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Facility Renovation Title I 60% Design

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THERMAL AND FLOW ANALYSES OF THE NUCLEAR MATERIALS STORAGE FACILITY RENOVATION TITLE I 60% DESIGN

by

Thad D. Knight, Robert G. Steinke, and Cathrin Müller

ABSTRACT

We are continuing to use the computational fluid dynamics code CFX-4.2 to evaluate the steady-state thermal-hydraulic conditions in the Nuclear Material Storage Facility Renovation Title I 60% Design. The analyses build on those performed for the 30% design. We have run an additional 9 cases to investigate both the performance of the passive vault and of an individual drywell. These cases investigated the effect of wind on the inlet tower, the importance of resolving boundary layers in the analyses, and modifications to the porous-medium approach used in the earlier analyses to represent better the temperature fields resulting from the detailed modeling of the boundary layers. The difference between maximum temperatures of the bulk air inside the vault for the two approaches is small. We continued the analyses of the wind effects around the inflector fixture, a canopy and cruciform device, on the inlet tower by running a case with the wind blowing diagonally across the inflector. The earlier analyses had investigated a wind that was blowing parallel to one set of vanes on the inflector. Several subcases for these analyses investigated coupling the analysis to the facility analysis and design changes for the inflector.

1. INTRODUCTION

Fluor Daniel, Inc., (FDI) completed the 30% phase of Title I design for the Los Alamos National Laboratory (LANL) Nuclear Materials Storage Facility (NMSF) renovation project with the release of the 30% design report.¹ This report provides the current set of drawings that describe the planned renovation and also includes discussion of the thermal analyses performed by FDI and LANL Groups TSA-10 and ESA-DE. Group TSA-10 is responsible for the LANL analyses of the passive vault, and Ref. 2 provides a summary of those analyses at the 30% design phase.

The analyses described in Ref. 2 used a porous-medium assumption to describe the drywell array for the calculations of the passive vault in order to generate quickly

information necessary for assessing the passive vault design. Reference 2 also describes the initial analyses for wind flowing past the inflector, a canopy and cruciform device, on one inlet tower to the vault. Then, the document provided an estimate of the maximum temperature of plutonium metal stored in the passive vault based on the bay analyses and earlier drywell and container analyses and experiments. That estimate was 221.8°F (105.4°C or 378.6 K) for a total heat load of 10 kW for one bay and 224.6°F (107.0°C or 380.2 K) for a total heat load of 20 kW in a single bay. These temperatures are significantly less than the applicable limit of 284°F (140°C or 413.2 K). Because of the modeling assumptions invoked in the drywell and container modeling, we believe that these estimated maximum temperatures are conservative.

During the 30–60% design phase, we in TSA-10 investigated the effects of the porous-medium assumption for describing the drywell array in the facility calculations. We developed a new drywell model to represent better the container holding fixtures inside the drywell and to track more accurately the boundary layers inside the drywell. We also extended the analyses of wind effects on the performance of the inlet-tower inflector. We then explored the coupling of the facility analyses to the wind-effects analyses by attempting to converge both analyses to the same flow. The following sections of this report document the status of these TSA-10 analyses. We used the computational fluid dynamics code CFX-4.2 (Ref. 3) to perform all of these new calculations.

2. PASSIVE-VAULT ANALYSES

During the 30–60% design phase, we ran an additional 8 cases to analyze the flow and temperature fields in one of the two bays in the passive vault. Of these 8 cases, 7 explored the effects of the porous-medium assumption that had been utilized in the earlier analyses of the passive vault. The concern about the porous-medium assumption arises from the fact that the boundary layers on drywell outside walls are not resolved. These boundary layers should rise vertically and should represent the hottest air in the storage bay. A comparison of the Grashof and Reynolds numbers suggests that these boundary layers may exert a dominant effect on the overall flow patterns and temperature distributions. The effect of the boundary layers is to impart a larger vertical velocity component than the porous medium produces. Also, relative to the porous medium calculations, the boundary layers make the distribution of heat addition to the air more heterogeneous and produce increased local variations in the temperature distributions in the air. These local variations in the air temperature lead directly to increased buoyancy forces, which drive natural convection inside the bay.

Table I summarizes the various analysis cases for the passive vault. The table includes cases 15–20 and 22. Cases 14 and 21 do not address the issue of porous medium and resolution of boundary layers in the storage bays. Subsequent sections will discuss cases 14 and 21. Cases 1–13 are described in Ref. 2.

TABLE I
ANALYSIS CASES TO EXPLORE BOUNDARY-LAYER EFFECTS
IN THE PASSIVE-VAULT CALCULATIONS

Case Number	Total Heat Source (W)	Description
15	150	A test model with two half drywells, 23,532 cells, and an air flow of $684.9 \text{ ft}^3 \cdot \text{min}^{-1}$ [1/8 of the $5479 \text{ ft}^3 \cdot \text{min}^{-1}$ flow calculated in facility case 4 (Ref. 2)]. The boundary conditions, also based on facility case 4, are specified as $V_x(z)$ and $V_z(z)$ at the inlet side and $P(z)$ at the outlet.
16	150	Case 15 with the $P(z)$ boundary condition replaced by a constant relative pressure based on the calculated outlet air temperature: $P(z) = 0.0 + 9.80665 \cdot \sum_k^K [\rho_{ref} - \rho_{out,k}(T_k)] \cdot \Delta z_k$.
17	150	Case 16 with an air flow of $287.5 \text{ ft}^3 \cdot \text{min}^{-1}$ [1/8 of the $2300 \text{ ft}^3 \cdot \text{min}^{-1}$ flow from facility case 12 (Ref. 2)]. This reduced flow was achieved by renormalizing the $V_x(z)$ and $V_z(z)$ distributions from case 16.
18	150	Case 16 with zero air flow. For this case, the $V_x(z)$ and $V_z(z)$ at the inlet boundary condition were set to zero.
19	150	Case 16 with an approximate doubling of the number of cells in each of the three directions for a total of 164,576 cells.
20	1250.6	A test model with 47 half drywells with 553,002 cells and an air flow of $684.9 \text{ ft}^3 \cdot \text{min}^{-1}$ [1/8 of the $5479 \text{ ft}^3 \cdot \text{min}^{-1}$ flow from facility case 4 (Ref. 2)]. This model applies the same nodalization to each drywell as was used in cases 15–18. As before, the inlet boundary condition is defined by $V_x(z)$ and $V_z(z)$ from case 4, and the exit boundary condition is $P(z)$, again from case 4. This model represents the entire length of the drywell array while taking advantage of symmetry planes between columns of drywells and through the centers of drywells in a given column.
22	10,005	Facility case 4 (Ref. 2) with its uniform heat-source distribution at a given elevation replaced by a heterogeneous distribution in the y-direction across the vault. With three cells per drywell in the y-direction, the volumetric heat source is increased by a factor of 3 locally and is defined in y-direction cell 1 for the first drywell, cell 6 for the second drywell, cell 7 for the third drywell, and cell 12 for the fourth drywell. This model still invokes the porous-medium assumption.

