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FFL-2 TEST REPORT
RESULTS AND ANALYSIS OF INITIAL TESTS

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Foreword

This report was prepared by the Reactor Engineering Department of the Astronuclear Laboratory of the Westinghouse Electric Corporation, Pittsburgh, Pennsylvania, 15236.

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1.0 CONCLUSIONS AND RECOMMENDATIONS

The purpose of the initial test series on the simulated heated seal system was to determine the adequacy of the analytical techniques used in the determination of the thermal and hydraulic characteristics of the lateral support system of the NRX-A reactors.

The heated seal test rig flow system comprises an annular flow channel characterized by a series of contractions and expansions. The heat source, which simulates heat fluxes resulting from conduction and internal heat generation in the reactor lateral support system is a graphite resistance element in the center of the test rig. Heat is removed from the test section by:

- 1) the hydrogen flowing in the annular flow channel and
- 2) the water flowing in the copper cooling jacket.

These simulate the effects of the seal coolant and the inner reflector coolant in the reactor lateral support system.

This test configuration and its attendant flow loop have demonstrated their capability for obtaining the desired test data over a limited range of operating conditions. Moreover, with continued testing and development of the model, it should provide greater insight into the thermal and hydraulic characteristics of the NRX-A lateral support and seal system over the complete range of reactor operation.

The test configuration was analyzed by the same basic methods and techniques used for the reactor lateral support system. When the hot seal analytical results were compared to the experimental data, the hydrogen flow rate was found to be within 20 percent and the gas temperature rise to be within 10 percent. This discrepancy is possibly attributed to the leakage between the seal lands and the outer barrel. The material temperature levels determined by analysis showed a maximum deviation of 80°R from the test data. This deviation is largely attributed to uncertainty in the boundary conditions existing in the test configuration and the uncertainty of the relationship of the experimental temperature measurements to mean temperatures.

Thus, on the basis of the agreement of the analysis with the data obtained in the first series of tests on the heated seal configuration, the level of confidence in the analysis of the reactor lateral support system has been significantly enhanced.

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So as to extend the knowledge of heat transfer characteristics of flow in the seal system region, it is recommended that testing be continued. Future testing should include the following:

- 1) Resolve uncertainties of temperature distribution in the inner and outer barrels.
- 2) Establish boundary conditions with more certainty.
- 3) Determine effects of interface leakage past the lands on flow rates and temperatures.
- 4) Extend comparison of analytical and experimental results to higher powers and higher heat fluxes to more nearly simulate reactor conditions.
- 5) Study operational characteristics at higher temperatures by utilizing preheated hydrogen in the annular cooling channel.

2.0 BACKGROUND

2.1 Justification for Test

In the NRX-A reactor, core lateral support is provided by spring-loaded plungers acting on the circumference of the filler strips through seal segments, and by an axially distributed pressure seal consisting of eighteen (18) rows of seal segments each having a characteristic impedance built-in by means of axial machined grooves. The lateral support springs are mounted on the outside diameter of the impregnated H4LM graphite inner reflector cylinders and the plungers pass through holes in this cylinder.

The coolant flow system consists of a primary flow channel, characterized by a series of contractions and expansions (i.e. flow contraction occurs on entering the seal segment grooves and expansion occurs as it enters the plenums between seal segments), and secondary interconnected flow channels formed by gaps between filler strips, the annular clearance between the plungers and the inner reflector cylinder, and the pores in the inner reflector cylinder. The analysis of the thermal and hydraulic characteristics of this complex system utilizes the SHYP-TOSS computer program.

Heretofore, these analytical techniques have only been checked on component tests at ambient temperatures^{1,2} (i.e. with no heat addition). The comparison of these ambient temperature results indicated excellent correlation with analysis for the seal pressure distributions and reasonable comparisons on the overall flow balance. However, since the addition of heat to the hydrogen coolant is a significant variable in determining the seal coolant pressure and temperature distributions and since the graphite material temperatures are important for both structural and corrosion considerations, component testing of a simulated heated seal system was initiated.

-
- 1 WANL-TME-792, Plugged Core Flow Test Report: Phase I Tests Prerequisite to NRX-A2, C. V. Buzza, et al, September 1964.
 - 2 WANL-TME-655, Analysis and Design Significance of B-3 Single Seal High Pressure Test Results from the Single Seal High Pressure Iso-Thermal Test Data, A. I. Miller, K. L. Rieke, 7 January 1964

2.2 Test Objectives

The general objectives of this test series are as follows:

- 1) To determine the adequacy of the analytical techniques used to predict the thermal and hydraulic characteristics of the reactor lateral support and seal system during power operations.
- 2) To determine the effect of progressive graphite corrosion on the thermal and hydraulic characteristics of a heated flow system similar to that which exists in the reactor.

Due to restrictions of time and design, the initial tests were limited to meeting only the first of these objectives at moderate heat fluxes, temperature levels, and increases in gas coolant temperature. The test model, designed and built to meet this limited objective, consisted of two hollow graphite cylinders bounding a hydrogen flow channel characterized by a series of three alternatively spaced flow contractions and expansions. The test model was heated on the inside diameter of the inner cylinder, and cooled by (1) hydrogen flowing in the annular channel between the graphite cylinders and by (2) water flowing through a copper jacket surrounding the outer cylinder. In essence, the hydrogen channel is affected by the same factors which would be found in the lateral support system of the NRX-A reactors. Thus, the comparison of the analytical results with the experimental data from this test constitutes a fair evaluation of the adequacy of similar analytical techniques used on the full scale reactor. This comparison, presented in this report, has been designated^{3, 4} a prerequisite to the NRX-A2 test.

