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HEDL-TME 72-66

May, 1972
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CLIRA Decay Heat Removal Test

Hanford Engineering Development Laboratory

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AT(45-1)-2170

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Hanford Engineering Development Laboratory

Richland, Washington

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Under Contract No. AT(45-1)-2170

HEDL-TME 72-66
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CLIRA Decay Heat Removal Test

Hanford Engineering Development Laboratory

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ABSTRACT

An electrically heated test facility was designed and fabricated for use in an ex-reactor investigation of heat transport during the handling of a solid wall flow tube type Closed Loop In-Reactor Assembly (CLIRA) while in the shielded, air-cooled Combined Loop Ex-Vessel Machine (CLEM). The heat generation zone was limited to the bottom three feet of an approximately 12-foot long vertical column of sodium contained in a simulated 3-3/4 in. OD CLIRA integral pot. Heat was transmitted radially from the sodium column (pot), across a "stagnant" argon gas gap, to the inner surface of a simulated CLEM cold wall. Finally, heat could be removed by axial forced air flow past the finned outer surface of the cold wall.

A heat generation rate of six KW was attained in tests without flow tube overflow, with a maximum air-cooling rate of 1020 CFM, and a heating element cladding temperature limit of 1000°F. Eight KW were attained in similar tests, except that the sodium was allowed to overflow the top of the flow tube and recirculate by natural convection. Tests with the sodium level below the top of the flow tube and heat generation rates above 1 to 2 KW resulted in an essentially linear, axial temperature gradient of about 5°F per inch in the region above the heated length. The formation of this gradient was attributed primarily to convective buoyancy-driven instabilities. In tests with heat generation rates of 1 to 2 KW, it was necessary to drastically reduce the air cooling rate to prevent freezing at the top of the sodium column.

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I. INTRODUCTION

This report describes the design and fabrication of an electrically heated test section that was used to simulate the heat transport associated with the handling, after irradiation and discharge from the reactor vessel, of a fueled Fast Flux Test Facility (FFTF) closed loop test assembly while in the air-cooled ex-vessel handling machine. Test results arising from ex-reactor investigations using this Facility are presented and discussed. The work was performed in response to Westinghouse Development Test Requirements Specifications (WDTRS) No. 6.18.⁽¹⁾

The overall purpose of the test was to establish the level of fission product decay heat that could be handled with a solid wall flow tube Closed Loop In-Reactor Assembly (CLIRA) while in a cold wall Combined Loop Ex-Vessel Machine (CLEM). In addition, problem areas, such as the possibility of freezing at low decay heat levels and the loss of forced convection cooling (cooling air fan failure), were investigated briefly.

The test facility consisted of a long, cylindrical, sodium-filled vessel, the simulated CLIRA, installed concentrically in a second cylindrical vessel based on the design of the CLEM cold wall. The heat generation zone (simulated fuel element decay heating) was limited to the bottom three feet of an approximately 12-foot long vertical column of sodium. Heat was transmitted radially from the heated and unheated lengths of the sodium-filled vessel (the integral pot), across a stagnant argon gas gap, to the inner surface of the CLEM cold wall. Finally, heat could be removed by forced convection with axial forced air flow past the finned outer surface of the cold wall.

The quantity of heat transported from the integral pot to the cold wall was a function of the axial temperature gradient established by conduction and convection in the vertical column of sodium. As a result of the single wall flow tube installed inside the integral pot, two modes of convective axial transport could be studied (i.e., convective mixing and convective recirculation). The flow tube ran essentially the entire length of the sodium column, surrounding the heated length. Below the heated length, at the bottom of the simulated CLIRA, openings were provided to connect the sodium inside the flow

tube to that contained in the annulus formed by the flow tube and the pot. However, at the top, the sodium level could be adjusted either above or below the top of the 12-foot long flow tube. In the latter case, tests were an investigation of axial heat transport by buoyancy-driven mixing (circulation) in nine-foot long quiescent pools of sodium, heated from below. In the case where the level was maintained above the top of the flow tube, the heated sodium could circulate up the flow tube, overflow, and then return to the heated length via the annulus formed by the flow tube and pot and the openings at the bottom.

Obviously, for significant convective recirculation to occur, temperatures inside the flow tube had to be greater than those in the annulus. Since the test facility design was based on the solid wall concept, radial thermal resistance of the flow tube was small. In other words, sodium inside the flow tube was closely coupled thermally to that in the annulus.

II. SUMMARY

A heat generation rate of six KW was attained in tests without flow tube overflow, with a maximum air-cooling rate of 1020 CFM, and a heating element cladding temperature limit of 1000° F. Eight KW was attained in similar tests, except that the sodium was allowed to overflow the top of the flow tube.

In tests with heat generation rates of 1 to 2 KW, it was necessary to drastically reduce the air cooling rate or turn the fan off to prevent the sodium from freezing at the top of the sodium column. The maximum power attained in the test to simulate fan failure was six KW. The sodium level was maintained below the top of the flow tube in this simulation of the loss of forced convection cooling. The heating element cladding temperature limit, however, was raised to 1200° F for this test, and cold wall temperatures up to 680° F were observed.

Tests with the sodium level below the top of the flow tube and with heat generation rates above 1 to 2 KW resulted in an essentially linear, axial temperature gradient of about 5° F per inch in the region above the heated portion of the simulated fuel bundle. The gradient was observed in tests where the maximum air cooling rate was used, as well as in the fan failure simulation with no air cooling. Flatter axial profiles were observed in tests with flow tube overflow. The region of maximum temperature shifted from the axial centerline toward the upper end of the three-foot long heated length when the sodium level was raised from below to above the top of the flow tube.

Analysis of the results from the evaluation of the reference handling technique (i.e., tests where the level was maintained below the top of the flow tube) indicate that, in general, only about half of the heat transmitted from the surface of the integral pot was associated with the three-foot long heated length. Therefore, the remaining heat transmitted radially from the unheated pot surface was first transported axially upward, resulting in the linear temperature gradients described above for tests with heat generation rates above 1 to 2 KW. Review of the linear nature of these gradients and their slope suggests that the axial heat flow was primarily the result of a convective mechanism.

In the tests with flow tube overflow, the observations that axial temperature gradients were even flatter and that the region of maximum temperature was shifted upward or downstream, both indicate that buoyancy-driven recirculation was the dominant mechanism for axial heat transport. Measurements taken during the eight KW run indicate that, at the same elevation, the average difference between the temperature of the sodium inside and outside the flow tube was about 50° F. Calculations based on the assumption that all heat transported radially from the unheated surface was first associated with the upflow of heat (mass) inside the flow tube, followed by downflow in the annulus formed by the flow tube and pot, result in a calculated mean value for mass flow rate of about 900 pounds per hour. The calculations also indicate that recirculation was established to the extent that up to 65 percent of the heat was transmitted from the unheated length.

In tests with air flow, values calculated for the average combined unit surface conductance for radial heat transfer from the heated and unheated lengths of the integral pot ranged from 3 to 4 and 2 to 3 BTU/(hr) (ft²) (°F), respectively. Although 80 to 90 percent of the heat flow was associated with radiation, the preceding values account for the effects of both convection and radiation across the stagnant argon gas gap that exists between the pot and the cold wall. Heat flow by radiation was calculated assuming an emissivity of 0.6, since tarnished (weathered) stainless surfaces were used in the fabrication of both the integral pot and the cold wall. A value of about 0.5 was calculated for the gray body shape factor.

III. TEST FACILITY DESCRIPTION

A. OVERALL DESCRIPTION

The electrically heated facility (see Figures 1a and 1b) described in this section was designed for use in an ex-reactor investigation of heat transport during the handling of an FFTF Closed Loop In-Reactor Assembly (CLIRA) while in the shielded, air-cooled, ex-vessel handling machine (CLEM).