Cases 15–19 were relatively quick running problems that allowed us to investigate the application of boundary conditions, the effects of imposed flow, and nodalization. Case 20 extended the modeling methodology to a complete column of drywells down the length of the storage bay. Case 22 begins to explore how to alter the porous medium model to obtain results similar to the detail model that tracks the boundary layers on the drywells.

2.1. Thermal-Hydraulic Models for the Drywell-Array Analyses

We created all of the models for these analyses with the CFX-4.2 MeshBuild preprocessor. Each model consists of multiple blocks with a three-dimensional Cartesian geometry and with a body-fitted grid. We used the CFX-4.2 Solver to generate steady-state calculations of the flow field and inside conducting solids (such as the drywell walls). The solver calculated convection and conduction heat transfer and used the k - ϵ turbulence model. The air was treated as weakly compressible, which results in the air density being a function only of temperature as defined by the ideal-gas equation of state. The pressure for determining the air density was the Los Alamos reference pressure⁴ of 76,976 Pa (11.16 psia). The air temperature at the inlet was 308.15 K (95°F), which corresponds to the historical maximum temperature⁵ recorded at LANL Technical Area 59.

Iterative convergence of the steady-state solution was determined by the following tests:

1. air mass flow in equals air mass flow out;
2. air enthalpy in plus the drywell-array heat source equals air enthalpy out;
3. the absolute value of the air-mass-flow residual ($|\Delta \dot{m}_k|$) summed over all mesh cells and divided by the air mass flow ($\dot{m}$) through the facility is $\sum |\Delta \dot{m}_k| / \dot{m} < 0.01$; and
4. the absolute value of the air-enthalpy-flow residual $|\Delta(\dot{m}_k \cdot e_k)|$ summed over all mesh cells and divided by the drywell-array heat source is $\sum |\Delta(\dot{m}_k \cdot e_k)| / (heat\ source) < 0.01$.

AEA Technology, the developer of CFX-4.2, claims that there is sufficient steady-state iterative convergence when the values of the latter two criteria are <1% (0.01).

For the models which were intended to resolve the boundary layers on the outside surfaces of the drywells (cases 15–20), we included no additional body force terms to account for the wall and form drag associated with flow around the drywells. In case

22, which invokes the porous-medium assumption, we used the correlation in Idelchik⁶ with the implementation described in Ref. 2.

During the initial calculations for case 15, we discovered that it was necessary to model the drywell wall as a conducting solid to provide good conduction heat transfer to mitigate hot spots that were forming in the turbulent eddies on the downstream sides of the drywells. Cases 15–20 include this feature and apply the heat source axial distribution as a volumetric source inside the drywell walls.

2.2. Results of the Drywell-Array Analyses

Reference 7 provides graphical results for cases 15–20 and 22. The use of symmetry planes to define the discrete models for cases 15–20 results in modeling only 1/8 of the total width of the bay and only 1/2 of each drywell represented in the model. Table I defines the cases and provides the total power simulated by the models. In all the models, the assumed drywell loading was 10 containers, each generating 15 W, on 16-in. centers starting from the bottom of the drywell. For cases 15–19, which model half drywells because of symmetry considerations, each half drywell generates a total of 75 W. For case 20, which models half drywells down the full length of the drywell array, only the first 16 drywells are fully loaded and generate 75 W, and the 17th drywell has a partial (67.5%) load to account for the total power corresponding to 1/8 of the design load for the bay.

Cases 15–19 were quick running versions of the discrete drywell model with two half drywells. We used these cases to gain insight into the CFX-4.2 calculation of a model that captured relatively complex interactions between boundary layer flow and bulk air flow through the facility. Case 20, which modeled the full length of the drywell array; included more than twenty times the number of cells in cases 15–18 and required a much longer run time to converge the solution. To minimize the number of computational cells in all of these cases, we took advantage of symmetry planes that run the length of the vault between the columns of drywells and that run through the axes of all of the drywells in a given column.

We defined the models for cases 15–20 with body-fitted grids in Cartesian coordinates, which allows the computational domain to adapt to the curve of the drywell walls in a horizontal plane. While the coordinates are defined as Cartesian, it is convenient to refer to the discretization of the grid as it relates to a given drywell in terms of radial (r), azimuthal (θ), and axial (z) dimensions.

To specify the nodalization for case 15, we used information from earlier sensitivity studies that investigated the size of cells required to resolve boundary layers. For each half drywell in the model, there were 37 axial cells; 15 radial cells, including 1 cell for the drywell wall thickness; and 20 azimuthal cells to span the half circumference of the drywell. The axial dimension of the cells was uniform through out the model, and the azimuthal dimension of the cells in the drywell wall was uniform. In the air space, the Δr associated with the radial discretization

near the drywell wall increased with increasing radius through the geometric-progression option available with CFX-4.2 MeshBuild preprocessor. The Δr for the first cell just outside the drywell (radial cell 2) was 0.1 in., which is about half of the boundary layer thickness. We defined two additional blocks (3x3 cells in the horizontal plane) for each drywell unit model to improve the orthogonality of the grid in the far corners away from the drywell. The resulting model had 11,766 cells for each half drywell, or 23,532 cells total.

The upstream boundary condition for case 15 specified the longitudinal velocity component V_x (down the bay) and the vertical velocity component V_z ; the V_y component is set to zero. Both V_x and V_z were specified as a function of vertical height z . The specification of V_x and V_z involved normalizing the velocity distribution at the inlet to the drywell array from facility case 4 (Ref. 2) to achieve the desired flow of $684.9 \text{ ft}^3 \cdot \text{min}^{-1}$, which is $1/8$ of the bay flow of $5479 \text{ ft}^3 \cdot \text{min}^{-1}$ from facility case 4. At the outlet to the case-15 model, we specified a pressure distribution, again based on the results from facility case 4. In developing these boundary conditions, we averaged the original distributions from facility case 4 in the y -direction (across the width of the bay) to smooth out variations in that direction.