3 Aerojet General Corporation ltr AGC775:SDN:445, Prerequisite Component Testing and Analysis for NRX-A2, 8 April 1964

4 WANL-TME-728, Component Testing Prerequisites to NRX-A2 Power Testing, S.L. Abramson, 20 March 1964

3.0 TESTS AND ANALYSIS

3.1 Description of Test

3.1.1 Description of Test Piece and Flow System

As shown in figure 1, the test piece housing is a cylindrical pressure vessel 10.75 inches in diameter and 21.75 inches long. The inlet flange of the vessel is stainless steel, the water cooled outlet flange is Inconel X-750. Two concentric graphite cylinders fit in the pressure vessel. The outer cylinder of H4LM graphite, impregnated after machining, according to Specification WANL-294545, Revision 3, is surrounded by a copper heat exchanger. The inner cylinder, of ZTA graphite, contains a heater element (figure 2). As shown in figure 3, three bands of lands and grooves are machined on the outside surface of the inner cylinder. The lands are sized to contact the inside surface of the outer cylinder; hydrogen flows through the grooves. On each band 18 grooves are machined to a nominal depth of 0.007 inch. The hydrogen flow to an inlet manifold on the housing is directed through four inlet ports (figure 4). These ports and an internal annular plenum chamber aid in the establishment of a good distribution of the entering gas.

The heater element, of HLM-85 graphite, has two separate sections whose heating paths parallel each other. Each section connects to two common graphite terminals into which are screwed water cooled copper electrodes. These electrodes extend through the inlet end of the pressure vessel, as shown in figure 4. Silicone rubber "O" rings fitted on boron nitride sleeves seal the electrodes; the boron nitride sleeves insulate the electrodes from the pressure vessel. (figure 5).

When the test piece is under pressure, the electrode end of the heater element seats on a stack of Marinite and aluminum oxide plates. This stack acts as a heat barrier and as an aligning fixture. At the other end of the heater element, in the bottom of the inner cylinder, an aluminum oxide cup acts as a heat barrier and would pilot the heater element should it distort severely off center.

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FIGURE 1 PRESSURE VESSEL

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Figure 2. Assembled Test Section of Heat Transfer Model

Figure 3. Disassembled Test Section of Heat Transfer Model

Figure 4. Subassemblies of the Heat Transfer Model

Figure 5. Subassemblies of the Heat Transfer Model

The water jacket surrounding the outer cylinder is constructed of copper pipe. Water flows to an inlet plenum of the exchanger from a one inch supply line. From the inlet plenum the water flows through 300 axial coolant passages in the same direction as the hydrogen flow in the inner cylinder grooves. The passages are 0.031 inch wide by 0.038 inch deep and exhaust to an outlet plenum chamber.

The tolerances on part dimensions for the test piece allow ample space for expansion in areas where fits are not critical. However, the tolerances on the outside diameter of the inner cylinder lands and the inside diameter of the outer cylinder, combined with the differential expansion of the parts and the end restraints, permit a probable clearance of the order of 0.0005 inch to occur between the lands and the outer cylinder at temperature levels experienced during the run reported.

Thermocouples, equipped with polarized jack and plug assemblies for quick connect and disconnect, measure temperature in five regions of the test piece: (1) heater element, (2) inner cylinder (3) hydrogen gas inlet and outlet temperatures, (4) outer cylinder, and (5) water jacket.

A tungsten/tungsten-26-percent-rhenium thermocouple at the axial mid-section and inside the heater element is an indication of the heater element temperature (figure 6). This heater element region temperature, designated HT1, is used as a control and is read out continuously on a Leeds and Northrup potentiometer during a test.

Four precalibrated chromel/alumel thermocouples, designated IBT1, IBT2, IBT3, and IBT4, measure the inner cylinder temperatures. IBT1 is in the first land and groove band; IBT2, IBT3, and IBT4 are in the second band at different distances from the inner cylinder wall. (figure 6 and 7). The thermocouples read out on separate Minneapolis Honeywell Multipoint recorders that have been internally cold-junction compensated.

Two copper/constantan thermocouples measure hydrogen inlet temperature, and three chromel/alumel thermocouples measure hydrogen outlet temperature. All five are shielded to prevent radiation effects by surrounding material. The inlet thermocouples, ST8 and ST9, are inserted through two opposite inlet ports. Thermocouples ST8 is in the middle of the internal annular plenum; ST9 is at the entrance to the test section. (figure 6). The three outlet thermocouples are in the same axial station in the test model. Thermocouples ST7 and ST6 are at $\theta = 10$ degrees and 30 degrees, respectively, opposite inner barrel grooves. Thermocouples

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Figure 6. Locations of the Thermocouples, WANL Curve No. 603008D

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Figure 7. Heat Transfer Fixture-Circular, 200 psi, 3500 F WANL Drawing No. NT701182 Rev. A

ST5 is at $\theta = 170$ degrees, opposite an inner cylinder land. (figures 6 and 7).

Five precalibrated chromel/alumel thermocouples measure the outer cylinder temperatures. Thermocouple OBT1 is opposite the first land and groove band, and thermocouple OBT 5 is opposite the third land and groove band. Both instruments are at $\theta = 45$ degrees and penetrate the outer cylinder wall to a depth of $11/32$ inch. Thermocouples OBT2, OBT3, and OBT4 are opposite the second land and groove band. OBT2 is at $\theta = 45$ degrees and penetrates the outer cylinder wall to a depth of $5/64$ inch; OBT3 is at $\theta = 120$ degrees and penetrates the outer cylinder wall to a depth of $11/32$ inch; OBT4 is at $\theta = 120$ degrees and penetrates the outer cylinder wall to a depth of $9/16$ inch. Like the inner cylinder instruments, the outer cylinder thermocouples read out on a Minneapolis Honeywell recorder that has been internally cold-unction compensated. Figures 6 and 7 show the location of the thermocouples.