The fuel assembly was modeled by 19 electrical heaters mounted in a hexagonal array and enclosed in a hexagonal duct. Overall length of each heater was 72 inches; however, only 36 inches was fabricated of material that had high enough electrical resistivity to generate uniform heat. Sheathed thermocouples simulated spiral wire wrap spacers of fuel pins and were used to give local sodium temperature information within the heater bundle. The axial and radial locations of the thermocouples are outlined in Figure 2.

As shown in Figure 1, the following CLIRA-CLEM components were simulated in the test assembly: CLIRA solid flow tube; CLIRA integral pot; CLEM inner cold wall; and the CLIRA instrument tube.

- The 19-pin fuel assembly was mounted in the solid wall flow tube. The flow tube was simulated by a 2-1/2 in. schedule 40, stainless steel pipe 12 feet long.
- The solid wall flow tube subassembly was then mounted into a simulation of the integral pot.
- The CLIRA integral pot was the outer sodium containment vessel. This unit was simulated by a 3-3/4-in. OD 0.120-in. wall stainless steel tube. The 176-inch long integral pot was sealed by welding on the bottom and by a mechanical seal on the top.
- The CLEM inner cold wall simulated the heat sink around the integral pot. During operation the annulus between the cold wall and pot contained argon cover gas. The cold wall was simulated by an

8-in., schedule 40 stainless steel pipe approximately 16 feet long. The emissivity of the cold wall was not measured, but is estimated to be above 0.5.

- The CLIRA instrument tube mock-up was located above and connected to the hexagonal duct. It was simulated by a 1-in. stainless steel tube about 8 feet long within an 8-foot length of 1-1/4 in., schedule 40 stainless steel pipe. It should be noted that the 1-in. tube was sealed on the bottom, so it did not contain sodium during testing.

The heat removal system simulating the CLEM system included a blower to supply the air for cooling and a thermally insulated air duct which enveloped the CLEM cold wall. Electrical trace heating was necessary on all external tubing containing sodium, as well as on the CLEM cold wall. As shown in Figure 2, a total of 88 test thermocouples was installed on the various components.

B. ELECTRICALLY HEATED BUNDLE

Nineteen commercial electrical heaters, arranged in a hexagonal array, were used to simulate the test fuel assembly. The heaters were single-ended, with electrical power brought to one end only; the sodium and other hardware of the test section completed the electrical circuit.

The heaters had a total length of 72 inches; however, only 36 inches of the total were heated to simulate the fueled length. Heating was uniform, with no simulation of axial or radial power gradients. The top 33 inches were unheated to simulate the fission gas plenum region. Mild steel with low electrical resistance was used for the cold portions of the heater, whereas coiled nichrome wire with high electrical resistance was used for the heated portion. Incalloy cladding was used as the heater sheath material and magnesium oxide powder provided the insulation. The outside diameter of the heater was a nominal 0.315 inch. Each heater had a power rating of 530 watts at 240 volts.

Figure 1a. Test Facility Assembly

Figure 1b. Test Facility Assembly

T/C LOCATION	NO. OF T/C'S AT LOCATION	INCHES ABOVE BOTTOM OF HEATED LENGTH	DATA ACQ. SYSTEM CHANNEL
O. D. of Cold Wall	4 at 90	18	4-7
O. D. of Cold Wall	4 at 90	78	8-11
O. D. of Cold Wall	4 at 90	138	12-15
O. D. of Integral Pot	4 at 90	0	16-19
O. D. of Integral Pot	4 at 90	18	20-23
O. D. of Integral Pot	4 at 90	28	24-27
O. D. of Integral Pot	4 at 90	48	28-31
O. D. of Integral Pot	4 at 90	64	32-35
O. D. of Integral Pot	4 at 90	78	36-39
O. D. of Integral Pot	4 at 90	108	40-43
O. D. of Integral Pot	4 at 90	136	44-47
Heating Element*	1	-3	48
Heating Element*	1	0	49
Heating Element*	3	8	50-52
Heating Element*	4	18	53-56
Heating Element*	4	28	57-60
Heating Element*	3	48	61-63
Heating Element*	3	64	64-66
O. D. of Instrument Tube	4 at 90	78	67-70
O. D. of Instrument Tube	4 at 90	108	71-74
O. D. of Instrument Tube	2 at 180	134	75&76
O. D. of Instrument Tube	2 at 180	135	77&78
O. D. of Instrument Tube	2 at 180	136	79&80
O. D. of Instrument Tube	2 at 180	137	81&82
O. D. of Instrument Tube	4 at 90	138	83-86
O. D. of Instrument Tube	2 at 180	140	87&88
O. D. of Instrument Tube	2 at 180	143	89&90
O. D. of Instrument Tube	2 at 180	161	91&92

NOTE: LOOKING DOWN FROM TOP -
NUMBERED CONSECUTIVELY
CLOCKWISE FROM 0°

TYPICAL AT ALL LOCATIONS
WHERE 4 T/C'S LOCATED AT 90°

* See Corresponding detail above

Figure 2. Thermocouple Identification/Location

A fitting was attached to the bottom of each heater to complete the electrical circuit and to seal the heater internals from the sodium. (See Figure 3). This fitting screwed onto the bottom cold end and shouldered against the sheath. An electron beam weld was used to complete the closure. This fitting was also slotted to key the heater into the grid to assure positioning.

Power was brought into the top end of the heater by an insulated electrical input lead. The leads selected were 17 feet long, stainless steel sheathed, magnesium oxide insulated with a solid copper center conductor. The copper conductor was No. 14 AWG and generated a negligible amount of heat. The outside of the lead was 0.125 inch.

The electrical connection was made by first drilling and tapping the top cold end of the heater for an O-80 external thread cut on the copper. A transition piece was fabricated in a conical shape that reduced the 0.315-inch heater diameter to the 0.125 inch lead diameter. This transition piece was electron beam welded to the heater sheath, the lead slipped through the transition piece, and screwed into the heater cold end. A final electron beam weld was then made between the transition piece and the heater lead. After welding was complete, each heater weld was dye penetrant tested and the entire heater and lead transition helium leak tested. Figure 4 shows the heaters with the lead wires attached.

Electron beam welding was selected as a means of closure after considerable difficulty was experienced using automatic TIG welding. The TIG welder lacked adequate controls to prevent melt-through of the lead sheath. When this happened, the heated portion created internal pressure which, in turn, created holes in the weld.

The table in Figure 2 indicates that temperature measurements were required at various axial and radial locations on the heater sheath. Mineral insulated, chromel-alumel thermocouples were used that had an outside diameter of 0.062 inch. No terminations were made in the sheathed thermocouple leads from the hot junction end until the sheath penetrated the CLEM cold wall.

The thermocouples were used as wire wraps on the heaters, down to the elevations specified in the table on Figure 2. From this elevation down to

Figure 3. Bottom End View of the Hexagonal Duct and Three of the Total of Nineteen Electrical Heaters Used to Simulate the Fuel

Figure 4. Overall View of Hexagonal Duct and Three of the Nineteen Electrical Heaters Used to Simulate the Fuel

the bottom of the heater, solid stainless wire of like diameter was used. The wire wraps were installed in a clockwise direction on a 12-inch pitch. Figure 4 illustrates three wire wrapped heaters prior to installation in the bundle. Spot welding was used to attach two-mil stainless steel tabs over the wire wrap to the heater sheath at 4-inch intervals.