The calculation of case 15 revealed nonphysical velocities downstream of each of the two drywells. In the region between the two drywells and downstream of the second drywell, the air velocities were downward in the calculation when we anticipated the flow would be upwards. We traced the cause of this behavior to the pressure boundary condition, which did not inherently match the temperature of air flowing out of the model and thus set up erroneous natural-circulation flows. To avoid creating nonphysical, natural-circulation flows at the outlet, the pressure boundary condition must reflect the correct gravity head associated with air flowing out.

Case 16 replaces the pressure boundary condition from case 15 with a calculated pressure distribution:

$$P(z) = P_{ref} + g \cdot \sum_k^K [\rho_{ref} - \rho_{out,k}(T_k)] \cdot \Delta z_k \quad (1)$$

where P_{ref} is the reference pressure specified in CFX-4.2 input, ρ_{ref} is the reference density corresponding to the reference temperature specified in the input, and g is the acceleration due to gravity [units are standard SI]. Substituting for P_{ref} and g for our case yields

$$P(z) = 0.0 + 9.80665 \cdot \sum_k^K [\rho_{ref} - \rho_{out,k}(T_k)] \cdot \Delta z_k \quad (2)$$

The pressure distribution defined by Eq. (2) needs to be applied iteratively in the calculation to maintain consistency between the calculated temperature of the air flowing into the pressure boundary and pressure defined by that boundary. We accomplished this application by defining the pressure boundary condition through the available user FORTRAN routines and updating the definition at the beginning

of 6 restarts to adjust the air density as required. The final calculation created the correct upward flow of air next to the drywells that results only from buoyancy forces generated by boundary-layer heating on the drywell surface.

Cases 17 and 18 repeat case 16 with the air flow at the inlet reduced to $287.5 \text{ ft}^3 \cdot \text{min}^{-1}$ and $0.0 \text{ ft}^3 \cdot \text{min}^{-1}$, respectively. The $287.5 \text{ ft}^3 \cdot \text{min}^{-1}$ flow corresponds to the $2300 \text{ ft}^3 \cdot \text{min}^{-1}$ flow in facility case 12 (Ref. 2), which modeled a total bay loading of only 600 W. These two cases together with the results from case 16 show how the air-flow pattern changes with reduced facility flow past the drywells. For the highest facility flow, case 16, the facility flow forced the velocity vectors nearly horizontal in the open channel between the two columns of drywells. For the low facility flow, case 17, the flow field in the open channel became disrupted and circulation patterns developed. For zero facility flow, case 18, the flow field became obviously buoyancy driven. It is apparent from temperature plots that at higher flows, more heat is swept away from the boundary layers. However, another important observation from case 16 for the highest facility flow was that there was significant buoyancy driven flow and recirculation effects between the two half drywells and downstream from the second half drywell in the regions shielded by the half drywells from the forced facility flow. These effects transported warm air toward the top of the bay and above the region of heat generation from the drywell.

Case 19 refined the computational mesh used in case 16 by generally halving the cell sizes in all three directions outside the boundary layers. We maintained the minimum Δr (0.1 in.) next to the drywell wall because we had previously shown that this minimum cell size was sufficient to resolve boundary-layer phenomena in air. The resulting model for each half drywell consisted of 74 axial cells; 26 radial cells, again including only 1 cell for the drywell wall thickness; and 40 azimuthal cells. Each of the two additional blocks that improve the orthogonality of the grid for each drywell unit model had 6×6 cells in the horizontal plane. The final model had a total of 164,576 cells.

We evaluated case 19 solely to verify that the coarser mesh of case 16 was sufficient to calculate accurately the thermal-hydraulic solution. A comparison of the results revealed only acceptably small differences between the two cases.

Case 20 represented an initial effort to model the entire length of the drywell array in sufficient detail to resolve the boundary layers on the individual drywells. We built this model using the same discretization scheme outlined above for case 15. As before, we took advantage of the available symmetry planes in the facility to limit the model to representing a single column of half drywells extending the length of the bay. This model represented 47 half drywells and included a total of 553,002 cells. The model is set up to represent facility case 4 (Ref. 2); therefore, only the first 17 drywells generate heat.

Figure 1 shows the temperature contours on several vertical slices down the length of the model. The first slice at the bottom of the figure is 3.81 cm (1.5 in.) away from the symmetry plane through the axes of the drywell. Subsequent slices are 11.43 cm

(4.5 in.), 19.05 cm (7.5 in.), 26.67 cm (10.5 in.), and 34.29 cm (13.5 in.) away from the symmetry plane through the axes of the drywells. In the lower three slices shown in the figure, the vertical white bars represent the intersections of the interiors of the drywells with the slice. The colored vertical bars show the drywell wall temperature at the intersection with the slice (almost unnoticeable in the scale of this figure) and the air temperature between the drywells in that slice. The drywell inside radius is 21.59 cm (8.5 in.), which results in the drywell extending only slightly beyond the slice at 19.05 cm (7.5 in.). The last two slices are through the free space between columns of drywells. In the context of this figure, the symmetry plane between two columns of drywells lies at 0.3810 m (15.0 in.), which is the full extent of the model in the y direction. In this figure, the facility air flow was from left to right.

The maximum temperature in this figure is 317.67 K (44.52°C or 112.14°F). However, this maximum temperature occurred only in limited regions of the drywell wall, and the hottest air in the boundary layers was less than this value. The maximum temperature in the bay air outside the boundary layers, which approximates the bulk-air temperature for heat transfer purposes, was closer to 314.55 K (41.4°C or 106.5°F). This bay air temperature is comparable to the maximum air temperature from facility case 4 (see Fig. 3 of Ref. 2), which is 313.35 K (40.2°C or 104.4°F). Therefore, the maximum error in the air temperature associated with the porous-medium assumption in facility case 4 is 1.2°C (2.2°F) too low. The impact of this error on the projected plutonium-metal temperatures given in Ref. 2 is small in view of the available margin to the temperature limit of 140°C (284°F).

