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THERMAL EVALUATION FACILITY FOR LMFBR SPENT
FUEL TRANSPORT

David A. Wesley
Thermal Test and Analysis Division

TRANSPORTATION TECHNOLOGY CENTER

Sandia Laboratories
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Thermal Evaluation Facility for LMFBR Space Fuel Transport *
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Abstract

A fuel-coolant mock-up of a 17 pin breeder reactor fuel assembly in a cylindrical pipe was initially designed and constructed by Oak Ridge National Laboratory (ORNL). It was transferred to Sandia where it was extensively modified and modified.

The 17 pin horizontal core assembly was installed in a smaller diameter stainless steel pipe which was 10 feet long - one-half the diameter of a shipping container. A shipping container was used as the outer wall. Two-hundred feet of the mock-up was completely covered with an active length of 4 feet and the remaining 16 feet was insulated with 18 inches of insulation. The insulation was 18 inches thick and was installed in the shipping container. The mock-up was installed in the shipping container and was tested in the shipping container. The mock-up was tested in the shipping container and was tested in the shipping container.

The space provided in the natural convective region is operated with the liquid in the mock-up fluid. Liquid media can only be used in water fluids in further development. The space provided is operated with the pressure from a near vacuum to a pressure of several atmospheres. A variation of the mock-up assembly is provided from horizontal to vertical in position, and the position of the assembly within the enclosing pipe can be varied.

In addition to the natural convective mode, a forced convective mode is available. Air is utilized as the working fluid. The air can either be directed through the box core or be undirected to flow through the core as well as the annular region between the core and container.

*This work supported by the United States Department of Energy (DoE).

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THERMAL EVALUATION FACILITY FOR LMFBR SPEND FUEL TRANSPORT

INTRODUCTION

In August of 1976 the Thermal Test and Analysis Division of Sandia Laboratories received an experimental mockup of a single Atomics International LMFBR fuel assembly. The mockup consists of 217 fuel tubes surrounded by a hexagonal sheath. This core assembly was housed within a 10-in-diameter pipe enclosure (cover tube). The mockup was designed and fabricated by the Reactor Division at Oak Ridge National Laboratory. A complete description of the experimental assembly is reported in reference 1.

Sandia Laboratories obtained this assembly for the purpose of conducting heat transfer tests. These tests are expected to yield local and circumferentially averaged heat transfer coefficients at the surface of the hex wrapper and cover tube for various heat transport fluids. The heat transfer coefficients are necessary in order to evaluate a number of conceptual designs of spent fuel shipping casks (SFSC).

The ability to transfer heat from the fuel assemblies to the environment is a major factor in the design of an SFSC. The heat transport controls the type of coolant, containment pressure, canister design, loading methods, and ultimately the shipping capacity of the SFSC.

The heat transport tests which were conducted at ORNL with this mockup were of limited value in obtaining the desired heat transfer coefficients. Attempts to model heat transfer coefficients from the ORNL data resulted in the identification of many areas where further data and/or improved instrumentation were

needed. These areas are discussed in detail in reference 2. Most importantly, data are needed on the extent of axial spreading of the decay heat within the annulus and of the effect of non-coaxial assembly and pipe. The accuracy of the ORNL heater power measurements was also not well defined. It was necessary, therefore, to redesign or modify much of the experimental assembly in order to obtain better quantified test results.

Those areas which were redesigned or modified are:

1. thermal sensor instrumentation
2. heater power supply,
3. test piece support structure
4. fuel assembly cover tube
5. fluid pressure and vacuum system

Changes in these areas will greatly enhance the control flexibility and the thermal information which can be extracted from the experimental unit.

The nature of the above changes eliminated the possibility of conducting tests with a liquid metal working fluid without further modification. The working fluids that can be tested include gases, water, and heat transfer oils (e.g., Dowtherm A).

Thermal Sensor Instrumentation

The determination of the local heat transfer coefficients at the hex wrapper surface required local heat flux and temperature measurements. These measurements can also be used to obtain the circumferential average heat transfer coefficients. The radial temperature profiler within the hex core were also desired. The peak pin temperature was of prime concern. With these objectives in mind, the test piece was instrumented with a number of thermocouples, thermocouple probes, and heat flux gauges.

The hexagonal fuel rod wrapper of the mockup was initially fitted with five thermocouples. The number and placement of thermocouples was considered to be inadequate for the purpose of determining the required objectives. In addition, the original instrumentation did not include heat flux gauges.

Heat flux gauges were judged to be an essential part of the instrumentation. A thermopile type of heat flux gauge was chosen. This type of gauge consists basically of a thin insulating wafer that is fitted with a thermopile. The specific gauge selected is the RdF Corp. model 20456. The wafer material of this gauge is Kapton which permits the gauge to be operated continuously at a temperature of 310°C (590°F). The thickness of the gauge is only 4 mils. This characteristic helps eliminate any hydrodynamic disturbance in the annulus between the hex wrapper and the cover tube that may result from the presence of the gauges. The nominal sensitivity of these gauges is 3.5 mv/w/in^2 ($4 \text{ mv/Btu/ft}^2\text{sec}$).

The thermocouples, which are installed on both the hex wrapper and cover tube, are 30 AWG type KX (Special limits of error, chromel-alumel) with an HF/D Kapton fused tape insulation.

The expected temperature gradients for gaseous coolants are much larger in the circumferential as opposed to the axial direction. Therefore, the circumference of both the hex wrapper and

cover tube are heavily instrumented at four equally spaced axial locations. The first axial location was at the end of the tube bundle heated section - 295.9 cm (116.5 in) from the outer face of the tube sheet* (see Fig. 5 in reference 1). The remaining three instrumented cross sections are equally spaced at 60.96 cm (24.00 in) intervals in a direction toward the tube sheet. Because the heated section is 121.92 cm (48 in) in length, this spacing placed an instrumented cross section at the center and both ends of the heated section.

Six heat flux gauges and twelve thermocouples comprise the thermal instrumentation on the hex wrapper at each axial location. The heat flux gauges are located at the center of each hex wrapper flat. Each heat flux gauge is flanked by thermocouples mounted at the one-quarter and three-quarter locations of the hex wrapper flats.

The thermopile type heat flux gauges are glued to the hex wrapper and cover tube surfaces with AP cement which is marketed by the R&F Corp. AP cement is a high temperature polymer which requires heat and pressure for a proper cure.

An effort was made to minimize any hydrodynamic disturbance that could be caused by the heat flux gauge and thermocouple wires on the hex wrapper surface by imbedding these wires within long channels cut in the surface. The wires were staked down in the channels and then the channels were filled with a potting compound. The potting compound was composed of two parts of powdered aluminum and one part of Aremco-Seal 529 by weight. Aremco Seal 529 is a single component plastic-ceramic composition in a liquid form that is marketed by Aremco Products, Inc. This mixture provided a low porosity filler with a thermal conductivity near that of the stainless steel hex wrapper surface. Fig. 1 shows a set of wires staked in place and a second set with cured potting compound prior to filing, sanding, and cleaning.

* The tube sheet end represents the "grapple" end of an LMFB fuel assembly.

Figure 1. Method used in mounting thermal sensors to the
wall fire

The thermal radiation transmission characteristics of the Kapton heat flux gauges permits a significant amount of heat to pass through the wafer without being detected by the thermopile. To eliminate potential errors, a 1 mil thick piece of 304 stainless steel foil was cemented to the exposed face of the gauge. This procedure minimizes the problem of radiant heat transmission through the gauge and helps ensure that the thermal radiation properties of the gauge face are similar to those of the hex wrapper surface.

The emittance of the cover tube, hex wrapper, and gauge face foil were measured with a 100°C Gier Dunkle Infrared Reflectometer (See Reference 3). The hex wrapper emittance measurements varied over the surface from 0.17 to 0.26 with a mean emittance value of 0.16. In the regions of the hex wrapper surface that contained the potting compound stripes, the emittance over a 2.0 cm (0.79 in) diameter spot rose to range from 0.25 to 0.29. The emittance of the gauge face foil was 0.14. Thus, an average difference of 0.04 emittance units exists between the hex wrapper surface and the gauge face foil. It was not possible to obtain thermal radiation property measurements of the cover tube I.D. due to dimensional limitations. The mill finish of the tube O.D. and I.D. appeared to the eye to be equal. The emittance of the O.D. was measured at 0.33. The emittance of the I.D. was, therefore, estimated to be 0.33. After the O.D. of the cover tube was instrumented, a coating of Pyromark 2000 flat black paint was applied. The emittance of this surface was measured at 0.95.

The heat flux gauge and thermocouple wires that emerge from the hex wrapper surface 16.5 cm (6.5 in) from the axial instrumentation location, were lead radially outward through holes in the cover tube wall. Four pairs of thermocouple and two pairs of heat flux gauge wires were placed through a 0.64 cm

(0.25 in) O.D. by 3.8 cm (1.5 in) long section of 304 stainless steel tubing. The tube was filled with General Electric RTV 630 which is a high temperature and pressure two part silicone resin. A seal between the tube O.D. and cover tube was provided with 780-FSS 1/4 x 1/4 Imperial Eastman stainless steel thermocouple connectors which were screwed into the threaded holes of the cover tube wall.

Each axial station of the cover tube O.D. was instrumented in the following manner. Eight thermocouples were spaced at 45 degree increments around the circumference of the pipe. The first thermocouple of each axial set is located at the 12 o'clock position. A heat flux gauge is mounted mid-way between each pair of thermocouples on the circumference of the pipe. The heat flux gauge and thermocouple wires were situated along the cover tube axis. The hydrodynamic characteristics at the cover tube O.D. were judged to be less critical than those at the hex wrapper surface. In place of imbedding the wires in the surface of the cover tube, the wires were covered with 1 mil thick strips of 304 stainless steel which had a 60 mil wide by 60 mil deep channel formed within each strip. The channel portion of a strip was placed over a wire pair and the strip was spot welded in place. The installation of a strip over a thermocouple wire pair is shown in Fig. 2.

Thirteen tubes were selected in the tube bundle for temperature measurements: They are tube numbers 1, 20, 23, 26, 29, 32, 35, 92, 98, 104, 110, 116, 122 (See Reference 1 for the numbering sequence). Five measured tubes are situated on a line between opposite corners of the hex wrapper.

The temperatures of the core tubes are monitored by thirteen multilevel temperature profiling assemblies-part number T500095-142K8C-24-12. These were purchased from ARI Industries,

Figure 2 Method Used in Mounting Microfilm to the Cover Tube

Inc. Each assembly is made of a 350.7 cm (142 in) long by a 0.316 cm (1/3 in) diameter sheath containing five type K thermocouples. The junctions within the sheath are axially spaced at 60.96 cm (24.00 in) increments corresponding to the stations where heat flux gauges are located. Since the core tube I.D. is somewhat larger than the probe sheath O.D., small beryllium copper rings were placed on the probe sheath where the thermocouple junctions were located. These rings, which were approximately 145 mils long and 16 mils thick, were placed on the sheath to improve the thermal contact between the tube I.D. and the measuring junction. A cross sectional illustration of the hex core and cover tube instrumentation is shown in Fig. 3.

Following the initial testing, a traversing mechanism was designed which allows positioning of the probes at a number of axial locations to map the axial temperature gradients. This mechanism, which is illustrated in Fig. 4, permits the probes to be drawn through a 76.2 cm (30 inch) range as compared to the 60.96 cm (24 inch) spacing between thermocouple junctions. This over-lapping permits cross-checking thermocouple readings and, therefore, temperature equilibrium.

The mechanism is driven by an electric motor-lead screw device. The electronic motor control package clearly indicates the axial location of the probes at any instant. Also, the electronic nature of the driving mechanism permits traversing the probes at any test piece elevation.

Eight thermocouples were installed near the upper channels of the mockup support structure. These are used to measure the environment (air) temperature which is required to determine the heat transfer coefficients at the O.D. of the cover tube. A pair of thermocouples are situated axially at each of the four instrumented axial measurement stations. One thermocouple of each pair is located near the 4 o'clock position from the test

LEGEND:

- f = SURFACE THERMOCOUPLES
- + = CORE TUBE THERMOCOUPLES
- = HEAT FLUX GAGES

Fig. 3 Cross-Sectional Illustration of the Sensor Locations

Fig. 4 Inner Core Thermocouple Probe Transport Mechanism.

piece center line while the other is located near the 8 o'clock position. The mean temperature of each axial thermocouple pair is expected to yield an adequate representation of the environment temperature near the measurement station.

In addition to the instrumentation mentioned previously, six "L" shaped traversing thermocouple probes can be used. The set of six probes would be used at any one of the four instrumented axial stations. The probes, with the housing mounted external to the cover tube, would be used to measure the radial temperature profile from the center of a hex wrapper flat to the surface of the cover tube. The construction of such a probe is illustrated in Fig. 5. The figure does not show that the thermocouple tube telescopes from a 188 mil dia. down to a 40 mil dia. at the probe end. The traversing action of the probe is initiated by rotating the chimney of the mechanism. Twenty revolutions are required to move the probe 2.54 cm (1.00 in).

