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NAA-SR-6304

TRW REPORT No. ER-4521

THE SNAP II POWER CONVERSION SYSTEM

TOPICAL REPORT No. 12

BOILER DEVELOPMENT

AEC RESEARCH AND DEVELOPMENT REPORT

PREPARED UNDER SUBCONTRACT
N 843 FS-101221 FOR ATOMICS
INTERNATIONAL, A DIVISION OF
NORTH AMERICAN AVIATION, INC.

TAPCO a division of
Thompson Ramo Wooldridge Inc.
CLEVELAND, OHIO

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512-148252-88

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Prepared under Subcontract N843FS-101221 for
Atomics International, A Division of North American Aviation Inc.

DATE

Published April 9, 1961
Prepared July 17, 1961

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8868

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ABSTRACT

SNAP II is the designation for a 3 KW auxiliary power unit to be used in a satellite vehicle. The SNAP II system consists of a reactor heat source, a mercury Rankine engine, and an alternator. The heat source and Rankine engine are connected by a primary fluid-to-mercury heat exchanger, or boiler. In the boiler, liquid mercury is converted to vapor to drive the turbine. This report summarizes boiler test results and design requirements. This work was performed under a subcontract to Atomic International as part of the Atomic Energy Contract AT (11-1)-GEN-8.

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1.0 INTRODUCTION

The SNAP II auxiliary power unit for a 3 kw electrical output consists of an Atomic International reactor heat source and a Thompson Ramo Wooldridge power conversion system. Thermal energy is transferred from the reactor to the power conversion system by a liquid metal primary fluid. Sodium, the initial primary fluid, has been replaced by NaK as a result of subsequent component investigations.

The major components of the closed Rankine cycle power conversion system are the NaK-heated once-through forced convection mercury boiler, an axial flow impulse turbine which extracts energy from mercury vapor, a condenser which returns the vapor to a liquid state and a mercury pump which supplies the condensed liquid to the boiler. The turbine is mounted on a single shaft with an alternator to generate 3 kw of electrical power, a mercury pump, and a permanent magnet induction NaK pump. The entire shaft assembly is enclosed in one housing and referred to as the combined rotating unit (CRU).

Computer simulation studies of the SNAP II system performed at TRW included an extensive non-linear study of the boiler which defined the interplay of preheat, boiling, and super-heat regions during boiler transients. The results are presented in "SNAP II Topical Report No. 3, Dynamic Analysis, TRW Engineering Report No. 3911".

For the most rapid attainment of a higher performance boiler to the established specifications the program has emphasized the continuous development of primary fluid-to-mercury heat exchanger test sections and full-scale boilers. Preliminary study led to the conclusion that helical configurations of once-through boilers would best satisfy the requirements. Accordingly, two test sections and one full-scale boiler have been built and tested. Information from the SNAP I program was also utilized. The test data were analyzed and used to develop a mathematical model from which were deduced parameters characteristic of mercury forced convection boiling systems. Test data were then plotted on the basis of these parameters and extrapolated to improved boiler designs.

A research program complementing the experimental approach was initiated in May 1960 to establish local boiling mercury heat transfer coefficients as a function of temperature differences, pressure, mass velocity, quality and wetting. These data will provide the detailed and comprehensive local heat transfer information necessary for advanced design boilers capable of high performance in space environment.

This report covers SNAP II boiler development efforts to September, 1960. A major portion of the work reviewed in this report was accomplished during the 1959 fiscal year.

2.0 SUMMARY

The SNAP II boilers which have been designed to date are summarized in Table I. As shown by test results from the three boilers which have been tested, a continuous progress in design has been achieved. These designs have been based on test data from both the SNAP I and SNAP II programs. As the quantity of data has increased, physical models describing the heat transfer process have been developed. These physical models provide the necessary correlation parameters which permit the extension of existing data to advanced design.

Preliminary Test Sections 1 and 2 were designed on the assumption that an all-vapor model which ignores the presence of the liquid phase during forced convection boiling could be used to describe the process quantitatively. The conventional Dittus-Boelter equation was applied with the increase in the vapor flow along the tube being ascribed to liquid evaporation. As shown in Table I, the assumption led to a design that fell short by about an order of magnitude since the exit qualities were only in the range of 10 percent, far less than required for complete vaporization. As a result, a revision in the concept of the mechanics of boiling was found necessary and a theoretical analysis was formulated, based on a "dry wall" or "dropwise"* type boiling phenomenon.

The test results of the Preliminary Test Sections and the SNAP I boiler were plotted on the basis of dry-wall boiling parameters containing the area mean temperature difference and mass velocity. A conservative design curve was established and used to design the Thirteen Tube Boiler. The design was found by test to be conservative, and the measured performance and the degree of conservatism were found to be within the expected spread in earlier test data.

Dropwise boiling pictures the heat transfer as occurring directly from the wall to the drop through a film created by the vapor being ejected from the underside of the drop. The drop is held against the wall by its inertial force induced by a swirl device. Heat transfer experiments performed with mercury droplets provided a more detailed understanding of the mechanics of dry-wall boiling. The theory thus developed compared favorably with the test data.

A boiling research program has been initiated to refine the design procedures presently available. In this program the detailed heat transfer data can be derived relative to a greater number of controlled variables than are attainable in prototype boiler tests.

The primary efforts of the boiler development program have been supplemented by other related work. These include porous bed boiler studies, mercury droplet boiling on hot plates in air, isothermal nitrogen - liquid mercury two-phase flow tests and sodium-nitrogen heat transfer tests. From these tests information has been obtained on the effect of various inserts on two-phase pressure drop, the phenomena of heat transfer to drops, and the effect of primary fluid side parameters on calculation of the mercury side parameters.

*i.e., heat transfer associated with a two-phase flow pattern consisting of drops dispersed in a continuous vapor phase.

TABLE I
SNAP II FORCED CONVECTION BOILERS (ALL HELICALLY COILED)

Boiler Design	Preliminary Test Section No. 1 (Plain Tube)	Preliminary Test Section No. 2 (Twisted Ribbon Insert)	Thirteen Tube Boiler	PCS I Boiler
Test Date	May, 1959	July, 1959	March, 1960	Untested
Type	Counter-flow	Counter-flow	Counter-flow	Counter-flow
Heat Source	Na	Na	Na	NaK
No. of Tubes	1	1	13*	7
Length	34 ft	34 ft	110 ft	62 ft
Tube O.D.	1/2 in	1/2 in	5/16 in	5/16 in
Tube Wall	0.049 in	0.049 in	0.035 in	0.028 in
Shell O.D.	1 5/16 in	1 5/16 in	1.900 in	1.900 in
Shell Wall	0.049 in	0.049 in	0.065 in	0.065 in
Internal Configuration	None	0.040 x 0.40 in Pitch Twisted Ribbon	0.051 in Swirl Wire 1 in Pitch	0.054 in Swirl Wire, 1 in Pitch
Helix Diam	17 3/4 in	17 3/4 in	19 1/2 in	19 1/2 in
No. of Coils	8 1/2	8 1/2	21 1/2	11 1/2
Weight:				
Dry	38 lb	42 lb	338 lb	160 lb
Primary				
Fluid Inv.	12 lb	12 lb	130 lb	50 lb
Hg Inv.	1 lb	1 lb	—	50 lb

PERFORMANCE OF BOILING SECTION AT DESIGN Hg FLOW AND PRIMARY FLUID TEMPERATURES

Design Hg Flow	16.0 lb/min	16.0 lb/min	17.0 lb/min	18.6 lb/min
Exit Quality	0.098	0.22	1.0	--
Heat Flux	4,160 Btu/hr-ft ²	12,600 Btu/hr-ft ²	6,000 Btu/hr-ft ²	--
Coefficient	8 Btu/hr-ft ² -°F	33 Btu/hr-ft ² -°F	45 Btu/hr-ft ² -°F	--
ΔT_m	486 °F	382 °F	135 °F	--
ΔP_{Hg}	6 psi	34 psi	101 psi	--

* Six valves allowed variation from 13 to 7 tubes.

3.0 SPECIFICATIONS

The SNAP II boiler, heated by the reactor coolant, converts liquid mercury into the superheated vapor which drives the turbine. Specifications for the Initial SNAP II boiler were based on the operating requirements established by the selection of cycle operating conditions.* Slight modification of these requirements established the following design conditions used to construct the Preliminary Test Sections 1 and 2:

Sodium Inlet Temperature	1200 °F
Sodium Outlet Temperature	1000 °F
Sodium Flow	44 lb/min
Sodium Pressure Drop Allowed	2 psi
Mercury Inlet Temperature	425 °F
Mercury Outlet Temperature	1150 °F
Mercury Inlet Pressure	140 psia
Mercury Outlet Pressure	100 psia
Mercury Flow	16 lb/min

As the system was evolved, changing component power requirements necessitated an increase in the quantity of energy from the boiler to drive the turbine. As shown below, flow and pressure conditions were changed and served as the design specification for the Thirteen Tube Boiler.

<u>Parameter</u>	<u>Specification</u>	
	<u>Original</u>	<u>Altered</u>
Sodium Flow	44 lb/min	46 lb/min
Mercury Flow	16 lb/min	17 lb/min
Mercury Outlet Pressure	100 psia	105 psia

The change in the reactor coolant from sodium to NaK further modified boiler requirements. The lower specific heat of NaK necessitated an increased flow rate of the primary fluid. An increased mercury flow was required to supply the additional quantity of vapor to the turbine because of the increased power demanded by the NaK pump.

*See Design Point Determination, SNAP II Topical Report No. 1, Thompson Ramo Wooldridge Inc., Cleveland, Ohio, ER-3619, AE-61A.

Because performance of the permanent magnet pump is lowered by the increased resistivity of NaK and the lower specific heat requires increased flow, a NaK pump requires approximately three times more power than a sodium pump. Table II compares the Thirteen Tube Boiler specifications to those used to design NaK-heated boilers for Power Conversion Systems I and II. Other implied boiler specifications require that the boiler have the capability of vehicle launch, zero-gravity operation, no production of net disturbing torques and long-life unattended operation.

