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M E M O R A N D U M

March 1, 1965

TO: E. C. NELSON

FROM: W. S. DURANT WSD
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[redacted]

HYDRAULIC AND HEAT TRANSFER TRANSIENTS DURING POWER RAMP - MARK VII-AL TESTS AT COLUMBIA UNIVERSITY

INTRODUCTION:

Transient tests were run at Columbia University on a Mark VII-AL mockup to reproduce the hydraulics and heat transfer in a single fuel assembly during a power ramp resulting from the accidental withdrawal of control rods from a reactor. These tests are one of a series to supply information to aid in establishing limits for reactor operation.

This memorandum presents: a) the background for the purpose of the tests, b) a description of the test equipment and procedure, c) the results, d) a discussion of inconsistencies in experimental measurements and the corrections applied to the data, and e) limitations on the application to SRP fuel assemblies. Calculation details are shown in the Appendix.

SUMMARY:

The results of two transient tests that mocked-up a power ramp resulting from an accidental withdrawal of control rods from a reactor where the reactor is not scrammed showed that:

1. The time required for the flow to decay to zero is significantly longer than for a pump shaft break incident.

[redacted]

2. The critical effluent temperature as determined from steady-state tests[4] may be exceeded slightly without damage to the fuel or reactor because of a relatively slow flow decay and an increase in lower end fitting pressure.

3. Where a great amount of nucleate boiling may be present as with the Mark VII-AL, the transducer signal for the end fitting orifice pressure drop may increase enough to actuate a Scram I before a complete flow instability occurs.

These data are not subject to rigorous analysis because of significant errors in flow and effluent temperature measurements, and because of the limited number of tests. The qualitative observations and limited calculations were made from data corrected to their probable values from results of previous tests and from energy balance calculations as shown in the Appendix. The test section was accidently destroyed during the third test, which prevented determination of the cause of the errors.

DISCUSSION:

Background

Technical Limits on effluent temperatures from fuel assemblies are specified to prevent bulk boiling during normal operation, and Minimum Margins from these Limits are specified to restrict boiling in the event of credible incidents[1]. For incidents where transient hydraulic and heat transfer data are available, some boiling may be allowed in a limited number of assemblies or subchannels during an incident to prevent undue restrictions on reactor power provided fuel melting or damaging overpressures in the reactor tank do not occur[2,3]. For incidents where data are limited or not available, Minimum Margins are specified on the basis that the minimum saturation temperature at the exit of the hottest subchannel will not be exceeded.

Steady-state tests do not supply adequate information to determine conditions at which reactor or assembly damage actually occurs, or the behavior of a fuel assembly during an incident. These transient tests were designed to obtain part of this information for a control rod withdrawal incident. The specific objectives and the relationship of these tests to the overall program are discussed in detail in DPST-63-362[5].

Description of Equipment and Test Procedure

The test section was a modified Mark VII-AL assembly mockup as shown in Figure 1. The mockup consisted of one channel of a quatrefoil with a stainless steel resistance heater of the same diameter as a Mark VII-AL fuel slug. Heat generation was uniform and the total heated length was 11.7 feet. The length was less than that for an actual assembly, but was a compromise to obtain similar heat generation to a 14-foot modified cosine.

Three quarters of the symmetrical end fitting was blocked-off such that the hydraulics in the single tube of the quatrefoil were representative of a full-size assembly. Interaction among the coolant discharged from fuel assemblies in the reactor was approximated by three peripheral jets in the quench tank. Deionized, deaerated water with a resistivity of about 0.2 megohm-cm was used as the coolant.

The flow loop is shown schematically in Figure 2. The hydraulics of the test section were not mechanically adjusted during the transients, but were solely dependent upon the characteristics of the system. Because a sufficient bypass flow to maintain constant header pressure during a flow instability in a single assembly was not available, the header pressure was maintained constant by a bank of back-pressure valves.

Initial flows to the axial and annular channels were preset separately and measured with turbine-type flow meters (Pottermeter). Pressures were measured just below the inlet of the axial and annular channels, just above the flow guide in the end fitting, at the monitor pin, and across the metering orifice holes with strain gage pressure transducers. Quench tank pressures were maintained constant by a standpipe.

Silver-tipped iron-constantan thermocouples with a response time of 0.1 second measured coolant temperatures at the inlet to the axial and annular channel, at the exit of the axial channel, at the monitor pin, and in the quench tank.

To begin the tests, the hydraulic and thermal conditions were set at approximately those of a fuel assembly at full power during normal operation. The power was ramped upward by manual control at a rate of about 1/3% per second. The ramp was continued until the coolant in at least one channel became unstable and began to oscillate. The electrical power was then cut off by a circuit breaker. Measurements of the following transients conditions were recorded on high-speed Offner oscillographs.

1. Input electrical power.
2. Axial channel flow
3. Annular channel flow
4. Axial inlet pressure
5. Annular inlet pressure
6. Header (plenum) pressure
7. Inlet temperature.
8. Monitor pin temperature
9. Axial exit temperature
10. Pressure at the flow guide
11. Monitor pin pressure
12. Shell hole pressure drop

Other variables were registered on potentiometers.

RESULTS:

The experimental results are plotted in Figures 3 and 4 and in Table I. The data were corrected as discussed in the next section of this report.

From Figure 4, as the power increased the effluent temperatures increased in phase. Only local perturbations were observed in flows and pressures for about 50 seconds. At that time, the monitor pin temperature began to increase at a more rapid rate than the axial channel effluent temperature, while the annular channel flow began to drop slowly. The difference between the two measured temperatures resulted from an increased ΔT in the annular channel coolant. As shown in the Appendix, nucleate boiling was calculated to have started in the annular channel which caused the flow reduction. The axial channel coolant remained constant and no nucleate boiling was calculated to be present in this channel.

Nucleate boiling was sensed by the transducer that measured the pressure drop across the metering orifices soon after the calculated time for start of boiling. The metering orifice pressure drop continued to increase until such that at about 90 seconds the increase in signal would have been large enough to actuate a Scram I in the reactors. At this time, the assembly flow had dropped about 5%, all of which had occurred in the annular channel. Up to about 90 seconds, no flow change occurred in the axial channel. At about this time, the flow began to decrease, probably as a result of the increase in pressure in the lower end fitting. Nucleate boiling in the axial channel was calculated to have begun at about 106 seconds.

About 15 seconds elapsed between the time when a sufficiently high ΔP signal to scram a reactor was sensed until the flow in one channel became completely unstable. The precise time when the actual critical effluent temperature was reached could not be obtained from these data because of the transient behavior of the system; however, the critical effluent temperature as calculated from steady-state data (122°C for this test section[4] at design flow) was attained at about 90 seconds, or about 15 seconds before the flow decayed to zero. The minimum saturation temperature had increased from 126°C to 130°C because of an increase in bottom fitting pressure. Because this decay time was not well defined, it is concluded only that the time was considerably greater than that (about 2 seconds) measured during the pump shaft break tests[3].

