ENCLOSURE 2

NUH32PHB-0400, Revision 1

Calvert Cliffs Nuclear Power Plant, LLC
June 19, 2012

AREVA TRANSNUCLEAR INC.	**Form 3.2-1 Calculation Cover Sheet** TIP 3.2 (Revision 4)	**Calculation No.:**	**NUH32PHB-0400**
		Revision No.:	**1**
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DCR NO (if applicable) : NUH32PHB-006	**PROJECT NAME:** NUHOMS® 32PHB System
PROJECT NO: 10955	**CLIENT:** CENG - Calvert Cliff Nuclear Power Plant (CCNPP)

CALCULATION TITLE:

Benchmarking of the ANSYS Model of the OS200FC Transfer Cask

SUMMARY DESCRIPTION:

1) Calculation Summary

This calculation benchmarks the thermal analyses of the OS200FC Transfer Cask loaded with 32PTH1 DSC using ANSYS against the calculation performed in [7] using Thermal Desktop® and SINDA/FLUINT when forced air circulation is used to improve the thermal performance of the system.

2) Storage Media Description

Secure network server initially, then redundant tape backup

If original issue, is licensing review per TIP 3.5 required?

Yes ☐ **No** ☒ **(explain below)** **Licensing Review No.:**__________

This calculation is prepared to support a Site Specific License Application by CCNPP that will be reviewed and approved by the NRC. Therefore, a 10CFR72.48 licensing review per TIP 3.5 is not applicable.

Software Utilized (subject to test requirements of TIP 3.3):	**Version:**
ANSYS	10.0
Calculation is complete: Originator Name and Signature: Venkata Venigalla *Venkat*	Date: 06/24/2010
Calculation has been checked for consistency, completeness and correctness: Checker Name and Signature: Davy Qi	Date: 06/24/2010
Calculation is approved for use: Project Engineer Name and Signature: Kamran Tavassoli	Date: 7/14/10

REVISION SUMMARY

REV.	DESCRIPTION	AFFECTED PAGES	AFFECTED Computational I/O
0	Initial Issue	All	All
1	Update the title for Table 5-6	1,2 4 and 21	None

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1.0 PURPOSE

Thermal Desktop® and SINDA/FLUINT were used in the previous applications such as Amendment 8 and Amendment 10 to NUHOMS® UFSAR [2] to simulate the forced air cooling option for transfer casks. The SINDA/FLUINT models for OS197FC and OS200FC transfer casks (TCs) were studied extensively and approved by NRC as documented in the SER to Amendment 10 [2]. In order to use ANSYS to simulate the forced air cooling option for the CCNPP-FC TC, the ANSYS model is validated in this calculation by benchmarking it against the Thermal Desktop® and SINDA/FLUINT model. The OS200FC TC loaded with 32PTH1 DSC and heat loads of 31.2 kW and 40.8 kW are considered for the benchmarking to envelope the conditions expected for the CCNPP-FC TC with a maximum heat load of 29.6 kW.

2.0 REFERENCES

1 U.S. Code of Federal Regulations, Part 71, Title 10, "Packaging and Transportation of Radioactive Material".

2 Updated Final Safety Analysis Report for the Standardized NUHOMS® Horizontal Modular Storage System for Irradiated Nuclear Fuel, NUH-003, Rev. 11.

3 Rohsenow, Hartnett, Cho, "Handbook of Heat Transfer", 3rd Edition, 1998.

4 Rohsenow, Hartnett, Ganic, "Handbook of Heat Transfer Fundamentals", 2nd Edition, 1985.

5 ASME Boiler and Pressure Vessel Code, Section II, Part D, "Material Properties", 1998 Edition through 2000 Addenda.

6 ANSYS computer code and On-Line User's Manuals, Version 10.0.

7 Calculation, "Thermal Analysis of OS200 Transfer Cask Loaded with 32PTH1 DSC", Transnuclear, Inc., Calculation No. NUH32PTH1-0450, Rev. 1.

8 NOT USED.

9 Calculation, "Thermal Analysis of MP197HB Transport Cask for Normal Conditions of Transport", Transnuclear, Inc., Calculation No. MP197HB-0401, Rev. 1.

10 Henninger, J. H., "Solar Absorptance and Thermal Emittance of Some Common Spacecraft Thermal-Control Coatings," NASA Scientific and Technical Information Branch, NASA Reference Publication 1121, 1984.

3.0 ASSUMPTIONS AND CONSERVATISM

All the assumption and conservatism considered in this calculation are the same as those described in [7] for the purpose of benchmarking. These assumptions and conservatism are listed in the following sections.

3.1 OS200FC Mass Flow Rate Model

The annulus between the DSC shell and the TC inner liner are divided into parallel, individual segments along the DSC axis. No circumferential air flow is considered between the parallel segments. Since the presence of circumferential flow will tend to exchange hotter air in the narrower segments of the annulus with cooler air in the wider segments of the annulus, ignoring the potential for circumferential flow will yield conservative temperature estimates for the peak temperatures on the DSC shell and TC inner liner [7].

Based on [7], an air flow rate of 450 cfm is considered for forced air cooling. To evaluate the air flow rate in each of the parallel segments, a constant pressure boundary condition is applied at the inlet such that the total mass flow rate at the outlet is equal to the total airflow rate of 450 cfm. Since the pressure drop through the annulus between the DSC shell and TC inner shell is the major factor controlling the amount of air flow rate in each segment, the mass flow rate model considers only the annulus over the length of the DSC to determine the mass flow rate through each segment.

3.2 OS200FC TC Model

Heat load is simulated by heat generation distributed uniformly over the basket length on the homogenized region. The fuel basket is assumed to be centered within the DSC cavity. As such, a 1.25 inch long helium filled gap is assumed between the top and bottom of the fuel basket region and the DSC's closure lid and bottom end plug. An approximate 13.25 inch long cask spacer is used to position the short DSC within the TC cavity. For the purposes of this calculation the spacer is modeled as 0.75-inch thick inner and outer cylinders enclosed with 0.75-inch thick top and bottom plates. The inner cylinder is assumed to have a 22 inch ID, while the outer cylinder is assumed to have a 66.5 inch ID [7].

For regions between the canister support rails (orientation 150 to 180°) convection from forced air flow is ignored due to the narrowness of the gap between the DSC and TC inner liner.

For the transfer operation in horizontal orientation, the lower halves of the TC cylindrical surfaces are not exposed to insolance. No solar heat flux is considered over these surfaces. To remove any uncertainty about the solar impact on the vertical surfaces, the entire surface areas of vertical surfaces are considered for application of the solar heat flux.

No convection is considered for the exterior vertical surface of the liquid filled neutron shield shell.

When the TC is in the horizontal orientation, the DSC is supported by the four canister rails depicted in Figure 2-2 of [7]. The thermal resistance between the DSC and the canister rails is assumed to be approximately 2.7 Btu/hr-in^2-ºF [7]. The material properties are listed in Section 4.3.

The grapple ring is not modeled in the current analysis; it is conservatively replaced with air.

Radiation heat exchange is considered between the DSC/Spacer and the cask inner liner, between the lead gap and structural shell and also between the fuel basket and the DSC top and bottom end cover plates by using the ANSYS [6] AUX12 processor.

The following gaps are considered in the OS200FC transfer cask model:

- 0.037" radial gap between the gamma shield and the structural shell [7].

0.037" radial gap assumed between the gamma shield and the structural shell is based on the calculated uniform lead gap at room temperature for OS200FC TC in [7]. This assumption is conservative since the lead gap reaches its maximum dimension of 0.037" at room temperature and decreases to 0" at the lead melt point.

4.0 DESIGN INPUT

4.1 Benchmarking Cases

The following cases shown in Table 4-1 are analyzed in this calculation to benchmark the thermal performance of the OS200FC TC with 32PTH1 DSC with forced air flow using ANSYS [6]. These benchmarking cases envelope the conditions expected for CCNPP-FC TC with 29.6 kW heat load with forced cooling. Design load cases such as the vertical loading where the forced cooling is not used are not considered in this evaluation.

Table 4-1 Design Load Cases for 32PTH1 DSC in OS200FC TC

		Applicable Conditions			
		Air Flow	Ambient Temperature	Insolation	Decay Heat
Case	Description	(cfm)	(°F)	Max (1)	(kW)
1	Normal Hot	450	106	x	40.8
2	Normal Hot	450	106	x	31.2

Notes:
(1) Insolation in accordance with 10CFR71.71(c) (1) [1].

4.2 Major Dimensions in the OS200FC TC Model

Major dimension of 32PTH1 DSC used in the OS200FC TC model are the same as those used in [7].

4.3 Thermal Properties of Materials in the OS200FC TC Model

Materials used in the OS200FC TC model are listed in Table 4-2 and are the same as those used in [7].

The material properties for SA240 Type 304, ASTM B29 Lead and NS-3 are listed in Table 3-1 of [7] and the same are used in this analysis. The material properties for the water filled neutron shield are listed in Table 3-4 of [7] for radial directions and Table 3-3 of [7] for axial directions. The material properties for the homogenized fuel basket are listed in Table 3-2 of [7].

The effective conductivity values used in this calculation are noted in Table 4-3 through Table 4-4. The axial effective conductivity for the the spacer are calculated using the methodology described in Section 5.3 of [9].

The heat transfer coefficients for the forced air flow over the DSC/TC annulus are calculated using the same correlations described in [7],Section 4.2 and are presented in MassFlow_ConvCoeff_32PTH1_31kW.xls for the 31.2 kW heat loads and MassFlow_ConvCoeff_32PTH1_41kW.xls for the 40.8 kW heat loads as noted in Table 8-3.