The individual heaters, with wire wraps and lead wires attached, were then assembled into a bundle. This was accomplished by laying the hexagonal duct in a horizontal plane across a table. Starting with a bottom row, the heaters were passed through the duct and onto the grids that were welded into the bottom of the hexagonal duct. The plates were designed so as to prevent the heaters from moving laterally in any direction. The heaters were to be removable, so no attempt was made to prevent them from being lifted upward when they were mounted in a vertical position. (See Figure 4).

C. HEXAGONAL DUCT

The hexagonal duct was a length of hexagonal tubing 74 inches long. The internal distance across opposite flats averaged 1.842 inches. Nominal duct wall thickness was 0.125 inch. To facilitate accurate centering of the hex duct inside the flow tube, nibs were welded to the corners and turned in the lathe to fit the inside diameter of the flow tube. Plates to key the heaters into location were welded in the bottom of the hexagonal duct. (See Figure 3). The plates (grid assembly) provided a triangular lattice of 0.4 inch.

D. SOLID WALL FLOW TUBE

The solid wall flow tube was simulated by using a 2-1/2 inch, schedule 40 pipe 144 inches long. The nominal inside diameter of this pipe was 2.47 inches. Three spacers were welded to the top end of the pipe, 120 degrees apart. The spacers provided concentricity on the top just as the support plate did on the bottom. (See Figure 5).

E. INSTRUMENT TUBE SUBASSEMBLY

This subassembly consisted of a 91-inch length of 1-1/4 inch, schedule 40 stainless steel pipe. The bottom end of this pipe had three 1 X 1/8 X 8-inch

Figure 5. Bottom End View of the Assembled Heaters Mounted Inside the Hexagonal Duct, CLIRA Solid Wall Flow Tube, and CLIRA Integral Pot. Figure 5 Inset. Support Plate Which Was Welded Inside the CLIRA Integral Pot to Support the CLIRA Solid Wall Flow Tube and the Hexagonal Duct Containing the Electrical Heaters.

MECHANICAL SEAL USED TO SEAL THE CLIRA INTEGRAL POT

Figure 6. Top End View of the CLIRA Integral Pot Assembly Mounted in the CLEM Cold Wall

long stainless steel bars welded to it, at 120-degree centerline spacing. The other ends of these bars were welded to alternate flats on the hexagonal duct. The top end of the pipe used three spacers to maintain concentricity with the integral pot. The welding of the bars to the hexagonal duct effectively created a single unit of the instrument tube, hexagonal duct, flow tube and heaters. This assembly rested, when in the vertical position, on a support plate and was removable from the integral pot. (See Figure 5).

Enclosed within the 1-1/4 inch pipe was a 90-7/8 inch long 1-inch OD X 1/32-inch wall tubing. This length of tubing had a plug welded in the bottom end. The thermocouple and heater leads were brought up through the annulus formed by the tube within the pipe. Stainless steel tabs spot welded to the tube secured the thermocouple and heater leads and was the only support provided to hold this tube in the correct vertical position.

F. INTEGRAL POT

The integral pot was the sodium and argon containing vessel of the assembly. It consisted of a 176-inch length of 3-3/4 X 0.120-inch wall stainless steel tube. This tube was sealed on the bottom with a welded pipe cap. The sodium inlet line to the integral pot was a 1/2-inch stainless steel tube that penetrated the pipe cap. A support plate was fabricated and welded on the inside of the bottom end of the integral pot (see Figure 5). This support ring has six 5/8-inch wide radial slots milled in it to provide clear channels between all concentric members within the integral pot. A land was also milled into the ring to key the hexagonal duct in position. (See Figure 5).

The top end of the integral pot was closed by means of a removable mechanical seal. The sodium outlet line was welded to the pot at 134 inches above the sodium inlet pipe. An argon line (for cover gas) penetrated the integral pot approximately 24 inches above the sodium outlet tube. A plate was welded to the top of the integral pot which supported it when installed in the CLEM cold wall. Mechanical seals, using a crushable lava sealant, were used to bring thermocouple and heater leads out of the integral pot. (See Figure 6).

G. CLEM COLD WALL

The CLEM cold wall also served as a secondary containment for the integral pot and all piping and lead wire penetrations into the CLEM were sealed. A removable, mechanical seal was provided for access to the integral not. (See Figure 6). When the top head was in place, a stagnant argon space encompassed the integral pot.

The CLEM cold wall was fabricated from a 156-inch length of 8-inch, schedule 40 stainless steel pipe. A 16 X 8-inch reducer was used to enlarge the top end to provide working room to penetrate the leads and tubing. Support plates were welded into the 16-inch diameter portion to support the integral pot. (See Figure 7). A 16-inch, schedule 10 pipe cap was used for the top closure of the CLEM. The cap was penetrated to provide the argon inlet port.

The bottom 148-1/2 inches of the CLEM had twenty slots milled around the outside circumference for installation of cooling fins. The fins were fabricated from two-inch wide by 1/8 inch thick stainless steel plates. The fins were placed in the shallow slots for location purposes, and short TIG welds were made at intervals along each fin. Each fin ran the entire length of each slot; however, each fin consisted of several lengths of material (see Figures 7 and 8).

H. AIR HEAT REMOVAL SYSTEM

The air duct that encompassed the CLEM cold wall subassembly was fabricated by rolling 1/8-inch steel plate to a 12-3/4-inch inside diameter (Figure 8). Flanges with gussets were installed on each end to provide seals and mounting support. Total overall length of the duct was 145 inches. This portion of the duct was thermally insulated during testing.

Height limitations of the test facility required that the air duct make a 90-degree turn on the bottom of the CLEM cold wall assembly and run on the horizontal plane. Air turbulence caused by the direction change was minimized by installing straightening vanes inside the elbow. The elbow also provided a base for the entire assembly to rest on the floor.

Figure 7. Top End View of the CLEM Cold Wall with Top Seal Removed

Figure 8. Bottom End View of the CLEM Cold Wall Mounted Inside the Air Heat Removal Duct

A horizontal run of 10-inch diameter duct approximately 10 feet long connected the elbow to an electrically driven blower. (See Figure 9). This duct contained an orifice plate and pressure taps for air flow measurements. Air flow out of the blower was regulated by a set of dampers. Air could be bypassed back into the room, rather than through the air duct for reduced testing flow rates. The blower had the capability of moving a maximum of 1020 CFM of air. Air at ambient room temperature was used.

I. INSTRUMENTATION

Temperatures were measured by use of chromel-alumel thermocouples. The sheathed thermocouples were long enough so that no terminations existed inside the CLEM cold wall. After the thermocouple penetrated the CLEM, plugs were attached to connect them to extension wires which ran to reference junctions. Copper wire was used between the reference junction and the data acquisition system. The data acquisition system had the capability to handle 100 inputs, and the outputs were stored on both magnetic tape and a printed paper tape for each temperature scan. The scan rate of the data acquisition system is 20 channels per second.

The flow rate of the air heat removal system was measured by use of an orifice plate installed downstream of the blower. The differential pressure created by the orifice was measured by use of a differential pressure gage. In addition to the orifice, a pitot tube, in conjunction with an inclined manometer, was used to measure the flow.

Electrical input power to the heaters was measured by digital voltmeters. A meter connected across the bus bars indicated the voltage, and a true RMS meter measured the voltage drop across a calibrated shunt to provide current indication.

