CNLM-4358

ENGINEERING REPORT ON THE DESIGN OF THE PRATT & WHITNEY AIRCRAFT FORCED CONVECTION ALKALI METAL VAPORIZATION CONDENSATION HEAT TRANSFER RIG

PRATT & WHITNEY AIRCRAFT

DIVISION OF UNITED AIRCRAFT CORPORATION

CANEL

MIDDLETOWN • CONNECTICUT

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ENGINEERING REPORT ON THE DESIGN OF THE PRATT & WHITNEY AIRCRAFT FORCED CONVECTION ALKALI METAL VAPORIZATION CONDENSATION HEAT TRANSFER RIG

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UNCLASSIFIED

CLASSIFICATION

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January 14, 1963

DATE

1. SUMMARY

A test rig has been designed to obtain basic two phase forced convection liquid metal heat transfer and pressure drop data for use in the design of liquid metal boilers. This engineering report includes background of the boiling problem, partial summary of work done by other investigators, the object, scope, engineering, and operation of this test rig, and a summary of reproducibility tests conducted in a water back-up rig.

2. INTRODUCTION

The object of this test is to obtain fundamental design type information that can be applied towards the design of a once-through or partial vaporization boiler. The test rig can therefore be justifiably described as a basic data producing heat transfer test rig as opposed to a materials evaluation, endurance, or component performance rig. More specifically, local boiling heat transfer coefficients will be determined in a tube with vapor quality being the independent variable at several flow rates, heat fluxes, and for different tube geometries.

Briefly described, the potassium is pre-heated to almost the boiling temperature in a radiant preheater and then heated and vaporized to some quality in a "work horse" boiler. The potassium is then vaporized an additional incremental amount in the test section and temperature differences and heat input are measured to obtain the local boiling heat transfer coefficient. The potassium is then superheated in a third boiler and condensed and cooled in a finned tube condenser. Heat losses are controlled by guard heaters, temperature differences are measured by differential thermopiles, and flow rates are measured with a thermal and an electromagnetic flowmeter. Heat is transferred from Haynes alloy No. 25 clad tubular immersion heaters through sodium condensing on the outside of the boiler tube. A constant wall temperature is achieved by this technique.

The stability problem, an inevitable by product, will be grossly investigated by changing the "hardness" of the system with accumulators and by adjustment of throttle valves upstream of the boiler and downstream of the condenser.

The first boiler tube to be tested has the shape of a sine wave with a developed length of 10.35 inches. High temperature (>1000F) components are fabricated from Haynes alloy No. 25. All other components in contact with liquid metal are fabricated from type 316 stainless steel.

A table of expected operating conditions and expected rig capability is included on the next page.

Table 1

Projected Rig Capability

		Design Capability	Maximum Capability*
1.	Fluid Tested	Potassium	Potassium
2.	Flow rate (two-phase)	20 lb/hr	30 lb/hr
3.	Method of Heating	Condensing Na	Condensing Na
4.	Loop Materials	Haynes alloy No. 25 and stainless steel	Haynes alloy No. 25 and stainless steel
5.	Vapor Mach. No. (at 1700F)	0.2	0.3
6.	Flow Stabilizer Device	Hoke THY 442 Mod 1	Hoke THY 442 Mod 1
7.	Tube ID (Test Section)	0.186 inch	0.186 inch
8.	Type of Test Section and Length		
	a. Type 1	Modified Sine Wave (10.35 inches)	Modified Sine Wave (10.35 inches)
	b. Type 2	Hockey Stick (9 inches)	Hockey Stick (9 inches)
9.	Pump Type	E.M. (Conduction Type)	E.M. (Conduction Type)
	a. Available System Capacity	1.15 gpm	1.15 gpm
	b. Available Pump at 1.15 gpm	135 psi	135 psi
10.	Maximum Potassium Temp.	1700F	1750F
11.	Maximum Potassium Pressure	57 psia	68 psia
12.	Maximum Sodium Temp.	1800F	1850F
13.	Maximum Sodium Pressure	33 psia	38 psia
14.	Maximum Haynes alloy No. 25 Temp. (on heater surface at Q/A 100,000 Btu/hr ft^2)	1815F	1865F
15.	Boiling K Heat Flux		
	a. First Boiler, Btu/hr ft^2 (Δx = 100%)	6.2×10^4	4.5×10^5
	b. Second Boiler, Btu/hr ft^2 (Δx = 5%)	2.2×10^4	4.04×10^5
	c. Third Boiler, Btu/hr ft^2 (Δx = 35%)	5.1×10^4	3.89×10^5
16.	Heater Characteristics (First Boiler)		
	a. Heat Flux, Btu/hr ft^2	1.3×10^4	1×10^5
	b. Centerline Temp.	1846F	2180F
	c. Heater Length	296 inches	296 inches
	d. Heater Diameter OD	3/16 inch	3/16 inch

Table 1

Projected Rig Capability
(Continued)

	Design Capability	Maximum Capability*
17. Heater Characteristics (Second Boiler)		
a. Heat Flux, Btu/hr ft^2	5.24×10^3	1×10^5
b. Centerline Temp.	1804F	2180F
c. Heater Length	43.5 inches	43.5 inches
d. Heater Diameter OD	3/16 inch	3/16 inch
18. Heater Characteristics (Third Boiler)		
a. Heat Flux, Btu/hr ft^2	1.3×10^4	1×10^5
b. Centerline Temp.	1846F	2180F
c. Heater Length	188 inches	188 inches
d. Heater Diameter OD	3/16 inch	3/16 inch
19. Preheater Requirements		
a. Boiling Test (Radiant)	0.9 Kw	0.9 Kw
b. Calibration Test (I^2R)	7 Kw	18 Kw
20. Maximum Cooling Requirements	10 Kw	10 Kw
21. Immersion Heater Power		
a. First Boiler (Δx = 100%)	5.1 Kw	33 Kw+
b. Second Boiler (Δx - 5%)	0.255 Kw	5 Kw
c. Third Boiler (Δx = 35%)	1.8 Kw	20 Kw+

*Maximums do not necessarily correspond to the same flow condition.
+Power supply limited to a lower heater power.

FORCED CONVECTION ALKALI METAL VAPORIZATION CONDENSATION HEAT TRANSFER RIG

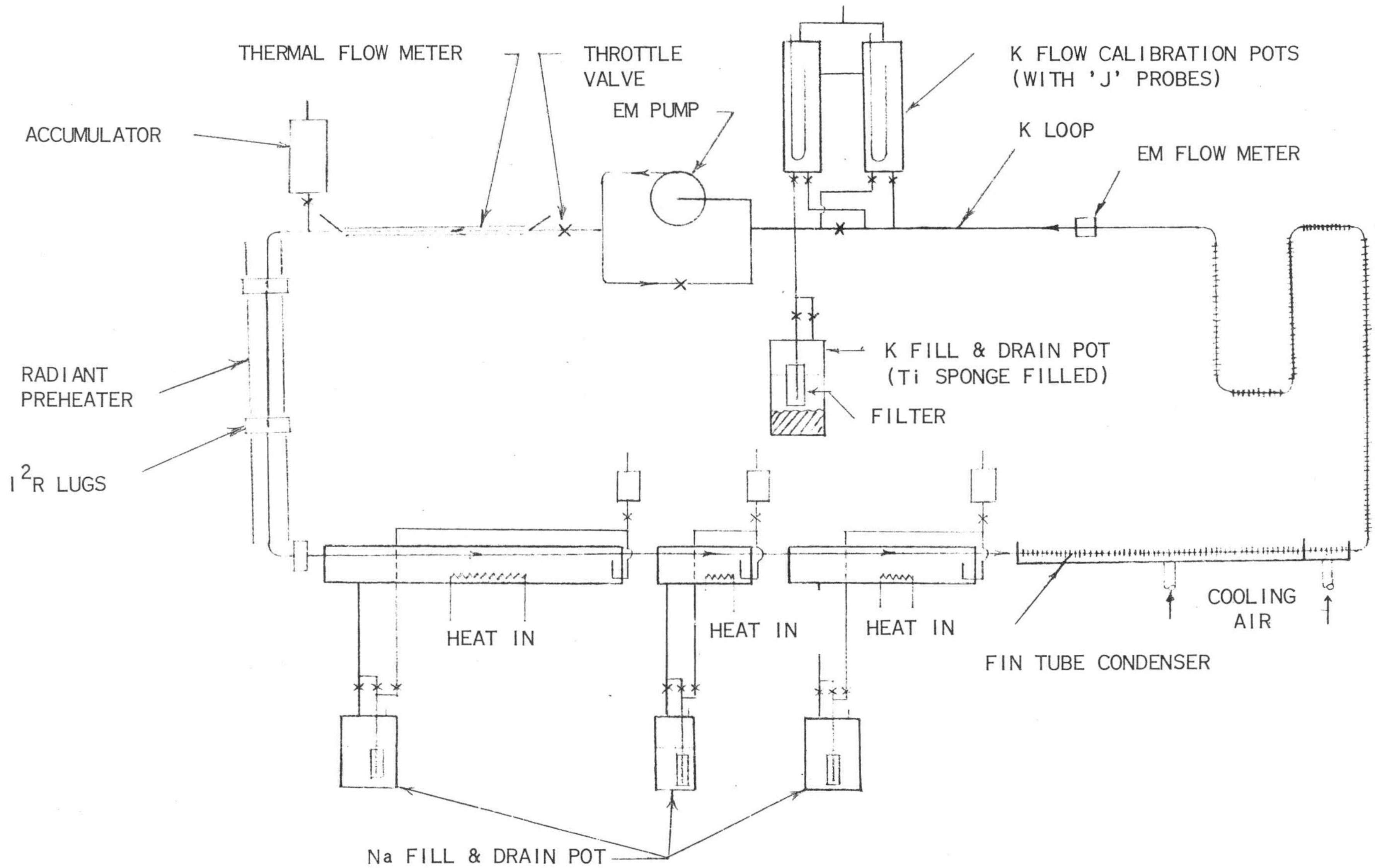

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FIG 1A

FORCED CONVECTION ALKALI METAL VAPORIZATION-CONDENSATION HEAT TRANSFER RIG

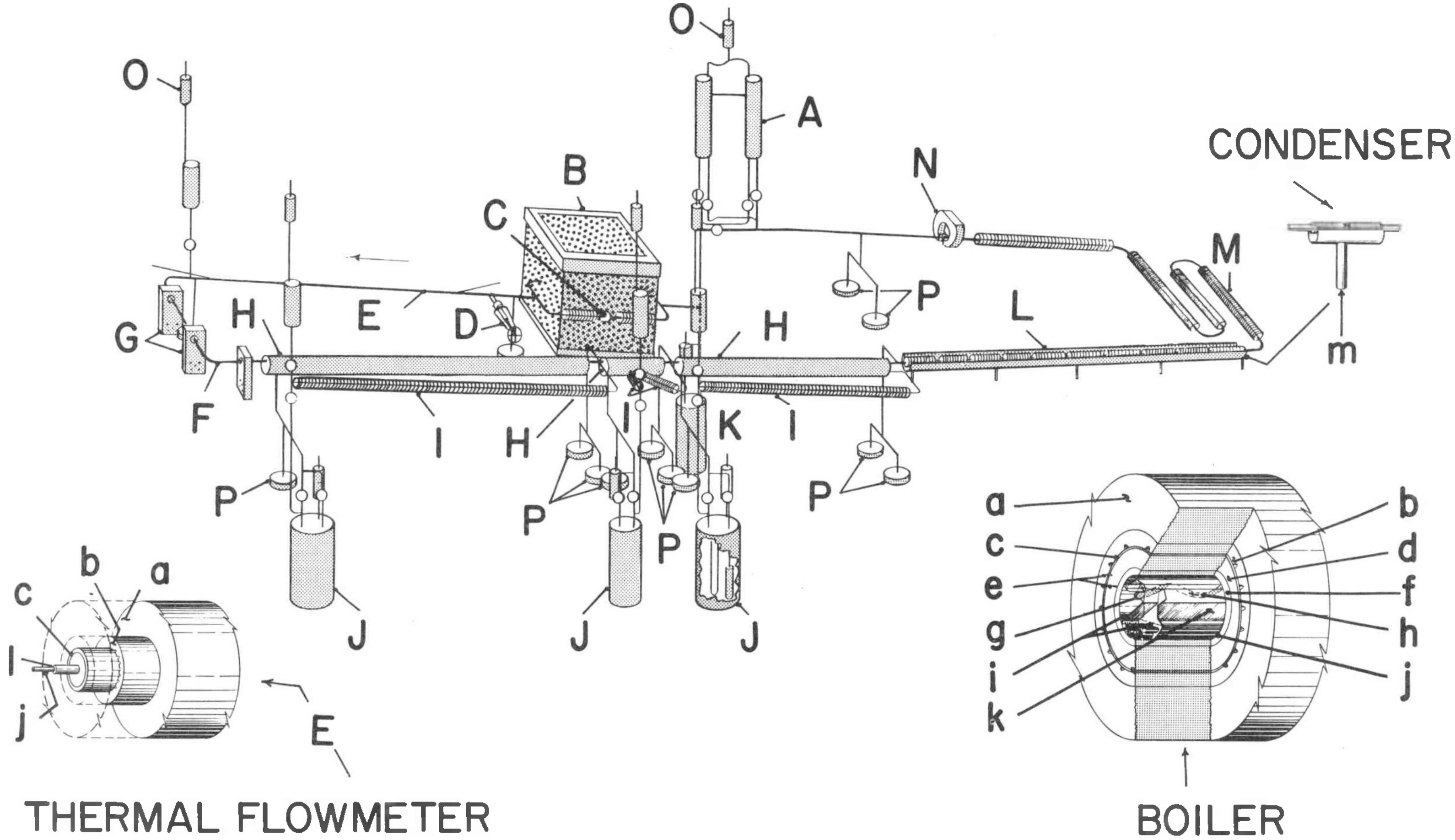

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FIG 1

FORCED CONVECTION ALKALI METAL VAPORIZATION-CONDENSATION HEAT TRANSFER RIG

A. VOLUMETRIC FLOW CALIBRATOR
B. E. M. PUMP
C. BYPASS VALVE
D. THROTTLE VALVE
E. THERMAL FLOW METER
F. RADIANT PREHEATER (LOW FLOW)
G. I^2R PREHEATER LUGS (HIGH FLOW)
H. BOILERS (3)
I. Na CONDENSER
J. Na FILL-DRAIN POT
K. POTASSIUM FILL-DRAIN POT
L. CONSTANT HEAT FLUX CONDENSER
M. NAT'L CONVECTION CONDENSER
N. E. M. FLOW METER
O. VAPOR TRAPS
P. PRESSURE TRANSMITTERS

a. INSULATION
b. CLAMSHELL HEATER
c. HEAT DISTRIBUTOR
d. INSULATION
e. THERMOCOUPLE
f. PRESSURE SHELL
g. TEST SPECIMEN
h. BOILING POTASSIUM
i. SODIUM LIQUID, SODIUM VAPOR
j. IMMERSION HEATER
k. BAFFLE
l. POTASSIUM LIQUID
m. COOLANT AIR

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FIG 1

3. FORCED CONVECTION VAPORIZATION PHENOMENA

When a subcooled liquid enters a heated tube, it experiences warming by forced convection. When this bulk fluid reaches a temperature somewhat below the saturation temperature, subcooled nucleate boiling at the heated surface may occur depending upon the temperature difference (t_W-t_S). A large increase in local heat transfer coefficient is experienced upon initiation of subcooled nucleate boiling due primarily to an increased mass transport mechanism. When the vapor enters regions of subcooled fluid in the fluid core, it collapses as quickly as the heat transfer and fluid properties allow. Some authors compare this collapse to cavitation. Finally when the temperature of the bulk fluid reaches the saturation temperature, net generation of vapor commences. Nucleate boiling continues to some quality depending upon fluid properties, flow, surface conditions, system, pressure and heat flux. At some point in the process depending upon fluid properties, heat flux, configuration, flow rate, etc., a partial and then complete blanket of vapor may cover the heated surface resulting in a drastic lowering of heat transfer coefficient. From the point a net generation of vapor commences to 100 percent quality, the temperature (saturation) of the fluid will decrease due to a frictional <u>and</u> momentum pressure drop: Fig 2 shows the relationship between pressure drop and temperature drop for saturated potassium at several temperature levels.

Heat transfer in forced convection boiling is predominently influenced by the type of flow regimes present. The flow regime may be any of at least seven defined regimes (Saturated, Wave, Slug, Annular, Froth, Plug, and Dispersed Flow). The Baker Chart (Fig 3) is an attempt to define these regimes. It may be noted that at a given flow rate, several regimes may exist in a single tube. As vaporization occurs, several complex and interacting phenomena occur:

a. Adjacent to the heated surface a thin layer of superheated liquid is formed in which bubbles nucleate and grow from some preferred spots on the surface. Additional vapor is generated by evaporization of superheated liquid which is adjacent to the wall and the vapor. Conventional boundary layer concepts are not meaningful under this condition. Heat transfer rate is greatly affected by the increased agitation and mass transfer during boiling.

b. Vapor is also formed by flashing of saturated liquid due to frictional <u>and</u> momentum pressure drop. The quality increase is insignificant however, even for large pressure drops.

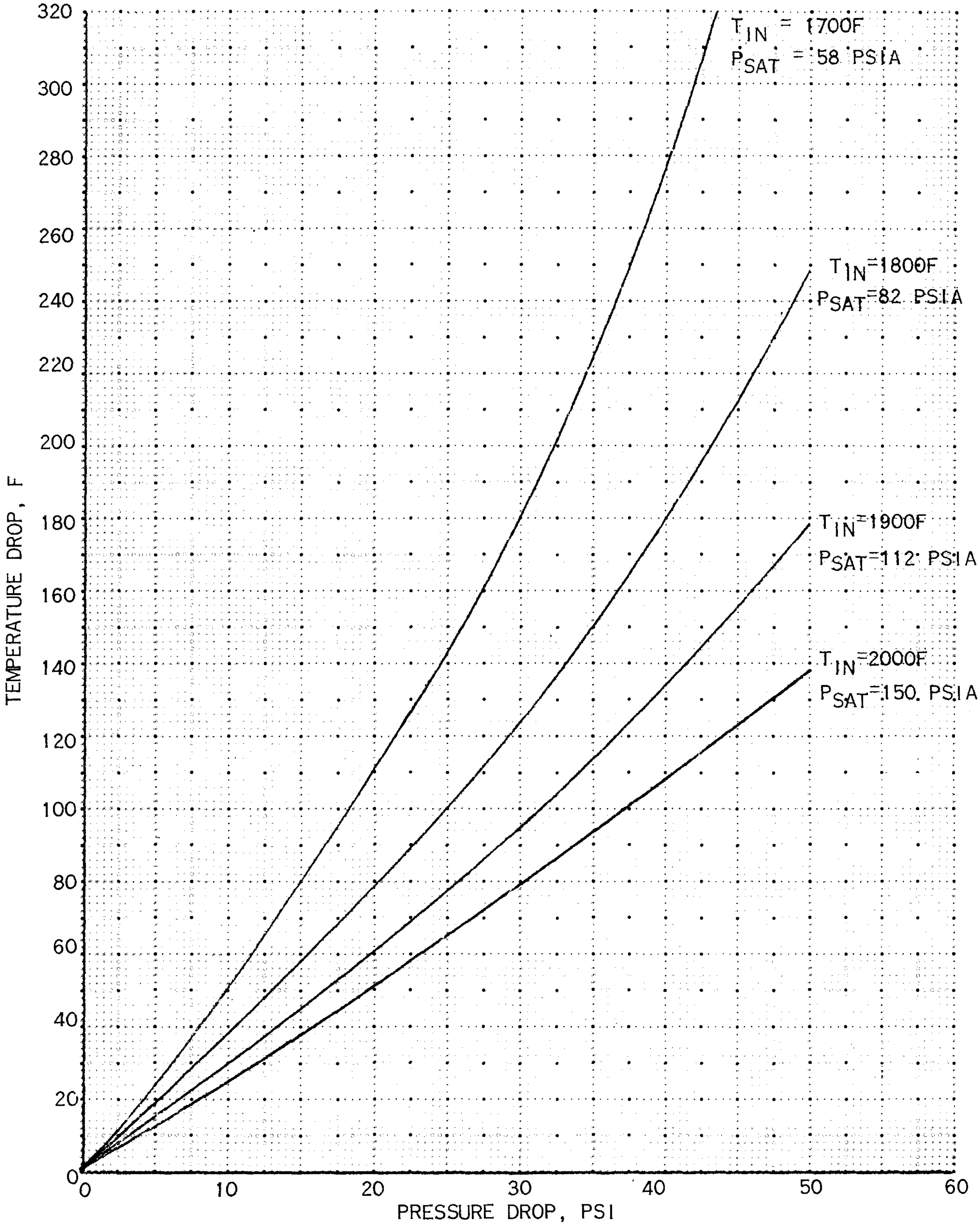
c. There are two phase pressure drops which are significantly larger than single phase pressure drops. The Martinelli and Lockart correlation however, correlates most data to $\pm$ 30 percent. It is believed that the disagreement may be due to the various heater materials, dissolved gasses, surface roughness and various flow regimes encountered by different investigators. Pressure drop is doubly important in boilers since <u>both pumping power and outlet temperature are affected</u>. Fig 2 shows that a pressure drop of 20 psi corresponds to a temperature drop of 80F for 1800F potassium.

d. When vapor is generated, the average velocity increases by the density ratio between the liquid and vapor (1:259 for potassium at 1800F). In a condenser, the momentum change results in a rise in static pressure unless the condensing length is very large and the frictional drop overcomes the pressure rise due to the momentun change.

