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**DEFICIENT COOLING
SECOND QUARTERLY PROGRESS REPORT
OCTOBER 1 - DECEMBER 31, 1969**

U.S. ATOMIC ENERGY COMMISSION
CONTRACT AT(04-3)-189
PROJECT AGREEMENT 55

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DEFICIENT COOLING
SECOND QUARTERLY PROGRESS REPORT
October 1-December 31, 1969

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SUMMARY

This report covers progress in the second quarter, October 1–December 31, 1969, for the Deficient Cooling Program, for which work was authorized on June 23, 1969.

Detailed planning and scheduling of one- and nine-rod tests to be performed in the third and fourth quarters were completed. Design and procurement of materials for one-rod power and flow cycling tests was completed. Procure-

ment of calrod heaters for the nine-rod tests was initiated. Calrods with both uniform and nonuniform axial heat flux variations were ordered.

Analyses and procedures for data reduction of system pressure and flow transients, as measured by either a turbine- or orifice-meter, were completed.

1. INTRODUCTION

1.1 GENERAL

The work on the Deficient Cooling Program is directed toward extension of out-of-pile heat transfer data in support of current nuclear-powered, water-cooled plants. A brief review of current technology reveals that the large bulk of data is of the steady- or quasi-steady-state class. Thus, a need exists for data taken under transient conditions of power, flow and pressure in combination with different geometries. Analysis and study of such data are expected to contribute to the following overall objectives:

- a. A firmer understanding and establishment of heat transfer safety limitations, and
- b. Extension of safe operating limits to higher power densities.

Deficient cooling conditions are considered to be those which result in inadequate cooling of the fuel. They may result in potentially excessive cladding temperature which, in combination with the internal gas pressure of the fuel, may cause some type of fuel disruption. Resultant temperatures depend on the mode or modes of heat transfer that arise during deficient cooling conditions. A brief description of these modes of heat transfer follows.

Three different modes of heat transfer can generally occur when heat is removed, during high quality conditions, from a surface such as a fuel rod: nucleate boiling, with a liquid film on the heated surface; transition boiling; and the liquid deficient region. Nucleate boiling occurs at very high surface heat transfer coefficients, usually in excess of 10,000 Btu/h-ft², with steady surface temperatures. Increase of heat flux eventually results in "film dryout" and a change of mode to transition boiling, which is usually characterized by temperature oscillations of varying amplitude on the heated surface. The transition mode of heat transfer exhibits relatively low heat transfer coefficients, on the order of 1000 Btu/h-ft² or less. Continued increase of heat flux eventually results in the liquid deficient regions,

with some liquid droplets impinging on the heated surface which is mostly steam cooled.

The heat flux at which transition is made from the nucleate to transition mode of boiling, as evidenced by onset of surface temperature oscillations, is usually termed the critical heat flux (CHF). For low qualities, it is also known as the departure from nucleate boiling (DNB), or burnout point. Considerable effort has been made on parametric definition of the CHF with pressure, coolant flow, vapor quality, and the geometry of the system. These parametric studies serve as design data and indicate the operating conditions beyond which fuel surface temperatures increase substantially, with greater risk to fuel cladding integrity.

Most of the work defining limits of CHF has been done with small incremental heat flux increases which would allow it to be classified as steady- or quasi-state data. However, the actual modes of heat transfer become more complex in some cases involving transients, when the dynamic characteristics of the fuel-coolant system come into play. Therefore, practical cases of transient conditions which are considered worthwhile for study and test under this Deficient Cooling Program are those involving:

- a. Transient reduction of flow,
- b. Transient increase of power,
- c. Oscillation of power into and out of the transition boiling region to determine effects of prolonged operation at CHF, and
- d. Transient decrease of pressure which causes flashing and potentially deficient cooling conditions due to increased void content.

Furthermore, it is proposed to study under transient conditions typical geometry changes such as those due to fuel spacer components and those caused by rod bowing and swelling. Such studies are expected to define more

clearly heat transfer safety limitations, and will eventually result in extension of safe operation to higher limits.

1.2 PROGRAM OBJECTIVES

The objectives expected to be accomplished under this program are as follows:

- a. Determine experimentally the CHF and temperature regimes which may occur in water reactor fuel assemblies due to nonstandard cooling conditions that result from power, flow, or pressure transients, and determine these values relative to steady-state data.
- b. Evaluate temperature regimes likely to be experienced by water reactor fuel assemblies in loss-of-coolant accidents, and determine areas of similarity or relationships between these regimes and those found in transient critical conditions.

- c. Evaluate consequences of fuel rod geometry changes on heat transfer performance from simulated tests on single, electrically-heated rods.
- d. Provide a plan for further in- and out-of-pile work necessary for more complete performance and definition of the performance of water reactor fuel under deficient cooling conditions.

1.3 ORGANIZATION OF PROGRAM

The outline of tasks, as originally proposed for the entire program, is shown in Table 1, with brief reference to the equipment and type of tests. It is intended to preserve these task reference designations in this and subsequent reports so that documenting of work and progress can be made in a systematic manner.

Tasks 3, 4, 5, and 8 involve test work. Chronological sequence of testing will probably proceed in the numerical task order.

Table 1
TASK DESIGNATION AND CONTENT

No.	Title	Content
1	Planning and Analysis	<ol style="list-style-type: none"> a. Review current technology and plan appropriate tests to integrate with required safety programs b. Analysis as necessary to determine test parameters
2a	Design	<ol style="list-style-type: none"> a. Design new equipment or modifications of existing equipment to carry out test plans
2b	Manufacture	<ol style="list-style-type: none"> a. Manufacture or purchase new equipment required
3	One-Rod Exploratory Transient CHF Tests	<p>Exploratory tests involving:</p> <ol style="list-style-type: none"> a. Steady-state and transient flow, power, and pressure b. Cycling of power and flow into the region of CHF transition boiling c. Prolonged exposure to CHF conditions d. Modifications of heaters for nonuniform axial heat flux

Table 1 (Continued)

No.	Title	Content
4.	Multirod Transient Tests	Tests involving: <ol style="list-style-type: none"> Steady-state and transient conditions Power and flow cycling Combinations of intrabundle clearance, power distribution, and spacer components with a. and b.
5.	One-Rod Tests with Geometry Changes	<ol style="list-style-type: none"> Effects of temperature and internal pressure of heater swelling Steady-state simulated swelling at constant coolant pressure drop
6.	Data Reduction and Analysis	<ol style="list-style-type: none"> As required for test Tasks 1, 3, 4, 5, and 8
7.	Project Administration, Reporting, and Recommendations	<ol style="list-style-type: none"> Administration of entire program Regular and topical reporting Future recommendations for continued efforts
8.	Multirod Geometry Changes	<ol style="list-style-type: none"> Steady-state CHF data with spacer components and simulated fuel rod bowing and swelling

1.4 ACTIVITIES COVERED BY THIS REPORT

The work covered by this second quarterly report is that in connection with Planning and Analysis, Task 1, and Design and Manufacture, Tasks 2a and 2b respectively.

