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TWO-PHASE PRESSURE DROP AND BURNOUT USING WATER FLOWING IN ROUND AND RECTANGULAR CHANNELS

by

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Chemistry

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Table of Symbols and Units[a]

c_p	Specific heat at constant pressure	$Btu/(lb_m)(F)$
D	Diameter (inside of conduit)	ft
D_e	Equivalent diameter of conduit	ft
E	Voltage	volts
f	Friction factor	dimensionless
g	Gravitational acceleration, local, usually taken as 32.2	ft/(sec)(sec)
g_c	Proportionality factor in Newton's law of motion	$32.2\ [lb_m/lb]\ [ft/(sec)(sec)]$
G	Mass velocity	$lb_m/(sec)(sq\ ft)$
h_{fg}	Latent heat of vaporization	$Btu/(lb_m)$
I	Current	amperes
L	Length of conduit	ft
P	Pressure	(lb/sq ft) or (in. of H_2O)
Q	Heat input	Btu/hr
v_f	Specific volume of saturated liquid	$cu\ ft/lb_m$
v_{fg}	Specific volume change due to vaporization	$cu\ ft/lb_m$
V	Velocity	ft/sec
W	Mass flow rate	lb_m/hr
x	Steam quality, as fraction by weight of steam	dimensionless
z	Vertical distance above any arbitrary datum plane	ft
t_s	Inlet subcooling	F
ρ	Density	$lb_m/cu\ ft$
ρ_f	Density of saturated water	$lb_m/cu\ ft$
ρ_{fg}	Density change due tp vaporization	$lb_m/cu\ ft$

[a] lb_m designates pound mass. All other lb designate pound force.

TWO-PHASE PRESSURE DROP AND BURNOUT USING WATER FLOWING IN ROUND AND RECTANGULAR CHANNELS

ABSTRACT

Two-phase pressure drop data and burnout data are reported for a vertical rectangular channel with upflow of water within the channel. Burnout data are also reported for upflow of water through a vertical pipe.

A comparison is made between experimental and calculated pressure drops. The Martinelli-Nelson method of calculating two-phase pressure drop is used with a suitable correction for hydrostatic head.

Burnout data indicate that vertical channels or tubes with upflow of water within the tubes will not overheat or burn out until the value of exit quality ranges from 70 to 100 per cent for the range of variables:

Inside diameter	0.1 to 1 inch
Length-diameter ratio	over 40
Heat flux	up to 300,000 Btu/(hr)(sq ft)
Inlet subcooling	up to 150F
Exit pressure	14.7 psia

Extrapolation to other ranges is not recommended.

The effect of length-diameter ratio on the value of exit quality at burnout is also given.

I. INTRODUCTION

In heat exchange apparatus incorporating many flow channels in parallel, each channel receives a share of the flow. The amount of the flow received by each channel depends upon channel geometry and the orientation of the channels with each other as determined by the connecting headers. The flow in a channel is governed by the total pressure drop across the channel. If the resistance to flow changes through any particular channel, the flow in that channel will change. In particular, if steam is formed in any channel during the flow of water through a parallel channel arrangement, certain channels may be deprived of water. The resistance to flow may cause a stoppage of water and consequent burnout. For this reason, a knowledge of the conditions for burnout and a means of calculating two-phase pressure drop of steam-water mixtures is desirable. There are methods in the literature[1,2] for estimating two-phase pressure drop. Some data[3,4] are available for determining burnout inside long round channels at pressures up to 2000 psia.

II. PURPOSE

The purpose of this experiment was to determine the conditions for burnout in a single vertical rectangular channel, a single inclined rectangular channel, and a vertical round channel; the heated length of the round channel was varied in some of the tests to determine the effect of L/D on burnout.

Two-phase pressure drop was measured for a single vertical rectangular channel in order to predict the performance of channels operating in parallel; additional tests were run using two inclined rectangular channels in parallel to determine if their performance could be predicted.

III. EQUIPMENT

The test equipment consisted of a water storage tank, electrically heated stainless steel test sections, and the necessary instrumentation to determine power input, surface temperatures, inlet water temperatures, and water mass flow rates.

A. Storage Tank

A 90-gallon stainless steel AISI type 304 water storage tank with heaters was used in some of the tests to obtain high inlet water temperatures. High water temperatures were maintained in the storage tank by using three calrod units, each dissipating about four kilowatts. The tank was not used for any burnout tests that required a supply pressure head greater than ten feet of water. For tests requiring a greater supply pressure head, the water was taken directly from the laboratory piping system. This provided about 80 psi.

B. Test Sections

The test sections for the two-phase pressure drop tests were 0.087 by 2.25 by 43-inch rectangular cross section stainless steel AISI type 347 channels. These channels were also used for burnout tests. A 0.94-inch ID type 347 pipe was also used in some of the burnout tests to determine the effect of length-diameter ratio on the value of exit quality at burnout. This pipe was designed such that the heated length could be varied from 8 inches to 36 inches.

Laboratory tap water was forced upward through these sections; heat was generated within the metal walls of the test sections by connecting directly across the secondary of a low-voltage high-current transformer. The exit of the channels and the pipe discharged to the atmosphere. In the particular case where two channels were connected for parallel flow at an inclination of 14.5 degrees, the entrance and discharge ends were attached to Lucite plenum chambers. Fig. 1 shows an assembly of the parallel flow apparatus with the Lucite plenum chambers. Fig. 2 shows a schematic diagram of the apparatus including storage tank, flow meters, and parallel flow arrangement.

C. Instrumentation

1. Power Input

The total power input from the transformer was calculated from measurements of voltage drop across the test sections and current flow; previous measurements indicated a power factor of unity, permitting direct calculation of heat input.

2. Surface Temperatures

A thermocouple circuit was designed to detect overheating of the test channels. The circuit consisted of five thermocouples connected in series with the potential being fed into a preamplifier and Brush recorder. This circuit permitted a rapid means of detecting overheating but did not indicate the position of overheating. Fig. 3 shows the arrangement of the thermocouple circuit.

3. Inlet Water Temperature

The inlet water temperatures were measured with a thermocouple and portable precision potentiometer. The inlet water thermocouple was connected to the outer surface of the test section upstream from the heated portion. This upstream section was made from copper sheet. Calculations indicated a maximum possible error in inlet water temperature of about two degrees using this method of measurement.

4. Flow Rate

Two visual type rotameters were used to indicate the mass flow rate of water to the test sections. These meters were located near the test section entrance. Precautions were taken to eliminate all air pockets from the flow line between the rotameters and the test sections. The indicated flow was therefore the true flow at the test section.

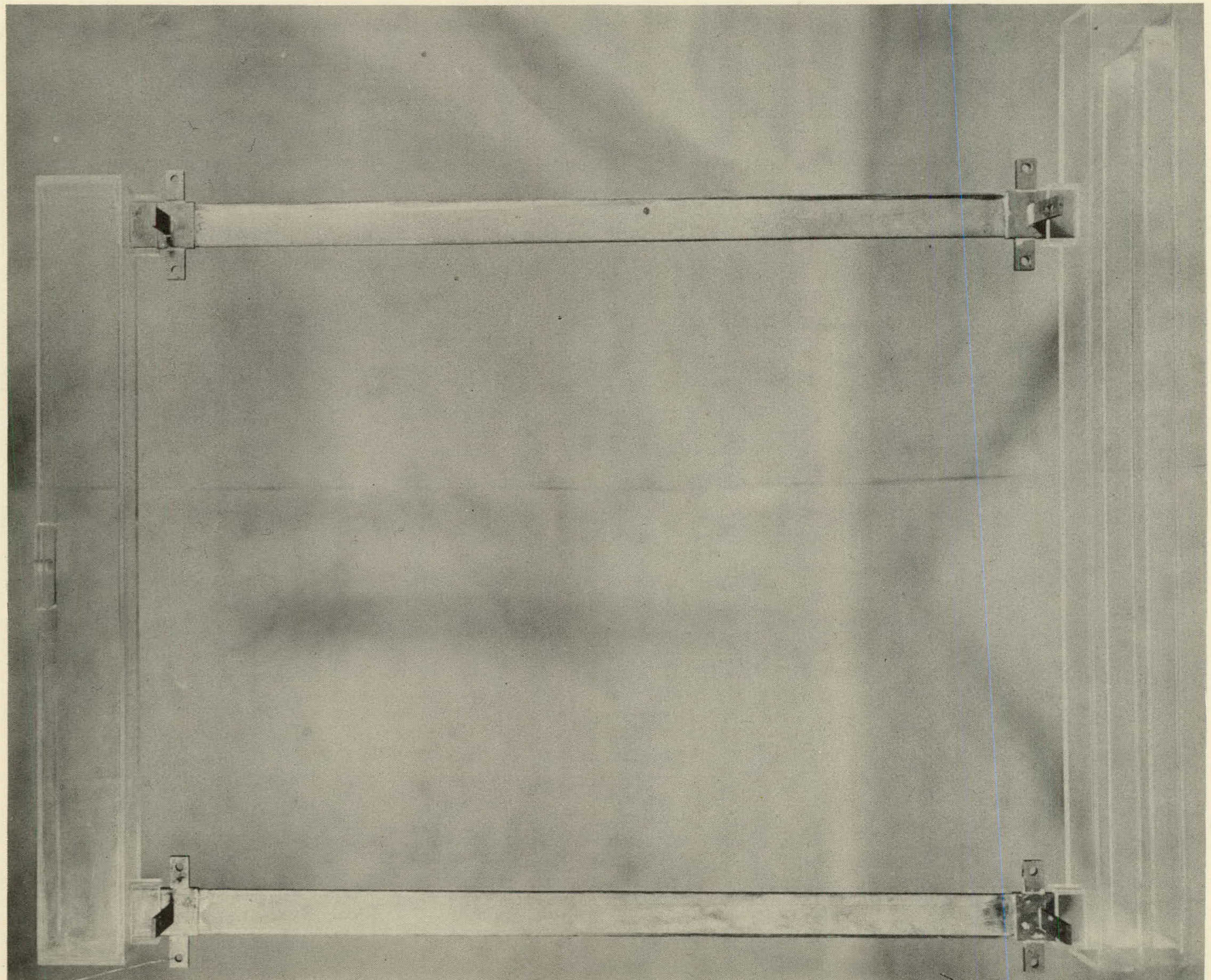

FIG. I
PARALLEL CHANNEL ASSEMBLY

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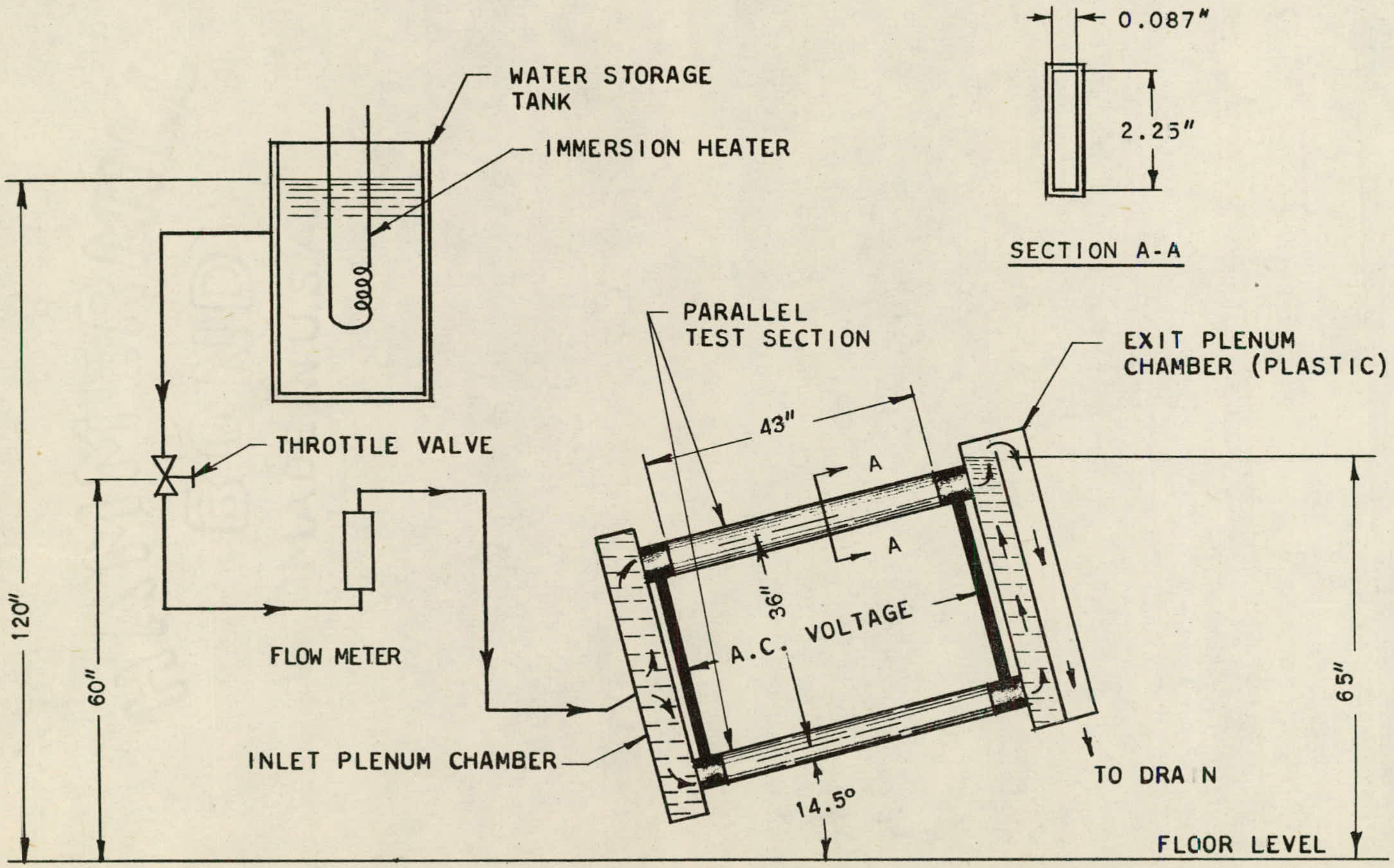
FIG. 2
LOW PRESSURE PARALLEL FLOW HEAT TRANSFER APPARATUS



