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ORNL/CSD/TM--239

DE87 005293

HEATING6 ANALYSIS OF INTERNATIONAL THERMAL BENCHMARK PROBLEM SETS 1 AND 2

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Date Published - October 1986

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HEATING6 ANALYSIS OF INTERNATIONAL THERMAL BENCHMARK PROBLEM SETS 1 AND 2

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ABSTRACT

In order to assess the heat transfer computer codes used in the analysis of nuclear fuel shipping casks, the Nuclear Energy Agency Committee on Reactor Physics has defined seven problems for benchmarking thermal codes. All seven of these problems have been solved using the HEATING6 heat transfer code. This report presents the results of five of the problems. The remaining two problems were used in a previous benchmarking of thermal codes used in the United States, and their solutions have been previously published.

1. INTRODUCTION

The first Nuclear Energy Agency Committee on Reactor Physics (NEACRP) Specialists Meeting on Heat Transfer Assessment of Transport Packages took place during May 1985 under the auspices of the Organization for Economic Cooperation and Development (OECD) in Paris. The meeting was attended by specialists from the United States, France, Belgium, Italy, Sweden, and the United Kingdom. The objectives of the meeting were to select a series of heat transfer problems related to transport casks and a schedule for implementing the calculational work. These problems would serve to test the relative validity of ("to benchmark") heat transfer computer codes currently in use. It was agreed that the exercise would be carried out in two phases. In phase 1 simple problems would be solved to demonstrate the ability of the codes to model heat transfer processes, and in phase 2 more complicated problems representative of shipping cask design would be solved. Four problems were agreed to for phase 1. At a subsequent meeting in October 1985, three additional problems were agreed to for phase 2.

The United States effort was coordinated by the Sandia National Laboratories Transportation Technology Center (SNL/TTC). Oak Ridge National Laboratory, in support of the SNL/TTC effort, developed solutions to the benchmark problems using the HEATING6^{1,2} computer code. Descriptions of these solutions are presented in this report.

2. HEATING6 DESCRIPTION

HEATING6 is a finite difference code designed to solve steady-state and/or transient heat conduction problems in one-, two-, or three-dimensional Cartesian, cylindrical, or spherical coordinates. The thermal conductivity, density, and specific heat may be both time- and temperature-dependent. The thermal conductivity may be anisotropic as well. Materials may undergo a change of phase. Heat generation rates may be dependent on time, temperature, and position. Boundary temperatures may be time- and position-dependent. The boundary conditions,

which may be surface-to-boundary or surface-to-surface, may be specified temperatures or any combination of prescribed heat flux, forced convection, natural convection, and radiation. The thermal efficiency of certain finned surfaces may be modeled. HEATING6 has an internal mesh generator that allows variable mesh spacing along each axis. The nodal locations are at the intersections of the grid lines.

HEATING6 allows user-defined thermal connections to be made between any two nodes in the mesh or from any node to a boundary. The number of these connections that can be included in the model is limited only by the size of the computer being used. These thermal connections can model conduction, radiation, convection, and natural convection heat transfer mechanisms, although their inclusion in the code was primarily to allow general radiation modeling.

For steady-state problems, the point-successive overrelaxation iterative method and a modification of the "Aiken delta squared extrapolation process" are used to solve the finite difference equations which approximate the partial differential equations. Many one- and two-dimensional steady-state problems may be solved more efficiently using the optional direct solution technique. Several numerical schemes to solve transient heat conduction problems are available in HEATING6. In general, the recommended method is the implicit technique which can range from Crank-Nicolson to a fully implicit procedure. The system of equations arising from this technique is solved by point-successive overrelaxation iteration. Transient problems may also be solved either by an explicit method that is stable for a time step of any size (Levy's method) or by using the stability-limited classical explicit procedure. Problems involving change-of-phase must be solved using one of the explicit procedures.

Much of the recent development on the HEATING6 code has been supported by the U. S. Nuclear Regulatory Commission (NRC), and HEATING6 is included in the NRC Standardized Computer Analyses for Licensing Evaluation (SCALE) system of codes. The version of HEATING6 used in the work reported here is the latest version available at the Department of Energy facilities in Oak Ridge.² Certain features used in these analyses were not available in earlier versions. This latest version will not be publicly available until the next release of the SCALE system. This is currently planned for calendar year 1987. The SCALE system is distributed by the Radiation Shielding and Information Center, Oak Ridge National Laboratory (P. O. Box X, Oak Ridge, TN 37831).