FIG 2

EFFECT OF PRESSURE DROP ON TEMPERATURE DROP FOR SATURATED POTASSIUM VAPOR

(REF: WADD TECHNICAL REPORT 61-96)



BAKER CHART

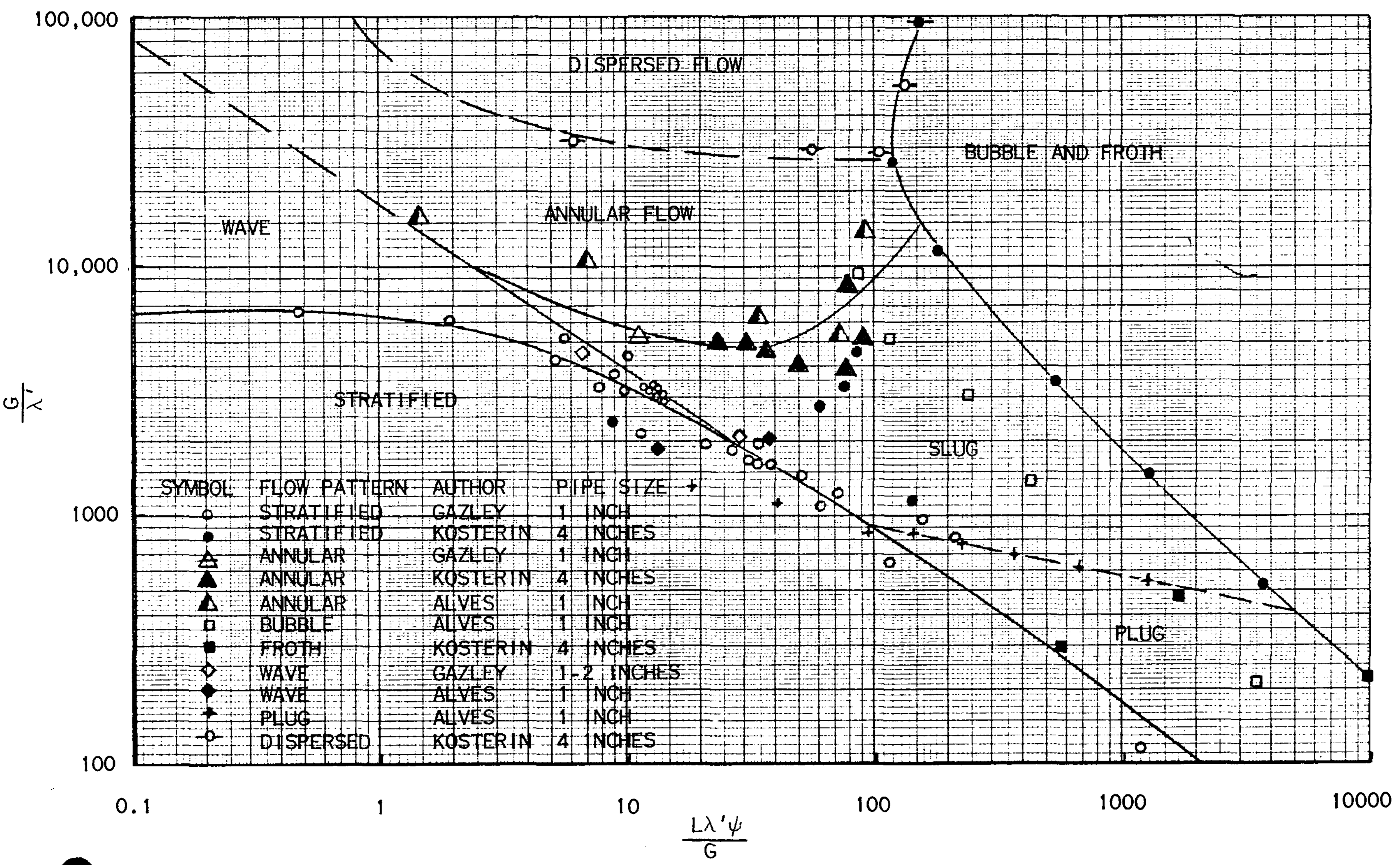


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FIG 3

e. The rise in velocity due to the generation of vapor has the effect of increasing the pressure drop and heat transfer.

f. Thermal entrance effects in the liquid phase have been studied by many investigators. It has been shown in single phase flow that heat transfer coefficients in thermal entrance regions vary from very large (approaching infinity) to the fully developed value. It is quite likely that some type of thermal entrance phenomena will occur also for subcooled, saturated, or two phase fluid entering a heated tube. These effects are not known at this time.

g. Stability is of primary importance in boiling-condensing systems. A great many variables affect the stability of a boiler. Among these are:

1. Amount of subcooling of fluid entering the boiler.
2. Variations with time in heat addition rate.
3. Variation with time in heat rejection rate.
4. Pump characteristics.
5. Frictional losses in system.
6. Flow Restrictive Devices
   (a) Sonic orifice (i.e., in turbine).
   (b) Subsonic orifices in power loop.
7. Effect of gas pockets which are inadvertently present.
8. Flow rate with its effect on flow regime and heat transfer rate.
9. Fluid properties.
10. System geometry.
11. Surface conditions
12. Pressure level.
13. System volume.

Due to the large number of variables, and since most of these variables are interdependent, stability is best arrived at by actual test and development of a prototype design. A fundamental heat transfer rig is helpful, however, in obtaining a better understanding of the stability problems in order to reduce the amount of large scale testing.

4. REVIEW OF LOCAL HEAT TRANSFER COEFFICIENT VERSUS LOCAL QUALITY CURVE

A review of the general shape of the functional relation of local heat transfer coefficient with local vapor quality for forced convection in a tube is desirable in order to describe the information a boiler designer would like to have in order to design a minimum weight Rankine Cycle System. Typical curves are shown in Figs 6 and 7. The test rig described in this report is designed to provide such curves at several flow rates and heat fluxes for several configurations. The slope of the curve is generally positive at low qualities due to an increased amount of boundary layer agitation and fluid mixing due to nucleate boiling and the ever-increasing fluid velocity. At some point in the process (40 to 90 percent vapor quality depending upon many factors) the heat transfer coefficient decreases rapidly to the gas coefficient at 100 percent vapor quality. Indications by some investigators are that at some quality just below 100 percent the local heat transfer coefficient is below the gas coefficient due to a mist type flow in which the fluid velocities are lower than at 100 percent quality and heat is transferred through the gaseous boundary layer by conduction only.

The point at which the local heat transfer starts to drop drastically may be called "Departure from Nucleate Boiling" (DNB). This point is influenced by many factors including:

a. Mass flux (See Fig 7)

b. Tube diameter and L/D ratio (via Prof. Warren Rohsenow)

c. Heat flux and Temperature Difference (See Figs 4 and 6)

d. Fluid properties (i.e., density, surface tension, viscosity, specific heat, etc.)

e. Tube configuration (Ref 22)

f. Coehesion and Adhesion Forces (i.e., wetting versus non wetting)

g. Surface roughness (Fig 7)

h. Surface cleanliness (Fig 5)

i. Aging of surface (i.e., outgassing)

j. Pressure level (i.e., size of bubbles)

The location of this point and the shape of the curve beyond this point is required to make an engineering decision on the relative merits of a once-through boiler generating 100 percent quality vapor versus a partial vaporization boiler separator. The location of this point is required to size the separator circuit. The shape of the curve beyond this point is predominently influential in establishing a once-through boiler length. The boiling resistence in the low quality regime is not a controlling factor, however, in a lithium-potassium boiler. (at $Q/A = 100,000$ Btu/hr ft^2 $\Delta T_{Boiling} = 17F$)

In order to quantitatively indicate the effect of the shape of the local boiling heat transfer coefficient versus local quality curve on the length of boiler tube required for a once-through boiler, several simplified but representative assumed curves are super-imposed on Fig 8. The effect of the gas coefficient on length is quite strong.

FIG 4

THE INTERACTION OF SOME OF THE PARAMETERS RELATING TO NUCLEATE BOILING

(REPRODUCED FROM REF. 19)

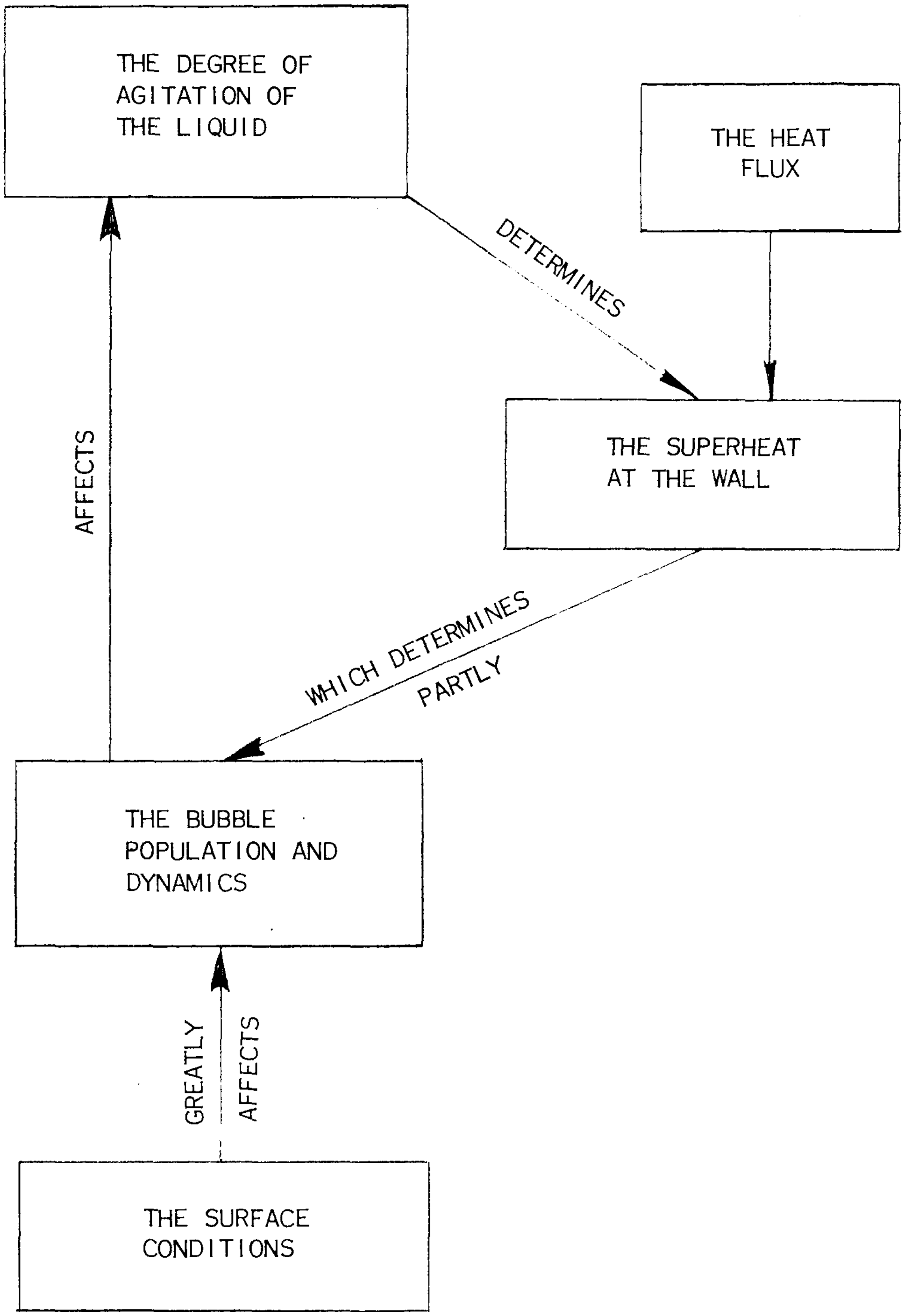


FIG 5

THE EFFECT OF DIFFERENT AMOUNTS OF ROUGHNESS ON THE HEAT TRANSFER IN NUCLEATE BOILING

(H = Q/ΔT, BTU/HR FT^2 F; N = BUBBLES/IN^2)

THIS FIGURE IS REPRODUCED FROM THE PAPER BY CLAUDE CORTY AND ALLAN S. FOUST

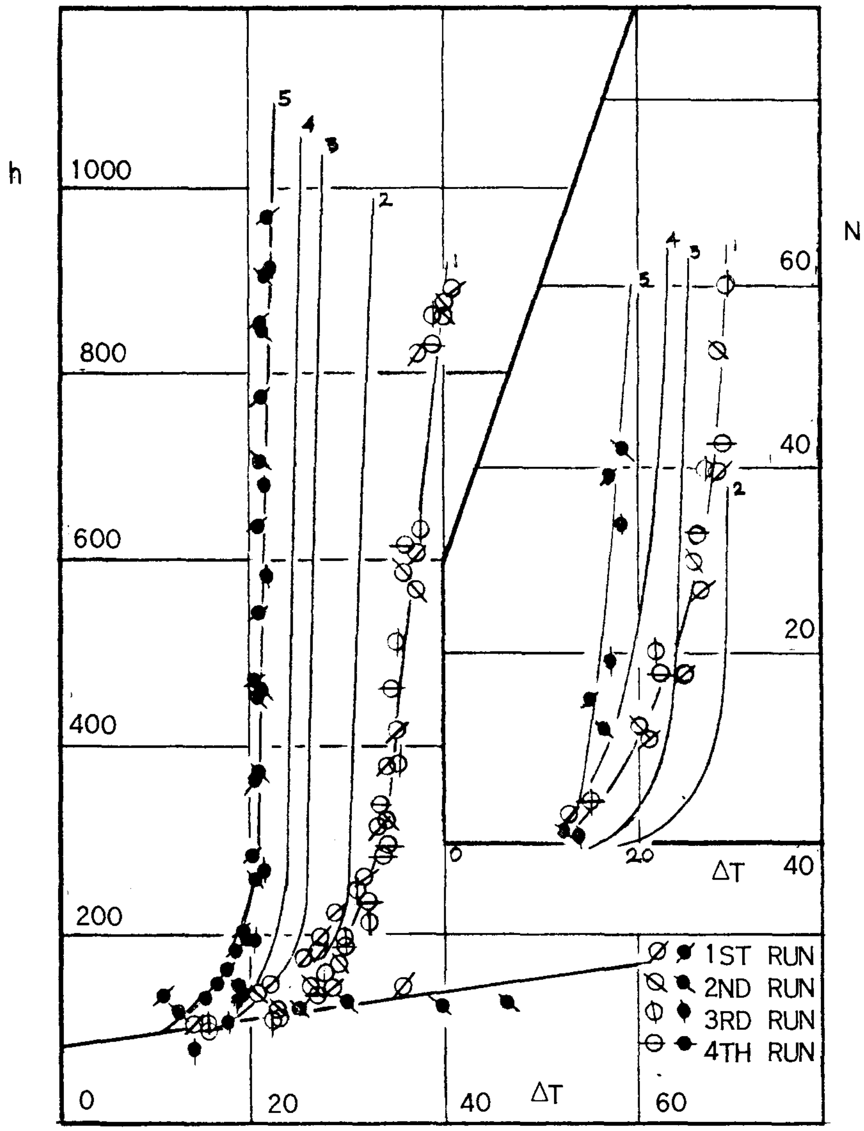


FIG 6

VARIATION OF HEAT TRANSFER COEFFICIENT IN ONCE-THROUGH SYSTEM AT CONSTANT FLOW AND VARIOUS CONSTANT HEAT FLUX VALUES

(REPRODUCED FROM REFERENCE 161)