TABLE II
COMPARISON OF BOILER SPECIFICATIONS

Parameter	Heat Source	
	Sodium *	NaK
Inlet Temperature	1200°F	1200°F
Outlet Temperature	1000°F	1000°F
Coolant Flow	46 lb/min	71.5 lb/min
Coolant Pressure Drop Allowed	2 psi	2 psi
Mercury Inlet Temperature	425°F	430°F
Mercury Outlet Temperature	1150°F	1150°F
Mercury Inlet Pressure	140 psia	230 psia
Mercury Outlet Pressure	105 psia	115 psia
Mercury Flow	17 lb/min	18.6 lb/min

* Designed to the altered specifications

4.0 DESIGN CONSIDERATIONS

4.1 Selection of Boiler Type

Flow arrangements and the placement of boiler tubes were the primary considerations in choice of boiler type. Recirculation and once-through forced convection boilers were reviewed to determine the best design configuration for the SNAP II boiler.

The design configuration selected is the inherently simple and compact once-through forced convection boiler which avoids the complications of recirculating pumps and separators and minimizes mercury inventory. A shell-and-tube arrangement of flow passages is used, with mercury flowing through the tubes and primary fluid through the shell. The pressure level of the primary fluid is considerably less than that of mercury (i.e., 300 to 25 psia), permitting a shell of minimum thickness. The tube wall thickness can be minimized and the mercury coefficient can be increased by use of small tubes. Minimum tube supports are used in the liquid metal primary fluid passages as pressure drop is at a premium and the mercury side coefficient is the controlling heat transfer resistance.

For the shell-and-tube configuration, large ratios of length to diameter were used in order to have better assurance that boiling was complete and liquid was not carried over. A small number of tubes could also be used, thereby minimizing tube-to-tube sheet fabrication problems. All prototype boilers were helically coiled (e.g., see Figure 25). Flows of the primary fluid and mercury flows may then be easily routed to minimize system gyroscopic torques. A symmetrical distribution of weight is also obtained.

The helical coil configuration permits differential thermal expansion between the tubes and shell by proper location of tubes during fabrication so that mechanical stresses are minimum at operating conditions. The specific helix diameter was governed by considerations of system integration with the vehicle.

4.2 Selection of Materials

The prime criteria for selection of materials are (1) compatibility with the working fluids and (2) high temperature strength. Workability and cost are also considerations, but the corrosiveness of mercury in conjunction with a closed system and the lightweight, long-life requirements make the enumerated considerations of major importance. In support of the boiler and other component programs an extensive materials investigation is being conducted by the Materials Research Division of TRW. A primary emphasis of this investigation is the determination of materials compatibility by test at or near operating conditions. Early results established certain of the 300 series stainless steels as preferred materials from the compatibility aspect. Continuing investigations are yielding materials which are superior in both corrosion resistance and high temperature strength to the materials initially established.

4.3 Additives

Additives have not been utilized in the once-through forced convection boiler designs discussed in this report for the following reasons:

1. Use of additives may pose significant problems from the viewpoint of overall system reliability. In the boiler itself a performance degradation with time will result if the additives deteriorate in their wetting effectiveness. With the current non-wetting design approach, an improved wetting surface with time leads to an increasing margin of performance. Formation of precipitants in the bearing and other small clearance components can adversely affect their performance characteristics. Moreover, from considerations of hydrodynamic stability of the condenser and subcooler, the non-wetting characteristics of mercury prove to be an advantage. However, the desirability of additives to mercury is in continual evaluation as their effects on component and system reliability become more clearly defined.
2. High temperature differences were expected in the boiling section (see Figure 1). Indications are that film boiling would then occur and additives would have been of limited benefit as shown in Figure 2.

Decreasing the temperature difference to make additives more effective would not significantly improve the heat flux in the high quality regions as film or dry wall boiling would occur regardless of wetting characteristics. This phenomenon has been observed in boiling tests using mercury with additives (see Figure 3) and in forced-circulation water (a wetting fluid) evaporators.*

4.4 Fabrication and Inspection Procedures

The prime requirement imposed on the ultimate (vehicle) SNAP II boiler is reliability. Since the reliability of a product is determined by the mechanical design and the fabrication and inspection procedure employed, these areas have received major emphasis. All boiler parts subject to pressure are designed with minimum thicknesses in accordance with Section VIII of the 1952 ASME Code for Unfired Pressure Vessels, the 1955 ASA Code for Pressure Piping B31.1, and the Liquid Metals Handbook, TID 5227. Conservative values were used for design temperatures and pressures. Currently available fabrication equipment and processes have established design values well above the minimum strength requirements for reliability.

From a mechanical design viewpoint, a most critical consideration is thermal stress and cycling in the mercury inlet region. In this area a fluid temperature difference of 600 to

*McAdams, W.H., Heat Transmission, Third Edition, New York: McGraw-Hill Book Co. Inc., 1954, p. 398.

800°F is possible across the tube sheets and tube walls (See Figure 1). A 700°F temperature difference results in a 20,000 psi thermal stress in stainless steel and a thermal cycling condition could exist due to unstable film boiling of the liquid mercury entering the boiler. The severity of these conditions has been minimized by design. For example, Figure 4 shows a primary fluid stagnant zone created by a loose fitting baffle plate in which the temperature difference between the NaK and the mercury across the tube sheet is kept within safe values from a thermal stress viewpoint. The stagnant zone also heats the mercury to the boiling temperature so that an unsafe temperature difference is never reached. Thermal stress and cycling are not so critical in the boiling and superheating regions because the high heat conductance of the primary fluid in conjunction with the low conductance of mercury vapor results in a small temperature difference across the separating structures. Detailed thermal stress considerations are required of all critical areas. Other possible critical areas are (1) joints of dissimilar metals which expand at different rates with temperature and (2) mechanical stresses due to thermal expansion.

Mechanical Internal design Improvements have been made by using boiler tube supports and spacers which have a minimum effect on primary fluid pressure drop and boiler reliability. Where internal design alternatives were possible, the selection was dictated by consideration of ease of fabrication and installation. The major fabrication problems in the NaK-to-mercury heat exchanger used in SNAP II are choice of welding techniques, coiling of the shell-and-tube heat exchanger, and maintenance of tube spacing during fabrication. Reliable welding techniques are insured by employing qualified welders who are required to conform to rigid welding specifications.

The tube-and-shell assembly is coiled by a coiler specifically made for the required configuration. Procedures to maintain the spacing from tube to tube and to the shell have been established. Radiographs of the completed assembly show that these procedures have resulted in good location of the tubes within the coiled shell. Minimum allowable wall thickness and bend radii required of the specific shell sizes must be considered in the design of the coiled assembly. A minimum tube diameter of 1/8 inch has been established because smaller sizes pose welding problems and plugging possibilities.* Reliability considerations also tend to dictate that the minimum number of tubes required for performance be used so that the number of tube-to-tube sheet welds are minimized.

*Fraas, A.P., "Design Precepts for High-Temperature Heat Exchangers". Preliminary copy of this report obtained from Mr. Fraas, P.O. Box Y, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

5.0 TEST AND DATA-REDUCTION PROCEDURES

The SNAP II boiler development test program has concentrated on seven major areas:

1. Overall performance of once-through forced convection mercury generators as defined by the area required to produce an exit quality of one-hundred percent, overall heat transfer coefficients and pressure drop.
2. Distribution of heat transfer surface areas in the preheating, boiling and superheating zones.
3. Determination of heat transfer coefficients of mercury under preheating, boiling and superheating conditions.
4. Mercury side hydrodynamic stability.
5. Effect of long-time operation.
6. Startup and shutdown operating procedures.
7. Effect of thermal cycling on material.

Data obtained in boiler tests are continuously applied to advance design techniques.

5.1 Test Procedures

A full-scale test rig using the SNAP II liquid metals was designed and fabricated to test SNAP II combined rotating units and SNAP II boilers. The original system used sodium, but eutectic NaK-78 was used in later tests. Figure 5 is a schematic of the test rig and instrumentation and Table III lists the capabilities of this performance test rig. Figure 6 shows the Preliminary Test Section No. 2 installed with a complete cover of 6 inches of Johns Mansville "cerafelt" insulation. All SNAP II boilers tested were insulated in this manner.

TABLE III

CAPABILITIES OF PERFORMANCE TEST RIG

Thermal Input	100kw
Maximum Sodium Temperature	1400°F
Maximum Sodium Pressure	100 psia
Sodium Flow	60 lb/min
(Converted to NaK flow)	100 lb/min
Mercury Pressure	300 psig
Mercury Flow	24 lb/min
Minimum Condenser Pressure	2 psia

The general test instrumentation used was common for liquid metal systems and is itemized in Table IV. The location of this instrumentation is generally described in the test loop schematic, Figure 5. The specific location of terminal temperature and pressure taps is typified by Figure 42 showing these details for the Thirteen Tube Boiler discussed in Section 7.1.4.

Based on previous liquid mercury experiments, accurate readings of the bulk temperatures were expected from the shielded immersion thermocouples shown in Figure 5. This is so because it has been shown* that only slight radial temperature variation exists and entrance effects are nil after 10 diameters.

Also shown in Figure 42 is the location of surface thermocouples on the shell of the Thirteen Tube Boiler. An insight into the variation of mercury tube temperature with length was thus obtained without compromising the reliability nor complicating the fabrication of the boilers. These surface temperature readings should be within a few degrees of the primary fluid temperature due to the high thermal conductivity of the liquid metal primary fluid and generally low heat fluxes. These temperature readings are also within a few degrees of the temperatures of the mercury tube wall, for the heat transfer conductance of the primary fluid is considerably greater than that of mercury.

These surface thermocouples were used to define the temperature distribution of the primary fluid with length which permits an approximate determination of (1) the distribution of heat transfer surface into preheating, boiling and superheating regions, (2) the heat transfer coefficients between the primary fluid and mercury during preheating, boiling, and superheating, (3) mercury side hydrodynamic stability, and (4) startup and shutdown characteristics. The relationship of primary fluid temperature distribution with length to these factors is better understood by considering Figure 43.