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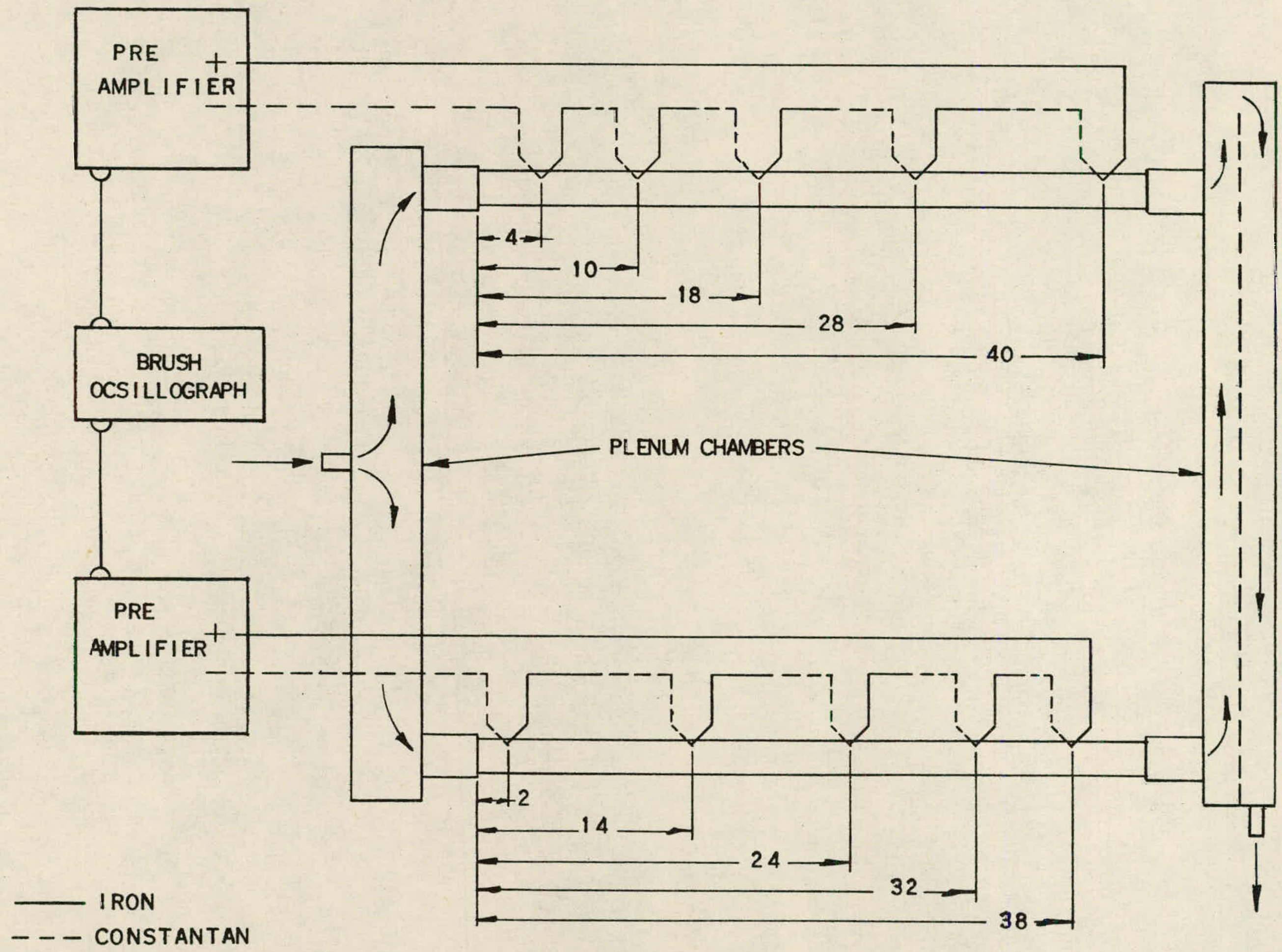

FIG. 3
THERMOCOUPLE WIRING DIAGRAM

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5. Pressure Drop

The total static pressure drop was measured at the bottom of the vertical channel by using a vertical water leg as shown in Fig. 4. In the case of large pressure drops, a combination mercury and water open end manometer was used. The exit pressure of the vertical channel was measured at various times and was always found to be less than 1/4-inch of water. The total static pressure as indicated by the manometer was therefore recorded as being the total static pressure drop.

IV. TEST PROCEDURE

Burnout: In the single vertical rectangular channel burnout tests, a fixed voltage was applied to the test section, and the flow was decreased until the surface temperature rose sharply. It was noticed that the surface temperature remained constant throughout the boiling region, independent of the flow rate. The flow when overheating occurred at the exit was recorded in addition to the power input and inlet temperature.

When the rectangular channel was inclined at 14.5 degrees from the horizontal, it was observed that the overheating did not occur at the exit but instead at some position about halfway along the channel. The position of overheating could not be determined accurately with the instrumentation used. The flow rate, power input, and inlet temperature were all recorded at the flow rate at which overheating occurred.

In the case of the round vertical channel, overheating occurred at the exit as in the case of the rectangular channel. In addition to recording the flow rate, power input, and inlet temperature; the heated length was also varied and recorded.

Pressure Drop: The two-phase pressure drop tests were made using a vertical rectangular channel with a constant heat flux and variable flow rate. The flow rate, power input, inlet water temperature, and pressure drop were measured.

Using the inclined parallel flow section, tests were run first with the upper channel heated, then with the lower channel heated, and finally with both channels heated uniformly. The inlet temperature, total power input, and total flow rate to each channel were measured.

Since corrosion of the heat transfer surface or a deposit on the heat transfer surface could have caused an increase in the measured pressure drop, frequent measurements of the pH of the laboratory water were taken in order to provide some indication of the conditions under which the test was being conducted.

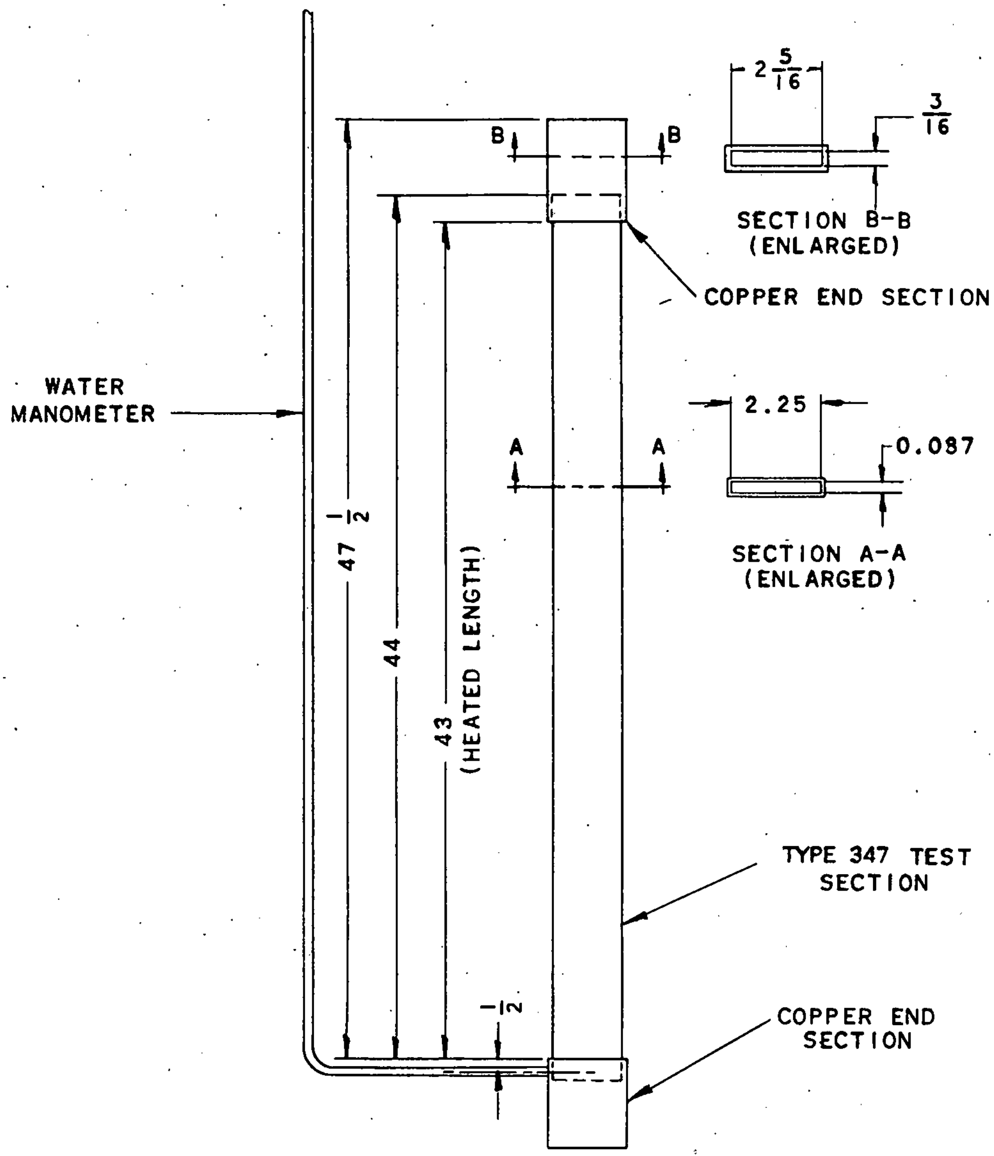

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FIG. 4
LOCATION OF PRESSURE DROP MANOMETER

V. EXPERIMENTAL DATA

Table I lists burnout data for a single vertical channel. Table II lists the burnout data for a single heated channel at an inclination of 14.5 degrees. Table III lists the burnout data for a vertical round tube with variable length. Table IV lists two-phase pressure drop data for a single vertical channel. Table V lists burnout data for parallel flow sections at an inclination of 14.5 degrees.

Table I

BURNOUT DATA FOR A SINGLE VERTICAL HEATED CHANNEL[a]

Mass Flow Rate, lb_m/hr	Burnout Heat Flux, Btu/(hr)(sq ft)	Exit[b] Quality
	80F Inlet Temperature	
41	34,300	1.07
29	26,200	1.16
26	22,100	0.99
41	34,300	0.97
	60F Inlet Temperature	
45	33,700	0.83
113	86,500	0.96
14	13,300	1.21
18	15,200	1.07
19	16,900	1.13
18.5	16,900	1.17
21	16,900	1.01
25	21,200	1.07
26	21,200	1.03
25	21,200	1.07
27	37,000	0.78
52	38,100	0.91
51	38,100	0.93
51	27,400	0.92
37	27,400	0.92
77.5	56,000	0.90
76	56,000	0.90
76	56,000	0.90

[a]Measurements made at 15 psia exit pressure

[b]Average Quality - 0.995 or 99.5%

Table II

BURNOUT DATA FOR A SINGLE CHANNEL WITH 14.5 DEGREE INCLINATION[a]

Inlet Temp., F	Mass Flow Rate, lb_m/hr	Burnout Heat Flux, Btu/(hr)(sq ft)
180	37	11,000
178	78	18,200
178	48	12,700
179	95	25,900
176	105	27,100
174	20	9,470
180	20	5,600
163	100	25,100
160	100	26,400
160	70	21,200
68	60	17,200
68	45	14,800
68	35	11,900

[a]Measurements made at 15 psia exit pressure.