A total of 215 thermal data channels record information from the test mockup. Of these, 56 are heat flux while the remaining 159 are temperature. As noted previously, 24 heat flux gauges are mounted on the hex wrapper surface while the other 32 are affixed to the O.D. of the cover tube. The number of thermocouples attached to the sub-assemblies are as follows: 32 on the cover tube, 48 on the hex wrapper surface, 65 in the core tubes, 6 for the traversing probes, and 8 for the environment temperature.

Fig. 5 Cross Section of
the Traversing
Temperature Probe
Mechanism

The Data Acquisition System

The magnitude of the heat flux gage and thermocouple outputs placed some severe operational requirements on the data acquisition system. Some of these requirements include high accuracy, micro-volt resolution, and high common and normal mode rejection. Since most of the data runs are to be made in the steady state condition the speed of the device was not a major consideration.

The system selected was a 240 channel Autodata 9 data logger. The system was fitted with an alarm system, a digital input, and a serial interface. The alarm system allows unattended operation of the experimental assembly while it is coming to a steady state condition. The digital input provided a means of linking the YEW power meter to the data acquisition system which in turn would process and store the power data. The serial interface allows the data system to be connected to a computer terminal for processing and storing the data.

With the Autodata 9 operating in the hi-resolution mode, the resolution is 0.001% (1 microvolt), the normal-mode rejection is 80 dB, and the common mode rejection is 180 dB. The speed is 2.4 channels per second. The Autodata 9 contains a built in averaging option that allows one or several channels to be averaged over a selected time period.

The computer terminal linked to the Autodata 9 was a Hewlett Packard Model 2647A Intelligent Graphics terminal. This terminal offers up to 9600 baud rate, local intelligence for editing, single computer connection, dual tape cartridges, and easy programming of the Autodata 9. The computer is linked to a HP13246B Hewlett Packard Thermal Printer which supplies hard copies.

Heater Power Supply

The power supply designed and constructed by ORNL was not shipped to Sandia Laboratories. A new system which provides a great degree of flexibility was designed to the following requirements: the power of all of the heaters must be controlled through a single manual operation; the power level of each individual heater must be regulatable with respect to the mean total power level; the total power level must be measureable; and the power of each heater must be measureable.

The new power supply design is shown in Figure 6. Note that only three DPDT switches, rheostats, fuses, and heaters of the 218 total are shown. Thus, the circuit is, in part, a massive parallel system. The specific electrical components are the following:

- (a) A Sola Electric - Solatron line voltage regulator - model number 31-46-337 - maximum output of 37.5 kVA.
- (b) A Superior Electric - Powerstat variable transformer - model number 1256 D-4PS - maximum output of 31.4 kVA.
- (c) A 20 to 1 water cooled stepdown transformer - specially built by Goodrich Division of Conrac Corp. - maximum output of 31.4 kVA.
- (d) A Weston current transformer - model 327, type 2 - maximum Error of 0.11%.
- (e) A Yokogawa Electric Works, LTD - Type 2503 Digital AC Power Meter - model 250311 - maximum error of $\pm (0.1\% \text{ of rdg} + 0.02\% \text{ FS} + 1 \text{ digit})$.
- (f) Ohmite Mfg. Co. - Rheostat Model K - 100 watt - Type 0442 (0 - 2 ohms)

Various fuses and switches comprise the remaining electrical components.

Fig. 6 Electrical Schematic of Heater Power Supply

Since the nominal heater resistance is 5.76 ohms the 2 ohm rheostats give a nominal zero to 35% power adjustment level. The heaters may either be set at some mean power value or adjusted to some predetermined radial power profile.

The large water cooled stepdown transformer drops the maximum voltage from 560 v to 28 v. The voltage stepdown is accompanied by a proportional rise in current. This large current (max 1120 amps) necessitates the use of a large common cable. The cable represents the line in Fig. 3 where the heater wires, opposite the fuses, terminate. A 1000 mcm cable is used for this purpose.

The individual wires and cable, which are approximately 9 m (29 1/2 ft) long, extend from the power control board (Fig. 7) to a specially designed buss which is located adjacent to the test piece. The buss configuration, consisting of two large copper plates and a number of electrical strip type connectors is shown in Figure 8. The thirteen tube thermocouple probes are also shown attached to the buss supports.

Figure 7 Power Supply Control Board Photograph

Figure 8 Power Supply Buss and Lead Wiring Photograph

Fluid, Pressure, and Vacuum Systems

Many of the original ORNL fluid, pressure, and vacuum systems were retained in the modified mock-up. The primary difference between the ORNL and the redesigned systems is that the new systems incorporate units that allow the test piece to be operated in a forced convective mode. Also, the heaters that were attached to the various tanks which permitted operation with liquid metals were not used in the new system.

A schematic illustration of the new system is shown on Fig. 9.

The natural convective mode components consist of the following units:

- a) A redesigned surge tank complete with sight tube
- b) A Welch vacuum pump with a capacity of 17.7 cfm
- c) A 60 and 150 psi Ashcroft compound gauges with a rated accuracy of 1% of full scale
- d) A 30 in. Hg Ashcroft vacuum gauge with a rated accuracy of 1% of full scale
- e) An inert gas supply
- f) A redesigned fill and drain tank
- g) The various regulating valves were manufactured by Whitey.

The leakage rate of the test piece was such that the obtainable vacuum was limited to approximately 100 microns of mercury. If the working fluid is a gas rather than a liquid, the surge and fill/drain tanks are not used in the natural convective mode.

The forced convective mode components consist of the following units:

- a) A building air supply that is rated at 153 scfm.
- b) A large Ingersoll-Rand oil and water vapor trap.
- c) Two model 3000 Fairchild pressure regulators.

- d) A pair of air filters.
- e) A Fischer and Porter rotameter rated at 76.5 scfm and an accuracy of 2%.
- f) A J-Tech ultrasonic flow meter with a 20 to 1 turn-down ratio and a rated accuracy of 1%.
- g) A Mensor 0 to 60 psia pressure gauge with a 0.1% full scale rated accuracy.
- h) A vacuum valve, globe valve, and two gate valves.

The building air supply feeds into a large outside storage tank. The flow rate for long time periods of running would be limited to near 120 scfm. The large outside storage tank, however, permits short time flow rates up to 400 scfm. Hardware was designed and constructed which allows the flow to be directed through the core or left undirected. In the undirected flow mode the greatest portion of flow occupies the annular region between the hex and cover tube I.D. surfaces. This is due to a substantial difference in pressure drop between the core and the annular region.

Test Piece Support Structure

Many changes were made to the original test piece support structure. The structure that came from ORNL was fitted with liquid metal shielding. I was decided that, in the near term, liquid metals would not be used in the assembly, therefore, all of the shielding was removed.

The original 25.4 cm (10 in) dia. cover pipe was supported by rollers spaced equally along the length of the support structure. This construction was undesirable in that each roller represented a heat sink from the pipe to the support structure. In the redesigned support the flange end of the test piece was mounted securely to the end structure bulkhead. The opposite end of the test piece is supported by a single adjustable roller. The roller assembly has a secondary spring loaded roller on top of the cover tube required to keep the test piece seated on the primary roller when the assembly is elevated above the horizontal.

Other sections of the structure that required major modifications were the surge and fill-drain tanks. The testing of liquids requires an expansion or surge tank. The unit designed for the present tests includes a sight tube and a means of tilting the tank backwards through an angle of 45 degrees. When the test piece is oriented in a vertical position the axis of the surge tank will not exceed 45 degrees.

The fill-drain tank is situated in basically the same location as the ORNL tank. The tank dimensions and lack of surface heaters differ from the ORNL tank. Fig. 10 graphically illustrates the changes made, while Fig. 11 shows the final assembly when the construction was completed.

Fig. 10 Illustration of Changes Made to the Support Structure

Figure 11. Photograph of the Modified Support Structure

Fuel Assembly Cover Tube

The original ORNL cover tube, which represents the shipping canister, was replaced with a new unit. The major changes that were built into the new cover tube include diameter dimensional changes, flange changes, and the addition of various probe and gauge wire ports.

The initial 10 in. pipe exceeded the estimated diameter of an actual shipping canister or spent fuel shipping cask basket opening. Therefore, a new cover tube was constructed from an 8 in. - schedule 40 - 304 stainless steel pipe. The cover tube length as measured from the tube plate face (Ref. 1) was left unchanged from the original ORNL design.

A new end flange was designed which provides a capability for offsetting the center line of the hex with respect to the center line of the cover tube. This offset simulates the hex laying in the shipping canister. A flange pair, with eccentric inside diameters, provides the desired offset. The cross section of the final design is shown in Fig. 12. This design permits the center lines of the hex and cover tube to coincide or the hex can be offset such that two corners of the hex clear the inside of the cover tube by 110 mils. Five other offsets between these two extremes are possible. Also, these flanges have a twelve bolt hole pattern which maintains symmetry between the hex and the cover tube, i.e., it prevents two adjacent corners of the hex from laying at different distances from the cover tube inside surface. Fig. 13 is a photograph that shows the eccentric flange which is welded to the cover tube.

The cover tube contains a number of 1/4 and 3/8 NPT holes which are ports for the hex instrumentation wires and the traversing thermocouple probes. The end of the cover tube opposite the flange contains a 2 NPT hole which acts as a vacuum port or a forced air inlet port which will be used

Fig. 12 Cross-Section of Eccentric Flange Pair

Figure 13. Inside view of the Soviet atomic centrifuge.

for forced convection studies.

The O.D. of the cover tube was painted with pyromark 2000 flat black paint. The thermal emittance measurements (Ref. 2) indicated that the emittance was $.85 \pm .01$.

Local Heat Transfer Coefficients

The local heat transfer coefficients are to be determined directly, i.e., through the use of heat flux gauges. Hence, the heat transfer coefficient, h_T , at a specific location on the hex surface would be calculated from the output of a specific heat flux gauge, q , divided by the surface to reference temperature difference.

$$h_T = \frac{q}{T_w - T_b} \quad (1)$$

Since each heat flux gauge is flanked by thermocouples (T_1 and T_2) the surface temperature at the heat flux gauge can therefore be estimated as the mean value of the thermocouple pair, i.e.,

$$T_w = \frac{T_1 + T_2}{2} \quad (2)$$

The reference temperature, T_b , is defined as the mean value of the surface temperature, T_w , and the temperature, T_3 , of the cover tube I.D. at the location where a radial line which passes through the center of a heat flux gauge terminates at the I.D. surface of the cover tube.

$$T_b = \frac{T_w + T_3}{2} \quad (3)$$

For non-isothermal surfaces this definition may appear somewhat arbitrary. It is, however, adopted for lack of a better definition. Fig. 12 illustrates the various temperatures for a specific heat flux gauge.

The temperature of the cover tube I.D. is not measured directly. It must therefore be calculated from the temperature and heat flux fields at the tube O.D. The solution of the two-

LEGEND:

- j = SURFACE THERMOCOUPLES
- + = CORE TUBE THERMOCOUPLES
- = HEAT FLUX GAGES

Fig. 1- Temperature and Heat Flux Locations Used in
The Heat Transfer Coefficient Determination

dimensional heat conduction equation yields the temperature at any point, (r, θ) , in the cover tube wall.

$$T(r, \theta) = a_0 + \sum_{n=1}^{\infty} (r^n - R_1^{2n} r^{-n}) (a_n \cos n\theta + b_n \sin n\theta) + \sum_{m=1}^{\infty} (r^m + R_1^{2m} r^{-m}) (a_m \cos m\theta + b_m \sin m\theta) \quad (4)$$

where

$$a_0 = \frac{1}{2\pi} \int_{-\pi}^{\pi} f_0(\theta) d\theta \quad (4a)$$

$$a_n = -\frac{1}{2\pi k n R_1^{n-1}} \int_{-\pi}^{\pi} f_1(\theta) \cos n\theta d\theta \quad (4b)$$

$$b_n = -\frac{1}{2\pi k n R_1^{n-1}} \int_{-\pi}^{\pi} f_1(\theta) \sin n\theta d\theta \quad (4c)$$

$$a_m = \frac{1}{2\pi R_1^m} \int_{-\pi}^{\pi} f_0(\theta) \cos m\theta d\theta \quad (4d)$$

$$b_m = \frac{1}{2\pi R_1^m} \int_{-\pi}^{\pi} f_0(\theta) \sin m\theta d\theta \quad (4e)$$

Here k represents the thermal conductivity of the stainless steel cover tube wall while R_1 represents the radius of the outside wall. If the temperature is desired at the I.D. the variable r is replaced by the constant R_0 which is the I.D. cover tube radius. The functions $f_0(\theta)$ and $f_1(\theta)$ are the circumferential temperature and heat flux profiles respectively. These functions were determined by fitting a cubic spline through the

eight temperature and heat flux data at a specific axial location. For a complete derivation of equation 4 above see Appendix A.