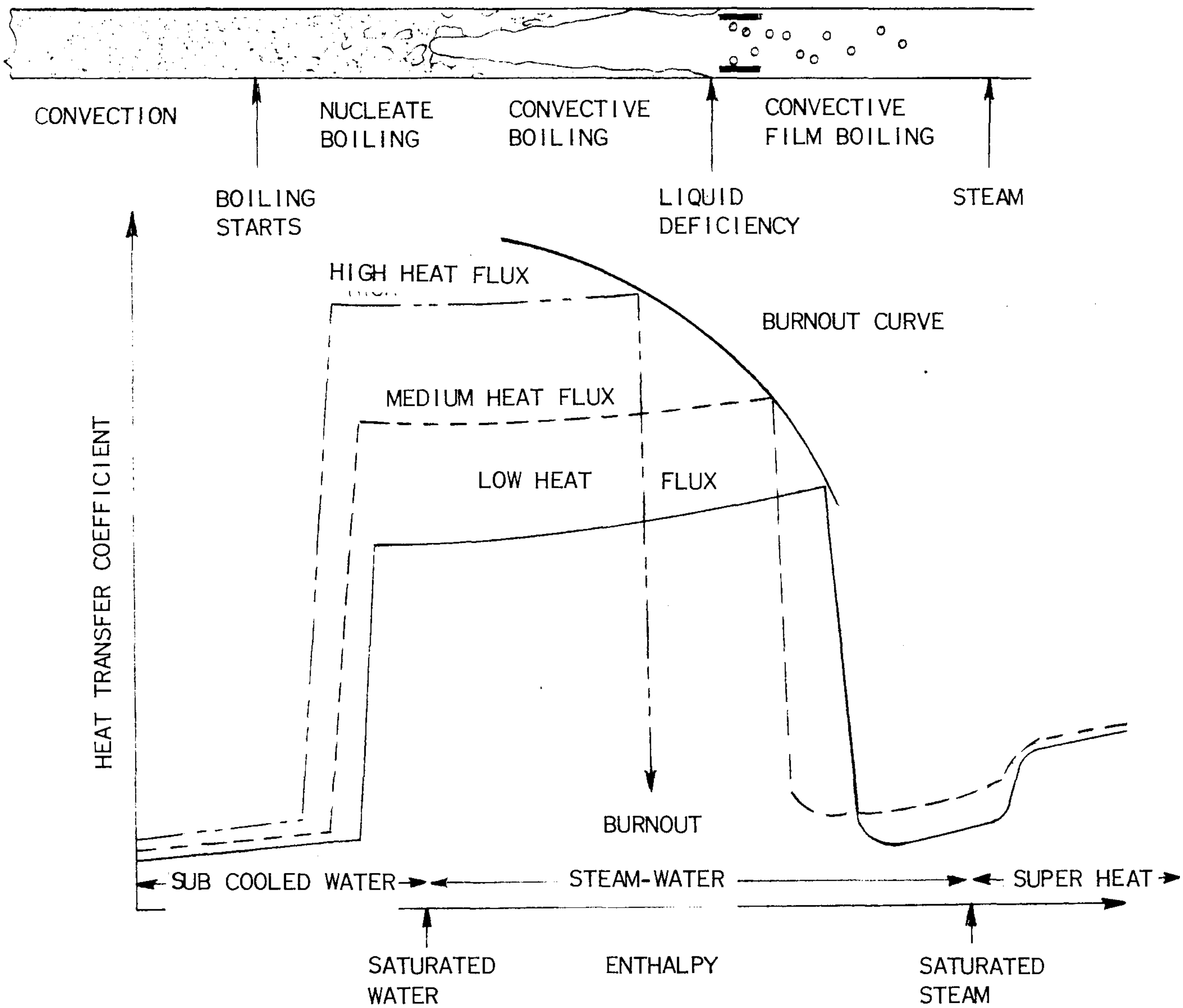


FIG 7

EFFECT OF FLOW RATE ON LOCAL OVERALL HEAT TRANSFER COEFFICIENT FOR BENZENE

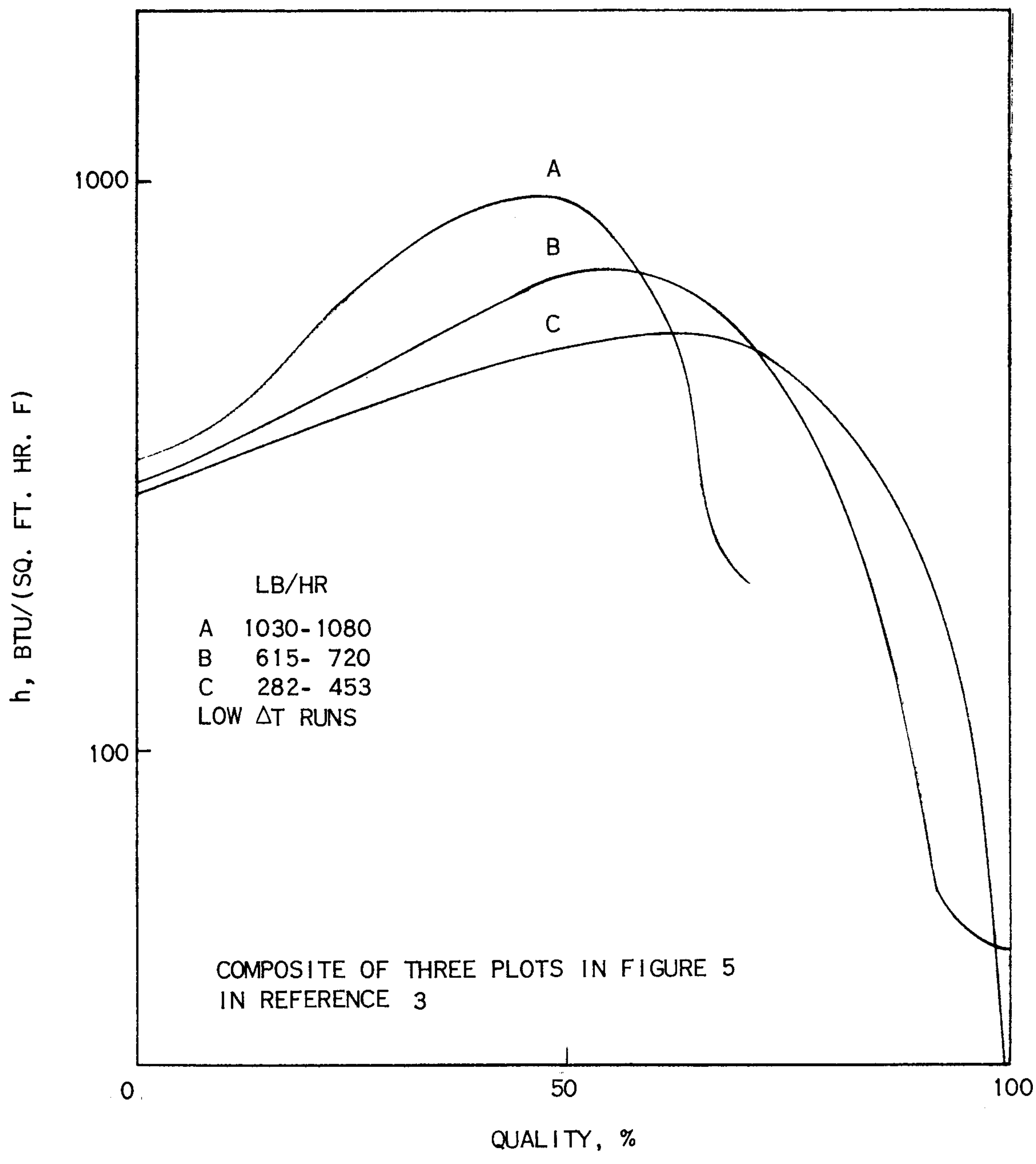
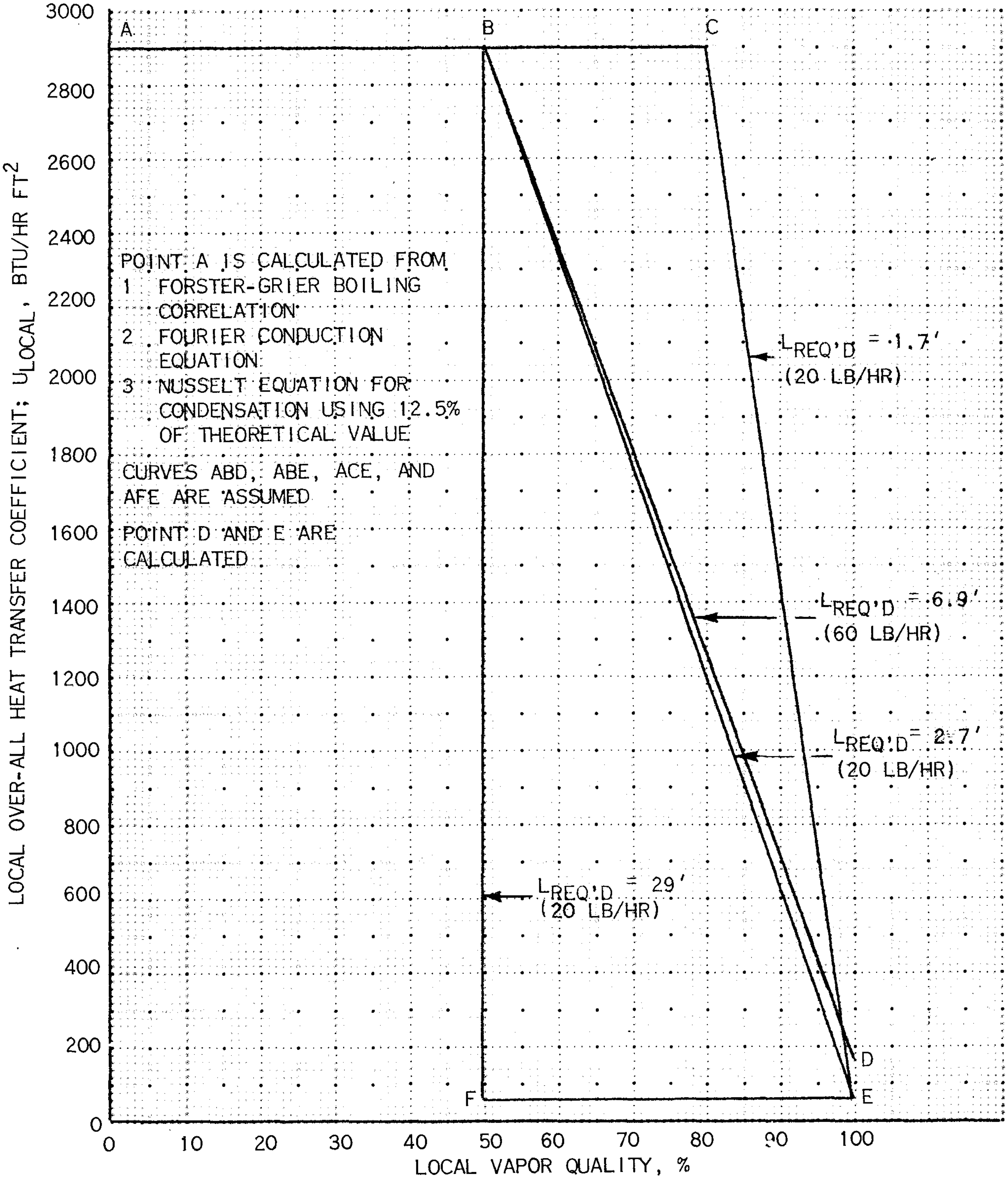


FIG 8

EFFECT OF SHAPE OF LOCAL OVER-ALL HEAT TRANSFER COEFFICIENT (U_{LOCAL}) CURVE ON THE LENGTH REQUIRED TO VAPORIZE POTASSIUM TO 100% QUALITY

(ΔT OVER-ALL = 100F)



5. PREVIOUS WORK IN FORCED CONVECTION BOILING

There has been very little work published on the subject of forced-convection boiling of liquid metals. There have been attempts to extend pool boiling and one-phase forced-convection correlations to this subject. In view of the radical departure of the physical picture of forced-convection boiling from either of these two regimes, it is difficult to find merit in this latter approach. Paragraphs 5.1 to 5.6 have been extracted from the excellent write-up of this subject by Roger Maurice Wright in Ref 1.

5.1 Dengler, and Dengler and Addoms

Dengler used water in an upflow system consisting of a 1-inch-inside diameter, 20-foot-long, vertical copper tube. Five 3-foot-long steam jackets were spaced along the tube and 21 thermocouples were embedded in the tube wall. Local heat fluxes were determined by collecting steam condensate from the specially designed steam jackets. Local pressures were obtained at stations between the steam jackets by a manometer system. Saturated liquid was introduced to the test section with outlet pressures ranging from 7.2 to 29 psia. Mass fluxes were varied from 12.2 to 280 lbm/sec ft^2. The mass vapor fraction (quality), x, varied from zero to 100 percent. Local volumetric vapor fractions were determined by a radioactive-tracer technique.

Dengler and Addoms postulated that the local heat-transfer coefficients at low flow rates and qualitities are governed by the combined influence of boiling and forced convection. As the linear velocity of the vapor-liquid mixture increases, it was proposed that nucleate boiling mechanism is suppressed, and a forced-convection heat transfer mechanism is the dominant factor. Their correlation for the region of suppressed nucleate boiling was

$$\frac{h_B}{h_o} = 3.5\, X_{tt}^{-0.5}, \tag{I-1}$$

where h_o is the heat-transfer coefficient that would be obtained if the flow were all liquid; it is calculated from the Dittus-Boelter equation

$$h_o = 0.023\, \frac{k}{D}\, Re^{0.8}\, Pr^{0.4} \tag{I-2}$$

The physical properties are those of the liquid evaluated at the local saturation temperature, and the Reynolds number, DG/μ_l, is based on the total mass flow rate. The Martinelli-Lockhart parameter X_{tt} is defined by

$$X_{tt} = \left(\frac{\rho g}{\rho f}\right)^{0.5} \left(\frac{\mu f}{\mu g}\right)^{0.1} \left(\frac{1-x}{x}\right)^{0.9} \tag{I-3}$$

This was originally developed for the correlation of pressure drop in two-phase, two-component isothermal flow.* Its possibility as a correlating parameter for heat transfer in two-phase flow was suggested by Lockhart and Martinelli.

*The sub-tt refers to turbulent-turbulent in categorizing the types of flow of the vapor and liquid phases.

In the entrance regions of the test section, heat-transfer coefficients were significantly larger than those predicted by Eq. (I-1). Dengler postulated that this was the region in which the nucleate boiling mechanism was predominant, whereas downstream, with higher linear velocities, boiling was suppressed. A temperature difference to initiate nucleate boiling ΔT_i, was defined by

$$\Delta T_i = 10(V_{avg})^{0.3} \tag{I-4}$$

and was applied as a criteria for nucleate boiling. The average stream velocity V_{avg} was defined by material balance relations and measured volumetric vapor fractions. Dengler obtained no correlation between the liquid velocity and ΔT_i when both liquid and vapor velocities were assumed equal. Then ΔT_i was nondimensionalized in an arbitrary manner, and the data were correlated by multiplying h_B/h_o by the factor

$$(0.673)\ \left[(T_w - T_b - T_i)\left(\frac{dP}{dT}\right)_{sat.}\ \frac{D}{\sigma}\right]^{0.1} \tag{I-5}$$

when the factor was greater than one. Although this factor was used to reduced the scatter of the data, its physical significance is not immediately apparent. Thermal entrance effects were not mentioned. From the results of Siegel and Sparrow, the first boiling section of about 36 inches (which contributed one of the five data points for each run) contained a thermal entrance region of some 24 inches. In view of the relationship of data points in the first heated region to the correlation, it is suggested that thermal entrance effects form a more plausible explanation than the proposed mechanism. It should also be mentioned, that since each one of the 36 inch boiling sections was used to obtain one value of the heat-transfer coefficient, these values are not true local coefficients.

5.2 Mumm

Mumm used water in an electrically heated, horizontal, stainless steel tube; it was 0.465 inch inside diameter and 7 feet long. Local heat-transfer coefficients were obtained for exit qualities up to 60 percent and for pressures from 45 to 200 psia. Heat fluxes ranged from 5.10^4 to $2.5\ 10^5$ Btu/ht ft^2, and mass fluxes ranged from 70 to 280 lbm/sec ft^2. Local heat-transfer coefficients for qualities less than 40 percent were correlated by

$$Nu_B = \left[4.3 + 5.10^{-4}\left(\frac{V_{fg}}{V_f}\right)^{1.64} x\right]\left(\frac{q}{Gh_{fg}}\right)^{0.464} (Re)^{0.808} \tag{I-6}$$

with a standard deviation of $\pm$ 10 percent. Here the V's are specified volumes. The quantity (q/Gh_{fg}) was first introduced by Davidson and has been called the boiling number, Bo.

5.3 Schrock and Grossman

Schrock and Grossman used water in an upflow system. They used electrically heated test sections of 0.1162-inch, 0.2370-inch, and 0.4317-inch inside diameter. Length varied from 15 to 40 inches. Mass fluxes for the small tubes varied from 197 to 911 lbm/sec ft^2; and for the largest tube, 49 to 69. Heat fluxes for the small tubes were 6.10^4 to $1.45.10^6$ Btu/hr ft^2 and for the large tubes $0.65.10^5$ to $2.46.10^5$. Pressures ranged from 42 to 505 psia, and exit qualities up to 59 percent. During the initial stages of the project heat transfer data were correlated in two flow regimes. For very low vapor qualities where nucleate boiling was considered predominant, the correlation was

$$\frac{h_B}{h_l} = 1.15.10^{-5}\, q \qquad \text{(I-7)}$$

The scatter of the data was large in this region. The authors believed that the relatively high coefficients obtained with the low qualities were not due to entrance effects. When the inception of net boiling occurred well within the heated test section ($L/D \sim 60$), the same effects were still observed. At higher qualities a vapor core-liquid annulus type of flow was postulated. These data were correlated with the Martinelli parameter

$$\frac{h_B}{h_l} = 2.5\, X_{tt}^{-0.75} \qquad \text{(I-8)}$$

Here h_l is the local, nonboiling heat-transfer coefficient that would be obtained if the liquid in the two-phase mixture were actually flowing alone and filling the tube. It was also calculated by the Dittus-Boelter equation.

In the final stages of their work, the correlation was modified. It was postulated that heat transfer is dependent on both boiling and forced-convection regimes. The boiling number and the Martinelli parameter, respectively, were used to express these contributions:

$$\frac{Nu_B}{Re^{0.8} Pr^{1/3}} = 1.7.10^2 \left[\left(\frac{q}{G h_{fg}}\right) + 1.5.10^{-4}\, X_{tt}^{-2/3} \right] \qquad \text{(I-9)}$$

The standard deviation was $\pm$ 35 percent.

5.4 Natural-Circulation Boiling in Vertical Tubes

Guerrieri and Talty presented data for the boiling of several organic liquids in natural-circulation vertical-tube evaporators. Tube diameters were 0.75 inch and 1.0 inch; tube lengths were about 6 feet. Heat fluxes were low (up to 17,400 Btu/hr ft^2). Outlet qualities varied from 2.8 to 11.6 percent. Heat-transfer coefficients were correlated in a manner similar to that of Dengler:

$$\frac{h_B}{h} = 3.4\, X_{tt}^{-0.45} \qquad \text{(I-10)}$$

A correction factor for nucleate boiling was also introduced. The physical significance of this correction is somewhat more apparent than that of the one used by Dengler:

$$\text{Correction Factor } 0.187\,(r^*/\delta)^{-5/9} \tag{I-11}$$

where r^* is the calculated radius of the minimum size of thermodynamically stable bubble for a given degree of superheat, and δ is the thickness of the laminar layer of liquid along the wall. When (r^*/δ) was greater than 0.049, it was physically interpreted to mean that flow velocities near the wall were large enough to prevent nucleation of vapor bubbles.

5.5 Evaporation of Refrigerants

Some work on the evaporation of refrigerants in forced flow through tubes has appeared in the literature. The data presented are usually for relatively low mass fluxes (less than 150 lbm/sec ft^2) and low heat fluxes (20,000 Btu/hr ft^2). However, vapor fractions (x) up to and over 90 percent are common. One recent paper summarizes previous work and presents new data (Ref 14). In the experiments, the difference between inlet and outlet vapor fractions was usually about 15 percent. Average heat-transfer coefficients were correlated by:

$$\frac{h_B D}{k} = 0.0225 \left(\frac{GD}{\mu_1}\right)^{0.75} \left(\frac{J\,\Delta x \lambda}{L}\right)^{0.375} \tag{I-12}$$

Where Δx is the change of vapor fraction x over the test section length L, λ is the latent heat of vaporization, and J is the mechanical equivalent of heat. As pointed out in the Discussion section of the paper, the measured coefficients were not true local coefficients, but average values. Also it was pointed out that true local coefficients would depend on the value of x rather than Δx.

5.6 Sterman, Morozov, and Kovalev

Sterman describes forced-convection boiling work carried on in the U.S.S.R. Data are presented for both the boiling of water up to 90 atmos. and the boiling of 95 percent ethyl alcohol at 2 atmos. The boiling tubes used were 120 to 140 mm (4.7 inches) in length and 16 mm (0.63 inch) in diameter. They were electrically heated by using the tube itself as a resistance element. To insure an adiabatic condition at the outer tube wall, the tubes were insulated and then completely surrounded by adjustable guard heaters. Heat fluxes up to 179,000 Btu/hr ft^2 were employed. Superficial velocities were about 6 to 10 ft/sec, and volumetric vapor fractions were varied from zero to 26.9 percent. It was stated that there was no effect due to increasing vapor fraction. However, there was no statement made as to the magnitude of the mass vapor fraction; at low pressures this could easily be less than one percent. Heat-transfer coefficients were correlated according to the following relation

$$\frac{Nu_B}{Nu} = 6150 \left[\left(\frac{q}{h_{fg} v_o\, g}\right) \left(\frac{\rho g}{\rho f}\right)^{1.45} \left(\frac{h_{fg}}{C_p T_s}\right)^{1/3} \right]^{0.7} \tag{I-13}$$

where the Nusselt numbers are for boiling and nonboiling (liquid only), v_0 is the superficial velocity, and T_S is the saturation temperature. All of the above bracketed quantities are dimensionless.

5.7 Preliminary Results of G.E.'s 300 Kw Boiling Potassium Heat Transfer Rig

General Electric is conducting several forced-convection boiling potassium heat transfer tests under NASA Contract NAS 5681. Preliminary results of their 300 Kw boiling potassium test (1300F boiling potassium primary loop, annular flow sodium secondary loop, Haynes alloy No. 25 material) were reported in their fifty quaterly (April 20, 1962).

