. 5 where the ABAQUS ordering for the principal stresses is $\sigma_1 \leq \sigma_2 \leq \sigma_3$. ABAQUS recovers the theoretical axial strain to within the precision of the code's line printer output file. The errors in the calculated maximum radial and circumferential stresses are 0.23% and 0.0321%, respectively.

The test model was then modified by adding a 12-in.-diam hole in the middle of the cylinder quadrant. Applying the same boundary conditions and internal pressure loading as in Fig. 5 the resulting ABAQUS solution for the maximum principal stress, σ_3 , is presented in Fig. 6. The stress distribution indicates high tensile stresses around the axis of the hole that is aligned with the longitudinal axis of the cylinder. The calculated peak maximum principal stress in tension is 57.328 ksi giving a stress concentration factor of 3.68 based on the maximum circumferential stress in the cylinder without a hole (15.561 ksi). No stress concentration data corresponding exactly to this geometry were found in the literature; however, two similar cases were located. For a hole in an infinite plate under uniaxial tension, the stress concentration factor is 3.0.⁴ Stress concentration factors for circular holes in shallow thin shells are given by Peterson⁵ where

Cylinder with 1000 psi Internal Pressure Loading

Geometry: Material Properties:
 Cylinder ID = 94 in. $E = 28.0 \times 10^6$ psi
 Wall Thickness = 3.125 in. $\nu = 0.3$

ABAQUS Solution
 Major Principal Stress: $14,558 \leq \sigma_1 \leq 15,561$ psi
 Minor Principal Stress: $-997.7 \leq \sigma_2 \leq 4.3$ psi
 Second Principal Stress: $-1.55 \leq \sigma_3 \leq 8.30$ psi
 Axial Strain: $\epsilon_z = -1.5596 \times 10^{-3}$

Lame Solution:
 Circumferential Stress: $14,556 \leq \sigma_\theta \leq 15,556$ psi
 Radial Stress: $-1000 \leq \sigma_r \leq 0$ psi
 Axial Stress: $\sigma_z = 0$
 Axial Strain: $\epsilon_z = -1.5596 \times 10^{-3}$

Fig. 5. ABAQUS solution for cylinder with internal pressure loading.

Cylinder with a Hole: 1000 psi internal pressure loading

Cylinder ID = 94 in. Material Properties:
 Wall Thickness = 3.125 in. $E = 28.0 \times 10^6$ psi
 Hole Diam. = 12 in. $\nu = 0.3$

Major Principal Stresses: σ_1
 ABAQUS Solution: $-1420 \leq \sigma_1 \leq 57,328$ psi
 Stress Concentration Factor = $57,328/15,561 = 3.68$

Fig. 6. ABAQUS solution for a cylinder under internal pressure loading with a hole.

"shallow" refers to a small hole to cylinder radius ratio, and "thin" refers to a small shell thickness to cylinder radius ratio. The geometry of Fig.6 does not fall strictly within the range of validity of the theoretical treatment given by Peterson, primarily because the test problem's hole to cylinder radius ratio is too large. A hole diameter of 12 in. was selected for the test problem to correspond approximately to the bore diameters of the nozzles. Peterson uses the following parameter to correlate the stress concentration factors

$$\beta = \frac{[3(1 - \nu^2)]^{1/4}}{2} \left(\frac{r}{\sqrt{Rt}} \right) = \frac{[3(1 - 0.3^2)]^{1/4}}{2} \left(\frac{6}{\sqrt{(48.5625)(3.125)}} \right) = 0.313 \quad , \quad (3)$$

where r is the radius of the hole, R is the mean radius of the cylinder, and t is the thickness of the cylindrical shell. From Fig.90 in Peterson's book, the stress concentration factor is approximately 3.2 for membrane stresses (tension) and approximately 3.7 for total stress (membrane plus bending) at a β of 0.31. The ABAQUS solution, therefore, falls within the expected levels of stress concentration for the case of internal pressure loading of a cylinder with a hole.