Figure 43 is a temperature distribution obtained in the Thirteen-Tube Boiler test discussed in Section 7.1.4. This plot gives a good indication of the heat transfer surface used for preheating, boiling and superheating. Knowledge of these areas and an estimate of the variation in mercury temperature (mercury saturation in the boiling section) permits derivation of heat transfer coefficients.** Section 5.2 discusses this procedure further.

*Kirillov, P.L. et al. "Heat Transfer in Pipes to a Sodium-Potassium Alloy and to Mercury," Journal of Nuclear Energy, Part B, Reactor Technology, Volume I, pp. 123-129. Pergamon Press Ltd., 1959. Translated from Atomnaya Energiya 6 (1959), 382.

**The effect of lateral conduction through the tube walls and primary fluid is expected to introduce a maximum error of about 10 percent in the determination of boiling heat transfer.

Since the surface temperatures indicate the distribution of heat transfer areas in the boiler, continually recorded variations of the surface temperatures with time provide a good measure of the transient characteristics of the boiler during startup and shutdown as well as during unstable operation. In fact, the surface temperature reading is a prime criterion for determining equilibrium.

Mercury flow was determined by the calibrated liquid orifice shown in Figure 5. For 100 percent vapor generation, the choked nozzle was utilized as a check on the liquid flow.

5.2 Data Reduction Procedures

This Section presents the procedures used to reduce the data obtained in the majority of the boilers tested. Procedures peculiar to a specific test are presented in the discussion of that test in Section 7.0. The discussion emphasizes the boiling portion of the boiler as that requires 80% of the heat transferred. The basic data provided by the boiler test instrumentation discussed in Section 5.1 above is summarized by means of Table IV and, for example, Figure 43. These measurements are then utilized to determine the various parameters of interest.

The first step that usually was taken was to determine the distribution of heat transfer area and heat transferred to the preheating, boiling and superheating occurring in a given boiler test. The distribution of heat transfer area is determined by means of the primary fluid temperature distribution obtained from shell surface thermocouples. Figure 43 is an example of such a plot and shows the estimated area distribution based on the rate of change with length of the primary fluid temperature (dT/dL). The unfamiliar terms "suppressed boiling" and "excess superheat" length have been introduced and require elaboration. A "suppressed boiling" length occurs when the mercury has effectively reached the primary fluid temperature but boiling cannot take place since the mercury saturation temperature has not been reached. Heat transfer is nil ($dT/dL = 0$) except for heat loss and the mercury flows through the tubes until the saturation temperature goes below the primary fluid temperature and boiling may begin. The "excess superheat" length is defined as the region at the mercury exit where no apparent heat transfer to the mercury exists ($dT/dL \approx 0$). There may be some superheating, but the resulting primary fluid temperature change is insignificant when compared to that for boiling. Due to the dry wall boiling phenomena occurring in the boiling section, most of the superheating occurs simultaneously with boiling. As a result, only a fraction of the total superheating occurs outside of the boiling section. It is difficult to estimate this effect accurately, but it is of minor significance as the heat transferred to boiling and preheating is about 96 percent of the total required.

TABLE IV

BASIC DATA PROVIDED BY TEST INSTRUMENTATION

<u>VALUES RECORDED</u>	<u>INSTRUMENTATION</u>
1. Flows	
(a) Primary fluid (Na or NaK)	Permanent magnet flowmeter
(b) Mercury	Calibrated orifice-Hg (with check provided by choked nozzle for X=100% when used)
2. Boiler (Pressures)	
(a) Primary fluid inlet and outlet	Diaphragm type transmitter to Bourdon gage
(b) Mercury boiler inlet	Bourdon gage
(c) Mercury boiler outlet	Bourdon gage
(d) Mercury superheater (separate from the boiler) inlet and outlet when used	Diaphragm type transmitter
3. Inlet and Outlet Fluid Temperatures	Immersed shielded thermocouples
4. Pumps	
(a) Primary fluid	Electromagnetic
(b) Mercury	Lapp (3)
5. Powers	
(a) Primary fluid heaters	Wattmeter
(b) Superheater	Wattmeter
(c) Primary fluid electromagnetic pump	Wattmeter

Definition of the preheat and boiling section in a plot such as Figure 43 also establishes the primary fluid temperature drop and thus the heat transferred in the respective sections. The heat transfer to the mercury was also calculated as follows:

$$Q_{pH} = W_{Hg} (h_{f_{x=0}} - h_{in}) \quad \text{Equation 1}$$

where:

Q_{pH} is the heat flow into the preheat section

W_{Hg} is the mercury flow

$h_{f_{x=0}}$ is the saturated liquid enthalpy at the saturation temperature corresponding to the pressure where boiling begins

h_{in} is the enthalpy of the mercury at entrance to the test section

$$Q_B = W_{Hg} (h_v - h_{f_{x=0}}) \quad \text{Equation 2}$$

where:

Q_B is the heat flow into the boiling section

h_v is the saturated vapor enthalpy

$h_{f_{x=0}}$ is the saturated liquid enthalpy as defined for Equation 1.

$$Q_{SH} = (W C_p \Delta T_{SH})_{Hg} \quad \text{Equation 3}$$

where:

Q_{SH} is the superheat heat flow

ΔT_{SH} is the superheat

C_p is the specific heat of the vapor

At the NaK design conditions of Table II, these equations give $Q_{pH} = 340$, $Q_B = 2280$ and $Q_{SH} = 102 \text{ Btu/min}$. These values show the importance of the boiling section.

The heat transferred and area of heat transfer are found for use in calculating heat transfer coefficients,

$$U_m = \frac{Q_{Hg}}{A \Delta T_m} \quad \text{Equation 4}$$

Q_{Hg} is calculated from:

$$Q_{Hg} = (WC_p \Delta T)_{\text{Primary Fluid}} - Q_{\text{Heat Loss}} \quad \text{Equation 5}$$

The mean temperature difference is found from:

$$\Delta T_m = \int_0^{A_{TOT}} \frac{\Delta T dA}{A_{TOT}} \quad \text{Equation 6}$$

The ΔT under the integral sign is $(T_w - T_{Hg})$. The wall temperature can be taken as the surface temperatures measured (see Section 5.1) while the mercury temperature can be based on the saturation temperature corresponding to the local boiling pressure. These have not been measured but must be estimated from the pressures existing in the inlet and exit of the boiler. Local values of pressure may be found as shown in Figure 43 by using gas friction factors and the total pressure drop in the boiler. Since the "excess superheat" length is essentially all vapor, its pressure drop is easily found using a one-phase friction factor as shown in Figure 48. The boiling section pressure drop is found from:

$$\Delta P_B = P_{Hg_{in}} - P_{Hg_{out}} - \Delta P_{ESH} - \Delta P_{SB} + \Delta P_{LOSSES} \quad \text{Equation 7}$$

where:

- $P_{Hg_{in}}$ is the measured mercury inlet pressure
- $P_{Hg_{out}}$ is the measured mercury outlet pressure
- ΔP_{ESH} is the pressure drop in the "excess superheat" length based on friction factors such as those of Figure 48.
- ΔP_{SB} is the pressure drop in the "suppressed boiling" length. Due to the low velocity of the mercury (about 1 ft-sec.) this is just a static head drop as the mercury flows upward.
- ΔP_{LOSSES} represents the entrance and exit losses.

Figure 43 depicts the pressure drops that may occur and shows the estimated saturation temperatures in the boiling section. The mean temperature difference of Equation 6 and the mean overall heat transfer coefficient of Equation 4 may then be determined. This overall coefficient is effectively the mercury side coefficient (e.g., $h_{Hg} = 50$ results

in only a 3 percent error). Thus, the assumption

$$h_{\text{Hg BOILING}} \approx U_{\text{BOILING}} \quad \text{Equation 8}$$

has been made in the test analyses of Section 7.1. Although higher mercury side coefficients can invalidate this assumption, in all future boiler designs the overall coefficient is used directly.

The correlation procedures of Section 6.0 require the use of mercury vapor properties in the boiling section. The values of these properties are taken at the average vapor temperature and pressure calculated from the vapor temperature at 100 percent quality, the saturation temperature at zero quality and the corresponding pressures.

In the testing of once-through forced convection mercury boilers it is difficult to verify the existence of 100 percent exit quality. Calorimeters were considered but the low specific heat of mercury vapor ($C_p \approx 0.025 \text{ btu/lb-}^\circ\text{F}$) allows a small heat loss to introduce errors that invalidate the resulting measurements. Temperature and pressure measurements were found to not give accurate thermodynamic state points. The mercury liquid and vapor may not be in thermal equilibrium and the liquid occupies such a small volume fraction that a thermocouple reads the vapor temperature only.

Three means have been established for the estimation of exit quality. A heat balance may be taken, comparing the thermal energy lost by the primary fluid minus the heat loss to that required to preheat, boil and superheat the mercury. The discussion of the Thirteen-Tube Boiler tests in Section 7.1.4 utilizes the heat balance approach to show that this boiler was generating 100 percent vapor (See Figure 46).

A choked nozzle may also be used to measure the vapor exiting from the boiler (See Figure 5, the test rig schematic). A small amount of moisture would not affect the choked nozzle characteristics significantly and the flow indicated by the nozzle inlet pressure and temperature and calibration curve would be the vapor flow. The resulting quality would be

$$X_{\text{exit}} (\%) = \frac{W_{\text{nozzle}}}{W_{\text{liquid entering}}} \times 100 \quad \text{Equation 9}$$

The liquid flow rate entering the boiler is measured by means of an orifice in the inlet line.

Figure 47 gives another means of utilizing a choked nozzle to determine that 100 percent vapor is generated.

The shell surface temperatures versus the distance along the coil are an indication of the heat being transferred by the NaK. The position of 100 percent mercury quality can be estimated by establishing the point at which the slope of the NaK temperature profile ($dT/dl = 0$) is zero. The heat transferred to superheat the vapor beyond the 100 percent

quality point is negligible, since a major portion of the vapor is being superheated during boiling and the total superheat required is estimated to be only four percent of the total heat transferred.