2. PLANNING AND ANALYSIS (TASK 1)

2.1 PLANNING

2.1.1 Personnel

E. E. Polomik contributed essentially full time during the quarter. R. T. Lahey, R. J. Ernst, and Y. H. Kong contributed on a part-time basis.

L. E. Schnebly reported to Core Development Engineering in November and will start full-time participation in this program in the next quarter.

2.1.2 Tests

Detailing and scheduling of one- and nine-rod tests to be performed in the third and fourth quarters was completed. These tests will comprise those shown in Task 3, of Table 1, except for tests involving prolonged exposure to Critical Heat Flux (CHF), and exploratory nine-rod tests in Task 8 on equipment available from the Project Agreement 44 Two-Phase Flow and Heat Transfer Program. The tests involving the prolonged effect of CHF have not been scheduled. They will be held in abeyance until preliminary tests on the existing facility can be made to determine whether water chemistry can be adequately controlled.

2.2 ANALYSIS

2.2.1 Transient in the System Pressure

The pressure in the system will be measured with a BLH pressure transducer connected to the top of the test section by a 20 foot length of 1/4 inch diameter stainless steel tubing. The transducer and interconnecting tubing are modeled as a single degree of freedom system. Such a system is described by the equation

$$\ddot{x} + 2L\omega_n \dot{x} + \omega_n^2 x = cP(t) \quad (2-1)$$

where x is the output of the transducer and $P(t)$ the true system pressure. Values for the damping coefficient, L , and the natural frequency, ω_n , were obtained experimentally. The constant, c , was evaluated using initial conditions.

By comparing x , the transducer output, with $P(t)$, the calculated true pressure, the extent to which the dynamic characteristics of the transducer affected the measurements was determined. It was found that, due mainly to the relatively high natural frequency of the transducer system, there were negligible differences of less than 1 psi in 1000 psi between the two. The complete analysis is given in Appendix A.

2.2.2 Transient Flow by a Turbine Vane Meter

The sensing element rotor of the turbine vane meter (Pottermeter) spins freely in the measured fluid and is instantaneously sensitive to flow rate changes. Therefore, the output signal reflects the true flow rate. However, a low pass filter is implicit in the Pottermeter frequency converter and thus, the Sanborn recorder, and the time lag this introduces into the data must be factored out. The transfer function for a low pass filter is known to be

$$H(S) = \frac{\alpha}{S + \alpha} \quad (2-2)$$

where α equals the inverse of the time constant. The differential equation which produces this transfer function was then derived. It is

$$f_1(t) = \frac{1}{\alpha} \frac{df_2(t)}{dt} + f_2(t) \quad (2-3)$$

where f_1 is the actual output of the Pottermeter and f_2 is the recorded output. The time constant was determined experimentally. The derivation of this equation and a copy of the computer program written to numerically solve it are given in Appendix B. The effect of the time lag (0.25 sec. time constant) on an actual run in the one-rod test section is seen in Figure 2-1.

2.2.3 Transient Flow by an Orifice Meter

A low pass filter is installed between the differential pressure transducer across the orifice and the recorder to reduce flow "noise." Hence, the same type of correction that was performed on the Pottermeter recordings above must be performed on the recordings of the differential pressure transducer signal. The time constant for this filter was also obtained experimentally. This operation produces the true transducer output from the recorded output.

The time required for the fluid to flow the distance between the two pressure taps on the orifice can be of the same order as the transient. Therefore the inertia of the fluid must be taken into account. By considering a control volume (Figure 2-2) and applying the principle of conservation of momentum the following equation was derived:

$$\frac{dw}{dt} + \frac{1}{2\rho l A_1} \left[\left(\frac{A_1}{A_2} \right)^2 - 1 + k \right] w^2 = (P_1 - P_2) \frac{A_1}{l} g_c \quad (2-4)$$

Figure 2-1. Flow Time Lag with the Turbine Vane Meter

P_1	PRESSURE AT SECTION 1
A_1	CROSSECTIONAL AREA AT SECTION 1
V_1	VELOCITY AT SECTION 1
P_2	PRESSURE AT SECTION 2
A_2	CROSSECTIONAL AREA AT SECTION 2
V_2	VELOCITY AT SECTION 2
M	MASS WITHIN THE CONTROL VOLUME
W	MASS FLOW RATE
F_0	FORCE REQUIRED TO HOLD THE ORIFICE PLATE STATIONARY
γ	SPECIFIC WEIGHT OF THE FLUID
ρ	DENSITY OF THE FLUID
l	DISTANCE BETWEEN PRESSURE TAPS
K	LOSS COEFFICIENT OF THE ORIFICE
C	ASME FLOW COEFFICIENT

Figure 2-2. Orifice Flowmeter

With this equation, given the true transient pressure differential ($P_1 - P_2$), the true transient flow, w , through the orifice can be calculated using a numerical integration program which was formulated. The derivation of the equation and a listing of the program are given in Appendix C.

The procedure, then, to determine the flow rate through the orifice consists of two steps. First, factor the

time lag out of the recorded signal to obtain the true transducer output. Second, taking into account the inertia of the fluid through the orifice, convert the true transducer output to the true flow rate by solution of the differential equation above. Figure 2-3 is a plot of the recorded flow and the totally corrected flow, as obtained on a typical run.

Figure 2-3. Flow Time Lag with the Orifice Meter

3. DESIGN AND MANUFACTURE (TASKS 2a, 2b)

3.1 ONE-ROD OSCILLATING POWER DESIGN

One of the tests in Task 3 will be to cause the power or heat flux to cycle between values above and below the critical value, over a long period of time. This test will determine the effect on the heater surface and is shown schematically in Figure 3-1. The apparatus designed for this purpose to test single rods is shown schematically in Figure 3-2.