Table 4-2 List of Materials in the OS200FC TC Model

Component	Mat # in ANSYS Model	Material
DSC Shell	1	SA240-Type 304
DSC Homogenized Bottom Plates	2,3	SA240-Type 304(1)
DSC Outer Top Cover Plate	4	SA240-Type 304
DSC Top Shield Plug	5	SA240-Type 304(1)
DSC Helium	33	Helium
DSC Homogenized Fuel Basket	41	Eff. Conductivity See Table 3-2 of [7]
TC Inner Shell	7	SA240-Type 304
TC Gamma Shield	8	Lead
TC Structural Shell	9	SA240-Type 304
TC Top NS-3 / Top Cover	10	SA240-Type 304
TC Top Cask Lid	11	XM-19
TC Inner/Outer Spacer Disc (Cylindrical Shell)	13	SA240-Type 304
TC Inner/Outer Spacer Disc (Top and Bottom Plates)	14	Effective Conductivity of SA240-Type 304 with 0.0625" Axial Gap
TC Outer Shell	17	SA240-Type 304
TC NS-3 Top and Bottom	18	NS-3
TC Neutron Shield – 1 (top & bottom)	21	Eff. Conductivity, See Table 3-4 of [7] (Shield Sections 1 & 17)
TC Neutron Shield – 2	22	Eff. Conductivity, See Table 3-4 of [7] (Shield Sections 2 to 13)
TC Neutron Shield – 3	23	Eff. Conductivity, See Table 3-4 of [7] (Shield Sections 14 to 16)
TC Cask Slide Rail	43	Effective Properties (See Table 4-4)
TC FLUID116 Flow Elements	31	--
TC LINK34 Convection Elements	61-75	--
TC LINK34 Convection Elements (At Entrance though wedges)	81	--
TC LINK34 Convection Elements (At Exit thought Top Cask Lid)	82	--
TC Air	32	Air
TC Gamma Shield Air Gap	34	Eff. Conductivity, See Table 4-1 of [7]
TC DSC/Cask Annulus Air Gap	36	Air

Note: (1) Assumed as SA240 Type 304 for consistency with [7]

Table 4-3 Thermal Conductivity of Spacer Disc

Top and bottom plates of Spacer

Plate thickness = 0.75 in
Gap thickness = 0.0625 in

Temp	k_SS304 [7]	Temp	k_air	k_air [3]	k_axial	k_radial
(F)	(Btu/hr-in-F)	(K)	(W/m-K)	(Btu/hr-in-F)	(Btu/hr-in-F)	(Btu/hr-in-F)
70	0.717	294.4	0.0257	0.0012	0.015	0.717
100	0.725	311.1	0.0269	0.0013	0.015	0.725
200	0.775	366.7	0.0308	0.0015	0.017	0.775
300	0.817	422.2	0.0345	0.0017	0.019	0.817
400	0.867	477.8	0.0381	0.0018	0.021	0.867
500	0.908	533.3	0.0415	0.0020	0.023	0.908
600	0.942	588.9	0.0449	0.0022	0.025	0.942
700	0.983	644.4	0.0482	0.0023	0.027	0.983
800	1.025	700.0	0.0514	0.0025	0.029	1.025
900	1.058	755.6	0.0545	0.0026	0.031	1.058
1,000	1.092	811.1	0.0576	0.0028	0.032	1.092

Table 4-4 Thermal Conductivity of Slide Rail

Thickness	0.12	in
Contact Resistance [7]	2.7	Btu/hr-in^2-^{o}F

Temp	k_SS304 [5]	k_eff
(^{o}F)	(Btu/hr-in-^{o}F)	(Btu/hr-in-^{o}F)
70	0.717	0.223
100	0.725	0.224
200	0.775	0.228
300	0.817	0.232
400	0.867	0.236
500	0.908	0.239
600	0.942	0.241
700	0.983	0.244
800	1.025	0.246
900	1.058	0.248
1000	1.092	0.250

Table 4-5 Air Thermal Properties [3]

Temperature (K)	Thermal conductivity (W/m-K)	Temperature (°F)	Thermal conductivity (Btu/hr-in-°F)
200	0.01822	-100	0.0009
250	0.02228	-10	0.0011
300	0.02607	80	0.0013
400	0.03304	260	0.0016
500	0.03948	440	0.0019
600	0.04557	620	0.0022
800	0.05698	980	0.0027
1000	0.06721	1340	0.0032

The above data are calculated base on the following polynomial function from [3].

$k = \sum C_i \, T_i$ for conductivity in(W/m-K) and T in (K)

For 250 < T < 1050 K	
C0	-2.2765010E-03
C1	1.2598485E-04
C2	-1.4815235E-07
C3	1.7355064E-10
C4	-1.0666570E-13
C5	2.4766304E-17

Specific heat, viscosity, density and Prandtl number of air are used to calculate heat transfer coefficients in APPENDIX A based on the following data from [3].

$c_p = \sum A_i \, T_i$ for specific heat in (kJ/kg-K) and T in (K)

For 250 < T < 1050 K	
A0	0.103409E+1
A1	-0.2848870E-3
A2	0.7816818E-6
A3	-0.4970786E-9
A4	0.1077024E-12

$\mu = \sum B_i \, T_i$ for viscosity (N-s/m^2)$\times 10^6$ and T in (K)

For 250 < T < 600 K	
B0	-9.8601E-1
B1	9.080125E-2
B2	-1.17635575E-4
B3	1.2349703E-7
B4	-5.7971299E-11

For 600 < T < 1050 K	
B0	4.8856745
B1	5.43232E-2
B2	-2.4261775E-5
B3	7.9306E-9
B4	-1.10398E-12

$\rho = P / RT$ for density (kg/m^3) with P=101.3 kPa; R = 0.287040 kJ/kg-K; T = air temp in (K)

$\mathrm{Pr} = c_p \, \mu / k$ Prandtl number

Table 4-6 Helium Thermal Conductivity [3]

Temperature (K)	Thermal conductivity (W/m-K)	Temperature (°F)	Thermal conductivity (Btu/hr-in-°F)
300	0.1499	80	0.0072
400	0.1795	260	0.0086
500	0.2115	440	0.0102
600	0.2466	620	0.0119
800	0.3073	980	0.0148
1000	0.3622	1340	0.0174
1050	0.3757	1430	0.0181

The above data are calculated base on the following polynomial function from [3]

$k = \sum C_i \, T_i$ for conductivity in (W/m-K) and T in (K)

For 300 < T < 500 K		for 500< T < 1050 K	
C0	-7.761491E-03	C0	-9.0656E-02
C1	8.66192033E-04	C1	9.37593087E-04
C2	-1.5559338E-06	C2	-9.13347535E-07
C3	1.40150565E-09	C3	5.55037072E-10
C4	0.0E+00	C4	-1.26457196E-13

4.4 Surface Properties of Materials

All the surface properties used in this calculation are the same as those described in Section 3 of [7] and are described below for reference.

An emissivity of 0.587 is assumed for the exterior surfaces of the 32PTH1 DSC, the inner shell of the TC, the exterior surface of the liquid neutron shield, and the stainless steel skin enclosing the NS-3 material at the top and bottom of the TC. For conservatism, an emissivity of 0.46 is assumed for the machined stainless steel surfaces of the top and bottom forgings of the TC [7].

An emissivity value of 0.6 is used for the inner surface of the structural shell to account for the expected surface oxidation that will occur during the lead pour process [7].

An emissivity of 0.587 is assumed for the radiation exchange between the fuel basket and the DSC inner end plates.

Solar absorptance values of 0.39 and 0.47 are given in [10] for rolled and machined stainless steel plates, respectively. For conservatism, it is assumed that the absorptivity and the emissivity of stainless steel are equal in this calculation. Solar absorptivity of 0.587 is used for the exposed stainless steel surfaces.

4.5 Design Criteria

The following criteria are considered for the maximum differences between ANSYS and SINDA/FLUINT models for the benchmarking purposes. :

(1) ±5°F for the maximum DSC shell temperature.

(2) ±5% for the heat removed by forced convection cooling.

(3) ±10°F for the air exit temperature.

5.0 METHODOLOGY

The NUHOMS® OS200FC TC contains design provisions for the use of forced air circulation to improve its thermal performance. The system will consist of redundant, industrial grade pressure blowers and power systems, ducting, etc. When operating, the fan system is expected to generate a flow rate of 450 cfm or greater, which will be ducted to the location of the ram access cover at the bottom of the cask.

The following are the steps to determine the maximum steady state temperatures of the DSC/TC components with the forced convection using ANSYS:

1. Assume a ΔT_{air} for initial runs, Calculate T_{exit} and T_{avg} based on the initial guess and the air properties based on T_{avg}.

   Where,

   $$T_{exit} = T_{amb} + \Delta T_{air}$$
   $$T_{avg} = (T_{amb} + T_{exit})/2$$
   $$T_{amb} = 106°F$$

2. Run Flow Rate Model described in Section 5.1 iteratively based on average properties of air calculated in previous step to compute the air mass flow rate in each DSC/TC annulus segment. (Run ID: "FlowRate_32PTH1_31kW" for 31.2 kW and "FlowRate_32PTH1_41kW" for 40.8 kW load cases as listed in Table 8-2)
3. Determine the heat transfer coefficients within the annulus based on the mass flow rates computed in Step 2 (see worksheet "Hc_data" for mass flow rate in lbm/hr and Hc_calc for convection coefficients in "MassFlow_ConvCoeff_32PTH1_31kW.xls" for the 31.2 kW load case and "MassFlow_ConvCoeff_32PTH1_41kW.xls" for the 40.8 kW load case as noted in Table 8-3)
4. Run Thermal Model (Run ID: "TR_32PTH1_31" for 31.2 kW and "TR_32PTH1_41" for 40.8 kW load cases as listed in Table 8-2) described in Section 5.2 based on mass flow rates and heat transfer coefficients calculated in Step 2 and Step 3.
5. Calculate T_{exit}, T_{avg}, and ΔT_{air} based on results from Thermal Model in Step 4.
6. If difference between assumed ΔT_{air} in Step 1 and calculated ΔT_{air} in Step 5 is less than 1°F, stop iterations, otherwise proceed to Step 7.
7. Rerun the Flow Rate Model described in Section 5.1 and Step 2 with air properties based on T_{avg} from Step 5.
8. If differences between air mass flow rates in each DSC/TC annulus segment from Step 7 and Step 2 are less than 0.1 lbm/hr, stop iterations, otherwise proceed to Step 9.

9. Repeat Steps 4 to 9 until the solution converges.

5.1 Flow Rate Model

The forced air enters the TC from the ram access opening and the airflow turns and enters the ten (10) flow paths formed by the 1.0" thick wedge segments welded to the TC's bottom. After the forced air exits from the flow paths formed by the wedge segments, the airflow turns and flows in the annulus between the DSC and the TC's inner liner. Given the gap between the DSC and TC varies with circumferential position, plus variances in the heating of the air, the airflow will distribute itself around the circumference of the DSC/TC inner liner, until an equal pressure drop is achieved everywhere.

For the purposes of this calculation, each half of the annulus is divided into 19 angular segments as shown in Table 5-1 with 0° at the top of the normally horizontal TC and 180° at the bottom. The mass flow rate along each of the 19 angular segments is calculated using the Flow Rate Model. The mass flow rates obtained from this model are used as input to the thermal model of the DSC/TC described in Section 5.2.

The 19 annular segments for forced air flow are modeled using FLUID116 elements with their length equal to the length of the DSC. The potential for circumferential airflow is conservatively ignored as discussed in Section 3.1. The flow area and hydraulic diameter for each annular segment are calculated based on the position of the DSC within the TC cavity. The determination of the gap between the DSC and the TC inner liner as a function of circumferential position was made considering a DSC shell outer diameter of 69.75 inches, a TC inner liner inner diameter of 70.5 inches, and two 0.120-inches thick slide rails that are located 12° from the centerline of the TC. The second set of sliding rails are conservatively ignored. Table 5-1 presents the calculation basis for the gap between the TC and DSC and the associated hydraulic diameter and air flow area.

