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TESTS ON HALF-SCALE FLOW MODEL OF 40-MW(E)
PROTOTYPE HTGR (PEACH BOTTOM ATOMIC
POWER STATION)

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REACTOR TECHNOLOGY

TESTS ON HALF-SCALE FLOW MODEL
OF 40-MW(E) PROTOTYPE HTGR
(PEACH BOTTOM ATOMIC POWER STATION)

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ABSTRACT

A half-scale clear plastic nonnuclear flow model of the 40-Mw(e) Peach Bottom High-temperature Gas-cooled Reactor (HTGR) vessel and internals was operated during the period from January, 1961, through October, 1961. The model was operated as an induced system using ambient air as the working fluid. The maximum Reynolds number achieved in the model was approximately one-half of the Reynolds number corresponding to the full-load design conditions in the prototype.

The original main objectives of the flow-model test program were:

1. To determine the pressure-drop characteristics of the reactor internal flow paths.
2. To study the flow patterns in the gas-coolant passages.
3. To measure how effectively the inlet gas would cool the reactor vessel wall.
4. To check the stability of the core and the core-restraining capabilities of the tilting reflector blocks.
5. To determine the effectiveness of the core-reflector seals.

The significant results obtained were:

1. The prototype reactor pressure-drop values extrapolated from the flow-model data indicated a pressure drop of 2.0 psi for a helium flow rate of 468,000 lb/hr and pressure of 350 psia, and a constant inlet and outlet temperature of 650°F. The corresponding conservatively calculated pressure-drop value was approximately 2.9 psi.

2. No areas of serious flow starvation were observed within the model during tests with flow through only one nozzle or through both nozzles. The inlet flow divided almost equally into the upward and downward directions. Regions where low velocities were indicated appeared to be turbulent and free from stagnation.

Completely closing the four off-center openings in the top head reduced the flow of air from the outer flow jacket to the inner flow jackets by only about 20%. This result supported the design approach of providing only one nozzle in the center of the top head for flow from the outer to the inner flow jackets.

3. The heat-transfer coefficients measured at the inner surface of the pressure vessel varied over a range from 50 to 200 Btu/(hr)(ft²)(°F),

at the design flow rate, and in general were about twice the calculated values for corresponding points.

4. The tilting reflector is a workable concept. No vibration or movement could be induced in the core by manual manipulation of various reflector blocks. There was no detectable vibration of the core during any mode of flow-model operation.

It was decided not to try to measure leakage across the reflector seal because the surfaces of the seal provided in the model were not felt to be comparable to those that will exist in the prototype, and accurate measurement of the leakage would have been very difficult. Results would have been of limited value because the seal design has now been changed.

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NOMENCLATURE

SYMBOLS

- A = area, ft^2
B. P. = barometric pressure, in. of mercury
 C_i = heater calibration factor
 C_p = specific heat
d = heater calibration constant
D = diameter, ft
E = thermocouple potential, millivolts
g = gravitational constant, 32.2 ft/sec^2
h = heat-transfer coefficient, $\text{Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$
i = surface-heater number
j = probe position
k = thermal conductivity, $\text{Btu}/(\text{hr})(\text{ft})(^\circ\text{F})$
K = loss coefficient
 ℓ = traverse position
L = length of heat-transfer surface, ft
M = mass flow, lb/sec
 M' = mass flow per foot of circumference, lb/sec/ft
N = Nusselt number
P = pressure, psi
Pr = Prandtl number
Q = volume flow, ft^3/sec
R = Reynolds number
s = heater calibration Reynolds number exponent
t = temperature, $^\circ\text{F}$
T = gap thickness, ft
V = velocity, ft/sec
 $\bar{V}$ = average velocity, ft/sec
v = specific volume, ft^3/lb
W = power, watts
y = head, in. of specified fluid
z = manometer level, in. of water
 μ = viscosity, lb/hr-ft
 ρ = density, lb/ft^3
 $\bar{\rho}$ = average density, lb/ft^3
 θ = direction of flow measured clockwise from perpendicular in a vertical plane, degrees

SUBSCRIPTS

a = air
he = helium
e = equivalent
in = inlet
m = model
out = outlet
p = prototype
r = reference atmospheric conditions
s = static or surface
t = total
v = velocity
w = water
x = conditions existing during test

I. INTRODUCTION

DESCRIPTION OF PROTOTYPE

The Peach Bottom High-temperature Gas-cooled Reactor (HTGR) pressure vessel and internals are to be maintained within safe temperature limits by the circulation of helium gas at a controlled temperature and pressure. Figure 1 shows the prototype reactor design which closely resembles the design at the time the flow model was constructed. Several modifications were subsequently made, but the basic arrangement in the final design remains the same.

The reactor pressure vessel is a vertical 14-ft ID carbon-steel cylinder with elliptical heads. The height is 35 ft 6-1/2 in. over-all, including the heads.

Helium coolant, at 350 psia and 634°F, enters the reactor vessel from the outer annulus of the concentric pipe circuit. By using the thermal shields and plenum shroud as baffles to direct the incoming gas along the vessel wall, the incoming gas removes both the nuclear heat generated in the vessel wall and the heat lost from the plenum shroud to the vessel, keeping the vessel wall at a temperature less than 700°F.

The incoming coolant flow is split as it enters the vessel--approximately 50% flowing upward and 50% flowing downward. The upward flow travels up along the vessel wall to the fuel-handling-equipment nozzles in the top head. At these nozzles, it turns down into the two annular spaces between the thermal shields and plenum shroud. (The final design has only one thermal shield in this region, thus eliminating one of the annular spaces.) After flowing down these annular spaces, the helium passes down between the reflector blocks.

The other 50% of the incoming coolant flows downward from the inlet nozzles along the vessel wall, then between the lower thermal shield and the vessel wall, and so into the space below the core support plate. A small portion of this flow is passed through cooling holes in the support plate and then into the core. (The final design has a solid support plate with no cooling holes.) The remainder passes up and around the periphery of the core support plate. At the periphery of the plate, the flow joins the gas coming down through the reflector blocks. The combined flow passes into the core and then up into a plenum above the core.

Fig. 1-- Early prototype reactor design

From the plenum, the gas passes into the inner concentric pipe and then on to the steam generators, exiting from the reactor at a temperature of approximately 1400°F.

The circumference of the core is surrounded by a neutron reflector of graphite approximately 2 ft thick, extending the full length of the fuel elements. This reflector consists primarily of two concentric rings, formed by graphite segments. The inner ring of segments is movable, each segment being pivoted at the bottom in such a manner that it can move radially inward. This movement allows the reflector to provide lateral support to the fuel elements near their upper ends. The force providing this firm yet thermally expandable restraint on the core results from the coolant pressure differential existing between the outer and inner faces of the tiltable reflector segments. Since gaps exist between the reflector segments, and the top of the reflector is not rigidly attached to the plenum shroud, seals are provided to prevent the coolant from bypassing the core.

NEED FOR THE MODEL

Stress analysis of the vessel indicated that thermal stresses in the pressure vessel would not exceed allowable limits unless a complete lack of heat transfer by convection should develop over an area larger than 3 ft in diameter.

Because of the three-dimensional character of the flow paths of the HTGR, it was extremely difficult to make an analytical prediction of flow distribution with any reasonable degree of accuracy. The accuracy of the calculated dynamic and frictional losses incurred by the fluid on passing through such a heterogeneous conduit was questionable, since these losses could be calculated only by the use of a series of velocities averaged over corresponding volume shapes and by selecting corresponding loss coefficients from the best available experience, subsequently adding these many local losses to obtain the total losses. Knowing the over-all pressure drop was of considerable importance for selecting the gas-circulating equipment. An experimental check on the pressure drop across the reactor section of the loop appeared very valuable.

The HTGR core and reflector design is unique in the manner in which freedom for thermal expansion of the components is provided by the tilting reflectors. Analyses indicated that the configuration was stable; however, the importance of core stability was considered so great that tests to confirm this stability were required.

For these reasons, it was considered very useful to establish a confidence factor for the flow and heat-transfer distributions, pressure

drop, and core stability of the HTGR prototype by studying the flow patterns in a geometrically similar model operating in the turbulent-flow condition.

BASIS FOR SELECTION OF EQUIPMENT

A study was conducted to determine the model size and the fluid to be used. In view of the problems in maintaining tolerances during manufacturing of a small model and the increased accessibility in a larger model for performing tests other than flow studies, as well as consideration of the blower requirements for a small model, it was decided to build the model to one-half scale. Plastic was chosen as the material, both from an economic standpoint and because of the practical advantage of being able to observe flow patterns with smoke. The practical choice for the fluid for a model of this size was air. Air at atmospheric pressure and room temperature also had the advantage of having dynamic properties similar to those of helium at prototype-design conditions, i. e., 350 psia and 650°F.

Operation of the flow model as an induced rather than pressurized system was considered desirable because of the strength properties of the plastic model and because such operation permitted the use of a lower-powered blower motor. The model was chosen to operate as an open loop, taking its suction from the atmosphere and discharging through the blower back to the atmosphere.

The selection of a blower size capable of producing about one-half the prototype full-load-flow Reynolds numbers was based upon the assurance of recognized fluid-mechanics authorities that there would be no trouble in extrapolating data from a model operating at 30%-flow conditions to the full design flow. In addition to the increased cost and greater power requirements of a larger blower, there was the additional consideration of the increased effects of compressibility at higher flow rates, which would have required tedious density corrections. Another factor was the increased strength of the plastic model that would have been required for flow rates higher than 50%.

TEST PROGRAM--GENERAL REMARKS

A plastic half-scale model was erected at General Atomic, and flow tests using this model were performed during the period January, 1961, through October, 1961. The maximum air flow through the model produced Reynolds numbers corresponding to about one-half the Reynolds numbers produced by full-load flow of helium through the prototype.

Data from the flow-model tests and a discussion of the extrapolation of these data to prototype conditions are presented in this report. It is very important that the extrapolated data be considered valid only for a prototype that is geometrically similar to the test model. Inasmuch as a number of changes in prototype design were made during and since model operation, special consideration should be given to the effects of these changes on the data extrapolated from results of flow-model tests.

The value used for the prototype-design helium flow rate throughout this report was 468,000 lb/hr. Present data indicate that 440,000 lb/hr is a more realistic value; however, the 468,000 lb/hr figure was used throughout this report so that direct comparisons among all the earlier and later analyses could be made.

All static-pressure readings were corrected to an arbitrary "standard" atmospheric pressure of 30 in. of mercury, temperature of 60°F, and relative humidity of 60%. The manner in which this correction was made is discussed in Section IV and in Appendix A.

II. DESCRIPTION OF APPARATUS

FLOW-MODEL TEST FACILITY

The flow-model test site, shown in Fig. 2, contained a fully integrated facility for large-scale tests of flow distribution, pressure drop, and convective heat transfer. The system was an open loop, using ambient air as the working fluid. The flow-model vessel, thermal shields, plenum shroud, and reflectors were made from clear acrylic plastic. The core support plate and fuel elements were made of aluminum. The layout of the flow model and locations of test probes is shown in Fig. 3. Figure 4 is an over-all photograph of the model before installation of the test probes, and Fig. 5 is a photograph looking down into the core area with the top head removed.

Air was drawn through the model by a centrifugal blower, driven by a 250-hp motor, capable of delivering 16,800 ft³/min at 71 in. of water vacuum and at 70°F. The blower was connected to the outlet side of the plastic model. Outlet-flow monitoring was provided by a recording, two-pen pressure gauge. Flow metering in each outlet duct was accomplished by means of an 18-in. venturi connected to a controller. The controllers were connected to 18-in. butterfly valves with pneumatic actuators. When the model was operated at full capacity, total flow controllability was within ±5% and repeatability within ±2%. Flow balance between the two outlet ducts was within ±5%.

PRESSURE INSTRUMENTATION

Test instrumentation was installed so that static pressures could be simultaneously indicated for a maximum of 400 different points. This indication was on four banks of manometers with separate variable-height reservoirs. The four manometer boards were set up to read gauge pressures to ±69 in. of water and were back-lighted for recording pressures by means of photographs. A photograph of a typical manometer board is shown in Fig. 6.