Seven copper/constantan thermocouples measure water temperatures in the heat exchanger and read out on a Minneapolis Honeywell Multipoint recorder. These thermocouples are made from ISA Special Limits of Error Wire and are considered matched. Thermocouples WT1, WT4, and WT5 are at the same radial position, $\theta = 0$ degrees, but at different axial positions; similarly, WT2, WT3, WT6, and WT7 are at $\theta = 180$ degrees but at different axial positions. (figures 6 and 7).

Pressure probes measure static pressures at the inlet to the test section, between the first and second land and groove bands, between the second and third land and groove bands and in the exit plenum chamber. The probes, designated P1, P2, P3, and P4 are at $\theta = 70$ degrees. (figures 6 and 7). The inlet and exit plenum probes read out on a recorder in the test control room; the other probes read out on gauges.

As shown in figure 8, the test piece is installed in a system of flow controls and data indicating and recording instruments. Automatic valves control the pressure of hydrogen gas supplied to and exhausted from the test model. Pressure levels as high as 250 psig and pressure differentials as high as 30 psi can be obtained. The flow to the test model is measured as the pressure drop across a calibrated sharp-edged orifice at an upstream gas density, and is indicated on a gauge. Another gauge indicates the pressure upstream of the orifice. A copper/constantan thermocouple measures the temperature of the gas upstream of the orifice and reads out on a

Figure 8. Schematic Flow Loop

Minneapolis Honeywell Multipoint recorder.

Water flow is regulated by a manual valve; a Wallace Tiernan rotameter measures and indicates the flow. Up to four gallons per minute can be obtained.

The electric power for the system is supplied by a direct current Westinghouse industrial welder supply, located in the control room. Two water cooled leads conduct current to the electrodes on the test model. A series of switches on a console control the power level up to a maximum of 40 kw. Power is calculated from measurements of current and voltage in the heater element. A high resistance dc voltmeter in parallel with the element measures the voltage; a shunt calibrated to read a 50 millivolt drop for every 1000 amperes measures the current.

As a safety precaution all flow connections to the test model contain a section of plastic tubing at least six inches long. The rest of the flow system is tied into a grounded water system.

3.1.1.1 Significant "as-built" dimensions

A complete dimensional inspection of the plenum seal, and seal groove sizes was made on the test hardware, as installed and assembled, so that the model used in analysis would reflect the "as-built" condition of the hardware. The results of the dimensional inspection are summarized in figure 9 and tabulated in detail in Appendix 5.1.

Figure 9. Results of Dimensional Inspection

3.1.2 Test Procedure

Specific step by step instructions for starting, running, and shutting down the system are prepared for each test run. The general procedure for all test runs is described here.

After the system has been purged and evacuated, the test piece is pressurized to 70 psia with helium through automatic valve VP1. (figure 8). Water is then turned on to the copper heat exchanger, the pressure vessel outlet flange, the electrodes, and the power leads. At the indicating control station, outlet pressure P4 is set at the desired value and, at the recording control station, gas flow is established by setting inlet pressure P1 higher than P4. Hydrogen is then substituted for the helium by simultaneously opening the hydrogen supply valve and shutting the helium supply valve.

Using the switches on the control console, power to the welder supply is turned on, and the current to the heater element is raised to the specified level. Current is raised slowly to prevent the occurrence of large transient thermal gradients across the test piece. Temperature HT1 is monitored continuously, and great care is taken not to exceed a startup heating rate of 500°F per minute. At elevated temperatures hydrogen will corrode the heater, and a limit of 3000°F was established in the interest of prolonging heater life.

Once the desired power level is reached, the final hydrogen flow conditions are set. Water flow is throttled to establish a minimum rise of 50°F between the heat exchanger inlet and outlet. Ample time is allowed for the system to reach equilibrium; P1, P4, ΔP_{01} , ΔP_{1-2} , ΔP_{2-3} , ΔP_{3-4} and the material and water temperatures must be stable.

During the test run the values of P1, P4, P01, ΔP_{01} , ΔP_{1-2} , ΔP_{2-3} , ΔP_{3-4} , PIR, POR, HT1, and heater element current and voltage are recorded on the log sheet, which is shown in table 5.3 in Appendix 5.2 for Build 3 Run 2. The Minneapolis Honeywell recordings are marked at points corresponding to log sheet points, and the pertinent data is tabulated in table 5.4 of Appendix 5.2.

After each test run the calibration of each pressure instrument is checked and the post calibration recorded. A rig consisting of two Heise gauges and a series of valves is used as a standard in plotting calibration curves before and after a test run for each pressure instrument.

Before and after each test run all thermocouples are checked for continuity. Multi-point recorders are checked point by point with a WANL Standards Laboratory potentiometer to ensure that the reading for a given millivolt input signal is correct. The Standards Laboratory periodically checks the voltmeter and ammeter used to measure power to the heater element. The rotameter is checked by measuring the amount of water which flows into a bucket in a prescribed time period. After every test run the flow system and the test piece are pressurized to 250 psig with helium and completely leak-checked.

The results of all pre- and post-run calibrations and any leaks discovered in the post-run check are logged on an auxiliary sheet, which is also shown in table 5.5 in Appendix 5.2 for Build 3 Run 2.

By choice of orifice size an attempt is made to use the pressure and flow measuring instruments in the upper portion of their measuring range. Estimated expected errors in measurements both on a relative and absolute basis take into consideration inherent accuracy of the instrument and the closeness to which it can be read under test stand conditions. Accuracies on a relative basis involve ability to reproduce data without reference to a standard; accuracies on an absolute basis involve comparison to a standard or to data from other test facilities. On a relative basis, at the level at which the readings were taken, estimated expected errors of pressure level are 2.5 percent; pressure differences, 2 percent; hydrogen weight flow measurements, 1.5 percent; water flow measurements, 2 percent; and voltage and amperage, 1 percent each.