The single rod test section will contain a 0.562 in. o.d. rod with a 6-foot heated length and a total resistance of 0.568 ohms. Power is supplied from one phase of a 2000 kW three-phase transformer. The primary circuit of a welding transformer is connected into the main line to the test section. The transformer secondary circuit is connected, with two banks of resistors, as shown. The secondary can be shorted by means of a magnetic contactor switch which is operated by a microswitch cam arrangement, the latter run by a variac. When the transformer secondary is shorted the ampere-turns in the primary and secondary are equal, and the net effect is essentially zero voltage in the primary. This means the full voltage E_0 is available to the test section. However, when the contactor switch is open, the load in the secondary causes a net primary voltage E_{TRP} in the transformer. When this happens the net volt-

age available to the test section is $E_0 - E_{TRP}$. Thus, cycling of the contractor switch, cycles the voltage to the test section, and results in a cyclic heat flux output from the test section heater. The period of power or flux cycling is varied by the variac which controls the period of the magnetic contactor switch, through the cam and microswitch.

Preliminary calculations showed that two banks of 1 kW, 115 V resistance heaters, with a pair in series and several series pairs in parallel (Figure 3-2) resulted in the desired flexibility and range of power variation for one-rod testing.

The electrical relationship in the transformer secondary circuit of Figure 3-2 is with nomenclature as shown on the figure:

$$\left(\frac{E}{I}\right)_{TRS} = \frac{R}{N_1} + \frac{R}{N_2}, \text{ and} \quad (3-1)$$

and

$$K_{TRS} = \frac{E_{TRS}^2}{R \left(\frac{1}{N_1} + \frac{1}{N_2} \right)} \quad (3-2)$$

Figure 3-1. Heat Flux versus Quality

Figure 3-2. Oscillating Power Schematic (Single Rod)

R is the resistance of a pair of heaters in series, or

$$R = 2 \left(\frac{115^2}{1000} \right) = 26.45 \text{ ohm.}$$

The calculations for a 10% swing in power about a heat flux of 550,000 Btu/h-ft² follow. Calculated values are shown in Table 2.

$$K_{TS} = \frac{\phi A}{3413} = \frac{(55 \times 10^6)(0.88)}{3413} = 141.8 \text{ kW,}$$

$$E_O = \sqrt{(141,800)(0.568)} = 283.8 \text{ V, and}$$

$$I_O = 283.8/0.568 = 499 \text{ amp.}$$

For a 10% power reduction in the test section, the magnetic contactor switch is closed and E_O is maintained constant at 283.8 V.

$$K_{TS} = 141.8 (0.9) = 127.6 \text{ kW,}$$

$$E_{TS} = \sqrt{127,800 (0.568)} = 269.2 \text{ V,}$$

$$I_{TRP} = I_O = 269.2/0.568 = 474 \text{ amp, and}$$

$$E_{TRP} = E_O - E_{TS} = 283.8 - 269.2 = 14.56 \text{ V.}$$

Also,

$$E_{TRS} = 22 E_{TRP} = 22 (14.56) = 320.4 \text{ V,}$$

$$I_{TRS} = \frac{I_{TRS}}{22} = 474/22 = 21.55 \text{ amp,}$$

and the power dissipation required in the transformer secondary is

$$K_{TRS} = 0.320 (21.55) = 6.92 \text{ kW.}$$

The effective resistance required in the transformer secondary is by equation (3-2)

$$R \left(\frac{I}{N_1} + \frac{I}{N_2} \right) = \frac{320.4}{21.55} = 14.85 \text{ ohms,}$$

where R, the resistance of two heaters in series is 26.45 ohms. Values of $N_1 = 3$ and $N_2 = 4$ are a practical solution for equation (3-3) and result in a test section power decrease of about 10.5%.

Values calculated above are shown on the first line of Table 2 below. Parametric calculations, for test section power reductions of 3-10%, were made for test section heat fluxes from 300,000-550,000 Btu/h-ft². These values for 6 and 10% power reductions are shown in Table 2. The number of heaters in the transformer secondary circuit are plotted on Figure 3-3 against the effective secondary resistance and test section power reduction for a 0.568 ohm heater in the test section.

Combinations of heaters required for typical test section power reductions were obtained from Figure 3-3. These are shown below and circled on Figure 3-3.

Nominal Test Section Power Reduction (%)	Series Pairs of Heaters in Parallel		Estimated Actual Power Reduction (%)
	N_1	N_2	
10	3	4	10.5
8	4	5	8.2
6	5	8	6.0
4	9	10	4.0

3.2 CYCLING FLOW DESIGN

Paragraph 3.1 above described a design for cycling the power so the heat flux can go below or beyond the mean actual steady-state CHF. Another method of obtaining cyclic CHF conditions is to cycle the flow. A graphical display of the latter process is shown on Figure 3-4, for a

0.563-inch o.d. test heater with a 6-foot heated length, in a 1.25-inch i.d. vessel, at 1000 psi.

The enthalpy balance over the heated length is

$$\phi PL = GA (\chi h_{fg} + \Delta h_s) \quad (3-4)$$

Table 2

Percent Decrease in Power When Secondary Switch Open	When Magnetic Contactor Switch in Secondary (Fig. 3-1) is Closed				When Magnetic Contactor Switch in Secondary (Fig. 3-1) is Opened										Pairs of 1kW Resistors in Parallel (Fig.3-1)	
	ϕ Btu/h-ft ²	EO (Volt)	ETS (Volt)	KTS (kW)	EO (Volt)	ETS (Volt)	KTS (kW)	ETRP (Volt)	KTRS (kW)	ETRS (Volt)	IO (Amp)	ITRS (Amp)	E ₁	E ₂	N ₁	N ₂
10	550,000	283.8	283.8	141.3	283.8	269.2	127.6	14.56	6.90	320.4	474	21.55	190.0	142.5	3	4
10	500,000	270.6	270.6	128.9	270.6	256.7	116.0	13.89	6.28	305.5	452	20.54	181.1	135.8	3	4
10	400,000	242.0	242.0	103.1	242.0	229.6	92.8	12.42	5.02	273.2	404	18.38	162.1	121.5	3	4
10	300,000	209.6	209.6	77.4	209.6	198.6	69.6	10.76	3.77	236.6	350	15.91	140.3	105.2	3	4
6	550,000	283.8	283.8	141.8	283.8	275.2	133.3	8.646	4.189	190.2	484	22.02	194.1	145.6	5	8
6	500,000	270.6	270.6	128.9	270.6	262.4	131.2	8.244	3.503	181.4	462	21.00	185.2	138.9	5	8
6	400,000	242.0	242.0	103.1	242.0	234.7	96.9	7.373	3.046	162.2	413	18.78	165.6	124.2	5	8
6	300,000	209.6	209.6	77.4	209.6	203.2	72.7	6.386	2.285	140.5	358	16.26	143.4	107.5	5	8

Basis: 0.563 in. o.d. heater, 6 ft heated length, 0.568 ohm

Figure 3-3. Transformer Secondary Load versus Test Section Power Reduction

Figure 3-4. Graphical Representation of Flow Cycling

with Btu, pound mass, hour, and foot units.