A more troublesome observation from the comparison of Fig. 1 here to the corresponding figure in Ref. 2 is that the porous-medium calculation of facility case 4 exhibited essentially no thermal stratification. As air flowed down the vault in the porous-medium calculation, the hottest air generally stayed at the vault elevation where the volumetric heat source was greatest. In contrast, the discrete representation of the drywells in case 20 resulted in a temperature distribution which indicates vertical air movement in the region of the first 17 drywells. This vertical movement of the warmer air resulted from the boundary layers rising up the walls of the heated drywells and led to the expected appearance of thermal stratification in the bay. Downstream from the 17th drywell, the stratification appeared to remain relatively unchanged because of the lack of rising boundary layers on the unheated drywells. The lack of thermal stratification in the porous-medium model appeared to be related to the uniformity across the bay of the flows at any given elevation and at any location down the bay and to the relatively small temperature differences that existed vertically at any location.

Finally, Fig. 1 indicates that there was some backflow from the pressure boundary condition applied at the exit from the drywell array (right side of the figure near the floor of the vault). This back flow was produced by a mismatch between the applied pressure boundary based on facility case 4 and the temperature of the air flowing out of the drywell array. The problem is essentially the same as that discussed above for case 16.

Case 22 is a variation of facility case 4 that investigates using a heterogeneous heat source to represent the heat generation in the drywells while maintaining the porous-medium description of the drywell array and the nodalization from case 4. In case 4, the heated portions of the drywells were distributed uniformly across the bay in 12 y-direction increments to represent the 4 drywells in a single row. There were therefore 3 cells in the y-direction to represent each of the 4 drywells. Case 20 redistributed the homogenous heat source into a heterogeneous distribution in the y-direction so that only one increment in three had heat generation. The initial approach was to concentrate the heat generation in cells 2, 5, 8, and 11 in the y-direction to heat preferentially the air in those cells and thus to generate the temperature differences that could produce buoyancy forces and ultimately enhance thermal stratification. However, the CFX-4.2 code calculates velocities at the cell centers, and at the top of the heated stacks of cells, which are only a single cell wide in the y-direction, the code encountered stability problems when the vertical flow at the top of the stack needed to turn and go in both y-directions. Unfortunately, a single y-component of the velocity at the cell center cannot go both ways simultaneously.

We revised the heterogeneous power distribution in the y-direction to heat cells 1, 6, 7, and 12. While this distribution exaggerated the heterogeneity of the heat source, it eliminated the difficulty determining in which direction to turn the flow at the top of the heated stacks, and the calculation ran. Figure 2 shows temperature contours at various vertical slices down the length of the bay. We adjusted the color scale in this figure to match that in Fig. 1 to facilitate comparisons between the two figures. Because of this adjustment, the maximum temperature of 317.67 K in the legend for Fig. 2 does not exist in the calculation. The maximum air temperature in case 22 was only slightly greater than in the original facility case 4. The improvement in thermal stratification was small. To produce thermal stratification with a porous-medium model that is similar to that from the discrete-nodalization case 20 would require that the heat source in the porous-medium be much more concentrated than what we modeled in case 22. Ultimately, to achieve an appropriate level of thermal stratification in the porous-medium models will require significantly finer discretization in the y-direction than in case 22.

A final comparison between cases 20 and 22 is shown in Fig. 3. This figure shows the pressure drop across the drywell array as a function of elevation for the two cases. The ΔP for the porous-media model in case 22 was approximately 27% higher than in the discrete model for case 20. Because case 20 captures the boundary layers, flow-separations, and recirculation on the downstream sides of the drywells, we assume that the ΔP from the discrete model is the better result. For the porous-medium model of the drywell array, we used a correlation from Idelchik⁶ to define the body-force terms necessary to give the pressure drop over the length of the array. It appears that the correlation overstates the flow resistance by a factor of 1.276. The total flow for case 22 is slightly less than the scaled flow for case 20 ($5447 \text{ ft}^3 \cdot \text{min}^{-1}$ vs. $5479 \text{ ft}^3 \cdot \text{min}^{-1}$). This difference between the flows is in the direction that minimizes the difference in pressure drops through the drywell array.

2.3. Significant Findings from the Drywell-Array Analyses

The following points summarize the more significant findings that were discussed in the previous subsection:

1. When applying a pressure boundary condition that is intended to represent the interior of a larger flow field, either from another calculation or from an experiment, care must be taken to match the assumed fluid density distribution in the pressure boundary to the fluid density of the flow into the pressure boundary. Eq. (1) describes the general form of the pressure distribution.
2. CFX-4.2 calculations showed that as the facility flow increased, the domination of the boundary-layer flows on the walls of the drywells decreased, but even at the highest facility flow, there were still significant boundary-layer flows in certain regions of the model that could produce thermal stratification that is more pronounced than porous-medium models would predict.
3. The discretization of the flow fields for cases 15–18 and 20 was sufficient to capture the interactions between the buoyancy-driven boundary-layer flows on the drywells and the forced facility flow.
4. The maximum air temperatures inside the drywell array that were reported in Ref. 2 may be slightly low, by 1.2°C (2.2°F), based on comparisons between the earlier facility case 4 and the discrete nodalization case 20. The impact of this error on the projected plutonium-metal temperatures given in Ref. 2 is small because of the available margin to the applicable temperature limit.
5. Porous-medium models appear to underpredict the thermal stratification in the drywell array when the flows are uniform and the vertical temperature differences are small. Resolution of this problem may require significantly finer discretization in the coordinate direction across the bay and a heterogeneous power distribution that produces power densities comparable to those between the drywell surface and the associated boundary layer.
6. The Idelchik correlation that we have been using in conjunction with the porous-medium models to specify the hydraulic losses through the drywell array appears to overstate those losses by a factor of 1.276 based on a comparison between the discrete model tracking the boundary layers and a porous-medium model.

3. DRYWELL AND CONTAINER ANALYSIS

The combined drywell and container model is needed to calculate an axial distribution of the heat flux at the outer surface of the drywell. This heat flux distribution can then be used directly as a boundary condition in a discrete model such as that in case 20 or to determine a volumetric heat source for use with porous-medium models. The drywell and container model can produce heat flux distributions for a variety of container loading schemes inside the drywell. This model also calculates the temperature distribution inside the drywell which determines the maximum temperature increase from the drywell to the container and plutonium metal.