The manometers were connected to static-pressure probes at the desired points with Tygon plastic tubing. A schematic diagram of a static pressure probe appears in Fig. 7.

Fig. 2--Drawing of HTGR flow-model test facility

Fig. 3 (continued)--Drawing of the one-half scale HTGR flow model

Fig. 4--Photograph of the HTGR plastic flow model

Fig. 5--Photograph of the HTGR flow-model interior (top head removed)

Fig. 6--Photograph of a vertical manometer board, showing typical data record of static- and total-pressure measurements

Fig. 7--Schematic diagrams of probes used to measure static pressure, total pressure, and temperature

VELOCITY INSTRUMENTATION

Velocity was measured by using directional total-pressure probes, each in conjunction with an adjacent static-pressure probe. A schematic diagram of a total-pressure probe is shown in Fig. 7.

A panel containing 10 inclined manometers reading to ± 2.00 in. of water and 10 manometers reading to ± 0.50 in. of water was provided for reading velocity pressures. Each of these inclined manometers was connected through a header to three test points, and each of these test points was also indicated on the vertical manometers for photographic record. A total of 50 total-pressure probes was available for use with the inclined manometers.

Thirteen of the fuel elements were instrumented in order to obtain records of the static and velocity pressures at both the top and the bottom of each element. The total-pressure probes used with the fuel elements consisted of a small (0.033-in. ID) steel tube permanently attached to each fuel element.

SMOKE-FLOW EQUIPMENT

The test facility was equipped for flow-visualization studies, using a noncorrosive, nonplating gray smoke. (The smoke proved to be mildly corrosive to the aluminum parts.) The smoke-generation system is shown in Fig. 8a. For the tests, a smoke bomb was placed in the hood and ignited. The blower pulled the smoke out of the hood and forced it through the flexible duct to a manifold, which had connections for 10 smoke probes. A petcock for draining residue was provided in the manifold. Each of the ten outlets could be turned on or off as desired. The smoke entered the flow model at the desired locations through the smoke probes, one of which is shown in Fig. 8b. The bypass, or return, line enabled smoke density to be varied by opening or closing the butterfly valve.

The commercially available smoke bombs, which are commonly used for fire-fighting training, produce a light-gray smoke with a minimum of residue and burn for approximately three minutes.

HEAT-TRANSFER INSTRUMENTATION

Convective heat-transfer coefficients were measured by inserting probes through the model walls to the inner surfaces at the points of interest. Each probe (see Fig. 9) consisted of a small, concentric electric patch heater insulated from an outer guard heater and instrumented with

Fig. 8a--Schematic diagram of smoke-generation system

Fig. 8b--Schematic diagram of smoke probe

Fig. 9a--Schematic diagram of original probe for measuring heat-transfer coefficient

Fig. 9b--Schematic diagram of modified probe for measuring heat-transfer coefficient

thermocouples to indicate patch-heater and guard-heater surface temperatures. The heat-transfer-coefficient probe as originally constructed and used is shown in Fig. 9a, and the probe, as modified by relocating the patch-heater thermocouple, is shown in Fig. 9b. By measuring local air temperatures (measured at four points in the model with thermocouple probes of the type shown in Fig. 7), it was possible to calculate approximate film coefficients. A total of 60 calibrated probes was available, with provisions for simultaneously connecting 40 to a portable instrument console for reading. The console had two potentiometers for use with surface and fluid thermocouples, two sets of dial indicators for heater power, and a battery power supply for the test heaters.

III. METHODS OF TESTING

The same general test procedure was used for all tests. After all instrumentation was connected for a specific run, the blower was turned on and the flow was adjusted to the prescribed settings. The flow rate was checked and adjusted periodically during each run. For tests with flow through both nozzles, the flow was adjusted to be equally divided, or balanced, between the two. For tests with flow through one nozzle only, the butterfly valve on the opposite inlet was closed.

The same values for the flow rate were used, whenever possible, for all tests. For flow through each nozzle, these flow rates were usually 2, 4, 6, 8, or 10 in. of water pressure differentials across the venturi meters. The maximum differential obtained for a two-nozzle flow was approximately 15 in. of water. For flow in only one nozzle, readings as high as 25 in. of water were obtained.

After the flow rate was adjusted to the desired value, a few minutes were allowed for the system to stabilize before data were taken. The period of time that the flow rate was maintained at a given value was varied depending on the type of data being taken. There was no set time interval for recording data. For static-pressure measurements, the data could be gathered by photograph in a few minutes, whereas tests such as velocity traverses and heater readings took considerably longer.

STATIC-PRESSURE MEASUREMENTS

The threads on the static-pressure probes were sealed with a thin film of silicone grease before they were screwed into the holes in the vessel. Tygon tubing was connected from each probe to a manometer board. After the desired flow rate was reached and the manometers had settled down, the static pressure was recorded by photographing the manometer boards; for the most part, the photographic recording was very successful. Static pressures could be read from 10-in. by 12-in. enlarged photographs to within ± 0.1 in. of water.

The wet- and dry-bulb temperatures of the ambient air, barometric pressure, relative humidity, and temperatures of the air in the model were recorded at frequent intervals. The absolute pressure at each venturi was also recorded during each run.

VELOCITY TRAVERSES

The general procedure for making velocity traverses was very similar to that for taking static-pressure readings. Before inserting a velocity probe, the width of the gap was measured, and the probe was marked for eight, equally spaced traverse depths. A Tygon line was run from each total-pressure probe to a vertical manometer, and a second line was run to the high-pressure side of an inclined manometer. The adjacent static-pressure tap was similarly connected to a vertical manometer and to the low-pressure side of the inclined manometer. The inclined manometers were provided with three-way valves so that pressures at three locations could be read on the same manometer. After determining the direction of flow by observing smoke patterns or by using wool yarn attached to the surface of the flow passages, the total-pressure probes were aimed directly into the flow. The flow direction was verified by rotating the probe to the position corresponding to the maximum manometer reading.

Pressures were recorded with all probes positioned at the same depth. Then all probes were moved out to the next depth position and the pressures were again recorded. This process was repeated until the gap traverse was completed.

The vertical-manometer-panel readings were recorded by photograph, and the inclined-manometer readings were entered into data sheets by test personnel. A record was kept of the probe directions and gap dimensions.

The wet- and dry-bulb temperatures of the ambient air, barometric pressure, relative humidity, temperature of the air in the model, and venturi absolute pressures were also recorded for each velocity traverse test.

SMOKE TESTS

The smoke probes, which were usually placed in a horizontal plane around the model, were connected to the smoke manifold with Tygon tubing. A maximum of eight probes were used at a time. The smoke direction was adjustable and was usually oriented downstream. Often, the valves to some of the probes were closed in order to emphasize the smoke patterns at the open probes. The flow rate was first established at about 2 in. of water. The smoke bomb was then ignited and placed on holders in the smoke hood after the smoke-hood blower was turned on. Smoke patterns were then observed visually at selected areas of the model and were recorded by sketches. The selected areas were also photographed on 16-mm movie film.

The initial studies were made with eight nozzles opened simultaneously; later, single nozzles were opened intermittently. This enabled study of an entire area, followed by a localized study. Another procedure used consisted in filling the study area with smoke from the nozzles before turning on the flow-model blower. In this way, the smoke dispersion could be watched. This method indicated that the smoke patterns did not change once the flow was established. All studies were recorded on film and corresponding information was entered in the data books. Complete flow studies were made of the outer jacket, with dual- and single-nozzle flows and with the top head in normal and in 45° counterclockwise positions.

HEAT-TRANSFER-COEFFICIENT MEASUREMENTS

The general operating procedure for heat-transfer tests was much the same as for the static-pressure and velocity measurements. The same supplementary data of temperatures and absolute pressures were recorded. The thermocouples were checked and the potentiometers were standardized before each test.

After the desired flow rate was reached, power to both the patch heater and the guard heater of each probe was turned on. The patch temperature was allowed to rise until it was 10° to 20°F above the air-stream temperature inside the model. The temperature of the probe wall near the guard heater was maintained equal to that of the patch in order to eliminate heat loss through the wall of the probe. The heaters were allowed to stabilize for about 5 min. before readings were taken. Temperature of the probe patch and wall and voltage and amperage to the patch heater were recorded. It was not difficult for an operator to become adept at maintaining the heater temperatures in balance so that the heater readings could be obtained with a minimum of delay.

IV. DISCUSSION OF RESULTS

STATIC PRESSURE AND PRESSURE DROP

Results of the flow-model tests indicated a drop in total pressure through the prototype vessel of 2.0 psi at the design full-load flow of 468,000 lb/hr of helium at a pressure of 350 psia and at a constant temperature of 652°F. (The flow rate of 468,000 lb/hr was used for all pressure-drop extrapolations.) At the more recently adopted value for full-load flow of 440,000 lb/hr, this pressure drop would be reduced to 1.78 psi. If the losses due to heating of the helium in the reactor fuel channels of the prototype, amounting to 0.6 psi, are added, the total pressure loss for normal operating conditions would be about 2.4 psi at a prototype helium flow rate of 440,000 lb/hr. This would compare to a calculated value of 3.2 psi for design operating conditions at a flow rate of 440,000 lb/hr.

The 468,000 lb/hr helium flow rate was used throughout the pressure-drop calculations to expedite comparison with all the earlier analyses. It is a simple matter to find the pressure drop at any desired flow rate or density, since the pressure drop is proportional to the square of the mass flow rate and is inversely proportional to the density.

A flow rate of 17.83 lb/sec of air through each nozzle of the model would produce Reynolds numbers in the model which would correspond to those in the prototype operating at the design flow of 468,000 lb/hr of helium. The maximum values obtained during the model tests were about one-half of this. The pressure-drop values which are quoted at design flow were extrapolated from the model data obtained for tests run at a maximum flow rate corresponding to about one-half the design value. The procedure used to express the flow-model pressure drop in terms of the prototype pressure drop is outlined in the sample calculations in Appendix A.

Since atmospheric conditions were not identical for tests performed at different times, it seemed advisable to correct all pressure-drop readings to some arbitrary standard atmospheric density for purposes of comparison of the several test measurements. The "standard atmosphere" used in this program was 0.0752 lb/ft³ density, 30 in. Hg pressure, 60°F temperature, and 60% relative humidity. Description of the correction method and its justification has been included in the sample calculations in Appendix A. In general, the method states that the pressure drop through a model is proportional to the square of the velocity (velocity head) at all

points through the model. If some arbitrary point of reference (the inlet) is selected, then the total drop through the model is proportional to some constant times the velocity head at this point. Since this constant is primarily a function of geometry, it should remain essentially the same for minor changes in inlet conditions. Thus, changing the atmospheric conditions while holding the flow rate constant amounts to predicting what the pressure drop (manometer reading) would be at the given point (inlet or outlet) if the flow-model system were operated with a standard-density atmosphere rather than the actual atmosphere existing at the time of the test.

The data taken in February are compared with those taken in May, 1961, in Fig. 10. Prior to the May tests, straightening vanes were installed in the inlet ducts, spacer blocks near the inlet nozzles were removed, and the fuel rods were completely cleaned. There was a noticeable reduction in the pressure loss, comparing the results from the earlier with those of the later tests; across the fuel elements alone, the reduction was significant. The conditions of the model during the May tests are believed to be more representative of prototype conditions than the conditions during the February tests.

When the curves of static-pressure drop (see Fig. 10) were extrapolated to a flow rate of 17.83 lb/sec per nozzle, the static-pressure drop for the May data was 118 in. of water. For the February data, the extrapolated value was 125 in. of water for a curve with a slope equal to that of the May data. The prototype pressure drop corresponding to the 118 in. of water static-pressure drop was calculated by using both the average density through the model and the density at the inlet to the vessel; the respective values for prototype pressure drop were 1.81 and 2.13 psi.