Table III

BURNOUT DATA FOR A 0.94-IN. ID VERTICAL HEATED PIPE[a]

Mass Flow Rate, lb_m/hr	Burnout Heat Flux, Btu/(hr)(sq ft)	Exit Quality	L/D Ratio
36	49,500	0.83	36
55	68,200	0.73	36
74	93,600	0.75	36
77	93,600	0.72	36
77	93,600	0.72	36
73	93,600	0.75	36
118	191,000	0.52	36
48	77,000	0.80	28.2
95	128,000	0.64	28.2
109	154,000	0.68	28.2
127	183,000	0.70	28.2
79	103,000	0.50	25.5
76	104,000	0.70	25.5
69	115,000	0.69	25.5
113	137,000	0.45	25.5
122	137,000	0.32	25.5
145	218,000	0.50	25.5
137	218,000	0.65	25.5
139	218,000	0.64	25.5
113	191,000	0.52	20
53	113,000	0.69	18.8
80	170,000	0.69	18.8
100	188,000	0.592	18.8
35	105,000	0.81	15.3
95	191,000	0.49	15.3
145	268,000	0.44	15.3
54	161,500	0.56	11.3
113	234,000	0.34	11.3
164	299,000	0.28	11.3
168	299,000	0.29	11.3
85	248,000	0.34	8.5
100	248,000	0.26	8.5
105	248,000	0.24	8.5
113	281,000	0.20	8.5

[a]Measurements made at 15 psia exit pressure and 60F inlet temperature

Table IV

TWO-PHASE PRESSURE DROP DATA FOR WATER FLOWING UPWARD THROUGH A SINGLE VERTICAL HEATED CHANNEL[a]

Total Mass Flow, lb_m/hr	Static Pressure Drop,[b] in. of H_2O	Heat Flux, Btu/(hr)(sq ft)	Total Mass Flow, lb_m/hr	Static Pressure Drop,[b] in. of H_2O	Heat Flux, Btu/(hr)(sq ft)
356	55.9	51,500	49	23.9	14,400
275	65.8	51,500	32	20	14,400
234	68.5	51,500	24	16	14,400
183	71.6	51,500	285	49.6	47,500
135	70.5	51,500	220	54.0	47,500
94	63.2	51,500	183	54.8	47,500
82	61.9	51,500	135	55.6	47,500
287	50.1	0	100	53.3	47,500
92	47.5	0	80	52.7	47,500
376	48.8	25,600	65	51.4	47,500
278	45.9	25,600	354	46.2	47,500
245	43.3	25,600	254	48.8	13,500
184	38.6	25,600	183	47.5	13,500
135	34.9	25,600	53	24.4	13,500
90	29.7	25,600	28	17.4	13,500
61	26.5	25,600	22	16.1	11,900
36	23.9	25,600	27	16.4	11,900
368	46.0	33,500	31	19.2	11,900
274	42.8	33,500	36	21.3	11,900
234	42.0	33,500	40	23.7	11,900
188	42.2	33,500	46	25.2	11,900
140	41.5	33,500	56	26.5	11,900
97	39.6	33,500	67	29.1	11,900
53	34.1	33,500	77	31.8	11,900
380	49.1	19,800	88	33.9	11,900
278	48.7	19,800	98	36.5	11,900
228	47.5	19,800	126	47.0	11,900
185	43.4	19,800	108	40.4	11,900
125	35.2	19,800	75	30.5	13,500
102	31.0	19,800	100	36.2	13,500
70	26.5	19,800	58	36.5	6,350
49	23.1	19,800	67	41.7	6,350
33	18.7	19,800	79	46.2	6,350
25	16.6	19,800	59	37.0	6,350
282	49.5	0	50	33.1	6,350
375	49.6	14,400	15	13.5	6,350
283	48.8	14,400	24	18.7	6,350
235	48.0	14,400	35	24.5	6,350
188	47.5	14,400	44	28.6	6,350
140	45.7	14,400	34	24.5	6,350
91	33.6	14,400	24	19.1	6,350
73	29.1	14,400	24	14.6	6,350

[a]Measurements made at 15 psia exit pressure and 60F inlet temperature
[b]Static pressure drop at no flow equals 46.5 inches H_2O

Table V

BURNOUT DATA FOR PARALLEL CHANNELS WITH 14.5 DEGREE INCLINATION[a]

Inlet Temp., F	Total Mass Flow, lb_m/hr	Burnout Heat Flux, Btu/(hr)(sq ft)	Inlet Temp., F	Total Mass Flow, lb_m/hr	Burnout Heat Flux, Btu/(hr)(sq ft)
77	66[b]	19,100	60	128[b]	36,700
70	25[b]	13,000	60	128[b]	36,700
68	86[b]	24,600	60	142[b]	36,700
68	120[b]	26,100	60	128[b]	36,700
68	114[b]	26,100	184	160[b]	55,500
68	110[b]	26,100	184	146[b]	55,500
61	52[b]	17,200	184	156[b]	55,500
61	32[b]	13,000	60	201[b]	55,500
--	zero[c]	8,800	60	210[b]	55,500
178	65[b]	17,800	60	210[b]	55,500
178	70[b]	17,800	75	130[d]	10,200
--	zero[c]	8,800	71	61[d]	5,330
193	119[b]	27,600	70	155[d]	13,000
195	119[b]	27,600	75	122[d]	10,500
195	114[b]	27,600	75	170[d]	15,350
195	136[b]	38,900	75	195[d]	17,900
195	137[b]	38,900	75	215[d]	21,100
190	146[b]	55,500	160	150[d]	9,700
190	146[b]	55,500	140	255[d]	12,300
60	201[b]	55,500	140	130[d]	8,900
60	203[b]	55,500	160	113[d]	7,980
60	204[b]	55,500	160	390[d]	15,350
60	119[b]	36,700			

[a]Measurements made at 15 psia exit pressure
[b]Bottom channel heated
[c]No burnout. Natural circulation boiling within channel
[d]Both channels heated - channel width changed from 2.75 in. to 2.25 in.

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VI. DISCUSSION

A. Burnout

The values of exit quality by weight at burnout were calculated as follows:

From a heat balance

$$Q = Wc_p\Delta t_s + Wxh_{fg} \tag{1}$$

or the exit quality is

$$x = \frac{1}{h_{fg}}\left[\frac{Q}{W} - c_p\Delta t_s\right] \tag{2}$$

The estimated accuracy may be found by taking differences in Eq. 1 and assuming no error in c_p or h_{fg}.

$$\Delta Q = Wc_p\Delta(\Delta t_s) + c_p\Delta t_s\Delta W + Wh_{fg}\Delta x + xh_{fg}\Delta W \tag{3}$$

Combination of Eqs. 1 and 3 gives an expression for the error in exit quality

$$\frac{\Delta x}{x} = +\left[\frac{c_p\Delta t_s}{xh_{fg}} + 1\right]\left[\left|\frac{\Delta Q}{Q}\right| + \left|\frac{\Delta W}{W}\right|\right] + \left|\frac{c_p\Delta(\Delta t_s)}{xh_{fg}}\right| \tag{4}$$

For these tests

$$Q = 3.41\,EI \tag{5}$$

and

$$\left|\frac{\Delta Q}{Q}\right| = \left|\frac{\Delta E}{E}\right| + \left|\frac{\Delta I}{I}\right| \tag{6}$$

so that

$$\frac{\Delta x}{x} = +\left[\frac{c_p\Delta t_s}{xh_{fg}} + 1\right]\left[\left|\frac{\Delta E}{E}\right| + \left|\frac{\Delta I}{I}\right| + \left|\frac{\Delta W}{W}\right|\right] + \left|\frac{c_p\Delta(\Delta t_s)}{xh_{fg}}\right| \tag{7}$$

It was approximated from the measurements that

$$\frac{\Delta E}{E} = 0.02$$

$$\frac{\Delta I}{I} = 0.02$$

$$\frac{\Delta W}{W} = 0.02$$

$$\Delta (\Delta t_s) = 5F$$

For water at 14.7 psia and 212F, c_p is 1.01 Btu/(lb)(F), and h_{fg} is 970 Btu/lb. The inlet subcooling for these tests was approximately 150F, and the exit quality ranged from 0.8 to 1.0. The maximum error in exit quality is then found from Eq. 7 to be

$$\frac{\Delta x}{x} \leq 0.07$$

1. Single Vertical Channels

Burnout data for a single heated vertical channel from Table I are plotted in Fig. 5, as a function of quality. The scatter of the data is about ± 0.20, indicating that burnout is likely to occur over a given range of exit quality rather than at a specific value of exit quality.

It is unlikely that burnout occurred under conditions of superheat at the channel exit. The values of exit quality greater than 1.0 should not be interpreted as burnout points but rather as an indication that burnout occurred when all of the water at the exit was evaporated.

2. Single Channel at 14.5 Degree Inclination

Burnout data from Table II are shown in Fig. 6. Burnout did not occur at the channel exit but at some point along the tube. The position at which the overheating occurred for the rectangular channel in the inclined position was not determinable with the instrumentation available. If the burnout condition depends solely upon the absence of water as indicated from the data of Fig. 5 for a vertical channel, then separation of flow must have occurred in the inclined tests. Evidence of this separation is shown in Fig. 7, in which overheating is shown to have occurred along the upper edge of the channel. Since flow separation probably is a function of velocity, the data in Table II were correlated by means of the following equation,

$$q''_{B.O.} \cong (10)^5 V \tag{8}$$

which was independent of the inlet water subcooling.