ϕ = Heat Flux
 P = Heater Perimeter
 L = Heated Length
 G = Mass Flux
 A = Flow Area
 χ = vapor weight fraction
 h_{fg} = Latent Heat
 Δh_s = Subcooling at entrance.

Equation (3-4) evaluation for the geometry under consideration, at a system pressure of 1000 psia, is

$$\phi = 0.007681 G(h_{fg} \chi + \Delta h_s), \quad (3-5)$$

or for 1000 psia and 20 Btu/lb subcooling

$$\frac{\phi}{10^6} = 4.987 \left(\frac{G}{10^6} \right) \chi + 0.1536 \left(\frac{G}{10^6} \right). \quad (3-6)$$

Operating lines for mass fluxes from $0.2 \cdot 1.0 \times 10^6$ lb/h-ft² are shown on Figure 3-4. Cyclic CHF conditions would be expected by maintaining heat flux at 300,000 Btu/h-ft² and cycling the flow between 0.25 and 0.20×10^6 lb/h-ft² positions 2-3; or between 0.3 and 0.2×10^6 mass flux, positions 1-3. Cycling CHF conditions at 750,000 heat flux would be expected by cycling flow between 1.0 and 0.75×10^6 mass flux, positions 4-5.

Cycling flow was designed into the control system of the 2 inch flow control which is shown in the lower right section of Figure 3-5. The valve is an Annin model 4560 with a fast response Domotor that positions the valve plug.

The control system for the 2-inch Flow Control Valve is shown schematically on Figure 3-6. When the solenoid actuating power is normally connected, position 1 is used for manual flow control. Two separate control systems are actuated by switch positions 2 and 3. When two different flows are set on positions 2 and 3, a fast flow transient can be obtained by switching from 2 to 3 or vice versa.

Cycling flow is obtained by rerouting the solenoid valve actuating power to a cam driven microswitch. The microswitch alternately energizes the two positions 2 and 3, thus varying the flow cyclicly between these preset flows. A variac in the circuit of the motor which drives the cam provides means for varying the period of flow cycling over a wide range.

3.3 FLOW SHUTOFF

An occurrence which may lead to deficient reactor core cooling is an intermittent loss of core flow as approx-

imated in Figure 3-7. The period T , during which coolant flow may be below normal, is estimated to be about 1 to 5 seconds. The time t during flow decay and rise is small compared to the shutoff time T , with t being on the order of 100 milliseconds compared to several seconds for T . A one-rod test is planned to investigate temperature excursions during such a loss of flow transient.

Accomplishment of a flow shutoff (Figure 3-7) can be realized with the basic flow control arrangement shown in the upper part of Figure 3-6, but with the cam-microswitch equipment replaced by a timer switch and drive as shown in Figure 3-8. The flow shutoff transient is then obtained by setting the desired flow on position 2, setting the air pressure on position 3 for full closed, and setting the timer switch for the desired time of flow shutoff. When the transient is desired, the start switch is closed. This deenergizes the number 2 solenoid; energizes the number 3 solenoid, and admits full control pressure to close the valve for the desired preset time.

The time of flow shutoff, t on Figure 3-7, was very small compared to several seconds of shutoff time T . Estimates were made for the speed of operation of the system and the results are tabulated below.

3.4 MANUFACTURE AND PROCUREMENT

Twelve calrods, 9/16 inch in diameter with an effective heated length of 6 feet and uniform heat flux, had been ordered from the General Electric Industrial Heater Division at Shelbyville, Indiana in the first quarter. An additional order for 18 of these rods was placed with the Watlow Electrical Manufacturing Company in anticipation of heater needs for the heavy nine-rod test schedule ahead. A detailed outline of these calrods is shown in Figure 3-9. These rods will all have stainless steel sheaths. The heater elements, in the 6-foot portion of the General Electric heaters, will be made of Nichrome wire insulated from the outer sheath by Boron Nitride. The Watlow heaters, not yet designed, will be of similar construction internally and have the same external dimensions.

The thermocouple arrangement for nine-rod CHF testing is shown in Figures 3-10 and 3-11. Two or three 0.020-inch o.d. stainless sheathed, chromel-alumel thermocouples are brazed into grooves in the exterior sheath of the rod (Figure 3-10). Stainless steel wire is used as a filler, along the length of the groove and in the runout portion, to minimize the distortion of surface heat flux. The sensing tips of the couples extend 3/8 inch into the heated length. This elevation has been found to be the mean position for CHF occurrence with uniform heat flux.

Figure 3-5. Schematic of One-Rod Test Arrangement

Figure 3-6. Schematic of Two-Inch Flow Control Valve Operation

Figure 3-7. Flow Shutoff

Table 3

$G (10^6)$ $\frac{\text{lb}}{\text{hr-ft}^2}$	w gpm	C_v Valve Coefficient	% Flow	Time for Valve Shutoff sec	Time for Timer sec	Total Time t , sec
0.30	4.06	1.61	4.6	0.042	0.03	0.072
0.40	5.4	2.14	6.1	0.055	0.03	0.085
0.60	8.1	3.21	9.1	0.082	0.03	0.112
0.80	10.8	4.27	12.2	0.109	0.03	0.139
1.00	13.5	5.34	15.2	0.136	0.03	0.196

Figure 3-8. Schematic of Flow Shutoff Control System

NOTES:

1. ELECTRO ETCH, IN AREA SHOWN, IDENTIFICATION MARK WITH 1/8 in. HIGH CHARACTERS.
2. GROUND IS AT TOP OF HEATER (TAPERED END).
3. INDICATED AREA FOR INSULATION OF SHEATH FROM ELECTRODE.

Figure 3-9. Calrod Type Heater for Use in Nine-Rod Test Section

Figure 3-10. Thermocouple Detail for Uniform Flux Calrod

Figure 3-11. Thermocouple Locations for CHF

Radial location of thermocouples in the calrods is shown in Figure 3-11. CHF usually occurs in the corner sections where an extra thermocouple has been placed for added protection. Thermocouples are arranged so CHF coverage is provided, between all rods and at two rod-sidewall locations, with a minimum number. The arrangement results in three types of instrumented rods—A, B, and C.

In addition to the 30 uniform flux calrods ordered above, an order was also placed with General Electric Industrial Heater Division for 12 calrod heaters with a non-uniform axial heat flux. Heat flux will increase from a low value at the bottom (flow inlet) end and increase to a maximum at the top (flow exit) end. The increase of flux along the length will vary approximately as a truncated sin curve.