It may be shown in passing that as long as the test piece remains in a horizontal position the heat fluxes at the eight heat flux gauge locations on the cover tube may be calculated approximately using only the eight temperature measurements. This results from the fact that for any realistic local cover tube temperature, the local Rayleigh number external to the cover tube always remains less than 10^8 . Since transition to turbulent flow generally occurs at Rayleigh numbers greater than 10^9 , the laminar flow and temperature fields have been calculated (see, for example, Ref. 4). These laminar results are usually presented for isothermal circular cylinders so a slight error is induced by using the results piece-wise for a non-isothermal circular cylinder. This effect is diminished since results of this type are usually correlated with the Rayleigh number to the one-fourth power, $\sqrt[4]{Ra_D}$.

The heat transport at the O.D. of the cover tube is a combination of radiant and convective contributions. Therefore, in addition to the preceding natural convective term, a radiant term of the form

$$q_{\text{rad}} = .85\sigma(T_s^4 - T_\infty^4) \quad (5)$$

must be included. The 0.85 emittance was nearly constant over the entire Pyromark 2000 flat black coated cylindrical surface. The surface temperature, T_s , and the environment temperature, T_∞ , are absolute temperatures. The symbol σ is the Stefan-Boltzmann constant.

The heat flux gauges at the hex surface yields an output that represents the combined radiant and convective contributions.

The heat transfer coefficient, h_T , in equation (1) represents, therefore, the coefficient of the total (radiant and convective) heat exchange.

If the heat flux data are taken directly from the gauges an error up to an estimated 10% can result. This error is caused primarily by a divergence of the heat flux field in the hex stainless steel sheath around the gauge. The analysis used to correct these gauge results is too complex to address in any depth here. It is, however, discussed in Appendix B.

Thermal Radiation Exchange

Determination of the natural convective coefficient, h , from the "total" heat transfer coefficient defined in equation (1) requires the thermal radiation heat flux contribution, q_{rad} . With the aid of equation (1), the natural convective coefficient may be defined as follows:

$$h = \frac{q - q_{\text{rad}}}{T_w - T_b} \quad (6)$$

The thermal radiation heat flux can be calculated from a thermal enclosure model which is illustrated in Fig. 15. The model approximates the circumferentially varying surface temperatures of the hex and cover tube with 30 isothermal surface elements. The 30 element enclosure requires the calculation of 900 configuration factors. Fortunately, due to the hex-cover tube geometry, many of the factors are either zero or identical to other non-zero factors. Known input to the radiation analysis requires the emittance, absorptance, and temperature of each of the 30 surface elements. Estimates of the thermal radiation properties can be obtained from the measurements that were addressed in the "Thermal Sensor Instrumentation" section. These measurements were obtained by assuming that the participating surfaces are gray, i.e., the emittance, ϵ , is equal to the absorptance, α . With the stainless steel hex and cover tube operating below the 500°F limit, the gray body approximation should induce little error.

The surface elements of the enclosure model exceed the number of measured temperatures. The temperature of the twelve I.D. cover tube elements can be obtained from the heat conduction solution which was addressed in the "Local Heat Transfer Coefficients" section. The temperature of the eighteen remaining

LEGEND:

- 1 = SURFACE THERMOCOUPLES
- + = CORE TUBE THERMOCOUPLES
- = HEAT FLUX GAGES

Fig. 15 Illustration of the 30 Surface Radiation Enclosure Model

box elements can either be estimated directly from temperature measurements or from the mean of a pair of temperature measurements.

The radiation transport analysis follows directly from the presentation in Ref. (5). Briefly, the analysis proceeds as follows:

$$\sum_{j=1}^{30} X_{ij} B_j = \sigma T_i^4 \quad (7)$$

where

$$X_{ij} = \delta_{ij} - \left(\frac{1 - \epsilon_i}{\epsilon_i} \right) F_{A_i - A_j} \quad (7a)$$

Here B_j is the radiosity of the j^{th} surface element, δ_{ij} is the Kronecker delta, and $F_{A_i - A_j}$ is the configuration factor between surface elements A_i and A_j . If the matrix inverse is taken of X_{ij} , a second matrix ψ_{ij} is obtained, i.e.,

$$\psi = X^{-1} \quad (8)$$

The heat flux at the various surface elements can now be given as

$$q_{\text{rad}_i} = \sum_{j=1}^{30} Z_{ij} \sigma T_j^4 \quad (9)$$

$1 \leq i \leq 30$

where

$$Z_{ij} = \frac{\epsilon_i}{1 - \epsilon_i} (\delta_{ij} - \psi_{ij}) \quad (9a)$$

The results obtained from equation (9) together with equation (6) is sufficient to estimate the local convective heat transfer coefficient, h .

Average Circumferential Heat Transfer Coefficient

The local heat transfer coefficients determined by the previous method are highly dependent upon the proper functioning of each individual thermal sensor. The malfunctioning of a specific sensor would grossly effect the outcome of the cubic spline curve fits through the data and hence the outcome of the heat transfer results. A different means of obtaining the results, therefore, would provide insurance of the validity of the results. Although no apparent means of obtaining the local heat transfer coefficients exists apart from the before mentioned method, the axially local-circumferentially average heat transfer coefficients may be obtained with another method.

Consider the following heat balance applied to the hex core.

Consider the finite element of length L of the hex core, and the axial heat transport into and out of the element, i.e., Q_{in} and Q_{out} . With the experimental assembly operating in the horizontal, natural convective mode, any axial convection contribution would probably not be significant. Also, due to the densely packed nature of the fuel rods within the hex core, any axial radiation heat transport is assumed to be small. The axial heat transport, therefore, would be largely conduction. By Fourier's

law of heat conduction Q_{in} and Q_{out} may be represented as:

$$Q_{in} = -Ak \left. \frac{dT}{dx} \right|_x \quad (10)$$

and

$$Q_{out} = -Ak \left. \frac{dT}{dx} \right|_{x+L} \quad (11)$$

The core is composed of a number of different materials. The area-thermal conductivity terms in the above equations are really the sum of the specific area-thermal conductivity contributions of the various materials within the core, i.e.,

$$Ak = \sum_{i=1}^n A_i k_i \quad (12)$$

The remaining terms in the heat balance are easily defined as:

Q_{prod} is the product of the volumetric heat production by the core heaters and the element volume, Q_{lost} is the heat that is lost by convection and radiation from the surface of the hex.

$$Q_{in} + Q_{prod} = Q_{lost} + Q_{out} \quad (13)$$

Substitution of equations 10 and 11 into equation 13 results in:

$$Q_{lost} = Ak \left. \frac{dT}{dx} \right|_{x+L} - Ak \left. \frac{dT}{dx} \right|_x + Q_{prod} \quad (14)$$

The first order temperature derivatives in equation 14 can be evaluated from the temperature field within the core which is supplied by the 13 core temperature probes and the probe traversing device. A total of 780 individual core temperatures exist from the

output of one data run. At any single core cross-section 13 temperatures are output. Since sizeable cross-sectional temperature gradients are expected to occur, a specific region of the cross-section was assigned to a specific thermocouple value which is located near the centroid of a region. These regions and thermocouple locations are shown in Fig. 16. Any variations in the thermal conductivity with respect to temperature would be taken into account with this method. Since the regions shown in Fig. 16 are of different areas, some of the measured temperatures will be weighted more heavily. This single drawback appears small in comparison to the advantages of this method. A cubic spline fit of the axial temperature data for each specific thermocouple probe is obtained for the evaluation of the first order temperature derivatives. Therefore, in terms of the 13 probes, equations 10 and 11 with the aid of equation 12 may be written as

$$Q_{in} = - \sum_{j=1}^{13} \sum_{i=1}^n (A_i k_i)_j \left\{ \frac{dT}{dx} \right\}_j \bigg|_x \quad (15)$$

and

$$Q_{out} = - \sum_{j=1}^{13} \sum_{i=1}^n (A_i k_i)_j \left\{ \frac{dT}{dx} \right\}_j \bigg|_{x+L} \quad (16)$$

The number of different materials in any given cross-section may be as large as five, i.e., $n=5$. The number and type of materials differ along the length of the core.

Equation 14 may now be rewritten as:

$$Q_{lost} = \sum_{j=1}^{13} \sum_{i=1}^n (A_i k_i)_j \left\{ \frac{dT}{dx} \right\}_j \bigg|_{x+L} - \sum_{j=1}^{13} \sum_{i=1}^n (A_i k_i)_j \left\{ \frac{dT}{dx} \right\}_j \bigg|_x + Q_{prod} \quad (17)$$

LEGEND:

- 1 = SURFACE THERMOCOUPLES
- + = CORE TUBE THERMOCOUPLES
- = HEAT FLUX GAGES

Fig. 16 Inner Core Area and Thermocouple Probe Locations

The terms Q_{lost} and Q_{prod} may also be rewritten as:

$$Q_{lost} = CL q_{lost} \quad (18)$$

and

$$Q_{prod} = \dot{q}AL \quad (19)$$

Where A is the total cross-sectional area of the hex element, C is the circumference of the hex, and $\dot{q}$ is the volumetric rate of heat production within the core. Since the hex element has a finite length, L , the heat flux, q_{lost} , is not in the strict sense an axially local circumferentially average value. The length, L , however is chosen to be 10.16 cm (4 inch) which is small enough compared with the total core length for q_{lost} to be considered to be essentially an axial local value.

The heat flux, q_{lost} , is composed of both radiation and natural convection components. Thus, with the circumferential average wall and bulk temperatures, T_w^{avg} and T_b^{avg} , the total heat transfer coefficient h_T^{avg} may be calculated from

$$h_T^{avg} = \frac{q_{lost}}{T_w^{avg} - T_b^{avg}} \quad (20)$$

The separate contributions of radiation and convection may be obtained with the aid of the radiation enclosure model which was presented in the Thermal Radiation Exchange section.

The preceding method is attractive since the heat transfer coefficients may be obtained using temperature data only. The heat transfer coefficients are also obtained without the use of any data used to obtain the local heat transfer coefficients addressed in the previous sections. The agreement, therefore, between the two sets of values would establish confidence in the results as a whole.

Concluding Remarks

Fig. 17 is a photograph of the completed full scale mockup assembly in a horizontal operational elevation. The photograph was taken prior to relocating the assembly to permit vertical elevation. With the assembly in the initial location a series of near vacuum ($\sim 90 \mu\text{m Hg}$) and helium (max of 0.1 atm) natural convective data runs were made.

Fig. 18 is a photograph of the mockup assembly with the unit elevated to a vertical position for a forced convective data run.

Figure 17 Completed Assembly Prior to Being Moved
to the New Location

Figure 15 Photograph of the Lock-1 in a Vertical Elevation

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Appendix A

Cover Tube Heat Conduction Solution

The two-dimensional heat conduction solution for the cover tube can be solved in closed form and numerical finite difference/element methods are not needed. The only difficulty that is encountered in solving this problem with the classical separation of variables technique is the two non-homogeneous boundary conditions at the O.D. of the tube. Specifically, these conditions arise from the temperature and heat flux profiles both being specified at this boundary, i.e.,

$$T(R_1, \theta) = f_0(\theta) \quad (1a)$$

$$-k \frac{\partial T}{\partial R}(R_1, \theta) = f_1(\theta) \quad (1b)$$

Here T is temperature and k is the average thermal conductivity of the pipe wall. The two remaining boundary conditions are homogeneous, i.e.,

$$T(r, \pi) = T(r, -\pi) \quad (2a)$$

$$\frac{\partial T}{\partial \theta}(r, \pi) = \frac{\partial T}{\partial \theta}(r, -\pi) \quad (2b)$$

Since the heat conduction equation is linear, the problem involving the non-homogeneous conditions at the O.D. may be solved by the superposition of solutions, i.e.,

$$T(r, \theta) = T_1(r, \theta) + T_2(r, \theta) \quad (3)$$

The superposition principle allows the original problem to be split into two separate problems. The solution to the original problem is, as Equation (3) states, the sum of the two separate

solutions. The two separate problems can be written as follows:

$$\frac{\partial^2 T_1}{\partial r^2} + \frac{1}{r} \frac{\partial T_1}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_1}{\partial \theta^2} = 0 \quad (4a)$$

$$T_1(R_1, \theta) = 0 \quad (4b)$$

$$-k \frac{\partial T_1}{\partial r}(R_1, \theta) = f_1(\theta) \quad (4c)$$

$$T_1(r, \pi) = T_1(r, -\pi) \quad (4d)$$

$$\frac{\partial T_1}{\partial \theta}(r, \pi) = \frac{\partial T_1}{\partial \theta}(r, -\pi) \quad (4e)$$

and

$$\frac{\partial^2 T_2}{\partial r^2} + \frac{1}{r} \frac{\partial T_2}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_2}{\partial \theta^2} = 0 \quad (5a)$$

$$T_2(R_1, \theta) = f_0(\theta) \quad (5b)$$

$$-k \frac{\partial T_2}{\partial r}(R_1, \theta) = 0 \quad (5c)$$

$$T_2(r, \pi) = T_2(r, -\pi) \quad (5d)$$

$$\frac{\partial T_2}{\partial \theta}(r, \pi) = \frac{\partial T_2}{\partial \theta}(r, -\pi) \quad (5e)$$