6.0 THEORETICAL ANALYSES AND DATA CORRELATION PROCEDURES

The primary objective of theoretical analyses in the SNAP II boiler development program is to provide means of correlating existing boiler data and to provide sufficient confidence in extrapolation toward prototype and final designs. Boilers were designed and tested to provide data and expose problem areas. Data were then analyzed and correlations were based on various mathematical models formulated to describe the forced convection boiling process. As these models and correlations were tested by subsequent boiler designs, appropriate changes were made in the theories.

The development of the boiler as an integral part of the power conversion system permits more rapid attainment of the component than is normally possible with a laboratory research approach. However, though proved adequate for development, the resulting heat transfer data were limited in range, since the system precluded variation of more than one parameter at a time. Inability to separate the effects of the several parameters required consideration of many other related works and some judgment in arriving at and interpreting the resulting correlations. Development of physical models describing the boiling process aided in the interpretation.

The theoretical considerations of this section are principally concerned with the two-phase heat transfer and fluid dynamics occurring in the boiling section of once-through forced convection mercury boilers. Preheating and superheating considerations are not discussed because (1) these sections of the boiler require a small fraction of the heat needed in the boiling section, (2) the heat transfer coefficients are generally higher, and (3) the phenomena are better understood. The heat transfer aspects of the forced convection boiling phenomena are emphasized in the discussion to follow, since the theoretical development is unique. Two-phase pressure drop has been extensively studied elsewhere* and the significant parameters established.

6.1 Model Development

The first boiler designs for the SNAP II boiler development program were based on the "forced convection to vapor only" physical model. This model was chosen because the high temperature differences and non-wetting between the mercury and wall would cause film boiling to occur (see Figure 1); also the high density of mercury would result in the major portion of the tube being filled with vapor. For example, at only one percent liquid by weight, 81 percent by volume is vapor at 133 psia.

The vapor-only model can be depicted as shown in Figure 7. The presence of liquid droplets can be ignored and the evaporated vapor assumed to be continually added as through a porous bed. A Dittus-Boelter form of equation with mass velocity for vapor only of G_{TOTX} then gives:

$$\frac{hD}{k_v} = 0.023 \left(\frac{D G_{TOTX}}{\mu_v} \right)^{0.8} \left(\frac{C_{p_v} \mu_v}{k_v} \right)^{0.4} \quad \text{Equation 10}$$

*SNAP II Power Conversion System Topical Report No. 8, "Mercury Condensing Research Studies," TRW ER-4442, and Gresham, W.A., Foster, P.A., Kyle, R.J., "Review of the Literature on Two-Phase (Gas-Liquid) Fluid Flow in Pipes," Part I, WADC Technical Report 55-422, Georgia Institute of Technology, June 1955.

Integration over a change in quality from 0 to 100 percent gives

$$\frac{h_{mD}}{k_v} = \frac{0.023}{1.8} \left(\frac{D G_{TOT}}{\mu_v} \right)^{0.8} \left(\frac{C_{P_v} \mu_v}{k_v} \right)^{0.4} \quad \text{Equation 11}$$

The expected heat transfer coefficients from Equation 11 were of the same magnitude of film pool boiling coefficients for pure mercury.

Boiler tests of Preliminary Test Sections No. 1 and No. 2 showed that the performance of boilers designed on the basis of the vapor-only model were off design requirements by almost an order of magnitude (as exit qualities of about 10 percent were obtained). It was concluded that the mechanics of the liquid dispersion was an important consideration. Swirl devices were then considered as means of forcing liquid droplets towards the heating surface. Information obtained on the effects of swirl devices in the SNAP I tests had shown that performance with swirl devices was still far from that desired.

Boiler data, including that with inserted swirl devices, were then correlated assuming a fog or misty flow model in which the Dittus-Boelter form of equation was again selected and all of the fluid properties were evaluated as a mixture. In such a model, all of the transport properties depend on the volume fractions of the phases in the mixture. For mercury at pressures far from the critical point, the vapor volume fraction is large compared to that of the liquid at qualities greater than 10 percent. As a result, the effective thermal conductivity and viscosity of the mixture are approximately equal to that of the vapor. To retain the form of the Prandtl number in the Dittus-Boelter type of equation the specific heat was defined as follows:

$$C_{P_{eff}} = \frac{dQ}{M_o dT} \quad \text{Equation 12}$$

where M_o is the total mixture weight, T is temperature, Q is heat transferred and $C_{P_{eff}}$ is effective specific heat at constant pressure. Figure 8 shows the process assumed. But,

$$\Delta Q = M_o \left[h_{fg} \Delta X + C_p (T_w - T_{sat}) \right] \quad \text{Equation 13}$$

C_p for mercury vapor is equal to 0.02517 Btu per pound at 1000°F., h_{fg} equals 124 Btu per pound and for $T_w - T_{sat}$ equal to 200°F, $C_p (T_w - T_{sat})$ equals 5 Btu per pound. Therefore, the sensible heat is small compared to the latent heat and can be ignored in computing Q . As a result

$$\Delta C_{P_{eff}} = \frac{M_o h_{fg} \Delta X}{M_o dT}$$

$$C_{P_{eff}} dT = h_{fg} dX$$

$$C_{p_{eff}} \int_{T_{sat}}^{T_w} dT = h_{fg} \int_{x=0}^{x=1} dX$$

$$C_{p_{eff}} = \frac{h_{fg}}{T_w - T_{sat}}$$

Equation 14

As defined $C_{p_{eff}}$ is not a property since its magnitude depends on a process. In effect, its definition arose from the desire to retain the Prandtl number form and the early observation of a dependence of the heat transfer coefficient on $T_w - T_{sat}$. In Section 6.2.1 the later drop model analysis justifies the term $h_{fg}/T_w - T_{sat}$.

As a result, the heat transfer equation assumed for the correlation was

$$\frac{h D}{k_v} = \text{Const.} \left(\frac{D G}{\mu_v} \right)^n \left(\frac{C_{p_{eff}} \mu_v}{k_v} \right)^m$$

$$\frac{h D}{k_v} = \text{Const.} \left(\frac{D G}{\mu_v} \right)^n \left[\frac{h_{fg} \mu_v}{(T_w - T_{sat}) k_v} \right]^m \quad \text{at any given time} \quad \text{Equation 15}$$

n was assumed equal to 0.8 (see Section 6.2.3 and m and constant can be determined from the correlation.

Although these parameters correlated the data, (see Figure 31) the exponent m was much larger than anticipated (0.3 or 0.4) for a single-phase fluid and the magnitude of the Nusselt number was much lower than expected if a constant of 0.023 is selected based on single-phase fluid heat transfer test. Apparently, the correct variables were used for correlation but not the correct physical model.

The model which proved most valid is discussed below (Section 6.2.1). This model is one in which heat is assumed transferred from the wall to the droplet through a small vapor film supporting the drop. The droplet is assumed small enough so that its surface is stable to disturbances and vapor bubbles do not form within the drops. Since the droplets are swirled as they pass along the tube, the inertial force on the droplet suppresses the vapor film under the drop and thereby improves the heat transfer.

Mercury droplets were evaporated on a hot plate in air and the time was measured to vaporize completely the drops of a given initial size (see Section 7.2.4). These data formed a

basis for theoretically predicting heat transfer coefficients under the droplets. The physics is similar to the "air-car" principle* in which gas is ejected from under the car and the resulting momentum produces a gas pressure which supports it. In droplets, the vapor is formed on the underside near the heated wall and is ejected radially outward. The latent heat required for evaporation is transferred across the supporting vapor film. The pressure and film thickness under the drop in a force field is computed from the momentum of the ejected vapor.

When the experimental drop data are applied to a droplet in a swirl tube in which radial body forces act on the drop, the time for evaporation becomes the droplet residence time in a tube. The residence time is related to the droplet velocity and the tube length as follows:

$$t_o = \int_0^{L_{TOT}} \frac{dL}{u_v \epsilon}$$

where u_v is the axial vapor velocity, ϵ is the slip ratio u_l/u_v , L is the tube length, and t_o is total resident time of the drop in the tube. This equation is integrated to give length required to vaporize the droplets completely. Although the actual path length of a mercury drop is determined by the swirl insert, the resident time depends on the axial velocity and tube length. Initial droplet sizes in boiler tubes are estimated from droplet stability considerations.

It is envisioned that a solid liquid slug in the preheat section of the boiler is heated to the saturation temperature at which point the interface becomes statically unstable since the surface vapor pressure equals the vapor pressure within the liquid. As a result, the interface breaks up and the slugs of mercury are swirled around until they assume the diameter of the swirl wire. The swirl wire is a helical wire coil, much like a spring, which is inserted into the tube so that intimate contact between the wire and tube inner wall is obtained. A drop larger than the wire diameter would be unstable since it would be sheared off by the relative velocity of the vapor. Thus, the initial droplet size is assumed equal to the wire diameter. This assumption is valid as long as the wire size is large compared to the laminar sublayer of the boundary layer. In all boilers the ratio of wire to tube diameters was kept constant. Experimental correlations based on forced convection dropwise boiling using this physical model have shown promising results. The detailed considerations are given in Section 6.2.1 of this report.

6.2 Heat Transfer Analyses

The classical picture of pool boiling heat transfer to a liquid indicates three successive regions: nucleate, transition, and film boiling. Enough data and correlations are available in the literature to estimate the boiling convection heat transfer coefficients for the case where a liquid film is in contact with the heating surface (nucleate boiling). However, a characteristic feature of heat transfer during the flow of liquid over a surface heated to a high temperature is the existence of a vapor film that considerably impairs the heat transfer rate.

*Haynes, A.L. and D.J. Jay. Mechanical Engineering, 82 (September, 1960), 62.

In general the actual physical situation in once-through forced convection boiling is one in which some of the liquid phase is dispersed in drop form in a continuous vapor phase and some forms a continuous film of liquid on the wall. At some point in the tube this liquid film adhering to the wall completely disappears by evaporation, and the liquid droplets are dispersed in the vapor and must be vaporized by transferring heat through the vapor. Consequently, the heat transfer coefficients decrease considerably.