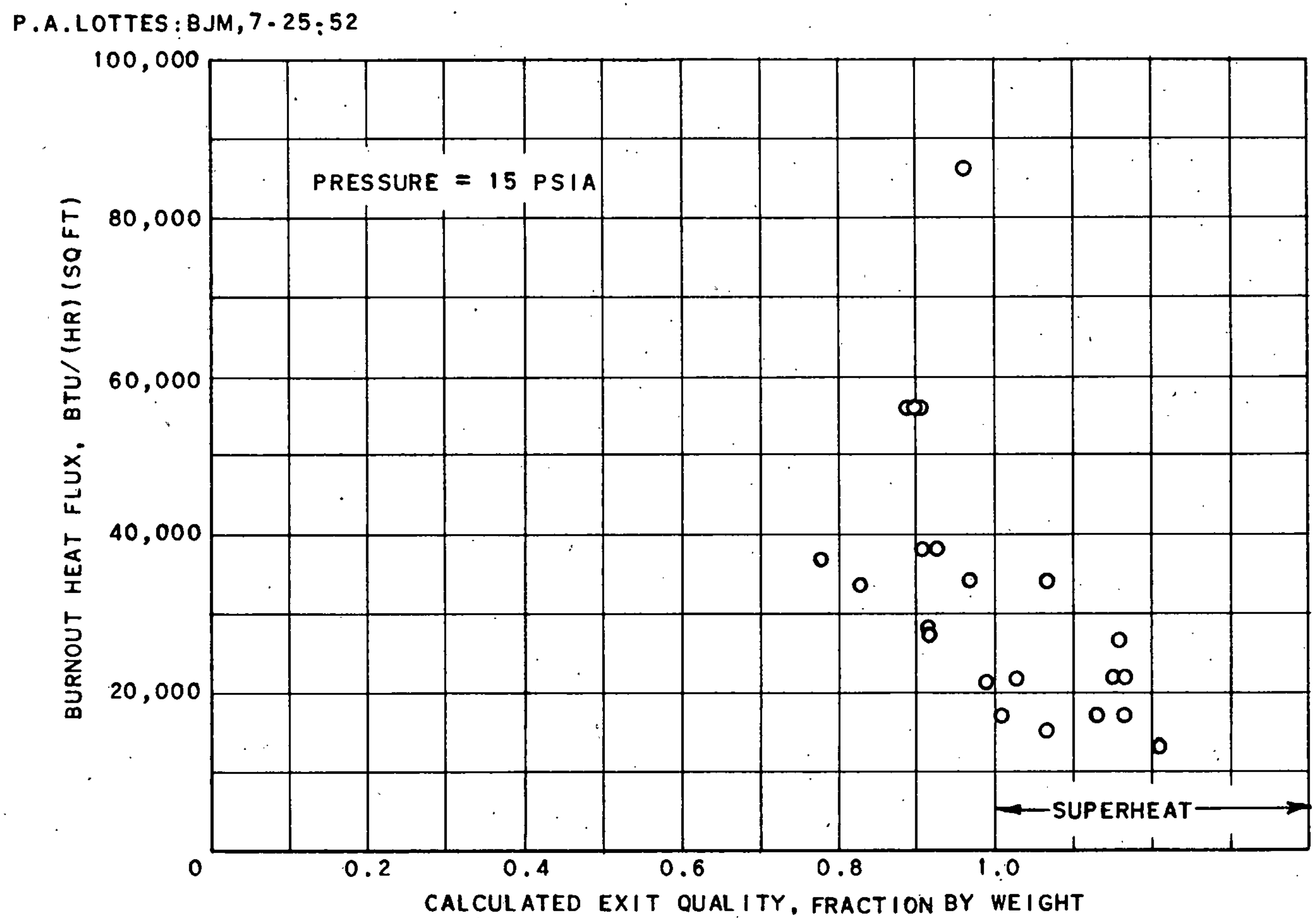

FIG. 5
BURNOUT DATA FOR A SINGLE HEATED VERTICAL CHANNEL

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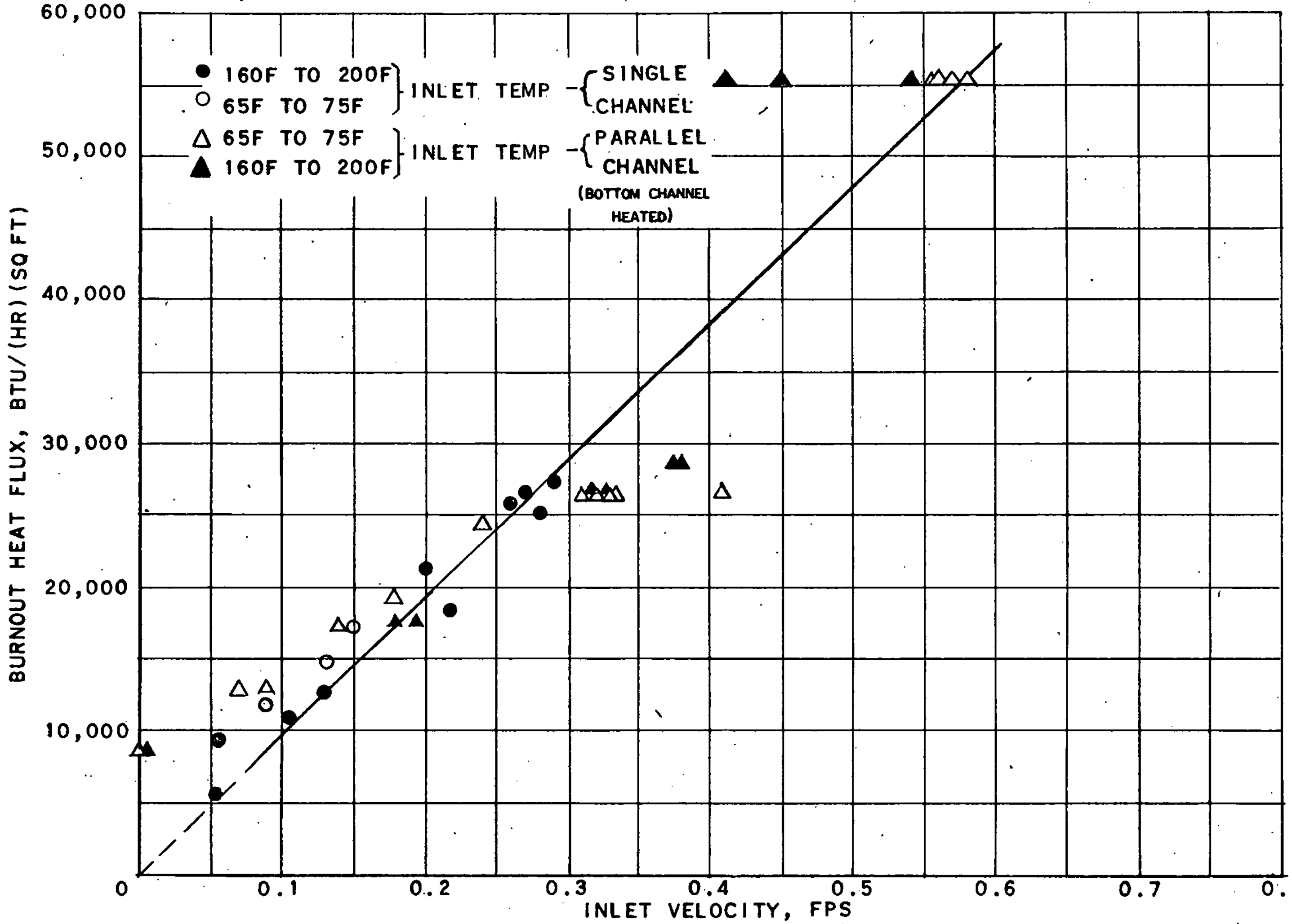

FIG. 6
BURNOUT DATA FOR SINGLE HEATED CHANNELS
WITH 14.5 DEGREE INCLINATION

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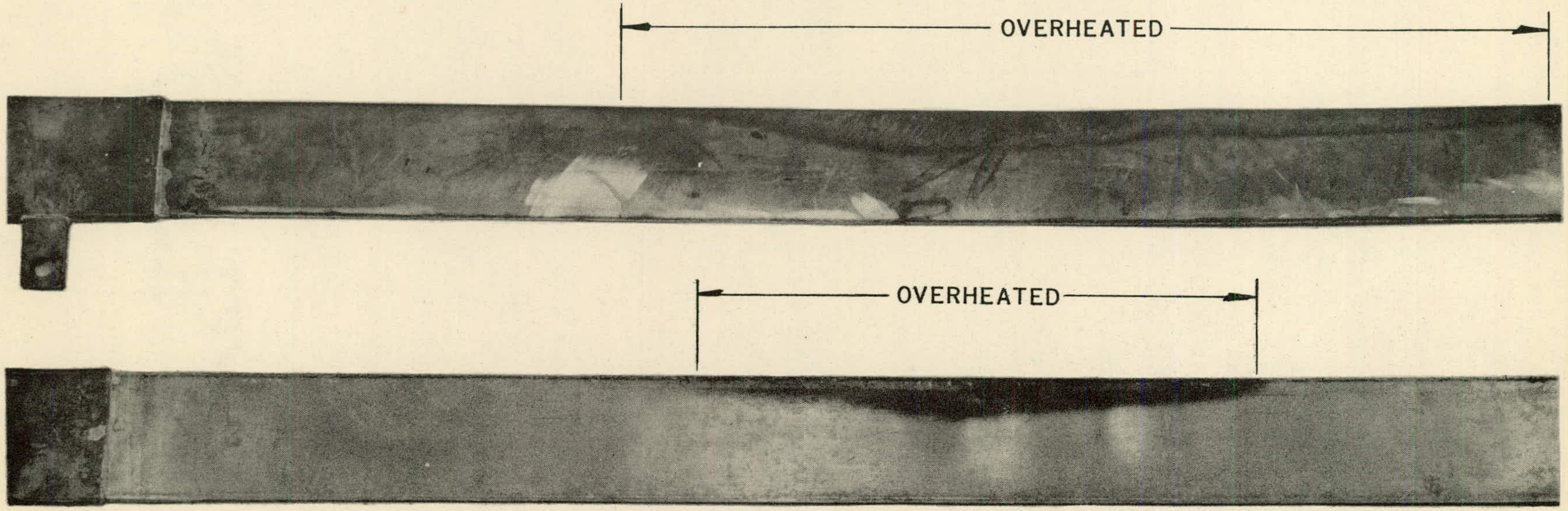

FIG. 7
AREAS OF BURNOUT ALONG UPPER EDGE
OF INCLINED FLOW CHANNELS

The data in Table V for parallel flow channels with the bottom channel heated are also shown in Fig. 6. These data match the single flow channel data since all the flow was diverted to the heated channel. A more detailed explanation of this flow phenomenon is given in the section on parallel channel effects.

3. Single Vertical Pipe

The effect of length-diameter ratio was studied by testing a 0.94 inch ID pipe. Data from Table III are plotted in Fig. 8. There was no apparent effect of heat flux on the value of exit quality at burnout over the range of variables tested; a result which might have been concluded from the information in Fig. 5. There was an effect of length-diameter ratio on the value of exit quality at burnout. Apparently the water had less chance of cooling the pipe wall as the heated length was decreased; i.e., there was less cooling per pound of water flowing since the length was reduced.

If the data from Fig. 8 are extrapolated to a length-diameter ratio of 250 corresponding to the rectangular channel, the extrapolation would show a value of exit quality of 0.9 to 1.0, which is in agreement with the data of Fig. 5.

B. Two-Phase Pressure Drop

Two-phase pressure drop data for the vertical 0.087 by 2.25 by 43-inch high rectangular cross section channel for heat fluxes from 6,350 to 51,500 Btu/(hr)(sq ft) are listed in Table IV and shown in Figs. 9 through 17. Each figure shows the experimental two-phase pressure drop curve and a corresponding isothermal pressure drop curve. The inlet subcooling for all of these tests was approximately 150F. The condition of sonic flow at the exit was not investigated in these tests.

The general equation for two-phase pressure drop without slippage is:

$$-dP = \frac{f\rho V^2}{2g_c D_e}\, dL + d\left(\frac{\rho V^2}{2g_c}\right) + \frac{g}{g_c}\rho\, dz \tag{9}$$

or

$$-dP - \frac{fG^2}{2g_c\rho D_e}\, dL + d\left(\frac{G^2}{2\rho g_c}\right) + \frac{g}{g_c}\rho\, dz \tag{10}$$

The over-all static pressure drop is then:

$$P_1 - P_2 = \frac{G^2}{2g_c D_e}\int_1^2 \frac{f dL}{\rho} + \frac{G^2}{g_c}\left(\frac{1}{\rho_2} - \frac{1}{\rho_1}\right) + \frac{g}{g_c}\int_1^2 \rho\, dz \tag{11}$$

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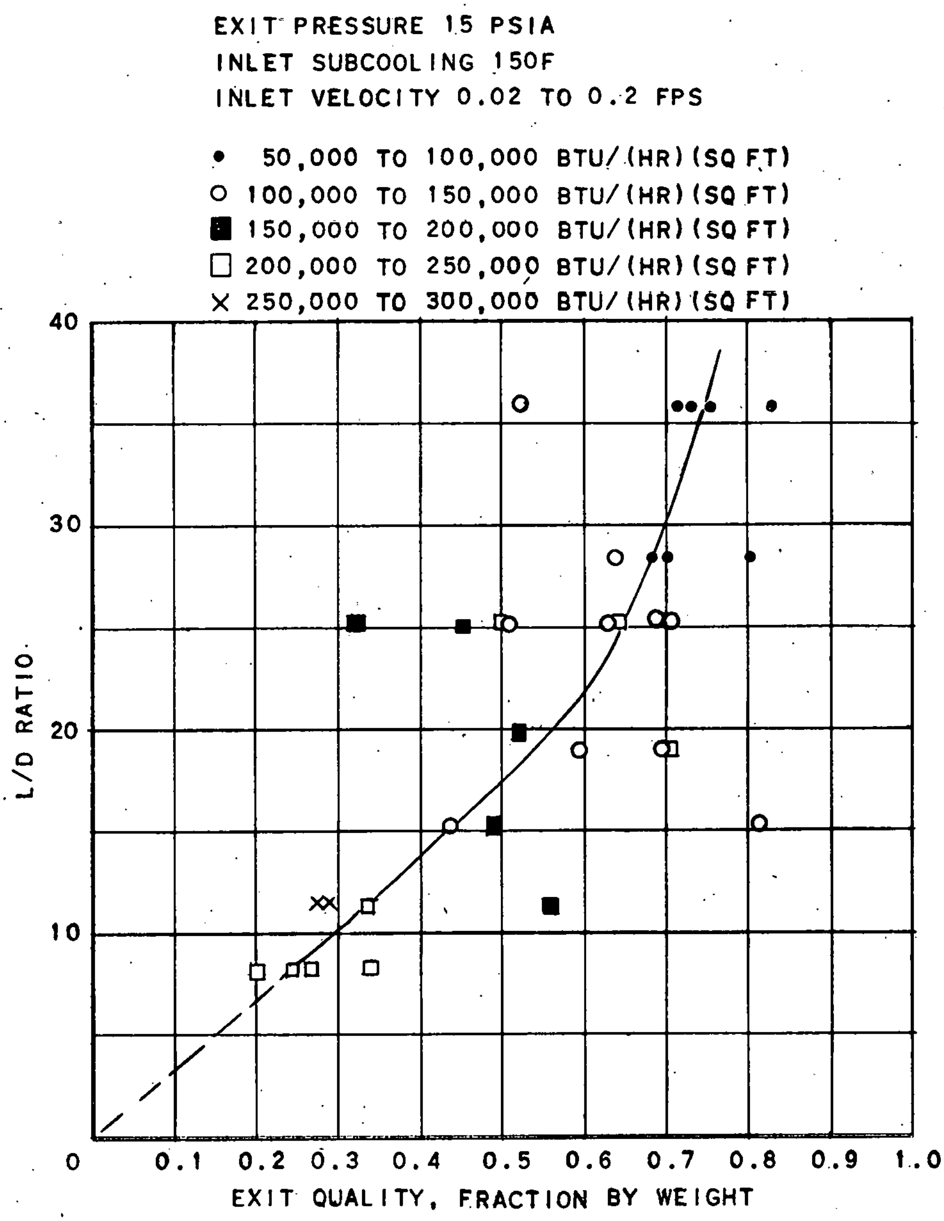

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FIG. 8
EFFECT OF L/D RATIO ON THE VALUE OF EXIT QUALITY AT BURNOUT FOR A 0.94-IN. ID VERTICAL TUBE-UPFLOW OF WATER