The solution to Equation (4a) is

$$T_1(r, \theta) = \sum_{n=1}^{\infty} (r^n - R_1^{2n} r^{-n}) (a_n \cos n\theta + b_n \sin n\theta) \quad (6)$$

where the Fourier coefficients a_n and b_n are given by

$$a_n = - \frac{1}{2\pi k n R_1^{n-1}} \int_{-\pi}^{\pi} f_1(\theta) \cos n\theta d\theta \quad (7a)$$

$n = 1, 2, 3, \dots$

and

$$b_n = - \frac{1}{2\pi k n R_1^{n-1}} \int_{-\pi}^{\pi} f_1(\theta) \sin n\theta d\theta \quad (7b)$$

$n = 1, 2, 3, \dots$

The solution to Equation (5a) is

$$\varphi_2(r, \theta) = a_0 + \sum_{m=1}^{\infty} (r^m + R_1^{2m} r^{-m}) (a_m \cos m\theta + b_m \sin m\theta) \quad (8)$$

where the Fourier coefficients a_0 , a_m and b_m are given by

$$a_0 = \frac{1}{2\pi} \int_{-\pi}^{\pi} f_0(\theta) d\theta \quad (9a)$$

$$a_m = \frac{1}{2\pi R_1^m} \int_{-\pi}^{\pi} f_0(\theta) \cos m\theta d\theta \quad (9b)$$

$m = 1, 2, 3, \dots$

$$b_m = \frac{1}{2\pi R_1^m} \int_{-\pi}^{\pi} f_0(\theta) \sin m\theta d\theta \quad (9c)$$

$m = 1, 2, 3, \dots$

The solution to the original problem is, therefore, the sum of equations (6) and (8) with the Fourier coefficients defined by Equations (7) and (9).

Appendix B

Heat Flux Gauge Measurement Errors

The additional thermal resistance of a thin disk affixed to a plate causes a local temperature increase and a divergence of the heat flux in the plate. The result is that the magnitude of the mean heat flux at the interface between the gauge and plate is less than the constant heat flux at the opposite face of the plate. This effect causes the gauge reading to be lower and therefore not indicative of the heat flux that is convected from the surface of the well-turbulent plate.

Differences between the thermal operating conditions and those existing during calibration will typically cause a change in the proportionality between the gage output and the heat flow through the gauge. Factors that cause this change are:

- (a) temperature dependence of the thermal conductivity of the gauge;
- (b) temperature dependence of the Seebeck coefficient associated with the thermopile;
- (c) significant departure from an axial one-dimensional heat flow through the disk.

If the heat flow through the disk is assumed to be one-dimensional then the convective heat transfer resistance and the thermal resistance can be lumped together. The result is an effective convective heat transfer coefficient h_e ,

$$h_e = \left(\frac{1}{h} + \frac{t_D}{k_D} \right)^{-1} \quad (1)$$

which applies to a disk sized area on the plate surface. Here t_D and k_D are the disk thickness and thermal conductivity respectively.

The method which was used to determine the temperature field within the plate is based on the observation that the classical separation of variables approach will yield solution representations for each of two regions of the plate. This solution method is believed to offer computational advantages over numerical finite-difference approaches.

The number of independent parameters characteristic of this problem are minimized by introducing the dimensionless quantities

$$\bar{z} = \frac{z}{t}, \quad \bar{r} = \frac{r}{a}, \quad \bar{b} = \frac{b}{a} \quad (1)$$

$$Bi = \frac{ht}{k}, \quad Bi_e = \frac{h_e t}{k} \quad (2)$$

$$\bar{T}_I = \frac{T_I - T_\infty}{T_w - T_\infty}, \quad \bar{T}_{II} = \frac{T_{II} - T_\infty}{T_w - T_\infty} \quad (3)$$

where T_w is the spatially uniform convective surface temperature in the absence of the disk. In the limiting case where b approaches infinity, T_w serves to specify the convective surface temperature at large $\bar{r}$. This temperature can be represented in terms of other thermal parameters as

$$T_w = \frac{w}{h} + T_\infty \quad (3)$$

The governing partial differential equations for regions I and II cannot be readily solved by the separation of variables method due to the nonhomogeneous spatially constant heat flux boundary at the plate surface. This obstacle can be removed through the use of the principle of superposition Bi . The superposition principle permits the temperature representation in each region to be expressed as the sum of the axial linear and spatially nonuniform temperature contributions.

$$\bar{T}_I = \theta_1 + Bi(1 - \bar{z}) + \frac{Bi}{Bi_e} \quad (4a)$$

$$\bar{T}_{II} = \theta_2 + Bi(1 - \bar{z}) + 1 \quad (4b)$$

Here θ_1 and θ_2 represent the dimensionless nonuniform contributions of region I and II respectively. The governing equations together with the resulting homogeneous boundary conditions expressed in terms of the variable θ appear as follows.

Region I

$$\frac{\partial^2 \theta_1}{\partial \bar{r}^2} + \frac{1}{\bar{r}} \frac{\partial \theta_1}{\partial \bar{r}} + \left(\frac{a}{t}\right)^2 \frac{\partial^2 \theta_1}{\partial \bar{z}^2} = 0 \quad (5a)$$

$$\theta_1(0, \bar{z}) \text{ is bounded} \quad (5b)$$

$$\frac{\partial \theta_1}{\partial \bar{z}}(\bar{r}, 0) = 0 \quad (5c)$$

$$\frac{\partial \theta_1}{\partial \bar{z}}(\bar{r}, 1) = -Bi_e \theta_1(\bar{r}, 1) \quad (5d)$$

Region II

$$\frac{\partial^2 \theta_2}{\partial \bar{r}^2} + \frac{1}{\bar{r}} \frac{\partial \theta_2}{\partial \bar{r}} + \left(\frac{a}{t}\right)^2 \frac{\partial^2 \theta_2}{\partial \bar{z}^2} = 0 \quad (6a)$$

$$\frac{\partial \theta_2}{\partial \bar{r}}(5, \bar{z}) = 0 \quad (6b)$$

$$\frac{\partial \theta_2}{\partial \bar{z}}(\bar{r}, 0) = 0 \quad (6c)$$

$$\frac{\partial \theta_2}{\partial \bar{z}}(\bar{r}, 1) = -Bi \theta_2(\bar{r}, 1) \quad (6d)$$

The boundary condition (6b) applies when the dimensionless radius $\bar{b}$ is finite. In the limit as $\bar{b}$ approaches infinity, equation (6b) is replaced with

$$\lim_{\bar{b} \rightarrow \infty} \theta_2(\bar{b}, \bar{z}) \text{ is bounded} \quad (7)$$

The two remaining relations that couple regions I and II can be formulated by noting that the temperature and the heat flux must be continuous at the interface between the two regions.

$$\theta_1(1, \bar{z}) + \frac{Bi}{Bi_e} = \theta_2(1, \bar{z}) + 1 \quad (8a)$$

$$\frac{\partial \theta_1}{\partial \bar{r}}(1, \bar{z}) = \frac{\partial \theta_2}{\partial \bar{r}}(1, \bar{z}) \quad (8b)$$

The solution of equations (5a) and (6a) by the separation of variables method yields the nonuniform temperature contributions for regions I and II.

$$\theta_1(\bar{r}, \bar{z}) = \sum_{n=1}^{\infty} A_n I_0(\lambda_n \bar{r}) \cos\left(\frac{t \lambda_n \bar{z}}{a}\right) \quad (9)$$

$$\theta_2(\bar{r}, \bar{z}) = \sum_{n=1}^{\infty} B_n \left\{ I_0(\delta_n \bar{r}) + \frac{I_1(\delta_n b)}{K_1(\delta_n b)} K_0(\delta_n \bar{r}) \right\} \cos\left(\frac{t \delta_n \bar{z}}{a}\right) \quad (10a)$$

Equation (9) remains unchanged when the radius b is taken in the limit as b approaches infinity. The solution representation of region II, however, changes to

$$\lim_{b \rightarrow \infty} \theta_2(\bar{r}, \bar{z}) = \sum_{n=1}^{\infty} B_n K_0(\delta_n \bar{r}) \cos\left(\frac{t \delta_n \bar{z}}{a}\right) \quad (10b)$$

The symbols I_x and K_x denote modified Bessel functions of the first and second kinds respectively. The symbols λ_n and δ_n represent eigenvalues which are defined by the following equations

$$\frac{t \lambda_n}{a} \tan\left(\frac{t \lambda_n}{a}\right) = Bi_e \quad (11)$$

$$n = 1, 2, 3, \dots$$

$$\frac{t \delta_n}{a} \tan\left(\frac{t \delta_n}{a}\right) = Bi \quad (12)$$

$$n = 1, 2, 3, \dots$$

The eigenvalues for the present investigation were determined by solving equations (11) and (12) with a numerical iteration scheme.

The A_n and B_n coefficients that appear in equations (9) and (10) have yet to be determined. The first step in this evaluation procedure is the substitution of the θ_1 and θ_2 parameters into equation (8a). By applying orthogonality to this equality, an expression for the A_n coefficients in

terms of the B_m coefficients will be obtained.

$$A_n = D_n + \sum_{m=1}^{\infty} B_m E_{mn} \quad (13)$$

Here, D_n and E_{mn} are defined as

$$D_n = \frac{\left\{ 1 - \frac{B_1}{B_1 e} \left\{ \frac{a}{\lambda_n t} \sin\left(\frac{\lambda_n t}{a}\right) \right\} \right\}}{I_0(\lambda_n) \left\{ \frac{1}{2} + \frac{a}{4t\lambda_n} \sin\left(\frac{2t\lambda_n}{a}\right) \right\}} \quad (14)$$

$$E_{mn} = \frac{\frac{a}{2t} \left\{ I_0(\delta_m) + I_1(\delta_m \bar{v}) K_0(\delta_m) \right\} \left\{ \frac{\sin\left[\frac{t}{a}(\lambda_n - \delta_m)\right]}{(\lambda_n - \delta_m)} + \frac{\sin\left[\frac{t}{a}(\lambda_n + \delta_m)\right]}{(\lambda_n + \delta_m)} \right\}}{I_0(\lambda_n) \left\{ \frac{1}{2} + \frac{a}{4t\lambda_n} \sin\left(\frac{2t\lambda_n}{a}\right) \right\}} \quad (15a)$$

When $\bar{v}$ is taken in the limit as $\bar{v}$ approaches infinity the E_{mn} expression becomes

$$\lim_{\bar{v} \rightarrow \infty} E_{mn} = \frac{\frac{a}{2t} K_0(\delta_m) \left\{ \frac{\sin\left[\frac{t}{a}(\lambda_n - \delta_m)\right]}{(\lambda_n - \delta_m)} + \frac{\sin\left[\frac{t}{a}(\lambda_n + \delta_m)\right]}{(\lambda_n + \delta_m)} \right\}}{I_0(\lambda_n) \left\{ \frac{1}{2} + \frac{a}{4t\lambda_n} \sin\left(\frac{2t\lambda_n}{a}\right) \right\}} \quad (15b)$$

Similarly, by substituting θ_1 and θ_2 into equation (8b) and applying orthogonality results in an expression for the B_m coefficients in terms of the A_n coefficients.

$$B_m = \sum_{k=1}^{\infty} A_k F_{km} \quad (16)$$

When the dimensionless radius $\bar{b}$ is finite, F_{km} is defined by

$$F_{km} = \frac{\lambda_k I_1(\lambda_k) \frac{a}{2t} \left\{ \frac{\sin \left[\frac{t}{a} (\lambda_k - \delta_m) \right]}{(\lambda_k - \delta_m)} + \frac{\sin \left[\frac{t}{a} (\lambda_k + \delta_m) \right]}{(\lambda_k + \delta_m)} \right\}}{\delta_m \left\{ I_1(\delta_m) - \frac{I_1(\delta_m \bar{b}) K_1(\delta_m)}{K_1(\delta_m \bar{b})} \right\} \left\{ \frac{1}{2} + \frac{a}{4t\delta_m} \sin \left(\frac{2t\delta_m}{a} \right) \right\}} \quad (17a)$$

In the limit as $\bar{b}$ approaches infinity the F_{km} expression changes to

$$\lim_{\bar{b} \rightarrow \infty} F_{km} = - \frac{\lambda_k I_1(\lambda_k) \frac{a}{2t} \left\{ \frac{\sin \left[\frac{t}{a} (\lambda_k - \delta_m) \right]}{(\lambda_k - \delta_m)} + \frac{\sin \left[\frac{t}{a} (\lambda_k + \delta_m) \right]}{(\lambda_k + \delta_m)} \right\}}{\delta_m K_1(\delta_m) \left\{ \frac{1}{2} + \frac{a}{4t\delta_m} \sin \left(\frac{2t\delta_m}{a} \right) \right\}} \quad (17b)$$

The substitution of equation (16) into equation (13) produces a relation between all of the A_n coefficients.

$$A_n = D_n + \sum_{k=1}^{\infty} A_k G_{kn} \quad (18)$$

Here G_{kn} is defined as

$$G_{kn} = \sum_{m=1}^{\infty} F_{km} E_{mn} \quad (19)$$

To obtain the value of each A_n coefficient by the use of equation (18) would require the solution of an infinite determinant. It can be shown that as successively larger $N \times N$ determinants are solved the value of each A_n approaches a definite limit. The value of this limit is what would be obtained if an infinite number of terms were taken. Extrapolation formulas have been utilized as a means of obtaining the same limit to a given number of significant figures.