The time intervals for these tests are applicable only to the power ramp used in the test. This ramp did not mockup any specific rod withdrawal rate for a given charge because of the large number of possible rates. Because the rate of change in power is relatively low, the precise ramp rate probably will have little effect on the transients. Ramp rates will be investigated, however, in tests with the Mark VE mockup to verify this assumption.

These tests, along with earlier steady-state tests[4], demonstrate the importance of nucleate boiling in the initiation and propagation of flow instability. The effects should not be ignored in application to fuel assemblies.

The small increase in header pressure near the end of the test delayed the flow rate to some undetermined extent; however, the same increase was observed during the pump shaft break tests.

The effect is presumed small, but the equipment is being modified for future tests to enable the header pressure to remain nearly constant throughout the tests.

The effluent temperature near the exit of the annular channel was not measured because earlier test data[4] showed that the thermocouples in the subchannels did not measure the mixed mean temperature, but registered about 5°C below that calculated from an energy balance. During Run 417-1, the metering orifice Δ P was not measured.

Only two transients were run. During preparations for the third transient, flow became unstable in the test section after several minutes at conditions that had been established as being in the stable flow region. This resulted in melting of the test section. Because instruments were on stand-by, no data were obtained. Examination of the test section failed to reveal positively the cause of the instability, but it was suspected that a pluggage in one subchannel of the annular channel initiated a flow reduction and subsequent melting.

Correction of Experimental Data

The power ramp tests were run immediately after completion of the pump shaft break tests[5]. During the analysis of the pump shaft break data, inconsistencies appeared in the outlet temperature measurements[6,7]. Later, it was discovered by Columbia University that inconsistencies were present also in the flow measurements[8,9]. The data cannot be corrected with any guarantee of accuracy; however, corrections have been made based on previous experiments, energy balances, and judgement. The results appear consistent.

The heater tube and lower end fitting used in these experiments were the same as used for steady-state tests[4]. Data from the steady-state tests were internally consistent and were used as a basis for comparison of flow-Δ P measurements in the axial coolant channel and across the metering orifices. Because of corrosion of the housings used in the steady-state tests, the annular channel flow-ΔP characteristics cannot be compared directly. Details of the calculations are presented in the Appendix, Sections IV and V.

Metering orifice Δ P measurements indicated that the total flow for the power ramp tests was about 3% greater than for the steady-state tests. Δ P measurements across the axial coolant channel indicate that the axial flow was about 11% greater for the power ramp tests. Because an 11% increase in axial flow is about equal to a 3% increase in total flow, it was assumed that the annular flow was measured correctly and that the error was confined to the axial flow measurement. The axial flow as shown in Table I is 11% greater than indicated by the uncorrected data.

The mixed mean effluent temperatures at the start of the transients were calculated from an energy balance, Equation 1:

$$\bar{t} = t_{in} + \frac{(1-x)q}{KF} \tag{1}$$

where:

$\bar{t}$ = mixed mean effluent temperature
t_{in} = plenum temperature
x = fraction of electrical power input lost to surroundings
q = electrical power input
F = assembly flow
K = coolant heat capacity at coolant inlet temperature

Calculation of channel effluent temperatures and monitor pin temperature depends strongly upon the heat split between the coolant channels. The heat split is a function of the temperature difference between the outer and inner surface of the heater. In turn, the surface temperatures are controlled by the heat flux, coolant velocity, channel geometry, and bulk coolant temperature. As shown in the Appendix, Section VII, the channel effluent temperatures were calculated by a trial-and-error procedure. The monitor pin temperature was then determined from the channel effluent temperatures and the calibration of the monitor pin. Calibrations of both a standard production monitor pin and a similar monitor pin used at Columbia University were made[10] and are shown in Figure 5. Monitor pin temperatures were calculated for both cases, and in either case, the monitor pin temperature shows little difference and is 1-2°C less than the mixed mean temperature.

Because the oscillograph operated on a null-balance principal, the difference between the indicated experimental temperatures during the transients and the calculated temperatures at the start of the transients was assumed to be correct. Energy balances during the transients and prior to the start of nucleate boiling confirmed this assumption. Calculation of the start of nucleate boiling showed that boiling started at about the expected conditions which further substantiates the calculated temperatures; however, this calculation is much less conclusive than the energy balance determination.

Measured pressures were probably correct, because the systems were calibrated frequently with compressed gas and a Heise pressure gage. Input powers as measured on the oscillograph and on a potentiometer were in agreement. The inlet temperatures were measured on the same potentiometer as the power.

Application to SRP Fuel Assemblies

Because of uncertainties in the data, it would be technically unsound to apply these results to current SRP fuel assemblies without reservation. Qualitative aspects of the tests, however, are of value in predicting the general behavior of assemblies during a power ramp and in the design of tests with the new Mark VE test section at Columbia University.

For assemblies where a large potential for nucleate boiling exists, such as Mark VII-AL and Mark VII-T, these tests showed that entrainment of vapor from the nucleate boiling will increase the ΔP signal across the metering orifices in the lower end fitting. The increase in signal was sufficiently large to have caused a Scram I in the reactors well in advance of flow instability in the hottest assembly. Although the Columbia mockup had uniform heat generation, the length of heated surface in nucleate boiling for the mockup was approximately the same as for an actual assembly with a cosine heat generation[4]. Nucleate boiling in a fuel assembly would occur nearer the center of the assembly than for the mockup; however, the mockup had about 6 inches of unheated spacer between the end of the heater and the end fitting. Unpublished observations from experiments in the SRL Heat Transfer Lab showed that nucleate boiling vapor may be entrained for several feet in a low subcooled liquid.

WSD:jj

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TABLE I

EXPERIMENTAL RESULTS RUN 417-1*

I Time, Sec.	II P_{ax} psig	III P_{guide} psig	IV Ax ΔP_{ts} psi	V P_{an} psig	VI An ΔP_{ts} psi	VII Mean Eff. Temp., °C
0	92	27.0	83	78	61	109.9
10	92	27.0	83	78	61	111.3
20	93	27.0	83	78	61	112.0
30	93	27.0	83	78	61	113.0
40	94	27.0	83	78	61	113.6
50	94	27.5	83	79	62	116.3
60	93	27.9	83	80	62	116.9
70	94	28.0	83	80	62	118.1
80	93	29.0	83	80	61	119.4
90	94	29.1	83	81	62	120.0
100	94	30.0	82	82	62	121.6
110	94	32.5	79	83	60	124.5
112	95	33.0	79	82	58	126.0
114	94	34.0	77	84	59	126.0
116	94	34.7	76	84	58	127.1
118	95	35.7	76	86	59	127.3
120	96	36.2	77	86	59	128.8
121	95	37.0	75	86	58	127.9
122	96	37.2	76	87	59	128.9
123	96	37.5	76	87	59	129.0
124	95	37.8	74	87	58	130.1
125	96	38.0	75	88	58	130.9
126	96	39.0	73	89	58	132.3
127	97	40.0	73	90	58	133.9

I	II	III	IV	V	VI	VII
128	98	42.0	72	91	57	136.9
129	100	44.9	71	93	55	143.9
130	109	48.7	76	96	-	-
131	106	54.5	67	120	-	-
132	98	-	-	106	-	-
133	91	-	-	94	-	-
134	90	-	-	80	-	-
135	90	26.5	82	77	61	-

* This table contains data not presented in Figure 1. See page 13 for explanation of column headings.