P.A.LOTTES:BJM,7-25-52

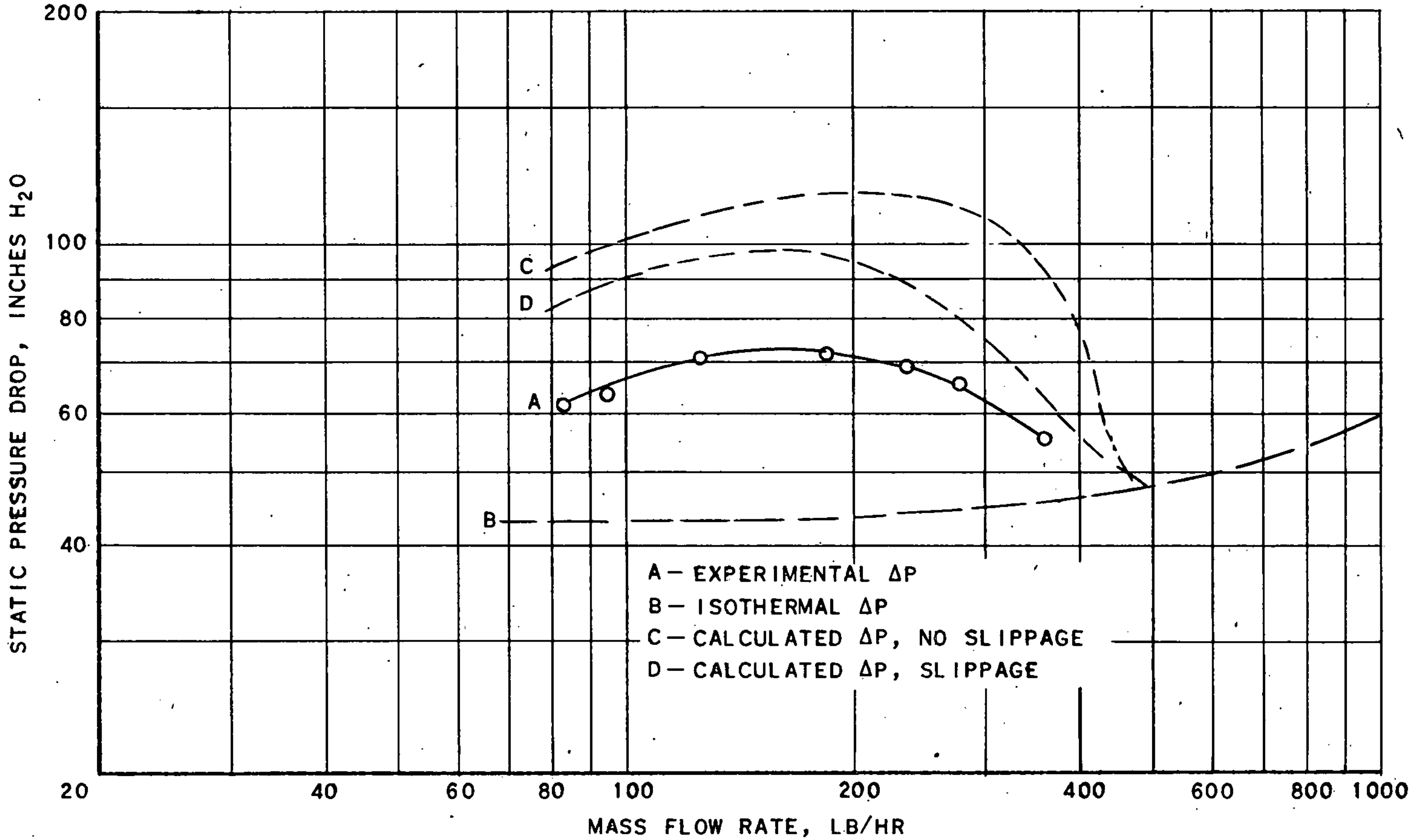

FIG. 9
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 51,500 BTU/(HR)(SQ FT)

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P.A.LOTTES:BJM,7-24-52

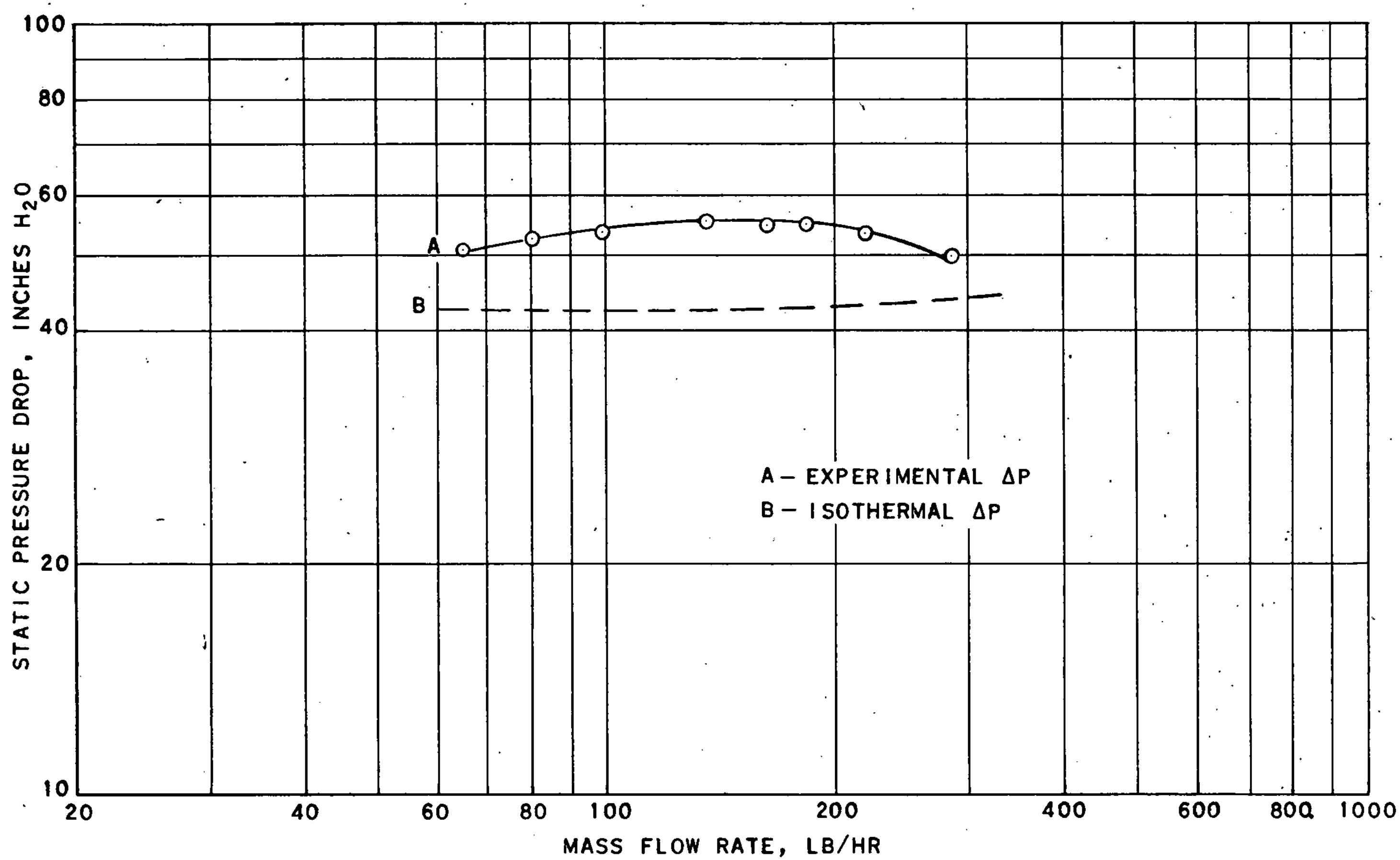

FIG. 10
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 47,500 BTU/(HR)(SQ FT)

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P.A.LOTTES:BJM,7-24-52

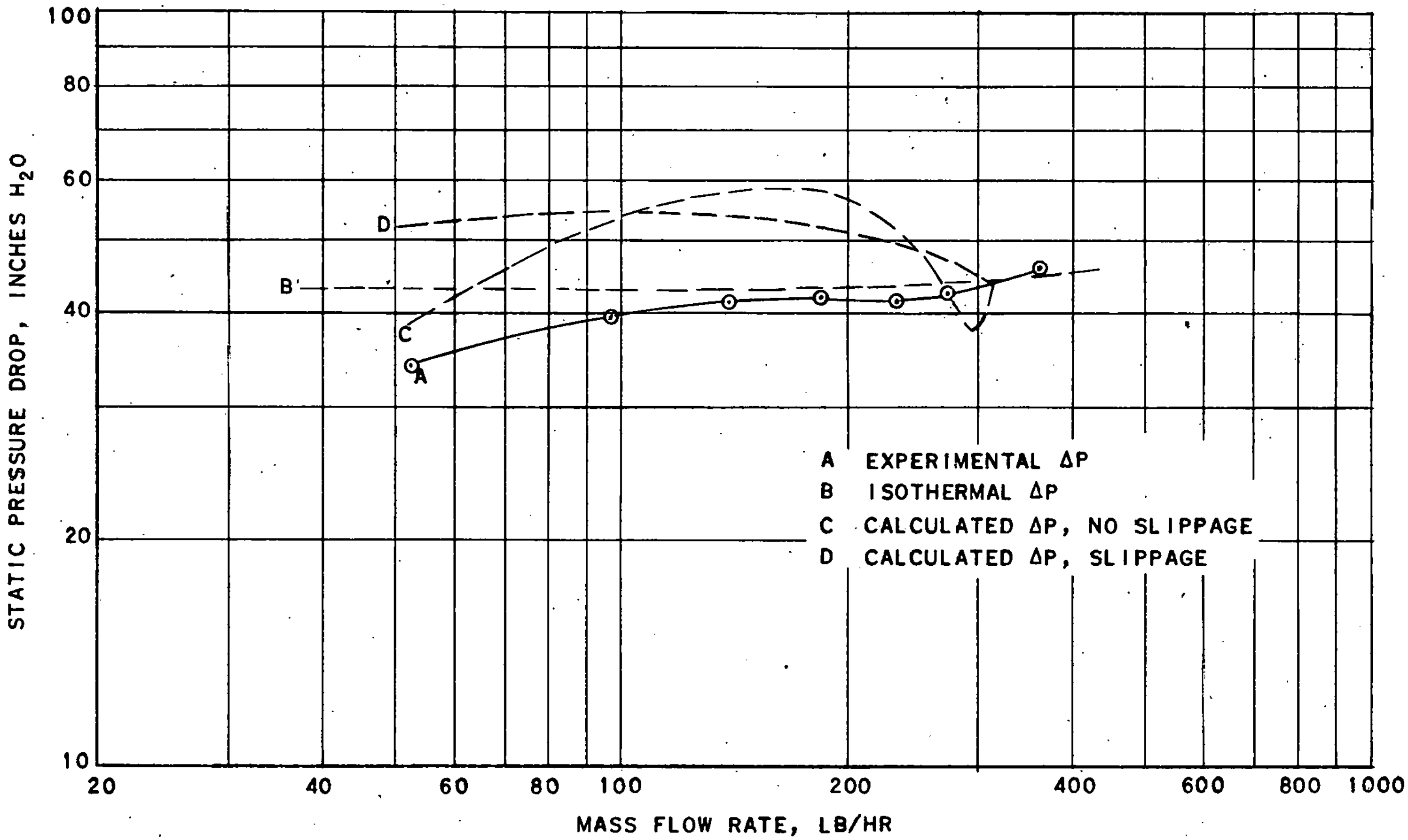

FIG. 11
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 33,500 BTU/(HR)(SQ FT)

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P.A.LOTTES:JH,7-25-52

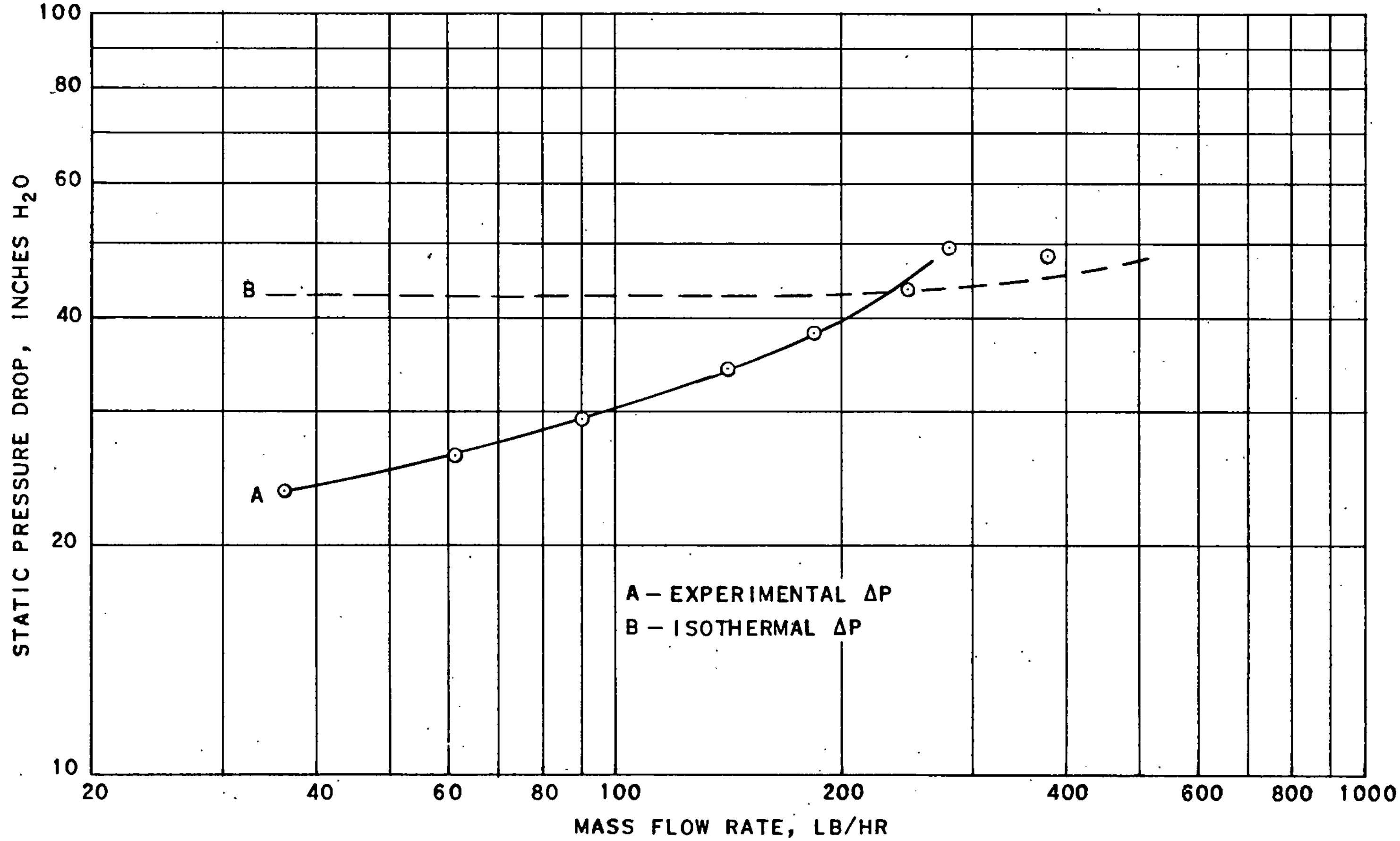

FIG. 12
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE
FOR SINGLE HEATED VERTICAL CHANNEL WITH
CONSTANT HEAT INPUT OF 25,600 BTU/(HR)(SQ FT)