J. SUPPORT FACILITIES

The test section was designed and built to be operated in the HTL-L loop enclosure in the 335 Building. (See Figure 10). The HTL-1 loop provided the source of sodium to fill the CLIRA-CLEM assembly. Cold-trapping was used to remove oxides in the sodium to a level of less than 10 ppm. After the assembly

Figure 9. Blower Assembly to Pass Air Over the CLEM Cold Wall

Figure 10. Overall View of the Entire CLIRA-CLEM Test Assembly Mounted in the HTL-1 Loop Enclosure

was filled with sodium, the CLIRA-CLEM was valved off from the HTL-1 loop and had no further sodium connection.

The electrical power supply for the heaters was located adjacent to the test section. The power supply has a maximum power rating of 100KW and is continuously variable on its output voltage from 0 to 144 volts DC. Operation of the heaters with one end grounded required the installation of a 1:1 isolation transformer prior to the rectifier section of the power supply. A 10-ampere fuse was also installed in each of the 19 heater leads to prevent a massive failure in case a single heater shorted out.

IV. DESCRIPTION OF TESTS

Test runs are summarized in Table 1. In the November tests (Run Nos. 1 through 11) emphasis was placed on the reference handling technique, and the sodium level was maintained below the top of the flow tube throughout the series. The sodium inside the solid wall flow tube was connected to the sodium contained in the annulus formed by the flow tube and integral pot only at the bottom, thus preventing total natural convection recirculation between the two chambers.

In the second, or January test series, temperature profiles and heat generation rates were investigated with the sodium level above the top of the flow tube (Run Nos. 12 through 16). Therefore, with an appropriate temperature (density) difference between the sodium inside and outside the flow tube, sodium could recirculate by rising inside the flow tube, overflowing, and then descending in the annulus between the integral pot and the flow tube. After the completion of tests with overflow, the sodium level was again lowered below the top of the flow tube and an attempt (Run No. 17) made to reproduce observations made in the earlier, or November, tests.

In general, facility startup was accomplished by raising the temperature of the sodium column to about 450° F with the external trace heating. The initial run of the day was made by turning off the trace heating, activating the internal 19-pin bundle, and adjusting the heat generation rate of the bundle to the desired value. Subsequent runs for the day were then made simply by re-adjusting the heat generation rate.

A. NOVEMBER TEST SERIES (EVALUATION OF REFERENCE HANDLING TECHNIQUE)

Testing was initiated on November 20, 1971, after a period of recirculation through the test facility via the HTL-T Loop to reduce oxygen concentration below 10 ppm. After lowering the liquid level below the top of the flow tube, the hot test facility (sodium temperature, 450 to 500° F) was isolated, by valving, from the loop. The trace heaters were then turned off and the heated fuel bundle adjusted to one KW for an initial shakedown run (Run No. 1). After a three-hour period without air cooling, the fan was turned on. Reduced

TABLE 1
TEST RUN SUMMARY

Run No.	Heat Gen (KW)	Date	Time at Power (Hr)	Sodium Level (above/below top of flow tube)	Cooling Air				Appendix I Figure No.
					Flow (on/off)	Inlet Damper (open/closed)	Inlet Temp. (° F)	Outlet Temp. (° F)	
1a	1	11/20/71	3.1	Below	Off	Open	--	--	A-2
1b	1	11/20-21-71	2.3	"	Partially on	"	--	--	A-3
2	5	11/22/71	2.0	"	Full (1020 CFM)	"	87	99	A-4
3	6	"	1.3	"	"	"	89	102	A-5
4	4	"	1.8	"	"	"	91	98	A-6
5	3	"	0.9	"	"	"	87	94	A-7
6	2	"	1.5	"	Partially on	"	--	--	A-8
7	1	"	2.0	"	Off	Closed	--	--	A-9
8	2	11/23/71	3.0	"	"	"	--	--	--
9	4	"	5.0	"	"	"	--	--	--
10	5	"	1.0	"	"	"	--	--	--
11	6	"	1.9	"	"	"	--	--	A-10
12	6	1/13/72	3.0	Above	Full (1020 CFM)	Open	69	85	A-11
13	7	"	1.0	"	"	"	70	88	--
14	8	"	2.6	"	"	"	65	90	A-12
15	3	"	3.3	"	"	"	67	75	A-13
16a	1	1/14/72	5.8	"	Off	Closed	--	--	A-14
16b	1	"	1.0	"	"	Open	--	--	--
16c	2	"	--	"	"	"	--	--	--
17	6	1/18/72	6.2	Below	Full (1020 CFM)	"	78	95	A-15

air flow rates were required, since extrapolation of data suggested the possibility of temperatures below 250° F near the top of the sodium column. Axial temperature gradients resulting in near freezing temperatures was the major problem uncovered during this initial test run.

The next set of runs (Run Nos. 2 through 7) were performed to determine the magnitude of the heat generation rate that would result in heating element temperatures at or near 1000° F. After heating to about 450° F, the trace heating was turned off and the heated fuel bundle adjusted to five KW. The maximum air cooling rate, about 1020 CFM, was being used within an hour. The test facility was held at power for an additional hour before equilibrium values were taken. After about 1 1/4 hours at six KW (to obtain additional data concerning temperature distributions) power was reduced to four, three, two and then one KW for about 2, 1, 1 1/2, and 2 hours, respectively. It was necessary to reduce the air cooling rate during the two KW run, again because of the possibility of freezing. Finally, on the following day, as a simulation of the loss of forced convection cooling, heat generation rates that would result in 1200° F heater temperatures and temperature profiles were investigated with the fan off (Run Nos. 7 through 11). The inlet air damper was closed during the final runs to prevent air circulation by natural convection.

B. JANUARY TEST SERIES

Attempts to restart the test facility in January, 1972 for testing with the sodium level above the top of the flow tube, (i.e., with possible convective recirculation between the chambers inside and outside the flow tube) resulted in the failure of five heating elements, Pin Nos. 53, 54, 56, 62 and 69: (see Figure 2). The reason for the failures is unknown. After removing the failed heating units from the electrical circuit and recirculating for sodium cleanup, the facility was again isolated from the loop and heated to about 450° F with trace heating.

At the time the facility was isolated, the sodium level was one to two inches above the top of the flow tube. The trace heaters were then turned off and the now 15-pin array adjusted to six KW, in an initial attempt to determine the magnitude of the heat generation rate that would result in heating element

temperatures at or near 1000° F. The assembly was held at six KW for about three hours before equilibrium temperatures were measured. Since the maximum temperature observed was only 857° F, power level was raised first to seven KW and then to eight KW. The latter power level was maintained for about 2 1/2 hours before equilibrium data were taken. Maximum temperature observed was 987° F. Power level was then reduced from eight to three KW and held at three KW for about three hours before additional equilibrium information was taken. Maximum air cooling rates were used throughout the above tests (Run Nos. 12 through 15). As shown in Table I, cooling air inlet temperatures were lower during the January test series.

On the following day (January 14), a preliminary evaluation of the power requirements needed to prevent freezing was also conducted while the sodium level was above the top of the flow tube. This test run (Run No. 16) was made without forced convection air cooling and using one KW of heat generation. Finally, on January 18, the sodium level was lowered below the top of the flow tube and a six KW run made to determine to what extent results from the earlier, November, tests could be reproduced.

V. TEST RESULTS

For tests using the maximum air cooling rate of 1020 CFM and a heating element cladding temperature limit of 1000°F, maximum heat generation rates attained during runs with and without flow tube overflow were eight and six KW, respectively. Tests with the sodium level below the top of the flow tube and heat generation rates greater than one to two KW resulted in an essentially linear, axial temperature gradient of about 5°F/inch in the region above the heated portion of the simulated fuel bundle. The gradient was observed in tests with the cooling air fan off, as well as in tests with the maximum air flow rate. Even flatter axial profiles were observed in tests with flow tube overflow. The region of maximum temperature shifted from the axial centerline toward the upper end of the 36-inch long heated length when the sodium level was raised from below to above the tip of the flow tube.