TABLE I (CONTINUED)

EXPERIMENTAL RESULTS RUN 417-2*

I Time, Sec.	II P_{ax} psig	III P_{guide} psig	IV Ax ΔP_{ts} psi	V P_{an} psig	VI An ΔP_{ts} psi	VII Mean Eff. Temp., °C
0	94	27.0	85	80	63	104.0
10	95	27.0	86	80	63	106.6
20	94	27.0	85	80	63	107.7
30	94	27.0	85	80	63	109.3
40	94	27.0	85	80	63	110.8
50	94	27.0	85	80	63	112.9
60	94	27.0	85	80	63	113.9
70	94	27.5	85	80	62	116.6
80	94	28.0	84	81	63	119.7
90	95	31.0	82	84	62	123.2
91	95	31.0	82	84	62	123.2
92	95	32.0	81	84	61	123.8
93	95	32.0	81	84	61	123.8
94	95	32.0	81	85	62	124.5
95	96	32.0	82	85	62	125.3
96	96	33.0	81	86	62	126.3
97	96	33.0	80	86	62	126.7
98	96	34.0	79	87	62	127.7
99	96	35.0	78	88	62	128.8
100	97	36.0	78	88	61	128.8
101	97	37.0	77	89	61	130.1
102	97	36.0	78	90	63	130.7
103	97	37.0	77	91	62	133.9

I	II	III	IV	V	VI	VII
104	98	40.0	74	93	61	135.7
105	100	44.0	72	99	63	139.0
106	104	48.0	72	120	79	151.6

* This table contains data not presented in Figure 2. See page 13 for explanation of column headings.

TABLE I (CONTINUED)

EXPLANATION OF COLUMN HEADINGS

I. Time from start of power ramp.

II. Measured pressure at top of axial coolant channels.

III. Measure pressure at top of guide in lower end fitting.

IV. Difference between Pax and Pguide with corrections for difference in coolant velocities, elevation, pressure tap location, transducer location, and square term losses.

V. Measured pressure at top of annular coolant channel.

VI. Difference between Pan and Pguide with corrections for difference in coolant velocities,elevation, pressure tap location, transducer location, and square term losses.

VII. Calculated mixed mean effluent temperature based on measured inlet temperature,power, and annular channel flow and on calculated axial channel flow. A 1% heat loss is included.

APPENDIX

I. START OF NUCLEATE BOILING IN ANNULAR CHANNEL

The criterion for the start of nucleate boiling is that the actual surface temperature must equal the surface temperature required to start nucleate boiling. Because the Mark VII-AL mockup has uniform heat generation, the surface temperature is highest at the bottom of the heater. The minimum temperature for the start of nucleate boiling is also at the bottom of the heater because the pressure and subcooling are lowest at this location.

A. Surface Temperature - Annular Channel

See Section X of Appendix for notations.

The surface temperature is calculated by trial-and-error from the following basic equations:

$$Q/A = h\ (t_s - t_b) \tag{1A}$$

$$h = 200\ (1+0.012\ t_f)\ \frac{v^{0.8}}{D_e^{0.2}} \quad \text{(Derived from Dittus-Boelter equation}^{(12)}\text{)} \tag{2A}$$

$$t_f = \frac{t_s + t_b}{2} \tag{3A}$$

Combine Equations 1A, 2A, and 3A

$$Q/A = \left\{200\ \left[1+0.012\left(\frac{t_s + t_b}{2}\right)\right] \frac{v^{0.8}}{D_e^{\ 0.2}}\right\}\left\{t_s - t_b\right\} \tag{4A}$$

Or:

$$t_s^{\,2} + 167t_s - \left[\frac{(0.833\ Q/A)(D_e^{\,0.2})}{v^{0.8}} + 167t_b + t_b^{\,2}\right] = 0 \tag{5A}$$

In quadradic form:

$$t_s = \frac{-167 \pm \sqrt{167^2 + 4\left[\frac{(0.833Q/A)(D_e^{\,0.2})}{v^{0.8}}+167t_b+t_b^{\,2}\right]}}{2} \tag{6A}$$

At 50 seconds from start of transient for Run 417-2

$Q/A = 414{,}000$ pcu/hr-ft^2
$D_e = 0.242$ inch
$v = 26.2$ ft/sec
$t_b = 116.5$°C

From Equation 6A, $t_s = 159$°C

B. Surface Temperature Required to Start Nucleate Boiling

The surface temperature required to start nucleate boiling is calculated from a method presented in Reference 12 and simplified to a graphical solution shown in Reference 11 and Figure 6.

[redacted]

[redacted]

Pressure at base of channel	= 27 psig
Saturation temperature	= 132°C
Coolant temperature	= 116.5°C
Subcooling	= 15.5°C
t_{lb} - t_{sat}	= 27°C
t_{lb}	= 159°C

At 50 seconds, the calculated surface temperature equals the calculated temperature required to start nucleate boiling. Actually, it would be expected that nucleate boiling would start prior to this time because spacer ribs increase the local heat flux and reduce the coolant velocity adjacent to the ribs.

II. LACK OF NUCLEATE BOILING IN AXIAL CHANNEL

A. Surface Temperature - Axial Channel

At 50 seconds from start of transient for Run 417-2:

$$Q/A = 572{,}000 \text{ pcu/hr-ft}^2$$
$$D_e = 0.5 \text{ inch}$$
$$v = 49.1 \text{ ft/sec}$$
$$t_b = 106°C$$

From Equation 6A, t_s = 149°C

B. Surface Temperature Required to Start Nucleate Boiling

Pressure at base of channel	= 27 psig
Saturation temperature	= 132°C
Coolant temperature	= 106°C
Subcooling	= 26°C
t_{lb} - t_{sat}	= 33°C
t_{lb}	= 165°C

Because t_{lb} is greater than t_s, no nucleate boiling occurs.