On an absolute basis all estimated expected errors are considered approximately double those quoted for the relative basis. It is estimated that the thermocouple readings on a relative basis are accurate within the tolerances listed below:

<u>Thermocouple</u>	<u>Tolerance (in degrees F)</u>
Copper/constantan	± 0.5
Chromel/alumel	± 2.0
Tungsten/tungsten-rhenium	± 30.0

Conduction effects of thermocouple sheaths as fitted in the graphite cylinders are considered to vary the temperature of the hot junction by no more than $\pm 15^{\circ}\text{F}$. The effect of thermal radiation

on the readings of these shielded stagnation thermocouples is considered negligible. At both the upstream and downstream locations of the stagnation thermocouples in the gas stream, the sheaths are immersed in the flowing hydrogen for several inches back from the hot junction; the conduction effect is considered negligible.

3.2 Test Results

Results obtained from the first series of tests demonstrate good trends as a function of hydrogen flow. This is demonstrated in figures 10, 11, 12, and 13. Figure 10 shows the inner barrel graphite temperatures decreasing, in general, as hydrogen flow increases with power input constant. Thermocouple IBT3 was producing random readings during this run; accordingly these results are not plotted. Figure 11 shows the outer barrel graphite temperatures decreasing, with the exception of OBT1, as hydrogen flow increased. This trend along with the flattening out of the IBT1 curve in the previous figure suggests an axial temperature distribution in the heater element which changes as a function of hydrogen flow. Note that IBT1 and OBT1 are both upstream of the other thermocouples measuring graphite temperatures. (figures 6 and 7). This change of heater element temperature distribution can profitably be investigated in future testing. Figure 12 plots decreasing values of water temperature in the heat exchanger as hydrogen flow increases. In addition, temperature levels plot smoothly as a function of axial location. (figures 6 and 7). The fact that three of the thermocouples are 180 degrees from the other four in a circumferential sense is indicative that there is good basic circumferential alignment in this Build 3 Run 2. Figure 13 shows the entrance and exit gas temperatures decreasing as functions of increasing hydrogen flow.

Thermocouple WT1 after test indicated a short on recheck of resistance level; the data read by the thermocouple, however, is considered valid and was included in the curves since the temperature levels as read during this run are compatible with similar previous runs. The presumption

Figure 10. Inner Barrel Material Temperatures - Hydrogen Weight Flow

Figure 11. Outer Barrel Material Temperatures - Hydrogen Weight Flow

Figure 12. Heat Exchanger Water Temperature - Hydrogen Weight Flow

Figure 13. Hydrogen Gas Temperatures - Hydrogen Weight Flow

is that the thermocouple became shorted after the run was completed.

3.3 Comparison of Test Data with Analysis

3.3.1 Method of Analysis

The thermal and hydraulic analysis of the B-3 hot seal rig was performed with the aid of the SHYP-TOSS chain routine designed for the IBM 7094 computer. The routine embodies:

- 1) The steady state SHYP (Seal Hydraulic Program), previously used for analysis of the G-3, Plugged Core, Test data¹ and
- 2) The steady state option of TOSS⁵ (Transient or Steady State Temperature Distributions).

The SHYP and the TOSS programs are linked by control programs which evaluate the results of the preceding program, prepare input, boundary conditions, execute iteration procedures between SHYP and TOSS, and compare conduction and convective results to determine compatible solutions within specified convergence limits. The compatible solution is defined as converged when the difference in wall channel surface fluxes, as calculated by succeeding iterations of TOSS, is less than one percent.

The hydraulic model, as utilized by SHYP, is a single annular flow channel consisting of a series of axial sections having alternating large and small flow areas, dimensioned to represent the flow channel in the experimental apparatus. Along the annulus walls, provisions have been made for heat addition or removal.

Neither flow past inner cylinder lands nor porous flow were incorporated in the hydraulic model. As extruded ZTA and impregnated H4LM graphites were used for the cylinder walls, porous flow was assumed to be negligible for pressure differentials in the test range.

1 WANL-TME-792, Plugged Core Flow Test - Phase 1 Tests Prerequisite to NRX-A2, C. V. Buzza, et al, September 1964

5 Modified Transient and/or Steady State (TOSS) Digital Heat Transfer Code, B. L. Pierce, April, 1964, unpublished.

The hydraulic analysis, which incorporates SHYP, determines the temperature and pressure distributions in the coolant flow channel and the heat transfer coefficient along the channel walls when convection occurs. The test conditions used in the hydraulic analysis were the experimental value of the overall axial pressure differential, and the temperature and pressure of the hydrogen at the entrance plenum.

The conduction model, as utilized by TOSS, is shown in figure 14. It includes that section of the inner and outer graphite cylinders between the axial midplane of the first seal and the axial mid-plane of the last seal. Temperatures of surfaces bounding the hydrogen flow channel which lie beyond the ends of the conduction model were obtained from the hydraulic analysis.

To simulate the effect of the heater in the experimental rig, a constant heat flux was applied to the inner surface of the model. The effective heating surface was assumed to commence at a plane axially located at the entrance of the first simulated seal and to terminate at a plane axially located at the exit of the last simulated seal.

In the analytic model, heat is transferred to the hydrogen coolant by convection from the cylindrical surfaces of the inner and outer cylinders which bound the flow channel, both in the plenums and in the simulated seal grooves.