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P.A.LOTTES:ES,7-25-52

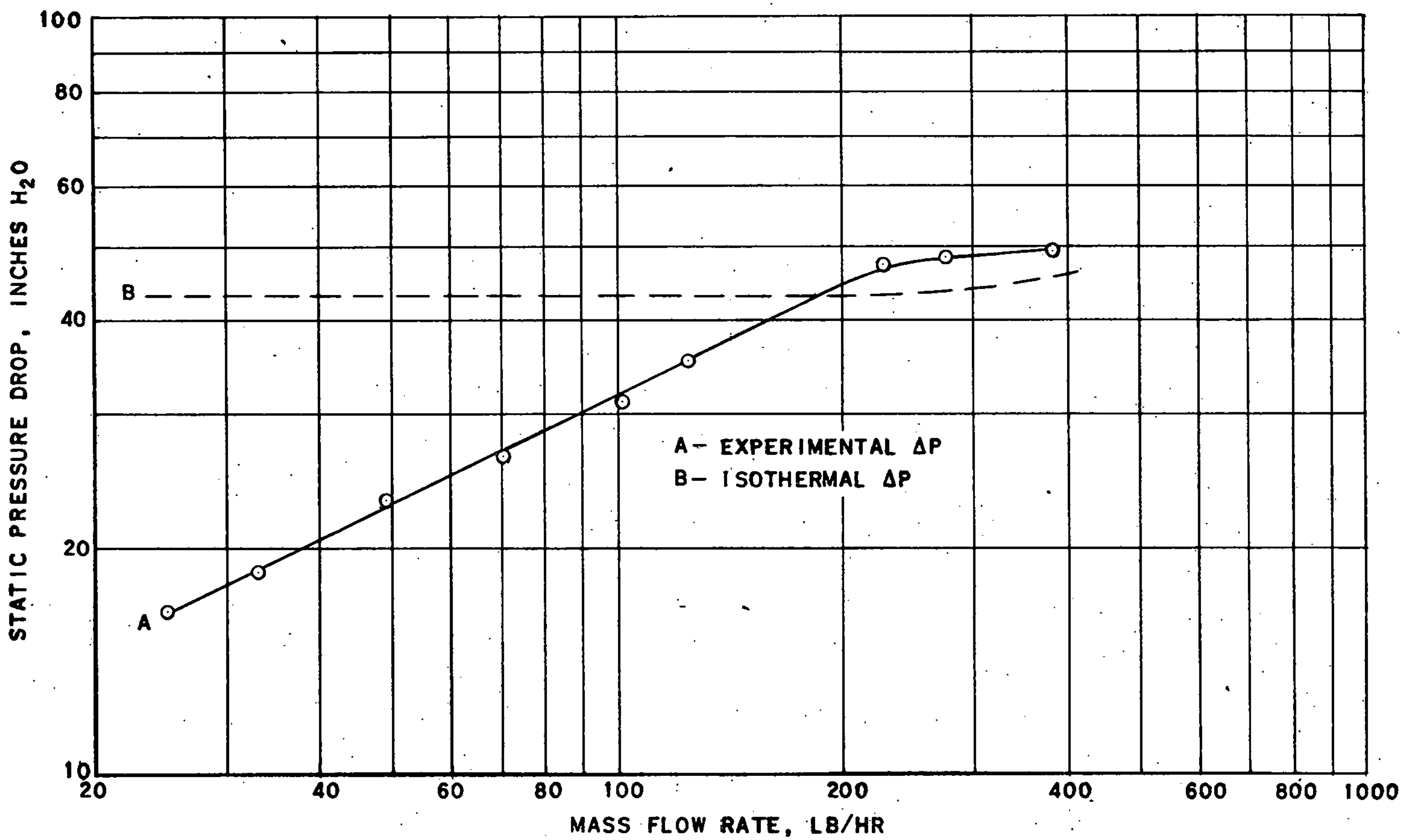

FIG. 13
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE
FOR SINGLE HEATED VERTICAL CHANNEL WITH
CONSTANT HEAT INPUT OF 19,800 BTU/(HR)(SQ FT)

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P.A.LOTTES:ES,7-25-52

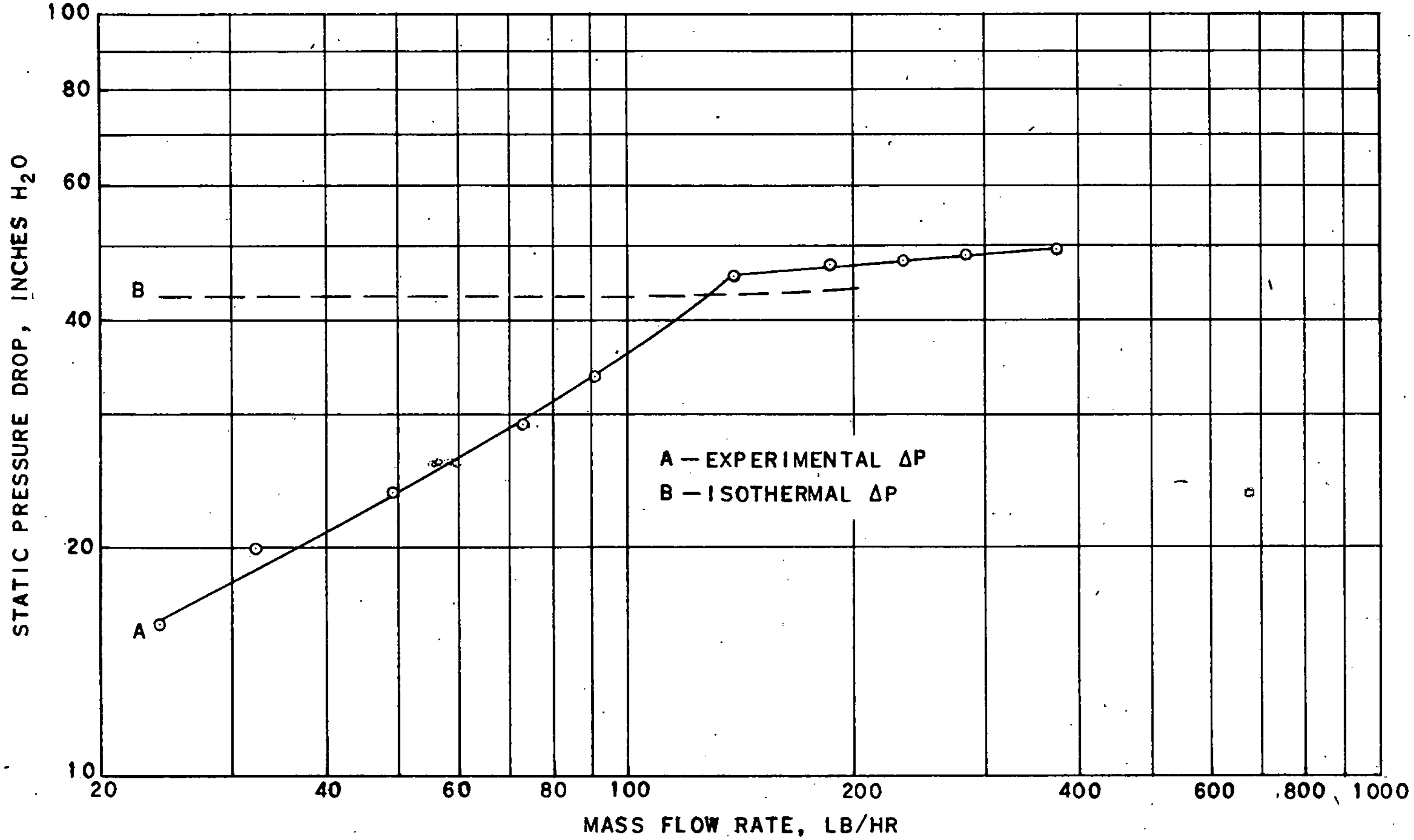

FIG. 14
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 14,400 BTU/(HR)(SQ FT)

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P.A.LOTTES:ES.7-25-52

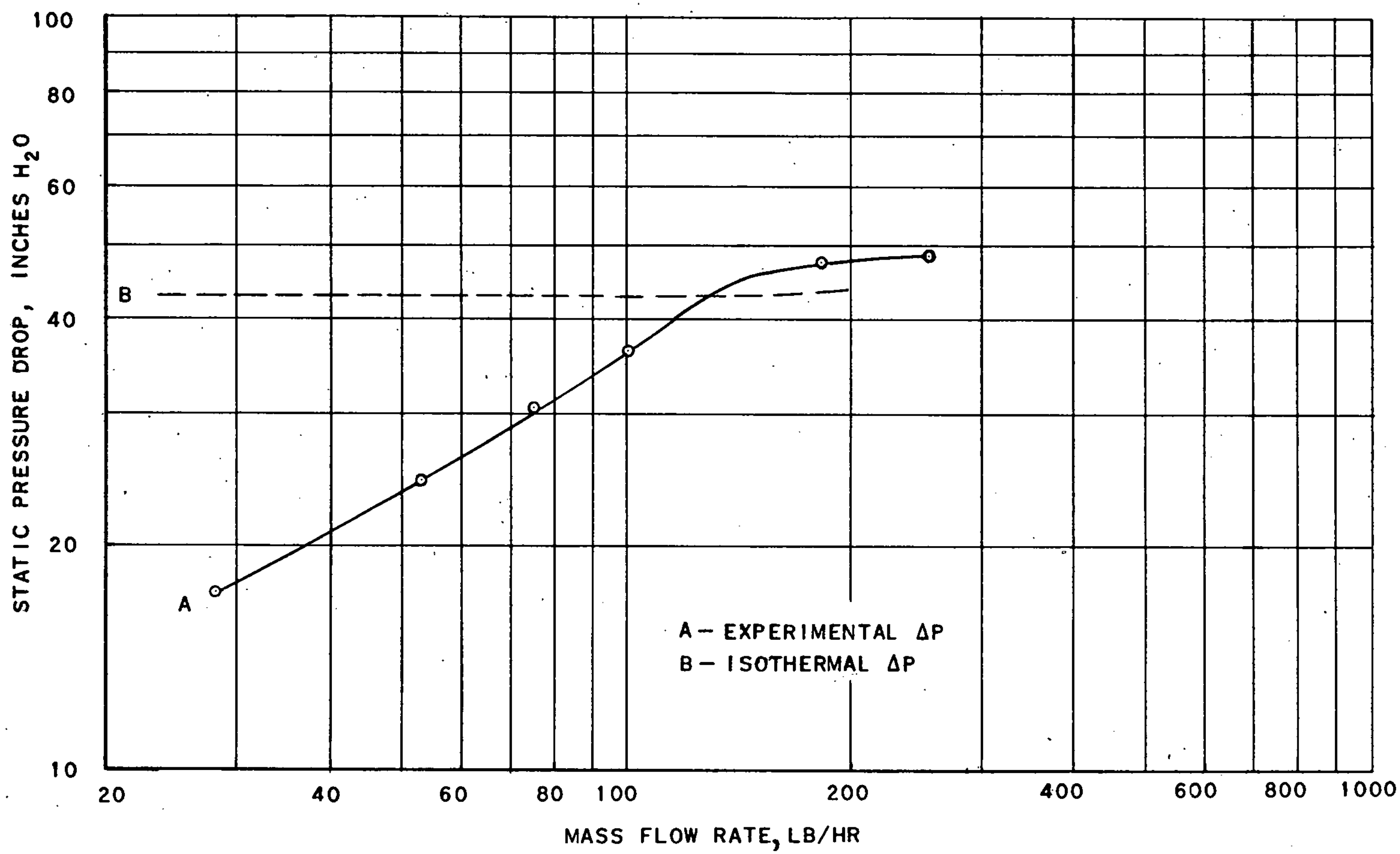

FIG. 15
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 13,500 BTU/(HR)(SQ FT)