This phenomenon of heat transfer to dropwise flow has been found with water (Figure 9) and magnesium amalgam (Figure 10), both wetting fluids. The point where the liquid film on the wall is completely evaporated off the heating surface is quite evident in both cases. The heat transfer coefficients are reduced by an order of magnitude and the value is even below the gas coefficient predicted by the Dittus-Boelter equation.

Drop formation in wetting fluids is due to instability of the liquid-vapor interface. As the quality increases, the interfacial shear stress due to the vapor velocity increases, generating a capillary wave at the interface. At a critical value of the vapor velocity, these waves become unstable and break up into liquid drops. For a non-wetting fluid such as pure mercury, the formation of the drops is due to both interfacial shear and the high surface tension and non-wetting properties of pure mercury. Adiabatic two-phase flow tests with liquid mercury (see Section 7.2.3) have shown that a continuous liquid film cannot be produced. The tendency of liquid mercury is to "roll up" into drops. This destabilizing effect is mainly attributed to the inherent properties of mercury (high density and surface tension) rather than two-phase flow disturbances at the interface.

This difference in drop formation in wetting and non-wetting fluids is important from a heat transfer point of view. In the case of pure mercury, the low heat transfer rates due to drop flow, or "dry wall boiling" as it is sometimes called, starts almost immediately at the preheater interface: no marked change from nucleate, to transition, to film boiling can be detected.

Because dropwise flow is characteristic of liquid mercury, an understanding of heat transfer in this mode is necessary to the design of SNAP II boilers. Laboratory investigations of drop evaporation have been in progress as described in Section 7.2.4.

6.2.1 Once-Through Forced Convection Boiling of Mercury

This section presents a theoretical analysis of the once-through forced convection boiling of mercury. Values of the coefficients predicted compare favorably with those determined from test. The analysis also indicates the important correlating parameters. A brief discussion of the analysis is presented initially to act as a summary.

In Figure 12, the experimental results of the hot plate experiments, the mean heat transfer coefficient to the droplet based on its projected area ($\bar{h}_\delta$) is plotted versus the temperature difference. The theoretical curve derived below (Equation 24) based on a vapor film

supporting the drop on the plate is also shown. At temperature differences greater than 200°F the agreement between the theoretical and measured results is good. At temperature difference below 200°F, the experimental $\bar{h}_\delta$ is higher than the theoretical curves. This is attributed to intermittent contact between the heated wall and droplet.

The analysis resulting in Equation 25b considers the case of the drop in point contact with the wall and the heat transfer resistance of importance becomes the liquid mercury of the drop. The $\bar{h}_\delta$ under these conditions is then computed from a droplet Nusselt number equal to 2. This implies that when the film becomes too thin, the major resistance to heat transfer will be in the liquid of the drop. Therefore, $\bar{h}_\delta$ can never exceed that $\bar{h}_\delta$ computed from a droplet Nusselt number equal to 2. The fact that the data of Figure 12 lie between the two curves means that a transition region exists with partial film boiling and partial liquid contact due to the droplet bouncing around on the surface similar to the oscillations of a mass-spring system. At the high ΔT this does not happen and the drop cannot penetrate the vapor film. The two stable regimes are the liquid contact regime and the vapor film regime, with a transitional regime between the two. Parallel heat flow paths also tend to explain the transitional character of the correlation since heat could flow through a point contact and through a vapor film surrounding the point contact. This mechanism would make $\bar{h}_\delta$ greater than the sum of the two $\bar{h}_\delta$'s based on wall contact and vapor film.

Figure 14 compares test results from the Thirteen Tube Forced Convection Boiler to the dropwise heat transfer model. The dotted line is based on the film theory and the solid line is corrected by the contact factor which is the ratio of $\bar{h}_\delta$ as measured on the hot plate to $\bar{h}_\delta$ computed from the film theory. The contact factor was assumed to be a function of the mean boiler ΔT . Actually, a mean contact factor should be computed over the range of ΔT from boiler inlet to boiler exit. In addition, the effect of radial acceleration on the contact factor should be experimentally determined. The dashed line shows the estimated curve used in August 1959 to design the Thirteen Tube Boiler (see Section 7.1.4). It was assumed that the slope would be uniform, that is, that the film theory would apply up to the design mean temperature difference. Since the actual slope increased, an overdesign was obtained. Although the dropwise heat transfer theory predicts the magnitude of the heat transfer coefficients and the shape of the curve, some of the points lie off the curve, indicating that drop mechanics are still not completely understood. Although the actual situation is much more complex than the simple model developed, this analysis has provided an insight into the phenomena and has established correlation procedures (see Section 6.2.3).

The following steps were taken in the analysis of the boiling process occurring during once-through forced convection boiling:

1. Analysis of evaporation of small drops on heated walls.
2. Attainment of an experimental correction of the result of the above analysis to account for the intermittent drop contact region of boiling where most of the forced convection data seems to exist.

3. Extension of the analyses of Items 1 and 2 to the case of once-through forced convection boiling.

Evaporation of Drops

Consider the film boiling drop model depicted in Figure 11. Assuming that the mean vapor film created by the vapor evolved from the drop is stable and supports the drop, the following momentum, energy and continuity equations may be written

Momentum:

$$\Delta P \pi \delta \delta' = \frac{\dot{m}_v u'}{g} \quad \text{Equation 16}$$

$$\Delta P \frac{\pi \delta^2}{4} = \frac{\pi \delta^3}{6} \rho_f N \quad \text{Equation 17}$$

Energy:

$$\dot{m}_v h_{fg} = \frac{k_v}{\delta} \frac{\pi \delta^2}{4} (T_w - T_d) \quad \text{Equation 18}$$

Continuity:

$$\pi \delta \delta' u \rho_v = \dot{m}_v \quad \text{Equation 19}$$

also:

$$\frac{k_v}{\delta} = h_\delta \quad (\text{laminar film}) \quad \text{Equation 20}$$

$$T_w - T_d = \Delta T \quad \text{Equation 21}$$

Combining these equations gives the drop heat transfer coefficient for a stagnant drop:

$$h_\delta = \left(\frac{2}{3} \frac{N}{\delta} \rho_v \rho_f g \right)^{1/4} \left(\frac{4 k_v h_{fg}}{\Delta T} \right)^{1/2} \quad \text{Equation 22}$$

Integration of this expression for a change in the drop diameter from its initial diameter δ_o to complete evaporation ($\delta = 0$) gives:

$$\frac{\bar{h}_\delta}{h_{\delta_o}} = \frac{4}{3} \quad \text{Equation 23}$$

where $\bar{h}_\delta$ is the mean drop coefficient for $\delta = \delta_0$ to $\delta = 0$ and h_{δ_0} is the heat transfer coefficient for the initial size drop. Thus,

$$\bar{h}_\delta = \frac{4}{3} \left(\frac{2}{3} \frac{N}{\delta_0} \rho_v \rho_f g \right)^{1/4} \left(\frac{4 k_v h_{fg}}{\Delta T} \right)^{1/2} \quad \text{Equation 24}$$

Correction to Account for Intermittent Contact Effect

Equation (24) is compared in Figure 12 to the experimental drop data obtained as described in Section 7.2.4. Figure 12 shows that the theoretical equation appears to reasonably fit the data for $\Delta T > 200^\circ\text{F}$.

An approximate maximum value of $\bar{h}_\delta$ can be postulated to exist when the drop has little or no contact resistance due to the vapor film. The major resistance to heat transfer then becomes the liquid itself and may be computed as follows:

CONTACT DROP MODEL

Droplet of diameter δ in contact with a heated wall, showing heat flow paths

Assume:

Mean length for heat conduction $\approx \delta/2$ (1/2 drop diameter)

Mean spherical surface area for conduction $\approx \frac{\pi \delta^2}{4}$ (for sphere with radius of $\delta/2$)

Reference surface area for convection coefficient $\approx \frac{\pi \delta^2}{4}$ (projected area)

Therefore,

$$Q = \frac{\pi \delta^2}{4} h_{\delta} \Delta T = \frac{k_f}{\delta/2} \frac{\pi \delta^2}{4} \Delta T$$

and

$$\frac{h_{\delta} \delta}{k_f} = 2 = Nu_{\delta} \quad \text{Equation 25(a)}$$

Figure 12 also compares this equation with the experimental data.

The mean drop diameter when a drop is completely vaporized is $\bar{\delta} = 5/9 \delta_o$ (by combining Equations 28 and 22 and integrating over the total evaporation time). Therefore,

$$\bar{Nu}_{\delta_o} = \frac{9 \times 2}{5} = \frac{18}{5} = \frac{\bar{h}_{\delta} \delta_o}{k_f} \quad \text{Equation 25(b)}$$

It may be seen that the experimental data is somewhere between the two theoretical curves of Equations 25(b) and 24. This is attributed to an intermittent contact phenomenon that occurs in the transition region between the stable vapor film region and the stable drop contact region, similar in nature to partial film boiling in pool boiling.

To account for the intermittent contact effect modification of Equations 22 and 24 by a factor of β and simplifying gives, respectively,

$$\bar{h}_{\delta} = \beta \left(\frac{2}{3} \frac{N}{\delta} \rho_v \rho_f g_c \right)^{1/4} \left(\frac{4 k_v h_{fg}}{\Delta T} \right)^{1/2} \quad \text{Equation 26}$$

and

$$\bar{h}_{\delta} = \frac{8}{3} \beta \left(\frac{2}{3} \frac{N}{\delta_o} \rho_v \rho_f g \right)^{1/4} \left(\frac{k_v h_{fg}}{\Delta T} \right)^{1/2} \quad \text{Equation 27}$$

Film Vaporization of Small Drops by Forced Convection

A drop being moved by vapor is depicted in Figure 13. The heat transferred from the wall to the drop is

$$h_{fg} \frac{dm}{dt} = h_{\delta} \frac{\pi \delta^2}{4} \Delta T$$

Since the mass is $m = \frac{\pi \delta^3}{6} \rho_f$

$$h_{\delta} = \frac{2 h_{fg} \rho_f}{\Delta T} \frac{d \delta}{d t} \quad \text{Equation 28}$$

In the model shown there would also be some heat transferred to the mercury vapor continually generated. This is small by comparison to the heat transferred to the liquid drops being evaporated. For example, for

$$T_w - T_{\text{sat}} = 300^{\circ}\text{F}; C_p \int Hg_{\text{vapor}} \approx 0.025 \text{ Btu/lb-}^{\circ}\text{F}$$

$$h_{fg} \approx 124 \text{ Btu/lb}$$

$$\frac{C_p \Delta T}{h_{fg}} = \frac{0.025 \times 300}{124} = 0.06 \quad \text{Equation 29}$$

Thus it takes only 6% as much heat to bring the vapor up to the wall temperature as is required to evaporate the liquid drops. Newly generated vapor must then be almost immediately heated to nearly the wall temperature. Furthermore, the heat transferred from the surrounding vapor to the drop can be assumed small if (1) the relative velocity between the drop and the vapor stream ($u - u\epsilon$) is low, (2) the vapor acts as an insulator to heat transfer, namely, has a low heat transfer coefficient and (3) temperature differences for the cases of interest are such that the radiant heat transferred is negligible.