For a single test the potassium temperature distribution is determined from the mixed mean inlet and outlet values and the saturation temperatures are obtained via pressure measurements. Using the slope of the temperature curves, the heat flux as a function of length is determined. The temperature difference between the wall and the boiling fluid is then obtained from the total temperature difference between the two streams using calculated liquid coefficient values for sodium and a calculated resistance for the wall. The three most significant achievements demonstrated to this author are that:

a. At a temperature difference (t_W-t_S) up to 60F, the data corresponds to the Forster-Zuber nucleate boiling correlation (at 1200F) with a heat flux of 4 x 10^5 Btu/hr ft^2.

b. The local heat transfer coefficient decreases sharply at 50 percent vapor quality to a value only slightly higher than a gas coefficient. The coefficient then increases gradually to 350 to 400 Btu/hr ft^2 F at which the maximum quality (74.5 percent) in their series was obtained. This author theorizes that in their vertical flow system and low flow rate of 145 lb/hr in a 0.88 inch inside diameter tube, the inertia and surface tension forces were insignificant compared to the gravity forces. Pool boiling resulted with an interface occurring at the 50 percent vapor quality location followed by a mist type of flow regime, and a drastic reduction in coefficient since heat rate was controlled by a vapor film. As the mist vaporized, the stream velocity increased with a resultant rise in coefficient. This phenomena is not expected in the Pratt & Whitney Aircraft test since the flow flux in the Pratt & Whitney Aircraft test is higher (3:1) and the tube diameter is smaller (4.7:1) which tends to increase the relative magnitudes of the inertia and surface tension forces compared to the gravity forces. The Pratt & Whitney Aircraft boiler is also tested in a horizontal position.

   No correlations were attempted in the latest G.E. Quarterly.

c. The materials effort in both their Haynes alloy No. 25 loop and their columbium loop seems to indicate a well organized practical approach to the fabrication problems, particularly in welding and heat treating.

5.8 Preliminary Results of Airesearch Boiling Potassium Heat Transfer Tests

The test specimen consists of a bored through nickel billet with temperatures measured at imbedded radial locations to allow extrapolations to obtain the inside wall temperature. A twisted ribbon was incorporated. This is similar to a technique used by ORNL. Due to thermocouple error and uncertainty in thermocouple location, the reproducibility in the heat transfer coefficient was not satisfactory. No trend in h as a function of exit quality was evident. The maximum quality indicated was 15.3 percent. The maximum heat flux was 320,000 Btu/hr ft^2. Very little instability ($\pm$ 3 psi) was noted, even for large degrees of subcooling (ΔT_s = 600F). A throttle valve was useful in reducing flow pulsations but not pressure pulsation. Hot trapping was accomplished with corrugated 0.004 inch zirconium foil. Temperatures up to 1483F were recorded. The test was terminated by loop failure. An incomplete liquid metal fill was indicated.

5.9 Preliminary Results of ORNL's Forced Convection Boiling Potassium Rig

Some results of ORNL's work was published in a paper delivered at the Brookhaven meeting earlier this year. The most significant results of this work appears to be:

a. The agreement of "stable flow" burnout with the correlating equations of Lowermilk, Lanzo, and Siegal for forced-convection boiling of water expressed by

$$(Q/A)_{critical} = 270 \left(\frac{G}{L/D}\right)^{0.85} (d)^{-0.2}$$

$$\text{for } \frac{G}{(L/D)^2} < 150$$

In as much as pressure level determines the size of the vapor bubbles and therefore affects burnout to a great extent, agreement with Lowdermilk's et al correlation, which does not include a pressure term, was aided by the fact that both tests were run at near atmospheric pressure. At atmospheric pressures, the ratios of ρ_l/ρ_v of H_2O and K differ by only 28 percent. Latent heats differ by only 13 percent.

b. Under unstable boiling conditions burnout occurred at heat fluxes only 60 percent as large as those maintained in a stable test and predicted by Lowdermilk.

c. Adjustment of a throttle valve at the boiler inlet was ineffective in controlling fluctuations. (Experience at CANEL in the water proof test, indicated that throttling with a sensitive needle valve was effective in controlling pressure fluctuations and throttling with a coarse needle valve was ineffective when the valves were in parallel and the flow controlled by one valve at a time. Gas pockets may offer a possible explanation).

d. The burnout heat flux for ORNL's test is initiated at about 2.5×10^5 at Δt of about 40F. Above this point, the slope of the Q/A versus Δt curve start to decrease towards a negative value (Fig 10, Ref 15).

e. The authors of the ORNL paper imply that water data in the high quality region may apply to liquid metals due to the similarity in fluid properties. (The liquid-vapor density ratio of water at 325F matches the liquid-vapor density ratio of potassium at 1800F).

6. PRESSURE DROP IN FORCED-CONVECTION BOILING

The total pressure gradient*--$(dp/dL)_{tpf}$, the pressure drop per unit length of flow channel--in forced-convection boiling is the sum of three contributions; friction losses, acceleration losses due to momentum changes, and losses (or gains) due to the hydrostatic head of the contents of the flow channel. Friction losses may be considered independently of the other two contributions; i.e., it was believed that local friction losses in the boiling system could be estimated from studies dealing with adiabatic two-phase flow. Acceleration and hydrostatic heat losses are both dependent on holdup; i.e., they are dependent on the velocities of the two phases and the fraction of the flow channel occupied by each phase. The holdup can be expressed in terms of the volumetric vapor fraction or the slip ratio (the ratio of the average vapor velocity to the average liquid velocity.

6.1 Two-Phase-Flow Frictional Pressure Loss

Recently much work on adiabatic two-phase flow friction losses has appeared in the literature. Most of the work has been experimental, resulting in emipirical correlation. One of the earliest pressure drop correlations and still one of the most quoted papers is that of Lockhart and Martinelli. They obtained two-phase friction losses in pipes, using dissimilar liquids and gases. They correlated their results with two parameters, ϕ_l and X_{tt}. Here ϕ_l is defined by

$$\phi_l = \left[\frac{(dp/dL)_{tpf}}{(dp/dL)_l}\right]^{1/2} \quad \text{and} \quad \phi_g = \left[\frac{(dp/dL)_{tpf}}{(dp/dL)_g}\right]^{1/2} \qquad \text{(I-14)}$$

It is the square root of the ratio of the two-phase frictional pressure gradients to the pressure gradient that would be obtained if the liquid phase filled the pipe and were flowing alone. Parameter X_{tt} was defined by Eq. (I-3) in the previous section. The square of ϕ_l can be considered a friction factor multiplier.

Some of the friction-loss papers have been theoretical, but each has had as its basis some idealized flow model.

6.2 Holdup Data

Very little applicable two-phase holdup data have been published; of those published there are no papers dealing with a downflow system. Lockhart and Martinelli in their pressure drop work obtained holdup data for dissimilar gases and liquids in horizontal pipes. Dengler obtained steam-water holdup data for his upflow boiling system. Marchaterre and Petrick review steam-water holdup data used in nuclear-reactor design.

6.3 Total-Pressure-Gradient Correlations

Martinelli and Nelson extended the work of Lockhart and Martinelli to the boiling system. This extension consisted of empirically modifying friction-factor multiplier values and vapor-fraction values to be more consistent at higher pressures. Then

*Unless otherwise specified the pressure gradient is a local or a point value.

frictional and accelerational pressure gradients were added and integrated over the length of the boiling tube. In this graphical integration, the heat flux and the vapor fraction were arbitrarily specified. The resulting pressure drop values (the pressure drop over the entire tube) were plotted against average pressure level and exit quality. In order to set limits on pressure-drop values, the procedure was carried out twice; once for the so-called homogeneous or fog-flow model where liquid and vapor velocities are assumed equal, and the second time for the modified volumetric vapor fraction data obtained by Lockhart and Martinelli. This latter case is sometimes referred to as the slip or stratified flow model. The first case would supposedly set the upper limit on pressure drop.

Schrock and Grossman correlated total pressure gradient values in the manner of Lockhart and Martinelli (the total pressure gradient replaced the frictional gradient in the definition of ϕ_l), and presented a simplified design procedure. Ninety-five percent of their data were correlated to 15 percent. Using Dengler's holdup correlation, they also obtained frictional pressure gradients. The correlation of this data was not nearly as good as that for the total pressure gradient; probably due to the inapplicability of the holdup data.

6.4 Pratt & Whitney Aircraft Two Phase Pressure Drop Computer Program

A Pratt & Whitney Aircraft report (APM 1122 written by D.N. Osella) describes the method for predicting pressure drop in a tube for two phase vaporizing or condensing flow when external body forces are negligible. The calculation procedure utilities Lockhart and Martinelli's two phase pressure drop correlation to obtain frictional pressure drops. The frictional pressure drop is added to the momentun drop (or gain) to obtain the drop in total pressure. A uniform heat flux is programed.

The program yields frictional pressure drop and momentum pressure change for two phase flow when the following input information is available:

a. Mass flow rate

b. Tube diameter (inside diameter)

c. Liquid and vapor density

d. Liquid and vapor viscosity

e. Vapor quality in

f. Vapor quality out

g. Tube length

7. EFFECT OF SURFACE CONDITIONS, DISSOLVED GASES, AND WETTING

7.1 Surface Conditions

The condition of the boiling surface affects the heat transfer characteristics to a considerable degree. The surface condition affects the bubble population which effects the entire process as shown in Fig 4 . In general there are at least four surface "defects" that are significant of which three are time dependent.

7.1.1 Surface Roughness

The number of nucleation sites and therefore bubble population increases as the surface roughness increases (i.e., Jakob suggests that the heat flux is proportional to the number of active nucleation sites at a given ΔT). The condition of the surface of the heater has been found, experimentally by P. Griffith and J. Wallis (Ref 11) to have a pronounced effect on both the slope and position of the nucleate boiling curve. Corty and Frost (Fig 5) summarized the effects of roughness on bubble population and heat transfer coefficient. At a constant temperature difference of 22F for example, the boiling heat transfer varies from 100 to 1000 Btu/hr ft^2 F and the bubble population varied from 1 to 30 bubbles per square inch depending upon the degree of surface roughness. Boiling correlation becomes difficult and those that do exist are probably representative of the "average" surface.

7.1.2 Adsorbed Gases

The effect of adsorbed gases has been observed to affect the coefficient. This phenomena is sometimes described as "surface aging" which is time dependent and initially results in a much higher boiling coefficient.

7.1.3 Heater Material

The heater material has been shown to affect the boiling coefficient to a very large extent. McAdams shows (in Ref 8) that at a constant Δt of 35F, the heat transfer coefficient for freshly polished chromium plate was 900 Btu/hr ft^2 F and for freshly polished copper the coefficient was 3200 Btu/hr ft^2 F, a factor of 3.55 to 1.

7.1.4 Boiler Scale

Boiler scale has been known to reduce capacity by 70 percent without apparent soiling. For this reason, reasonable effort will be made in this test rig to obtain a knowledge of the fluid and material analysis before and after the test. Samples of the boiling surface will be obtained. Decontamination however may spoil these samples.

7.2 Dissolved Gases

Dissolved gases affect the performance of the Rankine Cycle in at least four areas. Dissolved gases increases the boiling coefficient, decreases the burnout heat flux, decreases the condensing coefficient, and causes flow instability. The latter two have been observed by J. Petrek in the water proof test. The gases tend to increase the heat flux at a given LMTD since t_{sat} decreases as the gas content increases at a given system pressure. See Figs 9 through 11 in Ref 7. The dissolved gases affect the condensing coefficient as indicated in Section 16.1.2 and they affect flow stability due to extra gas pockets resulting in increased "spongeness".

7.3 Wetting versus Nonwetting

The effect of wetting versus nonwetting on boiling heat transfer is considerable. It is shown in Ref 7 that the unwetted boiling heat transfer coefficient is considerably less than the wetted coefficient by a factor of 100 to 1 at Δt_B = 35F for mercury with and without wetting agents. This is most probably due to the contact resistance of a nonwetting fluid. This problem is not anticipated in this test rig except possibly under start-up conditions.

8. SPECIFIC AIMS OF THIS TEST

The specific aims of this test are listed below.

8.1 Boiling Information

8.1.1 Obtain local and over-all boiling heat transfer coefficients as a function of local vapor quality.

8.1.2 Obtain local and over-all pressure drop data as a function of vapor quality and flow for a given geometry and compare with the Martinelli-Lockhart correlations.

8.2 Condensing Information

8.2.1 Obtain over-all heat transfer (U) coefficient as a function of quality change.

8.2.2 Obtain over-all pressure drop data as a function of quality change and compare with Martinelli-Lockhart correlations.

8.3 Stability Information

8.3.1 Qualitative effect of restriction device at boiler inlet and condenser outlet on stability, and a possible check with empirical theory.

8.3.2 Qualitative effect of restriction at condenser exit on stability.

8.3.3 Qualitative effect of accumulator (gas pocket) at boiler inlet.

8.4 Material Compatibility Information

8.4.1 Effect of boiling potassium and boiling sodium on Haynes alloy No. 25 in a bi-alloy loop.

8.4.2 Effect of condensing potassium on Haynes alloy No. 25 in a bi-alloy loop.

9. EXPERIMENTAL EQUIPMENT

9.1 General Flow System

In the design of the flow system, two radically different flow situations had to be considered. A low flow, low heat rejection system is required to achieve the basic objective. A high flow, high heat rejection system is required to calibrate the system.

The flow system consists of a closed loop charged with filtered hot-trapped potassium liquid and pumped with an MSA electromagnetic high pressure low flow pump, (MSA style I, 1/4 inch outside diameter tube). A head-flow curve and a system pressure drop curve are shown in Fig 9. A 1/4 inch Hoke type THY 442 Mod 1 bellows seal stellite seat throttle valve located downstream of the pump is included for stability control. All other valves are manually operated positive open and positive closing, bellows seal, stellite seat valves (Hoke THY 442 Mod 1 and HY 493 Mod 20). A thermal flowmeter and a preheater are included between the throttle valve and the inlet of the boiling section. The potassium is preheated via radiant heat or with I^2R heat depending upon the test condition. The test fluid then enters the first of three tube-shell boilers in which the potassium is vaporized on the tube side. Degassed sodium vapor is condensed on the shell side of the boiler.

The first boiler, considered the workhorse boiler, is designed to heat the fluid to saturation temperature and vaporize the fluid to any desired quality. The middle boiler, considered the test section, vaporizes another five percent (an incremental amount) of liquid in order that a local heat transfer coefficient can be determined for a small change in quality. The third boiler receives the mixture from the second boiler and heats the fluid to some value (10F) of superheat in order to provide a check on heat balance calculations. The fluid is then condensed and subcooled in an air cooled finned tube condenser.

The first portion of the condenser is cooled via forced convection through a device made from micrometallic filter material which almost completely surrounds the tube for five feet. It is expected that this device, which was designed to obtain uniform air flow to obtain a uniform heat flux, together with temperature measurements will evaluate the Martinelli-Lockhart correlation for pressure drop in two phase liquid metal systems. The latter portion of the condenser is cooled via radiation and natural convection.

An E.M. flowmeter is included to record instabilities in flow. Two surge tanks with special valving are included in order that the E.M. flowmeter and the thermal flowmeter can be calibrated via volume-time measurements. An accumulator with an isolation valve is included to test the effect of gas pockets at boiler inlets on stability.

Pressures, temperatures, and liquid level measurements are taken at various points in the rig. This instrumentation is further discussed in the instrumentation section of this report. Table 1 presents expected rig capability.

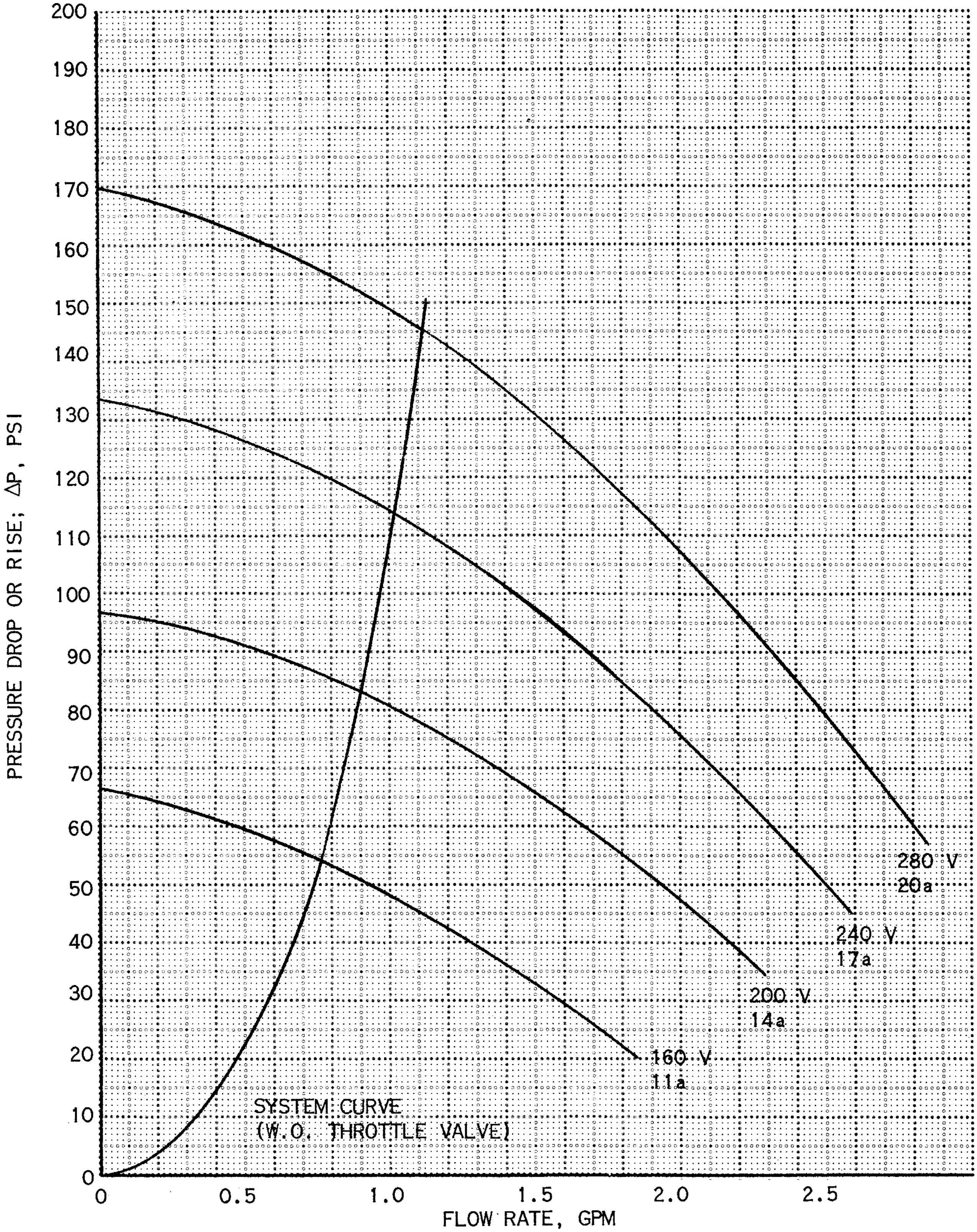
9.2 Heating System

Condensing sodium is utilized to transfer heat into the potassium system. Simply described, a pool of sodium is made to boil in the shell side of a heat exchanger and

FIG 9

PUMP PERFORMANCE AND SYSTEM PRESSURE DROP

(MSA STYLE 0.5-150 ELECTROMAGNETIC PUMP)



the vapor is allowed to condense on the "cold" potassium filled tube. This scheme has several distinct advantages over other types of heating systems.

a. It avoids a secondary loop heating system.

b. It avoids a direct I^2R heating system which is inherently more unstable.

c. No wall temperature measurements are required if the condensing coefficient is known or very large and heat loss from the boiler is insignificant. Condensing coefficients will be obtained experimentally. Guard heaters control heat losses.

d. No liquid metal immersion thermocouples are required if the condensing coefficient is known or very large and heat loss from the boiler is insignificant. Condensing coefficients will be obtained experimentally. Guard heaters control heat losses.

e. A uniform tube wall temperature is achieved.