CHF is expected to occur near the exit flow end where heat flux is a maximum. This design will permit the number of thermocouples, required to detect CHF, to be held to about 4-6 per rod which is a practical value for a nine-rod assembly. Details of the heat flux profile and thermocouple arrangement will follow in the next quarterly report.

Material required for power and flow oscillation, described in paragraphs 3.1 and 3.2 above, was ordered. This included the magnetic contactor switch, two banks of resistance heaters, microswitches, and cam.

Tests on the 2-inch flow control valve indicated the seat trim of 2 inches may be too large and that an equal percentage plug would give better transient performance than the linear plug available. An equal percentage plug was obtained for future testing.

4. PROJECT ADMINISTRATION AND REPORTING

A technical review was held November 6, 1969 with Mr. R. Scroggins at AEC headquarters in Washington, D.C. Organization of the program tasks, available facilities, and accomplishments were summarized. It was recommended that the program be modified to include 16-rod test bundles with heated lengths up to 12 feet, which would provide a better simulation of actual reactor fuel bundles. These tests could be run in the new 8000 kW Test Facility (ATLAS) which is presently under construction and

expected to be in operation the latter part of 1970. This new facility would be much more desirable as it is designed for transient testing and has the necessary control, instrumentation, and data acquisition systems.

A trip was made to ORNL at Oakridge, Tennessee to obtain information on calrod heater operation and manufacture, and discuss areas of their testing that was related to that in this program.

APPENDIX A

ANALYSIS OF THE SYSTEM PRESSURE TRANSDUCER TRANSIENT

The pressure in the test section will be measured with a BLH type GP general purpose pressure transducer. The transducer is connected near the top of the test section by 20 feet of 1/4-inch diameter stainless steel tubing. This analysis was performed to determine if the dynamic characteristics of the transducer and connecting tubing affected the recorded output of the transducer during a transient test.

Consider the transducer and tubing as shown in Figure A-1. Modeling a pressure transducer as a single degree of freedom spring-mass damper system is common practice.

First Newton's Second Law is applied to the inter-connecting tubing and fluid.

pressure forces—shear forces = inertia of the fluid

$$(P_{TS} - P_S) A_t - \frac{f \rho V_t^2}{8} \pi D L_t = \rho A_t L_t \frac{dV_t}{dt} \quad (A-1)$$

Based on the assumption of laminar flow in the tubing,

$$f = \frac{64\mu}{D \rho V_t} \quad (A-2)$$

Substitution of equation (A-2) into equation (A-1) and simplifying yields:

$$(P_{TS} - P_S) A_t - 8\pi \mu L_t V_t = \rho A_t L_t \frac{dV_t}{dt} \quad (A-3)$$

P_O	ATMOSPHERIC PRESSURE
P_S	PRESSURE IN THE TRANSDUCER
P_{TS}	PRESSURE IN THE TEST SECTION
A_t	CROSSECTIONAL AREA OF THE TUBING
D	DIAMETER OF THE TUBING
L_t	LENGTH OF THE TUBING
ρ	DENSITY OF FLUID
V_t	VELOCITY OF FLUID IN THE TUBE
X	POSITION OF PISTON IN TRANSDUCER MODEL (= THE RECORDED OUTPUT)
K	SPRING CONSTANT
B	DAMPING COEFFICIENT
A_S	AREA OF PISTON IN TRANSDUCER
f	FRICTION FACTOR

Figure A-1. System Pressure Transducer Schematic

To relate this equation to the pressure transducer, consider the equation of continuity between the tubing and the transducer.

$$\rho V_t A_t = \rho A_s \frac{dx}{dt}$$

hence

$$V_t = \frac{A_s}{A_t} \frac{dx}{dt} \quad (A-4)$$

and

$$dV_t = \frac{A_s}{A_t} \frac{d^2x}{dt^2} \quad (A-5)$$

Substituting equation (A-4) and (A-5) into equation (A-3) yields an equation which relates the dynamics of the fluid in the tube to motion of the piston in the transducer model.

$$(P_{TS} - P_S) A_t - 8\pi\mu L_t \frac{A_s}{A_t} \frac{dx}{dt} = \rho L_t A_s \frac{d^2x}{dt^2} \quad (A-6)$$

The next step is to eliminate P_S by expressing it in terms of x . To do this Newton's Second Law is applied to the transducer model. The result is

$$m \frac{d^2x}{dt^2} = -kx - B \frac{dx}{dt} + (P_S - P_O) A_s \quad (A-7)$$

If all pressures are taken to be gage pressures, then $P_O = 0$ and, upon rearrangement, equation (A-7) becomes

$$P_S A_s = m \frac{d^2x}{dt^2} + B \frac{dx}{dt} + kx \quad (A-8)$$

This is the needed relationship between P_S and x . Substituting this into equation (A-6), regrouping terms, and dividing through by the coefficient of d^2x/dt^2 we obtain

$$\frac{d^2x}{dt^2} + \left[\frac{8\pi\mu L_t \frac{A_s}{A_t} + \frac{A_t}{A_s} B}{\rho L_t A_s + m \frac{A_t}{A_s}} \right] \frac{dx}{dt}$$

$$\left[\frac{k \frac{A_t}{A_s}}{\rho L_t A_s + m \frac{A_t}{A_s}} \right] x = P_{TS} \left[\frac{A_t}{\rho L_t A_s + m \frac{A_t}{A_s}} \right] \quad (A-9)$$

Close observation indicates this to be identical in form to the equation for a second order single degree of freedom system. The standard form is

$$\frac{d^2x}{dt^2} + 2L\omega_n \frac{dx}{dt} + \omega_n^2 x = CF(t) \quad (A-10)$$

where

L = damping coefficient

ω_n = undamped natural frequency

C = constant

$F(t)$ = driving force (P_{TS})

This close correlation between equations (A-9) and (A-10) shows that the combination of transducer and connecting tubing is indeed a simple second order single degree of freedom system and that it can be described by equation (A-10). Because some of the parameters in equation (A-9) are unknown and difficult to determine, it was decided to use the form of equation (A-10) to describe the system.

The damping coefficient, L , and the natural frequency, ω_n , were obtained experimentally. The stainless steel tubing was disconnected from the test section and a step pressure pulse, from an initial pressure of 400 psi to a final of 750 psi, was imposed on the end of the tubing. An oscilloscope was connected directly to the transducer to record its response. From a trace of the response ω_n was readily determined as 99 radians/sec. Figure 1-13 in "Basic Feedback Control System Design" by C. J. Savart, Jr. was used to determine that L was 0.14.