An estimate for the distribution of the maximum principal stresses around the HFIR nozzles can be developed by the superposition of two linear elastic solutions for the case of a hole in an infinite flat plate under a uniaxial tensile load. Applying two uniaxial loads at 90° angles produces a combined biaxial load that is comparable to the loading on a HFIR nozzle. In Fig. 7, far-field tensile stresses have been applied to the four edges of a large flat plate which has a centrally located hole of radius a . For the combined loading cases depicted in Fig. 7, the distribution of tangential stresses σ_θ around the hole is given by⁶

$$\sigma_\theta = \frac{\sigma_A}{2} \left[1 + \left(\frac{a}{r} \right)^2 \right] - \frac{\sigma_A}{2} \left[1 + 3 \left(\frac{a}{r} \right)^4 \right] \cos(2\theta) + \frac{\sigma_A}{4} \left[1 + \left(\frac{a}{r} \right)^2 \right] - \frac{\sigma_A}{4} \left[1 + 3 \left(\frac{a}{r} \right)^4 \right] \cos \left(2 \left(\theta + \frac{\pi}{2} \right) \right) \quad , \quad (4)$$

where σ_A corresponds to the nominal vessel hoop stress (assuming a cylinder with internal radius, r_i , wall thickness, t , and no hole with an internal pressure load p_i). Given $p_i = 1.0$ ksi and applying the geometry of the HFIR pressure vessel, the vessel hoop stress is

$$\sigma_A = p_i \times \frac{(r_i + t/2)}{t} = \frac{1.0 \text{ ksi} \times (47 \text{ in.} + (3.125/2) \text{ in.})}{3.125 \text{ in.}} = 15.54 \text{ ksi} \quad . \quad (5)$$

The hoop stress load acting alone on the hole produces a peak tensile tangential stress at the top edge ($\theta = \pi/2$) of the hole $3\sigma_A$ and a maximum compressive circumferential stress of $-\sigma_A$ at the side ($\theta = 0$) of the hole. The circumferential stress distribution around the hole due to the meridional (longitudinal) stress load acting alone can be estimated by applying a far-field stress of $\sigma_A/2$ to the top and bottom edges of the plate. The superposition of these two solutions [as shown in Eq. (4)] results in a peak tangential stress at the top of the hole ($\theta = \pi/2$) of $2.5\sigma_A = 38.85$ ksi and a minimum tangential stress of 7.77 ksi at $\theta = 0$.

Fig. 7. Tangential stress distribution around a hole in an infinite flat plate. Linear superposition of two loadings.

4. FINITE-ELEMENT MODEL RESULTS

4.1 HB-2 NOZZLE STRESSES

Figure 8 shows two views (inside and outside) of the HB-2 ABAQUS model, developed using the PATRAN pre-processing code and translated into ABAQUS input file format. The finite element mesh was constructed with 11,123 nodes and 2392 elements which are primarily 20-node isoparametric hexahedra (bricks). Triangular regions, including the rounded inner rim of the nozzle and the bottom and top of the reactor vessel, are modeled with 15-node isoparametric wedges. These wedges are prismatic elements consistent with the quadratic basis functions employed with the 20-node hexahedra. The nozzle is positioned asymmetrically within the vessel quadrant corresponding to the asymmetry depicted in Fig. 2. The upper and lower extent of the pressure vessel (regions above and below the nozzle) are approximations to the real geometry. The location and thickness of the top head are correct; however, all other vessel penetrations above the nozzle and the extension to the hemispherical endcap have been neglected. The intent of these regions is to locate the upper and lower boundaries of the model sufficiently far away from the nozzle to have minimal impact on the stress gradients calculated near the nozzle. It was not within the scope of this analysis to determine the effect of other penetrations (see Figs. 1b and 2) in the reactor vessel on the state-of-stress in the nozzle. Figure 8 also shows the location of surface constraints applied to the model as reflective symmetry boundary conditions. These surfaces are aligned with the coordinate planes of the problem's global coordinate system. Since these are symmetry planes, a four-fold nozzle-placement symmetry is implied for the remainder of the vessel. From Fig. 2, one observes that the implied symmetry does not in fact exist; however, as the results (to be presented) for both the HB-2 and HB-3 nozzles will show, no significant stress gradients extend from the nozzles out to these symmetry planes, implying a negligible interaction between the nozzles in adjacent quadrants. This lack of interaction justifies the application of symmetry boundary conditions to the side planes of the model. Internal pressure loading (1.0 ksi) was applied on all interior surfaces including the inner bore of the nozzle.