III. START OF NUCLEATE BOILING IN AXIAL CHANNEL

A. Surface Temperature - Axial Channel

At 106 seconds from start of transient for Run 417-2:

$$Q/A = 681{,}000 \text{ pcu/hr-ft}^2$$
$$D_e = 0.5 \text{ inch}$$
$$v = 44.4 \text{ ft/sec}$$
$$t_b = 126.7°C$$

From Equation 6A, t_s = 177°C

B. Surface Temperature Required to Start Nucleate Boiling

Pressure at base of channel	= 48 psig
Saturation temperature	= 146°C
Coolant temperature	= 126.7°C
Subcooling	= 19°C

[redacted]

$$t_{lb} - t_{sat} = 31\,°C$$
$$t_{lb} = 177\,°C$$

At 106 seconds t_{lb} equals t_s, and nucleate boiling starts.

IV. COMPARISON OF METERING ORIFICE Δ P SIGNALS

The flow through the assembly for the 417 test series was corrected based on the metering orifice Δ P-flow relationships for the 400 series steady-state tests.

Run 417-2
Flow = 70.2 gpm measured
Δ P = 170 inches H_2O
Effluent Temp. = 104 °C

Run 401, pt. 4
Flow = 63.7 gpm measured
Δ P = 130 inches H_2O
Effluent Temp. = 82 °C

Corrected flow for Run 417-2

$$F_{calc} = 63.7\sqrt{\frac{170''\ H_2O}{130''\ H_2O} \times \frac{60.58\#/ft^3}{59.64\#/ft^3}} = 73.4 \text{ gpm}$$

$$\frac{F_{calc}}{F_{meas}} = \frac{73.4}{70.2} = 1.045$$

Run 406, pt. 6
Flow = 68.2 measured
Δ P = 154 inches H_2O
Effluent Temp. = 99 °C

Corrected flow for Run 417-2

$$F_{calc} = 68.2\sqrt{\frac{170''\ H_2O}{154''\ H_2O} \times \frac{59.87\#/ft^3}{59.64\#/ft^3}} = 71.8 \text{ gpm}$$

$$\frac{F_{calc}}{F_{meas}} = \frac{71.8}{70.2} = 1.023$$

Other steady-state data also indicated a 2-4 gpm correction in total flow. The sensitivity of the Δ P differences to total flow differences is insufficient to apply a correction to the axial channel flow, and should be used only as an indication that the total measured flow for the transient tests was low.

V. COMPARISON OF AXIAL COOLANT CHANNEL Δ P MEASUREMENTS

The flow through the assembly for the 417 test series is corrected based on the Δ P-flow relationships for the 400 series steady-state tests.

Run 417-2
Axial channel flow = 26.2 gpm measured
Δ P = 83.1 inches H_2O
Avg. coolant temp. = 76 °C

Run 401, pt. 3
Axial channel flow = 24.9 gpm measured
ΔP = 61.7 inches H_2O
Avg. coolant temp. = 56°C

Corrected flow for Run 417-2

$$F_{calc} = 24.9 \sqrt[1.8]{\frac{83.1'' \ H_2O}{61.7'' \ H_2O} \times \frac{61.52\#/ft^3}{60.82\#/ft^3}} = 29.6 \text{ gpm}$$

$$\frac{F_{calc}}{F_{meas}} = \frac{29.6}{26.2} = 1.13$$

Run 405, pt. 7
Axial channel flow = 26.4 gpm measured
ΔP = 73.4 inches H_2O
Avg. coolant temp. = 58°C

Corrected flow for Run 417-2

$$F_{calc} = 26.4 \sqrt[1.8]{\frac{83.1'' \ H_2O}{73.4'' \ H_2O} \times \frac{61.46\#/ft^3}{60.82\#/ft^3}} = 28.45 \text{ gpm}$$

$$\frac{F_{calc}}{F_{meas}} = \frac{28.5}{26.2} = 1.09$$

Run 406, pt. 6
Axial channel flow = 25.2 gpm measured
ΔP = 66.1 inches H_2O
Avg. coolant temp. = 66°C

Corrected flow for Run 417-2

$$F_{calc} = 25.2 \sqrt[1.8]{\frac{83.1'' \ H_2O}{66.1'' \ H_2O} \times \frac{61.20\#/ft^3}{60.82\#/ft^3}} = 28.7 \text{ gpm}$$

$$\frac{F_{calc}}{F_{meas}} = \frac{28.7}{26.2} = 1.10$$

Based on these tests, an 11% correction was applied to the transient data for Runs 417-1 and 2.

VI. MIXED MEAN EFFLUENT TEMPERATURE

The mixed mean effluent temperature was calculated from the following equation at the start of Run 417-2.

$$\bar{t} = t_{in} + \frac{(1-x)q}{KF} \tag{7A}$$

$$\bar{t} = 46.8 + \frac{(1-.01)1110}{0.263 \times 73.1} = 104.0°C$$

VII. DETERMINATION OF CHANNEL EFFLUENT TEMPERATURES

The procedure used to determine the channel effluent temperatures was as follows:

1. Assume a heat split between the annular and axial coolant channels.
2. Calculate the effluent temperatures for both channels from the measured power and inlet temperature and from the corrected flows.
3. Calculate the surface temperatures, from the assumed heat split, power, coolant velocity, bulk temperature, and channel equivalent diameter.
4. Calculate the heat split from the power, channel dimensions, and surface temperatures.
5. Repeat until the assumed heat split equals the calculated heat split.

Assume a heat split to the annular channel of 0.64

Power to outer annulus = $q(1-x)\eta$

Power = (1110x.99) 0.64 = 704KW

$$\Delta T = \frac{q(1-x)}{KF} = \frac{704}{0.263 \times 44.0} = 60.7°C \qquad (8A)$$

Annular channel eff. temp. = $T_{in} + \Delta T = 46.8 + 60.7 = 107.5°C$

Axial channel effluent temperature = $\frac{q(1-x)(1-\eta)}{KF} + t_{in}$

$$\frac{(1110 \times 0.99)(0.36)}{0.263 \times 29.1} + 46.8 = 98.5 \qquad (9A)$$

Calculate surface temperatures at mid-plane
$t_b = 77.1°C$ for annular channel

$$Q/A = \frac{q(1-x)\eta \times \text{conv.factor}}{A} = \frac{1110 \times 0.99 \times 0.64 \times 1896}{3.70} = 361{,}000 \text{ pcu/hr-ft}^2 \qquad (10A)$$

From Equation 6A
$t_s = 122°C$ for midplane of heater in annular channel

$t_b = 72.6$ for axial channel, and by like analogy,
$t_s = 116°C$ for midplane of heater in axial channel

To calculate heat split for cylinder with internal heat generation:

$$\eta = \frac{(D_o/D_i)^2}{(D_o/D_i)^2-1} - \frac{4\pi Lk(t_o-t_i)}{Q \ln(D_o/D_i)^2} - \frac{1}{\ln(D_o/D_i)^2} \qquad (11A)$$

Equation 11A reduces to the following for this test section:

$$\eta = 0.640 - 919 \frac{(t_o-t_i)}{Q}$$

[redacted]

$$\eta = 0.640 - 919 \left(\frac{122-116}{1110x.99x1896}\right) = 0.637$$

assumed η = 0.640
calculated η = 0.637
Now assume η = 0.637 and repeat the above
Calculated η now becomes 0.638
Because of the rapid convergence of η, η is considered as 0.638.
If η calculated at the top and bottom of the test section is approximately the same as for the midplane, it may be assumed that this value applies as the integrated average heat split and that the effluent temperatures calculated using this heat split are approximately correct.