The constant C was evaluated using steady state conditions. The Sanborn recorder was calibrated such that 0-50 divisions is equivalent to 600-1000 psi. To keep C a constant, it was necessary to let 0-50 divisions be equivalent to 0-400 psi and use the conditions

$$\frac{d^2x}{dt^2} = \frac{dx}{dt} = 0$$

$$x = 50$$

$$p = 400 \text{ psi}$$

After determining that C equals 1225, equation (A-10) was rewritten in the following form.

$$\frac{1}{C} \frac{d^2x}{dt^2} + \left(\frac{2L\omega_n}{C} \right) \frac{dx}{dt} + \frac{\omega_n^2}{C} x + 600 = P_{TS}$$

or

$$0.000816 \frac{d^2 x}{dt^2} + 0.0226 \frac{dx}{dt} + 8x + 600 = P_{TS} \quad (A-11)$$

This equation relates the pressure in the test section, through the dynamic characteristics of the transducer-tubing system, to the signal recorded on the Sanborn.

To determine what effect these dynamic characteristics had on the recorded signal, a pressure trace taken during a depressurization transient was fitted with a curve. This resulted in an equation for the pressure trace:

$$x = 46.614 - 23.214t + 12.9705t^2 - 3.305t^3 + .295t^4 \quad (A-12)$$

By differentiating equation (A-12) and substituting into equation (A-11), the corrected system pressure was obtained for any point in time.

This procedure was carried out for several cases and the results showed that, due mainly to the relatively high natural frequency of the transducer system, there were only negligible discrepancies of less than 1 psi in 1000 psi between the recorded transient pressure and the corrected true pressure at every point in time.

APPENDIX B

ANALYSIS OF THE TIME LAG INTRODUCED BY A LOW PASS FILTER (for Transient Flow by Turbine Vane Flow Meter)

The transfer function for a low pass filter is known to be

$$\frac{F_2(s)}{F_1(s)} = H(s) = \frac{\alpha}{s+\alpha} \quad (B-1)$$

where

$$F(s) = L\{f_1(t)\}$$

$$H(s) = L\{\alpha e^{-\alpha t}\}$$

Taking the convolution of $F_1(s)$ and $H(s)$, and noting that

$$F_2(s) = F_1(s) H(s)$$

results in the equation

$$F_2(s) = L\left\{\int_0^t f_1(\tau) \alpha e^{-\alpha(t-\tau)} d\tau\right\} \quad (B-2)$$

The inverse Laplace transform of equation (B-2) is:

$$f_2(t) = \int_0^t f_1(\tau) \alpha e^{-\alpha(t-\tau)} d\tau \quad (B-3)$$

Let

$$G(t,\tau) = f_1(\tau) \alpha e^{-\alpha(t-\tau)} \quad (B-4)$$

and therefore

$$f_2(t) = \int_0^t G(t,\tau) d\tau \quad (B-5)$$

Now differentiating equation (B-5) with respect to t yields (noting that the limits of integration are not constant);

$$\frac{df_2}{dt} = G(t,t) \frac{dt}{dt} - G(t,0) \frac{d0}{dt} + \int_0^t \frac{dG(t,\tau)}{dt} d\tau \quad (B-6)$$

Substituting equation (B-4) into equation (B-6),

$$\frac{df_2}{dt} = \alpha f_1(t) - 0 - \alpha \int_0^t f_1(\tau) \alpha e^{-\alpha(t-\tau)} d\tau$$

Noting that the last term on the right is $\alpha f_2(t)$, and rearranging gives,

$$f_1(t) = \frac{1}{\alpha} \frac{df_2(t)}{dt} + f_2(t) \quad (B-7)$$

With this equation the input $[f_1(t)]$ to a low pass filter can be determined from the output $[f_2(t)]$ once the time constant $(1/\alpha)$ of the filter is known.

J. E. Wood of General Electric has developed a Fortran - IV computer program to solve this equation. A listing of the program follows.


```

$ IDENT L325,RUE-PW300,EXPRESS
$ OPTION FORTRAN
$ FORTRAN
$ INCODE IBMF
CGLAG G L A G -- EXECUTIVE ROUTINE
C FLOW VERSUS TIME SHIFT GENERATES PCI FOR SCAT
COMMON /INPUT/ L0,ALPH(10),N,IGX(100),GTG(100),ID(10),CX(6)
COMMON /OTHER/ T(1000),G(1000),X(102),P(102),A(101),B(101),C(101),
1 XX(102),TMV(20),GTAB(20),TG(100),GT(100),CG(1000),GGG(1000)
DIMENSION IA(3,10)
DIMENSION SL(1000)
DATA(IA(1,01),I=1,3)/6HLO , -1, 0/
DATA(IA(1,02),I=1,3)/6HALPHA , 2, 0/
DATA(IA(1,03),I=1,3)/6HN , -12, 0/
DATA(IA(1,04),I=1,3)/6HTGX , 13, 0/
DATA(IA(1,05),I=1,3)/6HGTG , 113, 0/
DATA(IA(1,06),I=1,3)/6HID , -213, 0/
DATA(IA(1,07),I=1,3)/6HCX , 223, 0/
DATA MYPUN/6HITAP&T/
IPUN=IDFTAP(MYPUN,3,2)
IOUT=IDFTAP(MYPUN,2,2)
CX(1)=1.
CX(2)=0.
CX(3)=1.
CX(4)=1.
CX(5)=0.
CX(6)=1.
51 CONTINUE
CALL CASE(IARG)
IARG=0
CALL DTA(IA,10,3,1,80,IARG,L0)
IF(IARG.NE.0) CALL CASERR
DO 351 I = 1, N
TG(I)=CX(1)*(CX(2)+CX(3)*TGX(I))
GT(I)=CX(4)*(CX(5)+CX(6)*GTG(I))
351 CONTINUE
A(1)=TG(1)-(TG(2)-TG(1))
XX(1)=X(1)**2
P(1)=GT(1)
DO 21 I = 1, N