3.1. Thermal-Hydraulic Models for the Drywell and Container Analysis

Reference 7 describes the drywell and container model and its calculational results from case 21. The current geometry model depicts 10 heated containers on 16-in. vertical spacing. Each container assembly consisted of 2 sealed, nested containers with a cylindrical plutonium-metal ingot that generated 15 W. Inside the drywell, each container sits on a 16-in. diameter stainless-steel plate. The solution is a steady-state calculation that invokes the $k-\epsilon$ turbulence model and uses the weakly-compressible option to define the gas density. The boundary condition on the outside of the drywell is defined by a heat transfer coefficient based on the Grimsom correlation⁸ and an external temperature profile from the facility case-4 calculation in Ref. 2. The boundary conditions at the bottom and top of the drywell were adiabatic.

The geometry was modeled with two-dimensional, axisymmetric cylindrical coordinates with 80 radial intervals and 1124 axial intervals. This geometry discretization provides for 89,920 cells.

3.2. Results of the Drywell and Container Analysis

Figure 4 shows the calculated heat-flux distribution on the outer surface of the drywell from this calculation. The assumed external temperature distribution is shown across the top of the plot. The adiabatic boundary condition at the bottom of the drywell resulted in the bottom container being the hottest because this container could not reject heat from the bottom of the support plate on which it sits. All other containers reject heat through their support plates to the air space below. If heat-transfer paths from the bottom of the drywell were modeled, the bottom container would be better cooled, and the heat flux from the side of the drywell at the bottom would be lower.

With better cooling of the bottom container, the hottest container would be at location 5 for the external axial temperature profile used in this case. The maximum plutonium-metal temperature in container 5 was 93.0°C (199.5°F) for a cylindrical ingot. If the ingot were replaced by two buttons with minimum contact

to the container, the maximum plutonium-metal temperature would increase by 5.3°C (9.5°F), based on the thermal analysis in Ref. 9.

This CFX-4.2 calculation converged very slowly and did not conserve energy. At this time, we believe that the convergence and conservation problems resulted from the mesh being too fine inside the boundary layers and the conducting solids [$\Delta r_{\min} = 0.001$ m (0.04 in.)]. Previous experience with a similar model having a coarser mesh indicated that run time, convergence, and energy conservation would be acceptable. The path forward is to recast the model with a coarser mesh and to define realistic axial boundary conditions before running additional cases.

4. INLET-TOWER INFLECTOR ANALYSES

The inflector on the inlet tower, in conjunction with the deflector on the outlet stack, should promote flows in the specified direction through the passive vault under all wind conditions. For a no-wind condition, the inflector and deflector ideally should have no impact on the natural-draft flow through the vault. Under windy conditions, these devices work by increasing the relative pressure in the inlet tower and decreasing the relative pressure in the outlet tower. An overall assessment of either device must consider the effects of both.

The inflector at the top of the inlet tower, in the 30% design,¹ consists of a flat canopy with a four-vane cruciform that extends from the canopy down to the top of the tower and continues down inside the tower for a short distance. The horizontal cross-section of the inside of inlet tower is square, 7.5 ft (2.29 m) on a side. The canopy is a flat square with 16-ft (4.88-m) sides, and it is 7.5 ft (2.29 m) above the top of the inlet tower. The cruciform is centered on the tower and extends from the canopy to 2.5 ft (0.76 m) below the top of the tower. The vanes of the cruciform extend out to the inside surface of the tower and are straight and perpendicular to the tower surface they touch. The inflector design also includes screens that extend up from the top of the tower at the inside edge to the canopy.

The design of the deflector on the outlet stack was not complete in the 30% design.

Our analyses for the 30% design looked only at the inflector. Reference 2 documented four analysis cases:

Case 1: Models the inlet tower without any part of the inflector with boundary conditions specifying no wind and 5500-ft³·min⁻¹ air flow through the tower. This air flow is the nominal full flow based on the earlier facility case 4 analysis.²

Case 2: Repeats case 1 and adds the canopy and cruciform parts of the inflector but not the screens.

Case 3: Repeats case 2 and adds the screens to the inflector.

Case 4: Repeats case 3 and adds a horizontal wind¹⁰ at $7.5 \text{ m}\cdot\text{s}^{-1}$ (16.8 mph) from the west. Because of the symmetry of the model, a west wind is equivalent to the same wind from the north, east, or south. This wind is parallel to two of the vanes of the cruciform and perpendicular to the other two vanes.

The analysis results for these four cases showed that the hydraulic loss associated with the inflector under the condition of no wind was very small, -0.211 Pa (-0.00003 psi). The velocity of the wind added in case 4 represents an average maximum for a sustained wind; it is not intended to be the absolute maximum sustained wind or a maximum gust. The wind increased the pressure inside the tower significantly and would have increased the facility flow in a coupled calculation. On the basis of these four cases, the inflector appeared to meet its design goals as stated above.

The following subsections describe our continuing analyses that build off of case 4. Reference 11 documents these calculations in more detail.

4.1. Geometry Model for the Inlet-Tower Inflector Analyses

The geometry model encompassed a large air volume surrounding the inlet tower and inflector to minimize the effects of the domain boundaries on the flow in the vicinity of the tower and the inflector. The computational domain was a cube, 17.07 m (56 ft) on a side. This domain extended three times the tower outside width horizontally from the tower and vertically above the canopy and included the upper 7.47 m (24.5 ft) of the inlet tower. The bottom of the domain was just above the roof of the existing NMSF structure. We superimposed on the domain a nonuniform Cartesian grid of $\sim 100,000$ cells. The minimum cell dimension was 0.1 m (4 in.) in and around the inflector, and the geometric-progression option for generating the grid allowed the cell size to increase to 1.3 m (52 in.) at the domain boundaries. For some calculations, we refined the vertical grid inside the tower to a uniform 0.15 m (6 in.) vertical spacing.

The canopy, cruciform, and tower walls were modeled as no-flow interfaces. The effects of this approach were that the solid walls had zero thickness and that a wind could not impinge on the exposed edges of the cruciform and canopy. We modeled the screens in the inflector with one-cell thick regions, which allowed us to include a hydraulic resistances⁶ for the screens.

4.2. Thermal-Hydraulic Model for the Inlet-Tower Inflector Analyses

The CFX-4.2 Solver evaluated steady-state solutions for the flow field. Selected options included isothermal, incompressible flow and the $k\text{-}\epsilon$ turbulence model. Because we did not model heat-transfer effects, we could characterize the air with constant temperature and density. Also, there was no need for a buoyancy model.