η at top of heater = 0.640
η at bottom of heater = 0.636

Avg. integrated heat split = 0.638
From this heat split, the measured electrical power and inlet temperature, and the corrected flows:

$t_{\text{annular outlet}}$ = 107.3°C
$t_{\text{axial outlet}}$ = 98.9°C

VIII. DETERMINATION OF MONITOR PIN TEMPERATURE

$$t_{mp} = f_{an}\, t_{an} + f_{ax}\, t_{ax} \qquad (12A)$$

If the calibration of the Columbia University monitor pin applies, the monitor pin temperature (from Figure 5) is:

t_{mp} = 0.44 x 107.3 + 0.56 x 98.9 = 102.6°C

If the calibration of the standard production monitor pin applies, the monitor pin temperature is:

t_{mp} = 0.486 x 107.3 + 0.514 x 98.9 = 103.0°C

The difference is of no consequence in this analysis.

IX. MASS-ENERGY BALANCE AT START OF NUCLEATE BOILING

A mass-energy balance was made to show that the transient temperatures may be obtained from the Offner oscillograph traces when the starting effluent temperature is known even though the calibration was not correct.

At 50 seconds from start of transient for Run 417-2:

Electrical input	= 1.27MW
Axial channel flow	= 28.9 gpm
Inlet temperature	= 46.8°C
Heat split	= 0.362, to axial channel
Effluent temperature	= 106.1°C, from Offner charts

$$t_{ax} = \frac{1.27 \times .99 \times 0.362}{0.263 \times 28.9} + 46.8 = 106.7$$

$$106.1 \cong 106.7$$

X. NOTATIONS

A	Surface area of clad fuel tube exposed to coolant channel
D	Diameter
F	Total assembly coolant flow unless otherwise indicated
f	Fraction of coolant impinging on monitor pin thermocouple from the designated coolant channel[10]
h	Liquid film heat transfer coefficient
K	Heat capacity of coolant at inlet temperature
k	Thermal conductivity
L	Fuel column length
m	Fraction of heat that has entered coolant stream divided by total heat entering stream
n	Local heat flux divided by average heat flux
P	Pressure, psig
Q	Heat, pcu
q	Heat or equivalent electrical power, KW or MW
t	Temperature
$\bar{t}$	Mixed mean effluent temperature
v	Mean velocity of coolant
x	Fraction of power input lost to surroundings
Δ	Indicates a difference
η	Fraction of assembly heat that is transferred to annular coolant channel
π	3.1416...

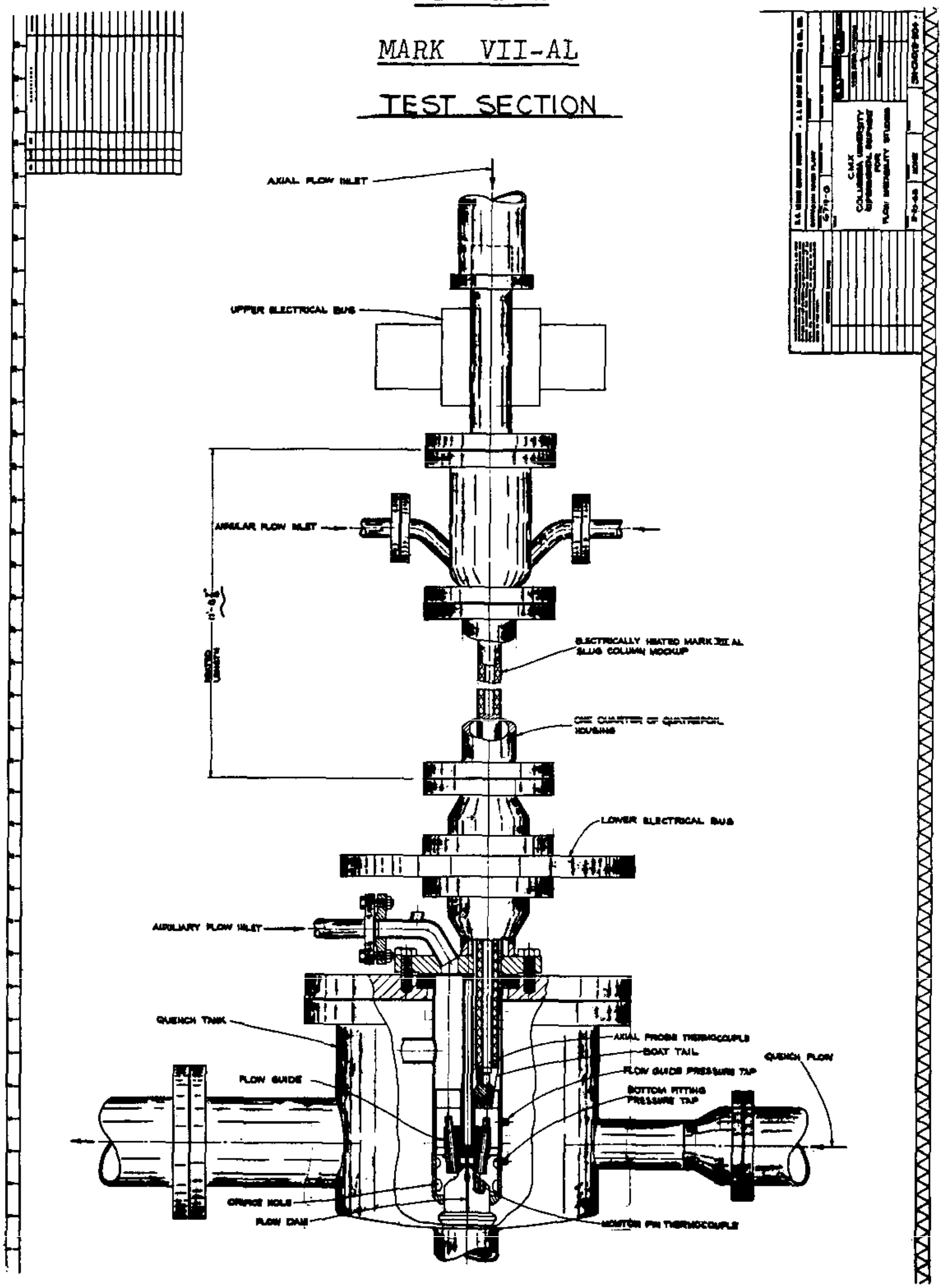
Subscripts

an	At exit of annular coolant channel
ax	At exit of axial coolant channel
b	Bulk Coolant
calc	Calculated value
e	Equivalent
f	Film condition, arithmetic average of bulk and surface conditions
in	At inlet to test section
i	Inner wall of heater
lb	Represents conditions for start of nucleate boiling
meas	Measured values
mp	Monitor pin
o	Outer wall of heater
s	Surface
sat	Saturation