A. EQUILIBRIUM RESULTS -- AXIAL TEMPERATURE PROFILES

The results presented in this section of the report, to describe the types of axial temperature profiles observed, were taken under equilibrium conditions. More specifically, the term equilibrium is used to indicate that the net rate of temperature change at the hottest region had slowed to 5 to 10°F per hour or less. Axial temperature profiles observed during tests with and without flow tube overflow are shown in Figures 11 and 12, respectively. Test results are summarized in the form of a plot of "center-line" temperature versus distance from the bottom of the heated length. The "center-line" temperature plotted for a particular elevation is the arithmetic average of values for the three to four thermocouples installed either on the heating elements or the outer surface of the instrument tube assembly at that elevation (measurements taken at individual locations during equilibrium conditions are tabulated in Appendix A for several of the test runs).

1. Level Below Top of Flow Tube (See Figure 12)

As shown in Figure 12, in the reference, or November tests with the sodium level below the top of the flow tube, an essentially linear, axial temperature gradient of about 5°F per inch was observed in the region above the heated

Figure 11. Axial "center-Line" Temperature Profiles (Sodium Level Above Top of Flow Tube/Overflow)

Figure 12. Axial "Center-Line" Temperature Profiles (Sodium Level Below Top of Flow Tube/No Overflow)

portion of the simulated fuel bundle. This gradient was observed in tests with and without air cooling and heat generation rates ranging from three to six kilowatts.

a. Fan On

In tests with the maximum air cooling rate, the 1000°F heating element temperature limit was reached with a heat generation rate of six KW (Run No. 3). The maximum temperature observed (1015°F) was associated with the center pin (Pin No. 53) of the 19-pin bundle and the axial center of the heated length (see Appendix A, Figure A-5). Operation at six KW in the January test series, again with the maximum air cooling rate and the level below the top of the flow tube, reproduced the essentially linear gradient of about 5°F per inch in the region above the heated length (Run No. 17). However, the maximum temperature attained was only 950°F, and was associated with two of the failed pins, the central and an adjacent pin (Pin Nos. 53 and 54, see Figure 2 and Appendix A, Figure A-15). On the average, heating element, instrument tube, and integral pot temperatures attained during the January test were about 50°F lower than corresponding values reached during the November test. The complete reason for the difference is unknown. As shown in Table I, inlet cooling air temperature was only 11°F higher in the earlier test. Cold wall temperatures opposite the hot spot (axial center of the heated length) were above the $130 \pm 15^\circ\text{F}$ limit specified in the development test requirement specification⁽¹⁾ in both of the six KW tests (see Appendix A, Figures A-5 and A-15). The products of the cooling air ΔT (13 and 17°F); flow rate (72 lb/min); and heat capacity (0.24 BTU/lb-°F) are four and five KW, or 67 and 83% of the power input for the November and January test, respectively.

b. Fan Off

Review of Figure 12 will reveal that both the top and bottom curves are associated with test runs where the cooling air fan was off (Run Nos. 11 and 7). The top curve is a summary of the loss of forced convection cooling simulation (cooling air fan failure) where six KW was attained using a 1200°F temperature "limit" (Run No. 11). Both the cooling air inlet and outlet were blanked during this test. The maximum temperature observed, 1162°F, was associated with the axial center of the heated length and Pin No. 54 (see

Appendix A, Figure A-10). The maximum cold wall temperature observed was 680°F. As in the case of the tests with maximum air flow, the slope of the temperature profile above the heated length was about 5°F per inch.

The bottom curve is associated with one of the low power runs where the fan was turned off to prevent freezing. For this one KW run (Run No. 7) it appears that there is insufficient buoyancy-driven mixing to fully establish the linear gradient and (as will be discussed later) the shape of the curve is more representative of a conductive axial heat transfer mechanism. (Also see Appendix A, Figure A-9.)

2. Level Above Top of Flow Tube (See Figure 11)

Axial temperature profiles obtained with the sodium level above the top of the flow tube and while using the maximum air cooling rate are shown in Figure 11. Comparison of Figures 11 and 12 will show that even flatter axial temperature profiles were associated with tests with flow tube overflow. The slope of the temperature-distance curves shown in Figure 11, at an elevation 64 inches from the bottom of the heated length, is about 3°F per inch.

As shown in Figure 11, temperatures approaching 1000°F were attained with a heat generation rate of eight KW (Run No. 14). The maximum temperature observed (987°F) was associated with Pin No. 58 and was found at an elevation 28 inches from the bottom of the heated length (see Appendix A, Figure A-12). The product of the cooling air ΔT (25°F); flow rate (72 lb/min); and heat capacity (0.24 BTU/lb-°F) corresponds to 95% of the power input, or about 7-1/2 KW.

B. EQUILIBRIUM RESULTS -- RADIAL TEMPERATURE DIFFERENCES

Differences between the "center-line" and the average integral pot outer surface temperature are presented for seven elevations and tests with and without overflow in Table II. Measurements taken during the eight KW run with flow tube overflow (Run No. 14) indicate that, on the average, the maximum difference between the temperature of the sodium inside and outside the flow tube (at the same elevation) was about 50°F. In calculating this average ΔT , it was assumed that the minimum temperature of the sodium outside the flow tube (i.e., the sodium inside the annulus formed by the flow tube and the integral pot) was the same as the temperature of the outer surface of the pot.

TABLE II
RADIAL TEMPERATURE DIFFERENCE ("CENTER-LINE"-POT WALL)

Elevation above bottom of heated length (in.)	ΔT (°F)			
	6KW Run "Stagnant"(a) Run No. 3 Fig. A-5	6KW Run "Stagnant"(a) Run No. 17 Fig. A-14	6KW Run (b) Overflow Run No. 12 Fig. A-11	8KW Run (b) Overflow Run No. 14 Fig. A-12
18	63	49	42	51
28	47	53	68	87
48	50	37	54	69
64	57	45	52	59
78	33	36	37	44
108	8	11	18	23
138	-3	-28	5	31

(a) Sodium level below top of flow tube. Sodium in flow tube connected to sodium in annulus formed by flow tube and integral pot only at bottom; axial heat transport by buoyancy-driven mixing and conduction.

(b) Sodium level above top of flow tube. Sodium in two chambers connected at both ends; buoyancy-driven recirculation possible.

C. EVALUATION OF FREEZING

A limited evaluation of the power requirements needed to prevent freezing was conducted while the sodium level was above the top of the flow tube (Run Nos. 16a, 16b, and 16c). These test runs were initiated with the fan off and using a heat generation rate of only one KW. In addition, the cooling air inlet damper was closed and the exhaust ports plugged with insulation. Test results are presented in Figure 13 in the form of plots of "center-line" temperatures at selected elevations versus time. As shown in Figure 13, the inlet damper was opened and the insulation removed from the outlet ports after about six hours of transient operation. This action was taken because the time rate of change of temperature at the 138-inch elevation had slowed to the point where it appeared that freezing temperature would not be reached for many hours, if ever. As shown in Figure 13, opening the air passage to permit air flow by natural convection initiated a linear rate of temperature decrease (at the 138-inch elevation) of about 0.2°F per minute.

Figure 13. One KW Investigation of Freezing (Run No. 16, Sodium Level Below Top of Flow Tube, Fan Off)

D. MISCELLANEOUS RESULTS

Figure 14, included to give some idea of the response time of the massive test facility, summarizes selected results for air cooled tests with relatively high heat generation rates. Data are summarized in the form of plots of "center-line" temperatures at elevations of 18, 78, 108, and 138 inches versus time. The first of these elevations is, of course, the axial center of the heated length. Time zero is the approximate time the desired power level was reached (i.e., shortly after the trace heaters were turned off and the 19-pin bundle energized). As shown in Figure 14, for the initial run of the day, equilibrium conditions were attained in the heated region in about two hours.