A two constant extrapolation formula that is reported in [B2] is

$$A_n^{(N)} = A_n^{(\infty)} + \frac{C_1}{N} + \frac{C_2}{N^2} \quad (20)$$

The $A_n^{(\infty)}$, C_1 , and C_2 unknowns in equation (20) can be evaluated by the solution of three simultaneous equations. Each equation requires a distinct $A_n^{(N)}$ value. These values can be obtained from the solution of three different A_n determinants. Each determinant must differ in order from the preceding by $N - 2$. Hence, only values obtained from successive odd or even determinants are suitable for use in solving equation (20).

The solution representation of the temperature field within region I provides a means of estimating several errors that may be present in a thermopile heat flux gauge reading. These errors originate from two separate sources: (a) the local divergence of the heat flux field within the plate and (b) the temperature dependence of the thermal conductivity and the Seebeck coefficient.

The local divergence of the heat flux field within the plate results in a difference between the heat flux at the interface between the gauge and plate and the spatially constant heat flux at the opposite face of the plate. Therefore, a fractional part of the energy that enters that area of the plate directly opposite the gauge will flow around the gauge. This net heat loss from region I gives rise to a misleadingly low gauge reading (i.e., a gauge error). The percentage difference between the heat entering region I and the heat lost through the interface between the gauge and plate is

$$\% \text{ Error} = \left\{ 1 + \frac{2}{B^2} \int_0^1 \frac{\partial \bar{T}_I}{\partial z}(\bar{r}, 1) \bar{r} d\bar{r} \right\} \times 100 \quad (21)$$

The temperature dependence of the thermal conductivity and the Seebeck coefficient of a specific gauge may change the proportionality between the heat flow through the gauge and the emf output. The mean temperature of the gauge is the basis for correcting the error in the output. If the gauge is thin and the temperature gradient within the gauge is not large, then the radial mean temperature of the interface between the gauge and the plate provides a good approximation of the mean gauge temperature. The radial mean interface temperature ($\bar{T}_{\text{mean}}$) can be determined from

$$\bar{T}_{\text{mean}} = 2 \int_0^1 \bar{T}_I(\bar{r}, 1) \bar{r} d\bar{r} \quad (22)$$

This approximation may be somewhat in error if the axial temperature gradient within the gauge is large and the governing thermal and electrical gauge properties change appreciably with temperature. If the axial temperature drop across the gauge is taken into account then equation (22) can be improved. The resulting dimensionless equation for the mean gauge temperature is

$$\text{Mean Temp.} = \bar{T}_{\text{mean}} \left(\frac{\frac{Bi}{2} + \frac{k_g t}{k_p t_g}}{\frac{Bi}{2} + \frac{k_g t}{k_p t_g}} \right) \quad (23)$$

Equation (23) provides an excellent estimate of the mean gauge temperature for all but the most severe operating conditions.

One-Dimensional Conduction Factor

As earlier remarked, a significant departure from axial one-dimensional heat flow through a thermopile heat flux gauge can lead to an error in the gauge reading. A locally distorted heat flux field at the edge of the gauge is inevitable. Reference [3] suggests that edge effects produce a negligible error in the gauge reading if the gauge is constructed with the thermopile junctions located at least five times the gauge thickness from the edge of the gauge. There is no indication that the recommended thermopile region was derived with consideration given to a spatially distorted temperature at the interface between the gauge and heat transferring surface. Equation (9) implies that a radially nonuniform temperature is typical at the interface between a gauge and plate. It is obvious that the heat flux field

within the gauge is governed by the entire conductive/convective thermal system. A specific gauge design, therefore, may not insure against an error due to a distorted heat flux within the gauge which results from an unfavorable thermal system.

A one-dimensional conduction factor, which provides an indication of when the mean heat flow through a thin disk can be considered to be axially one-dimensional, will be developed in the following analysis. The factor was formulated with consideration given to the entire heat flux gauge-plate thermal system. Recall that the previous plate heat conduction analysis utilized a simplified thermal boundary condition which was based on the assumption of an axial linear temperature gradient within a disk affixed to a plate. In view of the foregoing, it appears that the conduction factor would serve a twofold purpose: first, predicting when the previous plate heat conduction results are no longer valid and, second, indicating when an error may be present in a thermopile type heat flux gauge reading due to a significant departure from an axial one-dimensional heat flow through the gauge.

A disk of radius a and thickness t_g is attached to the plate. The axis of the disk coincides with the z coordinate of the plate. A dimensionless temperature is required in addition to those dimensionless parameters defined by Equations (2a), (2b), and (2c). This is the dimensionless temperature distribution of the disk and is defined as

$$\bar{T} = \frac{T_{\text{disk}} - T_{\infty}}{T_w - T_{\infty}} \quad (24)$$

It is convenient to express the dimensionless heat conduction equation of the disk in the integral form.

$$\left(\frac{t_g}{a}\right)^2 \int_0^1 \int_0^1 \frac{1+\tau g/\bar{r}}{\bar{r}} \left(\frac{\partial^2 \bar{T}}{\partial \bar{r}^2} + \frac{1}{\bar{r}} \frac{\partial \bar{T}}{\partial \bar{r}} \right) \bar{a} \bar{z} d\bar{r} = - \int_0^1 \bar{r} \frac{\partial \bar{T}}{\partial \bar{z}}(\bar{r}, 1 + \frac{t_g}{t}) d\bar{r} + \int_0^1 \bar{r} \frac{\partial \bar{T}}{\partial \bar{z}}(\bar{r}, 1) d\bar{r} \quad (25)$$

This equation allows for heat loss at the edge of the gauge.

The mean value theorem of integral calculus states that a $\bar{\zeta}$ exists within the thin gauge ($t_g \ll a$) such that $1 \leq \bar{\zeta} \leq 1 + \frac{t_g}{t}$ where

$$\int_1^{1+t_g/t} \frac{1}{\bar{r}} \left(\frac{\partial^2 \bar{T}}{\partial \bar{r}^2} + \frac{1}{\bar{r}} \frac{\partial \bar{T}}{\partial \bar{r}} \right) d\bar{z} = \frac{t_g}{t} \frac{1}{\bar{r}} \left\{ \frac{\partial^2 \bar{T}}{\partial \bar{r}^2} (\bar{r}, \bar{\zeta}) + \frac{1}{\bar{r}} \frac{\partial \bar{T}}{\partial \bar{r}} (\bar{r}, \bar{\zeta}) \right\} \quad (26)$$

For physically realistic convective boundary conditions at the edge of the gauge and significant differences between Bi_e and Bi , $\bar{\zeta}$ may be approximated as being equal to unity, i.e. $\bar{\zeta} \approx 1$. The boundary conditions where this approximation would prove invalid occur when the disk and plate interface temperature approaches a constant (Bi_e approaches Bi) or when the convective heat transfer coefficient at the edge of the disk becomes large when compared with that at the disk face. As Bi_e approaches Bi the problem defined in the previous analysis begins to disappear as would any interest in a one-dimensional conduction factor. Reference [B4] indicates that a large reduction in heat transfer occurs at the edge of the gauge as opposed to the opposite being true.

Both partial derivatives on the right hand side of equation (25) may be represented in terms of other thermal parameters. Noting that the local convective heat flux at the disk surface is given by $h(T_{\text{disk}} - T_\infty)$, it follows that

$$- \frac{\partial \bar{T}}{\partial \bar{z}} \left(\bar{r}, 1 + \frac{t_g}{t} \right) = \frac{k}{k_g} Bi \bar{T} \left(\bar{r}, 1 + \frac{t_g}{t} \right) \quad (27)$$

The local heat flux at the interface between the disk and plate must be conserved, that is

$$\frac{\partial \bar{T}}{\partial \bar{z}} (\bar{r}, 1) = \frac{k}{k_g} \frac{\partial \bar{T}}{\partial \bar{z}} (\bar{r}, 1) \quad (28)$$

Let a conduction factor C_f be defined as

$$C_f = \frac{t}{a} \left| \frac{\int_0^1 \bar{r} \left\{ \frac{\partial^2 \bar{T}_I}{\partial \bar{r}^2} (\bar{r}, 1) + \frac{1}{\bar{r}} \frac{\partial \bar{T}_I}{\partial \bar{r}} (\bar{r}, 1) \right\} d\bar{r}}{\int_0^1 \bar{r} \frac{\partial \bar{T}_I}{\partial \bar{z}} (\bar{r}, 1) d\bar{r}} \right| \quad (29)$$

Equation (9) in the previous analysis section enables one to easily evaluate the integrals in the C_f expression.

When equation (25) is expressed in terms of the preceding quantities developed in equations (26) through (29), a relation between the magnitudes of the radial, axial, and convective heat transport contributions is obtained

$$\frac{t}{a} \frac{k_r}{k} C_f = 1 + \left| \frac{Bi \int_0^1 \bar{r} \bar{T} (\bar{r}, 1) + \frac{t}{a} \frac{k_r}{k} d\bar{r}}{\int_0^1 \bar{r} \frac{\partial \bar{T}_I}{\partial \bar{z}} (\bar{r}, 1) d\bar{r}} \right| \quad (30)$$

Thus, when the product of the conduction factor, the disk to plate thermal conductivity ratio, and the disk thickness to radius ratio is much less than unity, i.e.,

$$\frac{t}{a} \frac{k_r}{k} C_f \ll 1 \quad (31)$$

the mean heat flow through the disk can be considered to be axially one-dimensional.

A more complete presentation of errors associated with thermopile heat flux gauges is addressed in reference [B5].

APPENDIX B

NOMENCLATURE

A_n ,	dimensionless series coefficient;
a ,	disk radius;
Bi ,	Biot number, ht/k ;
Bi_e ,	effective Biot number, $h_e t/k$;
B_m	dimensionless series coefficient;
b ,	plate radius;
$\bar{b}$,	dimensionless plate radius, b/a ;
$\bar{b}_\infty$,	dimensionless plate radius, equation (34);
C_F	conduction factor
h ,	convective heat transfer coefficient;
h_e ,	effective convective heat transfer coefficient, equation (1);
k ,	plate thermal conductivity;
k_g ,	disk thermal conductivity;
r ,	radial coordinate;
$\bar{r}$,	dimensionless radial coordinate;
T_I ,	region I temperature representation;
T_{II} ,	region II temperature representation;
T_{disk} ,	disk temperature representation;
T_w ,	plate surface temperature, equation (3);
T_∞ ,	temperature of external fluid environment;
$\bar{T}_I$,	dimensionless region I temperature;
$\bar{T}_{II}$,	dimensionless region II temperature;
$\bar{T}$,	dimensionless disk temperature;
t ,	plate thickness;
t_g ,	disk thickness;
w ,	uniform heat flux at the plate surface;

z , axial coordinate;
 $\bar{z}$, dimensionless axial coordinate, z/l .

Greek Symbols

λ_m , eigenvalue, equation (11);
 ϑ_1 , dimensionless region I temperature;
 ϑ_2 , dimensionless region II temperature;
 λ_η eigenvalue, equation (11).

Subscripts

k , integer value, ($0 < k \leq \infty$);
 m , integer value, equation (10);
 n , integer value, equation (11).

Superscripts

n , order of the A_n determinant;
 ∞ , infinite order A_∞ determinant.

Appendix B

References

- B1. Myers, G. E., Analytical Methods in Conduction Heat Transfer, McGraw-Hill Book Company (1971) pages 151-154.
- B2. Smythe, W. R., Flow Over Thick Plate with Circular Hole, J. Appl. Phy., 23, 447 (1952).
- B3. Harer, W. E., Jr., Thin Foil Heat Meter, Rev. Sci. Instr., 36, 1564 (1965).
- B4. Chilcote, R. E., A Review of Separated and Reattaching Flows with Heat Transfer, Int. J. Heat Mass Trans., 10, 783, (1967).
- B5. Wesley, D. A., Thin Disk on a Convectively Cooled Surface--Application to Heat Flux Measurement Errors, J. Heat Transfer, 101, 346, (1979).