3. PROBLEM DESCRIPTIONS AND SOLUTIONS

The phase 1 problems were designated US-1, US-2, UK-1, and UK-2. The phase 2 problems were designated FR-1, US-3, and UK-4. Problems US-1 and US-2 were previously investigated in a benchmarking of thermal codes used in the United States. These problems have already been documented³ and are not included in this report. Problem descriptions and solutions for the remaining five problems are presented below.

The problem descriptions and accompanying assumptions were extracted from the descriptions furnished by the participating countries to the fullest extent possible. However, in some cases the original problem descriptions were either incomplete or ambiguous. These descriptions were supplemented with additional written and verbal communications from the problem originators and the United States coordinator at SNL/TTC. It was not the intent of the authors to alter the problem definitions in any manner, but it is possible that this was done inadvertently.

3.1 THERMAL BENCHMARK UK-1, HEATED PIN ARRAY IN ISOTHERMAL ENCLOSURE

3.1.1 Objective

The objective of this problem is to test the code's ability to model direct and reflected radiation in a fuel pin bundle. The problem is based on thermal tests carried out on a simulated pressurized water reactor (PWR) fuel element in a gas environment.

3.1.2 Geometry and Dimensions

The model comprises a 16 x 16 array of heated and unheated pins (simulating fuel and control rod positions) in a square isothermal enclosure. The pin layout is presented in Fig. 1.

Enclosure:	
Internal dimension	242.7 x 242.7 mm
Pin array:	
Pin layout	16 x 16 square pitch
Number of heated pins	236
Number of unheated pins	20
Pin outside diameter	10.75 mm
Clad thickness	0.725 mm
Pin pitch	14.30 mm

3.1.3 Material Properties

Enclosure:	
Material	Unspecified
Surface emissivity	0.38
Pin cladding:	
Material	Unspecified
Conductivity	19.0 W/m-°C
Emissivity	0.8
Pin filler (power source):	
Material	Unspecified
Conductivity	0.0982 W/m-°C
Power per pin	8.156 W/m
Ambient gas:	
Material	99% helium and 1% air
Conductivity	$(0.14426 + 3.42 \times 10^{-4} T - 7.1471 \times 10^{-8} T^2)$ W/m-°C where T is in °C

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Fig. 1. Benchmark problem UK-1 geometry.

3.1.4 Heat Transfer Model

The internally generated heat is removed by conduction and radiation to the internal surface of the enclosure which is maintained at 181.2°C. The pins and the enclosure are assumed to be gray diffusely emitting and reflecting surfaces. The gas within the assembly does not absorb, emit, or scatter radiation.

Due to symmetry, only one-fourth of the assembly is modeled. There is additional symmetry in the problem which would have allowed the modeling of only one-eighth of the assembly, but geometric restrictions in HEATING6 made this impractical. The finite difference mesh is presented in Fig. 2. The filler for each pin is modeled as a square with an area equal to that of the actual pin. There are four nodes on the surface of each pin. Conductances for heat transfer in the clad around each pin were calculated external to HEATING6 and supplied as input to the model. The gas conductances between nodes were calculated internally by HEATING6. However, since the geometry of the model does not match the true geometry, the spacing between pins in the model had to be chosen such that the calculated conductance through the gas would match that of the physical problem.

The radiation exchange factors were also calculated external to HEATING6 and were input to the code. View factors from a quadrant of one pin to quadrants on other pins were determined numerically using the actual geometry (not the finite difference approximation). For pins that were not near the enclosure, 25 view factors to quadrants on other pins were deemed significant enough to be carried along to the next step in the calculation. Pins near the enclosure saw fewer surfaces. The next step was the determination of gray body exchange factors to account for reflected as well as direct radiation between pins. In this calculation the planes of symmetry bounding two sides of the model were approximated by diffusely reflecting surfaces with a reflectivity near 1.0. (The actual value used was 0.999 to avoid numerical difficulties.) Given the fact that the first row of pins is situated very close to the plane of symmetry, the assumption of a diffusely reflecting surface rather than a specularly reflecting surface should not introduce a significant error in the results.

When reflections are considered, some of the radiation emitted from a quadrant on one pin reaches every surface in the model. To model all reflections an inordinate number of radiation connectors would be required in the computer model. To avoid this, any connector for which the radiation exchange factor was less than 10^{-4} was eliminated from the model. This left approximately 10,000 pairs (two connectors are required to connect a pair of nodes). The number of radiation connectors could probably be reduced without significantly influencing the solution. A steady-state solution was obtained using the point-successive overrelaxation iteration method in HEATING6.

3.1.5 Results

Clad temperature distributions as a function of position along the lines A-B and A-C are depicted in Fig. 3. The results are presented for the true pin locations of the model rather than the modified finite difference mesh locations. These temperature distributions are also presented in Table 1.

Fig. 2. Benchmark problem UK-1 finite difference mesh.