FIGURE-1

MARK VII-AL

TEST SECTION



[redacted]

FIGURE 2

COLUMBIA UNIVERSITY FLOW LOOP FOR INSTABILITY TESTS

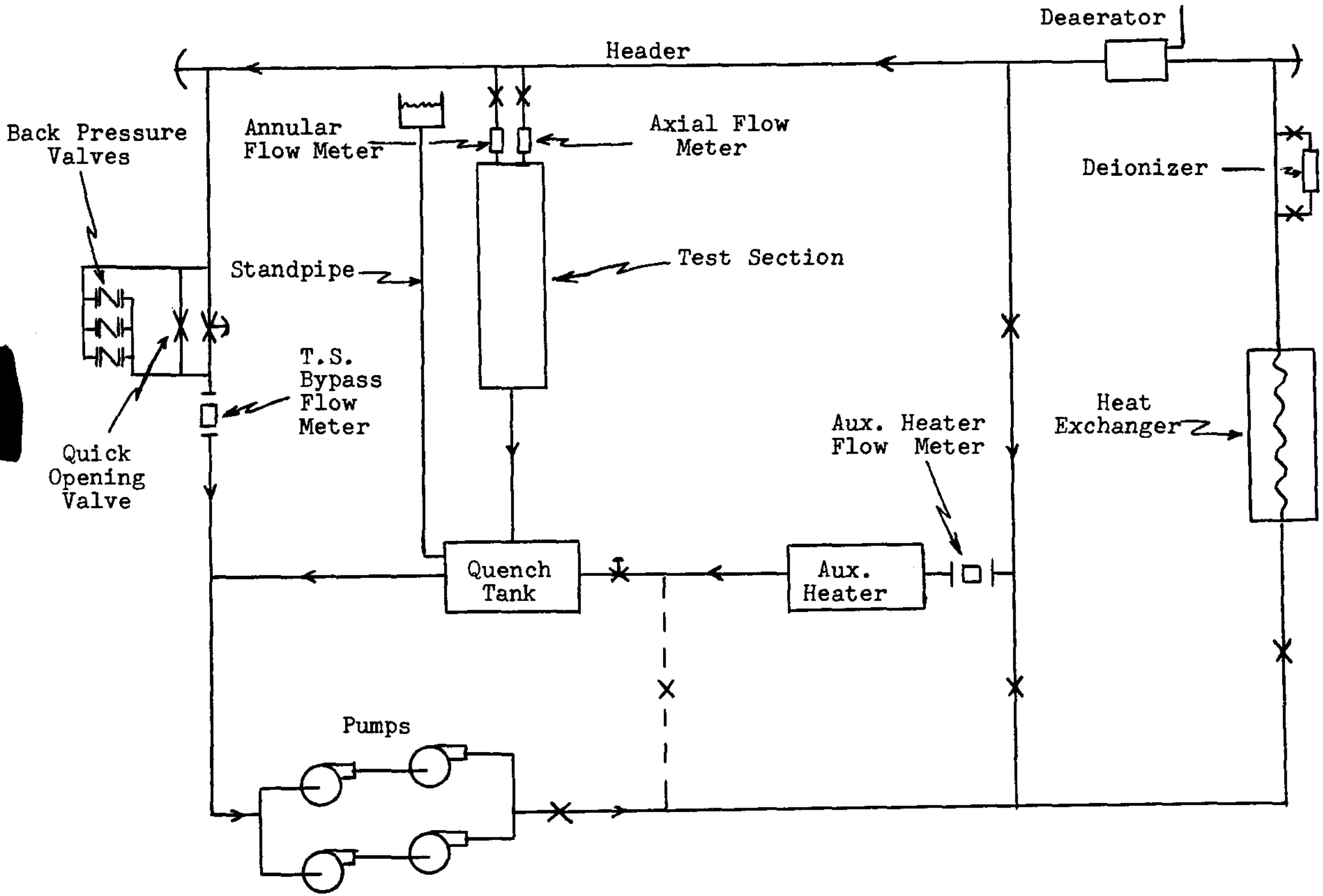


FIGURE 3