Data Reduction Computer Program Listing

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01960 Z1=ADFLT(Y*1)
01960 Z2=WPZ1
01960 Z4=Y*Z1
01970 Z5=REFSK(1,Z1,IN,KNDF,MZ1)
01980 Z6=REFSK(1,Z2,MV,KNDF,MZ1)
01990 Z7Z=(Z4/Z5)*Z1
02000 O1=(SIN(X1-Z1)) / (Z1*(X1-Z1))
02010 O2=(SIN(Y1-Z1)) / (Z1*(Y1-Z1))
02020 O3=X1*O2
02030 O4=(SIN(X1-Z1)) / (Z1*(X1-Z1))
02040 O5=(SIN(Y1-Z1)) / (Z1*(Y1-Z1))
02050 O6=O4*O5
02060 O7=X1*O5
02070 O8=(Y1-Z1) * ((1./O4)*Z1) * SIN(Z1*Z1)
02080 O9=O7/O8*Z1
02090 O10=Z1/O8
02100 A4 SUM=SLW*O7*O8
02110 IF(11.EC,2) GO TO 78
02120F CALC= 4 BY N DETERMINANT
02130 SS(11,11)=-SUM*O9
02140 GO TO 62
02150 DO SS(11,11)=1.0-(SUM*O9)
02160 NO CONTINUE
02170 A5 CONTINUE
02180 N1=N-2
02190 N2=N-4
02200 DO 103 I=1,N1
02210 DO 104 J=1,N2
02220 DO 110 I2=1,N1
02230 I10=I2-I21
02240 DO 142 J1=1,N1
02250 DO 150 K1=1,N1
02260F CALC= (N-2) BY (N-2) DETERMINANT
02270 I50 SS(11,K1)=SS(11,I1)
02280 I40 CONTINUE
02290 DO 142 J2=1,N2
02300 DO 170 K2=1,N2
02310F CALC= (N-4) BY (N-4) DETERMINANT
02320 I70 SS(11,J2,K1)=SS(11,J1,K1)
02330 I60 CONTINUE
02340 N0=50
02350 N01=64
02360 M02=46
02370 M01
02380 INT=0
02390F SOLVE 4 BY 4 DETERMINANT
02400 CALL SAYE(IN0,M01,SS,M0,INIT,IN,KF0)
02410F SOLVE (N-2) BY (N-2) DETERMINANT
02420 CALL SAYE(M01,M1,SS,M01,INT,IN,KF1)
02430F SOLVE (N-4) BY (N-4) DETERMINANT
02440 CALL SAYE(M02,M2,SS,M02,INT,IN,KF2)
02450F CALC= FOR EXTRAPOLATION 7*JTIME TO CALC. FOURIER COEFF
02460 DO 146 I3=1,N1
02470 FC(1)=0
02480 FC2=0
02490 FC3=0
02500 G1=1./FC1
02510 G2=1./FC1*FC(1)
02520 G3=1./FC2
02530 G4=1./FC1*FC(2)
02540 G5=1./FC3
02550 G6=1./FC1*FC(3)
02560 G7=(FC3*G6)-10*G4
02570 G8=G1*FC6-G4

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02500 G0=6**((F-F1)
02500 M0=(F+G)*K0
02500 F10=M0/(1+(F1000)/(G0000))
02510 G10=0.5*(F+M1000)/(M10000)
02520 F12=F02*(1+(101010000)/(M02010000))
02530 D02=F010000/(F012
02540 144*M0/(100000000)
02550 F10=(0.00000000)/(M000)
02560 SUM1=0.0
025700 CALC. SERIES TO DETERMINE FROD
02580 DO 45 LL=1,N2
02590 F2=0.00000000
02700 F1=F02
02710 E4=F02
02720 F5=3F501(F1,NV,MODE)/FYP(F-F1)
02730 F4=STN(F4)
02740 F5=EM(F5)
02750 F4=F7400(F4)
02760 SUM1=SUM1+F4
02770 45 CONTINUE
02780 FROD(NIP)=E1*SUM1
02790 370=300/11.-(FROD(NIP)/100.1)
02800 H2=0.02/(FROD(NIP))
02810 W1=L/(1.1/2)+(0.0331/F5001)
02820 F1(NIP,E0,1) GO TO 290
028300 PRINT OUT VARIOUS HEX VALUES
02840 PRINT 1,200,400,THEN,TIME
02850 1 FORMAT('T0=0F=*,F5,1,4F,00ALD=*,F6,1,5F,0T40=*,F6,1,5F,
02860*TIME=*,F6,1)
02870 PRINT 4,M2,M1,FROD(NIP),NIP
02880 4 FORMAT('M2=*,F6,1,2,5F,0HEF=*,F6,1,5F,0ERROR=*,1PF3,2,5F,
02890*TIME=*,F6,1)
02900 790 CONTINUE
02910 500 CONTINUE
02920 640 CONTINUE
02930 GO TO 600
02940 440 CONTINUE
029500 CALL SUBROUTINE TO CALC. HEAT CONDUCTION PROB. OF COVER TUBE
02960 CALL DAT(TOUT,ADUT,TM0,TM4L)
02970 DO 901 I=1,12
02980 TM=I+10
02990 901 TEMP(I)=TEM(I)
03000 PRINT IC, COVER TUBE TEMP. ABOVE HEX HEAT FLUX GAUGES
03010 PRINT*,F
03020 PRINT*,I,INHER TYPE SURFACE TEMPERATURE
03030 PRINT*,FLCC,(1) (2) (3) (4) (5) (6)
03040 PRINT 700,(TM4L(J),J=1,6)
03050 700 FORMAT(6(4X,F6,1))
03060 PRINT*,F
03070 600 CONTINUE
03080 SUMN1=0.0
03090 DO 971 I=1,18
03100 971 SUMN1=SUMN1+TEMP(I)
03110 TML(MJAC)=SUMN1/18.0
03120 SUMN2=0.0
03130 DO 972 J=1,12
03140 K=J+10
03150 972 SUMN2=SUMN2+TEMP(K)
03160 TML(MJAC)=SUMN2/12.
031700 CALL SUBROUTINE TO CALC. THERMAL RADIATION OF ENCLOSURE
03180 CALL RAD(TEMP,RAD)
031900 PRINT OUT TEMP. & HEAT FLUX OF EACH ENCLOSURE ELEMENT
03200 PRINT*,F
03210 PRINT*,SHEX. TEMP. AND RADIATION HEAT FLUX

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03220 PRINT*,# #
03230 PRINT*,# LOC. TEMP. 0 FLY LOC. TEMP. 0 FLUX LOC. TEMP. 0 FL
03240*UX#
03250 DO 973 I=1,7,1
03260 IX=I+1
03270 IY=I+2
03280 PRINT 974,I,TEMP(I),RAD(I),IX,TEMP(IX),RAD(IX),IY,TEMP(IY),RAD(IY)
03290 974 FORMAT(IX,*,*,I2,*)*,2Y,F5.1,2X,F5.1,2X,F5.1,6X,*,*,I1,*)*,2X,F5.1,2X,
03300*F6.1,5X,*,*,I1,*)*,2X,F5.1,2X,F5.1)
03310 973 CONTINUE
03320 DO 975 J=10,16,1
03330 JX=J+1
03340 JY=J+2
03350 PRINT 976,J,TEMP(J),RAD(J),JX,TEMP(JX),RAD(JX),JY,TEMP(JY),RAD(JY)
03360 976 FORMAT(I2X,*,*,I2,*)*,2Y,F5.1,2X,F5.1,5X,*,*,I2,*)*,2X,F5.1,2X,
03370*F6.1,5X,*,*,I2,*)*,2X,F5.1,2X,F5.1)
03380 975 CONTINUE
03390 PRINT*,# #
03400 PRINT*,# PIPE IN. TEMP. AND RADIATION HEAT FLUX#
03410 PRINT*,# #
03420 PRINT*,# LOC. TEMP. 0 FLUX LOC. TEMP. 0 FLUX LOC. TEMP. 0 FL
03430*UW#
03440 DO 977 K=19,24,1
03450 KX=K+1
03460 KY=K+2
03470 PRINT 978,K,TEMP(K),RAD(K),KX,TEMP(KX),RAD(KX),KY,TEMP(KY),RAD(KY)
03480 978 FORMAT(I2X,*,*,I2,*)*,2X,F5.1,2X,F5.1,5X,*,*,I2,*)*,2X,F5.1,2X,
03490*F6.1,5X,*,*,I2,*)*,2X,F5.1,2X,F5.1)
03500 977 CONTINUE
03510 601 CONTINUE
03520C READ IN HEATED FATAL POWER IN WATTS & TEST PIPE PRESSURE IN PSI#
03530 READ(1,1001)SM,PM,PMF
03540 1001 FORMAT(F7.2,F8.5)
03550 DO 1002 I=1,60
03560C READ IN CORE TEMP. MATRIX
03570 READ(1,1003)ITLX(I),J,J=1,13)
03580 1003 FORMAT(I1(I,F5.1))
03590 1002 CONTINUE
03600C CALL SUBROUTINE TO CALC. AXIAL CONDUCTION HEAT TRANSPORT
03610 CALL AVGH(ITLX,AFIX)
03620 SS04=.3413*PM
03630 APRO1=.1162*SS04
03640 APRO2=.1724*SS04
03650 APRO3=.03956*SS04
03660 APRO4=.10670*SS04
03670 AOCN1=AFIX(1)-AFIX(5)
03680 AOCN2=AFIX(13)-AFIX(17)
03690 AOCN3=AFIX(27)-AFIX(29)
03700 AOCN4=AFIX(39)-AFIX(41)
03710 AOCV(1)=(AOCN1+APRO1)/.4735
03720 AOCV(2)=(AOCN2+APRO2)/.4735
03730 AOCV(3)=(AOCN3+APRO3)/.4735
03740 AOCV(4)=(AOCN4+APRO4)/.4735
03750C PRINT OUT CTR. AVG. HEAT TRANSFER COEFFS
03760 PRINT*,# #
03770 PRINT*,# CTRUMPPENTIAL AVERAGE HEAT TRANSFER COEFFICIENTSX
03780 PRINT*,# #
03790 DO 1004 I=1,4
03800 PRINT 1005,I
03810 1005 FORMAT('AXIAL STATION=',I1)
03820 TCVT=(ITLX(I)+TWL(I))/2.
03830 WOCV=AOCV(I)/(ITLX(I)-TCVT)
03840 TFCV=(ITLX(I)+TCVT)/2.
03850C LOC. CUBIC COEFFS FOR HELIUM PAV. NO. PARA. LOC(1)FOR G R N**3)/V**2)

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04500 DO 300 I7P=1,N
04510 Y(I7P)=YOUT(I7P)
04520 300 Y(I7P)=ADIT(I7P)
04530 SUM=0.0
04540 DO 15 I4=1,4
04550 IS SUM=SUM+Y(I4)
04560C FOURIER COEF. CALC.
04570 A0=SUM/8.
04580 S1=(1.62726847E-04)*FRO**1.1)+(1-5.55842515E-16)*FRO**2.1)
04590=(17.14088972E-03)*AC1*(8.76351932)
04600 SUO=0.0
04610 DO 16 I9=1,N
04620 IS SUO=SUO+Y(I9)
04630 B0=MI/PI*C
04640C FOURIER COEF. CALC.
04650 B0=-(FRO*SUO*ALOC(MI))/(A.*SK1
04660 Y(25)=Y(1)
04670 Y(25)=Y(I)
04680 DO 20 J=1,4
04690 J0=J+8
04700 JL=J+16
04710 Y(JK)=Y(IJ)
04720 Y(JL)=Y(IJ)
04730 Y(IJK)=Y(IJ)
04740 20 Y(IJL)=Y(IJ)
04750 M=25
04760 30 J0 K=1,29
04770 UY(K)=0.0
04780 30 DDY(K)=0.0
04790 S0=25.
04800C FIT SPLINE TO FPM. DATA
04810 CALL SMOCT(P,F,DY,SO,A,B,C,D,F,R1,R2,Y1,T2,U,V,IFRM)
04820C FIT SPLINE TO HEAT FLUX DATA
04830 CALL SMOCT(M,KC,Y,DDY,S0,AA,PP,DD,D0,P,1,P2,U1,V2,JEPR)
04840 31 CONTINUE
04850C CALC. PARAMETEES TO EVALUATE FOURIER COEFS
04860 DO 40 L=1,N
04870 S1=
04880 S2=S*S
04890 S3=S*S*S
04900 S4=S*S*S
04910 SUMA=0.0
04920 SUMB=0.0
04930 SSUM=0.0
04940 DO 50 II=1,N
04950 JJ=II-1
04960 SS=JJ
04970 SS1=SS*PI**5
04980 SS2=(SS+1.)*PI**6
04990 SS3=(PI**7*(SS+PI**4))*S
05000 SS4=(PI**7*(SS+1.)*PI**4)*S
05010 KC=II+8
05020 Y1=(B(KK)*PI**5)+(C(KK)*PI**5/S)+(D(KK)*PI**6/S)-(6.*D(KK)*PI**4/S)
05030 X1=(B(KK)*PI**4/S)+(C(KK)*PI**4/S)+(D(KK)*PI**5/S)-(6.*D(KK)*PI**4/S)
05040 Y2=(2.*C(KK)*PI**4/S)+(4.*D(KK)*PI**5/S)
05050 X2=(2.*C(KK)*PI**4/S)+(4.*D(KK)*PI**5/S)
05060 Y3=(A(KK)/S)-(7.*C(KK)/S)
05070 X3=(A(KK)/S)-(7.*C(KK)/S)
05080 Y4=(A(KK)/S)-(7.*C(KK)/S)
05090 X4=(B(KK)/S)-(4.*D(KK)/(S**2))
05100 Y5=(B(KK)/S)-(4.*D(KK)/(S**2))
05110 Y1=SUM(SS2)
05120 Y1=SUM(SS4)
05130 Y2=SUM(S2)