The disadvantage that the process does not simulate the two loop system which is contemplated in Space Power Plant boilers can be somewhat discounted in that local heat transfer coefficients are obtained. The major disadvantage of the condensing scheme is apparent at high heat fluxes only. High heat fluxes are not contemplated in the first test since the region of most engineering interest is in the DNB portion of the tube. In the low quality region where the boiling coefficient is very high, the resistance offered by the nucleate film is small compared to the wall resistance and the shell side fluid resistance in a tube and shell type of boiler. A high heat fluxes, the condensing resistance becomes a significant portion of the over-all L.M.T.D. In as much as the condensing coefficient cannot be accurately calculated due to the large variance among investigators, a pretest condensing coefficient calibration will be performed. Fig 10 shows the relative magnitude of the condensing Δt_t, wall Δt_W, and boiling Δt_B as of a function of heat flux. Fig 11 indicates the possible discrepancy when calculating the condensing Δt_C, when using the several "suggested" values of the Nusselt Modulus that are available. It is believed that the large variance in the Nusselt Modulus among investigators may be due to additional resistance to heat flow due to noncondensibles. Fig 12 is a sketch showing the postulated mechanism to explain the insulating effect of noncondensibles. In order to insure that the sodium filling and degassing technique produces reproducible condensing coefficients, a water test simulating the second boiler was conducted. Although the test results do not agree with theory, the condensing coefficients are reproducible within five percent. (See Fig 16 in the appendix). The water test has confirmed the sodium cavity filling procedure which is briefly described below:

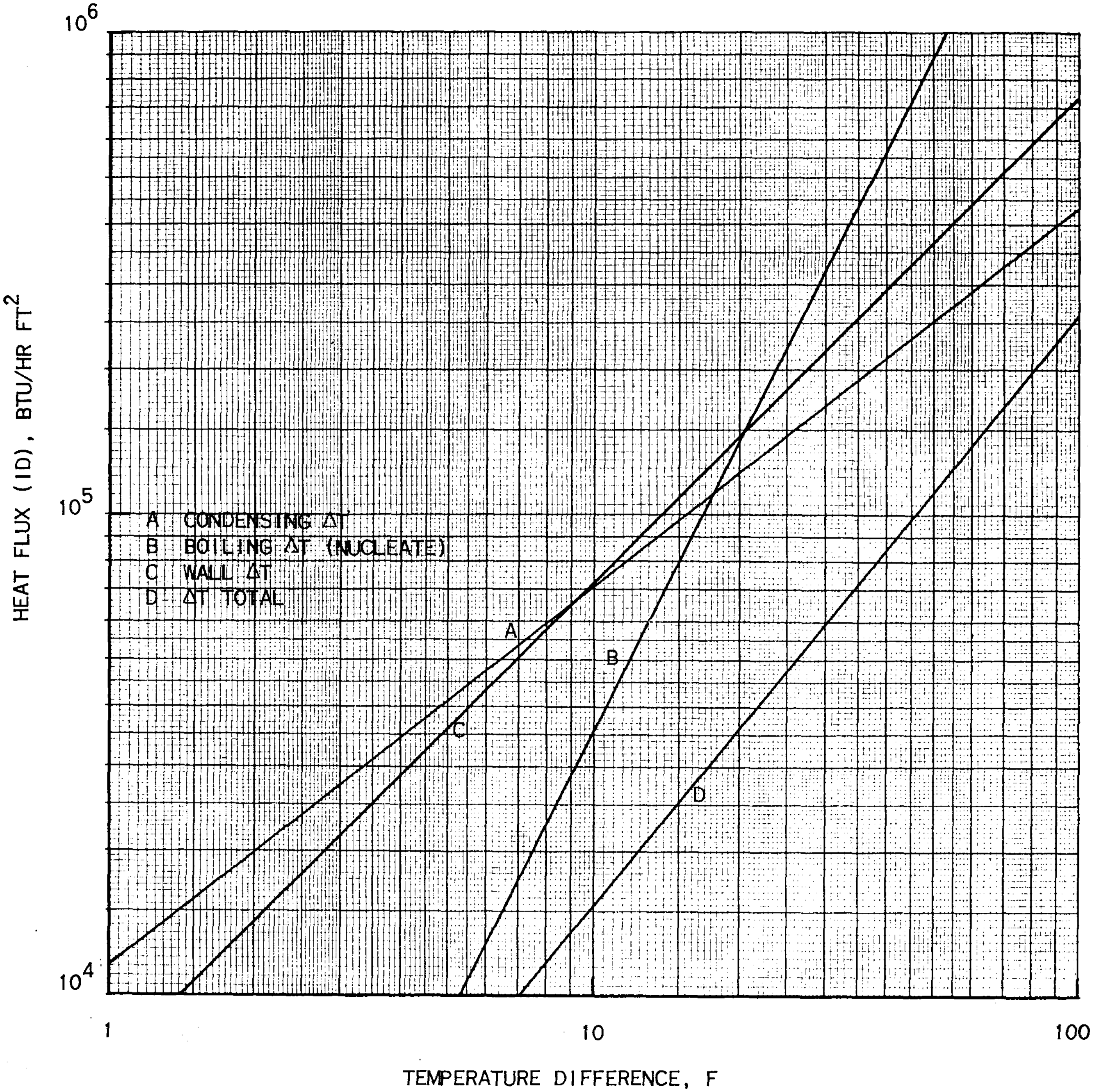
a. The cavity is purged and filled with purified helium.

b. The cavity is pressure filled with purified and filtered sodium. A 5 micron filter is an integral part of the fill-drain pot. The filling operation is stopped when the surge tank indicates the presence of sodium.

c. After proper valving, heat is added to the sodium which boils off thus boiling out all noncondensible gas. The noncondensibles are swept out via high velocity sodium vapor through a vent tube, and deposited back into the fill-drain pot together with condensed sodium.

FIG 10

HEAT FLUX VERSUS TEMPERATURE DIFFERENCES FOR 1800F SODIUM CONDENSING ON A HORIZONTAL 1/4 INCH DIAMETER TUBE (0.032 INCH WALL) CONTAINING NUCLEATE BOILING POTASSIUM



TEMPERATURE DROP VERSUS HEAT FLUX FOR SODIUM VAPOR CONDENSING ON A HORIZONTAL TUBE VIA NUSSELT THEORY

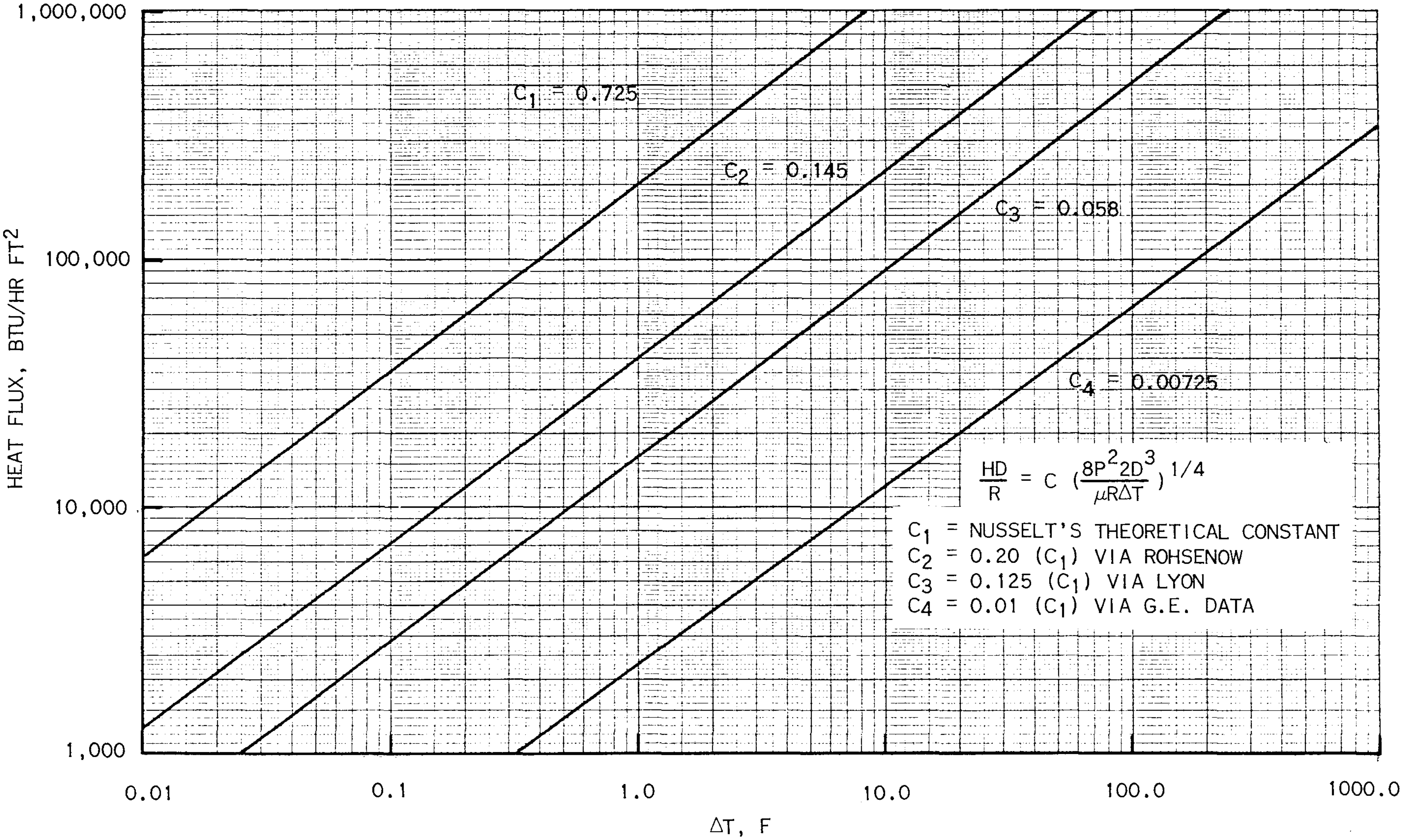
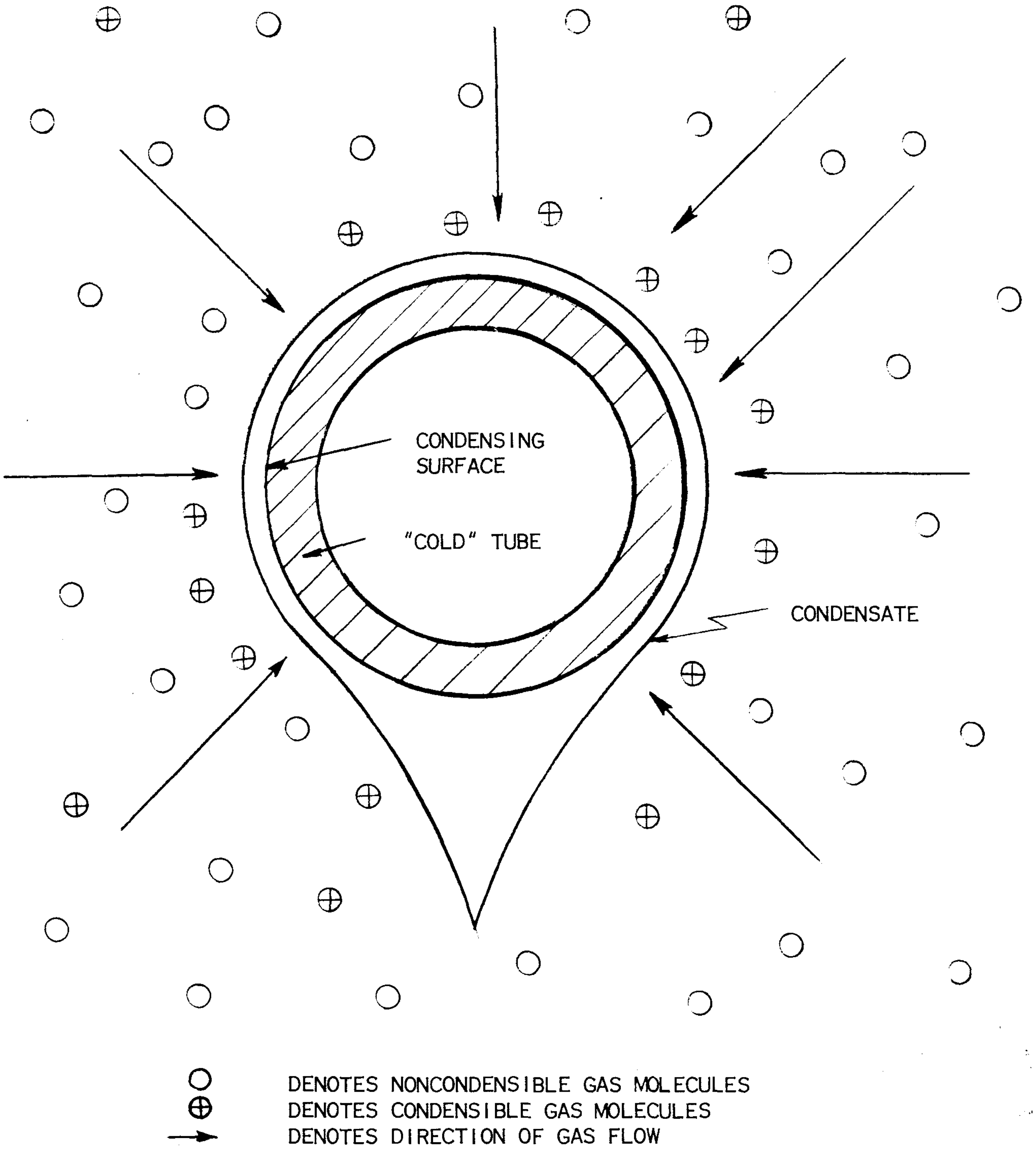


FIG 11

FIG 12

POSTULATED MECHANISM TO EXPLAIN INSULATING EFFECT OF NONCONDENSIBLE GAS ON A CONDENSER TUBE



SINCE THERE IS NO 'SINK' FOR NONCONDENSIBLE GAS MOLECULES, THEY TEND TO BUILD UP ON THE CONDENSING SURFACE DUE TO THE FORCE FIELD ERECTED BY THE FLOWING CONDENSING MOLECULES. THIS BLANKET OF NONCONDENSIBLE GAS MOLECULES TENDS TO INSULATE THE TUBE FROM THE CONDENSING VAPOR.

d. This process continues till the proper level is achieved as indicated by liquid metal level probes, verified by timing the venting process and hopefully confirmed by X-ray.

The sodium pool is heated and vaporized by Haynes alloy No. 25 clad tubular heaters immersed in the pool. Power is manually controlled by a powerstat. Voltage and current readings will be taken (and power checked with a watt meter) to enable a heat balance measurement. Heat loss will be made up by the guard heaters. Since only a very small heat flux exists at the outer surface of the boiler shell, sodium vapor temperatures are the same as the measured wall temperatures existing on the air side of the cavity. End losses will be established via pretest calibration utilizing the guard heaters.

All three boilers are identical except for length and the number and size of immersion heaters. A bare (high response) chromel-alumel thermocouple on the surface of the boiler shell signals a pyrovane control when the temperature (or pressure under boiling conditions) approaches an unsafe condition.

9.3 Instrumentation

A description of other than mundane instrumentation is included. Since accurate flow, liquid level and temperature measurements are imperative, the reader will notice a general redundancy of instrumentations associated with these quantities.

9.3.1 Absolute and Differential Temperature Measurements

Thermocouples are used sparingly throughout this rig to obtain the necessary temperature measurements. All thermocouples used in this rig are chromel-alumel couples (P&WA drawings 1029530 and 1029286. The thermocouples conform to CANEL specification CS-1927 Class II.

Generally two types of thermocouples are used in this rig. For absolute temperature measurements bare wire couples or cladded couples are tacked directly onto the surface per method 4 or 6 in CANEL drawing 1027164. For differential temperature measurements, a gang (thermopile) of three 1/8 inch diameter stainless steel sheathed insulated junction couples (in series) are wired to buck the signal from another gang of thermocouples. A differential millivolt signal is obtained.

The absolute temperature on the surface of the boiler (i.e., the temperature from which the coefficient is determined) is measured by electrically isolating one thermocouple from the gang of three insulated thermocouples. This measurement is checked by an optical pyrometer through a near black body emitter on the boiler surface.

9.3.2 Liquid Level Measurement

Liquid level measurements are taken at several places, primarily to establish the proper sodium level and to obtain a volume-time calibration. Two

"J" probes are installed in all tanks although only one probe will be electrically connected at any time. The "J" probes used in this rig are described in CANEL drawing L-101522. The level in all except the calibration tanks are indicated by millivolt indicators. The level in the calibration tanks are recorded on an oscillograph so that the slope of the curve represents the flow rate. A 1/4 inch change in liquid level represents a 0.44 millivolt/amp change.

9.3.3 Flow Measurement

An accurate flow measurement is required in order to determine the vapor quantity into the test section, the change in vapor quantity in the test section, and to obtain accurate heat balance checks. A description of each scheme is included:

9.3.3.1 Volume-Time Calibration Tanks

Two additional tanks with appropriate piping and valves are included in the system in order that the flow metering devices can be calibrated. in place. Fluid is pumped from one tank to another via the E.M. pump. The change in fluid level is recorded on an oscillograph to obtain a flow rate. The three metering devices used in this rig will all be calibrated by these tanks.

9.3.3.2 Electromagnetic Flowmeter

In order to obtain a satisfactory electrical signal in a device of this kind, it is desirable to obtain

1. a high fluid velocity
2. a high fluid electrical conductivity
3. a tube wall with infinite electric resistance
4. a strong magnetic field
5. good electrical contact between fluid and tube

Item 1 is fixed by tube geometry and flowrate, Item 2 is fixed by the test fluid and temperature, Item 3 is most restrictive since metal must be used as a containment vessel in a practical liquid metal system, Item 4 is restricted by available equipment and practice, and Item 5 is a function of the wetting characteristics between the fluid and the tube.

Since the ratio between necessary tube crossectional area needed to contain the liquid metal and fluid crossectional area increases as tube sizes decrease, an increasing amount of the voltage generated gets short circuited through the tube wall with a decrease in tube diameter.

It is believed that pinching of the tube is ineffective for this reason. Utilizing a 2000 gauss magnet, the signal strength expected at 20 lb/hr potassium at 800F in a 1/4 inch outside diameter x 0.035 inch wall tube is 0.17 millivolts. It is hoped that transient flow conditions can be spotted although random background noise may mask actual flow fluctuations. It may be noted that a steady signal from the E.M. meter indicates steady flow only in that portion of the loop and that flow instability can still exist elsewhere. The flowmeter will be calibrated in place via volume time measurements.

9.3.3.3 Thermal Flow Meter

The thermal flowmeter was considered a necessity for low flow steady state flow measurement. By measuring the temperature rise of potassium flowing through the meter the average absolute temperature and the heat rate into the potassium, the flow rate can be calculated. The accuracy of this meter is strongly dependent upon determining a net heat rate into the potassium. To this end, guard heaters are utilized and a no-flow heat loss test will be conducted. Tube wall temperature measurements will be taken in near zero heat flux regions via insulated junction thermocouples. This meter is inherently a low response instrument. Fig 1 shows the thermal flow meter in detail.

9.3.3.4 Valve

As a final back-up, the throttle valve which is located downstream from the pump can be used as a flow measurement device under most conditions encountered. A ΔP-flow curve can be established at any valve setting via in-place volume-time measurements. This technique is most promising at low flows and high pressure drops.

9.3.4 Differential Pressure Measurements

Differential pressure measurements are obtained across the three boilers and across the condenser via four Taylor Model 225TN114 differential pressure transmitters. An accuracy of two percent of full scale is expected. The system has a reasonably high response.

9.3.5 Pressure Measurements

Liquid metal pressures are taken at the pump outlet, preheater inlet, and condenser outlet via three Taylor Model 225TN5114 pressure transmitters. An accuracy of two percent of full scale is expected. A reasonably high response is expected.

9.3.6 Electric Power Measurements

Electric power into the immersion heaters is measured by Weston type 904-1902006, 904-1902004 multi-range voltmeters, and Weston type 904-2903001, 904-2903004 multi-range ammeters. An instrument accuracy of 1/2 percent of full scale is expected.

The power into the immersion heaters represents the heat into the potassium when all the heat loss is made up by the guard heaters.

9.3.7 Optical Pyrometer Measurements

Optical pyrometer temperature measurements at the boiler surfaces are observed to verify thermocouple readings and to give a quantitative indication of thermocouple drift. The pyrometer to be used in this experiment will be a Shawmeter two-color or a L & N optical pyrometer.

Observation will be made through a six inch long 3/8 inch diameter tube fastened to the surface of the boilers. A near black body emitter is expected. Observation will be made through air only.