Figure 14. Transient Data (Run Nos. 2, 3, 4 and 17)

VI. DISCUSSION OF TEST RESULTS

Analysis of the data from tests without overflow indicates that, in general, only about half of the heat transmitted from the surface of the integral pot was associated with the three-foot long heated length. Therefore, the remaining heat transmitted radially from the unheated pot surface was first transported axially upward, resulting in the linear temperature gradients described in the previous section. Review of the linear nature of these gradients and their slope suggests that, for tests with heat generation rates above one or two KW, axial heat flow was primarily the result of buoyancy driven mixing, i.e., a convective mechanism. In the tests with flow tube overflow, the observations that axial temperature gradients were even flatter and the region of maximum temperature was shifted upward or down stream, both indicate that buoyancy-driven recirculation was the dominant mechanism for axial heat transport. Calculations suggest that recirculation was established to the extent that about 65% of the heat transported radially from the integral pot was associated with the unheated length.

A. BUOYANCY-DRIVEN HEAT TRANSPORT

The fact that integral pot temperatures were maintained at relatively high values above the heated length suggests that convection, rather than conduction, was the overriding mechanism for axial heat flow. If heat were being transferred axially by conduction, axial exponential gradients similar to those observed in extended surfaces protruding from a heat source and losing heat to the surroundings by convection and radiation would be expected for the region above the heated length. More specifically, assuming an infinitely long cylinder or pin type fin, the temperature distribution would be approximated by (2)

$$T - T_s = (T_h - T_s)e^{-mx}, \text{ where} \quad [1]$$

T is the cylinder temperature at some axial distance x from the hot end;

T_s is the average temperature of the heat sink (surrounding cooling fluid, cold wall, etc.);

T_h is the temperature at the hot end;

m is a constant equal to $(4h_o/kD)^{1/2}$ for cylindrical geometry;

h_0 is the average combined unit-surface conductance (i.e., a value that accounts for the effects of heat flow by both convection and radiation between the surface of the cylinder and the heat sink);

k is the thermal conductivity of the cylinder material; and

D is the diameter of the cylinder

The type of gradient arising from the use of the conduction model, i.e., equation [1], is shown in Figure 15. As shown in this figure, temperature initially decreases and then tends to level off as distance from the hot end increases. For the geometry under consideration, the initial temperature drop is steep, about 30° F/in. , despite the relatively low value of h_0 used ($3 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$).

Also plotted in Figure 15 are the types of axial temperature gradients experimentally observed in tests with heat generation rates above one or two KW. It is immediately apparent that temperatures were maintained at relatively high values for both large and small values of x . This phenomenon, the fact that temperatures were maintained, suggests that a significant instability (mixing or circulation), initiated by density difference, was involved in the axial transport of heat.

1. Convective Mixing

The length of the linear portion of the temperature gradient observed in tests without flow tube overflow indicates that the convective mechanism was active over almost the entire unheated length. In fact, the linear temperature-distance relationships observed in tests without flow tube overflow can be derived from the assumption that only natural convection is involved in axial heat flow. More specifically, although the phenomenon is not the conventional one of transfer between a surface and fluid, under steady-state conditions, convective heat flow into the column above the heated length can be written as $hA'(T_h - T)$. This heat input can be equated to the heat lost by radiation and convection from the unheated length

$$hA'(T_h - T) = \pi h_0 D x (T - T_s) = q_c + q_r |_{\text{unheated}} \quad [2a]$$

FIG. 15

Figure 15. Axial Temperature Profile Calculated for Conduction Model

Rearranging the first two terms yields

[2b]

$$\frac{T_h - T}{x} = \frac{\pi h_o D (T - T_s)}{h A'}$$

In other words, the gradient, $(T_h - T)/x$, is equal to a constant and temperature is a linear function of distance (x). The h in the above equation is an average convection heat transfer coefficient based on the cross sectional area available for axial heat transfer, A' . As suggested above, h is not the conventional constant for transfer between a surface and a fluid, but an average convective heat transfer coefficient for the buoyancy driven mixing. As in the case of other Newtonian relationships for convection, the equation for axial heat transport by convection is simply a definition for h rather than a "...phenomenological law of convection..." for the buoyancy drive mass (heat) transport.⁽²⁾ Other terms associated with the above heat balance were defined earlier in this section of the report.

It should also be noted that for a column of liquid, heated from below, the threshold for convection is usually characterized by a critical Rayleigh number^(3, 4, 5). In other words, thermal or buoyancy-driven convection currents are established only after the critical Rayleigh number, the product of the Grashof and Prandtl numbers, is exceeded. For the CLIRA-CLEM geometry, the value calculated for the Rayleigh number, based on the unheated length and the axial gradient of 60° F per foot, is about 7×10^8 (see Appendix B, Section IV). Since convection was apparently well established, this value is higher than the critical value that would be associated with the onset of thermal instability.

2. Convective Recirculation

Tests with the sodium level above the top of the flow tube resulted in even flatter axial temperature gradients than those observed in tests without overflow. As shown in Figure 11, "center-line" temperatures at an elevation nine feet above the bottom of the heated length were 700 and 600° F for heat generation rates of eight and six KW, respectively. These temperatures are about 200 and 100° F higher than the "center-line" temperature observed at the nine-foot

level for the comparable six KW test without flow tube overflow. The obvious explanation for the more uniform axial distribution of heat (mass) is that buoyancy-driven recirculation was established during tests with flow tube overflow. The case for convective recirculation is supported by the fact that the region of maximum temperature was shifted from the axial center toward the upper or downstream end of the heated length when the sodium level was raised from below to above the top of the flow tube.

Review of the data, therefore, indicates that average radial temperature differences between sodium inside and outside the flow tube of about 40 to 50° F were sufficient to establish buoyancy-drive recirculation as the dominant mechanism for axial heat transport. As will be detailed later, in tests without overflow, calculations indicate that about half the heat transmitted radially from the surface of the integral pot was associated with the unheated length. In the case of flow tube overflow, recirculation was established to the extent that about 65% of the heat was transmitted from the unheated length.

B. HEAT TRANSFER COEFFICIENT CALCULATIONS

Radial temperature differences and axial temperature gradients observed were used to calculate values for h_o and h for five of the test runs (Run Nos. 5, 11, 12, 14 and 17). The calculations, admittedly oversimplified, were performed solely to provide orders of magnitude for the two heat transfer coefficients, and are summarized in Table III.