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05140 YYZ=COS(SS4)
05150 Y3=SIN(SS2)-SIN(SS1)
05160 Y3=SIN(SS4)-SIN(SS3)
05170 Y4=COS(SS3)-COS(SS1)
05180 Y4=COS(SS4)-COS(SS3)
05190 Z1=(X1**3)+(X2**3)+(Y1**3)+(X4**3)
05200 Z1=(X1**3)+(Y1**3)+(X2**3)+(Y3**3)+(X4**3)
05210 Z2=(X1**3)+(Y2**3)-(X3**3)+(X4**3)
05220 Z2Z2=(X1**3)+(X2**3)-(X3**3)+(X4**3)
05230 SUMR=SUMR+Z1
05240 SUMR=SUMR+Z2
05250 SSUMR=SSUMR+Z1
05260 SSUMR=SSUMR+Z2
05270 DO CONTINUE
05280 AT(L)=SUMR/(2.*PI*(RPO**5))
05290 RT(L)=SUMR/(2.*PI*(RPO**5))
05300 AOL(L)=-SSUMR/(2.*PI*SK**RPO**5)
05310 B3(L)=-SSUMR/(2.*PI*SK**RPO**5)
05320 NO CONTINUE
05330 CALC. FLURIEP SERIES AT PIPE IN. FOR SIX TEMP. LOCATIONS
05340 DO 61 LA=1,4
05350 LB=LA-1
05360 RL=LB
05370 CL=9L*PI/14
05380 SUMT=0.0
05390 SUMD=0.0
05400 DO 71 M=1,4
05410 CM=M
05420 F=Z.*G
05430 F1=(RR1**G)-(RPO**F)/(RR1**G)
05440 F2=(RR1**G)+(RPO**F)/(RR1**G)
05450 HF=F*CL
05460 G1=AT(M)*COS(HF)
05470 G2=BT(M)*SIN(HF)
05480 G3=AT(M)*COS(HF)
05490 G4=BT(M)*SIN(HF)
05500 G5=F2*(G1+G2)
05510 G6=F1*(G1+G4)
05520 SUMT=SUMT+G5
05530 SUMD=SUMD+G6
05540 DO CONTINUE
05550 TEQ3=AD*EC*SUMT+SIMQ
05560 TVAL(LA)=TEQ3
05570 48 CONTINUE
05580 CALC. FLURIEP SERIES AT PIPE IC. FOR TWELVE TEMP. LOCATIONS
05590 DO 61 LA=1,12
05600 LB=LA-1
05610 PL=LB
05620 CL=(9L*PI/14)*(PI/12.)
05630 SUMT=0.0
05640 SUMD=0.0
05650 DO 71 M=1,12
05660 CM=M
05670 F=Z.*G
05680 F1=(RR1**G)-(RPO**F)/(RR1**G)
05690 F2=(RR1**G)+(RPO**F)/(RR1**G)
05700 HF=F*CL
05710 G1=AT(M)*COS(HF)
05720 G2=BT(M)*SIN(HF)
05730 G3=AT(M)*COS(HF)
05740 G4=BT(M)*SIN(HF)
05750 G5=F2*(G1+G2)
05760 G6=F1*(G1+G4)
05770 SUMT=SUMT+G5