Maximum principal stress filled contours for the inside and outside of the HB-2 nozzle model are shown in Figs. 9a and 9b, respectively. The general trends are similar to those of the scoping calculations in that the larger tensile stresses are located above and below the nozzle bore along the longitudinal (vertical) axis of the reactor vessel. Significant nozzle stress gradients (as shown in Fig. 10) are localized within the immediate vicinity of the nozzle and do not extend to the side symmetry planes. The peak maximum principal stress is 46.69 ksi giving a stress concentration factor (as defined previously) of $46.69/15.561 = 3.0$. Maximum principal surface stress contours along the inner surface of the HB-2 nozzle are shown in Fig. 10. Eight points equidistant around the inner rim are labeled HB2-N1 to N8, and the magnitude and orientation of the maximum principal stress at each point are depicted in Fig. 10. The maximum principal stresses at eight locations (designated HB2-W1 to W8) along the weldment extending around the nozzle at a 15 in. radius are presented in Fig. 11. The stresses in the weldment were calculated using a separate submodel embedded in the geometry of the larger global model. The global model provides this submodel with displacement boundary conditions. All of the principal stresses at the points called out in Figs. 10 and 11 are given in Tables 1 and 2, respectively. In Table 2, the principal stresses as a function of distance from the inner vessel surface are also presented. These stresses follow lines that are parallel to the nozzle horizontal axis rather than normal to the inner surface of the vessel.

Fig. 8. Mesh layout for HB-2 nozzle ABAQUS model.

Fig. 9. Maximum principal stresses around HB-2 nozzle: (a) inside, and (b) outside of vessel.

Fig. 10. Maximum principal stresses for shallow surface flaws distributed around the rim of the HB-2 nozzle. View of interior surface of the HFIR pressure vessel.

Fig. 11. Maximum principal stresses for shallow surface flaws distributed around the weldment submodel of the HB-2 nozzle. View of interior surface of the HFIR pressure vessel.

Table 1. Principal Stresses Around Rim of HB-2 Nozzle

Position	Principal Stresses		
	σ_1 [ksi]	σ_2 [ksi]	σ_3 [ksi]
HB2_N1	-0.802	0.441	42.518
HB2_N2	-0.898	-0.333	23.053
HB2_N3	-0.898	-0.652	5.630
HB2_N4	-0.947	-0.468	24.084
HB2_N5	-0.804	0.380	40.675
HB2_N6	-0.795	-0.497	22.396
HB2_N7	-1.126	-0.961	4.418
HB2_N8	-0.777	-0.185	23.981