POWER RAMP TEST DATA FROM MARK VII-A1 MOCKUP AT COLUMBIA UNIVERSITY

Run 417-1

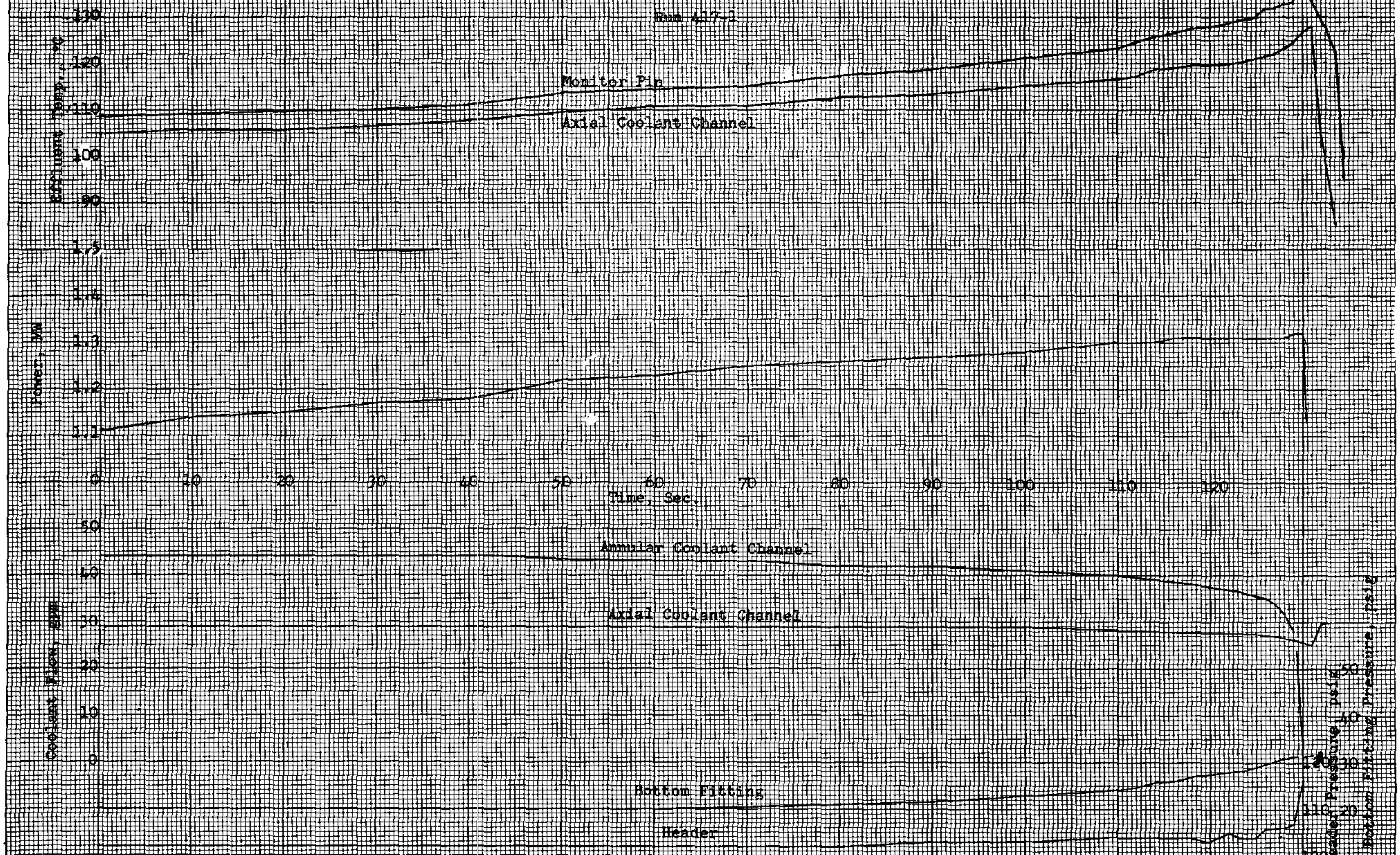
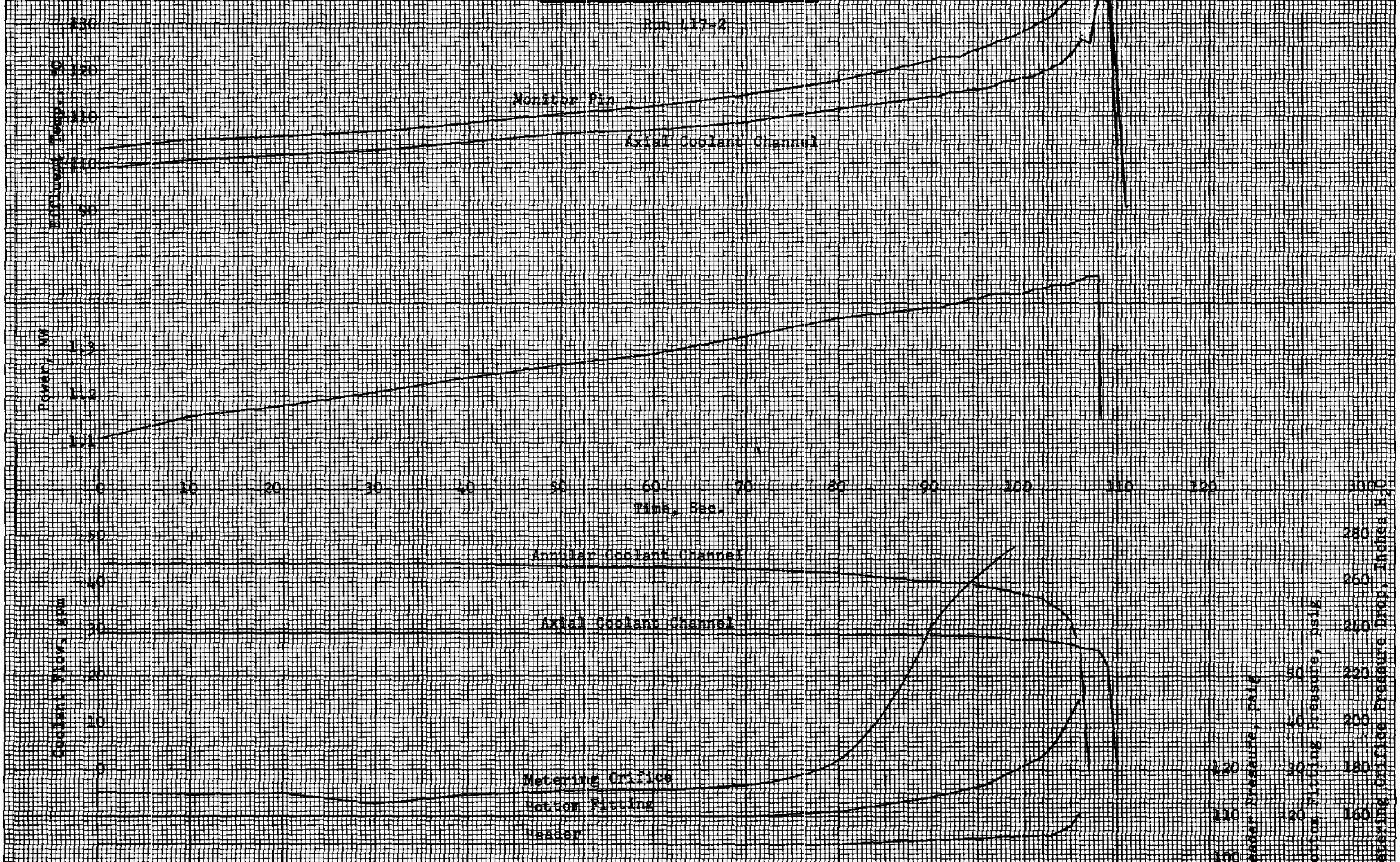