```

```

X(I+1)=TG(I)
XX(I+1)=X(I+1)**2
P(I+1)=GT(I)
21 CONTINUE
X(N+2)=TG(N)+(TG(N)-TG(N-1))
P(N+2)=GT(N)+(GT(N)-GT(N-1))
N1=N+1
DO 22 I = 2, N1
D=DTRM(XX(I-1),X(I-1),1.,XX(I),X(I),1.,XX(I+1),X(I+1),1.)
A(I)=DTRM(P(I-1),X(I-1),1.,P(I),X(I),1.,P(I+1),X(I+1),1.)/D
B(I)=DTRM(XX(I-1),P(I-1),1.,XX(I),P(I),1.,XX(I+1),P(I+1),1.)/D
C(I)=DTRM(XX(I-1),X(I-1),P(I-1),XX(I),X(I),P(I),XX(I+1),X(I+1),
1 P(I+1))/D
22 CONTINUE
K=1
T(1)=TG(1)
G(1)=GT(1)
GGG(1)=G(1)
DO 4 I = 2, N
DO 3 J = 1, 10
K=K+1
AJ=J
T(K)=(AJ/10.)*(X(I+1)-X(I))+X(I)
G(K)=((A(I)+A(I+1))*T(K)**2+(B(I)+B(I+1))*T(K)+(C(I)+C(I+1)))/2.
GGG(K)=G(K)
3 CONTINUE
4 CONTINUE
M=K-1
GG(1)=GT(1)
DO 701 IT = 1, L0
CA=1./ALPH(IT)
CB=1.
DO 31 I = 2, M
DF2=(G(I+1)-G(I-1))/(T(I+1)-T(I-1))
GG(I)=CA*DF2+CB*G(I)
31 CONTINUE
DF2=(G(K)-G(K-1))/(T(K)-T(K-1))
GG(K)=CA*DF2+CB*G(K)
DO 700 I = 1, K
G(I)=GG(I)
700 CONTINUE
701 CONTINUE

```

```

      TL=TG(N)-TG(1)
      GLA=1.0E10
      GLB=-1.0E10
      DO 501 I = 1, K
      IF (GG(I).LT.GLA) GLA=GG(I)
      IF (GG(I).GT.GLB) GLB=GG(I)
501 CONTINUE
      GL=GLB-GLA
      SL(1)=0.
      DO 502 I = 2, K
      SL(I)=SL(I-1)+SQRT(((T(I)-T(I-1))/TL)**2+((GG(I)-GG(I-1))/GL)**2)
502 CONTINUE
      TMV(1)=TG(1)
      GTAB(1)=GG(1)
      L=2
      DL=SL(K)/19.
      SLX=DL
      DO 503 I = 2, M
      IF (SL(I).LT.SLX) GO TO 503
      TMV(L)=T(I)
      GTAB(L)=GG(I)
      L=L+1
      SLX=SLX+DL
503 CONTINUE
      J=20
      TMV(J)=TG(N)
      GTAB(J)=GG(K)
      WRITE(IPUN,100) (ID(I),I=1,10)
100 FORMAT(4H ID ,10A6 )
      WRITE(IPUN,102)
102 FORMAT(7H NMV 20)
      WRITE(IPUN,103) ( TMV(I),I=1, J)
103 FORMAT(5H TMV ,5F10.4/5X,5F10.4/5X,5F10.4/5X,5F10.4)
      WRITE(IPUN,101) (GTAB(I),I=1, J)
101 FORMAT(5H GTAB,5F10.4/5X,5F10.4/5X,5F10.4/5X,5F10.4)
104 FORMAT(/////////)
      WRITE(10UT,104)
      WRITE(10UT,100) (ID(I),I=1,10)
      WRITE(10UT,102)

```

WRITE(100T,103) (1MV(1),1=1, J)

WRITE(100T,101) (GTAB(1),1=1, J)

GO TO 51

END

\$ FORTRAN

\$ INCODE IBMF

CDTRM D T R M -- FUNCTION SUBPROGRAM OF GLAG

C

FUNCTION DTRM(A1,B1,C1,A2,B2,C2,A3,B3,C3)

DTRM=A1*B2*C3+A2*B3*C1+A3*B1*C2-A3*B2*C1-A2*B1*C3-A1*B3*C2

RETURN

END

\$ EXECUTE

\$ LIMITS 5,24000,1000

\$ INCODE IBMF

)

 L@ 1

 ALPHA 35.714

)_AST

\$ ENDJOB

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APPENDIX C

TRANSIENT ANALYSIS FOR THE ORIFICE FLOW METER

The sharp edged orifice will be used to measure the flow rate during the depressurization transient tests. The time required for the fluid to flow the distance between the two pressure taps on the orifice can be of the same order as the transient. Therefore, it was necessary to be able to describe the dynamics of the flow through the orifice in order to determine the true flow rate from the indicated flow rate.

The principle of conservation of momentum was applied to the control volume indicated by the dotted lines in Figure 2-2. The principle of conservation of momentum states:

$$\frac{1}{g_c} \frac{d(mV)}{dt} = \sum F + \sum \frac{1}{g_c} \omega V$$

Application of this to the control volume yields

$$\frac{1}{g_c} \frac{d(mV)}{dt} = (P_1 - P_2) A_1 - F_O + \frac{1}{g_c} w (V_1 - V_2) \quad (C-1)$$

F_O is the force required to hold the orifice plate stationary, or to view it differently, it is the pressure and flow losses of the orifice. To evaluate F_O , consider the steady-state case of equation (C-1).

$$F_O = (P_1 - P_2) A_1 + \frac{1}{g_c} w (V_1 - V_2) \quad (C-2)$$

It is desired to express F_O in terms of velocity only, hence a relation between the pressure drop and the flow is required. By applying Bernoulli's equation at stations 1 and 2 such a relation can be obtained.

$$\frac{P_1}{\gamma} + \frac{V_1^2}{2g} = \frac{P_2}{\gamma} + \frac{V_2^2}{2g} + K \frac{V_1^2}{2g}$$

Rearranging results with,

$$P_1 - P_2 = \frac{\gamma}{2g} (V_2^2 - V_1^2 + K V_1^2) \quad (C-3)$$

where K is a loss coefficient for the orifice.

Substitution of equation (C-3) into equation (C-2) yields the derived results.

$$F_O = \frac{\gamma}{2g} (V_2^2 - V_1^2 + K V_1^2) A_1 + \frac{1}{g_c} w (V_1 - V_2) \quad (C-4)$$

This relationship is now put into equation (C-1) to give:

$$\frac{1}{g_c} \frac{d(mv)}{dt} = (P_1 - P_2) A_1 - \frac{\gamma}{2g} (V_2^2 - V_1^2 + K V_1^2) A_1 \quad (C-5)$$

The terms on the left can be written as

$$\frac{d(mv)}{dt} = m \frac{dv}{dt} + v \frac{dm}{dt}$$

and by the assumption of an incompressible fluid $\frac{dm}{dt} = 0$. Therefore, equation (C-5) becomes,

$$\frac{m}{g_c} \frac{dv}{dt} = (P_1 - P_2) A_1 - \frac{\gamma}{2g} (V_2^2 - V_1^2 + K V_1^2) A_1 \quad (C-6)$$