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Fig. 3. Clad temperature distribution along lines A-B and A-C.

Table 1. Clad temperature distribution along lines A-B and A-C

Line A-B			Line A-C		
Pin No.	Position (mm)	Temperature (°C)	Pin No.	Position (mm)	Temperature (°C)
1	5.4	357.9	1	5.4	358.1
3	14.8	353.7	2	8.9	355.9
3	25.6	351.7	2	19.7	354.9
6	35.1	343.1	4	23.2	350.3
6	45.8	340.1	4	34.0	348.6
10	55.3	327.9	7	37.5	341.3
10	66.0	323.9	7	48.3	339.0
15	75.5	307.2	11	51.8	329.7
15	86.3	302.1	11	62.6	326.7
21	95.7	278.7	15	66.1	313.6
21	106.5	273.0	16	76.9	309.6
28	116.0	250.7	22	80.4	291.5
28	126.7	244.8	22	91.2	286.3
36	136.2	217.0	29	94.7	260.2
36	146.9	210.7	29	105.5	253.2
Enclosure	161.5	181.2	Enclosure	114.2	181.2

3.2 THERMAL BENCHMARK UK-2, A PLANE FINNED SURFACE

3.2.1 Objective

The objective of this problem is to test the ability of codes to represent (1) radiation from one surface to another and (2) radiation reflected from one surface to another.

3.2.2 Geometry and Dimensions

The model consists of the external finned surface of a flask shown in Fig. 4. A flat plate comprises the base, which has rectangular fins on its exterior surface. Dimensions for the base plate and fins are given below.

Fin thickness	10 mm
Fin pitch	60 mm
Fin length	150 mm
Base thickness	100 mm

3.2.3 Material Properties

Fin and Base:	
Material	Mild steel
Thermal conductivity	50.0 W/m-K
Specific heat capacity	500.0 J/kg-K
Density	7800.0 kg/m ³
Emissivity	1.0 (case 1)
	0.8 (case 2)

3.2.4 Heat Transfer Model

Heat transfer on the outside (finned) surface of the flask is by natural convection and radiation. The environment emissivity is 1.0. The gas within the fin cavity does not absorb, emit, or scatter radiation. Radiative view factors (for case 1 with a surface emissivity of 1.0) and radiative exchange factors (for case 2 with a surface emissivity of 0.8) were calculated external to HEATING6 and supplied as input to the code. Initially, the external ambient temperature is 38.0°C, and the natural convection heat transfer coefficient is $2.0 |T_s - T_a|^{1/3}$ W/m²-K. To simulate an engulfing fire, the ambient temperature is raised to 800.0°C, and the natural convection heat transfer coefficient is raised to a value of 10.0 W/m²-K. After 30 min the external conditions return to their initial values.

Heat transfer on the inside surface is by natural convection. The natural convection heat transfer coefficient is $500.0 |T_s - T_f|^{1/3}$ W/m²-K, where T_s is the local inside surface temperature; and T_f is the fluid temperature inside the flask. The temperature of the fluid contained in the flask is a constant 100°C throughout the transient.

The finite difference mesh is presented in Fig. 4. The steady-state solution is obtained using the direct solution technique. An initial time step of 0.1 s is used for the fire transient solution using the Crank-Nicolson implicit technique. The time step is increased by 10% each time step until a value of 10 s is reached. Ten seconds is used until the end of the fire transient. At this point, the time step is reduced to 0.1 s, and the process of incrementing it by 10% is repeated. As before, when the time step reaches a value of 10 s, it is held constant until the completion of the passive cooling transient.

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Fig. 4. Benchmark problem UK-2 geometry with finite-difference mesh superimposed.

3.2.5 Results

Locations 1, 2, and 3 (identified on Fig. 4) are the fin tip center, the fin root midway between fins, and the inside wall surface midway between fins, respectively. The temperature, as a function of time for these locations, is presented in Table 2 and Figs. 5, 6, and 7 for a fin surface emissivity of 1.0. The external surface temperature as a function of position (measured from location 1 along the fin surface toward location 2) for times 0.0 and 30.0 min is presented in Fig. 8 and Table 3.

The calculations were repeated for a fin surface emissivity of 0.8. These results are presented in Figs. 9 through 12 and Tables 4 and 5.