In the experimental rig, the outer cylinder is cooled by water flowing in a close-fitting copper jacket with axial fins on its inner diameter to increase the effective heat transfer area. The heat transfer coefficient of the finned surface was determined by correcting the heat transfer coefficient obtained from the empirical equation suggested by Sieder and Tate for laminar flow in rectangular channels⁶ for the weighted fin efficiency.

⁶ Principles of Heat Transfer, F. Kreith, International Textbook Company, Scranton, Pa. 1958 (page 359)

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Figure 14. SHYP-TOSS Model

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$$h = 1.86 \left(\frac{K}{D_e} \right) \left(\frac{Re Pr D_e}{L} \right)^{0.33} \left(\frac{\mu_b}{\mu_s} \right)^{0.14} \quad (1)$$

where,

- D_e = Equivalent Diameter of Channel
- h = Heat Transfer Coefficient
- L = Length of Channel
- Pr = Prandl Number
- Re = Reynolds Number
- μ_b = Fluid Viscosity Evaluated at Bulk Temperature
- μ_s = Fluid Viscosity Evaluated at Surface Temperature

The effective film coefficient on the inner diameter of the water jacket was determined by the equation

$$h_{eff} = \frac{f (Ah)_{fin surface} + (Ah)_{outer surface}}{(A)_{inner surface}} \quad (2)$$

where,

- A = Area of surface
- f = Geometry and Material Factor (Obtained by Methods Discussed in reference 4)
- h = Heat Transfer Coefficient of Surface

The heat transfer coefficient thus calculated was applied as a boundary condition on the outer diameter of the outer cylinder.

In establishing this boundary condition it was assumed that:

1) the radial conduction of the copper jacket was sufficiently high to assure a negligible temperature difference between its two surfaces, and that

4 Process Heat Transfer, D. Q. Kern, McGraw-Hill Book Company, N Y , 1950, pg 515

2) the axial conduction in the copper jacket was sufficiently small to assure a negligible effect on surface temperatures.

A separate study showed that, for realistic values of axial thermal gradients and radial thermal fluxes in the copper jacket, the above assumptions were valid.

Water temperature was determined experimentally at seven stations along the channel and this temperature distribution was applied as a boundary condition on the outer diameter of the outer cylinder in the conduction model to facilitate calculation of the heat flux to the cooling water.

3.3.2 Results of Analysis and Comparison With Experimental Data

Boundary conditions for use in the analysis were obtained from the experimental data of Build 3, Run 2, Point 7. This data point was selected for detailed analysis because the high heater power coupled with the low hydrogen flow rate resulted in relatively large differences in the material and coolant temperature levels in various sections of the rig, and relatively small variations between symmetrically located temperature and pressure sensors. This data point showed a minimum of inconsistencies in the measured radial temperature gradients through the graphite cylinders compared to the other data points at higher hydrogen flow rates.

Test conditions, determined experimentally, which were applied to the model were as follows:

- 1) Total hydrogen gas pressure drop across test section (16.0 psi).
- 2) Hydrogen gas pressure and temperature at the entrance plenum (217.3 psia and 598 R, respectively).
- 3) Electrical power to the heater from which the heat flux on the inner surface of the inner cylinder was determined (26.5 kw).
- 4) Axial temperature distribution in the water coolant as shown in figure 19 and a heat transfer coefficient of $588 \text{ Btu/hr-ft}^2\text{-R}$ for coolant at the outer diameter of the outer cylinder.

Using the experimental hydrogen gas pressure drop, the inlet plenum pressure, the inlet plenum temperature, and power input to the model, the hydraulic part of the SHYP-TOSS chain routine calculates the temperature and pressure of the hydrogen gas at three points in each simulated seal flow channel and each chamber between the simulated seals and the hydrogen gas flow rate.

The calculated pressure distributions are compared to the experimental pressure measurements in figure 15. The calculated values for the pressure drops across individual seals are compared to the experimentally determined values in table 1.

The calculated hydrogen flow rate was 0.0076 lb/sec as compared to an experimental value of 0.0092 lb/sec. The calculated temperature rise in the hydrogen gas was 25 R higher than the experimental value. The fact that the calculated flow rate was 18% lower and the temperature rise was 8.5 % higher than the experimental measurements indicates that part of the total flow may be by-passing the seal constrictions via interface leakage at the seal lands. This hypothesis is partially substantiated by analyses of the tolerance build-up and differential thermal expansion of the inner and outer cylinders which indicate that a clearance of about 0.5 mils could exist between the seal lands and the outer cylinder during test operation. If this quantity of leakage did occur, the calculated pressure distribution would be expected to remain quite similar to that shown in figure 15 but the calculated gas temperature rise would decrease.

Inspection of the experimental data shows that the static pressure drop per seal decreases along the length of the flow channel. Since the density of the hydrogen gas continuously decreases as it flows through the channel due to continuous heat addition and pressure losses (friction, sudden contractions, sudden expansions) while the area of the grooves remain approximately constant (See Appendix B), the static pressure drop per seal would be expected to increase. As the intermediate static pressure taps were very close to the exit of the grooves (1/8 and 3/16 inches respectively), these taps could be affected if kinetic energy of the gas were not dissipated in turbulence upon entering the plenum. The causes of the apparent discrepancy between analysis and experiment should be resolved by further investigations.

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Figure 15. Hot Seal Test, Point 7, Run 2, Build 3

Table 1

Tabulation of Pressure Drop Across Simulated Seals

	Impedance	
	Experimental Value psi	Calculated Value psi
Pressure Drop Across First Simulated Seal	6.0	4.7
Pressure Drop Across Second Simulated Seal	5.6	5.8
Pressure Drop Across Third Simulated Seal	4.4	5.5
Total Pressure Drop*	16.0	16.0

* Total pressure drop was used to calculate hydrogen flow rate and individual seal impedances.