As indicated earlier, values for h_o account for the effects of heat flow from the surface of the integral pot by both convective and radiation, i.e.,

$$h_o = \frac{q_c + q_r}{A(T-T_s)} = \frac{q_c + q_r}{\pi D x (T-T_s)} \quad [3]$$

The convective heat flow rate, q_c , was calculated from the relationship

$$q_c = UA(T-T_s) = \pi U D x (T-T_s) \quad [4]$$

The overall coefficient, U , for convection heat transfer across the argon-filled annulus was estimated using the relation presented in Kreith⁽²⁾ for vertical enclosed air spaces:

TABLE III
CALCULATED VALUES - HEAT TRANSFER COEFFICIENTS
(Run Nos. 5, 11, 12, 14 and 17)(a)

Run No.	Heat Input (BTU/hr)	$q_c + q_r$ (BTU/hr)		h_o BTU/(hr)(ft ²)(°F)		h BTU/(hr)(ft ²)(°F)	V (ft/sec)	m (lb/hr)	Re	Appen. A Fig. No.
		Heated(b)	Unheated(c)	Heated(b)	Unheated(c)					
5(d)	10245	5456	6420	3.1	2.0	300	.005	--	--	A-7
11(e)	20490	9365	9438	8.1	5.6	400	.007	--	--	A-10
12(f)	20490	6572	10717	3.4	2.6	13400	.22	860	3300	A-11
14(g)	27320	8586	14778	4.0	3.0	14800	.25	950	4200	A-12
17(h)	20490	8080	9272	3.9	2.4	425	.007	--	--	A-15

- (a) See Appendix B for sample calculations.
- (b) 0 to 36 in. from bottom of heated length
- (c) 36 to 138 in. from bottom of heated length
- (d) 3 KW run without overflow, air flow rate 1020 cfm
- (e) 6 KW run without overflow, fan off
- (f) 6 KW run with overflow, air flow rate 1020 cfm
- (g) 8 KW run with overflow, air flow rate 1020 cfm
- (h) 6 KW run without overflow, air flow rate 1020 cfm

$$U = \frac{k}{b} 0.0317 Gr^{0.37}, \text{ where} \quad [5]$$

U is the over-all heat transfer coefficient;

k/b is the thermal resistance per unit area for pure conduction, the ratio of fluid thermal conductivity to the thickness of the gas layer; and

Gr is the Grashof number.

Radiant heat transfer per unit time, q_r , was calculated from the relationship

$$q_r = A \mathcal{F} \sigma (T^4 - T_s^4), \text{ where} \quad [6]$$

A is the surface area associated with a length x of the integral pot, $\pi D x$;

$\mathcal{F}$ is the gray-body shape factor, $1/[1/\epsilon_1 + (A_1/A_2)(1/\epsilon_2 - 1)]$;

σ is the Stefan-Boltzmann constant; and

ϵ is the emissivity.

The integral pot was divided into four lengths for the purpose of estimating heat flow rates from the surface (see Appendix B). This division was made to increase the accuracy of the radiant heat flow estimates which is based on non-linear considerations (temperature raised to the fourth power). The heat transferred by radiation was calculated assuming an emissivity of 0.6, since tar-nished (weathered) stainless steel surfaces were used in the fabrication of both the integral pot and the cold wall. A value of about 0.5 was calculated for the gray-body shape factor.

For the case where the sodium level was maintained below the top of the flow tube (i.e., where heat was transported axially by buoyancy-driven mixing) values for h were calculated from equation [2a] which, when rearranged, yields:

$$h = \frac{\pi h_o D x (T - T_s)}{A' (T_h - T)} \quad [2c]$$

A value of 60° F per foot was used for the temperature gradient, $(T_h - T)/x$, and a value of 0.04 ft² for transverse cross sectional area, A', the area normal to the axial length and associated with buoyancy-driven mixing. The latter value was used, since about 40% of the cross sectional area inside the integral pot is taken up with either the flow tube, hex-duct and the 19-pin assembly, or the flow tube and instrument tube assembly, depending on elevation.

For the case where the sodium level was maintained above the top of the flow tube (i.e., where heat was transported by buoyancy-driven recirculation) it was assumed that all heat lost from the unheated surface was first associated with the upflow of mass (heat) inside the flow tube followed by downflow in the annulus formed by the flow tube and integral pot. Therefore, the convective heat transfer coefficient for this case was defined as

$$h = \frac{q_c + q_r}{A'' \Delta T'} \text{ , where} \quad [7]$$

$\Delta T'$ is the driving force (in this case, the average temperature difference between the sodium inside and outside the flow tube); and

A'' is either the cross sectional area inside or outside the flow tube (a mean value of 0.02 ft^2 was used in calculating h).

Review of Table III will reveal that the heat transfer coefficients calculated for convective mixing (Equation No. 2c) are significantly lower than those calculated for convective recirculation (Equation No. 7). One of the main reasons for the difference is that the driving force for the mixing case is an order of magnitude higher than that for recirculation. For tests without overflow, as a result of the linear gradient of 5° F per inch, an overall ΔT of about 500° F is established in the long annular columns of sodium. In contrast, the driving force associated with the overflow case, the difference between the temperature of the sodium inside and outside the flow tube, is only about 50° F . In addition, the area term is a factor of two lower for the overflow case.

Finally, the values calculated for h can be related to velocity by relating the heat lost from the unheated length to the axial mass and heat transport within the sodium column. Symbolically,

$$q_c + q_r|_{\text{unheated}} = m C_p \Delta T = \rho A v C_p \Delta T = h A \Delta T. \quad [8]$$

Rearranging the last two terms yields

$$v = \frac{h}{\rho C_p} \quad [8]$$

The physical significance of the velocity calculated in the above manner cannot be totally defined, especially for the cases without flow tube overflow (no recirculation).

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APPENDIX A
TEMPERATURE MEASUREMENTS

(Run Nos. 1a, 1b, 2, 3, 4, 5, 6, 7, 11, 12, 14, 15, 16a and 17)

Figure A-1. Thermocouple Identification Chart.

Temperature (°F)

HTR	438	473	535	541	515		449	404	384		362		404
BUNDLE &			538	543	524		447	403	383	380	362	362	403
INST. TUBE			537	540	516		445	404	380		361		403
				540	522			404					

A-2

	468		529	488		452	396	381		357		373	
INT. POT	468	467	527	529	504	499	448	437	395	-	375	375	374
	469		524		509		435		382		374		372

COLD
WALL

176
172 190
180

203
199 210
206

268
280 264
259

DATE 11/20/71
TIME AT POWER 3.1 Hours
SODIUM LEVEL Below flow tube

POWER 1 KW
AIR FLOW None
AIR T_i --
AIR T_o --

Figure A-2. Temperature Measurements - Run No. 1a

Figure A-3. Temperature Measurements - Run No. 1b

Figure A-5. Temperature Measurements - Run No. 3

DISTANCE FROM BOTTOM OF HEATED LENGTH (IN.)

Temperature (°F)

	739	774	849	874	815	750	641	584		434		348
HTR			856	870	827	742	650	561 574		432 443		359 348
BUNDLE &			859	853	802	687	653	559		437		364
INST. TUBE				860	854							

A-6

	780	823	795	711	635	567		425		376
	781 777	816 822	814 779	695 679	641 -	560 549		431 425		374 376
INT. POT	781	819	791	667	566	496		428		374

2

	144	123	115
	135 163	121 139	112 121
COLD WALL	145	131	118

DATE 11/22/71
 TIME AT POWER 1.8 Hours
 SODIUM LEVEL Below flow tube

POWER 4 KW
 AIR FLOW 1020 CFM
 AIR T_i 91
 AIR T_o 98

Figure A-6. Temperature Measurements - Run No. 4

Figure A-7. Temperature Measurements - Run No. 5

DATE 11/22/71
 TIME AT POWER 1.5 Hours
 SODIUM LEVEL Below flow tube

POWER 2 KW
 AIR FLOW About 500 CFM
 AIR T_i 110
 AIR T_o 270

Figure A-8. Temperature Measurements - Run No. 6

DATE 11/22/71
 TIME AT POWER 2.0 Hours
 SODIUM LEVEL Below flow tube

POWER 1 KW
 AIR FLOW None
 AIR T_i --
 AIR T_o --

Figure A-9. Temperature Measurements - Run No. 7

DISTANCE FROM BOTTOM OF HEATED LENGTH (IN.)