For the case of no wind, we applied atmospheric-pressure boundary conditions to all four sides and the top of the computational domain. At the bottom of the computational domain and outside the tower, we applied a free-slip boundary that prevented flow in or out at that boundary but did not exert a shear stress on the flow field. At the inside bottom of the tower, we applied either a mass-flow boundary condition or, when the flow out of the tower bottom was to be zero, a free-slip boundary. The mass-flow boundary allowed the velocity distribution to develop naturally across the boundary when a nonzero flow was specified for the bottom of the tower, i.e., the velocity distribution varied according to the local pressure distribution, constrained only by the total flow through the boundary. When we applied a wind boundary condition, we changed the atmospheric-pressure boundaries on the sides of the computational domain where the wind entered to velocity boundaries. These velocity boundary conditions specified the necessary velocity components to produce the correct wind velocity and direction.

To be consistent with the facility calculations previously, we assumed that the temperature of the air was 308.15 K (95°F) (Ref. 5), which corresponds to a historical maximum for Los Alamos, and that the atmospheric pressure was 76,976 Pa (11.16 psia) (Ref. 4). The air density corresponding to this temperature and pressure was $0.87023 \text{ kg}\cdot\text{m}^{-3}$ ($0.0543 \text{ lb}_m\cdot\text{ft}^{-3}$).

4.3. Analysis Results for the Inlet-Tower Inflector

Our new analyses concentrated on cases involving wind. We continued to use a wind speed of $7.5 \text{ m}\cdot\text{s}^{-1}$ (16.8 mph) from Ref. 10 as averaged maximum wind speed. As indicated previously, this velocity did not represent an absolute maximum for either a sustained wind or a wind gust. We considered two directions for the wind:

parallel wind—this wind blows from the west and is parallel to two of the cruciform vanes and perpendicular to the other two vanes; the symmetry of the model causes a wind from the north, east, or south to produce the same results; and

diagonal wind—this wind blows from the southwest, northwest, northeast, or southeast and enters the inflector along a diagonal.

Because the inflector presents different frontal areas to these two winds, we expected the results to be different in otherwise similar calculations. For the current analyses, we assumed that the vertical component of the wind velocity is zero—a horizontal wind.

Table II shows the new analyses cases that have been run. All case numbers beginning with 4 built off of the case 4 results in Ref. 2 with a parallel wind. Case numbers beginning with 5 were with a diagonal wind. The letters appended to case number identified variations that were applied in a particular calculation. The applications of the letters was consistent so that case 4b could be compared to case 5b to determine the effect of changing the wind direction.

TABLE II
ANALYSIS CASES FOR THE INLET-TOWER INFLECTOR

Case Number	Description	Relative Pressure at Bottom of the Tower (Pa)
4a	5500-ft ³ ·min ⁻¹ air flow at the base of the tower (iteration 1 with facility case 4)	+6.67 to +6.8
4aa	9700-ft ³ ·min ⁻¹ air flow at the base of the tower (iteration 2 with facility case 14)	+4.93 to +5.04
4b	no air flow at the base of the tower	+8.04 to +8.23
4c	2 quadrants closed off at the top of the tower, no air flow at the base of the tower	+4.97 to +5.98
4d	8-vane cruciform, no air flow at the base of the tower	+7.63 to +7.72
4e	refined mesh inside tower, no air flow at the base of the tower	+8.4 to +8.8
4f	extended cruciform, no air flow at the base of the tower	+7.46 to +8.43
4g	cruciform open in middle, no air flow at the base of the tower	+2.44 to +2.54
5a	5000-ft ³ ·min ⁻¹ air flow at the base of the tower (iteration 1 with facility case 4)	-15.2 to -14.8
5b	no air flow at the base of the tower	-6.54 to -6.3
5c	2 quadrants closed off at the top of the tower, no air flow at the base of the tower	+2.6 to +2.8
5d	8-vane cruciform, no air flow at the base of the tower	-3.29 to -2.04
5e	refined mesh inside tower, no air flow at the base of the tower	-6.78 to -5.92
5f	extended cruciform, no air flow at the base of the tower	-6.97 to -0.95
5g	cruciform open in middle, no air flow at the base of the tower	-8.48 to -5.78

As indicated earlier, Ref. 11 documents the analysis results with brief descriptions of each case together with appropriate pressure and velocity-vector plots.

Table II summarizes the effectiveness of the inflector by listing the calculated range of relative pressures at the inside bottom of the inlet tower. The range of pressures is from the minimum to the maximum value. These relative pressures are given

in pascals and can be converted to psi by dividing by 6894.757 Pa/psi. The relative pressure is calculated from the absolute pressure by subtracting off both the reference pressure (76,976 Pa) and a gravity head ($\rho \cdot g \cdot \Delta z$). By this definition, atmospheric pressure corresponds to 0.0 Pa relative pressure, and the relative pressure has no dependence on elevation. Therefore, only hydraulic losses and flow effects such as velocity, stagnation, turning, separation, and eddies can cause the relative pressure to change. In the following discussion, the terms pressure and relative pressure will be used interchangeably.

Cases 4a and 4aa represent an iteration between the inlet-tower inflector analyses and the facility analysis cases 4 and 14 (Ref. 2 and 7, respectively). To run this type of calculation in a coupled mode would require a much larger, and therefore longer running, facility model in terms of the number of cells. Such a model would have increased nodalization detail in and around the inlet tower and the inflector, and then it would have to extend far away from the inlet tower to get to the free-field wind. We ran facility case 4 with the assumption of no wind around the inlet tower, and from that calculation, we determined that total facility flow was $5479 \text{ ft}^3 \cdot \text{min}^{-1}$; the relative pressure at the middle elevation of the inlet tower was -0.563 Pa. We applied a facility flow of $5500 \text{ ft}^3 \cdot \text{min}^{-1}$ to the bottom of the inlet tower in inflector case 4a together with a parallel wind of $7.5 \text{ m} \cdot \text{s}^{-1}$. This second calculation indicated that the corresponding pressure in the inlet tower was $\sim 6.71 \text{ Pa}$, much higher than in the previous facility calculation. Case 4a also supplied the component velocities V_x , V_y , and V_z across one screen of the inflector and pressures on the other three screens to form boundary conditions on the corresponding screens in the facility calculation. A new facility calculation, case 14, was run with these boundary conditions (applied as if a south wind were blowing) to produce a new facility flow, $9693 \text{ ft}^3 \cdot \text{min}^{-1}$, and a new tower pressure of 5.65 Pa. We then ran case 4aa of the inflector model with the same wind conditions as in case 4a but with the updated facility flow of $9700 \text{ ft}^3 \cdot \text{min}^{-1}$. This final iteration of the inflector model produced a tower pressure of $\sim 4.99 \text{ Pa}$.