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05790 SUM7=SUMC+GA
05790 71 CONTINUE
05800 TEMO(IA)=20+90*SUM7+SUMO
05810 61 CONTINUE
05820 RETURN
05830 END
05840C THERMAL RADIATION SOLUTION OF THE ENCLOSURE
05850 SUBROUTINE RAD1(T,AD)
05860 DIMENSION E(30),T(30),F(30,30)
05870 DIMENSION X(30,30),TT(30),AMEG(30),R(30,30),IN(30)
05880 DIMENSION QA(10),AD(30)
05890 DO 10 I=1,10,1
05900 1A=I+1
05910 1B=I+2
05920C EMISSANCE OF HEX. SURFACE
05930 E(IA)=.11
05940 10 F(10)=.14
05950 00 15 J=10,30
05960C EMISSANCE OF COVER TURE 10. SURFACE
05970 15 E(J)=.33
05980 00 20 K=1,16,3
05990C EMISSANCE OF HEAT FLUX GAUGE SURFACE
06000 20 E(K)=.14
06010 00 40 MI=1,30
06020 00 45 MJ=1,30
06030 45 F(MI,MJ)=0.0
06040 40 CONTINUE
06050C CONFIGURATION FACTORS OF THE ENCLOSURE
06060 F(1,10)=.4477
06070 F(1,20)=.0523
06080 F(1,29)=.0523
06090 F(1,30)=.4477
06100 F(2,19)=.5774
06110 F(2,20)=.1197
06120 F(2,29)=.2327
06130 F(2,30)=.2702
06140 F(3,19)=.1197
06150 F(3,20)=.5774
06160 F(3,21)=.2702
06170 F(3,22)=.1127
06180 F(4,19)=.0523
06190 F(4,20)=.4477
06200 F(4,21)=.4477
06210 F(4,22)=.0523
06220 F(4,19)=.0327
06230 F(5,20)=.2702
06240 F(5,21)=.5774
06250 F(5,22)=.1197
06260 F(6,21)=.1197
06270 F(6,22)=.5774
06280 F(6,23)=.2702
06290 F(6,24)=.2327
06300 F(7,21)=.0523
06310 F(7,22)=.4477
06320 F(7,23)=.4477
06330 F(7,24)=.0523
06340 F(8,21)=.2327
06350 F(8,22)=.2702
06360 F(8,23)=.5774
06370 F(8,24)=.1197
06380 F(9,23)=.1197
06390 F(9,24)=.5774
06400 F(9,25)=.2702
06410 F(9,26)=.0327

```

06420 F(10,23) = -.0523
 06430 F(10,24) = -.4477
 06440 F(10,25) = -.4477
 06450 F(10,26) = -.0523
 06460 F(11,23) = -.0427
 06470 F(11,24) = -.2702
 06480 F(11,25) = -.5774
 06490 F(11,26) = -.1197
 06500 F(12,25) = -.1197
 06510 F(12,26) = -.5774
 06520 F(12,27) = -.2702
 06530 F(12,28) = -.0427
 06540 F(13,25) = -.0523
 06550 F(13,26) = -.4477
 06560 F(13,27) = -.4477
 06570 F(13,28) = -.0523
 06580 F(14,25) = -.0427
 06590 F(14,26) = -.2702
 06600 F(14,27) = -.5774
 06610 F(14,28) = -.1197
 06620 F(15,27) = -.1197
 06630 F(15,28) = -.5774
 06640 F(15,29) = -.2702
 06650 F(15,30) = -.0427
 06660 F(16,27) = -.0523
 06670 F(16,28) = -.4477
 06680 F(16,29) = -.4477
 06690 F(16,30) = -.0523
 06700 F(17,27) = -.0427
 06710 F(17,28) = -.2702
 06720 F(17,29) = -.5774
 06730 F(17,30) = -.1197
 06740 F(18,19) = -.2702
 06750 F(19,20) = -.0427
 06760 F(19,29) = -.1197
 06770 F(19,30) = -.5774
 06780 F(19,31) = -.1197
 06790 F(19,32) = -.0415
 06800 F(19,33) = -.0223
 06810 F(19,34) = -.1441
 06820 F(19,35) = -.0114
 06830 F(19,36) = -.0337
 06840 F(19,37) = -.0651
 06850 F(19,38) = -.0661
 06860 F(19,39) = -.0428
 06870 F(19,40) = -.0651
 06880 F(19,41) = -.0337
 06890 F(20,21) = -.0415
 06900 F(20,31) = -.3933
 06910 F(20,51) = -.1441
 06920 F(20,18) = -.0223
 06930 F(20,19) = -.0337
 06940 F(20,20) = -.0114
 06950 F(20,21) = -.0337
 06960 F(20,22) = -.0651
 06970 F(20,23) = -.0428
 06980 F(20,29) = -.0661
 06990 F(20,30) = -.0651
 07000 F(21,31) = -.1441
 07010 F(21,51) = -.3933
 07020 F(21,61) = -.0315
 07030 F(21,81) = -.0223
 07040 F(21,19) = -.0651
 07050 F(21,20) = -.0337

07068 F(21,21) = -.8114
 07078 F(21,22) = -.3337
 07088 F(21,23) = -.0651
 07098 F(21,24) = -.0563
 07108 F(21,30) = -.0424
 07118 F(22,3) = -.0223
 07128 F(22,5) = -.0615
 07138 F(22,6) = -.3933
 07148 F(22,8) = -.1841
 07158 F(22,19) = -.0663
 07168 F(22,20) = -.0651
 07178 F(22,21) = -.0137
 07188 F(22,22) = -.0114
 07198 F(22,23) = -.0337
 07208 F(22,24) = -.0651
 07218 F(22,25) = -.0424
 07228 F(23,6) = -.1841
 07238 F(23,8) = -.3933
 07248 F(23,9) = -.0615
 07258 F(23,11) = -.0223
 07268 F(23,21) = -.0424
 07278 F(23,22) = -.0651
 07288 F(23,23) = -.0337
 07298 F(23,24) = -.0114
 07308 F(23,26) = -.0337
 07318 F(23,25) = -.0651
 07328 F(23,26) = -.0663
 07338 F(24,6) = -.0223
 07348 F(24,8) = -.0115
 07358 F(24,9) = -.3933
 07368 F(24,11) = -.1841
 07378 F(24,21) = -.0663
 07388 F(24,22) = -.0651
 07398 F(24,23) = -.0337
 07408 F(24,24) = -.0114
 07418 F(24,25) = -.0337
 07428 F(24,26) = -.0651
 07438 F(24,27) = -.0424
 07448 F(25,9) = -.1841
 07458 F(25,11) = -.3933
 07468 F(25,12) = -.0615
 07478 F(25,14) = -.0223
 07488 F(25,22) = -.0424
 07498 F(25,23) = -.0651
 07508 F(25,24) = -.0337
 07518 F(25,25) = -.0114
 07528 F(25,26) = -.0337
 07538 F(25,27) = -.0651
 07548 F(26,26) = -.0651
 07558 F(26,9) = -.0223
 07568 F(26,11) = -.0615
 07578 F(26,12) = -.1841
 07588 F(26,14) = -.3933
 07598 F(26,23) = -.0663
 07608 F(26,24) = -.0651
 07618 F(26,25) = -.0337
 07628 F(26,26) = -.0114
 07638 F(26,27) = -.0337
 07648 F(26,28) = -.0651
 07658 F(26,29) = -.0424
 07668 F(27,12) = -.1841
 07678 F(27,14) = -.3933
 07688 F(27,15) = -.0615
 07698 F(27,17) = -.0223


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      08340 250 CONTINUE
      08350 MD=30
      08360 M=30
      08370 N=30
      08380 IAT=4
      08390C MATRIX INVERSION
      08400 CALL SAXPBY(N,M,Y,B,INIT,IN,KEP)
      08410C ANY ERRORS IN MATRIX INVERSION
      08420 PRINT*, ' '
      08430 PRINT 501,KEP
      08440 501 FORMAT(2X,'NO. OF CALC. ERRORS=',IX,I1)
      08450 DO 110 L=1,30
      08460 SUM=0.0
      08470 DO 120 LL=1,30
      08480 SUM=SUM+ELL(LL)*AMEG(LL)
      08490 120 CONTINUE
      08500 BR(LL)=SUM
      08510 110 CONTINUE
      08520 DO 130 K=1,30
      08530C CALC. THERMAL RADIATION HEAT FLUX
      08540 AQE(K)=(EL(K))* (AMFG(K)-BR(K))/ (1.0-EKE(K))
      08550 130 CONTINUE
      08560 RTURN
      08570 FNO
      08580C CALC. AVG. CIR. HEAT TRANSFER COEFS & AXIAL HEAT TRANSPORT
      08590 SUBROUTINE AVGM(T,FLK)
      08600 DIMENSION T(60,13),X(60),DT(60,13),TY(60),DY(60),X(60),Q(60)
      08610 DIMENSION AFL(60),C(60),D(60),Q(62),R1(62),R2(62),Y1(62),Y2(62)
      08620 DIMENSION U(62),V(62),FLT(60,13)
      08630 DO 30 I=1,13
      08640 DO 40 JJ=1,60
      08650 T(I,JJ)=T(I,J,I)
      08660 40 CONTINUE
      08670 DO 50 KK=1,60
      08680 LL=KK-1
      08690 MM=LL*2
      08700 AM=MM
      08710 EKK)=AM/12.
      08720 50 CONTINUE
      08730 N=60
      08740 S=N
      08750 BM=0.
      08760 DO 70 NN=1,60
      08770 70 JY(NN)=98
      08780C FIT SPLINE TO EACH SET OF AXIAL TEMP. DATA
      08790 CALL SHOC(N,T,Y,DY,S,A,P,C,D,R,R1,R2,T1,T2,U,V,YDD)
      08800 DO 80 JJ=1,60
      08810C CALC. THE AXIAL FIRST DERIVATIVE
      08820 80 DT(I,J)=R1(I)
      08830 30 CONTINUE
      08840C CUBIC COEFS OF WFLUM THERMAL CONDUCTIVITY
      08850 R1=2.0833E-11
      08860 R1=-4.0833E-08
      08870 C1=1.0263E-04
      08880 D1=7.0444E-02
      08890C CUBIC COEFS OF MAG. OXIDE THERMAL CONDUCTIVITY
      08900 R2=-7.2878E-04
      08910 R2=4.1495E-05
      08920 C2=-5.1197E-02
      08930 D2=31.8666
      08940C CUBIC COEFS OF NICHROME W THERMAL CONDUCTIVITY
      08950 R3=1.1760E-11
      08960 R3=4.1733E-08
      08970 C3=5.7122E-03

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00000 01-0-2011
00000F CURIC COEFF OF STAINLESS STEEL THERMAL CONDUCTIVITY
00000 001-0.02727E-01
00010 001-0.00000E-00
00020 001-0.00000E-00
00030 001-0.00000E-00
00040 001-0.00000E-00
00050 001-0.00000E-00
00060 001-0.00000E-00
00070 001-0.00000E-00
00080 001-0.00000E-00
00090F CURIC COEFF OF NICKEL THERMAL CONDUCTIVITY
00100 001-0.02727E-01
00110 001-0.00000E-00
00120 001-0.00000E-00
00130 001-0.00000E-00
00140 001-0.00000E-00
00150 001-0.00000E-00
00160 001-0.00000E-00
00170 100 IF(I,0.0,1) GO TO 175
00180 IF(I,0.0,1) GO TO 185
00190 001-0.00000E-00
00200 100 IF(I,0.0,1) GO TO 195
00210 IF(I,0.0,1) GO TO 195
00220 001-0.00000E-00
00230 100 IF(I,0.0,1) GO TO 165
00240 100 CONTINUE
00250 001-0.00000E-00
00260 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00270 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00280 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00290 IF(I,0.0,1) GO TO 227
00300 001-0.00000E-00
00310 120 IF(I,0.0,1) GO TO 220
00320 001-0.00000E-00
00330 120 IF(I,0.0,1) GO TO 220
00340 227 ARKS=(1.00000E+00)/166.
00350 ARKS=(1.00000E+00)/166.
00360 ARKS=(1.00000E+00)/166.
00370 TAK=ARKS*ARKS/166.
00380 FLX(I,J)=TAK*NT(I,J)
00390 001-0.00000E-00
00400 220 ARKS=(1.00000E+00)/166.
00410 ARKS=(1.00000E+00)/166.
00420 ARKS=(1.00000E+00)/166.
00430 TAK=ARKS*ARKS/166.
00440 FLX(I,J)=TAK*NT(I,J)
00450 001-0.00000E-00
00460 229 ARKS=(1.00000E+00)/166.
00470 ARKS=(1.00000E+00)/166.
00480 ARKS=(1.00000E+00)/166.
00490 TAK=ARKS*ARKS/166.
00500 FLX(I,J)=TAK*NT(I,J)
00510 126 CONTINUE
00520 001-0.00000E-00
00530 130 CONTINUE
00540 001-0.00000E-00
00550 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00560 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00570 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00580 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00590 AKST=(AKST(I,J)**1.0)+(AKST(I,J)**2.0)+(AKST(I,J)**3.0)
00600 IF(I,0.0,1) GO TO 237
00610 001-0.00000E-00

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09620 130 IF (J.EQ.K) GO TO 230
09630 DO 139 K=8,13
09640 139 IF (J.EQ.K) GO TO 239
09650 237 ARKS=(.2950*AKSS)/144.
09660 ARKL=(.0553*AKST)/144.
09670 ARMO=(.0231*AKMO)/144.
09680 ARNI=(.0181*AKNI)/144.
09690 APKG=(.2331*AKGS)/144.
09700 TAK=ARKS+ARKL+ARMO+ARNI+APKG
09710 FLX(I,J)=-TAK*DT(I,J)
09720 GO TO 136
09730 23A ARKS=(.3539*AKSS)/144.
09740 ARKL=(.0737*AKST)/144.
09750 ARMO=(.0277*AKMO)/144.
09760 ARNI=(.0097*AKNI)/144.
09770 ARKG=(.3764*AKGS)/144.
09780 TAK=ARKS+ARKL+ARMO+ARNI+APKG
09790 FLX(I,J)=-TAK*DT(I,J)
09800 GO TO 136
09810 239 ARKS=(.11809*AKSS)/144.
09820 ARKL=(.1319*AKST)/144.
09830 ARMO=(.0519*AKMO)/144.
09840 ARNI=(.0581*AKNI)/144.
09850 ARKG=(.8529*AKGS)/144.
09860 TAK=ARKS+ARKL+ARMO+ARNI+APKG
09870 FLX(I,J)=-TAK*DT(I,J)
09880 136 CONTINUE
09890 GO TO 110
09900 145 CONTINUE
09910 DO 148 J=1,13
09920 AKSS=(A4*Y(I,J)**2.)+(B4*Y(I,J)**2.)+(C4*Y(I,J)**2.
09930 AKMO=(A2*Y(I,J)**3.)+(B2*Y(I,J)**2.)+(C2*Y(I,J)**2.
09940 ARNI=(A3*Y(I,J)**2.)+(B3*Y(I,J)**2.)+(C3*Y(I,J)**2.
09950 AKGS=(A1*Y(I,J)**3.)+(B1*Y(I,J)**2.)+(C1*Y(I,J)**2.
09960 IF (J.EQ.1) GO TO 247
09970 DO 148 K=2,7
09980 148 IF (J.EQ.K) GO TO 248
09990 DO 149 K=8,13
10000 149 IF (J.EQ.K) GO TO 249
10010 247 ARKS=(.3297*AKSS)/144.
10020 ARMO=(.0461*AKMO)/144.
10030 ARNI=(.03161*AKNI)/144.
10040 APKG=(.2331*AKGS)/144.
10050 TAK=ARKS+ARMO+ARNI+APKG
10060 FLX(I,J)=-TAK*DT(I,J)
10070 GO TO 146
10080 248 ARKS=(.3902*AKSS)/144.
10090 ARMO=(.0553*AKMO)/144.
10100 ARNI=(.0193*AKNI)/144.
10110 ARKG=(.3764*AKGS)/144.
10120 TAK=ARKS+ARMO+ARNI+APKG
10130 FLX(I,J)=-TAK*DT(I,J)
10140 GO TO 146
10150 249 ARKS=(.11877*AKSS)/144.
10160 ARMO=(.1037*AKMO)/144.
10170 ARNI=(.0382*AKNI)/144.
10180 ARKG=(.8529*AKGS)/144.
10190 TAK=ARKS+ARMO+ARNI+APKG
10200 FLX(I,J)=-TAK*DT(I,J)
10210 146 CONTINUE
10220 GO TO 110
10230 155 CONTINUE
10240 DO 156 J=1,13
10250 AKSS=(A4*Y(I,J)**2.)+(B4*Y(I,J)**2.)+(C4*Y(I,J)**2.

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10260 AKM0=(I2*Y(I,J))**2.1*(R2*Y(I,J))**2.1*(C2*Y(I,J))**2
10270 ARN1=(I3*Y(I,J))**2.1*(R3*Y(I,J))**2.1*(C3*Y(I,J))**2
10280 ARNL=(I4*Y(I,J))**2.1*(R4*Y(I,J))**2.1*(C4*Y(I,J))**2
10290 AKS=(I5*Y(I,J))**2.1*(R5*Y(I,J))**2.1*(C5*Y(I,J))**2
10300 IF(J.EQ.1) GO TO 267
10310 DO 158 K=2,7
10320 158 IF(J.FQ.K) GO TO 268
10330 GO 159 K=4,13
10340 159 IF(J.FQ.K) GO TO 269
10350 159 ARKS=(1.1262*AKSS)/144.
10360 159 ARN1=(.0241*AKND)/144.
10370 159 ARN1=(.0241*AKND)/144.
10380 ARNL=(.0251*AKNL)/144.
10390 ARKG=(.2311*AKGS)/144.
10400 TAK=ARKS+ARM0+ARN1+ARNL+ARKG
10410 FLX(I,J)=TAK*Y(I,J)
10420 GO TO 156
10430 258 ARKS=(1.1902*AKSS)/144.
10440 ARN1=(.0555*AKND)/144.
10450 ARN1=(.0555*AKND)/144.
10460 ARNL=(.0577*AKNL)/144.
10470 ARKG=(.1748*AKGS)/144.
10480 TAK=ARKS+ARM0+ARN1+ARNL+ARKG
10490 FLX(I,J)=TAK*Y(I,J)
10500 GO TO 156
10510 259 ARKS=(1.0774*AKSS)/144.
10520 ARN1=(.0377*AKND)/144.
10530 ARN1=(.0377*AKND)/144.
10540 ARNL=(.0461*AKNL)/144.
10550 ARKG=(.0529*AKGS)/144.
10560 TAK=ARKS+ARM0+ARN1+ARNL+ARKG
10570 FLX(I,J)=TAK*Y(I,J)
10580 156 CONTINUE
10590 GO TO 110
10600 155 CONTINUE
10610 DO 166 J=1,13
10620 AKSS=(I6*Y(I,J))**3.1*(R6*Y(I,J))**2.1*(C6*Y(I,J))**2
10630 AKM0=(I2*Y(I,J))**2.1*(R2*Y(I,J))**2.1*(C2*Y(I,J))**2
10640 ARN1=(I3*Y(I,J))**2.1*(R3*Y(I,J))**2.1*(C3*Y(I,J))**2
10650 AKS=(I5*Y(I,J))**2.1*(R5*Y(I,J))**2.1*(C5*Y(I,J))**2
10660 IF(J.EQ.1) GO TO 267
10670 DO 168 K=2,7
10680 168 IF(J.FQ.K) GO TO 268
10690 GO 169 K=4,13
10700 169 IF(J.FQ.K) GO TO 269
10710 267 ARKS=(1.3247*AKSS)/144.
10720 ARN1=(.0411*AKND)/144.
10730 ARNL=(.0411*AKNL)/144.
10740 ARKG=(.2311*AKGS)/144.
10750 TAK=ARKS+ARM0+ARN1+ARNL+ARKG
10760 FLX(I,J)=TAK*Y(I,J)
10770 GO TO 166
10780 268 ARKS=(1.7902*AKSS)/144.
10790 ARN1=(.0555*AKND)/144.
10800 ARNL=(.0577*AKNL)/144.
10810 ARKG=(.1748*AKGS)/144.
10820 TAK=ARKS+ARM0+ARN1+ARNL+ARKG
10830 FLX(I,J)=TAK*Y(I,J)
10840 GO TO 166
10850 269 ARKS=(1.0774*AKSS)/144.
10860 ARN1=(.0377*AKND)/144.
10870 ARNL=(.0461*AKNL)/144.
10880 ARKG=(.0529*AKGS)/144.
10890 TAK=ARKS+ARM0+ARN1+ARNL+ARKG

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10900 FLX(I,J)=-TAN*OF(I,J)
10910 100 CONTINUE
10920 110 CONTINUE
10930 00 310 I=1.40
10940 SUM=0.0
10950 CALC. THE AXIAL HEAT CONDUCTION AT EACH AXIAL STATION
10960 00 320 J=1.13
10970 320 SUM=SUM+FLX(I,J)
10980 310 APLX(I)=SUM
10990 RETURN
11000 FNO

```

Sample Data Listing

[illegible]

Probe
↓