Temperature (°F)

	783	791	819	849	836		771	713	678		593		558
HTR			822	852	857		774	701	659 668		587 591		556 561
BUNDLE &			828	842	834		725	689	650		583		558
INST. TUBE				833	839								

A-11

INT. POT

	791		804	779		712	683	657		577		555
	792 789		798 805	- 775		691 719	657 -	631 649		566 577		552 555
	792		800	767		691	608	569		564		551

COLD
WALL

129
120 147
130

119
117 138
129

120
117 127
124

DATE 1/13/72
TIME AT POWER 1.0 Hours
SODIUM LEVEL Above flow tube

POWER 6 KW
AIR FLOW 1020 CFM
AIR T_i 69
AIR T_o 85

Figure A-11. Temperature Measurements - Run No. 12

DATE 1/13/72
 TIME AT POWER 2.6 Hours
 SODIUM LEVEL Above flow tube

POWER 8 KW
 AIR FLOW 1020 CFM
 AIR T_i 65
 AIR T_o 90

Figure A-12. Temperature Measurements - Run No. 14

DATE 1/13/72
 TIME AT POWER 3.3 Hours
 SODIUM LEVEL Above flow tube

POWER 3 KW
 AIR FLOW 1020 CFM
 AIR T_i 67
 AIR T_o 75

Figure A-13. Temperature Measurements - Run No. 15

DATE 1/14/72
 TIME AT POWER 5.8 Hours
 SODIUM LEVEL Above flow tube

POWER 1 KW
 AIR FLOW None
 AIR T_i --
 AIR T_o --

Figure A-14. Temperature Measurements - Run No. 16a

Figure A-15. Temperature Measurements - Run No. 17

APPENDIX B
SAMPLE CALCULATIONS AND BACKGROUND INFORMATION
USED IN THE DEVELOPMENT OF TABLE III

I. CALCULATION OF h_o

A. ESTIMATE OF U FOR USE IN $q_c = UA(T-T_s)$

$$U = \frac{k}{b} 0.0317 Gr^{0.37} \text{ (Ref. 2)}$$

$$Gr = \frac{\rho^2 g \beta (\Delta T) b^3}{\mu}$$

For estimate of coefficient in argon gap in tests with fan on, use values for properties at 550°F and $\Delta T = 425^\circ F$.

$$\rho = 0.054 \text{ lb/ft}^3$$

$$\beta = \frac{\rho_s/\rho - 1}{T - T_s} = \frac{T/T_s - 1}{T - T_s} = \frac{1}{T_s} = \frac{1}{125+460}$$

$$b = 0.176 \text{ ft}$$

$$\mu = 24 \times 10^{-6} \text{ lb/ft-sec}$$

$$k = 0.016 \text{ BTU/hr-ft-}^\circ F$$

$$U_{550} = \left(\frac{0.016}{0.176} \right) 0.0317 \left[\frac{(0.054)^2 (32.2) (425) (.176)^3}{(585) (24 \times 10^{-6})^2} \right]^{0.37}$$

$$\approx 0.4 \text{ BTU/hr-ft}^2\text{-}^\circ F$$

For estimate of coefficient in argon gap in tests with fan off, use values for properties at 850°F and $\Delta T = 250^\circ F$.

$$U_{850} = \left(\frac{0.020}{.176} \right) 0.0317 \left[\frac{(0.042)^2 (32.2) (250) (.176)^3}{(1310) (30 \times 10^{-6})^2} \right]^{0.37}$$

$$\approx 0.2 \text{ BTU/hr-ft}^2\text{-}^\circ F$$

B. ESTIMATE OF $\mathcal{F}$ FOR USE IN $q_r = A \mathcal{F} \sigma (T^4 - T_s^4)$

$$A_1 \mathcal{F}_{1-2} = \left[\frac{1}{A_1} \left(\frac{1}{\epsilon_1} - 1 \right) + \frac{1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right) + \frac{1}{A_1 \bar{F}_{1-2}} \right]^{-1}$$

For cylindrical geometries involved, $\bar{F}_{1-2} = 1$

$$F_{1-2} = \frac{1}{\frac{1}{\epsilon_1} + \frac{D_1}{D_2} \left(\frac{1}{\epsilon_2} - 1 \right)}$$

Using value of 0.6 for ϵ_1 and ϵ_2

$$F_{1-2} = \frac{1}{\frac{1}{0.6} + \left(\frac{3.75}{7.98} \right) \left(\frac{1}{0.6} - 1 \right)} \cong 0.5$$

C. DETERMINATION OF T AND T_s (CALCULATION OF q_c AND q_r)

As shown in Figure B-1, average integral pot surface and cold wall temperatures were plotted against distance from the bottom of the heated length. Average values for three-foot lengths were then obtained and used in the equations presented in A and B, above, to calculate values for q_c and q_r . Detailed calculations are presented in Fig. B-1. The temperatures and calculations presented in this figure are for Run No. 17, a six-KW run with the maximum air cooling rate and the sodium level below the top of the flow tube.

D. h_o

Values for h_o were calculated for the heated length and three unheated lengths using equation [3] (see Figure B-1). Values presented in Table III of the text for the unheated length are arithmetic averages of values for zones 2, 3, and 4. In this case $(3.1 + 2.2 + 1.8)/3 = 2.4$ BTU/(hr)(ft²)(°F)

$$h_o = \frac{q_c + q_r}{A(T - T_s)} \quad [3]$$

II. CALCULATION OF h

As indicated in the text, for the runs without flow tube overflow, h was calculated using equation [2c] and $(T_h - T)/x$ and A were 60°F/ft and 0.04 ft², respectively. An arithmetic average of values for zones 2, 3, and 4 was used for $T - T_s$; in this case $(580 + 410 + 310)/3 = 433^\circ\text{F}$. Specifically,

$$h = \left[\frac{\pi h_o D}{A} \right] \left[\frac{T - T_s}{1} \right] \left[\frac{x}{T_h - T} \right] = \left[\frac{(\pi)(2.4)(.313)}{(.04)} \right] \left[\frac{433}{1} \right] \left[\frac{1}{60} \right] \quad [2c]$$

$$= 425 \text{ BTU/(hr)(ft}^2\text{)(}^\circ\text{F)}$$

Figure B-1. Average Pot Wall and Cold Wall Temperatures for Run No. 17
(for use in calculating q_r , q_c , h and h_o)

III. CALCULATION OF VELOCITY, V

Velocity was calculated from Equation [8]

$$V = \frac{h}{\rho C_p} = \frac{425}{(54)(.31)} = 25 \frac{\text{ft}}{\text{hr}} = 0.007 \frac{\text{ft}}{\text{sec}} \quad [8]$$

As indicated in the text, for tests without overflow, the physical significance for V is not totally defined.

IV. CALCULATION OF RAYLEIGH NUMBER

Rayleigh Number calculations were based on the following values:

$$g = 32.2 \text{ ft/sec}^2$$

$$\alpha = 1.3 \times 10^{-4} \text{ } ^\circ\text{F}^{-1}$$

$$A = 60^\circ\text{F/ft } (\Delta T/L)$$

$$L = 8.5 \text{ ft}$$

$$\nu = 3.3 \times 10^{-6} \text{ ft}^2/\text{sec } (\mu/\rho)$$

$$K = 6.9 \times 10^{-4} \text{ ft}^2/\text{sec } (k/\rho C_p)$$

$$\begin{aligned} \text{Rayleigh No.} &= Gr \cdot Pr = \left(\frac{\rho^2 g \alpha \Delta T L^3}{\mu^2} \right) \left(\frac{C_p \mu}{k} \right) = \frac{\rho^2 g \alpha A L^4 C}{\mu k} \\ &= \frac{g \alpha A L^4}{\nu K} \end{aligned}$$

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