Since the model inlet was arbitrarily selected as the point of reference, it might seem reasonable that the density at this point should have been used in computing the pressure drop through the prototype from the model data. However, it should be noted that the pressure drop through a model is a function of the local conditions of density and velocity existing at each individual point in the model. Although it was not possible to know these exact conditions throughout the model, it could be hypothesized that the total pressure drop was more nearly proportional to the average density from model inlet to outlet. If this were true, it made no difference that the air in the model was expanding due to the decreasing pressure, since the pressure loss was a function of local conditions and not of those at the inlet. Thus, computing the pressure drop on the basis of the average density through the model should have given an approximation to the drop in the prototype, where there will essentially be a constant density for the isothermal-flow conditions.

Since the densities within the flow model varied with the flow rate, the densities which had to be used in computing prototype pressure drop

Fig. 10--Static-pressure drop from inlet to outlet and across fuel elements (balanced two-nozzle flow through flow model; readings corrected to standard atmosphere)

were those that would exist if there were to be the prototype full-load-flow Reynolds number at the inlet. Inlet, outlet, and average densities extrapolated to full Reynolds number flow are plotted in Fig. 11.

The approximate static-pressure profile through the model for one-nozzle flow and for two-nozzle flow are plotted in Figs. 12a and b, respectively. The section from the top of the reflectors to the bottom head was not shown in detail because in the final prototype design one of the thermal shields has been eliminated. Actually, there will be loss and subsequent recovery of static head in this section of the vessel, as a result of convergence and divergence of flow.

Figure 13 is similar to Fig. 10, except that it compares the static-pressure drop for one-nozzle flow with that for two-nozzle flow. As may be seen in Fig. 13, the drop at corresponding rates of total flow through the model was almost identical.

A loss coefficient, K , for the model was evaluated using the same method as that for correcting the pressure readings to a standard atmospheric density. The prototype loss coefficient was obtained by extrapolating the model loss coefficient to the full-load flow rate. The values of K for both one- and two-nozzle flow are shown in Fig. 14. For two-nozzle flow, K was equal to 9.0 when computed using inlet density, and 8.0 when computed using average density.

Since densities and areas at the inlet and outlet were slightly different, it was more accurate to take velocity pressure into account and to express pressure drop in terms of total pressure rather than static pressure. This was done in calculating the values shown in Figs. 14 through 18.

In evaluating K , the question again arose as to whether the average density through the model or the density at the inlet should have been used. The argument previously given still holds; the coefficient evaluated for average density should be the closer approximation to the coefficient for flow of essentially incompressible fluid in the prototype. The loss coefficient was evaluated for both average and inlet densities. Prototype pressure drop using average density was 1.94 psi, compared with 2.18 psi using inlet density.

It was interesting to ask what would happen if standard air should expand ideally until the value of measured pressure drop were reached and if the resulting pressure drop were then corrected to that for incompressible flow, maintaining the assumption that the loss with a changing density is proportional to the average density between the atmosphere and the point in question. These calculations were performed for isothermal and adiabatic expansions, and the results are shown in Fig. 15. Loss coefficients were

Fig. 11--Air density versus air flow rate, extrapolated to full-load Reynolds number flow (readings corrected to standard atmosphere)

Fig. 12a--Static-pressure profile in flow model for one-nozzle flow of 9.35 lb/sec through east nozzle only (2/28/61 test)

Fig. 12b--Static-pressure profile in flow model for balanced two-nozzle flow of 8.9 lb/sec/nozzle (5/8/61 test)

Fig. 13--Static-pressure drop from inlet to outlet for one-nozzle and for two-nozzle flows through model (readings corrected to standard atmosphere; 2/28/61 test)

Fig. 14--Loss coefficient for drop in total pressure through flow model

Fig. 15--Drop in total pressure, inlet to outlet, for balanced two-nozzle flow; pressure drop corrected from densities with ideal expansion to incompressible flow (readings corrected to standard atmosphere; 5/8/61 test)

Fig. 16--Loss coefficient for drop in total pressure corrected from ideal expansion through model to incompressible flow (5/8/61 test)

Fig. 17--Drop in total pressure, inlet to outlet, for balanced two-nozzle flow; pressure drop corrected from densities at measured pressures to incompressible flow (readings corrected to standard atmosphere; 5/8/61 test)

Fig. 18--Loss coefficient for drop in total pressure corrected from measured densities to incompressible flow (5/8/61 test)

also calculated, and these results are shown in Fig. 16. There was, of course, now no question about which density to use in calculating the loss coefficient or the prototype pressure drop, since the readings were corrected to incompressible flow; the density of the standard atmosphere was used. The results of these calculations are summarized in Table 1.

Table 1

SUMMARY OF PRESSURE DROP THROUGH HTGR PROTOTYPE, AS CALCULATED BY SEVERAL METHODS (NOT INCLUDING CORE-HEATING LOSS)

	Pressure Drop		Pressure Drop Calculated from Loss Coefficient, K	
	Ref. Fig. No.	Extrapolated to Full Flow (psi)	Ref. Fig. No.	Extrapolated to Full Flow (psi)
Two-nozzle Flow (468,000 lb He/hr)				
Static-pressure drop corrected to standard atmosphere using—				
Average density	10	1.81		
Inlet density	10	2.13		
Total-pressure drop corrected to standard atmosphere using—				
Average density			14	1.94
Inlet density			14	2.18
Total-pressure drop corrected from ideal expansion to incompressible flow for—				
Isothermal expansion	15	2.13	16	1.98
Adiabatic expansion	15	2.27	16	2.13
Total-pressure drop corrected from measured densities to incompressible flow	17	2.25	18	2.01
One-nozzle Flow (234,000 lb He/hr)				
Pressure drop corrected to standard atmosphere using—				
Average density	13	0.62	14	0.58
Inlet density	13	0.59	14	0.58

It was now logical to ask "why not take the data for the actual expansion through the model and calculate values for pressure drop to individual points, adjusting them to incompressible flow as was done for isothermal and adiabatic expansions?" The results of such calculations are shown in Fig. 17; the corresponding prototype pressure drop was

2.25 psi. Values of K were also calculated using the pressure drops corrected to incompressible flow and standard air density. These data are shown in Fig. 18; the corresponding prototype pressure drop was 2.01 psi. This value is considered to be the most refined of all. The extrapolation of K in Fig. 18 is considered better than the extrapolation of pressure drop in Fig. 17 because it was not necessary to use logarithmic paper.

This method for computing prototype pressure drop was essentially the same as the average-density method used earlier and shown in Figs. 10 and 14. The main distinction was the manner of taking the average density into account. In the earlier method, the actual pressure drop across the model (corrected to standard atmosphere) was used in conjunction with the average density across the model. This combination was used in Fig. 10, where the actual pressure drop across the model was extrapolated. The same combination was used for computing the loss coefficient values for Fig. 14. In the later method, used in the calculations for Figs. 15 through 18, the pressure losses from atmospheric pressure to the inlet and to the outlet pressures were predicted for incompressible flow on the basis of the average density between atmosphere and the inlet or outlet. Then the loss between these points for incompressible flow was used in conjunction with the standard atmospheric density.

Values for prototype pressure drop calculated by the various methods are listed in Table 1. It may be noted that the difference between any two values is not very great. Although it is not readily apparent from the pressure-drop summary, the results for model pressure loss computed by the last-discussed incompressible-flow method were very similar to the results for isothermal expansion; this suggests that flow through the model if it had been incompressible might have closely approximated isothermal flow.

VELOCITY, FLOW-DISTRIBUTION, AND SMOKE STUDIES

A study of velocities and flow patterns in the model indicated that the flow divided almost equally upward and downward upon entering the model. Attempts to predict the actual flow distribution in the model based on the limited number of velocities measured were inconclusive; however, diagrams to show the results that were obtained are included along with the discussions of the results.

A wide variation was observed in the magnitude of individual velocities, but smoke testing showed no areas of serious flow starvation. Areas where very low velocities were indicated were found to be turbulent areas. The smoke studies proved to be a great aid in the decision to eliminate the four off-center openings of the top head. Movies of the smoke patterns

proved useful, as details in the movies could be studied to bring out information not observable at the time of testing.

The primary points at which velocity data were taken were

1. Around the east and west inlets.
2. In the outer passage on a plane 30 in. above the inlet.
3. In the outer passage on a plane 30 in. below the inlet.
4. In the outer passage on a plane 48 in. below the inlet.
5. In the outer passage on a plane near the bottom of the core.
6. On selected dummy fuel elements.

The average velocities around the circumference of the east and west inlet annuli, where eight point traverses were taken at eight stations equally spaced around the circumference, are shown in Figs. 19 and 20.

All point velocities were calculated according to the formula

$$V = 18.25 \sqrt{\frac{y}{\rho}} ,$$

where y is the velocity head, in inches of water, and ρ is the local air density, in lb/ft^3 .

The average velocities were determined from velocity profile graphs with the aid of a planimeter. Profiles plotted for each station indicated typical, well-established, turbulent-flow patterns with flat fronts; however, a few of the profiles were quite irregular. There is a striking similarity between Figs. 19 and 20--the peak of high velocity in both inlets was in line with the upper outside region of the elbows in the inlet ducting. It should be kept in mind when examining these two figures that the key diagram is being viewed from the inside of the vessel looking out.

It was felt that the variation in velocity around the inlet duct had an influence on the flow distribution in the model. Consequently, in April, 1961, straightening vanes were installed in the inlet ducts to decrease the velocity variations.

Several methods were tried for establishing flow distribution patterns in the outermost gap of the reactor vessel based on velocity data. The most significant point of agreement among the various methods was that the flow split about equally upward and downward in all cases. The most promising method was an attempt to find the upward or downward component of mass flow at each point using the expression

$$M' = \rho \bar{V} \cos \theta T \quad \text{lb/sec/ft of circumference ,}$$

where ρ is the density, in lb/ft^3 ; $\bar{V}$ is the average velocity of each point,

Fig. 19--Profile of average velocities at east inlet duct; venturi flow at east and west inlets was 10 in. of water (3/6/61 test; calculated mass flow, 8.30 lb/sec; annular area, 0.984 ft²; air density, 0.0735 lb/ft³)

Fig. 20--Profile of average velocities at west inlet duct; venturi flow at east and west inlets was 10 in. of water (3/6/61 test; calculated mass flow, 8.38 lb/sec; annular area, 0.988 ft²; air density, 0.0735 lb/ft³)

in ft/sec; θ is the angle from vertical clockwise to the direction of flow, in degrees; and T is the thickness of the gap, in feet.

Plots of the outer-jacket velocity components and average velocities for the May, 1961, tests with flow in both nozzles are shown in Fig. 21. Velocity profiles for each individual point covered in Fig. 21 are included in Appendix B, Figs. 33 and 34. Most of the profiles show flat-faced patterns, indicating turbulent flow.

An average of the vertical components of mass flow in the outer passage at the eight points in each plane gave a relative indication of 54% of upward and 46% downward flow, for a test run on May 10, 1961, with a flow rate per nozzle of 8.95 lb/sec. The total flow computed from these data agreed within 10% with the flow indicated by the venturi meters. Computations for other tests by methods using the same basic idea produced similar results, and some of the March, 1961, tests showed even better agreement.

An attempt was made to predict flow patterns from the velocity data to compare with the smoke-test patterns. The most successful method found was to divide the circumference of the vessel into segments proportional to the mass flows calculated for the individual points. The results for one test in March and one in May are shown in Figs. 22 and 23. Areas that were not near the velocity points were sketched in on the basis of observations made during the smoke tests. Since there was such a wide variation in velocity around the model, the representative area did not always coincide with the velocity from which it was calculated. The sketches were somewhat imaginative and should not be interpreted as being a complete representation of actual flow distribution.