Table 5-1 TC/DSC Annulus Hydraulic Diameter and Flow Area Calculation

Section	Angle Segment		Angle (degrees)	Cask "X" Location (in)	Cask "Y" Location (in)	DSC "X" Location (in)	DSC "Y" Location (in)	Hydraulic Diameter (m)	Flow Area (m^2)
1	-5	5	0	0.000	35.250	0.000	34.614	0.0323	2.479E-03
2	5	15	10	6.121	34.714	6.011	34.092	0.0321	2.464E-03
3	15	25	20	12.056	33.124	11.844	32.541	0.0315	2.419E-03
4	25	35	30	17.625	30.527	17.324	30.007	0.0305	2.345E-03
5	35	45	40	22.658	27.003	22.289	26.563	0.0292	2.245E-03
6	45	55	50	27.003	22.658	26.587	22.309	0.0276	2.121E-03
7	55	65	60	30.527	17.625	30.090	17.372	0.0257	1.978E-03
8	65	75	70	33.124	12.056	32.688	11.897	0.0236	1.819E-03
9	75	85	80	34.714	6.121	34.301	6.048	0.0214	1.648E-03
10	85	95	90	35.250	0.000	34.875	0.000	0.0191	1.473E-03
11	95	105	100	34.714	-6.121	34.390	-6.064	0.0167	1.296E-03
12	105	115	110	33.124	-12.056	32.856	-11.958	0.0145	1.125E-03
13	115	125	120	30.527	-17.625	30.316	-17.503	0.0124	9.637E-04
14	125	135	130	27.003	-22.658	26.844	-22.525	0.0105	8.177E-04
15	135	145	140	22.658	-27.003	22.546	-26.869	0.0089	6.914E-04
16	145	155	150	17.625	-30.527	17.551	-30.398	0.0076	5.888E-04
17	155	165	160	12.056	-33.124	12.012	-33.002	0.0066	5.131E-04
18	165	175	170	6.121	-34.714	6.101	-34.598	0.0060	4.667E-04
19	175	185	180	0.000	-35.250	0.000	-35.136	0.0058	4.510E-04

The friction factor along the length of the DSC/cask annulus is calculated as:

$f = (1.58 * \ln \mathrm{Re} - 3.28)^{-2}$ [Section 4.2.2.2 of 7]

Table 5-2 lists the friction factors as a function of Reynolds numbers.

Table 5-2 List of the Friction Factors in the Mass Flow Model

Re	f	4*f	Re	f	4*f
1	0.093	0.372	1500	0.015	0.058
100	0.063	0.250	1750	0.014	0.055
200	0.039	0.154	2000	0.013	0.052
300	0.030	0.122	3000	0.011	0.046
400	0.026	0.105	4000	0.010	0.041
500	0.023	0.094	5000	0.010	0.039
600	0.021	0.086	6000	0.009	0.037
800	0.019	0.075	8000	0.008	0.034
1000	0.017	0.069	12500	0.007	0.030
1250	0.016	0.063	22500	0.006	0.025

The areas, hydraulic diameters, and friction factors calculated for the 19 annular segments are applied as real constants to the FLUID116 elements. The friction factors are applied using the TB,FCON command as function of temperature and Reynolds number.

The total mass flow rate based on 450 cfm discharge from the fan for the 40.8 kW load case is calculated as follows:

$$\dot{m} = 450\ cfm * \rho_{air,average\ temp}$$
$$= 450 * \frac{(0.3048)^3}{60} * 0.986 \frac{kg}{s}$$
$$= 0.2094\ kg/s$$

Where,

$\dot{m}$ = Total Mass Flow Rate, (kg/s)

$\rho_{air,average\ temp}$ = Density of air based on Average Air Temperature (kg/m^3)

The air density in the above equation is calculated based on a pressure drop of 6 inches water gauge taken from calculations presented in Section 4.2.2.1 of [7] and an initial air exit temperature of 273°F for the 40.8 kW heat load case. The final air exit temperature is determined iteratively through the steps shown in Section 5.0. The same evaluation for 31.2 kW heat load case and an initial air exit temperature of 239°F results in a total mass flow rate of 0.2151 kg/s.

The forced air introduced in the annular gap between the DSC and the cask distributes itself based upon the flow area and hydraulic diameter. The Flow Rate Model computes the air flow rate in each annular segment based on achieving an equal pressure drop over any segments of the annulus. The Flow Rate Model for determining the mass flow rates is shown in Figure 5-2.

A constant volumetric airflow rate of 450 cfm is assumed to evaluate the air mass flow rate in each of the parallel segments. A constant pressure is applied at the inlet of the air flow into the DSC/TC annulus and the mass flow at the outlet is computed for the flow along the 19 annular segments. The pressure at the inlet is iteratively changed until the total mass flow rate at outlet of the 19 annular segments is equal to total mass flow rate of 0.2094 kg/s and 0.2151 kg/s for the 40.8 and 31.2 kW load cases respectively.

The mass flow rates obtained for each of the 19 angular segments for use in the OS200FC TC thermal model along with the hydraulic diameters and flow areas are presented in Table 5-3 for the 40.8 kW and 31.2 kW load cases.

Table 5-3 Mass Flow Rates Along Each Annular Segment

	40.8 kW	31.2 kW		
Section	Massflow (lbm/hr)	Massflow (lbm/hr)	Hydraulic Diameter (in)	Flow Area (in^2)
1	100.3	102.9	1.27	3.84
2	99.3	101.8	1.26	3.82
3	96.3	98.7	1.24	3.75
4	91.3	93.7	1.20	3.63
5	84.3	86.9	1.15	3.48
6	76.1	78.3	1.09	3.29
7	67.2	69.0	1.01	3.07
8	57.8	59.5	0.93	2.82
9	48.8	50.1	0.84	2.56
10	40.2	41.3	0.75	2.28
11	31.8	32.7	0.66	2.01
12	24.6	25.4	0.57	1.74
13	18.6	19.1	0.49	1.49
14	13.8	14.2	0.41	1.27
15	10.1	10.5	0.35	1.07
16	7.5	7.8	0.30	0.91
17	5.8	6.0	0.26	0.80
18	4.8	5.0	0.24	0.72
19	4.5	4.7	0.23	0.70

5.2 OS200FC TC Model

A half-symmetric, three-dimensional finite element model of OS200FC TC loaded with 32PTH1 DSC simulating forced air flow is developed using ANSYS Version 10.0 [6] to provide the maximum component temperatures for the benchmarking purposes.

The model contains the cask shells, cask bottom plate, cask lid, canister shell, spacer and canister end plates with a homogenized basket.

SOLID70 elements are used to model the components including the gaseous gaps. Surface elements SURF152 are used for applying the insolation boundary conditions. Radiation between the homogenized fuel basket and the canister inner cover plates, along the gap between canister and TC inner liner and also along the gap between the gamma shield and structural shell is modeled using the AUX12 processor with SHELL57 elements used to compute the form factors.

Decay heat load is applied as a uniform volumetric heat generated throughout the homogenized region of the basket. The volumetric heat generation rate is calculated as:

$$q''' = \frac{Q}{\pi\,(D_i/2)^2\,L_b}$$

q''' = Volumetric Heat Generation Rate (Btu/hr-in^3)
Q = decay heat load (Btu/hr) (to convert from kW multiply by 3412.3)
D_i = Canister inner Diameter (in)
L_b = Basket length (in)

The applied decay heat values in the model are listed in Table 5-4

Table 5-4 Decay Heat Flux

Heat Load (kW)	Heat Load (Btu/hr)	D_i (in)	L_b (in)	Decay heat flux (Btu/hr-in^2)
31.2	106464	68.75	162	0.1770
40.8	139222			0.2315

The insolance is applied as a heat flux over the TC outer surfaces using average insolence values from 10CFR71 [1]. The insolance values are averaged over 12 hours and multiplied by the surface absorptivity factor to calculate the solar heat flux. The solar heat flux values used in OS200FC TC model are summarized in Table 5-5.

Table 5-5 Solar Heat Flux

Surface Material	Shape	Insolance over 12 hrs [1] (gcal/cm^2)	Solar Absorptivity (1)	Total solar heat flux averaged over 12 hrs (Btu/hr-in^2)
Stainless Steel	Curved	400	0.587 (2)	0.501
	Flat vertical	200	0.587 (2)	0.250

Notes:
(1) See Section 4.4 for surface properties.
(2) Solar absorptivity of stainless steel is taken equal to its emissivity.

Convection and radiation heat transfer from the cask outer surfaces are combined together as total heat transfer coefficients. The total heat transfer coefficients are calculated using free convection correlations from Rohsenow Handbook [3] and are incorporated in the model using ANSYS macros. These correlations are described in APPENDIX A. The ANSYS macros used in this calculation are listed in Section 8.0.

During transfer when the cask in a horizontal orientation, the canister shell rests on two slide rails in the TC. These rails are flat stainless steel plates welded to the inner shell of the TC. The thickness of the slide rail is 0.12".

The angle between the lower rail and the vertical plane is 12 degree. Considering this configuration shown in Figure 5-1, the distance between the centerline of DSC and centerline of the cask are calculated as follows.

$$R_2^{\,2} = R_1^{\,2} + x^2 - 2\,R_1\,x\,Cos(\alpha)$$

With

$R_1 = D_{i,\,TC}\,/2 - t_{rail}$
$R_2 = D_{o,\,DSC}\,/\,2$
$\alpha = 12^\circ$
x = Distance between the canister and TC centerlines
$D_{i,\,TC}$ = Inner diameter of TC
$D_{o,\,DSC}$ = Canister outer diameter
t_{rail} = cask slide rail thickness = 0.12"

The calculated value for x is listed in Table 5-6. In the ANSYS model, the canister is shifted down by the amount of x in the Cartesian y-direction within the TC cavity.

Table 5-6 Distance between 32PHB Canister and TC Centerline

DSC Type	$D_{i,TC}$ (in)	$D_{o,DSC}$ (in)	R_1 (in)	R_2 (in)	α (degree)	x (in)
32PTH1	70.5	69.75	35.13	34.88	12	0.26

Forced air circulation through the annulus of the DSC/TC is modeled using the FLUID116 and LINK34 elements. The FLUID116 element models the forced air flow along the axial length of the DSC/cask annulus by conducting heat and transmitting the fluid between its nodes, whereas the LINK34 elements model the convection from the DSC/TC surfaces due to the forced air flow. The FLUID116 elements are modeled such that they are connected to the LINK34 convection elements.

The mass flow rates obtained from the Flow Rate Model described in Section 5.1 for each of the annular segments from 0° to 150° are applied to the FLUID116 elements using the "SFE,,,hflux" command.