Table 2. Temperature history at selected locations (emissivity = 1.0)

Time (min)	Temperature (°C)		
	1	2	3
0	76.0	93.2	97.3
5	475.8	136.6	104.5
10	596.4	177.4	116.1
15	637.4	208.4	124.9
20	653.0	227.8	130.3
25	659.9	239.1	133.5
30	662.8	245.3	135.3
35	338.9	199.2	131.2
40	231.1	165.3	123.1
45	173.9	142.1	116.5
50	140.3	126.8	111.5
55	119.2	116.6	108.0
60	105.6	109.8	105.4
65	96.5	105.2	103.4
70	90.4	102.0	101.9
75	86.2	99.6	100.7
80	83.3	97.9	99.6
85	81.2	96.4	98.7
90	79.7	95.4	98.2

Fig. 5. Temperature history for location 1 (emissivity = 1.0).

Fig. 6. Temperature history for location 2 (emissivity = 1.0).

Fig. 7. Temperature history for location 3 (emissivity = 1.0).

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Fig. 8. Fin surface temperature distribution at selected times (emissivity = 1.0).

Table 3. Fin surface temperature distribution at selected times (emissivity = 1.0)

Distance along fin (cm)	Temperature (°C)			
	0 min	30 min	60 min	90 min
0.0	76.0	662.8	105.6	79.7
0.25	76.0	662.9	105.6	79.7
0.50	75.9	662.6	105.5	79.6
1.50	76.1	654.3	105.8	79.8
2.50	76.4	642.0	106.1	80.2
3.50	76.9	626.8	106.4	80.6
4.50	77.5	609.2	106.7	81.2
5.50	78.2	589.6	107.0	81.8
6.50	79.0	568.1	107.2	82.6
7.50	80.0	544.7	107.5	83.5
8.50	81.1	519.2	107.7	84.4
9.50	82.3	491.6	107.9	85.5
10.50	83.6	461.5	108.2	86.7
11.50	85.0	428.9	108.4	88.0
12.50	86.6	393.3	108.6	89.4
13.50	88.3	354.5	108.9	91.0
14.50	90.2	312.1	109.3	92.7
15.50	92.3	263.8	109.7	94.6
16.50	93.0	249.9	109.8	95.2
17.50	93.2	245.3	109.8	95.4
18.00	93.2	245.3	109.8	95.4

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Fig. 9. Temperature history for location 1 (emissivity = 0.8).

Fig. 10. Temperature history for location 2 (emissivity = 0.8).

Fig. 11. Temperature history for location 3 (emissivity = 0.8).

Fig. 12. Fin surface temperature distribution at selected times (emissivity = 0.8).

Table 4. Temperature history of selected locations (emissivity = 0.8)

Time (min)	Temperature (°C)		
	1	2	3
0	76.2	93.3	97.3
5	447.8	135.1	104.3
10	575.8	174.0	115.6
15	623.1	203.9	124.1
20	642.0	223.0	129.5
25	650.5	234.3	132.7
30	654.3	240.6	134.5
35	344.0	196.9	130.6
40	234.5	164.3	122.8
45	176.3	141.7	116.3
50	141.9	126.7	111.5
55	120.5	116.7	108.0
60	106.5	109.9	105.4
65	97.2	105.3	103.4
70	91.0	102.0	101.9
75	86.7	99.7	100.7
80	83.7	97.9	99.6
85	81.6	96.5	98.8
90	80.0	95.5	98.2

Table 5. Fin surface temperature distribution at selected times
(emissivity = 0.8)

Distance along fin (cm)	Temperature (°C)			
	0 min	30 min	60 min	90 min
0.0	76.2	654.3	106.5	80.0
0.25	76.2	654.3	106.5	80.0
0.50	76.2	654.1	106.4	80.0
1.50	76.4	646.8	106.7	80.2
2.50	76.7	635.5	106.9	80.5
3.50	77.1	621.4	107.2	80.9
4.50	77.7	604.8	107.4	81.4
5.50	78.4	586.0	107.6	82.1
6.50	79.2	565.2	107.8	82.8
7.50	80.1	542.3	108.0	83.7
8.50	81.2	517.3	108.2	84.6
9.50	82.3	490.0	108.3	85.7
10.50	83.6	460.1	108.5	86.8
11.50	85.1	427.5	108.7	88.1
12.50	86.7	391.7	108.9	89.5
13.50	88.4	352.6	109.1	91.1
14.50	90.2	309.5	109.4	92.8
15.50	92.4	260.0	109.8	94.7
16.50	93.0	245.5	109.9	95.3
17.50	93.3	240.6	109.9	95.5
18.00	93.3	240.6	109.9	95.5

3.3 THERMAL BENCHMARK FR-1, SODIUM TRANSPORT— PHASE CHANGE

3.3.1 Objective

The objective of this problem is to test the code's ability to model the phenomenon of phase change without volume change in the transient state.

3.3.2 Geometry and Dimensions

The model comprises a radial section of the transport cask shown in Fig. 13. The cask dimensions are given below starting from the center and moving toward the surface.