Many peculiar patterns observed during the smoke tests were confirmed by the velocity tests. Figure 22 shows a region in the top section where the flow was apparently downward. This caused difficulty in interpreting the March data with respect to distinguishing the upward flow from the downward flow. In Fig. 23, for the May data, the flow in this region appears to be more nearly as expected. This suggests that the outer-jacket spacer blocks, which were positioned near the entrance nozzles of the model during the March tests, had a pronounced effect on the flow distribution. These spacers were removed prior to the May tests since there were no similar spacers in the prototype. Apparently as a result of their removal, a complete change took place in the area above the east nozzle, where the highest velocities occurred in March and the lowest velocities occurred in May.

The velocity, direction, and magnitude for outer-gap tests run with flow through only one nozzle are shown in Figs. 24 and 25. These tests indicated that there were no areas with a complete lack of flow during flow

Fig. 21--Upward and downward components of velocity in outermost annulus for balanced two-nozzle flow; arrows indicate actual directions and magnitudes of flow

Fig. 22--Local velocities and projected regions of equal mass flow, based on 24 points (tests of 3/6/61-3/8/61, with balanced two-nozzle flow of ~ 9 lb/sec of air at each inlet; without inlet straightening vanes and with spacer blocks adjacent to nozzles)

Fig. 23--Local velocities and projected regions of equal mass flow, based on 16 points (tests of 5/10/61, with balanced two-nozzle flow of ~ 9 lb/sec of air at each inlet; with inlet straightening vanes installed and with spacer blocks removed)

Fig. 24--Upward and downward components of velocity in outermost annulus for flow through east nozzle only; arrows indicate actual directions and magnitudes of flow

Fig. 25--Upward and downward components of velocity in outermost annulus for flow through west nozzle only; arrows indicate actual directions and magnitudes of flow

through either nozzle. It is particularly interesting that the general flow pattern remained independent of the nozzle through which the flow was passing, although the direction of flow at individual points changed quite noticeably. It should be noted from Figs. 24 and 25 that the actual magnitude of velocity at several points was quite high, as is indicated by the length of the arrows, but the profile does not indicate this because the vertical components were small. Individual flow profiles for these tests are included in Appendix B, Figs. 35 through 38. In all tests some areas of apparent low velocities were observed, notably near the east nozzle, as is indicated by the velocity diagrams of Figs. 21, 24, and 25. However, visual inspection of smoke patterns in these areas usually revealed a disturbed, swirling, nondirectional, turbulent type of flow with no complete stagnation, but with a small relative component of velocity.

It would be questionable to predict flow conditions at any particular point in the prototype based on the model flow. The eccentricities of the structural parts of the model, which cause a variation in dimensions such as gap widths, are almost certain to differ in the prototype compared with the model. Any attempt to establish a relation between the spacing of parts in the model and the flow distribution did not prove conclusive, but there was no doubt that the variations in channel dimensions did have an effect on the distribution of flow. The test results indicated that the gap tolerance specified in the prototype manufacturing specification will be adequate to prevent any harmful localized flow starvation.

A test was run to study the effects of throttling the nozzles in the top head. The results indicated that the flow distribution changed very little as the four off-center nozzles were throttled. When the four off-center nozzles were closed completely, the flow to the inner passages was decreased by only about 20% and the increase in pressure drop was negligible. These results confirmed that there was no need to provide control of the top-head nozzle openings in order to allow for a change in the leakage through the seal at the top of the reflectors. As a result, it was decided to eliminate the four off-center nozzle openings in the top head of the prototype and to provide only the center nozzle for flow from the outer to the inner passage. A flow balance valve will be provided in this nozzle for initial adjustment of the flow distribution in the prototype.

During the throttling tests, flow rates down through the inner, middle, and outer gaps were measured. A mass flow balance with the total flow indicated approximately 45% down the outer gap, 22% down the middle gap, and 33% down the inner gap. Further investigation in this area was not indicated, as these flow paths were modified by prototype design refinements, namely, the elimination of one thermal shield, insulation of the plenum, and provision of a valve for initial adjustment of flow in the remaining inner gap.

HEAT-TRANSFER STUDIES

In the zones studied, the flow-model-test heat-transfer coefficients extrapolated for helium at prototype-design conditions showed a wide band of values at prototype full-load Reynolds number ranging from 50 to 200 Btu/(hr)(ft²)(°F). In most zones, this was approximately twice the results of the other methods of calculation where test heater data were not used. Details and comparisons are shown in Table 2; each zone should be considered separately.

Table 2
ZONE AVERAGES OF THE HEAT-TRANSFER COEFFICIENTS
FOR HELIUM FOR PROTOTYPE FULL-LOAD FLOW

Zone	(1)	(2)	(3)	(4)	(5)	(6)
	May Test Calc.*	Elec. Analog	March Test	May Test	March Test	May Test
			Extrapolated by 0.8 Slope		Graphically Extrapolated	
A	68	58	122	117	75	82
B	--	30	132	117	100	96
C(upper)	--	58	119	---	78	--
C(lower)	53	56	99	93	110	55
D	--	171	210	---	160	--
E(upper)	--	---	---	160	---	230
E(lower)	--	---	---	105	---	79
Average	60	75	136	118	105	108

* Calculated using measured test velocities and the Dittus-Boelter equation, and ignoring heater data.

The original heater calibrations were performed using a small rectangular duct. The recalibrations, after the testing, were performed in a larger heated rectangular duct of a diameter equivalent to the flow-model configuration; the recalibrations indicated that the heater calibrations should be increased by factors varying widely and averaging about 1.42 to 1.49. This correction was incorporated into all test data.

The energy equation used for the reduction of test data was based on the electrical power supplied to the heater, the heater-calibration factor determined by experimentation, the difference between heater and air temperatures, and a factor for conversion of units. The equations and procedures are outlined in the following appendixes: Appendix C, "Computer

Programs for Velocity of Flow-model Coolant and for Heat-transfer Film Coefficients"; Appendix D, "Mathematical Method of Computing Flow-model Heat-transfer Coefficients for Air"; and Appendix E, "Simplified Graphical Procedure for Estimating Flow-model Heat-transfer Coefficients for Air." The final-test heat-transfer-coefficient values as presented were for the most part the product of computer program results, September heater recalibrations, and extrapolations to full flow in the prototype. During the program, values were checked by the longhand method; the short-cut graphical procedure was used after the insignificance of some of the variables was established. The test data were extrapolated to full prototype flow by multiplication by the factor of the prototype Reynolds number over the flow-model Reynolds number to the 0.8 power, in accordance with the Dittus-Boelter relation, inasmuch as the influence of temperature, pressure, and humidity variables were insignificant compared with velocity.

The empirical calculations were based on the Dittus-Boelter relation using an equivalent diameter (twice the gap) for an annulus configuration. The Reynolds numbers as measured on the flow model were used in one case, and the velocities determined through the use of an electrical analog were used in another. A graphical presentation of the Dittus-Boelter equation is given in Fig. 26. Extrapolation of the theoretical values to prototype full-flow Reynolds number was based on the ratio of Reynolds numbers to the 0.8 power as were the heater-test data.

Reasons for the variation of results between the theoretical and the test values can be given as follows:

1. The empirical formulas that were available for use in this study were not precisely applicable to a three-dimensional flow model. The L/D_e ratios for many areas of the flow model would have to have been much higher for the equations to apply directly. Based on information from several heat-transfer texts, inlet heat-transfer coefficients can approach twice the values of those at high L/D_e conditions; this approach appears to have been confirmed in the flow model.
2. The greater spread between the calculated data and the test data at the lower flows indicated that a stabilized turbulent condition did not always exist at all points on the flow model.
3. Smoke studies indicated that at many locations the air flow was not unidirectional; because of this, there was no uniform relation between inlet flow and flows at individual heater locations.
4. Velocity data had to be collected at a time different than the heat-transfer runs. Because of significant flow pattern variations with small inlet variations, the flow past individual heaters had to vary also.

Under these circumstances, the factor of two in comparing test results with calculated data is considered well within experimental limits.

Fig. 26--Calculated heat-transfer coefficient for air in the flow model as a function of equivalent diameter and Reynolds number

As the model heater data varied widely, it was grouped, averaged, and compared by various approaches in order to study any possible trends which might be observed. A range of inlet-nozzle flow conditions was covered twice, by 10-, 7-, and 4-in. of water (pressure drop across flow-measuring venturi) dual-nozzle flows in March, and by 10-, 8-, and 6-in. flows in May. These data for the most part were treated separately as the flow patterns were different, the heaters used in some cases were different, the averages of the results differed, and studies in all zones were not repeated. Sample heat-transfer surface maps appear in Appendix F.

The value of 468,000 lb/hr prototype helium flow was used throughout the reduction of flow-model data to expedite the comparison of test heat-transfer results with the earlier analytical calculations. A decrease of approximately 4% of the values would result from the use of the prototype design value of 440,000 lb/hr helium flow.

Comparisons of Prototype Full-flow Heat-transfer Coefficients

The prototype full-flow heat-transfer coefficients obtained by various methods are listed in Table 2 as the averages of all locations within any given zone of the flow model.

The values in column 1 of Table 2 were calculated by using the measured test velocity at each location and by using the Dittus-Boelter relation, ignoring test-heater power readings. Only the more stable 10-in. velocities of zones A and lower C in the May test were studied in this manner.

The values in column 2 of Table 2 were taken from the electrical analog by averaging the values at locations geometrically similar to the flow-model heater locations. Detailed comparisons of electrical analog with flow-model-test results are shown in Figs. 27a and b, which disclose that the electrical analog results were for the most part considerably lower than the test results. In the D zone, the electrical analog data compared most closely with the test results. A reverse relation between the near and far heater locations exists between the March and May test data, the March data (Fig. 27a) comparing more closely with the electrical analog results. It is quite possible that in the May testing a path of less resistance existed to the sides of the inlet nozzle (because of cylindrical curvature) thereby causing high velocities and high heat transfer at far points. The electrical analog data closely compare with the column 1 calculated data.

The values of column 3 of Table 2 and Fig. 28 show the March test results obtained as follows: Heat-transfer coefficients for helium, as derived from flow-model data by the computer program described in

Fig. 27--Electrical analog results compared with flow-model heat-transfer coefficients calculated for helium; (left) March data and (right) May data

Fig. 28--Heat-transfer coefficient for helium versus inlet-duct Reynolds number flow; 0.8 extrapolation from 10-in. flow point to full Reynolds number flow

Appendix C, were plotted at each heater location on a scaled surface map. Values were checked by the mathematical method or by a graphical method. The original calibration multiples obtained from a small rectangular duct for the various patch heaters varied widely but, in the initial programming, necessarily had to be fed into the computer. Later calibration multiples were obtained in September, 1961, on some of the already used patch heaters. As a recalibration was not available on every heater, an average Reynolds number of all available flow-model heater locations was obtained for each flow. The new calibration multiple of each used heater for that average Reynolds number was then compared with the original calibration multiple for that heater and average Reynolds number; an average of these ratios became the best available correction factor to bring the nonrecalibrated heaters up to date. Where available, the exact local corrections were used at the exact Reynolds number on the recalibrated heaters, and the above-described average corrections were used at the remaining heater locations. The 10-in. -data values were then extrapolated through a 0.8 power of the Reynolds number ratio to full prototype flow in accordance with the Dittus-Boelter equation. The averaged values in each zone are shown in column 3 of Table 2. Column 4, for the May data, is similar to column 3.

A study was also performed using average corrections at all heater locations because of the possible influence of the high and low percentages of recalibrations within any given zone. The results of this study, which are not tabulated, showed little difference.

The estimated heat-transfer coefficients for full prototype flow are listed in column 5 of Table 2 and shown in Fig. 29. These coefficients were obtained simply by graphically estimating average lines through the 10-, 7-, and 4-in. water flow points for the various zones. This was done with the realization that although some of these points might be questionable, some general trend might be observed. This approach was not meant to indicate that the power of the Reynolds number in the Dittus-Boelter equation should be varied to apply to the flow model but merely that the data were studied in this manner to indicate a degree of deviation from a hoped-for trend, regardless of reasons.

The data in column 6 are similar to these in column 5, but are for the May data, the 10-, 8-, and 6-in. water flows being used.