Based on the mass flow rates obtained for each of the annular segments from 0° to 150°, the convection heat transfer coefficients for the DSC/TC annulus are computed using the correlations for flow within ducts and pipes. The convection heat transfer coefficients are computed as a function of the local hydraulic diameter, the Reynolds number, and the

thermophysical properties of air. These convection heat transfer coefficients are applied to the LINK34 elements using the mpdata,hf/mp,hf commands.

The correlations for the convection coefficients are identical to those in [7] and are taken from equations 7, 43, 44, 45, 57, and 57a from Chapter 7 of [4] as follows:

For $0.5 < Pr < 2000$ and $10^4 < Re < 5x10^6$:

$$Nu = \frac{h_c D_h}{k} = \frac{Re \times Pr \times f/2}{1.07 + 12.7(Pr^{2/3} - 1)(f/2)^{0.5}}$$

$$Re = \frac{V \times \rho \times D_h}{\mu}$$

$$f = (1.58 \times \ln Re - 3.28)^{-2}$$

For $0.5 < Pr < 2000$ and $3000 < Re \leq 10^4$:

$$Nu = \frac{h_c D_h}{k} = \frac{(Re - 1000) \times Pr \times f/2}{1.0 + 12.7(Pr^{2/3} - 1)(f/2)^{0.5}}$$

For $0.5 < Pr < 2000$ and $0 < Re \leq 3000$:

$$Nu = \frac{h_c D_h}{k} = 2.035 \times (x^*)^{-(1/3)} - 0.7, \qquad \text{for } x^* \leq 0.01$$

$$Nu = 2.035 \times (x^*)^{-(1/3)} - 0.2, \qquad \text{for } 0.01 < x^* < 0.06$$

$$Nu = 3.657 + 0.0998/x^*, \qquad \text{for } x^* \geq 0.06$$

Where:

Nu = Nusselt number
h_c = convection coefficient
D_h = hydraulic diameter
k = thermal conductivity of fluid at film temperature
V = flow velocity
ρ = density of fluid at the film temperature
μ = dynamic viscosity
Pr = Prandtl number
f = friction factorRe = Reynolds number
x^* = entry length factor = $x/Re/D_h/Pr$
x = length of duct/pipe

Forced convection is omitted conservatively and conduction is assumed in the region between the canister support rails (i.e., approximately 150° to 180°) due to the narrowness of the gap

between the DSC and the TC inner liner. Based on the above correlations and the mass flow rates from Section 5.1 the heat transfer coefficients for the annular segments from 0° to 150° are calculated and are presented in Table 5-7 and Table 5-8 for the 40.8 and 31.2 load cases, respectively.

The material properties used in the OS200FC TC model are listed in Section 4.0.

The geometry of the OS200FC TC Model is shown in Figure 5-3 through Figure 5-5.

Typical boundary conditions for the Thermal Model of OS200FC TC are shown in Figure 5-6 through Figure 5-7.

The OS200FC transfer cask with 31.2 kW and 40.8 kW heat loads and forced cooling are analyzed under steady-state conditions using SINDA/FLUINT in [7]. Therefore, steady-state analyses are performed using ANSYS to compare the maximum component temperatures for the 32PTH1 DSC in an OS200FC transfer cask with 31.2 kW and 40.8 kW heat loads with forced cooling.

Table 5-7 Heat Transfer Coefficients in the DSC/TC Annulus for Forced Air Flow (40.8 kW Load Case)

	Heat Transfer Coefficients (Btu/hr-in^2-^{o}F)								
Temp (^{o}F)	Wedge[(1)]	Section 1	Section 2	Section 3	Section 4	Section 5	Section 6	Section 7	Section 8
110	0.004	0.027	0.027	0.027	0.027	0.026	0.025	0.024	0.023
210		0.028	0.028	0.028	0.028	0.027	0.026	0.025	0.024
310		0.029	0.029	0.029	0.028	0.028	0.027	0.025	0.024
410		0.030	0.030	0.029	0.029	0.028	0.027	0.026	0.024
510		0.030	0.030	0.030	0.029	0.029	0.028	0.026	0.024

	Heat Transfer Coefficients (Btu/hr-in^2-^{o}F)							
Temp (^{o}F)	Section 9	Section 10	Section 11	Section 12	Section 13	Section 14	Section 15	Exit at Top Lid [(1)]
110	0.022	0.020	0.009	0.009	0.011	0.012	0.014	0.016
210	0.022	0.020	0.010	0.011	0.012	0.014	0.016	
310	0.022	0.010	0.011	0.012	0.013	0.015	0.018	
410	0.022	0.011	0.012	0.013	0.015	0.017	0.020	
510	0.011	0.012	0.013	0.014	0.016	0.018	0.022	

Notes:

(1) The lowest heat transfer coefficient is used for the Wedge and Exit at Top for conservatism

Table 5-8 Heat Transfer Coefficients in the DSC/TC Annulus for Forced Air Flow (31.2 kW Load Case)

Temp (°F)	Wedge(1)	Heat Transfer Coefficients (Btu/hr-in^2-°F) Section 1	Section 2	Section 3	Section 4	Section 5	Section 6	Section 7	Section 8
110	0.004	0.028	0.028	0.028	0.027	0.027	0.026	0.025	0.024
210		0.029	0.029	0.028	0.028	0.028	0.027	0.026	0.024
310		0.030	0.029	0.029	0.029	0.028	0.027	0.026	0.025
410		0.030	0.030	0.030	0.030	0.029	0.028	0.027	0.025
510		0.031	0.031	0.031	0.030	0.029	0.028	0.027	0.025

	Heat Transfer Coefficients (Btu/hr-in^2-°F)							
Temp (°F)	Section 9	Section 10	Section 11	Section 12	Section 13	Section 14	Section 15	Exit at Top Lid (1)
110	0.022	0.021	0.009	0.009	0.011	0.012	0.014	0.016
210	0.023	0.021	0.010	0.011	0.012	0.014	0.016	
310	0.023	0.010	0.011	0.012	0.013	0.015	0.018	
410	0.023	0.011	0.012	0.013	0.015	0.017	0.020	
510	0.011	0.012	0.013	0.014	0.016	0.018	0.022	

Notes:

(1) The lowest heat transfer coefficient is used for the Wedge and Exit at Top for conservatism