From continuity it is known that,

$$V_2^2 = V_1^2 \left(\frac{A_1}{A_2} \right)^2$$

and it is also known that,

$$m = \rho A_1 \ell$$

$$\frac{\rho}{g_c} = \frac{\gamma}{g}$$

$$w = \rho A V$$

Substitution of all of these relations into equation (C-6) gives the final result.

$$\frac{dw}{dt} + \frac{1}{2\rho\ell A_1} \left[\left(\frac{A_1}{A_2} \right)^2 - 1 + K \right] w^2 = (P_1 - P_2) \frac{A_1}{\ell} g_c \quad (C-7)$$

In actual usage the equation was written in the form

$$\frac{dw}{dt} + \frac{w^2}{2\rho\ell A_T} B = \Delta P \frac{A_1 g_c}{\ell} \quad (C-8)$$

where

$$B = \left(\frac{A_1}{A_2} \right)^2 - 1 + K$$

To determine a numerical value for B , consider a steady-state case of equation (C-8) and make the substitution $w = \rho A_T V_T$. The subscript T refers to the throat of the orifice.

$$V_T^2 = \left[\frac{1}{B} \left(\frac{A_1}{A_T} \right)^2 \right] \frac{2g_c}{\rho} \Delta P \quad (C-9)$$

The equation normally used for steady-state conditions is

$$V_T^2 = C^2 \frac{2g_c}{\rho} \Delta P \quad (C-10)$$

where C is the ASME flow coefficient ($C = 0.604$).

By comparison of equations (C-9) and (C-10) it becomes readily apparent that

$$B = \frac{1}{C^2} \left(\frac{A_1}{A_T} \right)^2 \quad (C-11)$$

Determining the value of B from this relationship insures that the equation used for reduction of the transient data, equation (C-8), is exactly reducible to the equation used for previous steady-state data reduction.

A Fortran - IV computer program was written using the APED library subroutine GLANCE to numerically integrate equation (C-8). A copy of the program follows.

```

$ IDENT E320,RJE-PW300,NONPRO
$ OPTION FORTRAN
$ FORTRAN
$ INCODE IBMF
CFLOW ORIFICE - DYNAMIC CORRECTION
C UNITS FOR INPUT PARAMETERS
C DENSITY - LBM/FT3 FLOW1 - LBM/SEC
C AREA1 - IN2 STEP - SEC
C PDIST - IN TMV - SEC
C GTAB - LBF/IN2
  DIMENSION D(10), Y(10), R(10), IA(3,10)
  COMMON /INPUT/ DENSITY, AREA1, PDIST, STEP, GTAB(100), TMV(100), FLOW1,
  CNMV, ID(10)
  DATA(IA(1,1), I=1,3)/6HDENSITY, 1, 0/
  DATA(IA(1,2), I=1,3)/6HAREA1, 2, 0/
  DATA(IA(1,3), I=1,3)/6HPDIST, 3, 0/
  DATA(IA(1,4), I=1,3)/6HSTEP, 4, 0/
  DATA(IA(1,5), I=1,3)/6HGTAB, 5, 0/
  DATA(IA(1,6), I=1,3)/6HTMV, 105, 0/
  DATA(IA(1,7), I=1,3)/6HFLOW1, 205, 0/
  DATA(IA(1,8), I=1,3)/6HNMV, -206, 0/
  DATA(IA(1,9), I=1,3)/6HID, 207, 0/
  DATA MOUT/6HSYSOUT/
  DATA MINT/6HSYSIN /
  IPUN=IDFTAP(MOUT,3,2)
  IOUT=IDFTAP(MOUT,2,2)
  IIN =IDFTAP(MINT,1,2)
15 CALL CASE(IARG)
  L1=-1
  L2=-2
  IBIP=0
  CALL DTA(IA,10,3,1,80,IBIP,DENSITY)
  IF (IBIP.EQ.L1) CALL CASERR
  B3=32.2*12.*AREA1/PDIST
  B2=(2.*AREA1*PDIST*DENSITY)/1728.
  B=193.233
  Y(1)=FLOW1
  T=TMV(1)
  TA=T*AA
  K=C
  GSTEP=STEP
  AA=.01
  M=2

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```

SERR=.0002
ERROR=.002
MODE=1
WRITE(10UT,43) (ID(I),I=1,10)
WRITE(10UT,40)
WRITE(10UT,44)
WRITE(10UT,41) T,Y(1)
10 CONTINUE
PRINT=0.
21 IF(TA.LT.TMV(M)) GO TO 20
M=M+1
GO TO 21
20 CONTINUE
PD=GTAB(M-1)+(TA-TMV(M-1))*((GTAB(M)-GTAB(M-1))/(TMV(M)-TMV(M-1)))
D(1)=PD*B3*AA-(Y(1)*Y(1))*AA*B/B2
CALL GLANCE(MODE,T,GSTEP,D,Y,R)
IF(MODE.EQ.0) PRINT=1.
IF(MODE.EQ.L1) PRINT=1.
IF (ABS(.11*(Y(1)-R(1))).GT.ERROR) PRINT=0.
IF(MODE.EQ.L2) PRINT=1.
TA=T*AA
IF(PRINT.EQ.1.) GO TO 30
GO TO 31
30 IF(K.EQ.800) GO TO 32
K=K+1
GO TO 31
32 WRITE(10UT,41) TA,Y(1)
K=0
31 CONTINUE
IF (ABS(.11*(Y(1)-R(1))).GT.ERROR) MODE=-1
IF (ABS(.11*(Y(1)-R(1))).LT.SERR) MODE=-2
IF (TA.GE.TMV(NMV)) GO TO 15
GO TO 10
40 FORMAT(29X,4HTIME,11X,4HFLOW)
41 FORMAT(25X,F10.4,5X,F10.4)
43 FORMAT(////,28X,10A6,/)
44 FORMAT(27X,8H(SECOND),7X,9H(LBM/SEC),/)
END

```


\$ LIMITS 3.25000..700

\$ INCODE IBMF

)

DENSTY 46.2963

AREA1 6.6052

PDIST 2.

STEP .1

FLOW1 2.05

ID 60H RUN NUMBER 62

NMV 20

TMV	0.	0.1100	0.3200	0.8200	1.3000
	1.5400	1.8800	2.0500	2.1500	2.3000
	2.4500	2.8200	3.2300	3.4700	3.9000
	4.3000	4.7000	5.1000	5.5500	6.0000
GTAB	0.8990	1.0537	0.8771	0.6170	0.8953
	1.3962	1.7473	2.4924	2.8118	3.2980
	3.7927	3.7929	3.2558	2.8242	2.2869
	1.7849	1.3429	0.9299	0.4968	0.0724

\$

LAST

\$ ENDJOB

*

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