Variation of Results for Individual Heaters

After the May testing, an investigation was launched to determine the most likely causes of the observed scatter in the data. Data points for all flows and for all positions of a large number of heaters were gathered and plotted, local Reynolds numbers versus heat-transfer coefficients for

Fig. 29--Heat-transfer coefficient for helium versus Reynolds number flow extrapolated to full Reynolds number flow

air. This plotting was grouped, all data for one heater together, and was also treated separately for different flow-model positions (different equivalent diameters) and different dates of testing. Calculated results (ignoring heater data) were superimposed on the test data. For purposes of simplification, only one heater (10-A) is discussed here.

The results for flow-model heater positions and for the dates tested are shown in Fig. 30, and the test data are compared with results calculated not using heater data. Data for a position in zone A for both the March and May runs are shown in Fig. 30a. The difference in Reynolds number can be accounted for by the removal of the outer-jacket spacers after the March test, and perhaps the nonlinear heat-transfer-coefficient test results can in part be accounted for by the spacers being in place. Note that the 10-in. flow point falls between the 7-in. and 4-in. points. The May results appear reasonable.

Figure 30b shows the data for a C-zone position in the March test and appears to be good data. Note that the Reynolds numbers were spread further than in Fig. 30a. This was most likely caused by the difference in location.

Data from a position in zone C (see Fig. 30c) were more questionable than the A-zone data which included spacer blocks. Note that here also the 6-in. point is between the 10-in. and 8-in. points, although two of these points are close enough together to be interchanged and still be within the Reynolds number band of accuracy. The 10-in. test point is reasonable, the 8-in. is very questionable, and the 6-in. could only be reasonable if the Reynolds number were much lower. One factor should be kept in mind--the velocity data could not be taken at the same time that the heater data were collected. A surprising fact about the data shown in Fig. 30c is that the 10-, 8-, and 6-in. heater data points were run all on the same day, a characteristic not common to some of the other heaters. The scatter shown is typical of much of the test data and most likely was caused by more than one variable.

All the 10-A heater points for all flows and all dates are combined in Fig. 31. The slight spread of the calculated data, where heater power and temperature were ignored, would indicate a higher degree of accuracy of this method. To assess the accuracy of the test data, one would have to consider heater reliability and repeatability, variation of surface velocity between the date of measurement and the date of heater operation, and degree of accuracy of instrumentation.

Top-dome Locations and Time Stabilization

A separate experiment was performed on the flow model in

Fig. 30--Comparison of calculated with computed local-heater Reynolds number versus heat-transfer coefficient for air, at three different locations: (a) in the A zone; (b) and (c) in the C zone

Fig. 31--Heat-transfer coefficient at heater 10-A location, for all flows and all tests

October, 1961, to study heat-transfer-coefficient data at the top dome. An attempt to determine the effect of time stabilization was combined with this experiment. Heaters with little velocity dependence, calibrated in the heated duct, were used in this experiment because local Reynolds numbers and velocity data were not available and the jacket gap varied widely. The results are shown in Fig. 32. The slopes of the curves should not be taken seriously, as it is to be assumed that each location did not change in a degree exactly proportional in local Reynolds number to change of flow-model total mass flow. Smoke tests can be accepted as proof of this.

The top-dome heat-transfer-coefficient test results showed the usual scatter of data of similar flows collected at different dates. (Note that heater 2-B showed a 25% change at the 10-in. flow between the two dates.) Much of the previous flow-model data of similar flows was by necessity collected at different dates.

Thirty-minute stabilization had little effect, and the effect was close to being the same at all three locations. A far greater difference occurred between the same flow at a different date.

CORE-STABILITY STUDIES

The stability of the core with the tilting-reflector-block assembly was qualitatively investigated. A rod was attached near the top (free end) of four representative reflector blocks. The rod projected radially out through clearance holes in the thermal shield and pressure vessel, so that movement of the reflector block could be induced and/or measured from the outside during model operation.

Under shutdown conditions, the blocks could easily be pulled free of the core. While flow was being produced through the model, the blocks were drawn in firmly against the core and held by the pressure differential between the outside and inside of the reflector block. No vibration was induced at any condition of model operation, which ranged from zero to over one-half prototype Reynolds number for two-nozzle balanced flow, and from zero to full prototype Reynolds number for one-nozzle flow.

At various flow conditions, individual reflectors were retracted approximately $3/4$ in.; this operation involved forces as high as 150 lb. The reflectors were then released suddenly and allowed to slam into the core assembly. No noticeable movement of core elements resulted.

Static-pressure measurements in the gaps at the inside and outside of the reflector blocks provided differential pressure data necessary for checking the total force across the blocks. This was proved to be the force needed to pull the block free of the core, demonstrating that the block was free to tilt.

Fig. 32--Heat-transfer coefficient at top-dome locations

The conclusion reached was that the core-reflector arrangement as modeled was very stable under all conditions of model operation.

CORE-REFLECTOR SEAL TESTS

The original seal in the model did not fit with the tolerances necessary to provide sealing. Consequently, the gap was plugged with a rubber strip. The prototype seal design was undergoing revision during model testing. A careful survey of the requirements on tolerances of fits and surface finishes and the difficulty of instrumentation in the model led to the decision not to make the costly modification of simulating the revised prototype design in the model. A test was performed to investigate the leakage between a piece of graphite bearing against steel in a section which typically might be used in a seal. Results of this test have been used in designing the seal.

V. CONCLUSIONS

The most significant conclusions which may be drawn from the results of the flow-model test program are the following:

1. The tilting reflector is a workable concept. There was no indication of instability in the core region during model operation. No vibration could be induced by manual manipulation of the reflector blocks.
2. The pressure drop through the model indicated that the pressure drop through the prototype vessel will be slightly less than values obtained by calculation. The best estimate of prototype reactor vessel and core pressure drop at full-load helium flow rate based on flow-model data is 2.0 psi, as compared with a calculated value of 2.9 psi, at a helium flow rate of 468,000 lb/hr, pressure of 350 psia, and constant temperature through the reactor of 650°F.
3. It will be more desirable to have only one nozzle in the center of the top head for flow from the outer to inner cooling passages than the five nozzles in the original prototype design. Completely closing the four off-center nozzles reduced flow to the inner passages by only 20%. A valve to provide initial adjustment of flow distribution between the outer and inner passages (during precritical prototype testing) will be provided in the top center nozzle of the prototype.
4. There will be no areas with a serious deficiency of coolant flow in the prototype vessel with flow through either one or both of the nozzles, although there may be localized turbulent areas with small components of velocity in the general direction of flow. The flow should be sufficient in all areas to prevent serious localized hot spots from developing.

The results in some areas of study were not as specific as might have been desired. It would have been preferable to have had a model capable of full Reynolds number flow for determining pressure drop, since extrapolation from half-flow data could not be done with as much confidence as had been predicted at the time the flow model was designed. However, since the effects of compressibility were relatively small in the range in which the model was operated, the primary consideration was the accuracy to which the graphical extrapolation could be made; it is estimated that this error does not exceed $\pm 10\%$. The experimental heat-transfer coefficients did not provide a valid confirmation of calculated values as had been hoped, since, in general, they were about twice the calculated values.

Several recent design changes in the prototype reactor vessel geometry were not reflected in the flow-model results. It was not possible to test the reflector seal in the model as had originally been anticipated. On the other hand, the test program provided valuable information which was not originally anticipated, such as the data about flow through the nozzles in the top head which resulted in the decision to eliminate the four off-center nozzle openings.

Appendix A

PRESSURE-DROP DATA AND SAMPLE CALCULATIONS

HTGR FLOW-MODEL STATIC-PRESSURE DATA FOR INLET
AND OUTLET DUCTS

Date	Flow Rate (lb/sec/nozzle)		Static Pressure (inches of water)			
			Inlet Metal Duct		Outlet Metal Duct	
	East	West	East	West	East	West
2/22/61	4.20	4.20	-1.42	-1.56	-9.10 (Avg)	
2/22/61	5.90	5.90	-2.76	-3.01	-16.50 (Avg)	
2/22/61	7.70	7.70	-4.69	-5.10	-27.50 (Avg)	
2/22/61	8.22	8.22	-5.00	-5.41	-30.88 (Avg)	
2/22/61	8.68	8.68	-5.54	-5.97	-34.50 (Avg)	
2/22/61	9.14	9.14	-6.16	-6.69	-38.38 (Avg)	
2/24/61	5.82	5.82	-2.72	-2.94	-17.75 (Avg)	
2/24/61	7.04	7.04	-4.11	-4.50	-25.83 (Avg)	
2/24/61	8.05	8.05	-5.28	-5.83	-33.48 (Avg)	
2/24/61	8.90	8.90	-6.41	-6.99	-40.64 (Avg)	
2/24/61	9.88	9.88	-7.44	-8.13	-52.26 (Avg)	
2/28/61	6.04	6.0	-2.57	-2.84	-17.55 (Avg)	
2/28/61	7.13	7.16	-3.98	-4.35	-25.37 (Avg)	
2/28/61	8.20	8.25	-5.35	-5.75	-33.22 (Avg)	
2/28/61	9.05	9.05	-6.31	-6.76	-40.16 (Avg)	
2/28/61	5.95	0.3	-2.68	----	-7.77	----
2/28/61	7.25	0.5	-4.19	----	-11.10	----
2/28/61	8.40	0.7	-5.28	----	-14.53	----
2/28/61	9.35	0.8	-6.33	----	-17.89	----
2/28/61	10.50	0.9	-8.04	----	-22.25	----
2/28/61	12.75	1.0	-11.31	----	-30.75	----
2/28/61	14.2	1.2	-14.43	----	-38.75	----
5/8/61	7.0	7.05	-5.11	-4.91	-24.25	-24.10
5/8/61	8.0	8.07	-6.76	-6.36	-31.20	-31.03
5/8/61	8.93	8.95	-8.12	-7.71	-38.75	-38.45
5/8/61	10.7	10.7	-11.63	-11.07	-55.90	-55.30

HTGR FLOW-MODEL VENTURI AND BAROMETRIC PRESSURE DATA

Date	Venturi Differential (in. water)		Venturi Absolute Pressure (in. Hg)		Flow Rate (lb/sec/nozzle)		Atmospheric Properties			
	East	West	East	West	East	West	Barometer (in. Hg)	Temp. (°F)	Rel. Hum. (%)	Density (lb/ft ³)
2/22/61	2	2			4.2	4.2	29.52	55	89	0.0752
2/22/61	4	4			5.9	5.9	29.52	55	89	0.0752
2/22/61	7	7			7.7	7.7	29.52	55	89	0.0752
2/22/61	8	8			8.22	8.22	29.52	55	89	0.0752
2/22/61	9	9			8.68	8.68	29.52	55	89	0.0752
2/22/61	10	10			9.14	9.14	29.52	55	89	0.0752
2/24/61	4	4	28.65	28.62	5.82	5.82	29.86	71	20	0.0740
2/24/61	6	6	28.08	28.08	7.04	7.04	29.86	71	20	0.0740
2/24/61	8	8	27.57	27.52	8.05	8.05	29.86	71	20	0.0740
2/24/61	10	10	27.10	27.05	9.10	8.90	29.86	69	30	0.0747
2/24/61	Max	Max	26.50	26.18	9.88	9.88	29.86	69	30	0.0747
2/28/61	4.15	4.05	28.68	28.65	6.04	6.00	29.75	64	32	0.0748
2/28/61	6.11	6.16	28.12	28.08	7.13	7.16	29.75	64	32	0.0748
2/28/61	8.28	8.33	27.58	27.51	8.20	8.25	29.75	65	31	0.0748
2/28/61	10.24	10.21	27.08	27.03	9.05	9.05	29.75	65	31	0.0748
2/28/61	4.06	0.03	29.35	-----	5.95	0.3	29.72	68	48	0.0741
2/28/61	6.22	0.05	29.12	-----	7.25	0.5	29.72	68	48	0.0741
2/28/61	8.30	0.07	28.90	-----	8.40	0.7	29.72	68	48	0.0741
2/28/61	10.27	0.09	28.69	-----	9.35	0.8	29.72	67	40	0.0742
2/28/61	13.15	0.10	28.38	-----	10.50	0.9	29.72	67	40	0.0742
2/28/61	18.79	0.13	27.80	-----	12.75	1.0	29.72	67	40	0.0742
2/28/61	25.13	0.18	27.28	-----	14.2	1.2	29.72	67	40	0.0742
5/8/61	5.9	6.0	28.22	28.18	7.0	7.05	29.73	67	76	0.0738
5/8/61	7.85	8.0	27.73	27.68	8.0	8.07	29.73	67	76	0.0738
5/8/61	9.95	10.0	27.25	27.18	8.93	8.95	29.73	67	76	0.0738
5/8/61	14.75	15.0	26.10	26.00	10.7	10.7	29.73	67	76	0.0738

SAMPLE CALCULATIONS OF HTGR PRESSURE DROP

The calculations outlined here show how the flow-model pressure-drop data were extrapolated to yield HTGR prototype pressure drop. The calculations are discussed in terms of specific test data which have been plotted in Figs. 10 through 18.