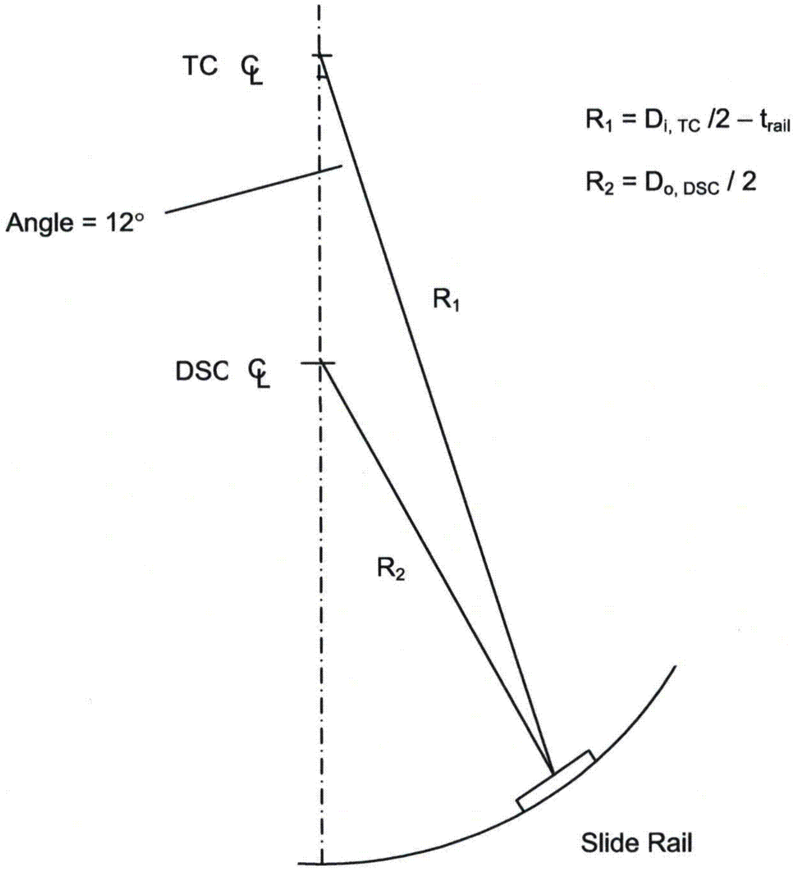


Figure 5-1 Location of 32PTH1 DSC within OS200 FC TC

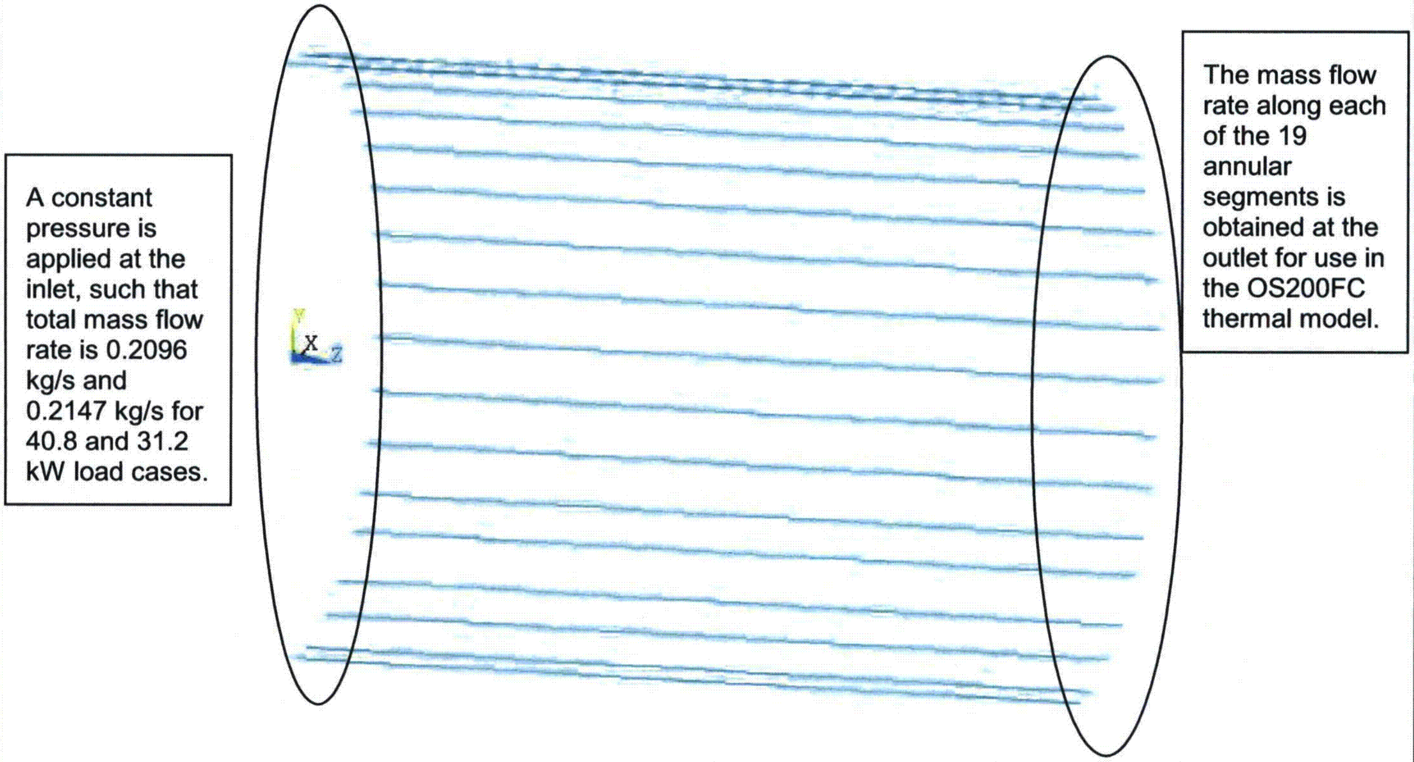


Figure 5-2 Finite Element Mesh of Flow Rate Model with FLUID116 Elements

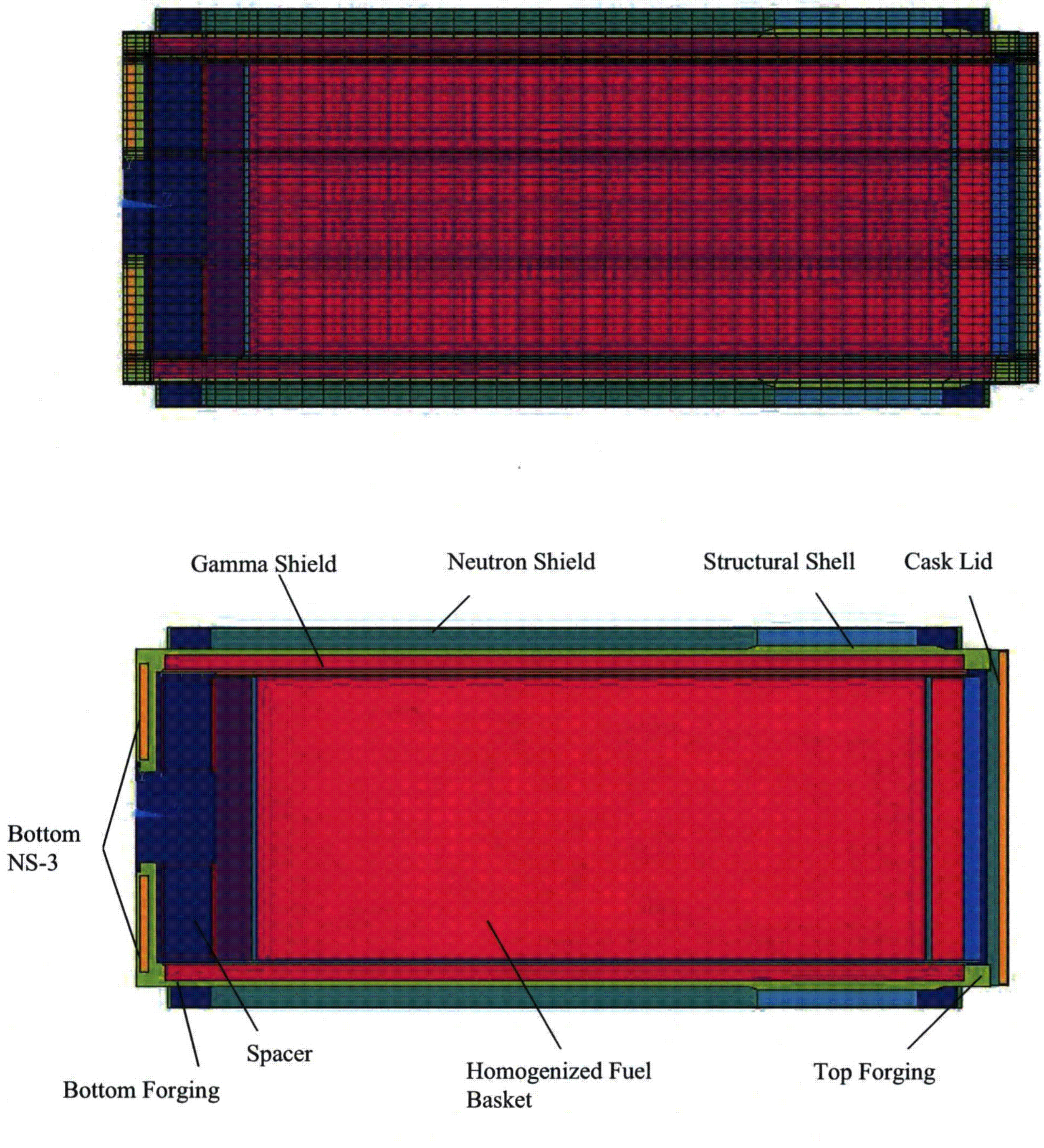


Figure 5-3 Finite Element Model of OS200FC TC with 32PTH1 DSC

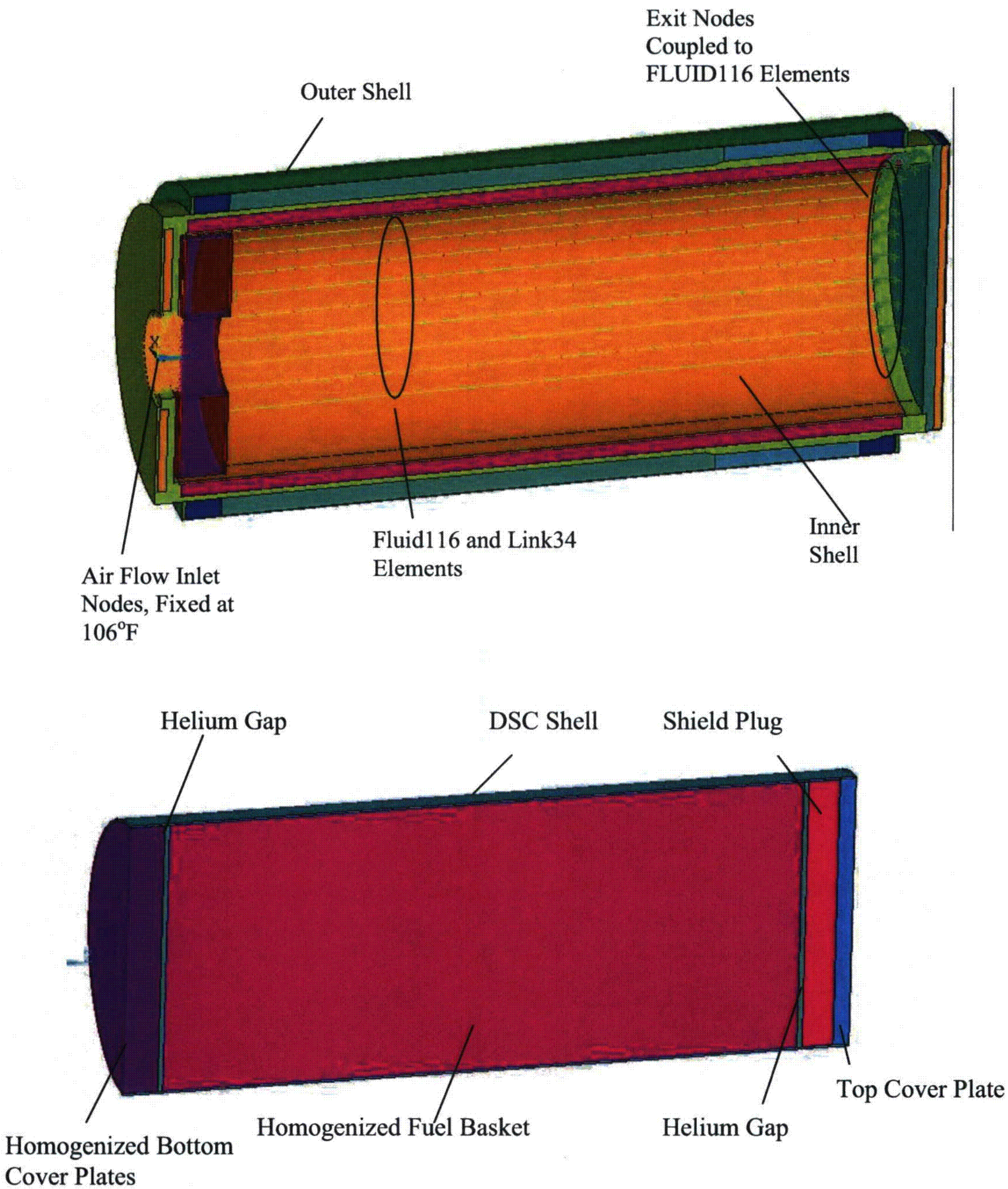
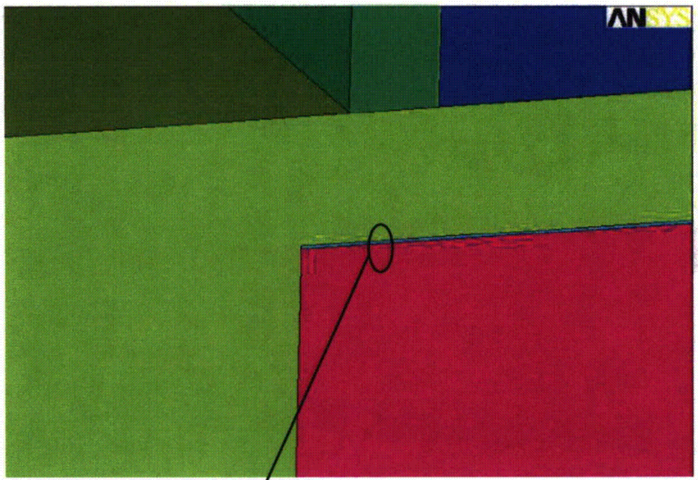
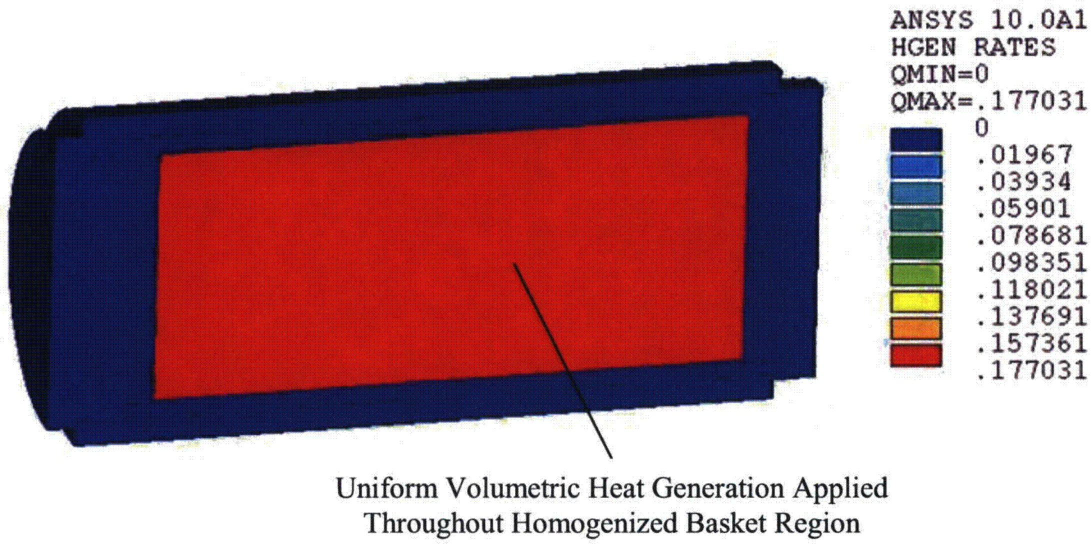