9.4 Preheater Design

A potassium preheater is required to heat the potassium liquid from 900F to 1700F for the low flow (20 lb/hr) boiling condition, and from 1000F to 1500F for the high flow (100 to 350 lb/hr) calibration condition.

A 22 inch radient clamshell heater is used to heat the fluid to the saturation temperature during the boiling tests. Radiant preheating was selected in order to minimize the possibility of material burnout in the event of a five second flow reversal during two phase flow. Burnout is prevented by temperature monitoring at the preheater outlet. An analysis of the radiant preheater is included in the appendix. Fig 13 illustrates the potassium axial temperature profile during radiant preheating.

9.5 Boilers

The design of the boilers involved the determination of:

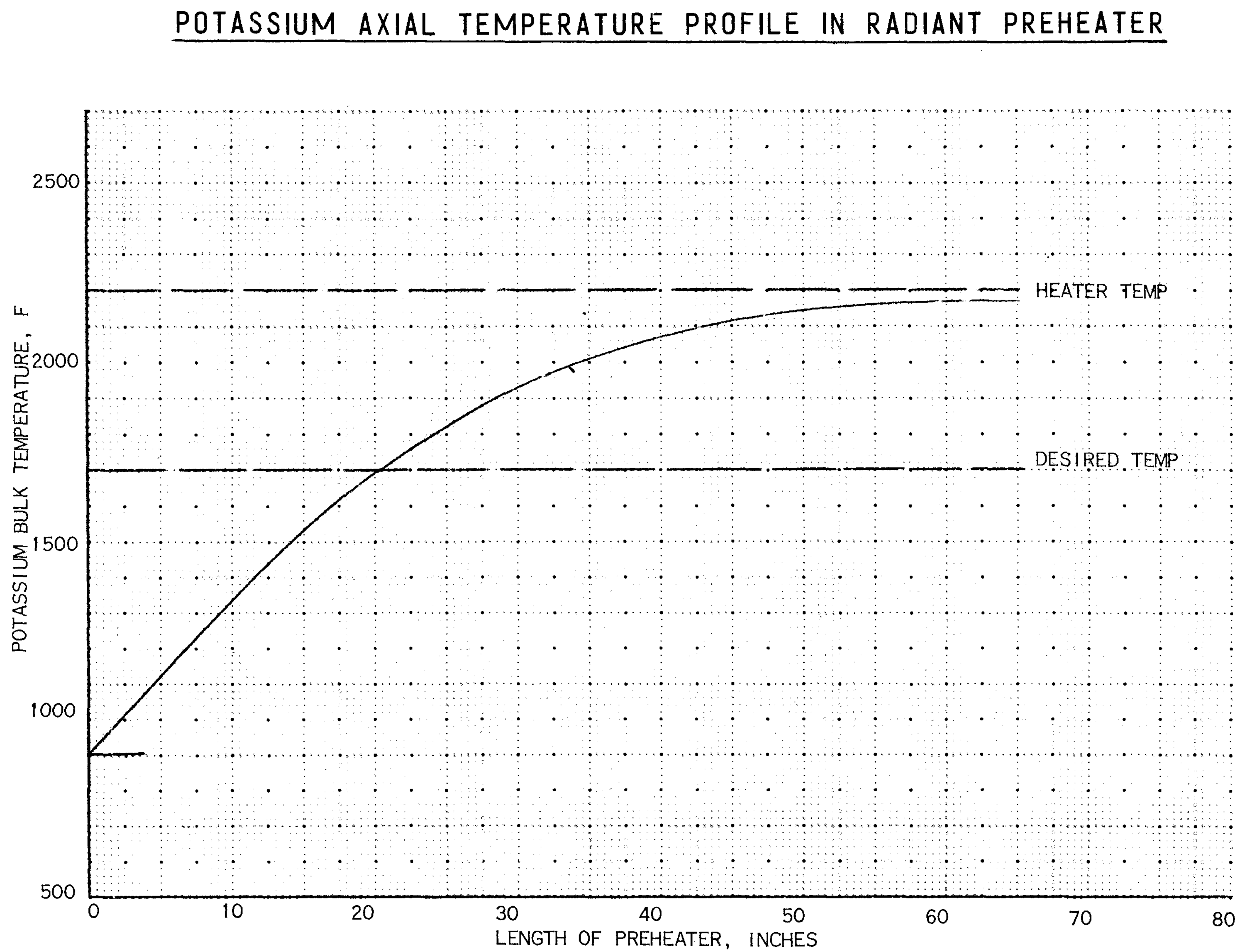
1. Boiler tube length
2. Heater design
3. Mechanical and fabrication details
4. Filling, draining and venting requirements and procedures
5. Heat loss control

A description of the boilers and heaters is included below. The analysis of the heaters and determination of boiler tube length is included in sections 16.2 and 16.4

9.5.1 Boiler Design

a. Potassium Side: (Tube side)

The first test section to be tested is fabricated from a 1/4 inch outside diameter x 0.032 inch wall weld-drawn Haynes alloy No. 25 tubing bent



CNLM - 4358
FIG 13

in a modified sine wave shape as illustrated in Fig 1 and described in CANEL Design Layout L-101567-8. It is hoped that the inertia forces induced by the curved tube and experienced by the liquid phase will force the fluid against the heated surface (depending upon velocity, viscosity, density, pitch, curvature and diameter). However, it is also conceivable that these constantly reversing forces will tend to produce liquid flow in the core of the tube if the forces on the liquid particles reverse the path of the particle before the particle reaches the heated surface. Erosion is also possible at high velocities.

b. Sodium Side: (Shell side)

The design of the shell side was dictated by the following requirements:

1. Generate sodium vapor at 1800F
2. Support the boiler tube
3. Remove non-condensibles in the sodium liquid and vapor
4. Contain saturated sodium liquid at 1800F
5. Support and contain immersion heaters
6. Design for minimum heat loss

To this end, sodium at 1800F is vaporized by Haynes alloy No. 25 clad immersion heaters and contained in a 3-1/2 inch outside diameter x 1/4 inch wall Haynes alloy No. 25 pipe. The tube to header joint is prepared via diffusion bonding, flaring, and welding. This technique is described in CANEL specification CS-119. The tube is supported by several vertical plates and the shell headers. The tube is protected from splashing and dripping by a splash guard which is also supported by the vertical plates. Non-condensibles are removed via boiling and venting through a tube.

9.5.2 Immersion Heater Design

Joulian heat is added to the sodium by calrod type resistance heaters. The resistance wire is B&S 18 gauge Kanthal A-1 alloy. The wire is sheathed in 0.187 inch diameter (outside diameter) by 0.016 inch wall Haynes alloy No. 25 weld-drawn tubing. Magnesium oxide (MgO) electrically insulates the wire from the sheath. The heaters conform to specifications SKN 17109 through SKN 17112.

An analysis was performed to estimate the heat flux capability of these heaters in boiling sodium. A heat flux of 100,000 Btu/hr ft^2 and a maximum Kanthal centerline temperature of 2180F is possible (at 1850F sodium temperature). It is expected that these heaters will operate at a maximum heat flux of approximately 52,000 Btu/hr ft^2 at 1846F centerline temperature (at 1800F sodium temperature). This analysis is included in the appendix.

An evaluation test of these heaters will be conducted in a sodium environment at 1800F in order to ascertain that normal operating conditions can be achieved and to establish a maximum heat flux limit.

9.6 Heat Rejection System

A finned air cooled tube acts as a condenser and cooler during test conditions and as a cooler only during calibration conditions.

In the first five feet of the condenser, heat is transferred to air by forced convection through a sintered steel filter in order to obtain a uniform heat flux and check the Martinelli-Lockart two phase pressure drop correlation. Except for a one additional foot of forced convection cooling section downstream of the five foot uniform heat flux section, the fluid is cooled to 800F by natural convection and radiation. Fig 1 pictures the device used to insure a uniform air flow rate.

9.7 Material and Atmosphere Control

9.7.1 Material Control

High temperature environments dictate the use of special material control practices to avoid contamination of materials and systems. A brief description of the controls to be used is included below.

The stainless steel and Haynes alloy No. 25 alloys are purchased and inspected to AMS specification except for Haynes alloy No. 25 tubing which is purchased to CF 245 specification. Upon receipt of this material at CANEL, standard CANEL "liquid metal use" material inspections, are conducted. Haynes alloy No. 25 samples will be analyzed for composition before and after the test. An outgassing procedure to remove adsorbed gases will be conducted prior to filling the loop.

The sodium will be filtered twice, once upon charging of the fill-drain tanks, and again through a 5 micron sintered stainless steel filter integral within the fill-drain tanks. A material analysis will be made prior to filling the sodium fill-drain pot and upon termination of the test. A 20 ppm oxygen content is expected in the sodium.

The double distilled sodium free potassium, which will be used in this test will be purified via hot trapping and filtered upon charging of the fill-drain pot at CANEL by the liquid metal fill group, hot trapped again with titanium sponge at 1400F for 24 hours inside the fill-drain pot, and filtered (5 micron) upon charging the loop. A material analysis will be made prior to filling the potassium into the fill-drain pot and upon termination of the test.

9.7.2 Atmosphere Control

Purified helium per CANEL specification CS-1983 will be used as the cover gas. Gas purity is to be achieved by the cryogenic method (CS-300A) utilizing activated charcoal absorbers. The gas system will be prepared according to CANEL specification FPS-E-363 to prevent undo contamination. The gas system will be 100 percent welded construction downstream of the purifiers. No continuous monitoring of the cover gas is anticipated although possible. Filling of the loop will proceed only when the loop purity reaches less than five ppm oxygen and -90F dewpoint.

9.8 Test Limitations

The test rig was designed to provide design type boiling heat transfer data at several flow rates for many types of tube configurations. Due to inherent limitations in the test scheme, in the design, and in instrumentation, certain limiting features are presently apparent.

a. Due to very small flow rates in single tubes of multi-tube boilers, flow measurement becomes a serious problem in establishing the quality level and quality change with reasonable accuracy. At very low flow rates all three flow measuring schemes used in this test become inaccurate, with the valve ΔP method offering the most promise.

b. In as much as temperature differences in the nucleate boiling regime are very small, the data in the low quality range is expected to look like "buck shot" due to the inability in obtaining an accurate absolute temperature measurement ($\pm$ 1F) in the 1800F range. In the high quality range, high temperature differences are expected and this problem diminishes. Optical pyrometers with near black body type emitters will be utilized to check and perhaps minimize this thermocouple error.

c. Due to the size of the test boiler (dictated by a minimum heat loss requirement), only a few types of tube shapes can be tested (modified since wave, short "hockey sticks") unless the boiler cavity is increased in size. No straight tubes will be tested due to the high thermal stresses inherent in redundant structures.

d. In a single length heated tube, the flow regime develops and changes as the fluid traverses down the tube length. In "breaking up" the tube, both dimensionally and by heating method (discontinuous heating and discontinuous sine wave), the flow regime may not represent that which would occur in a continuous tube. The heat transfer coefficients obtained from this test become representative of this test configuration only.

e. As in all flow systems the pressure drop is a parabolic function of the velocity. A 3:1 range in flow starts to exceed the range of pressure drop instrumentation. It becomes evident from this that pressure drop data can only be obtained for moderate ranges in flow without excessive ΔP instrumentation.

f. Fluid properties at elevated temperatures are not accurately know. Viscosity of potassium is not known above 1500F. Latent heat and vapor density are determined by theoretical means. The thermal conductivity of vapor is estimated, the density of liquid is extrapolated, vapor pressure data exists to 1400F only, etc., etc.

g. In order to obtain good data in the low quality regime, (low Δt) a high heat flux is required. It is possible to achieve a high heat flux with the present design; however, only over-all coefficients can be determined due to the large amount of heat (large quality change) transferred to the potassium at high heat fluxes.

h. The material of the boiler tube (Haynes alloy No. 25) does not simulate the material of interest. The result of this disparity is to lower the operating pressure (and coefficients) added to the problem that the effect of heater material on the boiling heat transfer coefficient is not known.

9.9 Fabrication

The loop and all component parts will be fabricated at CANEL to assure high quality welds necessary for liquid metal service. All welds in the high temperature portion of the loop will be X-rayed and dye checked except for the tube to header joints which will be welded and diffusion bonded and cannot be X-rayed. These joints will be dye checked only. Most of the welds in the low temperature region of the loop will be X-rayed and dye checked except for those which are inaccessible for X-raying. The loop will be leak checked at various stages of assembly and at the completion of assembly. All tube bends, machined tees and elbows, will be dye checked.

The loop and all component parts will be assembled according to CS-309 Class II (good cleanliness) specifications in the stand area under positive filtered air pressure inside a temporary clean room enclosure.

9.10 Loop Support System

The loop support system is illustrated in CANEL Facilities Drawing CR103841. Briefly described, the pump is rigidly fastened to an angle iron frame which is lagged to the floor. The potassium and sodium fill and drain pots, the boilers and the potassium condenser are supported by vertical wires rigidly connected to a heavy structural support frame. The rest of the loop is supported by cables with counter weights and stops.

10. EXPERIMENTAL PROCEDURE

10.1 Boiler Heat Loss Calibration

Heat leakage from and to the test boiler, must be determined or controlled in order to obtain an accurate measurement of the boiling heat transfer coefficient, quality level, and quality change. Heat leakage is through the cylindrical portion of the pressure shell, the ends and through the various connections on the boiler. A significant amount of heat flows between boilers through the interboiler connections, especially at high qualities when significant temperature differences are expected between boilers. Heat losses are controlled via guard heaters or heat dams.

Three heat loss calibration procedures are possible. They may be described as 1) Heat Make-up by Immersion Heaters, 2) Heat Make-up by Guard Heaters, and 3) Heat Dam Method. The procedures are as follows:

1. Heat Make-up by Immersion Heaters

   With no flow, and an arbitrary setting on the guard heaters, determine power required in immersion heaters to hold boilers at an expected operating temperature. Via proper setting of the guard heaters, the heat to be made up by the immersion heaters can be held to a minimum. This can be done for several temperature levels and for several temperature differences between boilers. The no flow power into the immersion heaters represents the heat loss at particular guard heater setting, and for definite boiler temperature levels and differences.

2. Heat Make-up by Guard Heaters

   With no flow, and no power into the immersion heaters, increase power into the guard heaters until the temperature level is maintained. This can be done with several temperature levels and with several temperature difference between boilers.

3. Heat Dam Method

   By measuring the temperature difference between the boiler shell and the guard heaters, a zero ΔT can be obtained and heat loss out the cylindrical surfaces of the vessel can be held to negligible amounts.

Of the three schemes, the Heat Make-up by Guard Heater holds the most promise. The advantage is that it is simpler than method (1) and includes end loss which cannot be included in (3). It does, however, depend upon a constant thermal conductivity for the insulating materials.

10.2 Condensing Coefficient Calibrations

The condensing coefficient for liquid metals on horizontal tubes cannot be reliably calculated due to the variance among investigators. A calibration will be performed to establish the condensing Δt as a function of heat flux at the expected vapor temperature. Comparison with Nusselt Theory will be made.

The condensing coefficient and scale coefficient (fowling factor) will be obtained via a Wilson Plot technique. The Wilson Plot (Ref 8) technique utilizes the heat resistance concept which states that the total resistance is a sum of the condensing resistance, a scale resistance, a wall resistance, and a potassium boundary layer resistance. Therefore:

$$\frac{1}{U} = [R_{vapor} + R_{scale} + R_{wall}] + R_{liquid} \tag{1}$$

Only a small error is assumed if $R_V + R_{SC} + R_W = R$. is assumed a constant. The resistance of liquid metal is a function of $V^{.4}$ (From Lubarsky and Kaufman Equation, $Nu = 0.025\ Pe^{.4}$). Therefore equation (1) can be rewritten as

$$\frac{1}{U_o} = R + \frac{1}{CV^{.4}} = \frac{A(LMTD)}{Q} \tag{2}$$

A plot of $1/U$. versus $1/V^{.4}$ should yield a straight line on linear co-ordinates. The intersection of this line at $1/V^{.4} = 0$ should be equal to $R_V + R_S + R_W$. Assuming initially $R_{SC} = 0$ and calculating R_W, R_V can be determined. By repeating the calibration at the termination of the test, any change in R can be contributed to a scale build-up.

10.3 Boiling Test

Once all the calibrations are performed, the boiling test can proceed. The following method will be employed in conjunction with the "Heat Make-up by Guard Heater" method.

1. Set desired flow rate.
2. Heat the potassium to the saturation temperature via radiant preheater and vaporize to any desired quality in the first boiler.
3. Vaporize an additional five percent of the potassium in the second boiler.
4. Vaporize to at least 10F superheat in the third boiler.

Measure the surface temperatures of all three boilers and repeat all three steps with five percent incremental increases in quality in the first boiler until the entire quality range is covered. Pressure drops in all three boilers will be measured and recorded.

10.4 Condensing Test

Since pressure drops in liquid metal condensing and boiling systems are necessary evils in any Rankine Cycle Space Power System, it was decided to obtain additional pressure drop information with which to check the Martinelli-Lockart correlation and in particular the Pratt & Whitney Aircraft pressure drop code. By insuring a constant Q/A for which the code was derived and by knowing the flow rate, the condensing length, the fluid properties, and the quality change, a ΔP can be calculated by theory and compared to by test. The flow rate is determined by the flowmeters described in other sections of this report, the condensing length is established by the temperature profile as measured by wall thermocouples along the condenser tube, and the quality change is known once the inlet quality is known. After correcting the measured pressure drop by subtracting the small calculated single phase pressure drops, theory and test results can be compared.

10.5 Stability Test

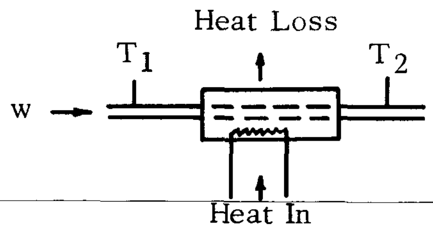
As stated previously, stability is dependent upon many interdependent factors. One cause of instability, noticed in the water proof test and reported by some investigators in the effect of increased loop "sponginess" due to trapped gas pockets. [Unless a vacuum fill technique is employed, it is highly probable that any practical system will have trapped pockets located somewhere in the system.]

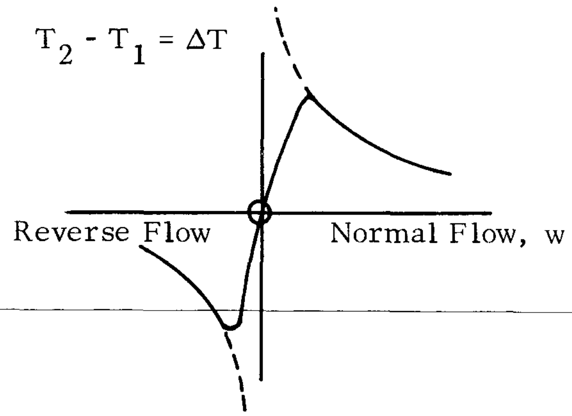
Large gas pockets can be added to the system (outside the main stream) either at the boiler inlet via the accumulator or at the condenser exit via the potassium surge tanks. The effect of these pockets will be indicated by the flowmeters, pressure gauges, or thermocouples.

It is hoped that the qualitative information obtained from this test will aid in understanding the stability problem.

10.6 Thermal Flowmeter Heat Loss Calibration

The thermal flowmeter concept is valid only if the net heat input can be accurately determined. At low flows (i.e., low net heat input required), the heat loss can become a significant part of the net heat input. The graph below indicates a possible effect of a large heat loss.





It becomes important to establish a condition in which the heat loss is known. This is done in two ways:

1. Under normal running conditions, the power to the guard heaters is adjusted to obtain a zero Δt (zero heat loss) across the insulation between the heaters and the flowmeter tube. Heat losses out the ends are negligible.

2. With zero flow and zero immersion heater power, maintain normal operating temperatures with the guard heaters. Any power into the immersion heaters therefore is transferred to the fluid.

   The insulated thermocouples are placed approximately two to three inches (10 to 15 diameters) of isothermal mixing length from the heating section in order to obtain a bulk temperature.