	Outer Radius	Thickness
Power source	0.0750 m	—
Sodium	0.1725 m	0.0975 m
Stainless steel container	0.1745 m	0.0020 m
Container/cask body gap	0.1750 m	0.0005 m
Steel cask body	0.5000 m	0.3250 m
Neutron shield	0.6500 m	0.1500 m

3.3.3 Material Properties

Power source:	
Material	Unspecified
Thermal conductivity	5.0 W/m-°C
Heat capacity	3.3×10^6 J/m ³ -°C
Heat generation	2800.0 W/m
Sodium:	
Material	Sodium
Thermal conductivity	Liquid (135.5290-0.1673 T) W/m-°C Solid (90.6038-0.048523 T) W/m-°C where T is in °C
Density	890.0 kg/m ³
Specific heat	1338.0 J/kg-°C
Latent heat of fusion	113.5×10^3 J/kg
Melt temperature	97.0°C
Container:	
Material	Stainless steel
Thermal conductivity	20.0 W/m-°C
Heat capacity	3.9×10^6 J/m ³ -°C
Surface emissivity	0.5

Fig. 13. Benchmark problem FR-1 geometry (half section of cask).

Cavity:	
Material	Air
Thermal conductivity	Table 6
Body:	
Material	Steel
Thermal conductivity	54.0 W/m-°C
Heat capacity	3.9x10 ⁶ J/m ³ -°C
Surface emissivity	0.5
Neutron shield:	
Material	Composite medium
Thermal conductivity	10.0 W/m-°C
Heat capacity	6.5x10 ⁵ J/m ³ -°C

Table 6. Thermal conductivity of air

Temperature (°C)	Thermal conductivity (W/m-°C)
0	2.442x10 ⁻²
100	3.182x10 ⁻²
200	3.868x10 ⁻²
300	4.494x10 ⁻²
400	5.077x10 ⁻²
500	5.629x10 ⁻²
600	6.150x10 ⁻²

3.3.4 Heat Transfer Model

Heat transfer across the air filled gap is by conduction through the air and thermal radiation between the gap surfaces. The ambient temperature is initially 0.0°C. External heat transfer is by convective and radiant exchange with the ambient. The coefficient of global exchange between the cask and the ambient is 30.0 W/m²-°C. From 0 to 30 min the ambient temperature is raised to 800.0°C, and the coefficient of global exchange is increased to 200.0 W/m²-°C. From 30 to 90 min the ambient temperature is returned to 0.0°C, and the coefficient of global exchange is reduced to 30.0 W/m²-°C.

The finite difference mesh is presented in Fig. 13. The nodes within each material are evenly spaced. The initial steady-state solution was determined with a direct solution technique. The transient results were calculated using a classical explicit solution technique with a time step determined by the stability criterion.

3.3.5 Results

The transient temperatures for locations 1, 2, 3, 4, 5, and 6, corresponding to the radial locations 0.65 m, 0.5 m, 0.1750 m, 0.1745 m, 0.075 m, and 0.0 m, respectively, are presented in Table 7 and Figs. 14 through 19.

Table 7. Temperature history of selected locations

Time (min)	Temperature (°C)					
	1	2	3	4	5	6
0	22.85	34.54	43.21	85.00	89.56	134.12
5	545.30	50.04	43.21	85.01	89.56	134.12
10	606.07	81.00	43.31	85.01	89.56	134.12
15	628.63	106.62	44.48	85.12	89.62	134.12
20	639.23	127.30	47.95	85.59	89.94	134.14
25	645.58	144.73	53.88	86.69	90.80	134.23
30	650.17	160.00	61.77	88.61	92.38	134.52
35	323.66	165.98	71.03	91.41	94.81	135.19
40	222.86	158.11	81.10	94.81	97.00	136.28
45	167.20	148.94	90.95	96.23	97.55	138.02
50	134.83	141.05	99.43	97.02	97.82	139.43
55	115.21	134.72	106.00	97.06	98.29	140.49
60	102.86	129.77	110.74	97.15	98.71	141.33
65	94.79	125.88	113.94	97.18	98.98	141.99
70	89.31	122.79	115.96	97.28	99.28	142.53
75	85.46	120.29	117.08	97.29	99.46	142.97
80	82.64	118.20	117.71	101.38	102.43	143.34
85	80.51	116.42	118.39	109.92	110.96	144.16
90	78.83	114.86	118.84	116.38	117.62	147.18

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Fig. 14. Temperature history for location 1.

Fig. 15. Temperature history for location 2.

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Fig. 16. Temperature history for location 3.

DWG. NO. K/G-88-1317
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Fig. 17. Temperature history for location 4.