Clearly, this computational procedure was converging to a consistent set of wind conditions for the facility model and facility flow for the inflector model. However, the process was cumbersome and slow. The above process should have included the deflector on the outlet stack, but that design was not complete. The addition of the outlet deflector would increase the complexity of the information exchange among the required calculations. These difficulties raise the question of whether or not this is the correct approach for evaluating the inflector (and deflector) design. With limitations on budget and time, we should explore other alternatives.

The analysis effort can provide two pieces of information about the inflector and the deflector. The first piece is a determination of the hydraulic loss associated with each device. This information does not require modeling the wind and can be obtained through analyses like case 3 for the inflector. If the hydraulic loss is significant, it can then be added to the facility model. The second piece of information that is needed concerns the effectiveness of inflector and deflector to

enhance facility flow under windy conditions. Because these devices should enhance, that is, increase, the flow, it is conservative to perform the facility calculations under conditions of no wind, which produces the minimum facility flow. It is not necessary to analyze the inflector and deflector coupled to the facility model for a variety of wind conditions. Rather, we only need to show that the inflector and deflector satisfy the design goal of increasing the flow for windy conditions.

An approach to demonstrating that the inflector and deflector meet the design goals is set up separate models like the one we used for the inflector calculations, but with zero facility flow, and apply wind boundary conditions to both the inflector and the deflector. The effectiveness of the combination of inflector and deflector for a given wind condition, in a pass/fail grading system, can be determined by the difference in relative pressures between the inlet tower and the outlet stack ($P_{\text{inlet}} - P_{\text{outlet}}$). For the case of zero wind, this pressure difference would be zero, and the inflector and deflector have no effect. For a given wind, if this pressure difference is positive, then the inflector and deflector designs pass. If the pressure difference is negative, the design fails for that case. It is important to look at the pressure difference because, as Table II indicates for some cases, the relative pressure at the bottom of the inlet tower can be negative. For such a case to receive a passing grade, then the relative pressure in the outlet stack must be less than in the inlet tower.

Case 5a changes the wind direction by 45° , from a parallel wind to a diagonal wind in combination with a facility flow boundary condition of $5000 \text{ ft}^3 \cdot \text{min}^{-1}$. Because the facility flow is somewhat lower, case 5a was not an exact counterpart to case 4a for determining the effect of wind direction on the inflector performance. Because the calculated pressures at the bottom of the inlet tower were -15 Pa for case 5a and $+6.7 \text{ Pa}$ for case 4a, it is clear that wind direction has a big impact on the performance of the current inflector design and that the acceptability of the inflector and deflector must be judged on the basis of combined performance.

Cases 4b and 5b essentially repeat cases 4a and 5a for zero facility flow. In both cases the tower pressures increased as expected with the reduction in facility flow. However, the diagonal wind still produced negative pressures at the bottom of the tower. The next several cases investigated small design changes in an attempt to drive the diagonal wind case towards positive pressures.

Cases 4c and 5c investigated the effect of blocking flow in two quadrants of the inflector by setting the horizontal flow area to zero in those quadrants at the top of the inlet tower. For the diagonal wind, the blocked quadrants were the ones that extended to the sides from the perspective of the wind; the upstream and downstream quadrants remained open. This modification does not represent a viable design alternative for several reasons, but it was illustrative of the flow processes that result in negative pressures at the base of the tower. Case 5c was the only case with the diagonal wind that produced positive pressures in the tower ($+2.7 \text{ Pa}$). In case 4c, the tower pressures were reduced but still positive. It appears

that improved performance of the inflector occurs when the areas of positive and negative pressure across the top of the tower, corresponding to regions of flow into and out of the tower, are similar in size. For the current inflector design in a parallel wind, the areas of inflow and outflow at the top of the tower are essentially equal, corresponding to the upstream and downstream sides of the inflector, and all of the case 4 perturbations produced positive pressures at the bottom of the inlet tower. However, the current design with a diagonal wind has an outflow area (three quadrants) at the top of the tower that is three times the size of the inflow area (one quadrant), and all of the case 5 perturbations except 5c produced negative pressures. The model perturbation in case 5c forced equal areas for inflow and outflow.

In cases 4d and 5d, we extended the cruciform concept to include a total of 8 vanes extending out radially from the center. This modification placed the additional vanes midway between the original vanes so that there are now vanes every 45° around the canopy. The dimensions of all vanes were the same so that the outer edge of each vane was the same distance from the center of the cruciform. Because of the existing rectangular grid and because the new vanes run diagonally through the grid, the new vanes have a stair-step shape as they follow cell interfaces. These additional vanes help to direct flow into the tower and make the presentation of the cruciform to the wind less dependent on the direction of the wind. Figure 5 shows the horizontal velocity field 0.7536 m (2.47 ft) below the canopy and 1.5324 m (5.03 ft) above the top of the tower. The figure shows how the air enters and circulates in the cruciform.

The pressures at the base of the tower for case 4d ranged from +7.63 to +7.72 Pa; these pressures were only slightly lower than the corresponding case 4b with the standard cruciform. However, the pressures in case 5d ranged from -3.29 to -2.04 Pa, and although they were still negative, they were much higher than for case 5b. Figure 6 shows the pressure distribution on a vertical slice through the tower and cruciform for case 5d. Figures 7 and 8 show the horizontal pressure distributions for cases 4d and 5d, respectively, at an elevation of 0.7536 m (2.47 ft) below the canopy. These two figures illustrate that both wind directions interacted with the 8-vane cruciform in a more uniform way than with the standard cruciform design.

For cases 4e and 5e, we refined the mesh inside the tower by making the discretization in the vertical direction more uniform and generally smaller than in the original model. The results here show only small differences in the range of pressures in the tower from cases 4b and 5b. We wanted to refine the mesh in order to extend the cruciform further into the tower.