Figure 5-4 OS200FC TC Finite Element Model, Components



0.037" radial gap between structural shell and gamma shield

Figure 5-5 Gaps in OS200FC Transfer Cask Model



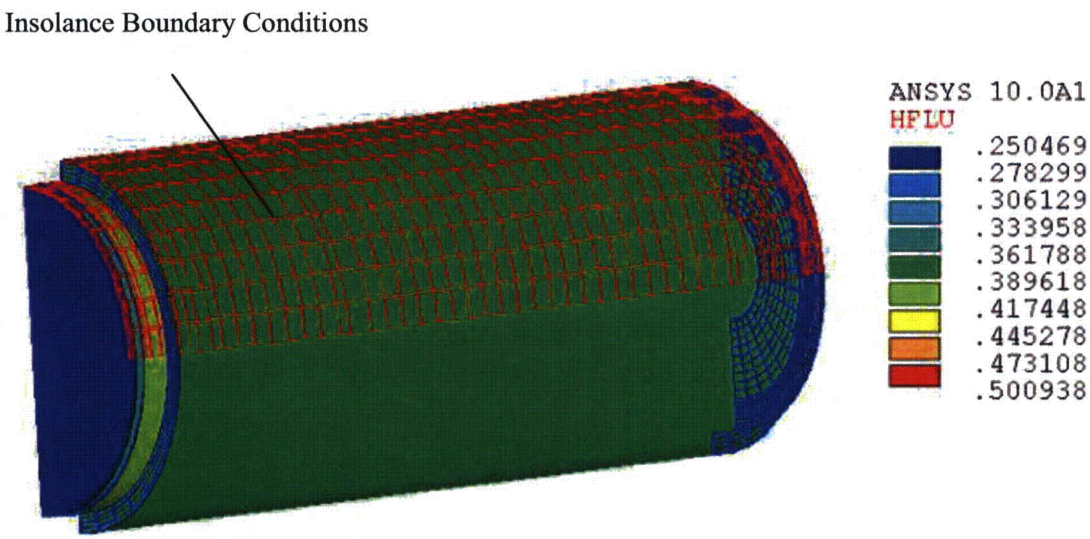


Figure 5-6 Typical Decay Heat and Insolance Boundary Conditions

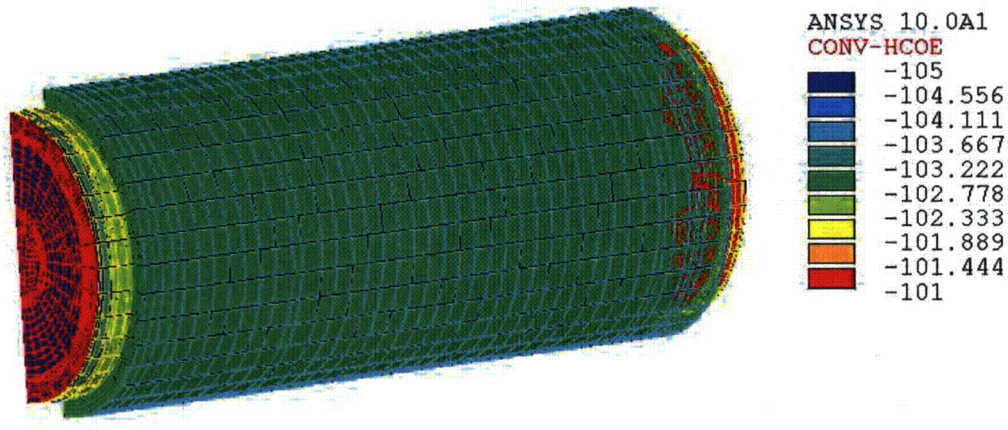


Figure 5-7 Typical Convection and Radiation Boundary Conditions

6.0 RESULTS AND DISCUSSION

Table 6-1 and Table 6-2 present a summary of the maximum/average temperatures for the various components of the OS200FC TC obtained using the methodology described in Section 5.0 using ANSYS and presents a comparison to the temperatures obtained using SINDA/FLUINT in [7] for the design cases noted in Table 4-1.

Table 6-1 Steady State Operations with FC, (40.8 kW), [°F]

	Temperature (°F)		ΔT (°F)
Component	Normal Hot ANSYS	Normal Hot SINDA/FLUINT [7]	$T_{ANSYS} - T_{SINDA}$
Max. DSC Shell	435	431	+4
Inner Liner	347	339	+8
Gamma Shield	339	333	+6
Structural Shell	288	283	+5
Neutron Shield, Max. / Avg.	283 / 216	278 / 210	+5 / +6
Neutron Shield Outer Skin	271	267	+4
Forced Air, Inlet / Exit	106 / 273	106 / 275	0 / -2
Bulk Average NS-3	207(1)	206	+1
Closure Lid	263	272	-9
Top Forging	292	299	-7
Bottom Forging	178	169	+9

Notes:

(1) The Bulk Average NS-3 is the maximum of the bulk average temperatures obtained for NS-3 at top and bottom.

Table 6-2 Steady State Operations with FC, (31.2 kW), [°F]

Component	Temperature (°F) Normal Hot ANSYS	Temperature (°F) Normal Hot SINDA/FLUINT [7]	ΔT (°F) $T_{ANSYS} - T_{SINDA}$
Max. DSC Shell	373	370	+3
Inner Liner	299	293	+6
Gamma Shield	294	289	+5
Structural Shell	251	247	+4
Neutron Shield, Max. / Avg.	247 / 195	243 / 192	+4/+3
Neutron Shield Outer Skin	237	235	+2
Forced Air, Inlet / Exit	106 / 239	106 / 243	0 / -4
Bulk Average NS-3	191(1)	188	+3
Closure Lid	234	236	-2
Top Forging	258	263	-5
Bottom Forging	166	156	+10

Notes:

(1) The Bulk Average NS-3 is the maximum of the bulk average temperatures obtained for NS-3 at top and bottom.

As seen from the data presented in Table 6-1 and Table 6-2, the maximum temperature difference between the DSC shell temperatures obtained from the ANSYS and SINDA/FLUINT calculation in [7] is +3°F for the 32PTH1 DSC with 31.2 kW and +4°F for the 32PTH1 DSC with 40.8 kW. Further, the maximum difference in the average exit air temperature is -4°F for both 40.8 kW load case and -3°F for 31.2 kW load case.

In addition, the maximum difference between the exit air temperatures assumed in Section 5.1 (273°F and 239°F for the 40.8 and 31.2 kW load cases respectively) and exit air temperatures obtained from the thermal model as presented in Table 6-1 and Table 6-2 (273.4°F and 239.3°F for the 40.8 and 31.2 kW load cases respectively) is -0.4°F. Therefore, no further iterations are required.

The maximum heat removed by forced convection from the OS200FC TC is summarized in Table 6-3 for methodologies using ANSYS and SINDA/FLUINT. The maximum difference is -3% less heat removed by forced cooling for the 31.2 kW 32PTH1 DSC using ANSYS methodology.

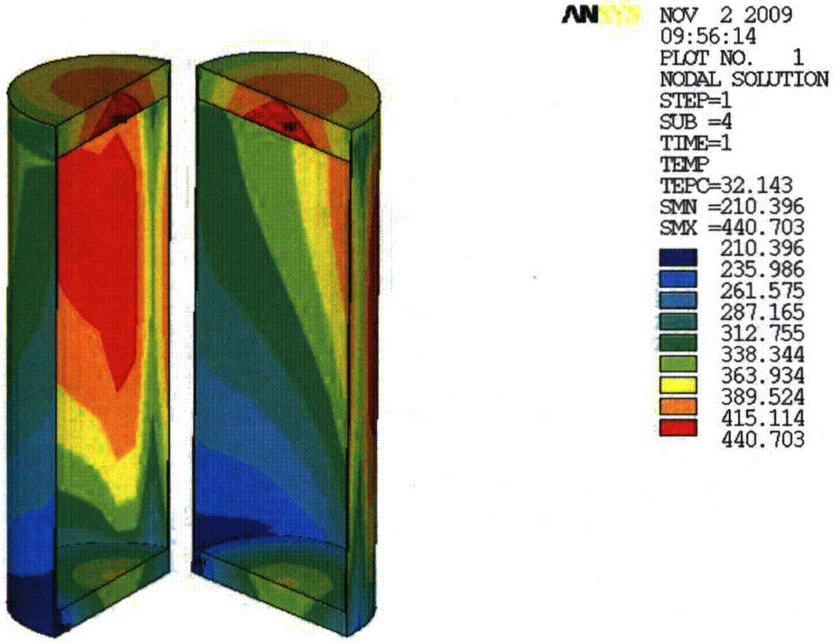
As seen from the results presented in Table 6-1 and Table 6-2, the maximum temperatures for the closure lid and the top forging are lower by -9°F and -7°F, respectively for the 40.8 kW 32PTH1 DSC and -2°F and -5°F, respectively for the 31.2 kW 32PTH1 DSC. The SINDA/FLUINT model used in [7] models the convection between the DSC end plates, interior surfaces of the top forging and the interior surfaces of the top closure lid. In the current ANSYS analyses, the convection in the region between the DSC top end plate, interior surfaces of the

top forging and the interior surfaces of the top closure lid is not modeled and heat transfer through conduction is assumed. This results in a lower heat being removed from the DSC outer surface due to forced convection and is conservative for the calculation of the fuel cladding and DSC top cover plate temperatures. However this also results in reduced heat transfer to the top forging and closure lid resulting in lower temperatures, but since the temperatures of these components are significantly below the code limits of 800°F [7], there is no effect on the thermal performance of the OS200 TC.