11. EXPERIMENTAL SEQUENCE

The sequence tabulated below includes all necessary steps between the time the last weld is made to the time the boiling heat transfer data is obtained. Detailed operating instructions are being prepared by J. Petrek and will be issued at a later date.

11.1 Rig Preparation

11.1.1 Gas System

a. Check cryogenic system (Ref CS-300A) to ascertain that CS-1933 gas purification specification has been fulfilled.

b. Purge for several hours with purified gas through all gas lines to remove any gaseous impurities.

11.1.2 Potassium System

a. Evacuate loop and fill drain pot to 10.0 microns

b. Back fill with 5 psig purified helium

c. Repeat (a) and (b) several times

d. Evacuate to 10.0 microns

e. Outgas loop at 400F for several hours

f. Repeat (a) through (e) until impurity level falls below 5 ppm oxygen and -90F dewpoint

g. Evacuate to 1200 microns

h. Vacuum or pressure fill potassium fill-drain pot with purified filtered potassium at 600F

11.1.3 Sodium System

a. Evacuate to 100 microns

b. Back fill with 5 psig purified helium

c. Repeat (a) and (b) several times

d. Evacuate to 10.0 microns

e. Outgas cavities at 400F for several hours

f. Repeat (a) through (e) until impurity level falls below 5 ppm oxygen and -90F dewpoint

g. Evacuate to 10.0 microns

h. Vacuum or pressure fill sodium fill-drain pot with purified filtered sodium at 400F

11.1.4 Vacuum fill potassium loop from potassium fill-drain tank till both surge tanks are approximately 1/2 full.

11.1.5 Pressure fill sodium cavities (3) until sodium is indicated in the sodium surge tank.

11.1.6 Set sodium level in the sodium cavities by the procedure outlined in Section 9.2.

11.1.7 Start single phase flow in the potassium system to check out pump.

11.1.8 Check out potassium surge tank calibration procedure.

11.1.9 Calibrate E.M. flowmeter at 800F.

11.1.10 Calibrate thermal flowmeter in the 800F to 900F range.

11.1.11 Dump potassium loop at 800F and fill three times (hot flush).

11.1.12 Hot trap potassium for 24 hours at 1400F after each flush.

11.1.13 Refill loop at 600F (pressure fill) start single phase flow.

11.1.14 Check out I^2R preheater.

11.1.15 Calibrate to obtain condensing coefficients as a function of Q/A and Δt and reduce data via Wilson Plot method.

11.1.16 Reduce flow rate and increase heat into the first boiler (with second and third boiler at zero capacity) until a 10F superheat is noted at the boiler exit. The temperature at the boiler surface is not to exceed 1800F. Establish curve of potassium temperature versus flow rate with 10F superheat and 1800F sodium temperature. This establishes that the limits of the system stability trends can be observed at this time.

11.1.17 Conduct boiling test as outlined in Section 10.3 at the flowrate and temperature obtained in Section 11.1.16.

11.1.18 Repeat test at lower flowrates.

12. DATA REDUCTION

12.1 Boiling Data

12.1.1 Heat Transfer Data

The boiling heat transfer coefficient is calculated from

$$Q_1 = h_B A_{ID} (Tw-Ts)$$

where Q_1 is equal to the heat supplied to the immersion heaters in the test boilers when the "Heat Make-up by Guard Heaters" technique is used, A_{ID} is the surface area of the tube based on the inside diameter, t_s is the saturation temperature of the potassium averaged between inlet and outlet conditions and obtained by measuring the temperature and pressure at the inlet and outlet. The wall temperature (Tw) is obtained by subtracting an experimental condensing Δt and a calculated wall Δt from the temperature measured on the outside surface of the cavity pressure shell.

The quality level entering the test section is determined via

$$Q_1 = mc\,(ts-ti) + mLx$$

where Q_1 is the heat supplied to the immersion heaters in the work horse boiler when the "Heat Make-up by Guard Heaters" technique is used. The flowrate (m) is determined by the various flow measuring schemes, ts is the saturation temperature as determined by temperature and pressure measurements, ti is the measured inlet temperature, L is the latent heat at saturation conditions, and Cp is the specific heat at constant pressure, and X is the quality at the outlet of the "work horse" boiler. The quality change in the test section is determined by

$$\Delta Q_2 = mL\Delta X$$

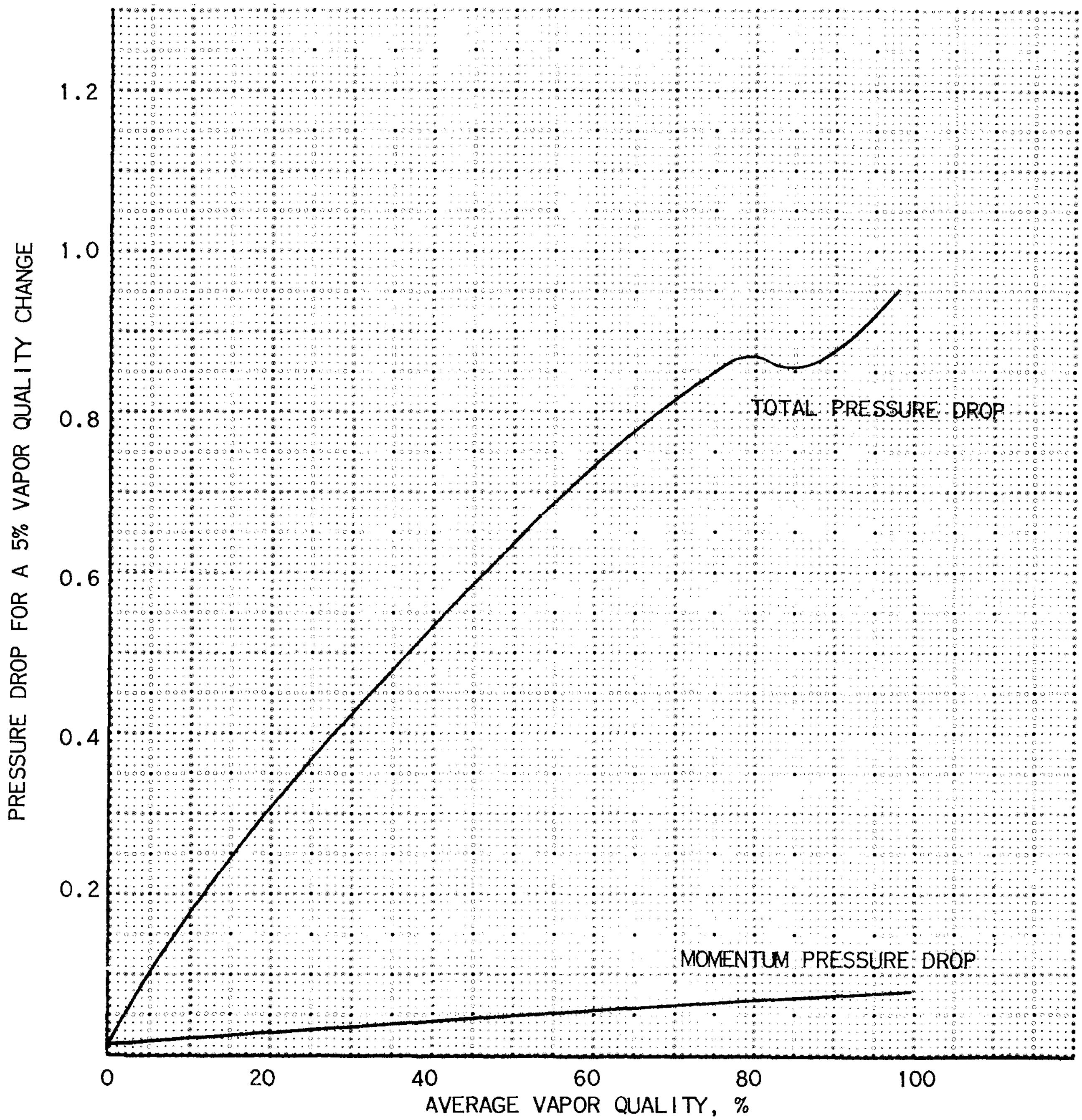
where ΔQ_2 is the heat added to the boiling potassium in the second cavity, m is the mass flow rate of the potassium, L is the latent heat of vaporization at saturation conditions, and ΔX is the quality change. Data reduction will be via hand and machine (Bendix) compulations. Correlations with existing information will be attempted.

12.1.2 Pressure Drop Data

Pressure drop across all boilers will be measured and compared with predicted pressure drops. Fig 14 presents the thoretical prediction for pressure drop in a straight tube for five percent incremental charges in quality for any quality from zero to 100 percent.

FIG 14

PRESSURE DROP CHARACTERISTICS FOR 1700F POTASSIUM UNDERGOING A 5% QUALITY CHANGE AT 20 LB/HR FOR A TUBE LENGTH OF 9 INCHES AND A TUBE INSIDE DIAMETER OF 0.186 INCH



12.2 Condensing Data

12.2.1 Heat Transfer Data

The over-all heat transfer coefficient is calculated from

$$Qc = UcAc\ (\Delta T)$$

where Qc is the heat rejected by the condenser for any quality change. Ac is the area required to accomplish this change as determined by condenser temperature profile and the ΔT is determined by taking the difference between the average saturation temperature and the average air temperature. Fig 15 presents theoretical pressure drops in the condenser.

12.2.2 Pressure Drop Data

The condensing pressure drop is determined from

$$\Delta P_c = \Delta P_m - (\Delta P_{friction})_{superheat\ section} - (\Delta P_{friction})_{subcooled\ section}$$

where $(\Delta P)_{superheat}$ is the calculated pressure drop experienced by the potassium from the point of pressure measurement to the point at which the liquid is cooled to 100 percent quality and $\Delta P_{subcooled}$ is the calculated pressure drop experienced by the potassium to the point of pressure measurement from the point at which the quality was zero percent. ΔP_m is the measured pressure drop. Data reduction will be via hand and machine (Bendix) calculations.

12.2.3 Correlations

Heat transfer and pressure drop correlations will be attempted. The use of the boiling number shows promise for boiling correlations, at least up to approximately 60 percent quality. The Lockart and Martinelli correlations will be compared with test results.

12.3 Stability Data

The information obtained from this test is generally of a qualitative nature. However, if throttling the boiler inlet and/or condenser outlet induces stability, this pressure drop across the throttling devices can be recorded and compared with the theoretical pressure drop required. Qualitative information on the effect of gas pockets at the boiler inlet and condenser exit will be recorded. Pressure fluctuations can be recorded if considered desirable.

12.4 Material Information

Haynes alloy No. 25, stainless steel, potassium, and sodium samples will be analyzed and photographed before and after the test in order to obtain a record of the possible change in material analysis and surface conditions.

PRESSURE DROP CHARACTERISTICS FOR 1700F POTASSIUM CONDENSING INSIDE A 0.186 INCH DIAMETER ROUND TUBE AT 20 LB/HR CONDENSING FROM 100% QUALITY

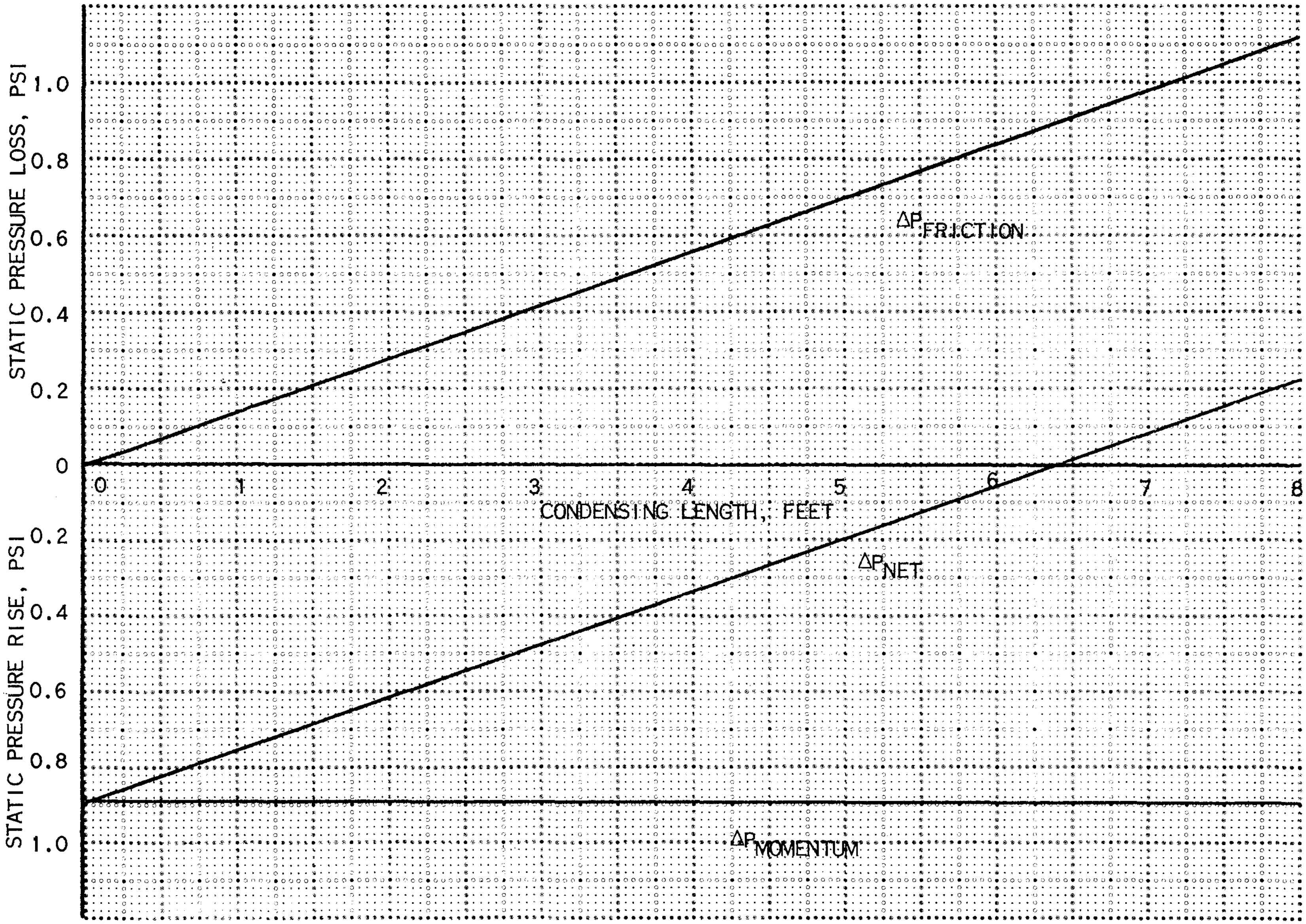


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FIG 15

13. CONCLUSION

The test rig described in this report is designed to obtain design type heat transfer and hydrodynamic information from which a high temperature potassium boiler can be properly sized. It is expected that the rig described in this report will also enhance Pratt & Whitney Aircraft's knowledge of boiling liquid metals systems as well as liquid metal boilers. The complete design of liquid metal boilers however, requires a thorough knowledge of material compatibility, liquid metal purity requirements and system requirements.

Evidence has also been presented in this report indicating the effect of the many variables involved. Due to the effect of the heater surface, heater material, scale, and pressure on the boiling heat transfer coefficient, it is felt by the writer that the final design of the boiler will evolve with extensive testing of full scale prototype boilers operating at the design temperature levels with the design materials under design conditions.

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15. NOMENCLATURE

Letters		Units
t	Temperature	F
V	Velocity	ft/sec
h	Heat Transfer coefficient	Btu/hr ft^2 F
h	Enthalpy	Btu/hr
k	Thermal conductivity	Btu/hr ft F
D	Diameter	ft
G	Mass flux	lb/hr ft^2
Q	Heat rate	Btu/hr
A	Area	ft^2
μ	Viscosity	lb/hr ft
ρ	Density	lb/ft^3
q	Heat flux = Q/A	Btu/ft^2
L	Length	ft
J	Mechanical heat equivalent	ft lb/Btu
λ	Latent heat of vaporization	Btu/lb
c	Specific heat	Btu/lb
P	Pressure	psf
g	Gravity constant	ft/sec^2
R	Thermal resistance	hr ft^2 F/Btu
m	Flow rate	lb/hr
γ	Surface tension	dynes/cm
μ'	Viscosity	centipoise

Subscripts

w	Wall
s	Saturation
B	Boiling film
tt	Turbulent liquid-turbulent vapor
l, f	Liquid
g	Gas

Subscripts

fg	Properties in two phase regime
tpf	Two phase flow
c	Condense
sc	Scale
v	Vapor

Dimensionless Parameters

Re	Reynolds Number
Pr	Prandl Number
Nu	Nusselt Number
B_o	Boiling Number
x	Vapor quality (weight fraction of vapor)
X	Martinelli-Lockart parameter
F	Form factor
σ	Emissivity
λ'	$[(\rho_g/0.075)(\rho_l/62.3)]^{1/2}$ (Baker Chart)
ψ	$73/\gamma\ [\mu'_L (62.3/\rho_l)^2]^{1/3}$ (Baker Chart)

16. APPENDIX

16.1 Results of Water Proof Test

A water proof test of a boiler section simulating the test section boiler was tested to:

1. Check venting technique and the effect of venting on the reproducibility of the condensing coefficient.

2. Check the effect of non-condensibles in the water vapor on the condensing coefficient.

16.1.1 Reproductability

As a result of these tests it is concluded that the venting technique appears successful in that the results are reproductable when the venting operation is carried on for a sufficient period of time with vigorous boiling. Fig 16 is a graph of Q/A versus Δt which includes a comparison with a theoretical analysis by Nusselt. The difference between the test and theory is a constant temperature difference of 6F which has not been accounted for. The data can also be reduced to the arbitrary equation

$$Q/A = 1.85 \left(\frac{R}{D}\right) \left(\frac{g\rho^2 \lambda D^3}{\mu k}\right)^{1/4} (\Delta t)^{1/2}$$

Further tests may be conducted to determine the origin of the 6F error and to further confirm reproductability.

16.1.2 Effect of Non-Condensibles

An example of the effect of non-condensibles on the condensing coefficient, occurred in one of the first tests which indicated a very large ΔT_c of 108F. When the venting was continued for a longer period of time, the ΔT_c (at constant Q/A) was 35F which checked quite closely with theory and was reproductable at different occasions.