Cases 4f and 5f extended the 4-vane cruciform an additional 2.90 m (9.5 ft) down the inlet tower. There were no substantive changes in the pressures at the bottom of the tower for either case relative to cases 4e and 5e. However, the range of pressures expanded in both cases because the extended cruciform forced flow perturbations further down the tower.

In cases 4g and 5g, we opened up the center of the cruciform to flow by removing the inner third of each vane. The effect of this change was to create venturi-like flows through the center of the cruciform for both wind directions, which served to reduce the pressures in both cases relative to cases 4e and 5e.

At this point, we discussed the potential for changes to the inflector design with FDI. Given the many possibilities and the complexities of some of the alternatives, the decision was made to wait for the results of the 60% design phase before continuing the analyses of the inflector and deflector.

4.4. Significant Findings from the Inlet-Tower Inflector Analyses

The following points summarize the more significant findings that were discussed above as part of the inlet-tower inflector analyses:

1. The coupling of separate calculations for the facility, the inflector, and the deflector will work, but the process is tedious and cumbersome. We recommended a different approach for assessing the performance of the combination of inflector and deflector under windy conditions that is relatively simple and straightforward. It involves determining under conditions of zero facility flow if the wind effects on the inflector and deflector increase or decrease the ΔP across the facility.
2. The relative sizes of the areas of positive and negative pressure at the top of the tower significantly affect the performance of the inflector. These areas correspond to regions of flow into and out of the tower. The bigger the inflow area is relative to the outflow area, the better the performance is in terms of creating higher pressures at the bottom of the tower.
3. A cruciform with 8 or more vanes helps to mitigate the effects of changing the wind direction. However, with the current square design of the tower, the pressures at the bottom of the tower with the diagonal wind were still negative. If increasing the number of vanes in the cruciform is a viable alternative, then we need to investigate changing the inlet tower design to a circular cross section to make the inflector and tower combination more nearly axisymmetric.

5. CONCLUSIONS

We have continued our thermal and flow analyses to support the NMSF Renovation Project. Sections 2.3 and 4.4 above have summarized our significant findings based on this latest round of calculations.

We have demonstrated the capability to run detailed calculations to track boundary layers on the heated drywells and the interactions between these boundary layers and facility flow. Limitations to this capability include the long run time to converge the calculations and the use of symmetry planes to reduce how much of

the storage bay has to be modeled. The effects of these limitations include reducing the number of different calculations that can be run and an inability to calculate variations across the width of the bay. Also, extending such a model through the inlet tower and the outlet stack involves approximations in order to match wall drag and other hydraulic losses.

The initial detailed calculation of the drywell array using boundary conditions from the earlier facility case 4, which analyzed the facility with a porous-medium representation of the drywell array, suggests that the maximum air temperatures in the porous-medium calculations may be low by as much as 1.2°C (2.2°F). We believe that this estimate of the error in the maximum air temperature is conservatively large because the maximum bay air temperature in the discrete-model case is very near the top of the bay and well above the highest container inside the drywells. At lower elevations in the discrete model, the calculated temperatures outside the regions of the boundary layers are cooler. The impact of this error on the projected plutonium-metal temperatures made for the 30% design report (see Ref. 2) is small given the available margin to the applicable temperature limit.

It appears that the porous-medium approach for modeling the drywell array can be modified to represent better the thermal stratification that was apparent in the discrete-model calculation. The changes involve increasing the number of cells across the width of the bay significantly in order to specify a heterogeneous power distribution that produces power densities that are comparable to the power densities in the boundary layers on the drywells. The interpretation of the results from such a model is more complex.

The drywell and container analysis described earlier is preliminary and not well converged. We believe that the convergence problems are related to our attempt to include much finer geometric detail in the model than we had previously. Even with the problems, the calculation shows promise for coupling directly temperature profiles from facility calculations to the maximum temperature of stored plutonium metal. The path forward is to recast the model to reduce some of the detail and to develop better boundary conditions at the top and bottom of the drywell.

With regard to the inlet-tower inflector analyses, we demonstrated the capability to link facility calculations to separate calculations for the inflector and deflector in order to get converged solutions for facility flow and wind effects. The process is tedious and cumbersome. We proposed a much simpler methodology for assessing the performance of the inflector and deflector that relies on only standalone calculations of inflector and deflector and that avoids the need to converge multiple calculation to a consistent set of boundary conditions. The methodology relies on determining the implied ΔP across facility for zero facility flow. If the ΔP is positive, then for that particular combination of inflector and deflector designs and wind condition, the designs are acceptable. If the ΔP is negative, then the designs are unacceptable for that set of conditions.

6. REFERENCES

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Fig. 1.
Air temperature in
y-direction (x,z)
planes of the 47
half-drywell model
of case 20

Air Temperature (K)

3.1767E+02
3.1609E+02
3.1450E+02
3.1291E+02
3.1132E+02
3.0974E+02
3.0815E+02

y = 0.3429 m (13.5 in.)

y = 0.2667 m (10.5 in.)

y = 0.1905 m (7.5 in.)

y = 0.1143 m (4.5 in.)

y = 0.0381 m (1.5 in.)

Fig. 2.

Air temperature in
y-direction (x,z)
planes of the hetero-
geneous heat-source
model of case 22

Air Temperature (K)

3.1767E+02
3.1608E+02
3.1450E+02
3.1291E+02
3.1132E+02
3.0974E+02
3.0815E+02

y = 2.7432 m (9 ft)

y = 2.1336 m (7 ft)

y = 1.5240 m (5 ft)

y = 0.9144 m (3 ft)

y = 0.3048 m (1 ft)

Fig. 3. Drywell-array vault inlet-to-outlet air-pressure difference.

Fig. 4. Axial heat-flux profile from the outer surface of a drywell with 10 heated cans.

Fig. 5. Case 5d. Air Flow in Inlet Tower. Cut through Z plane 0.7536 m below canopy (inflow area).

Fig. 6. Case 5d. Pressure in Inlet Tower. Cut through Y plane 0.4656 m away from center of cruciform on windward side of cruciform.

Fig. 7. Case 4d. Pressure in Inlet Tower. Cut through Z plane 0.7536 m below canopy (inflow area).

Fig. 8. Case 5d. Pressure in Inlet Tower. Cut through Z plane 0.7536 m below canopy (inflow area).