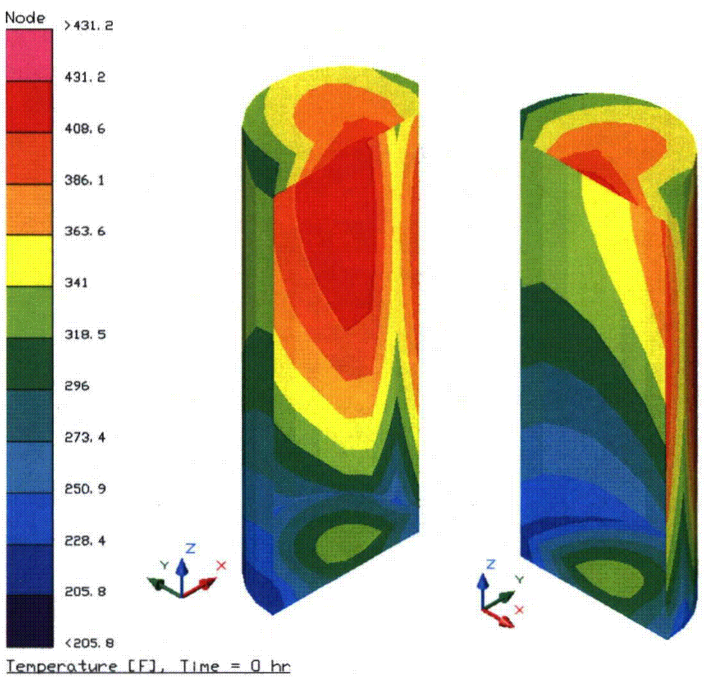
Table 6-3 Heat Removed by Forced Cooling

		T_{exit}	$T_{initial}$	T_{avg}	T_{avg}	C_p	Mass Flow	m.cp.ΔT	% Difference in Q_{FC}
Heat Load	**Methodology**	**(°F)**	**(°F)**	**(°F)**	**(K)**	**(J/kg-K)**	**(kg)**	**(kW)**	
31.2	ANSYS	239	106	173	351	1011	0.213	15.91	-3
	SINDA/FLUINT [7]	243	106	175	352	1011	0.213	16.39	
40.8	ANSYS	273	106	190	361	1012	0.208	19.53	-1
	SINDA/FLUINT [7]	275	106	191	361	1012	0.208	19.76	

The temperature profiles for the OS200FC and the 32PTH1 DSC with forced convection using ANSYS and SINDA/FLUINT [7] are shown in Figure 6-1 and Figure 6-2 for the 40.8 kW heat load and Figure 6-3 and Figure 6-4 for 31.2 kW heat load, respectively. The temperature profiles of the 32PTH1 DSC Shell and the OS200FC TC noted above from ANSYS and SINDA/FLUINT are similar. As seen from figures, the maximum temperature on the DSC Shell occurs towards the top of the DSC in both ANSYS and SINDA/FLUINT due to the forced air flow, which enters the OS200FC TC at the bottom and exits at the cask top lid.

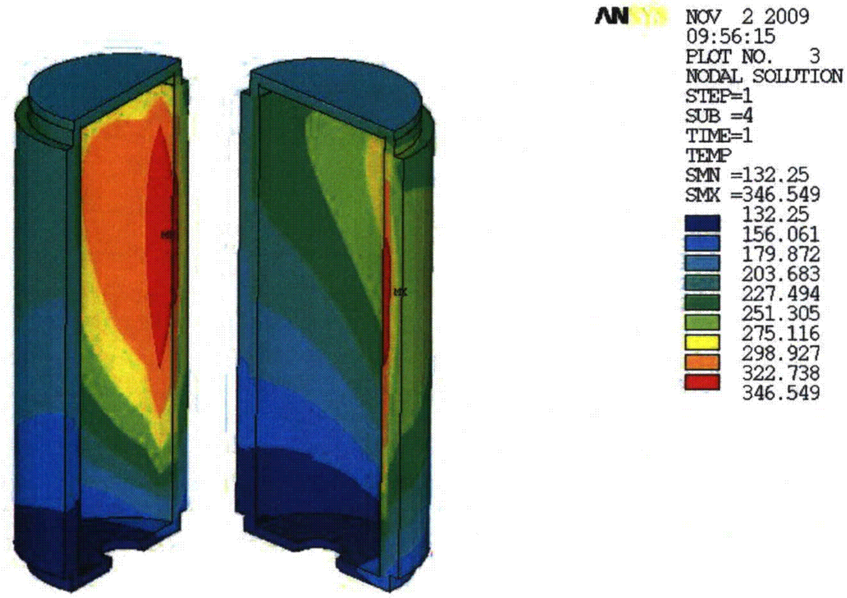


ANSYS

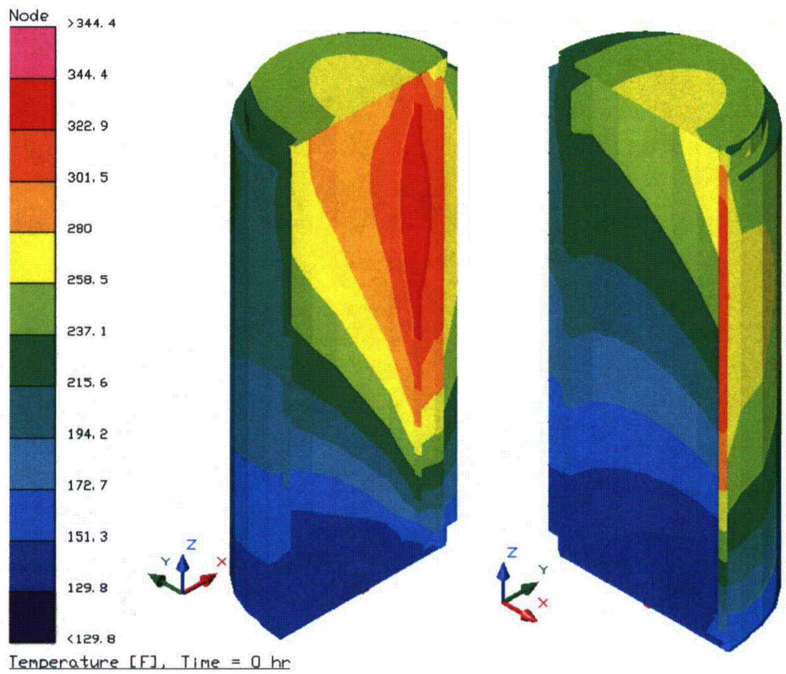


SINDA/FLUINT [7]

Figure 6-1
Temperature Distributions 32PTH1 DSC Shell @ 40.8 kW

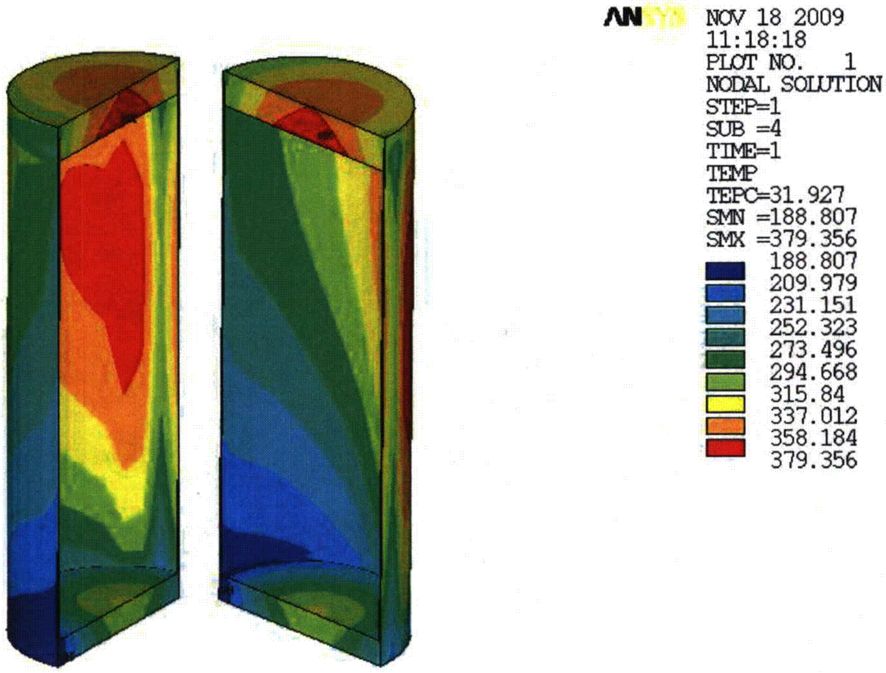


ANSYS

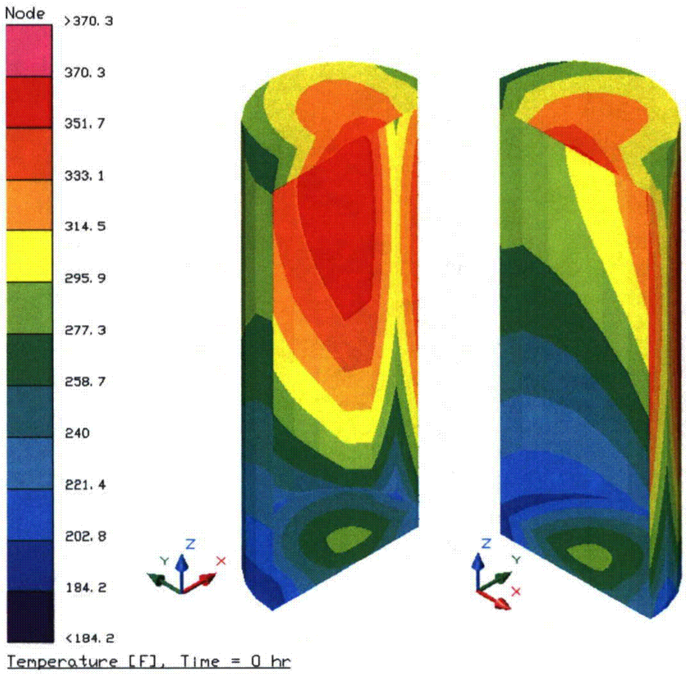


SINDA/FLUINT [7]