**Table 2. Principal Stresses Around Weld Line of HB-2 Nozzle
as a Function of Distance from the Inner Wall**

Position	Principal Stresses			
	R [in.]	σ_1 [ksi]	σ_2 [ksi]	σ_3 [ksi]
HB2_W1	0.00000	-1.027	8.234	23.431
	0.78125	-0.678	7.841	20.360
	1.56250	-0.343	7.408	17.291
	2.34381	-0.146	7.108	14.429
	3.12497	0.055	6.858	11.557
HB2_W2	0.00000	-0.940	4.149	18.558
	0.80099	-0.686	4.734	17.652
	1.60193	-0.534	5.294	16.839
	2.40223	-0.227	5.674	15.991
	3.20262	0.119	5.917	15.270
HB2_W3	0.00000	-1.080	4.246	6.817
	0.82144	-0.782	6.278	9.046
	1.64293	-0.546	8.257	11.265
	2.46437	-0.232	10.104	13.492
	3.28581	0.141	12.006	15.730
HB2_W4	0.00000	-0.959	1.963	16.149
	0.80099	-0.692	3.513	15.535
	1.60193	-0.496	4.995	15.018
	2.40223	-0.204	6.307	14.413
	3.20262	0.123	7.423	14.011
HB2_W5	0.00000	-1.063	5.572	18.921
	0.78125	-0.668	6.285	16.583
	1.56250	-0.359	6.932	14.224
	2.34372	-0.164	7.645	12.043
	3.12497	0.114	8.281	10.029
HB2_W6	0.00000	-0.982	0.596	14.358
	0.80095	-0.708	3.233	14.495
	1.60189	-0.546	5.700	14.811
	2.40223	-0.217	7.784	15.137
	3.20253	0.179	9.390	15.978
HB2_W7	0.00000	-1.100	3.174	3.643
	0.82145	-0.817	6.033	7.250
	1.64290	-0.604	8.795	10.880
	2.46439	-0.258	11.295	14.299
	3.28583	0.154	13.848	17.744
HB2_W8	0.00000	-0.948	3.000	16.965
	0.80099	-0.687	4.381	16.717
	1.60189	-0.552	5.644	16.646
	2.40232	-0.238	6.608	16.491
	3.20256	0.145	7.314	16.591

4.2 HB-3 NOZZLE STRESSES

The ABAQUS model for the HB-3 nozzle is depicted in Fig. 12. The finite element mesh consists of 10,431 nodes and 2230 elements which are primarily 20-node isoparametric hexahedra (bricks). Triangular regions, including the rounded inner rim of the nozzle and the bottom and top of the reactor vessel, are modeled with 15-node isoparametric wedges. The nozzle is located approximately in the middle of the vessel quadrant. The extent of the model, the boundary conditions, and the internal pressure loading are the same as those used in the HB-2 nozzle analysis.

The maximum principal stress distributions for the inside and outside of the HB-3 nozzle model are shown in Figs. 12a and 12b, respectively. The general trends are similar to those of the scoping calculations and the HB-2 nozzle results in that the larger tensile stresses are located above and below the nozzle bore along the longitudinal (vertical) axis of the reactor vessel. As in the HB-2 results, these stress gradients are localized in the nozzle and do not extend to the side symmetry planes. The maximum major principal stress is 44.19 ksi, giving a stress concentration factor of $44.190/15.561 = 2.8$. Maximum principal surface stress contours along the inner surface of the HB-3 nozzle are shown in Fig. 13. Eight points around the inner rim are labeled HB3-N1 to N8, and the magnitude and orientation of the maximum principal stress at each point are depicted in Fig. 13. The maximum principal stresses at eight locations (designated HB3-W1 to W8) along the weldment extending around the nozzle at a 10.5 in. radius are presented in Fig. 15. As with the HB-2 model, the stresses in the weldment were calculated using a separate submodel embedded in the geometry of the larger global model. The global model provides this submodel with displacement boundary conditions. All of the principal stresses at the points called out in Figs. 14 and 15 are given in Tables 3 and 4, respectively. In Table 4, the principal stresses as a function of distance from the inner vessel surface are also presented. These stresses follow lines that are parallel to the nozzle horizontal axis rather than normal to the inner surface of the vessel.

Table 3. Principal Stresses Around Rim of HB-3 Nozzle

Position	Principal Stresses		
	σ_1 [ksi]	σ_2 [ksi]	σ_3 [ksi]
HB3_N1	-1.003	6.224	42.255
HB3_N2	-0.764	-0.750	20.446
HB3_N3	-1.010	-0.755	7.655
HB3_N4	-0.158	-0.105	21.729
HB3_N5	-0.979	6.039	41.386
HB3_N6	-0.281	0.126	21.682
HB3_N7	-0.852	-0.511	7.753
HB3_N8	-0.264	-0.017	23.615

Fig. 12. Mesh layout for HB-3 nozzle ABAQUS model.

Fig. 13. Maximum principal stresses around HB-3 nozzle: (a) inside, and (b) outside of vessel.