DWG. NO. K/G-86-1316
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Fig. 18. Temperature history for location 5.

DWG. NO. K/G-88-1315
(U)

Fig. 19. Temperature history for location 6.

3.4 THERMAL BENCHMARK US-3, BUSS CASK NORMAL TRANSPORT

3.4.1 Objective

The objective of this problem is to test the steady-state solution technique in a complex geometry with temperature-dependent properties and boundary conditions.

3.4.2 Geometry and Dimensions

The problem to be modeled is a loaded BUSS cask during normal transport. The stainless steel cask contains a stainless steel basket which holds 12 strontium fluoride filled canisters. The basket/cask assembly is shown in Fig. 20. The cask has circular, external fins with rectangular cross sections as shown in Fig. 21. The canisters are of a double wall stainless steel construction with an air-filled gap between the walls. The dimensions for all components are given below.

Canister:

Strontium fluoride radius	2.62 cm
Inner canister wall thickness	0.24 cm
Inner/outer wall gap (air filled)	0.20 cm
Outer canister wall thickness	0.28 cm
Canister/basket gap (air filled)	0.31 cm
Canister length	50.10 cm

Basket:

Basket outer radius	25.70 cm
Basket/cask gap (air filled)	0.18 cm
Number of canister holes	12
Canister hole diameter	7.30 cm

Cask:

Cask inner radius	25.88 cm
Cask wall thickness (excluding fins)	33.00 cm
Fin height	10.20 cm
Fin width	2.50 cm
Fin spacing	7.50 cm

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Fig. 20. Benchmark problem US-3 geometry (half section of basket/cask assembly).

Fig. 21. Benchmark problem US-3 fin geometry.

3.4.3 Material Properties

Cask, basket, canister walls:

Material	Stainless steel
Thermal conductivity	Table 8
Density	7817.0 kg/m ³
Specific heat	460.0 J/kg-°C
Emissivity	0.5

Power source:

Material	Strontium fluoride
Thermal conductivity	0.46 W/m-°C
Density	2600.0 kg/m ³
Specific heat	8400.0 J/kg-°C
Power per canister	267.0 W

External environment:

Material	Air
Density	Table 9
Kinematic viscosity	
Specific heat	
Conductivity	
Prandtl number	

Table 8. Thermal conductivity of stainless steel

Temperature (°C)	Thermal conductivity (W/m-°C)
0.0	16.3
100.0	17.0
200.0	17.0
300.0	19.0
400.0	19.0
600.0	22.0
800.0	26.0
1000.0	31.0

Table 9. Thermal properties of air at 101.325 kPa

Temp. (K)	Density (kg/m ³)	Kinematic viscosity x10 ⁶ (m ² /s)	Specific heat x10 ³ (kJ/kg-K)	Conductivity (W/m-K)	Prandtl number (---)
273	1.2930	13.30	1.004	24.07	0.717
283	1.2473	14.18	1.004	24.86	0.714
293	1.2047	15.08	1.004	25.63	0.712
300	1.1766	15.75	1.005	26.14	0.711
303	1.1650	16.00	1.005	26.37	0.710
313	1.1277	16.95	1.005	27.09	0.709
323	1.0928	17.91	1.006	27.80	0.708
333	1.0600	18.89	1.007	28.51	0.707
343	1.0291	19.89	1.008	29.21	0.706
350	1.0085	20.63	1.008	29.70	0.706
353	1.0000	20.91	1.008	29.89	0.705
363	0.9724	21.95	1.009	30.58	0.704
373	0.9463	23.01	1.010	31.26	0.703
400	0.8825	26.00	1.013	33.05	0.703
450	0.7844	31.78	1.020	36.33	0.700
500	0.7060	37.99	1.029	39.51	0.699
550	0.6418	44.56	1.039	42.60	0.698
600	0.5883	51.50	1.051	45.60	0.699
650	0.5431	58.80	1.063	48.40	0.701
700	0.5043	66.41	1.075	51.30	0.702
750	0.4706	74.32	1.087	51.10	0.703
800	0.4412	82.56	1.099	56.90	0.703
850	0.4153	91.10	1.110	59.70	0.703
900	0.3922	99.90	1.121	62.50	0.702
950	0.3716	109.00	1.131	64.90	0.705
1000	0.3530	118.30	1.141	67.20	0.709
1100	0.3209	138.00	1.160	73.20	0.705
1200	0.2942	159.00	1.177	78.20	0.705

3.4.4 Heat Transfer Model

Heat transfer across all gaps is by conduction through the air filling the gap and by radiation between the gap surfaces. This heat transfer is modeled as one-dimensional (i.e., a node can only exchange heat with the node directly across the gap from itself). The external boundary condition is radiation and convection heat transfer with the ambient. The fins are not modeled directly, but their influence is accounted for by modifying the boundary conditions such that the net radiative and convective heat transferred in a height of 7.5 cm (the distance between centers of adjacent fins) accurately reflects the physical situation. Handling the effect of the fins in this manner allows the problem to be modeled in two dimensions.