TWO-NOZZLE FLOW

FIGURE 10

The following flow-model data were recorded during a test run on 5/8/61:

Air flow rate	7.02 lb/sec/nozzle
Barometer	29.73 in. Hg
Atmosphere	
Temperature	67°F
Relative humidity	76%
Atmospheric density	0.0738 lb/ft ³
Manometer readings, static-pressure drop:	
To model inlet	-5.01 in. of water
To model outlet	-24.17 in. of water

The pressure drop through the model should follow the relationship:

$$\Delta P = \frac{K \rho V^2}{144 \times 2g} \quad .$$

Since $V = M/\rho A$,

$$\Delta P = K \frac{M^2}{144 \rho A^2 \times 2g}$$

or

$$\Delta y_w = K \frac{M^2}{5.2 \rho A^2 \times 2g}$$

Since atmospheric conditions varied between different runs, pressure-drop readings should be referred to some standard condition so that they may be compared. The reference atmosphere arbitrarily selected was 30 in. Hg, 60°F, and 60% relative humidity, with a standard density of 0.0752 lb/ft³.

The relationship between actual test conditions and those of the "standard" atmosphere may be expressed as

$$\frac{\Delta y_r}{\Delta y_x} = \frac{K_r \frac{M_r^2}{144 \rho_r A_r^2 2g}}{K_x \frac{M_x^2}{144 \rho_x A_x^2 2g}} .$$

Since, for the same flow rate, $M_r = M_x$, $K_r = K_x$, and $A_r = A_x$, then

$$\frac{\Delta y_r}{\Delta y_x} = \frac{\rho_x}{\rho_r} .$$

For the flow-model data cited, the following values at the inlet and at the outlet, corrected to a reference atmosphere, were obtained:

$$\Delta y_r = \frac{\rho_x}{\rho_r} \Delta y_x = \frac{0.0738}{0.0752} \times (-5.01) = -4.91 \text{ in. of water at inlet,}$$

$$\Delta y_r = \frac{\rho_x}{\rho_r} \Delta y_x = \frac{0.0738}{0.0752} \times (-24.17) = -23.70 \text{ in. of water at outlet;}$$

therefore, the drop in static pressure, inlet to outlet, was

$$\Delta y_r = (-23.70) - (-4.91) = -18.79 \text{ in. of water.}$$

The pressure-drop relationships between prototype and model should be comparable at equivalent Reynolds numbers ($R_p = R_m$). The following are the prototype conditions at full flow:

Flow	468,000 lb/hr
or	65 lb/sec/nozzle
at	350 psia, 652°F
Helium density .	0.1155 lb/ft ³

The Reynolds number is defined by

$$R = \frac{\rho DV}{\mu} = \frac{M}{Q} \times \frac{DV}{\mu} = \frac{MD}{A\mu} .$$

The relationships between the model and the prototype are:

$$D_p = 2D_m$$

$$A_p = 4A_m$$

$$\frac{M_p D_p}{A_p \mu_p} = \frac{M_m D_m}{A_m \mu_m}$$

$$M_m = M_p \frac{A_m}{A_p} \times \frac{D_p}{D_m} \times \frac{\mu_m}{\mu_p}$$

$$\mu_p = 0.079 \text{ lb/hr-ft}$$

$$\mu_m = 0.0437 \text{ lb/hr-ft}$$

Then the model flow rate, M_m , corresponding to the prototype-design full-flow Reynolds number is

$$M_m = 65 \times \frac{1}{4} \times 2 \times \frac{0.0437}{0.079} = 17.9 \text{ lb/sec/nozzle} .$$

The model static-pressure drop extrapolated to 17.9 lb/sec/nozzle flow in Fig. 10 is 118 in. of water.

If we assume the loss coefficient, K , is the same for the model and for the prototype,

$$\Delta P_p = \left(\frac{M_p}{M_m} \right)^2 \left(\frac{A_m}{A_p} \right)^2 \frac{\rho_m}{\rho_p} \Delta P_m .$$

Here, the model density, ρ_m , must be that extrapolated from densities at 60°F and 60% relative humidity, since the pressure-drop readings were corrected to these atmospheric conditions.

FIGURE 11

A plot of these densities is shown in Fig. 11. For example, the absolute pressure at the inlet is

$$y_{in} = 30 \text{ in. Hg} - \left(4.91 \text{ in. H}_2\text{O} \times \frac{1 \text{ in. Hg}}{13.6 \text{ in. H}_2\text{O}} \right) = 29.64 \text{ in. Hg} .$$

The density, ρ_a , at 29.64 in. Hg, 60°F, and 60% relative humidity is 0.0747 lb/ft³.

The average model air density extrapolated to full-load flow is 0.0595 lb/ft³. The prototype pressure drop based on this average air density in the model, where the model pressure drop is

$$\Delta P_m = 118 \text{ in. H}_2\text{O} \times 0.0361 \text{ psi/in. H}_2\text{O} = 4.26 \text{ psi} ,$$

is

$$\Delta P_p = \left(\frac{65}{17.9} \right)^2 \left(\frac{1}{4} \right)^2 \times \frac{0.0595}{0.1155} \times 4.26 = 1.81 \text{ psi} .$$

The prototype pressure drop based on the inlet density in the model is

$$\Delta P_p = \frac{0.070}{0.0595} \times 1.81 = 2.13 \text{ psi} .$$

FIGURE 12b

The data shown in Fig. 12b is the same data as that shown in Fig. 10 for the 8.9 lb/sec/nozzle air flow rate.

FIGURE 13

The two-nozzle balanced flow data plotted in Fig. 13 were obtained by performing the same type of corrections as were performed for the data of Fig. 10.

FIGURE 14

The loss coefficient for the model may be expressed as

$$K = \frac{5.2 \Delta y \ 2g \ \rho_m \ A_m^2}{M_m^2} .$$

For a model inlet area, A , of 0.985 ft^2 ,

$$K = \frac{5.2 \Delta y \times 64.4 \rho_m (0.985)^2}{M_m^2} = \frac{325 \Delta y \rho_m}{M^2},$$

where ρ_m is the density referred to standard atmosphere, as in Fig. 11.

The total model pressure, y_t , for Fig. 14 is

$$y_t = y_s + y_v,$$

where

$$y_v(\text{in. H}_2\text{O}) = \left(\frac{\rho_m}{\rho_w} \right) y_m(\text{in. of air}) = \frac{12 M^2}{\rho_w \rho_m A^2 \times 2g} = \frac{M^2}{334.5 \rho_m A^2}.$$

The velocity head at the model inlet is

$$y_{v_{in}} = \frac{(7.02)^2}{334.5 \times 0.0747 \times (0.985)^2} = 2.03 \text{ in. H}_2\text{O},$$

and for A_{out} of 1.21 ft^2

$$y_{v_{out}} = \frac{(7.02)^2}{334.5 \times 0.0713 \times (1.21)^2} = 1.41 \text{ in. H}_2\text{O}.$$

Thus,

$$y_{t_{in}} = (-4.91) + (+2.03) = -2.88 \text{ in. H}_2\text{O},$$

and

$$y_{t_{out}} = (-23.70) + (+1.41) = -22.29 \text{ in. H}_2\text{O}.$$

The pressure drop from inlet to outlet is thus

$$\Delta y_t = (-22.29) - (-2.88) = -19.41 \text{ in. H}_2\text{O}.$$

Based on these calculations, the loss coefficient, using average density, is

$$K = \frac{325 \times 19.41 \times 0.0730}{(7.02)^2} = 9.35 .$$

Using inlet density,

$$K = \frac{325 \times 19.41 \times 0.0747}{(7.02)^2} = 9.57 .$$

The prototype pressure drop calculated using K is

$$\Delta P_p = K \frac{M^2}{144 \rho A^2 \times 2g}$$

for a prototype inlet area of 4.03 ft². Solving this equation,

$$\Delta P_p = K \frac{(65)^2}{144 \times 0.1155 \times (4.03)^2 \times 64.4} = 0.242K \text{ psi} .$$

The loss coefficient, K, based on average model density and extrapolated to full flow is 8.0 in Fig. 14. The corresponding prototype pressure drop is

$$\Delta P_p = 0.242 \times 8.0 = 1.94 \text{ psi} .$$

FIGURE 15

If air expands isothermally from atmosphere through the model system, it would follow the relation

$$Pv = \frac{P}{\rho} = \text{constant} .$$

For this case, the density at the inlet would be

$$\rho_{in} = \rho_r \left(\frac{P_{in}}{P_r} \right) ;$$

for the flow-model data cited at the beginning of this section (p. 63)

$$\rho_{in} = 0.0752 \left(\frac{30 - \frac{4.91}{13.6} \text{ in. Hg}}{30 \text{ in. Hg}} \right) = 0.0743 \text{ lb/ft}^3 .$$

If air expands adiabatically, it will follow the relation

$$P_v^{1.4} = \frac{P^{1/1.4}}{\rho} = \text{constant} ,$$

and the inlet density would be

$$\rho_{in} = 0.0752 \left(\frac{30 - \frac{4.91}{13.6}}{30} \right)^{1/1.4} = 0.0748 \text{ lb/ft}^3 .$$

If the pressure drop in any expansion is approximately proportional to the average density between the atmosphere and the point in question (inlet or outlet),

$$\Delta P = K \frac{M^2}{144 \bar{\rho} A^2 \times 2g} .$$

However, if the flow is incompressible, it is proportional to atmospheric density:

$$\Delta P(\text{incompressible}) = K \frac{M^2}{144 \rho_r A^2 \times 2g} ,$$

$$\frac{\Delta P(\text{incompressible})}{\Delta P(\text{isothermal or adiabatic})} = \frac{\bar{\rho}}{\rho_r} ,$$

or

$$\Delta y(\text{incompressible}) = \left(\frac{\rho_r + \rho_{in}}{2 \times \rho_r} \right) \times \Delta y(\text{atmosphere to inlet}).$$

If the model pressure drop to the inlet is corrected to incompressible flow from isothermal expansion,

$$y_s = \left(\frac{0.0752 + 0.0743}{2 \times 0.0752} \right) \times -4.91 = -4.88 \text{ in. H}_2\text{O} ,$$

$$y_v = 2.03 \text{ in. H}_2\text{O} ,$$

and

$$y_t = -4.88 + 2.03 = -2.85 \text{ in. H}_2\text{O} .$$

When the model pressure drop to the inlet is corrected to incompressible flow from adiabatic expansion,

$$y_s = \left(\frac{0.0752 + 0.0748}{2 \times 0.0752} \right) \times -4.91 = -4.89 \text{ in. H}_2\text{O} ,$$

and

$$y_t = -4.89 + 2.03 = -2.86 \text{ in. H}_2\text{O} .$$

The pressures at the outlet are calculated in the same manner, replacing inlet with outlet values in the above equations.