Proceeding then, the resident time of the drop is related to the tube length by

$$dt = \frac{dL}{u \epsilon} \quad \text{Equation 30}$$

where ϵ is the slip ratio and less than 1. The slip ratio can be estimated by assuming that the laminar shear on the underside of small spheres equals the drag at low sphere Reynold's number (see sketch), namely;

$$C_D = \frac{24}{Re} = \frac{24 \mu}{\delta \rho_v (u - \epsilon u)}$$

$$\frac{F_o}{A} = \mu \frac{du}{dy} = \frac{C_D \rho_v (u - \epsilon u)^2}{2g}$$

The drop projected area is the same for both the shear and the drag.

$$\frac{du}{dy} = \frac{u \epsilon}{k/h_{\delta}}$$

By manipulation, these equations give

$$\epsilon = \frac{12 k}{h_{\delta} \delta} \quad \text{Equation 31}$$

Also for a drop being swirled in a tube the number of g's acceleration N 's (see sketch),

$$N = \frac{u_T^2}{rg} = \frac{2}{Dg} (u \epsilon \tan \alpha)^2$$

$$u_v = \frac{GX}{\rho_v} \quad \text{Equation 31a}$$

$$N = \frac{2}{Dg} \left(\frac{GX \epsilon \tan \alpha}{\rho_v} \right)^2 \quad \text{Equation 32}$$

Combining Equations 26 and 28, and Substitution of Equations 30, 31, 31(a) and 32 gives:

$$\frac{d\delta}{dL} = \beta \left(\frac{2}{9D} \right)^{1/3} \left(\frac{\rho_v \tan \alpha}{\rho_f} \right)^{2/3} \left(\frac{k \Delta T}{h_{fg} G X} \right)^{1/3} \quad \text{Equation 33}$$

Assuming $\frac{dX}{dL} = \text{const}$ or $X = \frac{L}{L_T}$ and integrating this equation between the limits δ_o to $\delta = 0$;
 $L = 0$ to L_T ; $X = 0$ to $X = 1$ results in

$$L_T = \frac{2}{3} \frac{\delta_o}{\beta^{4/3}} \left(\frac{9D}{2} \right)^{1/3} \left(\frac{\rho_f}{\rho_v \tan \alpha} \right)^{2/3} \left(\frac{h_{fg} G}{k \Delta T} \right)^{1/3} \quad \text{Equation 34}$$

Further, the mean tube heat transfer coefficient for complete vaporization is

$$\bar{h} = \frac{G D}{4 L_T} \frac{h_{fg}}{\Delta T_m} \quad \text{Equation 35}$$

Where D is the tube diameter; substituting the above expression for L_T , equating ΔT and ΔT_m , and rearranging gives:

$$\frac{\bar{h} D}{k} = \frac{6}{8} \tan \alpha^{1/3} \beta^{2/3} \left(\frac{\rho_v}{\rho_f} \right)^{4/3} \left(\frac{D}{s_o} \right)^{2/3} \left(\frac{G D}{\mu} \right)^{2/3} \left(\frac{\mu h_{fg}}{k \Delta T_m} \right)^{2/3} \quad \text{Equation 36}$$

This equation is plotted in Figure 14 where it is compared with experimental forced convection data from Section 7.1.4. The initial paragraphs of this section discuss this comparison in detail.

6.2.2 Non-Wetting Porous Bed Boiler

Low heat transfer coefficients were experienced in early once-through forced convection mercury boiler tests. Other means of generating mercury vapor thus became of interest. One such scheme is a non-wetting porous bed boiler. Its operation is analyzed below.

In the non-wetting porous bed a hemispherical liquid mercury droplet is heated from behind through the mass of pressurized liquid mercury in contact with the heat transfer surface as in Figure 15. In the porous bed the liquid mercury is prevented from entering the bed by its surface tension and non-wetting characteristic. The amount of pressurization possible depends on the size of the pores. The pressure difference between the liquid and the vapor just above the liquid interface can be expressed by a force balance, namely:

$$P_f - P_v = \frac{4 \sigma}{D} \cos \theta \quad \text{Equation 37}$$

where P_f is liquid pressure, P_v is vapor pressure, D is pore diameter, σ is surface tension, and θ is contact angle. The liquid pressure P_f should be greater than the saturation pressure corresponding to the heating surface temperature to prevent vapor formation within the liquid.

To test the porous bed phenomenon the mercury porous bed boiler setup shown in Figure 16 was used. The various heat transfer resistances involved in the porous bed are (1) lead coefficient, (2) stainless steel container conductance, (3) liquid mercury coefficient, and (4) an apparent heat transfer coefficient of the porous bed due to the flow resistance and an associated temperature drop. Preliminary estimates indicated that in the temperature range tested the major resistance to heat transfer was due to the flow resistance resulting from the mass transfer of vapor from the liquid mercury to the vapor side of the bed. The

apparent heat transfer coefficient for the mass transfer of vapor through pores may be computed as follows:

Assume a single pore to be cylindrical and the vapor flow to be laminar. The vapor at (1) in Figure 17 is saturated at the temperature of the liquid mercury:

$$dP = \frac{32 \mu u dL}{D^2} \quad * \quad \text{Equation 38}$$

where P is pressure, μ is viscosity, U is velocity, L is length, and D is diameter.

Also:

$$u = \frac{\dot{m}}{A_o \rho} \quad \text{Equation 39}$$

$\dot{m}$ - mass flow rate

ρ - density

A_o - cross-sectional area of pore

Combining Equation 38 and 39 we have,

$$dP = \frac{32 \mu \dot{m} dL}{D^2 A_o \rho} \quad \text{Equation 40}$$

The apparent heat transfer coefficient can be expressed as

$$\dot{m} h_{fg} = h A (T_1 - T_2) \quad \text{Equation 41}$$

also let $\phi = A_o/A$, the porosity, h_{fg} - latent heat, T-temperature, h-heat transfer coefficient, A-total bed area; combining Equation 40 and 41 we have,

$$dP = \frac{32 \mu h (T_1 - T_2) dL}{D^2 \phi \rho h_{fg}} \quad \text{Equation 42}$$

As a first approximation assume $\rho = \frac{\rho_1 + \rho_2}{2}$; $\mu = \frac{\mu_1 + \mu_2}{2}$. By integration we have,

* Rouse and Howe, Basic Mechanics of Fluids, New York: Wiley and Sons, 1953, p. 120.

$$\left\{ \frac{(P_1 - P_2)}{(T_1 - T_2)} \frac{(\rho_1 + \rho_2)}{(\mu_1 + \mu_2)} \right\}_{\text{Sat.}} \frac{D^2}{32} \phi \frac{h_{fg}}{L_T} = h \quad \text{Equation 43}$$

the subscript "Sat" indicates that conditions at 1 and 2 of Figure 17 should be evaluated at saturated vapor conditions.

Equation 43 can also be written as,

$$\left\{ \frac{(P_1 - P_2)}{(T_1 - T_2)} \frac{(P_1/T_1 + P_2/T_2)}{(\mu_1 + \mu_2)} \right\}_{\text{(Sat)}} \frac{D^2 \phi h_{fg}}{32 L_T R} = h \quad \text{Equation 44}$$

where R is the gas constant.

Another approximation can be made by assuming $\rho = \frac{P}{RT}$ in Equation 42

$$dP = \frac{32 \mu h (T_1 - T_2) RT dL}{D^2 \phi P h_{fg}} \quad \text{Equation 45}$$

Assume T to be equal to the bath temperature (isothermal throttling). Integration of Equation 45 results in,

$$\frac{P_1^2 - P_2^2}{T_1 - T_2} = \frac{64 \mu R T_{pb} L_T h}{D^2 \phi h_{fg}}$$

Letting $\mu = \frac{\mu_1 + \mu_2}{2}$ we have

$$\left\{ \frac{(P_1 - P_2)}{(T_1 - T_2)} \frac{(P_1 + P_2)}{T_{pb} (\mu_1 + \mu_2)} \right\} \frac{D^2 \phi h_{fg}}{32 L_T R} = h \quad \text{Equation 46}$$

Equation 46 is not much different from Equation 44; therefore, the correlating parameter is

$$\left\{ \frac{(P_1 - P_2)}{(T_1 - T_2)} \frac{(P_1/T_1 + P_2/T_2)}{(\mu_1 + \mu_2)} \right\}_{\text{sat.}} \quad \text{Equation 47}$$

Assuming h approximately equal to U the overall heat transfer coefficient

$$U = \frac{\dot{m} h_{fg}}{A (T_{pb} - T_2)} \quad \text{Equation 47a}$$

where T_2 is the saturation temperature corresponding to the boiler pressure (vapor space pressure).