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P.A.LOTTES:ES.7-25-52

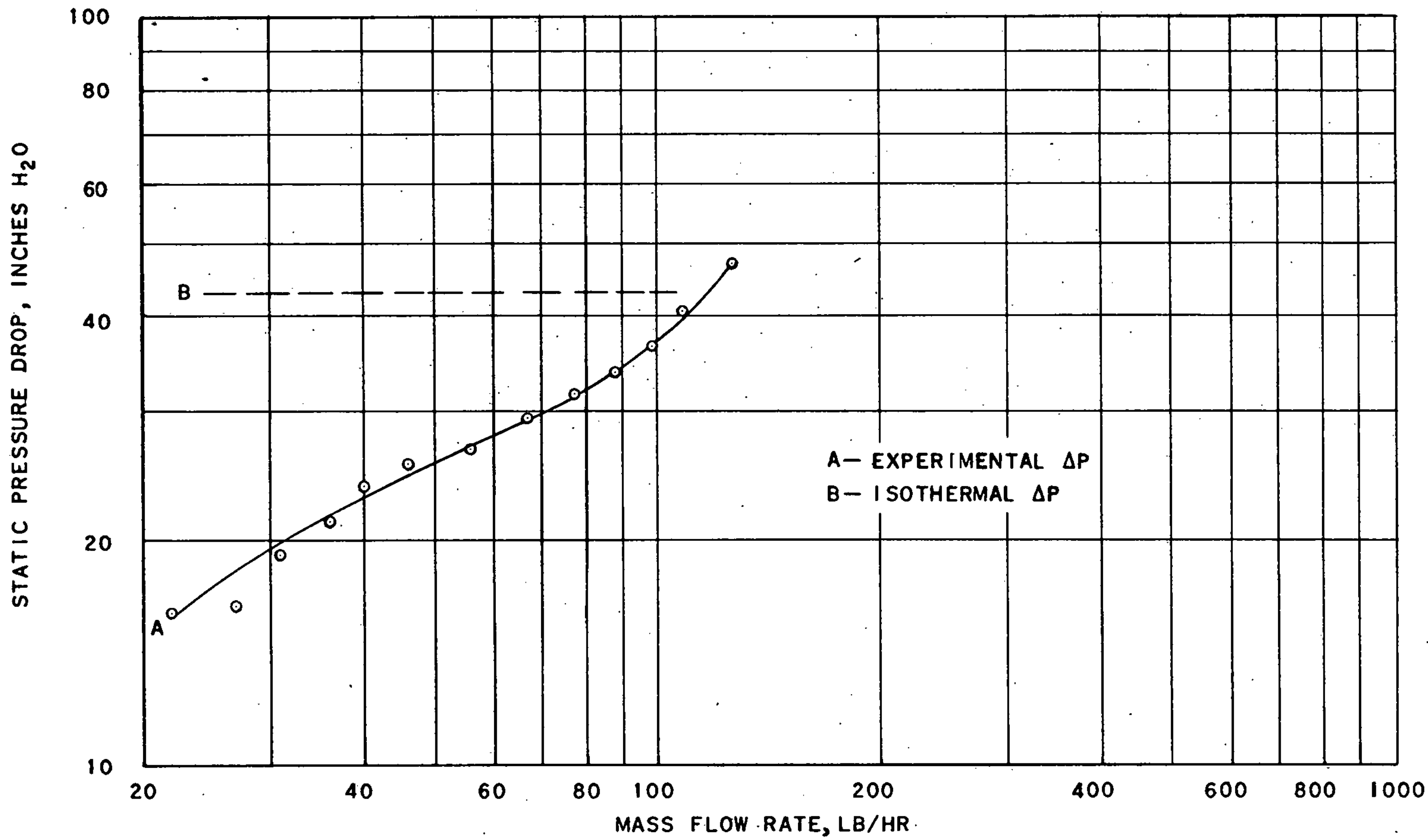

FIG. 16
TWO-PHASE PRESSURE DROP VS. MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 11,900 BTU/(HR)(SQ FT)

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P.A.LOTTES:BJM,7-24-52

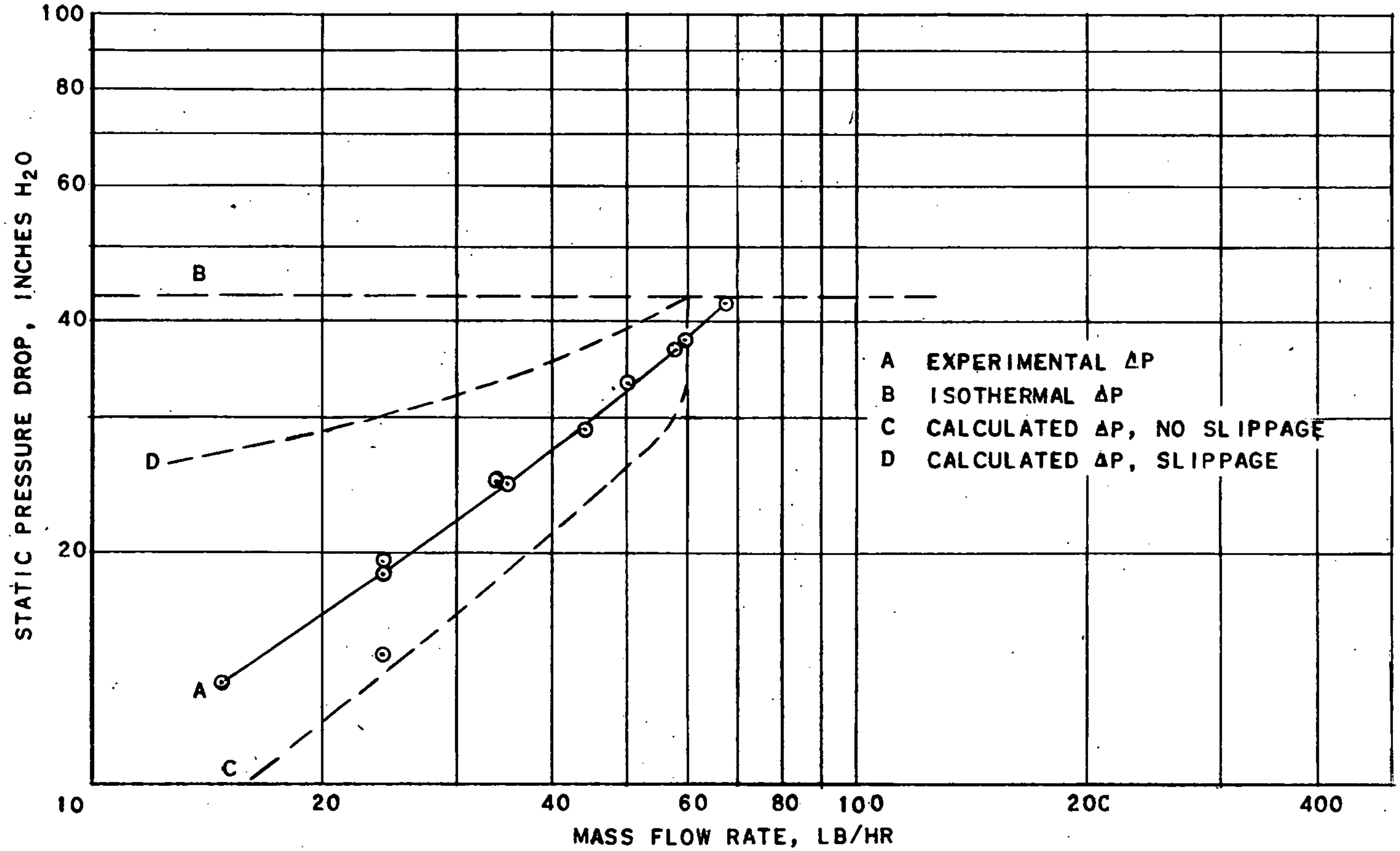

FIG. 17
TWO-PHASE PRESSURE DROP VS MASS FLOW RATE FOR SINGLE HEATED VERTICAL CHANNEL WITH CONSTANT HEAT INPUT OF 6,350 BTU/(HR)(SQ FT)

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The first two terms on the right-hand side of Eq. 11 are the frictional pressure drop and the momentum pressure drop, respectively. The last term is the change in hydrostatic head in the vertical direction "z." For vertical two-phase flow upward with constant heat generation along a tube, saturated water at the tube inlet, and equal steam and water velocities at any cross section (no slippage) with exit quality of x:

$$\int_1^2 \rho\, dz = \frac{z_2 - z_1}{x v_{fg}} \log_e \left[1 + \frac{x v_{fg}}{v_f} \right] \qquad (12)$$

For a vertical tube with negligible water velocities and a steady, uniform slippage of bubbles at constant velocity up through the water, the density gradient is linear up the tube and

$$\int_1^2 \rho\, dz = \left(\rho_f - \frac{\rho_{fg}}{2} \right) \left(z_2 - z_1 \right) \qquad (13)$$

Several two-phase pressure drop curves were calculated according to the method prescribed by Martinelli and Nelson[1] for two-phase flow through a horizontal tube of constant cross section, with the proper correction for hydrostatic head as given by Eqs. 12 and 13. Results of these calculations are compared in Figs. 9, 11, and 17. The hydrostatic head correction appears to be quite important at low heat fluxes, and the two-phase frictional pressure drop is important at the higher heat fluxes.

It is expected that the true hydrostatic head would lie between the values predicted by Eq. 12 and Eq. 13. Experimental evidence of this is shown in Fig. 17, where the hydrostatic head term is very large compared to the friction and acceleration terms. The calculated values bracket the experimental data.

The values of two-phase friction given by Martinelli and Nelson are apparently too high as indicated in Figs. 9 and 11. For these two values of heat flux, the hydrostatic head term was small compared to the friction and acceleration losses. The Martinelli-Nelson method is still useful when applied to vertical channels with the proper correction for hydrostatic head, since the calculated values were higher than the experimental values.

C. Parallel Channel Effects

The parallel flow problem is complicated in that various orientations of the channels with respect to the vertical will affect the flow distribution in parallel channels. This is especially true when the hydrostatic head becomes a sizeable amount compared to the friction and acceleration pressure losses. If the channels are in any position other than vertical, density differences between the inlet and exit headers will affect the flow distribution. In fact, the flow may actually reverse in some channels.

In order to demonstrate the effect of headers, consider the two channels A and B in Fig. 18.

The static pressure drop between points 1 and 2 through channel A is equal to the sum of the dynamic pressure drop and the hydrostatic pressure drop. The dynamic pressure drop is defined as the sum of the friction and acceleration losses.

$$P_1 - P_2 = \Delta P_A + \int_1^2 \rho_A \sin\theta \, dL \approx \Delta P_A + \bar{\rho}_A L \sin\theta \qquad (14)$$

where

$P_1 - P_2$ = total static pressure drop between points 1 and 2

P_A = dynamic pressure drop between points 1 and 2

ρ_A = density of fluid at any point in channel A

θ = angle of orientation with respect to the horizontal

L = length along channel

$\bar{\rho}_A$ = average density in channel A

If the headers are assumed large such that the velocities in the headers are small, the pressure drop between points 1 and 3 and between 2 and 4 will be largely hydrostatic pressure drop. Therefore

$$P_1 - P_3 \approx -d\rho_{in} \cos\theta \qquad (15)$$

$$P_3 - P_4 \approx \Delta P_B + \bar{\rho}_B L \sin\theta \qquad (16)$$

$$P_4 - P_2 \approx d\,\rho_{out} \cos\theta \qquad (17)$$

Adding Eqs. 15, 16, and 17 gives

$$P_1 - P_2 \approx \Delta P_B + \bar{\rho}_B L \sin\theta + d \cos\theta \qquad (18)$$

Equating 14 and 18

$$\Delta P_A + \bar{\rho}_A L \sin\theta \approx \Delta P_B + \bar{\rho}_B L \sin\theta - d\cos\theta \left(\rho_{in} - \rho_{out} \right) \qquad (19)$$

This equation could also have been written as

$$(P_{in} - P_{out})_A = (P_{in} - P_{out})_B - d\cos\theta \left(\rho_{in} - \rho_{out} \right) \qquad (20)$$

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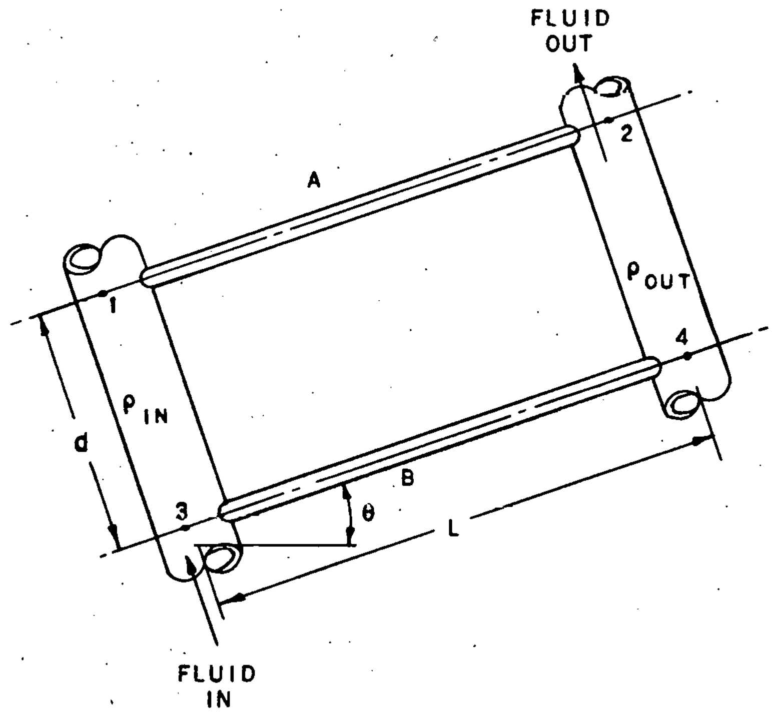

P.A.LOTTES:ES,10-27-52

FIG. 18
PARALLEL FLOW CHANNELS

Equation 20 states that channel A has less pressure drop than channel B by an amount equal to the difference in hydrostatic head of the inlet and exit headers. If both channel A and B receive the same amount of heat, channel A will operate at a lower flow rate than channel B. A critical condition might occur if θ were small, ρ_{out} were small in comparison to ρ_{in}, and d were large. Under these conditions the right side of Eq. 20 could become negative, causing a reversal of flow in channel A with resulting burnout of channel A.