Fig. 14. Maximum principal stresses for shallow surface flaws distributed around the rim of the HB-3 nozzle. View of interior surface of the HFIR pressure vessel.

Fig. 15. Maximum principal stresses for shallow surface flaws distributed around the weldment submodel of the HB-3 nozzle. View of interior surface of the HFIR pressure vessel.

**Table 4. Principal Stresses Around Weld Line of HB-3 Nozzle
as a Function of Distance from the Inner Wall.**

Position	Principal Stresses			
	R [in.]	σ_1 [ksi]	σ_2 [ksi]	σ_3 [ksi]
HB3_W1	0.00000	-1.056	7.203	20.024
	0.80021	-0.726	7.216	18.278
	1.60041	-0.392	7.189	16.549
	2.40062	-0.160	7.356	14.863
	3.20082	0.015	7.570	13.202
HB3_W2	0.00000	-1.103	4.688	18.248
	0.78381	-0.642	5.292	17.362
	1.10445	-0.586	4.952	17.590
	2.35137	-0.156	6.287	16.055
	3.13510	0.040	6.514	15.819
HB3_W3	0.00000	-1.244	5.853	7.671
	0.78125	-0.628	7.178	9.908
	1.56255	-0.180	8.406	12.182
	2.34381	-0.093	9.535	14.300
	3.12511	0.132	10.761	16.408
HB3_W4	0.00000	-1.118	2.627	16.423
	0.78381	-0.637	4.250	15.901
	1.56762	-0.255	5.808	15.430
	2.35137	-0.148	7.197	15.326
	3.13510	0.043	8.281	15.554
HB3_W5	0.00000	-1.093	4.662	17.366
	0.80021	-0.706	5.872	16.174
	1.60041	-0.332	7.037	14.970
	2.40062	-0.130	8.396	13.787
	3.20082	0.049	9.818	12.633
HB3_W6	0.00000	-1.226	2.717	21.102
	0.84026	0.520	4.792	21.056
	1.68043	1.791	7.332	21.090
	2.52069	1.662	9.119	19.716
	3.36094	1.468	10.835	18.406
HB3_W7	0.00000	-0.985	4.251	11.429
	0.86199	-0.734	6.016	11.891
	1.72384	-0.439	7.756	12.385
	2.58583	-0.144	9.424	13.343
	3.44773	0.092	11.118	14.271
HB3_W8	0.00000	-1.227	4.438	23.158
	0.84026	0.555	5.844	22.712
	1.68043	1.666	7.887	22.316
	2.52069	1.548	8.600	20.389
	3.36094	1.409	9.176	18.604

5. SUMMARY AND CONCLUSIONS

Three-dimensional linear elastic stress analyses of the HFIR HB-2 and HB-3 nozzles have been performed. Finite element models were developed using the PATRAN pre-processing code and translated into ABAQUS input file format. A scoping analysis using simple geometries with internal pressure loading was carried out to assess the capabilities of the ABAQUS/Standard code to calculate maximum principal stress distributions within cylinders with and without holes. These scoping calculations were also used to provide estimates for the variation in tangential stress around the rim of a nozzle using the superposition of published closed-form solutions for the stress around a hole in an infinite flat plate under uniaxial tension. From the results of the detailed finite element models, peak stress concentration factors (based on the maximum principal stresses in tension) were calculated to be 3.0 for the HB-2 nozzle and 2.8 for the HB-3 nozzle. The results of both the HB-2 and HB-3 nozzle analyses indicate that no significant stress gradients extend from the nozzles out to the side symmetry planes, implying a negligible interaction between nozzles in adjacent quadrants. Submodels for each nozzle were built to calculate the maximum stress distribution in the weldment region around the nozzle, where displacement boundary conditions for the submodel were automatically calculated by ABAQUS using the global nozzle model results. Maximum principal stresses are plotted and tabulated for eight positions around the nozzle. The HB-2 and HB-3 nozzle PATRAN databases and ABAQUS input models have been archived to be available for any future studies.

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