6.2.3 Forced Convection Once-Through Mercury Boiling Heat Transfer Correlation Procedures

The theoretical analysis of Section 6.2.1 above has shown that once-through forced convection mercury boiling correlations should consider the dimensionless parameters (see Equation 36).

$$Nu = C Re^m K^n$$

where:

$$Nu = \frac{h_B D}{k_v}$$

$$Re = \frac{GD}{\mu_v}$$

$$K = \frac{\mu_v h_{fg}}{k_v \Delta T_m}$$

h_B is the mean boiling heat transfer coefficient for a change in quality from 0 to 100 percent.

D is the tube diameter.

k_v is the vapor thermal conductivity based on the average vapor temperature.

μ_v is the vapor viscosity based on the average vapor temperature.

G is the mass velocity.

h_{fg} is the change in enthalpy during boiling.

ΔT_m is the area mean temperature difference.

The theoretical analysis also gives an approximate value to the exponent m of about $2/3$ for stable film boiling under mercury drops. The exponent n becomes greater when ΔT decreases due to drop contact with heated wall (see Section 6.2.1). The theoretical approach is not precise, however, and experimental data must be considered in determining the exponents to be used in correlating the once-through forced convection mercury

boiling data. As previously indicated, tests of SNAP II development boilers to date have not permitted the individual variation of all significant parameters. Consequently, a detailed check of the theoretical analysis is deferred until data from the boiling mercury research program is made available.

For example in the Thirteen Tube Boiler test it was impossible to hold the integrated mean ΔT between the tube wall and mercury constant and also obtain a large variation in Reynolds number. For this reason a definite relationship between Nusselt number and Reynolds number was unobtainable. A research effort such as that discussed in Section 9.0 is required to provide such information. The utilization of other experiments was required.

Figure 18 relates the heat transfer coefficient and inlet mass velocity for the once-through forced convection boiling of mercury in a sloped bare tube. These variables are transformable into the axial flow Nusselt number - Reynolds number relationship:

$$Nu_{\text{Axial Flow}} = \text{Const. Re}^{1.0} \quad \text{Equation 48}$$

The effect of a swirl device on the Nusselt number is given in Figure 19 for a wide range of pitch to tube diameter ratios. In equation form

$$\frac{Nu_{\text{Vortex Flow}}}{Nu_{\text{Axial Flow}}} = \text{Const. Re}^{-0.2} \quad \text{Equation 49}$$

Equations 48 and 49 combined suggest that a Reynolds number exponent of 0.8 be used. The theoretically determined value of 0.67 verifies.

In summary, once-through forced convection mercury boiling data should be correlated by comparing $\frac{Nu}{Re^{0.8}}$ to $\left(\frac{\mu h_{fg}}{k \Delta T_m} \right)$. All such data are compared on this basis in the section that discusses test results, (Section 7.0) of once-through forced convection boilers.

6.3 Boiling Pressure Drop Correlation For Once-Through Forced Convection Boiling of Mercury

It has been shown that two-phase pressure drop can be correlated by means of a Φ factor defined as follows: *

$$\Phi = \frac{\Delta P_{TP}}{\Delta P_v} \quad \text{Equation 50}$$

*SNAP II Power Conversion System Topical Report No. 8, "Mercury Condensing Research Studies," TRW Report No. ER-4442.

In this equation ΔP_{TP} is the actual two-phase pressure drop and ΔP_v the friction pressure drop which would occur if the vapor (gas) phase were flowing separately. For boiling, Φ was taken to be the ratio of the pressure drop for evaporation of a zero quality liquid to the computed pressure drop which would occur for the vapor only flowing without liquid present. The pressure drop for vapor only is computed as though the increase in vapor fraction along the tube were accomplished by addition of the vapor rather than by evaporation. The vapor flow was assumed to increase linearly along the tube length. In the boiler tests momentum and acceleration effects were negligible by comparison and the friction pressure drop is taken as the measured.

For the complete change in state (i.e., 0 to 100 percent for boiling and 100 to 0 percent for condensing), two-phase pressure drop correlations should be based on

$$\Phi = F \left[Re_o \left(\frac{\bar{v}_g}{10} \right)^{1.25} \right] \quad \text{Equation 51}$$

where Re_o is the boiling section exit vapor Reynolds Number, $\bar{v}_g$ is the mean vapor specific volume evaluated at average boiling section pressure in cubic feet per pound.

The specific application of this correlation to boiling test results is presented in Section 7.1.4. Figure 20 illustrates the effectiveness of Equation 51 in correlating two-phase pressure drop. This figure compares pressure drop data for the Thirteen Tube Boiler (see Section 7.1.4), the inclined and horizontal mercury condensers, the Martinelli * correlation for viscous-turbulent flow for an average quality of 50% and adiabatic-isothermal two-phase tests in tubes that were bare and with spring inserts. As can be seen the Φ for boiling tubes with spring inserts is lower than the Φ for condensing tubes without inserts because of the difference in flow pattern of the liquid drops. In condensers the drops were randomly distributed in the tube cross-section whereas in boilers the drops congregated on the heating surface due to swirl.

The low exit quality data obtained during the Preliminary Test Section No. 2 test has also been correlated by means of Equation 51 modified to account for the exit quality being less than 100%. See Figure 32.

Another method for correlating two-phase pressure drop is by means of the integration of the fog-flow equation

$$dP = \frac{G^2}{g_c} \left[v_g dX + \left(\frac{f dL}{2 D_e} \right) (v_f + X v_g) \right] \quad \text{Equation 52}$$

*Lockhart, R.W. and R.C. Martinelli. "Proposed Correlation of Data for Isothermal Two-Phase Two-Component Flow in Pipes," Chem. Engr. Prog., Vol. 45, No. 39, 1949.

For $v_g \gg v_f$ and a constant heat flux $\left(\frac{dX}{dL} = \text{const}\right)$, this equation becomes

$$\Delta P = G^2 \frac{v_g}{g_c} X_e \left(1 + \frac{fL}{4D_e}\right) \quad \text{Equation 53}$$

For large L/D 's, as is usually the case,

$$\Delta P = \frac{v_g X_e}{g_c} f G^2 \frac{L}{4D_e} \quad \text{Equation 54}$$

can be used since the acceleration term ($v_g dX$ in Equation 52) is negligible.

6.4 Forced Convection Once-Through Boiling Stability Considerations

There is no single cause for flow instability. Because several effects may cause flow instabilities, there may be many solutions to the problem of damping out the fluctuations. Slug flow (alternate bubbles and liquid slugs) in a gravity field will give rise to flow fluctuations. In small tubes capillary slug flow can cause flow fluctuations. A tube in a vertical gravity field filled with a bubbly liquid at the local saturation temperature is in a state of unstable equilibrium: if a bubble happens to form (e.g., near the top of the tube) which can be considered connected to a constant pressure plenum, a reduction in hydrostatic pressure causes the liquid to pass into the superheated state and cause more bubbles to form. This "snowballs" into a greater pressure reduction, with the tube finally ejecting the liquid by the resulting high vapor flow.

Another phenomenon somewhat similar in nature is the flow oscillations caused by pressure pulses which travel up and down a tube as in the operation of an aeropulse or organ pipe. The mechanics of this type of oscillation does not depend on gravity. The equation for computing the frequency of oscillation is quite simple and can be found in texts on propulsion or physics, namely:

$$f = \frac{a}{4L} \quad \text{cps}$$

where f is frequency, a is velocity of sound propagation and L is length of pipe.

In the flow of a two-phase fluid in which the phases are well mixed, the sonic velocity of the mixture can be considerably less than the sonic velocity in either of the two phases. As a result, very low frequency pulses can be observed in long boiler tubes. One-tenth cycle per second pressure pulses were observed in the SNAP I boilers discussed in Section 7.1.1.

A considerable amount of practical experience has been accumulated at TRW on the problem of hydrodynamic instabilities in heated channels. It has been observed that

TABLE V
BOILER INSTABILITIES DURING TEST

Boiler	$\frac{\text{Pressure Drop}}{\text{Inlet Pressure}} \times 100$	Remarks
SNAP I See Section 7.1.1	6.7% at 225 psia inlet $G = 56 \text{ lb/sec-ft}^2$	Subcooling had to be reduced to less than 100°F ($T_{\text{Sat}} - T_{\text{in}}$) to prevent pressure oscillations.
STE Boiler See Section 7.1.3	16% at 247 psia inlet $G = 56 \text{ lb/sec-ft}^2$	Subcooling had to be reduced to less than 300°F ($T_{\text{Sat}} - T_{\text{in}}$) before pressure oscillations were eliminated. Pressure changes of approximately 40 psi with frequencies of 0.088 cps were noted during unstable operation. Flow rates varied by 20%.
Thirteen Tube Boiler See Section 7.1.4	58% at 260 psia inlet $G = 130 \text{ lb/sec-ft}^2$	580°F subcooling ($T_{\text{Sat}} - T_{\text{in}}$) was observed without pressure oscillations. When inlet pressure was reduced to approximately 90 psia (43 psia outlet) pressure oscillations were noted.

pressure drop or friction ($f L/D$)* across the boiler, the amount of subcooling at the boiler entrance and the boiler pressure level are the variables by which an experimenter can control the pressure oscillations in a boiler. In boilers designed for large pressure drops (large percentage of inlet pressure), no amount of subcooling could induce pressure fluctuations, whereas in those boilers designed for small pressure drops, subcooling had to be eliminated to stop pressure fluctuations. In boilers with intermediate pressure drops, pressure fluctuations could be induced at will by controlling the amount of subcooling. Table V summarizes some observations of boiler instabilities during tests. Low pressure levels below the design point as during startup also produced instabilities, as noted in Table V.

In a forced convection system in which gravity effects are minimized, fluctuations are observed in flow systems in which several possible flow rates exist for the same total pressure. The total pressure drop through a smooth, heated tube during a change in

* f - friction factor, L - tube length, D - tube diameter

phase assuming fog flow is expressed as:

$$\Delta P = Aw^3 - Bw^2 + Cw \quad \text{Equation 55}$$

where ΔP is the pressure drop of the fluid undergoing change in phase, w is the weight flow, and A , B , and C are constants involving fluid properties, friction factor, heat flux and geometry. Such an equation gives three different flow rates corresponding to a single pressure drop value and thus unstable operating conditions may be present (see Figure 21, Curve A). Derivatives of this equation establish the conditions for stability. The tangent to the slope curve drawn through the origin point of the coordinates $\frac{d(\Delta P)}{dw}$, w represents the first derivative of the required pressure drop across the orifice or flow restriction needed for stability. This is the additional pressure drop required to convert the form of the ΔP versus w curve from a to b in Figure 21.