16.2 Estimate of Immersion Heater Capability

The heater capability is dependent upon the environment and the properties of the heater materials. An analysis of heater temperatures requires a knowledge of:

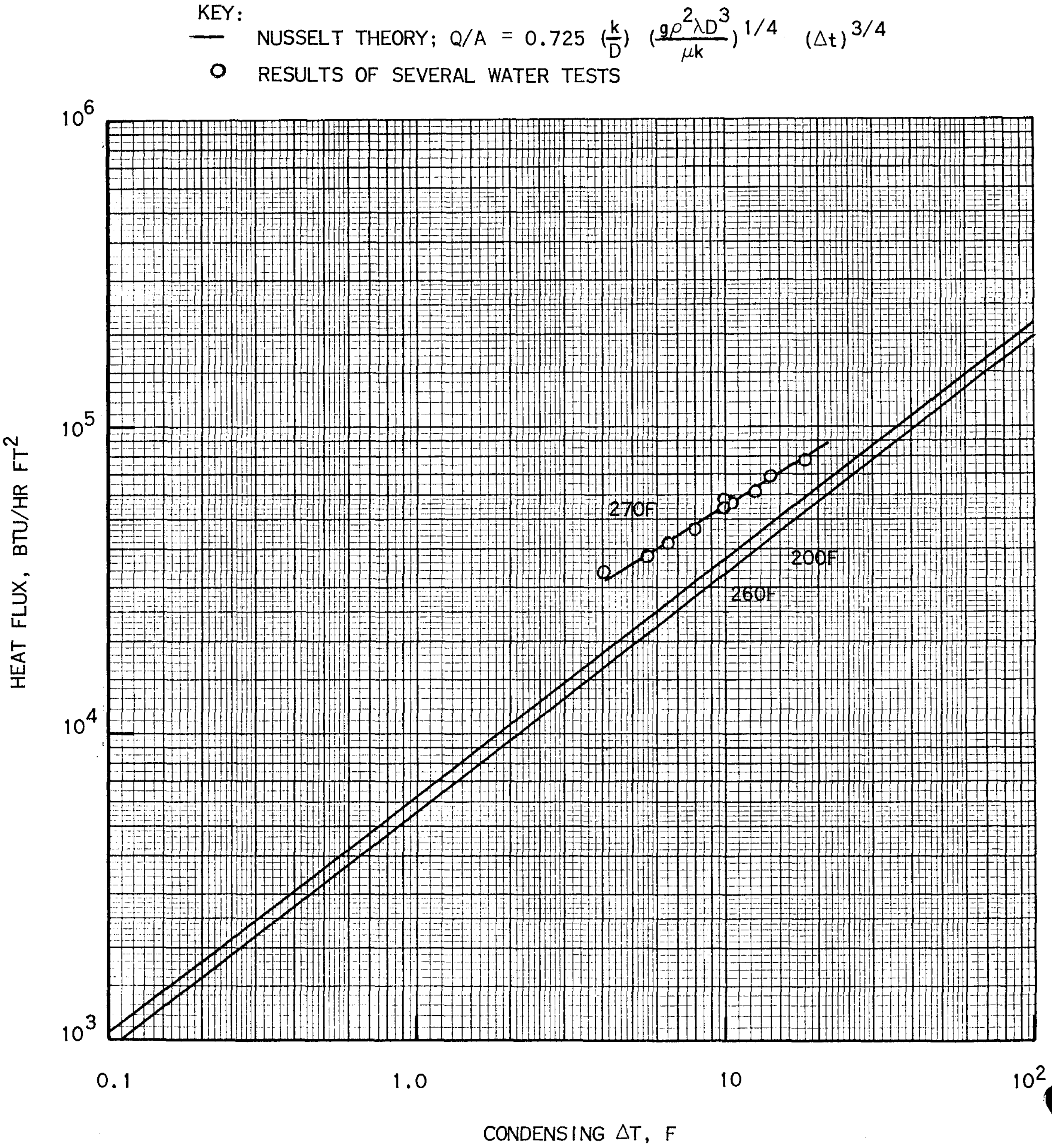
1. Burnout heat flux for a horizontal tubular heater in a boiling sodium environment. (To obtain limiting Q/A).

2. ΔT due to all the thermal resistances in the heater.

3. ΔT due to all the thermal resistances outside the heater.

FIG 16

COMPARISON OF NUSSELT THEORY AND TEST DATA FOR WATER CONDENSING ON A 1/4 INCH DIAMETER TUBE



Item 1 was obtained from Ref 19 which describes the results of burnout tests of a Mo-Al_2O_3 tubular heater in a sodium bath. See Fig 17 for the results of this data extrapolated to a burnout heat flux of 840,000 Btu/hr ft^2. Their minimum data extrapolates to 460,000 Btu/hr ft^2 at the operating pressure of this rig. A maximum heat flux of 100,000 Btu/hr ft^2 was assumed to be a safe (4.6 safety factor) heat flux for this rig.

Item 2 was evaluated via Fourier conduction equation, assuming no contact resistances between the MgO insulation, the sheath, or the Kanthal wire. A MgO conductivity of 3.7 Btu/hr F ft was used. A Haynes alloy No. 25 conductivity of 13 Btu/hr F ft was obtained from Ref 20.

Item 3 was evaluated using the Forster-Grief correlation for nucleate boiling.

Fig 18 presents the results of temperature drop calculations. At a maximum heater heat flux of 100,000 Btu/hr ft^2 the Kanthal centerline temperature is 330 degrees higher than the pool temperature. At a pool temperature of 1850F, the maximum Kanthal temperature is 2180F which is 280F below the maximum recommended operating temperature and about 570F below the melting point of the Kanthal heating wire.

16.3 Preheater Analysis

16.3.1 Preheater Flow Regime

The function of the preheater is to preheat the fluid from an initial temperature of 900F to 1600F or as high as 1750F. In as much as some investigators have found the degree of subcooling to be a factor in promoting instability (presumed to be due to subcooled nucleate boiling), it was desirable to prevent boiling in the preheater section, or to at least predict if boiling did or did not occur in the preheaters. To this end, a curve of Q/A versus ΔT was constructed in the forced convection regime using

$$Pr = 4.8 + 0.025\, Pe^{0.8}$$

at several temperatures (subcooled by zero F, 17F, 25F, and 100F) and in the nucleate boiling regime using the Forster-Grief correlation at 1400F, 1600F, and 1800F. The intersection of the forced convection curve and boiling curve defines the approximate point at which nucleate boiling begins. From these curves (Fig 19), it can be established that at 19,200 Btu/hr ft^2 (radiant heater input at 20 lb/hr), the potassium will boil (subcooled nucleate boiling) in the preheater if subcooled less than 6F to 7F. At 149,000 Btu/hr ft^2 (I^2R heater input at 350 lb/hr), the potassium will boil in the preheater if subcooled less than 18F to 25F.

16.3.2 Radiant Preheater Analysis

The radiant preheater is used to heat the potassium to the proper temperature level during the boiling runs (low flow). An analysis was performed to determine the proper preheater length. This analysis is included below and the results are shown in Fig 13.

FIG 17

BURNOUT HEAT FLUX FOR A TUBULAR HEATER IN A SODIUM BATH

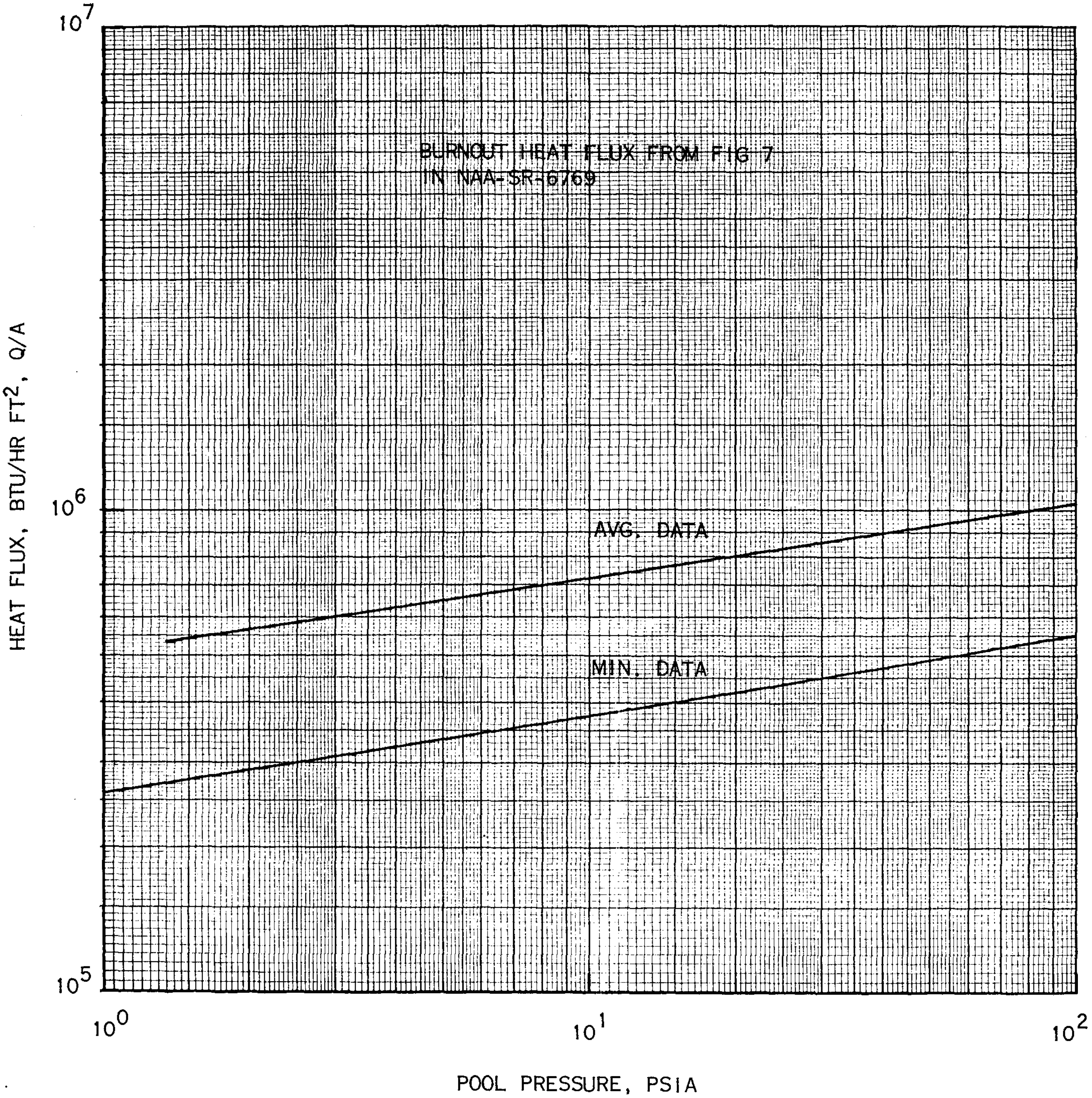


FIG 18

EFFECT OF HEAT FLUX ON IMMERSION HEATER TEMPERATURE DIFFERENCES

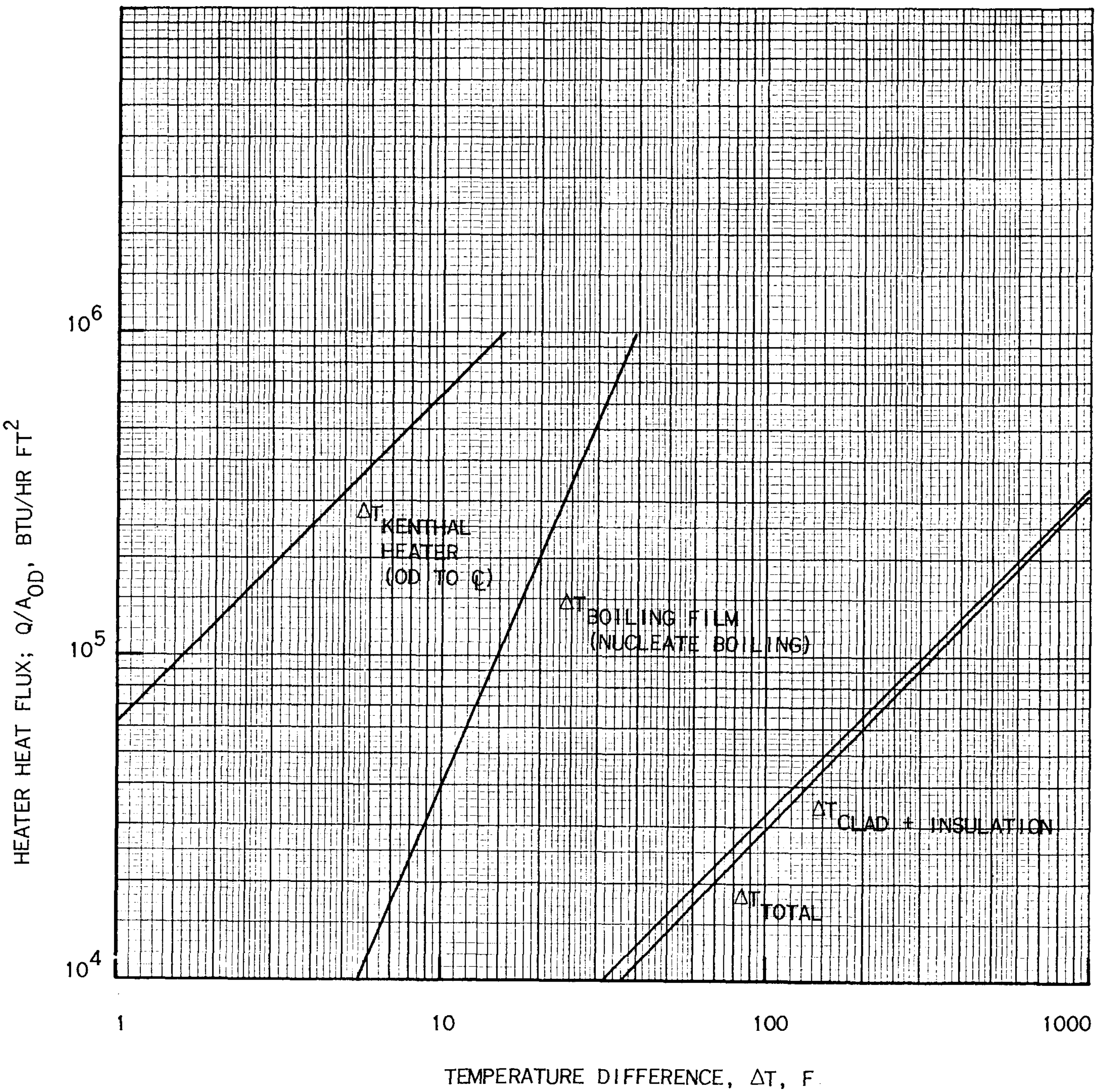
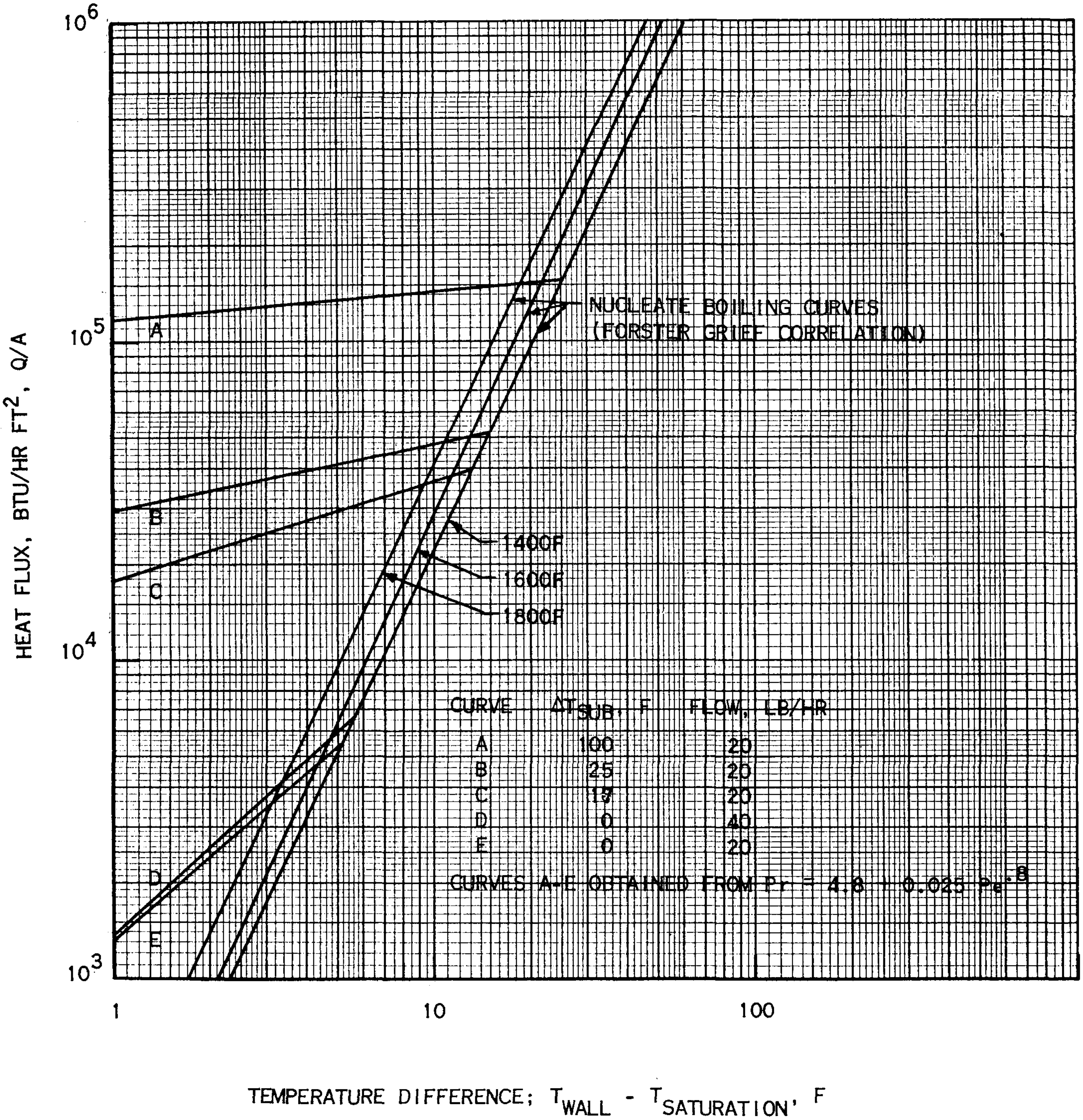


FIG 19

EFFECT OF SUBCOOLING ON THE HEAT FLUX REQUIRED TO INITIATE NUCLEATE BOILING OF POTASSIUM



1. The radiant heat flow into the tube is represented by

$$Q_{net} = F\sigma A_s (T_h^4 - T_t^4) \tag{1}$$

for the entire length of heater or

$$(Q_{\Delta x}) = F\sigma \ \Delta(A_s) \ (T_h^4 - T_T^4) \tag{2}$$

for an incremental length Δx.

2. Heat flow into the tube equals heat flow into the potassium

Therefore:

$$(Q_{\Delta x}) = mc \ \left[(T_F)_{x + \Delta x} - (T_F)_x \right] \tag{3}$$

and

$$T_T = T_F + \Delta T_D \tag{4}$$

where

a. $Q_{\Delta x}$ is the heat added to the potassium in the length Δx

b. T_h is the temperature of the heater

c. T_T is the temperature of the tube

d. T_F is the temperature of the fluid

e. ΔT_D is the sum of the wall and potassium boundary layer temperature drop

where the subscripts indicate the location along the tube.

3. Equating (2) and (3) and combining with (4) yields

$$\frac{mc}{F\sigma A_s} = \frac{T_h^4 - (T_F + \Delta T_D)^4}{(T_F)_{x + \Delta x} - (T_F)_x}$$

where

$$A_s = \pi D(\Delta x)$$

4. Constants:

$\sigma = 0.174 \times 10^{-8}$ Btu/hr ft^2 F^4

D = 0.0312 feet

Δx = 0.25 feet

T_i = 2660R (2200F)

$$F = \text{Form Factor} = \frac{1}{1/\epsilon_1 + D_1/D_2\,(1/\epsilon_2 - 1)}$$

where: ϵ_1 = 0.70

D_1 = 0.375

ϵ_2 = 0.38

D_2 = 1.5

$(T_F)_{x=0}$ = 900F

m = 20 lb/hr

ΔT_F = 20.5F

c_p = 0.185 Btu/lb F

This equation was programed on the Bendix computer. The results of this analysis are shown in Fig 13.

16.4 Boiler Calculations

Paradoxically, the proper sizing of the length of a boiler tube required to vaporize potassium from zero to 100 percent quality required the very information which this test is designed to produce. The boiler tube length is therefore, calculated from an assumed function of local heat transfer coefficient and a local vapor quality for a LMTD of 100F. Fig 8 illustrates several assumed curves. The boiler tube length required to vaporize to 100 percent quality varies from 2.7 feet to 29 feet depending upon the assumed curve. The final length selected in the first boiler is 4.75 feet. This length is based on:

1. Curve ABE in Fig 8 which was assumed the most reasonable plus a preheating length (six inches) and a safety factor.

2. An excessive pressure and temperature drop for longer lengths (Figs 2 and 20).

It is valuable to point out the very strong effect of the gas (100 percent vapor quality) coefficient and the shape of the curve in the high quality region on the boiler tube length. For curve ABE, 31.6 percent of the boiler tube length is required to go from 95 percent to 100 percent vapor quality, while only 21.8 percent of the boiler tube length is required to go from zero to 50 percent vapor quality. If the gas coefficient were constant from 95 percent vapor quality to 100 percent vapor quality, it would take 28 feet of tubing to go from 95 percent to 100 percent vapor quality.

EFFECT OF BOILER LENGTH ON FRICTIONAL PRESSURE DROP FOR A ZERO TO 100% VAPOR QUALITY CHANGE IN A 0.186 INCH DIAMETER BORE TUBE AT VARIOUS FLOW RATES FOR 1700F POTASSIUM