Due to symmetry it is necessary to model only one-eighth of the cask. The two-dimensional finite difference mesh used is given in Fig. 22. The model was defined in three parts: one for each of the fuel canisters and one for the cask/basket assembly. The thermal connections between the parts (conduction through the air in the canister/basket interstitial gap and radiation across the gap) were calculated external to HEATING6 and supplied to the code as input.

The external surface convective heat transfer coefficient was determined from an empirical correlation for natural convection heat transfer from a smooth horizontal cylinder given by:

$$Nu = 0.13 Ra^{0.333}$$

The fluid properties in the Nusselt (Nu) and Rayleigh (Ra) numbers are evaluated at the arithmetic mean of the local surface temperature and the ambient temperature. The heat removed from the surface by convection is given by:

$$q_c = \eta h (A_s + A_t) (T_b - T_\infty) ,$$

where

- η = fin efficiency (= 0.8),
- h = heat transfer coefficient determined from Nusselt number,
- A_s = surface area of cavity formed by adjacent fins,
- A_t = surface area of tip of fin,
- T_b = temperature at the base of the fin (cask surface temperature in two-dimensional model),
- T_∞ = temperature of the environment.

Fig. 22. Benchmark problem US-3 finite difference mesh.

It is assumed that there is no radiation from the ends of the fins. The net radiation loss from the enclosure formed by adjacent fins is given by:

$$q_r = \frac{\sigma(T_s^4 - T_\infty^4)}{\frac{1-\epsilon_s}{A_s\epsilon_s} + \frac{1}{A_sF_{s-\infty}} + \frac{1-\epsilon_\infty}{A_\infty\epsilon_\infty}},$$

where

- σ = Stefan-Boltzmann constant,
- T_s = cask surface temperature,
- T_∞ = temperature of the environment,
- A_s = surface area of cavity formed by adjacent fins,
- A_∞ = area between fin tips that along with A_s completely encloses the cavity between adjacent fins,
- ϵ_s = emissivity of fin surface,
- ϵ_∞ = emissivity of environment,
- $F_{s-\infty}$ = configuration factor (= 0.212).

A steady-state solution was determined using the point-successive overrelaxation technique in HEATING6.

3.4.5 Results

The maximum temperatures occurring in various components in the model are presented in Table 10. The locations where these maximum temperatures occurred are identified in Fig. 22.

Table 10. Maximum temperatures occurring in various components

Component	Temperature (°C)	Location
Strontium fluoride	523.7	1
Inner cladding	432.1	2
Outer cladding	371.8	3
Basket	296.5	4
Cask	105.4	5,5'
Cask surface	55.7	Uniform temperature on surface

3.5 THERMAL BENCHMARK UK-4, WATER-FILLED CYLINDER IN A FIRE

3.5.1 Objective

The objective of this problem is to test the anisotropic conductivity option in conjunction with radiation and convection boundary conditions.

3.5.2 Geometry and Dimensions

The model (shown in Fig. 23) comprises a closed cylindrical vessel partially filled with water. The axis of the vessel is vertical.

External height	1000.0 mm
Wall thickness	10.0 mm
Outside diameter	200.0 mm
Inside diameter	180.0 mm
Water level above base	700.0 mm
Height of dry wall	280.0 mm

3.5.3 Material Properties

Vessel:	
Material	Steel
Thermal conductivity	50.0 W/m-°C
Density	7800.0 kg/m ³
Specific heat	500.0 J/kg-°C
Emissivity	0.8
Water:	
Material	Water
Thermal conductivity	Horizontal 5000.0 W/m-°C Vertical 0.6 W/m-°C
Density	1000.0 kg/m ³
Specific heat	4200.0 J/kg-°C
Emissivity	1.0

The conductivity of water in the horizontal direction is enhanced to 5000.0 W/m-°C from the physical value of 0.6 W/m-°C to simulate the effect of natural convection in the water.

Fig. 23. Benchmark problem UK-4 geometry with finite difference mesh superimposed.

3.5.4 Heat Transfer Model

The initial temperature of the vessel and the water is a uniform 38°C. From 0 to 30 min a uniform heat flux of 10 kW/m² is applied to the outer curved surface of the vessel. The lid and base outside surfaces are adiabatic. Internal heat transfer is by natural convection from the vessel to the water with a convective coefficient given by:

$$h_w = 500 |T_i - T_w|^{1/3},$$

where

T_i = local vessel inside surface temperature (°C)

T_w = local water temperature (°C).