FIGURE 4

POWER RAMP TEST DATA FROM MARK VII-AL MOCKUP AT COLUMBIA UNIVERSITY

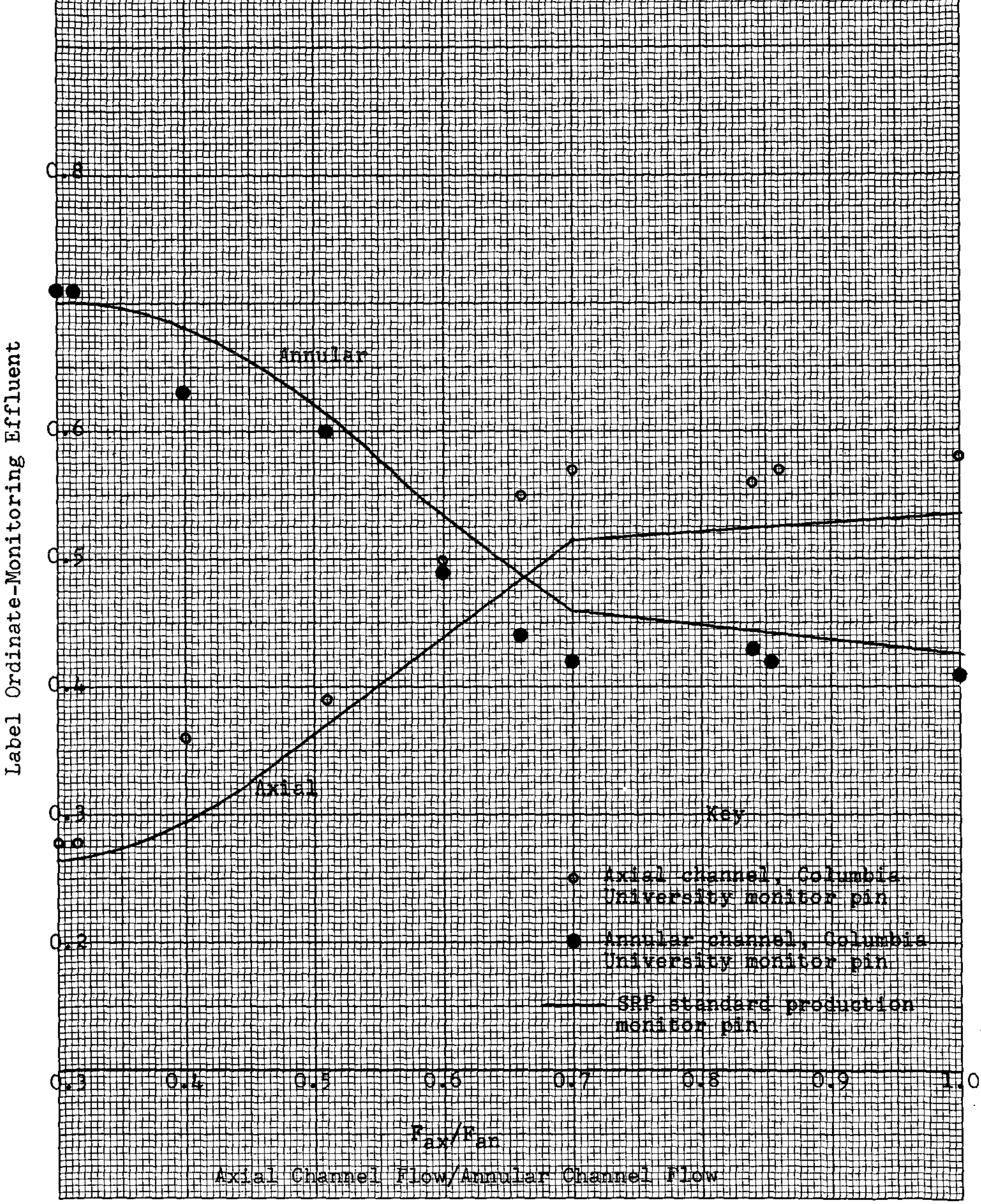
Run 117-2



[redacted]

FIGURE 5

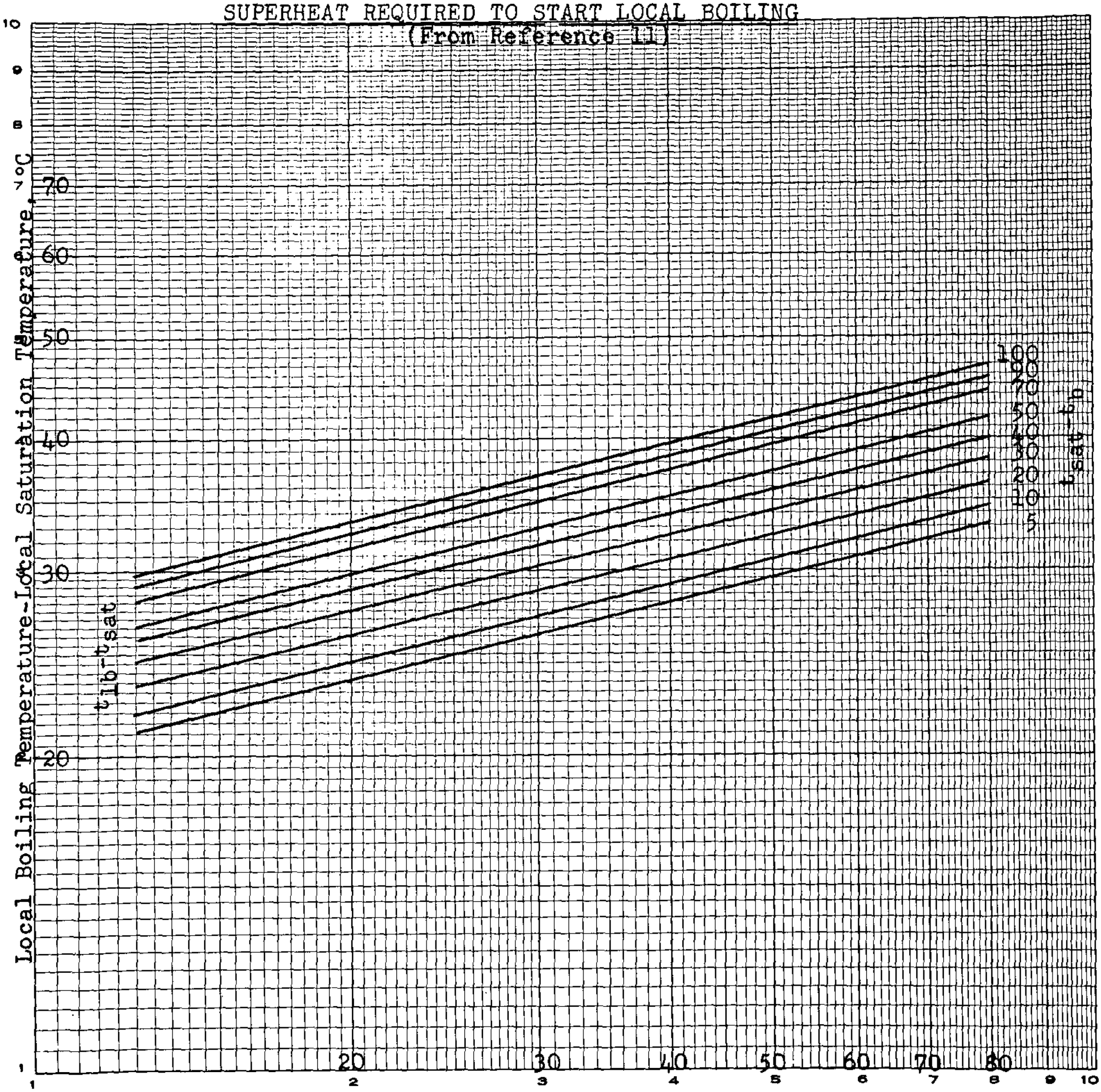
MONITOR PIN CALIBRATION TESTS
(From Reference 10)



[redacted]

FIGURE 6

SUPERHEAT REQUIRED TO START LOCAL BOILING
(From Reference 11)



[redacted]