Figure 6-2
Temperature Distributions OS200 TC @ 40.8 kW

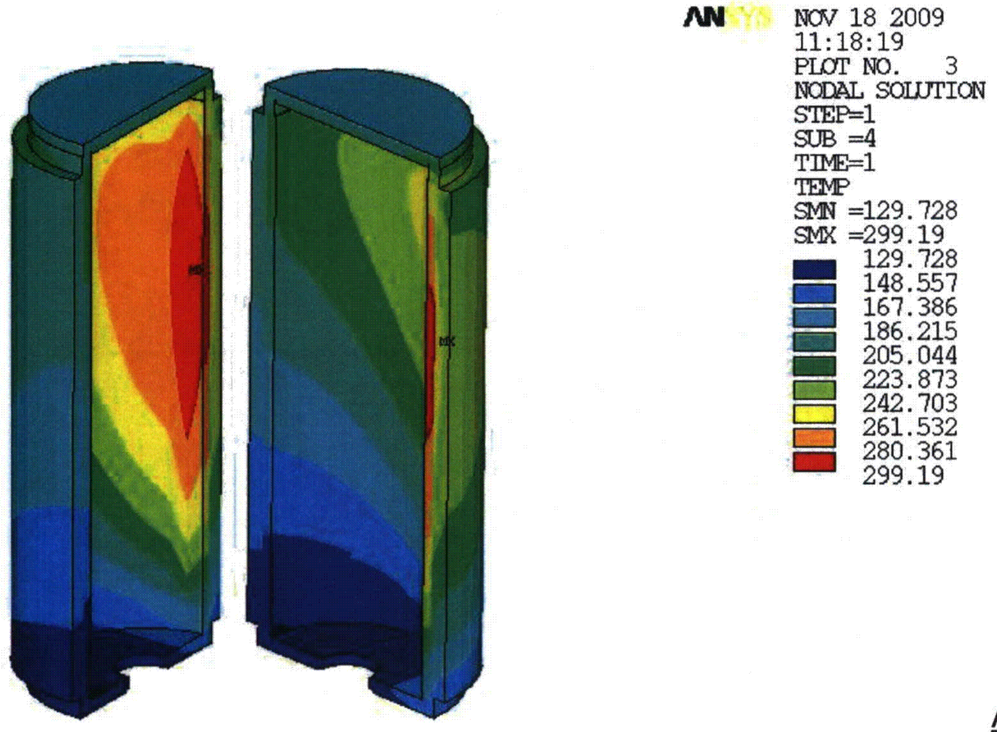


ANSYS

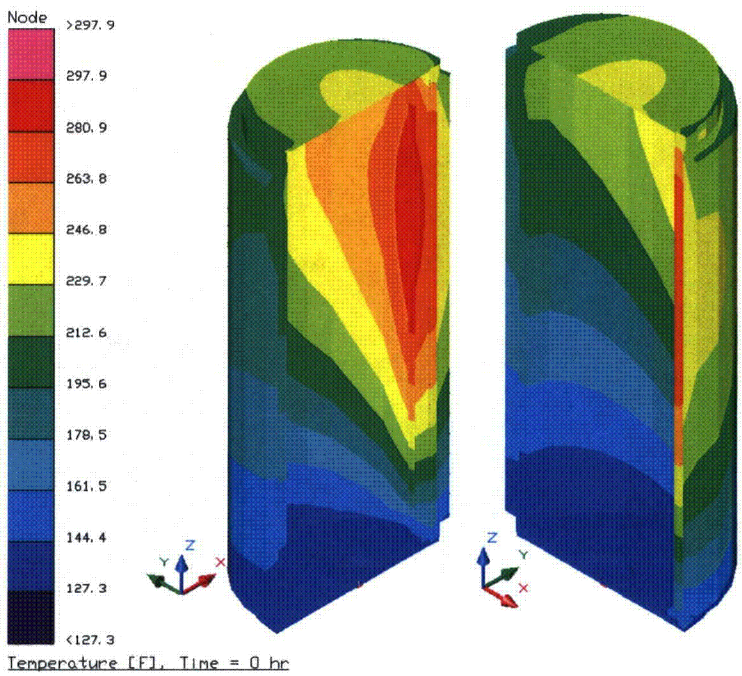


SINDA/FLUINT [7]

Figure 6-3
Temperature Distributions 32PTH1 DSC Shell @ 31.2 kW



ANSYS



SINDA/FLUINT [7]

Figure 6-4
Temperature Distributions OS200 TC @ 31.2 kW

7.0 CONCLUSION

Based on the results presented in Section 6.0, thermal analyses of OS200FC TC with forced air flow and 32PTH1 DSC for 31.2 and 40.8 kW using ANSYS satisfies all the criteria described in Section 4.5. Table 7-1 summarizes the maximum differences between the ANSYS and SINDA/FLUINT for the thermal analyses of OS200FC with 32PTH1 DSC for 31.2 and 40.8 kW heat loads.

Table 7-1 Maximum Differences between OS200FC TC Thermal Analysis using ANSYS and SINDA/FLUINT

	Max ΔT (°F)
Component	**T_{ANSYS} - T_{SINDA}**
Max. DSC Shell	+4
Forced Air, Exit	-4
	Difference in Q_{FC} (1- $Q_{FC, SINDA}$ / $Q_{FC, Ansys}$)
Max Difference in Heat Removed by Forced Cooling	-3%

As seen from Table 7-1, the maximum difference between the DSC shell temperatures is within ±5°F, between the air exit Temperature is within ±10°F, and between the heat removed by forced convection cooling is within ±5%. Hence all the criteria set in Section 4.5 for the thermal analyses of OS200FC/32PTH1 DSC with forced air flow are satisfied. In conclusion, the ANSYS model is validated to use for the thermal evaluation of CCNPP-FC TC equipped with forced air convection cooling.

8.0 LISTING OF COMPUTER FILES

All the runs are performed using ANSYS version 10.0 [6] with operating system "Linux RedHat ES 5.1", and CPU "Opteron 275 DC 2.2 GHz" / "Xeon 5160 DC 3.0 GHz".

A list of the files to create the finite element model of OS200FC with 32PTH1 DSC is shown in Table 8-1.

Table 8-1 List of Geometry Files

File Name (Input and Output)	Description	Date/Time
OS200_32PTH1	Macro to create geometry of OS200FC with 32PTH1 DSC	10/08/2009 05:01 PM

A summary of ANSYS runs is shown in Table 8-2.

Table 8-2 Summary of ANSYS Runs

Run Name	Description	Date / Time for Output File
31.2 kW Load Cases		
FlowRate_32PTH1_31kW	Flow Rate Model to determine the mass flow rates for 31.2 kW Heat Load.	10/12/2009 01:07 PM
TR_32PTH1_31	OS200FC with 32PTH1 DSC and Forced Convection- 31.2 kW	11/18/2009 11:18 AM
FlowRate_32PTH1_41kW	Flow Rate Model to determine the mass flow rates for 40.8 kW Heat Load.	10/20/2009 01:47 PM
TR_32PTH1_41	OS200FC with 32PTH1 DSC and Forced Convection- 40.8 kW	11/02/2009 09:56 AM

ANSYS macros, and associated files used in this calculation are shown in Table 8-3.

Table 8-3 Associated Files and Macros

File Name	Description	Date / Time
HCL_OS200.MAC	Total heat transfer coefficients for horizontal cylindrical surface	06/16/08 09:26 AM
VPL_OS200.MAC	Total heat transfer coefficients for vertical flat surface	06/16/08 09:32 AM
Mat_OS200+32PTH1.inp	Material properties for OS200FC TC and 32PTH1 DSC	09/30/09 10:58 AM
MassFlow_ConvCoeff_32PTH1_31kW.xls	Spreadsheet for calculating the hydraulic diameters, friction factors, mass flow rates and heat transfer coefficients	11/18/2009 11:41 AM
MassFlow_ConvCoeff_32PTH1_41kW.xls		11/18/2009 11:42 AM

APPENDIX A TOTAL HEAT TRANSFER COEFFICIENTS

Total heat transfer coefficient, h_t, is used to combine the convection and radiation heat transfer together.

$$h_t = h_r + h_c$$

Where,

h_r = radiation heat transfer coefficient (Btu/hr-in^2-°F)
h_c = free convection heat transfer coefficient (Btu/hr-in^2-°F)

The radiation heat transfer coefficient, h_r, is given by the equation:

$$h_r = \varepsilon F_{12}\left[\frac{\sigma(T_w{}^4 - T_{amb}{}^4)}{T_w - T_{amb}}\right] \quad \text{Btu/hr-in}^2\text{-°F}$$

Where,

ε = surface emissivity
F_{12} = view factor from surface 1 to ambient = 1
σ = 0.1714×10^{-8} Btu/hr-ft^2-R^4
T_w = surface temperature (°R)
T_{amb} = ambient temperature (°R)

Surface emissivity values are discussed in Section 4.4.

The following equations from Rohsenow handbook [7], Section 4.4 are used to calculate the free convection coefficients.

Horizontal Cylinders:

$$Ra = Gr\,\mathrm{Pr} \quad ; \quad Gr = \frac{g\,\beta\,(T_w - T_\infty)\,D^3}{\nu^2}$$

$$\mathrm{Nu} = \left\{0.60 + \frac{0.387\mathrm{Ra}_D{}^{1/6}}{\left[1 + (0.559/\mathrm{Pr})^{9/16}\right]^{8/27}}\right\}^2$$

$$h_c = \frac{Nu\,k}{D}$$

with
g = gravitational constant =9.81 (m/s^2)
β = expansion coefficient = 1/T (1/K)
T = absolute temperature (K)
ν = kinematic viscosity (m^2/s)

D = diameter of the horizontal cylinder (in)
k = air conductivity (W/m-K)

The above correlations are incorporated in ANSYS model via macro " HCL_OS200.MAC" listed in Section 8.0.

Vertical Flat Surfaces:

$$Ra = Gr\,\mathrm{Pr} \quad ; \quad Gr = \frac{g\,\beta (T_w - T_\infty) L^3}{\nu^2}$$

$$\mathrm{Nu}_L = \frac{2.8}{\ln\left(1 + 2.8/\mathrm{Nu}^T\right)} \text{ with}$$

$$Nu^T = \overline{C}_l\, Ra^{1/4} \qquad \text{with } \overline{C}_l = \frac{4}{3}\left[\frac{0.503}{\left(1 + (0.492/\mathrm{Pr})^{9/16}\right)^{4/9}}\right]$$

$$\mathrm{Nu}_t = C_t^V \mathrm{Ra}^{1/3} \text{ with}$$

$$C_t^V = \frac{0.13\,\mathrm{Pr}^{0.22}}{(1 + 0.61\,\mathrm{Pr}^{0.81})^{0.42}}$$

$$Nu = \left[(Nu_l)^m + (Nu_t)^m\right]^{1/m} \qquad \text{with} \quad m = 6 \quad \text{for } 1 < Ra < 10^{12}$$

$$h_c = \frac{Nu\,k}{L}$$

with
g = gravitational constant =9.81 (m/s^2)
β = expansion coefficient = 1/T (1/K)
T = absolute temperature (K)
ν = kinematic viscosity (m^2/s)
L = height of the vertical surface (in)
k = air conductivity (W/m-K)

The above correlations are incorporated in ANSYS model via macro " VPL_OS200.MAC" listed in Section 8.0.