The prototype pressure drop for Fig. 15 was calculated using the method of Fig. 10, with the reference density used as the model density ($\rho_m = 0.0752 \text{ lb/ft}^3$). Corrected from isothermal expansion, the pressure drop of the model was

$$\Delta P_m = 110 \text{ in. H}_2\text{O} \times 0.0361 = 3.97 \text{ psi} ,$$

and that of the prototype was

$$\Delta P_p = \left(\frac{65}{17.9} \right)^2 \left(\frac{1}{4} \right)^2 \times \frac{0.0752}{0.1155} \times 3.97 = 2.13 \text{ psi} .$$

FIGURE 16

For this figure, the same method of calculation was used as for Fig. 14. Since $\rho = \text{constant}$,

$$\rho_m = 0.0752 \text{ lb/ft}^3 .$$

For the constant density of 0.0752 lb/ft^3 ,

$$K = \frac{325 \Delta y \rho_m}{M^2} = \frac{325 \times 18.79 \times 0.0752}{(7.02)^2} = 9.31 .$$

The prototype pressure drop corrected from isothermal expansion, with $K(\text{extrapolated to } 17.9 \text{ lb/sec}) = 8.2$, is

$$\Delta P_p = 0.242 \times 8.2 = 1.98 \text{ psi} .$$

FIGURE 17

Utilizing the isothermal and adiabatic method of Fig. 15 with the actual model data:

$$\rho_{x_{in}} = 0.0747 \text{ lb/ft}^3 ,$$

$$y_{in}(\text{incompressible}) = \left(\frac{0.0752 + 0.0747}{2 \times 0.0752} \right) \times -4.91 = -4.88 \text{ in. H}_2\text{O} ,$$

$$y_t = -4.88 + 2.03 = -2.85 \text{ in. H}_2\text{O} ,$$

$$y_{out}(\text{incompressible}) = \left(\frac{0.0752 + 0.0713}{2 \times 0.0752} \right) \times -23.70 = -23.07 \text{ in. H}_2\text{O} ,$$

$$y_t = -23.07 + 1.41 = -21.66 \text{ in. H}_2\text{O} ,$$

and the model pressure drop is

$$\Delta y_t = (-2.85) - (-21.66) = 18.80 \text{ in. H}_2\text{O} .$$

With the model pressure drop extrapolated to 17.9 lb/sec,

$$\Delta y_t = 116 \text{ in. H}_2\text{O} ,$$

or

$$\Delta P_m = 116 \times 0.0361 = 4.19 \text{ psi} .$$

The prototype pressure drop is

$$\Delta P_p = \left(\frac{65}{17.9} \right)^2 \left(\frac{1}{4} \right)^2 \times \frac{0.0752}{0.1155} \times 4.19 = 2.25 \text{ psi} .$$

FIGURE 18

Using the same method as described for Figs. 14 and 16,

$$K = \frac{325 \Delta y_p}{M^2} = \frac{325 \times 18.80 \times 0.0752}{(7.02)^2} = 9.31 .$$

K extrapolated to 17.9 lb/sec is 8.3, and the prototype pressure drop is

$$\Delta P_p = 0.242 \times 8.3 = 2.01 \text{ psi}.$$

ONE-NOZZLE FLOW

For the one-nozzle flow conditions, the corrections were made in the same manner as described above for two-nozzle flow.

From Fig. 13, the model pressure drop extrapolated to 17.9 lb/sec is $\Delta y_m = 32$ in. H_2O . From Fig. 11, the average model density extrapolated to 17.9 lb/sec is $\bar{\rho} = 0.0685$ lb/ft³. The model pressure drop is

$$\Delta P_m = 32 \times 0.0361 = 1.15 \text{ psi},$$

and the corresponding prototype pressure drop is

$$\Delta P_p = \left(\frac{65}{17.9} \right)^2 \left(\frac{1}{4} \right)^2 \frac{0.0685}{0.1155} \times 1.15 = 0.56 \text{ psi}.$$

From Fig. 14, the loss coefficient extrapolated to 17.9 lb/sec is

$$K = 2.4,$$

and the corresponding prototype pressure drop is

$$\Delta P_p = 0.242 \times 2.4 = 0.58 \text{ psi}.$$

Appendix B

SUPPLEMENTARY CURVES OF VELOCITY-TEST DATA

Fig. 33--Velocity profiles at points 1a/01A-08A, ~30 in. above inlet, for balanced two-nozzle flows

Fig. 34--Velocity profiles at points 1c/09A-16A, ~48 in. below inlet, for balanced two-nozzle flows

Fig. 35--Velocity profiles at points 1a/01A-08A, ~30 in. above inlet, for flow through east nozzle only

Fig. 36--Velocity profiles at points 1c/09A-16A, ~48 in. below inlet, for flow through east nozzle only

Fig. 37--Velocity profiles at points 1a/01A-08A, ~ 30 in. above inlet, for flow through west nozzle only

Fig. 38--Velocity profiles at points 1c/09A-16A, ~48 in. below inlet, for flow through west nozzle only

Appendix C

COMPUTER PROGRAMS FOR VELOCITY OF FLOW-MODEL
COOLANT AND FOR HEAT-TRANSFER FILM COEFFICIENTS

COMPUTER PROGRAM FOR VELOCITY OF FLOW-MODEL COOLANT

INPUT RAW DATA

		<u>Insert Once</u>		
		Run No.		
		Flow rate, nozzle No. 1	M_{a1}	lb/sec
		Flow rate, nozzle No. 2	M_{a2}	lb/sec
		Barometric pressure	B. P.	in. Hg
<u>Readout</u>	{	<u>Insert/For Every Point</u>		
		Probe position	j	
		Traverse position	ℓ	
		Velocity pressure	ΔP	in. H ₂ O
		Reference static level	Z_1	in. H ₂ O
		Point static level	Z_2	in. H ₂ O
		Internal air temperature	E_a	mv
		Equivalent diameter	D_e	ft

INFORMATION STORED

$$\begin{aligned}
 K_4 &= 18.3 \\
 \rho_{he} &= 0.115 \text{ lb/ft}^3 \\
 \mu_{he} &= 0.0805 \text{ lb/hr-ft} \\
 K_5 &= 0.4912 \\
 K_6 &= 0.03613
 \end{aligned}$$

CALCULATIONS

$$\text{Readout } \left\{ (1) \rho_a = \frac{0.05825}{(E_a + 10.80)} [K_5 (\text{B. P.}) - K_6 (Z_2 - Z_1)] \right. \quad \text{lb/ft}^3$$

$$\begin{array}{l}
 \text{Readout} \left\{ \begin{array}{l}
 (2) \quad \bar{V} = K_4 \sqrt{\frac{\Delta P}{\rho_a}} \quad \text{ft/sec} \\
 (3) \quad \mu_a = 0.04155 + 0.002861 E_a \quad \text{lb/hr-ft} \\
 (4) \quad V_{he} = \frac{\mu_{he}}{2\mu_a} \times \frac{\rho_a}{\rho_{he}} \times V_a \\
 (5) \quad M_{he} = \frac{2\mu_{he}}{\mu_a} \times M_a \quad (\text{calculated for both nozzles 1 and 2})
 \end{array} \right.
 \end{array}$$

COMPUTER PROGRAM FOR FLOW-MODEL HEAT-TRANSFER
FILM COEFFICIENTS

INPUT RAW DATA

		<u>Insert Once</u>	
		Run No.	
<u>Readout</u>	Flow rate, nozzle No. 1	M_{a1}	lb/sec
	Flow rate, nozzle No. 2	M_{a2}	lb/sec
	Barometric pressure	B. P.	in. Hg
	<u>Insert For Every Point</u>		
	Surface heater No.	i	
	Heater position	j	
	Heater volts	E	v
	Heater amperes	I	amp
	Surface temperature	E_s	mv
	Internal air temperature	E_a	mv
	Reference manometer level	Z_1	in. H_2O
	Local manometer level	Z_2	in. H_2O
	Equivalent diameter	D_e	ft
	ΔP		in. H_2O
	Average velocity	$\bar{V}_a$	ft/sec

INFORMATION STORED

$$\mu_{he} = 0.0805 \quad \text{lb/ft-hr}$$

$$k_{he} = 0.146 \quad \text{Btu/(hr)(ft)(}^{\circ}\text{F)}$$

$$K_2 = 56.2$$

$$K_4 = 18.3$$

$$K_5 = 0.4912 \frac{\text{lb/in.}^2}{\text{in. Hg}}$$

$$K_6 = 0.03613 \frac{\text{lb/in.}^2}{\text{in. H}_2\text{O}}$$

CALCULATIONS

$$(1) \rho_a = \frac{0.05825}{(E_a + 10.80)} [K_5 (\text{B. P.}) - K_6 (Z_2 - Z_1)] = \text{lb/ft}^3$$

$$(2) \mu_a = 0.04155 + 0.002861 E_a = \text{lb/ft-hr}$$

$$\text{Readout} \left\{ (3) R = \frac{\rho_a V_a D_e}{\mu_a} \right.$$

$$(4) C_i = f(R) \quad (\text{i selects the equation corresponding to the heater})$$

$$\text{Readout} \left\{ (5) h_a = \frac{C_1 K_2 EI}{(E_s - E_a)} = \text{Btu/(hr)(ft}^2\text{)}(^{\circ}\text{F)} \right.$$

$$(6) k_a = 0.01395 + 0.001133 E_a = \text{Btu/(hr)(ft)}(^{\circ}\text{F)}$$

$$\text{Readout} \left\{ \begin{array}{l} (7) h_{he} = \frac{k_{he}}{2 k_a} h_a = \text{Btu/(hr)(ft}^2\text{)}(^{\circ}\text{F)} \\ (8) M_{he} = \frac{2\mu_{he}}{\mu_a} M_a = \text{lb/sec} \quad (\text{run for both nozzles}) \\ (9) N = \frac{h_a D_e}{k_a} \end{array} \right.$$

Appendix D

MATHEMATICAL METHOD OF COMPUTING FLOW-MODEL
HEAT-TRANSFER COEFFICIENTS FOR AIR

MATHEMATICAL METHOD OF COMPUTING FLOW-MODEL
HEAT-TRANSFER COEFFICIENTS FOR AIR

1. The 1-B heater in position 5 on run No. 2 of March 7, 1961, is used as an example.
2. First, watts are obtained as the product of volts and amperes, and the temperatures (surface temperature of heater and air temperature) are obtained via the millivolt reading from a table for copper-constantan thermocouples.
3. The difference of temperature is obtained by subtracting heater-surface temperature from air temperature.
4. Next, the calibration factor, C_i , must be obtained. For heater probes which are independent of Reynolds number, this C_i factor is a pre-determined constant. If it is dependent on R , the intermediate step of acquiring R must be done as follows:

$$R = \frac{\rho_a \bar{V}_a D_e}{\mu_a} \times 300 .$$

- a. Here, ρ_a is density in lb/ft^3 and is obtained via the barometric-pressure (B.P.) curve of the appropriate relative humidity and temperature versus density chart. The barometric pressure refers to the pressure within the air passage and is obtained by

$$P = \text{B.P.} - (Z_2 - Z_1) \frac{1}{13.7} .$$

In the example used, this is $\text{B.P.} = 29.77 - [(8.5 - 4.0)(1/13.7)]$, where B.P. is the outside absolute pressure in inches of Hg, Z_2 is static pressure in the air passage and Z_1 is the static reference point on the manometer board, and, since these Z_2 and Z_1 are read in inches of water, the $1/13.7$ is necessary to convert to inches of Hg. The relative humidity must also be known so that the appropriate chart can be used. In the example, ρ is 0.0746.

- b. $\bar{V}_a$ is in ft/sec and is the average velocity in the air passage obtained from data at this point on a similar run (same mass flow) of another date. $\bar{V}_a$ in the example is 25.5 and was obtained by

drawing a scaled curve of the velocities through the passage, finding the area with a planimeter, and dividing by the passage width.