The calculated temperatures of the hydrogen gas at the exit plenum was within 25 degrees of the experimental value. As the hydrogen gas temperature was measured only at the entrance and exit plenum, it is not possible to compare the calculated temperature distribution with the experimental values. However, the difference between the value calculated for the overall gas rise and that obtained experimentally would be reduced if the effects of interface leakage at the seal lands were considered. A separate study indicated that with the measured flow rate passing through the annulus, the hydrogen gas exit temperature was reduced 15 R.

The calculated material temperature distribution in the graphite inner and outer cylinders, together with the temperatures of the hydrogen coolant and water are shown in figures 16, 17, and 18. The experimentally determined temperatures have been included for comparison. Figures 17 and 18 show that the temperatures measured by thermocouples IBT 1 and IBT 4 are both over eighty degrees higher than the calculated temperatures at their respective positions. Both OBT 3 and OBT 4 are over 70 degrees lower than the calculated values.

The remaining calculated temperatures, show less discrepancy when compared with experimental data. These smaller discrepancies could be the result of the following analytical uncertainties.

- 1) The heater may be acting more as a constant temperature source than as a constant heat flux source in the low temperature region of the first seal.
- 2) The "adiabatic end" boundary condition may not be realistic. The material temperature in the region of the first seal would increase if the preceding region of the entrance plenum were heated.

The temperature levels show fair agreement, although the temperatures measured in the graphite may not be the average circumferential temperatures at that radius. The experimental radial temperature profiles thus obtained for the center seal do not progress reasonably. This condition may be attributable to circumferential temperature gradients and material non-homogeneities in the cylinders, although to maintain such gradients in the graphite would require circumferential heat fluxes of the same order as the radial fluxes. Circumferential temperature distribution should be the subject of continued analytical and experimental investigations.

Figure 16. Axial Temperature Distributions, Build 3, Run 2, Point 7

Figure 17. Radial Temperature Profile at Center Seal

Figure 18. Radial Temperature Profile at First Seal

To improve the analysis, the boundary condition must be known with more accuracy. The reliability of the boundary conditions are discussed in detail in section 3.3.3 Adequacy of Analytical Techniques, of this report.

The SHYP-TOSS chain routine incorporates a flux balance program to assure convergence within one percent along the hydrogen coolant channel. Thus the solution insures compatibility of heat fluxes and temperature rises in this channel. Similar compatibility is not assured for the outer boundary since only the film coefficient and an "initial" water temperature distribution were input as boundary conditions. The SHYP-TOSS program computes a heat flux at this outer boundary and, when combined with the known water flow rate and surface areas, a water temperature distribution can be calculated. This "calculated" water temperature distribution is compared to the "initial" temperature distribution and to the experimental data in figure 19. As can be seen, reasonable compatibility exists throughout the simulated seal region. The differences between the temperatures used in calculations and experimental points increases along the flow path. In the region of the first two seals, the maximum difference is 6°F. In the region of the last seal, this difference increases to 15°F.

The overall heat balance for the experimental rig is plotted in figure 20 as a function of the hydrogen flow rate while maintaining constant power to the heater. When calculated values of the heat removed by the hydrogen and heat removed by the water are compared with similar experimental values the maximum error is less than 5 percent.

3.3.3 Adequacy of Analytical Techniques

Prior use of both the SHYP and TOSS computer programs separately, has shown that both give reasonably reliable results, being limited primarily by the accuracy of the model to reflect the actual test conditions and the degree of detail with which the test article configuration could be modeled. The principle limitations on the accuracy of the heated seal test analysis are in establishing accurate boundary conditions for the analytical model. In the analysis, the following conditions were imposed upon the model.

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Figure 19. Axial Temperature Distribution in Water Channel

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Figure 20. Overall Heat Balance, Build 3, Run 2

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1) The effect of the heater was to produce a constant flux upon the inner surface of the inner graphite cylinder from the entrance of the first simulated seal to the exit of the third simulated seal. There was no heating beyond these points.

2) The temperature distribution in the water channel was established by experimental flow rate and dimensional data of the test rig.

3) There was no heat loss by either conduction or radiation through the ends of the conduction model.

The generally favorable agreement between experimental data and calculated values for selected variables as shown in table 2, indicates that the analytical techniques appear to be adequate to describe the thermal and fluid flow characteristics of the test system.

As the same analytical techniques were used to determine material temperatures, axial pressure distribution, gas temperatures, and flow rates in the reactor lateral support system with heat addition, the first results of this test series provide a basis upon which to assure a higher degree of confidence in the calculated thermal characteristics of the reactor lateral support seal system. The differences between the reactor lateral support system and the hot seal test are in the boundary conditions and the heat source. The model has an external heat source (electrical heater) whereas the reactor has internal heat generation.

Since the boundary conditions used in the analysis of the lateral support system of the NRX-A2 differ from those used in the analysis of the hot seal test, any error which may be induced by uncertainties in specifying the boundary conditions of the latter, would not necessarily affect the adequacy of the NRX-A2 lateral support system analysis. However, for purposes of comparing the test results with the analysis of the hot seal test, it is desirable to consider possible errors which might result from the use of improper boundary conditions.