In the experiments with flow in the parallel channels several effects were observed in addition to the numerical data recorded in Table IV. When only the top channel was heated, the level of water in the exit header fell to a level below the baffle as shown in Fig. 2, and the flow in the lower channel stopped. The flow recorded was therefore the flow in the top channel. When the lower channel was heated, the flow in the unheated top channel was stopped because the head available was not sufficient to cause flow in the top channel. The flow recorded was therefore the flow in the lower channel. Finally, with both channels uniformly heated, flow occurred in both channels. However, the flow did not divide uniformly, and in all cases the upper channel burned out first. The flow recorded was therefore the sum of the flow in the top and bottom channel. The flow in either the top or bottom channel, could not be calculated from the total flow. The experimental data indicated that a burnout would occur in the upper channel when steam is formed in the outlet header as a result of steam produced in the lower channel. The tests were not very extensive because it was soon recognized that the experimental behavior of the parallel channels was very dependent on the design of the outlet header. However, by making certain assumptions regarding the density in the outlet header, it was felt that the equations for parallel channels and the data from single channels could be used to predict the behavior of parallel channels.

D. Corrosion

Tap water from the laboratory supply was used in these tests. The pH of the water was 9.2 at room temperature. One test section was used for all of the single channel vertical burnout tests and all of the two-phase pressure drop tests. The estimated time of operation of this channel is about 18 hours.

At the completion of the tests, the channel was cut open in three places for inspection of the inside surfaces. Figs. 19, 20, and 21 show these surfaces at various positions along the channel. Fig. 19 shows a section where net boiling generally began. Fig. 20 shows a section that was always under conditions of vigorous boiling, and Fig. 21 shows a section near the top that was subject to overheating many times during the burnout tests.

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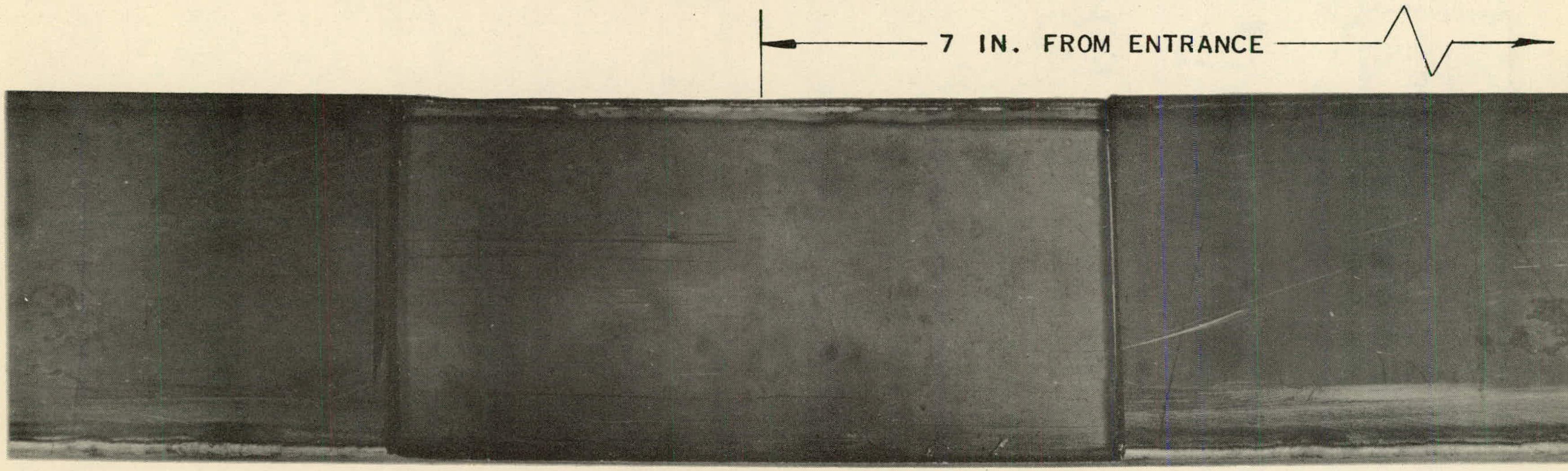

FIG. 19
INNER SURFACE OF TEST CHANNEL NEAR INLET

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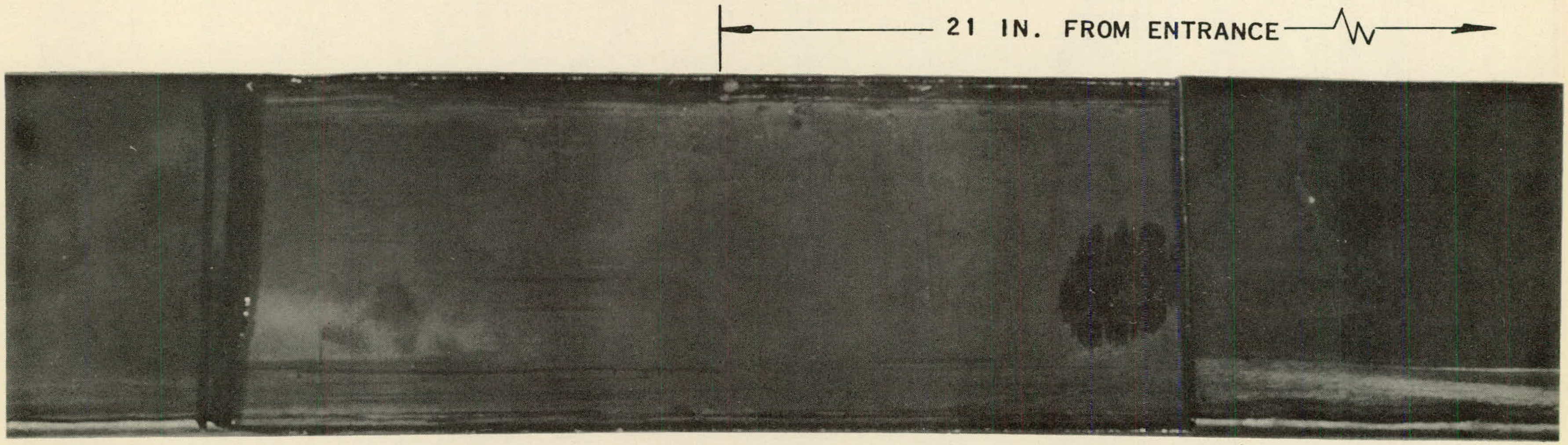

FIG. 20
INNER SURFACE OF TEST CHANNEL NEAR MIDDLE

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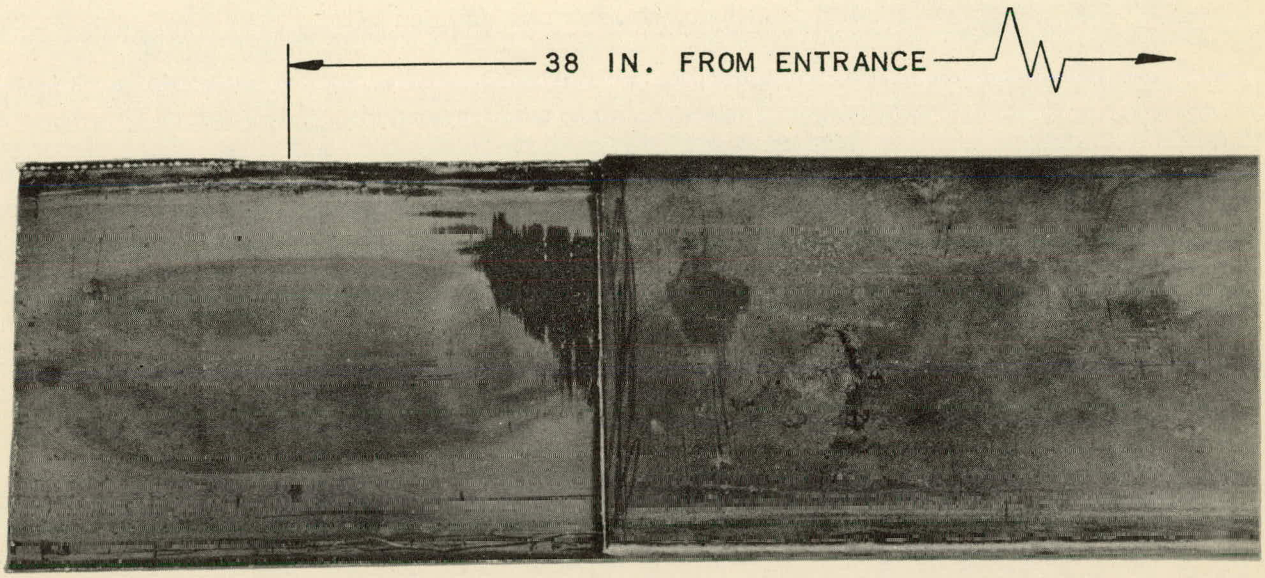

FIG. 21
INNER SURFACE OF TEST CHANNEL NEAR EXIT

394 042

These photographs indicate quite clearly the absence of scale or corrosion. It may be seen that the inside surfaces are cleaner than the exterior surfaces of the test section.

The lack of scale or products of corrosion confirms the data of UCLA;[5] these data indicated that treated water with a pH ranging from 9 to 10 would considerably reduce scaling or corrosion of metal surfaces in contact with the water.

VII. CONCLUSIONS

A. Burnout

Vertical channels or tubes with upflow of water within the tubes will not overheat or burn out until the value of exit quality ranges from 70 to 100 per cent for the range of variables listed. Extrapolation of the data to other ranges is not recommended.

Inside diameter	0.1 to 1 in.
Length-diameter ratio[(a)]	over 40
Heat flux	up to 300,000 Btu/(hr)(sq ft)
Inlet subcooling	up to 150F
Exit pressure	14.7 psia

No burnout predictions can be made for "inclined" tubes or channels other than the specific geometry tested in this experiment. Predications are not possible using the data obtained owing to the lack of information of the type of flow or the degree of separation of the vapor phase from the liquid phase.

B. Two-Phase Pressure Drop

The Martinelli-Nelson method of predicting two-phase friction and acceleration losses for the flow of a two-phase mixture through a horizontal pipe of constant flow area may be used to estimate losses in a vertical pipe or channel. The results will give maximum pressure loss to be expected due to friction and change in momentum of the fluid flowing.

The limits of the hydrostatic pressure loss may be evaluated from Eqs. 12 and 13. These losses together with the friction and momentum losses will give the maximum pressure loss over the range of variables tested in this experiment.

(a)The effect of length-diameter ratio is shown in Fig. 8.

C. Corrosion

The lack of scale or corrosion products observed during this experiment showed that use of water with an initial pH of 9.2 at room temperature reduced or eliminated corrosion.

VIII. REFERENCES

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2. Carl Gazley, Jr. and O. P. Bergelin, "Proposed Correlation of Data for Two-Phase, Two-Component Flow in Pipes," Chem. Eng. Progress, 45, 45-48 (1949).

3. W. H. McAdams, "Heat Transmission," McGraw-Hill Book Co., New York, 1942, pp. 325-333.

4. W. M. Rohsenow and J. A. Clark, "Local Boiling Heat Transfer to Water at Low Reynolds Numbers and High Pressure," Technical Report No. 4, Massachusetts Institute of Technology, DIC Project No. 6627, July 1, 1952.

5. H. Buchberg et al., "Final Report on Studies in Boiling Heat Transfer," University of California, Department of Engineering, Report COO-24, March, 1951.

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