- c. D_e is the equivalent diameter in inches, 3.844, and has been obtained via D_e equations as twice the passage width.
- d. The term μ_a is the viscosity of the air in lb/hr-ft and is obtained from the appropriate viscosity versus temperature chart. The passage-air temperature is used here; μ for the example is 0.0435.
- e. The "300" is necessary to convert to common units for cancellation, as R is a nondimensional number.
- f. The Reynolds number in the example becomes 52,700.
- g. After the Reynolds number has been obtained, C_i can be taken from the appropriate heater-probe calibration chart. In the example, which is heater 1-B, C_{1-B} is 0.226. The end points of this curve were obtained by the equation

$$C_i = d(R)^s = 0.380 (52,700)^{-0.0476} = 0.226 ,$$

where d and s are related to the deviation of the heater probe from its theoretical efficiency and related to a variation with Reynolds number.

5. With C_i having been obtained, the film coefficient, h , for the heater probe within the given test conditions can be obtained by the equation

$$h = 2505 \frac{CW}{t_s - t_a} ,$$

where C is the calibration factor for the heater probe under the given conditions; W is the power in watts; t_s and t_a are the heater surface and air temperatures, respectively, in $^{\circ}\text{F}$, and 2505 is necessary for the conversion to common units.

6. In the example of probe 1-B in position 5 on run 2 of March 7, 1961, the calculation is as follows:

$$h_a = 2505 \frac{0.226 \times 0.194}{76.2 - 62.4} = 7.88 \text{ Btu/(hr)(ft}^2\text{)(}^{\circ}\text{F)} .$$

Appendix E

SIMPLIFIED GRAPHICAL PROCEDURE FOR ESTIMATING
FLOW-MODEL HEAT-TRANSFER COEFFICIENTS FOR AIR

SIMPLIFIED GRAPHICAL PROCEDURE FOR ESTIMATING FLOW-MODEL HEAT-TRANSFER COEFFICIENTS FOR AIR

Data must be obtained from a velocity run of conditions similar to those of the subject-heater run. The heater-calibration factor can be computed prior to the heater run when a heater is placed at a given location. The remaining factors are to be computed from heater data at the time of the heater run.

INFORMATION GAINED FROM THE VELOCITY RUN

1. Location is identified via the master location drawing 32-SK-2232 (part of which is reproduced in Fig. 3).
2. D_e , the equivalent diameter, is twice the measured jacket gap at the given location.
3. V is the approximate velocity taken from Fig. 39, which includes a 0.9 multiplier to cause an approach to average velocity. The velocity head used in this curve is to be taken at the center of the gap. (True average velocity, if available from more accurate calculations, can be used here.)
4. Reynolds number, R , is estimated from Fig. 40 by using the recorded D_e and V_a .

INFORMATION GAINED BEFORE THE HEATER RUN

5. The heater number identifies the heater used and is the means whereby the appropriate calibration multiple can be found.
6. The calibration multiple, C_i , is taken from the appropriate heater curve, similar to Fig. 41, using the known local Reynolds number.

COMPUTATION AT THE TIME OF THE HEATER RUN

7. W is the product of the volts and amperes, and can be obtained by straight multiplication of heater data.
8. The product of CW can be obtained by straight multiplication.

Fig. 39--Air velocity versus velocity head

Fig. 40--Reynolds number as a function of equivalent diameter and air velocity

Fig. 41--Heater calibration factor versus Reynolds number

9. $E_s - E_a$ is the difference between the millivolt reading of the heater-surface thermocouple and the millivolt reading of the passage-air thermocouple. This must be obtained arithmetically from the heater-run data.

10. The approximate heat-transfer coefficient for air, h_a , is taken directly from Fig. 42, using $E_s - E_a$ and the product CW .

Fig. 42--Heat-transfer coefficient, h , for air as a function of heater-probe values

Appendix F

SAMPLES OF FLOW-MODEL OUTER-JACKET SURFACE MAPS

FLOW-MODEL OUTER-JACKET SURFACE MAPS

During the various studies of the flow-model heat-transfer program, the values were mapped on scaled surfaces along with other pertinent information. Samples of these maps are shown in Figs. 43 and 44. Computer heat-transfer coefficients are given for the 10-in. flows in each of the March and May runs at each heater location so that comparisons can be made. Extrapolation to full prototype flow was done by a 0.8 slope from the 10-in. -flow model flows only.

Fig. 43--Heat-transfer coefficient for helium extrapolated from 10-in., two-nozzle flow-model data by $(642, 550/290, 000)^{0.8} = 1.887$ (March data)

Fig. 44--Heat-transfer coefficient for helium extrapolated from 10-in., two-nozzle flow-model data by $(642,550/290,000)^{0.8} = 1.887$ (May data)

Appendix G

SUMMARY OF RESULTS OF TEST PROGRAM OUTLINED IN
PRELIMINARY HAZARDS SUMMARY REPORT

SUMMARY OF RESULTS OF TEST PROGRAM OUTLINED IN PRELIMINARY HAZARDS SUMMARY REPORT

The Peach Bottom Atomic Power Station Preliminary Hazards Summary Report included a brief résumé of the results of the flow-model test program up to the time the Hazards Report was published in August, 1961, and also presented an outline of the further tests in the program. A summary of the significant results of the test program and a brief discussion of the specific items listed under the heading of "Further Testing" in the Preliminary Hazards Summary Report are included in this Appendix.

PHASE I--CONFIRMATION OF EXISTING DATA AND INVESTIGATION OF ADDITIONAL AREAS

1. Pressure-drop Data

Repeat pressure-drop measurements after removing spacer blocks near the entrance nozzles to get more reliable pressure-drop data:

Pressure-drop measurements were repeated after the spacer blocks near the nozzle entrances were removed. There was a slight reduction in pressure drop after the blocks were removed. The blocks were to assure proper spacing of the annuli in the flow model only, and bore no resemblance to prototype features.

2. Heat-transfer Data

Run tests simulating reactor normal and emergency operating conditions and normal shutdown and startup transients. Film-coefficient data were to be used by the vendor to calculate stresses in the pressure vessel:

Heat-transfer measurements taken during low-flow conditions such as would exist during startup or shutdown did not produce meaningful results. The velocities at such low flows were too small to be measured accurately, so the reliability of any heat-transfer coefficients measured would have been no better than these velocities. Heat-transfer-coefficient tests that were made were run at flow rates that will be encountered during normal operation, i. e., in the range of 25% to

60% of full prototype flow. Heat-transfer measurements were also made with flow through only one nozzle, which is an emergency condition. Because of the lag in time necessary for the heater readings to stabilize, it was not feasible to vary the flow rate while taking heat-transfer readings. At low flow rates, corresponding low film coefficients produce small, transient, thermal stresses, thus making low-flow-rate heat-transfer measurements unnecessary. It appears that if the heat-transfer coefficients obtained were extrapolated to a low flow rate, such as 10%, the coefficients would be sufficient for adequate cooling of the vessel. Since the flow-model coefficients were predominantly higher than the calculated values, the conservatively calculated values were used by the vendor for stress analyses.

Measure film coefficients in the upper head and cylindrical part of the vessel (a) on vertical planes through the nozzles and midway between the nozzles and (b) on a horizontal plane through the nozzles:

Film coefficients were measured in the upper head and cylindrical parts of the vessel. The test results indicated coefficients of about $80 \text{ Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$ in this region at design flow.

Measure film coefficients in the lower head in the vicinity of the core support structure to determine efficiency of cooling in this area:

Extrapolated film coefficients based on model tests were about $80 \text{ Btu}/(\text{hr})(\text{ft}^2)(^\circ\text{F})$ in this region at design flow.

3. Flow-rate Data

Determine flow rates up and down in the outer gap:

The flow divided almost equally upward and downward upon entering the model, confirming the intentions of the design.

PHASE II--DESIGN DATA FOR THE UPPER THERMAL SHIELDS AND PLENUM

1. Pressure-drop Calculations

Check the reliability of the calculations for pressure drop in the inner concentric gaps between the thermal shields and between the inner

thermal shield and plenum shroud:

Conservatively calculated values for pressure drop were generally higher than those measured in the model. This can best be explained by considering the complex nature of the flow passages and the difficulty in predicting loss coefficients to account for the geometry and the surface characteristics of the materials.

2. Top-nozzle Orifices

Determine the quantitative effects of the top-nozzle orifices on pressure drop:

The effect on pressure drop of varying the four off-center orifices in the top head from fully open to fully closed was negligible, and flow to the inner passages was reduced only 20%; this permitted a change in design in which essentially all flow to the top head would pass through the center orifice.

3. Plenum Shroud

Estimate the amount of heat flowing through the plenum shroud in the prototype:

Subsequent to setting this test requirement, a decision was made to insulate the plenum. The amount of heat flowing through the insulated plenum shroud in the prototype was calculated to be 1,600,000 Btu/hr, which causes heating of core inlet gas by less than 3° F.

4. Heat-transfer Data

Determine film coefficients at four points on the inner and outer surfaces of the plenum shroud and below the exit nozzle:

Heat-transfer measurements were not taken in the model on the plenum surfaces, since accurate experimental data were not required after the design decision was made to insulate the plenum.

Observe smoke patterns in the inner gap between the thermal shield and the plenum shroud at the exit nozzles:

Observations of smoke patterns near the exit nozzles indicated no problems of uncooled regions in these areas.

5. Flow-rate Data

Measure flow rates in the gaps between the thermal shields and the plenum shroud:

Flow rates down through the middle gap and down the outer gap were measured. A mass flow balance with the total flow indicated approximately 45% down the outer gap, 22% down the middle gap, and 33% down the inner gap, which is a satisfactory distribution. In the final design, one thermal shield has been eliminated, and the flow to the inner gap before nuclear startup will be adjusted by a valve in the center-nozzle opening in the top.

Measure flow rates across the core support plate:

The core plate in the model contained cooling holes in accordance with the original prototype design. Tests were run to study the crossflow of coolant through the core. A baffle covering the holes beneath a cluster of about 100 fuel elements was installed on the core plate. Pressure and velocity measurements were obtained at four vertical levels on the fuel elements to determine the extent to which the air passing around the baffle recombined. The tests indicated that the mass flow rate along the center element had returned to the average value through the core at a level near the top of the fuel elements. This test indicated that there would be no serious problems from a flow standpoint if there were no holes in the core plate; only a small portion of the flow was designed to go through the core plate. Calculations indicated that the core plate would be adequately cooled without holes, and the great problems of neutron shielding caused by the holes led to the decision to have a solid core plate.

PHASE III--DESIGN DATA FOR THE LOWER THERMAL SHIELD, SIDE REFLECTOR, AND REFLECTOR SEALS

1. Reflector Seals

Determine the leak rate through the reflector seals:

It was decided not to attempt to measure seal leakage in the half-scale flow model for the following reasons:

(a) scaling difficulties--sealing surfaces were believed to be difficult to scale to prototype operation condition; (b) measurement of leakage very difficult--leakage through several parallel sources could not be separated by pressure-drop data, and it would have been most difficult to achieve reasonable accuracy in making velocity measurements in the model setup; (c) essentially, a full-scale section of graphite seal was tested in a separate facility for leakage between a graphite-to-steel joint, and the values from these tests were used in the final seal design.

2. Core-restraining load

Find the restraining load exerted on the core by the reflector during operation:

The restraining effect of the tilting reflectors on the core was sufficient to hold the core stable, with no vibration, under all conditions of model operation, i. e., flow rates were tested from zero to 60% of full flow. This is discussed in more detail on page 53.

PHASE IV--ITEMS WITH POSSIBLE OR EXISTING DESIGN CHANGES

1. Core Plate

Study the effects due to proposed design changes of the core plate resulting in elimination of cooling holes in the core plate and elimination of the lower thermal shield under the core plate:

This is discussed under Phase II, item 5, above.

2. Vessel and Thermal Shield

Investigate in more detail the effects of manufacturing tolerances of vessel and thermal-shield diameters on flow distribution:

Some effort was made to establish a connection between the variation in gap widths and flow distribution, but no conclusive results were obtained. Design of the model prevented flexing of the shells to distort the gaps. Since the variation in the width of the gaps in the model did not seem to cause any large variations or serious problems in flow distribution in the model, the subject was not explored thoroughly.