The first boundary condition, that of constant heat flux on the inner surface of the inner graphite cylinder, was applied under the assumptions that the graphite resistance heater has a constant resistance along its effective length and that the distance between the heater surface and the inner surface was sufficiently small to preclude axial radiation and conduction

Table 2
Comparison of Experimental Data Calculated Results

	<u>Experimental</u>	<u>Calculated</u>
Hydrogen flow rate	0.0092 lb/sec	0.0078 lb/sec
Water flow rate	2 gpm	2 gpm*
Heater power	26.7 kw	26.7 kw *
Pressure at entrance plenum	217.3 psia	217.3 psia*
H ₂ temperature at entrance plenum	598 R	598 R*
Pressure at exit plenum	201.3 psi	201.3 psi
H ₂ temperature at exit plenum	897	919 R
Pressure Drop - Hydrogen Channel	16.0 psi	16.0 psi*
Temperature rise - hydrogen channel	299 R	321 R
Temperature rise - water channel	51 R	59 R
Water Temperatures		
WT 1	82 F	90 F
WT 2	90 F	90 F
WT 3	105 F	102 F
WT 4	111 F	108 F
WT 5	115 F	122 F
WT 6	142 F	132 F
WT 7	133 F	149 F
Material Temperatures		
IBT 1	969 R	870 R
IBT 2	928 R	953 R
IBT 4	1010 R	925 R
OBT 1	734 R	703 R
OBT 2	750 R	760 R
OBT 3	709 R	780 R
OBT 4	711 R	800 R
OBT 5	766 R	825 R

*Indicates data which were used as input to analysis.

affecting the local surface flux. This assumption would be in error if the electrical resistance in the heater were not constant but highly influenced by internal variations, or by surface corrosion. It is also possible that non-isotropic properties in the graphite or high thermal conductivity could cause the element to vary from a constant heat flux device, especially in the regions of low material temperatures near the first seal. To obtain a better understanding of the boundary condition it will be necessary to obtain a better knowledge of the heater characteristics and to incorporate a model of the heater in the analysis.

The temperature distribution in the water channel was determined experimentally and applied to the analytical model. The heat transfer coefficient along the surface was calculated by empirical equations and adjusted so as to apply directly to the outer surface of the outer graphite cylinder. In so doing, it was assumed that the graphite to copper interface thermal resistance, the temperature drop across the copper jacket, and axial heat flow in the jacket were all negligible, and thus, could be ignored. A separate study indicated that for realistic values of temperature and heat flux, the axial heat flow in the copper, the radial temperature drop in the copper, and the graphite to copper interface thermal resistance were all small.

To assure a balance between heat flux to the water and temperature rise, it would be necessary to iterate along the water channel as well as along the hydrogen channel. However, the discrepancy between the two resulting temperature distributions did not prove to be sufficient to justify this.

The final boundary condition is that of adiabatic ends. The result is a region near the end of the analytical model which varies significantly from the test model. To allow for this, the test model is normally extended beyond the farthest point where reliable calculations are desired. However, in the present case the region of analytical interest extended between the axial stations corresponding to the entrance plenum of the first seal and exit plenum of the third seal.

4.0 BIBLIOGRAPHY

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5.0 APPENDIX

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5.0 APPENDIX

5.1 "As-built" Dimensional Data

Table 5-1

Outside Diameter of Inner Cylinder at Seal Lands

Reading No.	Outer Diameter of Seal		
	Seal*	Seal*	Seal*
	(1)	(2)	(3)
1	3.8738	3.8736	3.8737
2	3.8738	3.8737	3.8737
3	3.9739	3.8736	3.8736
4	3.8739	3.8738	3.8737
5	3.8739	3.8738	3.8737
6	3.8740	3.8740	3.8736
7	3.8740	3.8739	3.8735
8	3.8740	3.8740	3.8736
9	3.8739	3.8738	3.8737
Avg.	3.8739	3.8738	3.8736

*Seal (1) refers to the flow constriction closest to the open end of the inner barrel.

Seal (2) refers to the flow constriction closest to the center of the inner barrel.

Seal (3) refers to the flow constriction closest to the closed end of the inner barrel.

Table 5-2

Seal Groove Dimensions

Reading No.	Width			Depth		
	Seal (1)	Seal (2)	Seal (3)	Seal (1)	Seal (2)	Seal (3)
1	.4340	.4387	.4426	.0062	.0065	.0070
2	.4338	.4380	.4435	.0067	.0066	.0071
3	.4354	.4379	.4452	.0066	.0065	.0071
4	.4356	.4378	.4435	.0067	.0068	.0072
5	.4361	.4376	.4427	.0067	.0067	.0073
6	.4367	.4376	.4432	.0067	.0069	.0073
7	.4347	.4391	.4378	.0062	.0069	.0073
8	.4357	.4387	.4393	.0066	.0070	.0072
9	.4364	.4379	.4388	.0062	.0067	.0071
10	.4354	.4392	.4382	.0064	.0065	.0073
11	.4362	.4390	.4398	.0066	.0066	.0070
12	.4362	.4391	.4400	.0069	.0068	.0070
13	.4349	.4382	.4425	.0069	.0068	.0071
14	.4358	.4392	.4440	.0068	.0070	.0076
15	.4355	.4384	.4438	.0069	.0070	.0078
16	.4350	.4386	.4433	.0068	.0071	.0077
17	.4350	.4390	.4454	.0063	.0067	.0075
18	.4352	.4398	.4441	.0067	.0065	.0075
Avg.	.4354	.4385	.4421	.0066	.0068	.0073

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5.2 Log Sheets and Data Tabulations for Build 3 Run 2

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[illegible]