There is no heat transfer by conduction or convection in the vapor space above the water, but radiation exchange takes place between the surfaces of the vessel and the water.

From 30 to 60 min there is natural convection and radiation heat loss from the outer curved surface of the vessel to a 38°C ambient. The convection coefficient is given by

$$h_e = 1.3 |T_o - T_a|^{1/3},$$

where

T_o = local vessel outside surface temperature (°C),

T_a = ambient temperature (38°C).

The finite difference mesh used for this problem is presented in Fig. 23. Gray body radiation exchange factors were calculated external to HEATING6 and were input to the code to account for the radiant energy reflected around the cavity above the water in the vessel. A Crank-Nicolson implicit procedure was used to solve the problem. The time step was initially 0.1 s and was increased by 10% each time step until a value of 10 s was reached. The 10 s time step was used until a time of 30 min into the transient (the end of the fire) at which time it was once again reduced to 0.1 s. The process of increasing the time step 10% each time step was repeated until a time step of 10 s was again reached. A 10 s time step was used for the rest of the transient.

3.5.5 Results

Plots of temperature vs time for the water surface immediately adjacent to the vessel wall (location 1 on Fig. 23) and for the lid at the center inside surface (location 2) are presented in Figs. 24 and 25, respectively. The same information is presented in tabular form in Table 11. The temperature profile along the centerline of the water from the base to the water surface at times of 15, 30, 45, and 60 min is presented in Table 12 and Fig. 26. The temperature profile on the inside surface of the vessel from base to lid at times of 15, 30, 45, and 60 min is presented in Fig. 27 and Table 13.

Fig. 24. Temperature history of water surface temperature adjacent to vessel wall.

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Fig. 25. Temperature history of inside center lid surface.

Table 11. Temperature history of selected locations

Time (min)	Temperature (°C)	
	Water surface adjacent to wall	Inside center lid surface
5	62.79	64.34
10	96.97	118.55
15	131.93	178.78
20	167.23	239.56
25	203.00	298.39
30	239.41	354.09
35	233.51	347.30
40	213.59	302.85
45	197.18	263.79
50	183.97	232.22
55	173.05	206.89
60	163.79	186.41

Table 12. Water temperature profile on the centerline at selected times

Height above base (cm)	Temperature (°C)			
	15 min	30 min	45 min	60 min
0	82.55	126.53	120.51	115.49
10	79.53	122.98	120.36	115.89
20	79.66	123.08	120.24	115.70
30	79.66	123.11	120.29	115.82
40	79.67	123.18	120.69	116.83
50	79.75	124.29	124.24	122.56
60	82.41	137.47	145.12	142.95
70	131.77	239.33	197.27	163.83

Fig. 26. Water temperature profile on the centerline at selected times.

Fig. 27. Temperature profile on the inside surface of vessel at selected times.

Table 13. Temperature profile on the inside surface of vessel at selected times

Height above base (cm)	Temperature (°C)			
	15 min	30 min	45 min	60 min
0	88.16	132.10	119.48	114.59
10	88.43	131.90	118.72	114.30
20	88.56	131.99	118.56	114.12
30	88.58	132.02	118.61	114.26
40	88.58	132.11	119.14	115.46
50	88.71	133.50	123.44	121.65
60	92.34	148.58	145.50	141.10
70	147.18	249.40	188.18	158.06
80	230.28	367.68	205.91	157.27
90	239.61	399.73	227.59	166.02
98	209.04	380.35	248.69	177.78

ACKNOWLEDGMENTS

Funding for this work was supplied by the Transportation Technology Center of Sandia National Laboratories. Special thanks goes to reports editor M. G. Culbertson and secretary L. W. Winborn who prepared this report for publication.

REFERENCES

1. D. C. Elrod, G. E. Giles, and W. D. Turner, *HEATING6: A Multidimensional Heat Conduction Analysis with the Finite-Difference Formulation*, ORNL/NUREG/CSD-2/V2, Union Carbide Corp., Nuclear Div., Oak Ridge Natl. Lab., October 1981.
2. K. W. Childs et. al., *HEATING: A Computer Program for Multidimensional Heat Transfer Analysis (Version 6.1)*, to be published as updated section F10 in ORNL/NUREG/CSD-2/V2, at Oak Ridge Natl. Lab.
3. C. B. Bryan and K. W. Childs, *Thermal Codes Benchmarking - HEATING6 Results*, K/CSD/TM 57, Martin Marietta Energy Systems, Inc., Oak Ridge Gaseous Diffusion Plant, August 1985.

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