FLUID FLOW LAB
LOG SHEET

SMALL CIRCULAR HIGH PRESSURE HIGH TEMPERATURE
HEAT TRANSFER TEST BUILD NO. 3

TEST RECORD BOOK NO 1735739 RUN NO 2 SHEET 1 OF 1
TIME OF DAY 1345 DATE 9 July 54

ENGINEER J.R. Perry, S. V. Babin, R.T. Steffen OPERATOR FW, Mr. R. Lazzaro

BAR PRESSURE 29.22 @ 78 °F TEST GAS Hydrogen

SUPPLY PRESSURE BEFORE 1,500 PSI SUPPLY PRESSURE AFTER 1,200 PSI

SUPPLY TEMP BEFORE 71 °F SUPPLY TEMP AFTER 75 °F

VACUUM PRIOR TO TESTING 10 INHG ABSOLUTE

VACUUM AFTER TESTING 10 INHG ABSOLUTE

CONFIGURATION

DIMENSION A 007 INCHES TYPICAL

ORIFICE SIZE

ORIFICE OF 2.331 INCHES

Table 5.3 Fluid Flow Lab, Log Sheet, Small Circular High Pressure, High Temperature, Heat Transfer Test, Build No. 3, Run 2

BUILD NO. 3 DATE 7-9-64
 RUN NO. 2 TEMPERATURE IN DEGREES F

SMALL CIRCULAR HIGH PRESSURE HIGH TEMPERATURE HEAT TRANSFER TEST

FLUID FLOW LAB
TABULATION SHEET
TEST SERIES FFL-2

POINT	H ₂ FLOW	AP ₁₋₂	AP ₂₋₃	AP ₃₋₄	INPUT	HTI	TEMP	OBT1	OBT2	OBT3	OBT4	OBT5	ST6	ST7	CT8	IBT1	IBT2	IBT3	IBT4	CT5	CT6	ST5	WT1	WT2	WT3	WT4	WT5	WT6	WT7	TO1	ST8	ST9	HEE*	REMARKS
NO	LB/SEC	PSID	PSID	PSID	KW	°F	RP*	1 16	2	3	4	5 21	6 17	7	8	9 15	10 18	11 24	12	13 22	14 19	20 23	1 11	2 12	3 13	4 14	5 15	6 16	7 17	8 18	9 19	10 20	21 22	
1	.0109	7.9	7.5	5.9	26.45	2335		284	273	221	229	298	↑	402	100	483	400	↑	507	160	140	380	82	88	101	109	113	133	124	82	140	130	92	
2	.0121	9.6	9.1	7.4	26.45	2335		288	262	209	221	292		391	100	480	370		492	148	130	360	82	88	100	109	112	129	122	81	129	128	95	
3	.0128	10.5	10.3	8.0	26.18	2285		290	253	200	221	288		380	100	479	350		486	140	128	340	82	88	99	109	112	127	120	81	126	127	97	
4	.0142	11.3	10.6	8.5	26.35	2190		299	244	188	219	281		368	100	480	320		475	131	120	320	82	87	97	108	110	123	115	80	121	127	100	
5	.0153	12.1	11.3	9.0	26.55	2100		300	237	180	217	281	N.G.	358	100	479	309	N.G.	470	130	120	305	82	87	97	108	110	122	115	79	120	125	100	
6	.0100	7.1	6.6	5.2	26.75	2430		275	276	231	244	300		417	100	491	430		530	161	141	401	82	89	102	110	113	136	127	80	142	129	96	
7	.0092	6.0	5.6	4.4	26.70	2610		274	290	249	251	306		477	100	509	468		550	175	150	430	82	90	105	111	115	142	133	81	146	131	95	
8	.0084	5.1	5.0	3.6	26.55	2700		271	301	262	260	309		451	100	521	500		574	186	156	451	82	90	106	111	116	144	135	82	150	134	95	
9	.0103	6.3	5.9	4.2	0	0		82	82	82	82	82	↓	82	82	82	82	↓	82	82	82	82	81	82	82	82	81	82	81	81	82	82	85	

* RP = RECORDER POINT HEE = HEAT EXCHANGER EXIT

READING BEFORE TEST															
PSIG			PSID				INH ₂ O		PSIG		GPM		MV		REMARKS
INPUT	P 1	P 4	INPUT	ΔP 1-2	ΔP 2-3	ΔP 3-4	INPUT	ΔP OI	INPUT	P OI	INPUT	H ₂ O FLOW	INPUT	T HTI	
0	3.0	3.0	0	0	0	0	0	0	0	0	0	0			Volt meter and Amp meter sent to standards Lab.
75	6.0	6.0	5	3.0	5.0	5.0	25	2.5	100	100	2	2			
150	9.0	9.0	10	10.0	10.0	10.0	50	5.0	200	200	4	4			
225	12.0	12.0	15	15.0	15.0	15.0	75		300	300	6				
300	15.0	15.0	20	20.0	20.0	20.0	100		400	400	8				
READING AFTER TEST															
0	3.0	3.0	0	0	0	0	0	0	0	0	0	0			Volt meter and Amp meter sent to standards Lab.
75	6.0	6.0	5	5.0	5.0	5.0	25	2.5	100	100	2	2			
150	9.0	9.0	10	10.0	10.0	10.0	50	5.0	200	200	4	4			
225	12.0	12.0	15	15.0	15.0	15.0	75		300	300	6				
300	15.0	15.0	20	20.0	20.0	20.0	100		400	400	8				

FLUID FLOW LAB
AUXILIARY LOG SHEET
SMALL CIRCULAR HIGH PRESSURE HIGH TEMPERATURE
HEAT TRANSFER TEST BUILD NO 3

TEST RECORD BOOK NO 1735740 RUN NO 2 SHEET 1 OF 1
TIME OF DAY 0950 hrs DATE 14 July 64
ENGINEER R.J. Steffen OPERATOR F. Wagner
BAR PRESSURE 29.45 @ 83 °F TEST GAS Helium
CALIBRATION STANDARD PNEUMATIC Hesse Gauge
CALIBRATION STANDARD THERMOCOUPLE VTVM

READING BEFORE TEST																							ST 5	ST 8	ST 9	TO 1	Bar Temp	REMARKS																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																				
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