The analysis shows that when the restrictive diameter approaches infinity or the restriction pressure drop required approaches zero, the following stability criterion results:

$$B - \sqrt{3AC} \leq 0$$

from which the critical value of subcooling ($T_{\text{Sat}} - T_{\text{in}}$) can be computed as a function of boiling pressure for a tube without restrictions. A curve such as that in Figure 22 can be drawn.

The Thirteen Tube Boiler referred to in Table V and discussed in Section 7.1.4 showed flow instabilities at low pressures but as the boiler load was increased to the design point, stability was obtained. Further analysis shows that (1) there is a pressure above which the boiling section is stable for all values of subcooling, (2) there is a value of subcooling ($T_{\text{Sat}} - T_{\text{in}}$) below which the boiling section is stable at any pressure level, and (3) there is a value of fL/D above which the boiling section is stable at any pressure level. These effects are illustrated in Figure 23.

If by test or analysis boiling instabilities are indicated for a fixed value of inlet subcooling, increased frictional resistance is required as shown in Figure 23.. This is best accomplished by means of swirl devices, if possible, as heat transfer coefficients are correspondingly increased. Localized restrictions, on the other hand, play no part in the heat transfer.

Boilers can be developed that will operate in zero gravity as well as on the ground. Flow fluctuations in which gravity plays a role can be eliminated by reducing the importance of the gravity effect. This may be accomplished by means of high velocities (high Froude numbers) and swirl "g" fields.

7.0 ANALYSIS OF TEST RESULTS

7.1 Forced Convection Mercury Boiler Tests

This section analyzes and discusses the once-through forced convection mercury boiler tests and supplemental tests run on the SNAP II program. Sections 7.1.1 and 7.1.3 also discuss SNAP I once-through forced-convection mercury boiler tests. Comparison of the heat transfer and pressure drop data is based on the correlation procedures derived in Section 6.0, "Theoretical Analysis and Data Correlation Procedures."

7.1.1 SNAP I Boiler

The first useful information on the once-through forced convection boiling of mercury was obtained from testing the SNAP I boilers designed by The Martin Company. A set of these boilers (two in parallel) was tested at TRW. Each boiler had 23.7 feet of coil wrapped in a 4.6 inch diameter helix. A 3/8-inch outer diameter, 1/4-inch inner diameter stainless steel tube was used. The tube had a 0.109-inch plugged center tube with a 0.051-inch wire in the annular mercury flow passage. Electrically heated lead surrounded the coil, producing the package shown in Figure 24. The data obtained in these tests were less than desired since the exit quality and effective boiling length could not be accurately measured. From the data available, however, the three points shown in Figures 29 and 30 were obtained at the following conditions:

$$W_{Hg} = 0.93 \text{ lb/min - tube}$$

$$T_{Hg_{in}} = 900^{\circ}\text{F (subcooled)}$$

$$P_{Hg_{in}} = 225 \text{ psia}$$

$$T_{Hg_{out}} = 1350^{\circ}\text{F (measured)}$$

$$P_{Hg_{out}} = 210 \text{ psia}$$

$$P_{\text{lead bath}} = 15 \text{ psia}$$

$$T_{\text{lead bath}} = 1400 - 1450^{\circ}\text{F}$$

$$X_{\text{exit}} = 75\%$$

It should be noted that a superheated exit temperature reading was obtained although the exit quality was less than 100 percent. The vapor and liquid are not in equilibrium and the vapor temperature has the major effect on the thermocouple reading.

The heat transfer data obtained during these tests are shown along with bare tube data in Figures 28 and 29. The SNAP I boiler heat transfer coefficients are greater than those obtained from bare single tube boiler tests run at TRW by approximately a factor of 3. Since the exit qualities for the bare tube boiler were less than one and for the SNAP I boiler approximately one, an extrapolation was made in computing this factor which assumes that both the heating surface and heat transfer required to obtain a quality of one for the bare tube boiler are directly proportional to the exit quality. This is attributed to the radial accelerations imposed on the flowing mercury droplets by the spring inserts. Data obtained by other experimenters* on single-phase fluids (water, air, etc.) also indicated an improvement in the heat transfer coefficient of approximately a factor of 3 for swirl inserts. On the basis of Figure 28, the factor of 3 appears to be reasonable. The data reduction procedures of Section 5.2 were used to obtain these points. Application of the pressure drop equation (Equation 54) at the design conditions results in a boiler friction factor of:

$$f_b \simeq 0.16$$

During the operation of the two boilers in parallel, an unstable operation was noted. This unstable condition is utilized in Table V in Section 6.4 as one point in a once-through forced convection mercury boiling coordinate system. Stable operation was obtained by providing preheat before the high ΔT lead heated part of the boiler. The SNAP I data provided the only available swirl device data at the time of the design of the Thirteen-Tube Boiler. During the initial phases of the program these data were also the only high quality vapor data available.

7.1.2 Preliminary Test Sections

7.1.2.1 No. 1 Plain Tube

The first liquid metal (Na) to mercury boiler tested in the SNAP II Boiler Development Program was tested in May, 1959. The boiler had 34 feet of 1/2-inch tube (0.049-inch wall) in a 1-5/16-inch tube (0.049-inch wall) helically coiled on a 17-3/4-inch inner diameter with a 3-3/16-inch lead. All material was Type 347 stainless steel. Sodium flowed in the annulus with mercury flow in a counter-current direction in the 1/2-inch tube. Figure 25 is a photograph of the boiler.

Instrumentation and testing procedures were basically as discussed in Section 5.1 (in Figure 5 the external to boiler superheater path being used). Surface thermocouples as discussed in Section 5.2 were used to measure the variation of sodium temperature with length. Figure 26 resulted. These variations of sodium temperature indicated the approximate transition from preheating to boiling and allowed a definition of a quality or boiling section length. Table VI presents the steady state data points obtained, while Table VII is the data reduced according to the procedures of Section 5.2.

*Gambill, W.R. et al, "Heat Transfer, Burnout, and Pressure Drop for Water in Swirl Flow Through Tubes with Internal Twisted Tapes," Oak Ridge National Laboratory Report 2911, March 28, 1960.

TABLE VI
PRELIMINARY TEST SECTION NO. 1
TEST DATA

RUN	FLOW RATE		POWER		TEMPERATURE, °F				PRESSURE, PSIG			
	W _{Na} Lb/Min	W _{Hg} Lb/Min	KW Na		Na In	Na Out	Hg In	Hg Out	Na In	Na Out	Hg In	Hg Out
1	4.3	25	23.0		1303	880	435	864	14.4	14.3	60	54
2	4.3	24.6	19.0		1199	810	430	855	13.2	13.2	60	53.5
3	4.6	25	18.0		1098	745	427	849	14.9	14.7	60	53.5
4	5.0	10.7	10.5		1015	890	603	842	14.9	14.3	54	52
5	4.7	17.7	12.2		1020	815	560	840	12.0	12.0	56	53
6	4.6	24.8	15.9		1020	688	433	845	13.25	13.25	58	53.5

TABLE VII
PRELIMINARY TEST SECTION NO. I
REDUCED DATA

Run	P_{Hg} Avg. psia	Heat Rate, Btu/hr		Vapor Section Only			Quality, %
		Q_{Na} Across Boiler	$Q_{Preheat}$	Heat Flux Q/A , Btu/Hr-Fr ²	ΔT_m °F	U , Btu/Hr-°F-Fr ²	
1	71.7	34,000	20,900	4,440	336	13.1	6.97
2	71.7	30,200	20,400	3,760	259	14.5	6.56
3	71.7	29,300	20,600	3,430	180	19.0	4.63
4	67.7	11,230	5,000	1,510	131	11.5	7.74
5	69.2	17,350	9,670	1,880	119	15.8	5.76
6	69.7	21,550	19,975	3,020	104	29.8	0.86

Accurate determination of heat transfer coefficients was somewhat hampered as the low exit qualities (< 30 percent) made it impossible to determine the exit thermodynamic state points by pressure and temperature measurements. In addition, the small amount of total heat transferred required more accurate estimates of heat loss than were available.

The heat transfer data obtained in the testing of this boiler are summarized by means of Figures 27 and 28 as suggested in Section 6.2.3. Figure 29 is also presented to bring out the effect of quality and to obtain a comparison with all one-phase gas flow based on Dittus-Boelter Equation No. 10.

A definite trend of the data is shown in Figures 27 and 28. It should be noted that the combined preheating and boiling section has a generally higher heat transfer coefficient than the boiling section alone due to the higher coefficients in the preheat section. Figures 28 and 29 also show the points obtained from the SNAP I boiler tests at TRW (See Section 7.1.1) adjusted by a factor of 3 to account for the swirling action of the mercury drops as a result of the wire swirl device. The "Theoretical Considerations" of Section 6.2.1 (Equation 27) indicate the importance of a centrifugal force in the equation. The mercury and sodium pressure drops through this boiler were so low that correlations were impractical.

7.1.2.2 No. 2 Twisted Ribbon Insert

This boiler was tested in July, 1959 under conditions approximating those of the boiler discussed in Section 7.1.2.1. To determine the effect of a swirl device on the boiler performance, this boiler was identical to that discussed in Section 7.1.2.1, but a 0.40 inch by 0.040 inch twisted ribbon with a 1-foot pitch was inserted.

Materials, instrumentation, insulation and testing procedures were basically the same as those of Preliminary Test Section No. 1. One improvement was the use of thermocouples buried in the 6 inch insulation to give an idea of the insulation temperature distribution to check heat loss. The resulting estimate of heat loss was about 60 Btu/min. This could not be accurately determined, although an approximate check was obtained by comparing the sodium heat loss with no mercury flow, heat flow through the insulation based on the insulation thermocouples, and calculated heat loss based on surface conditions and surface heat transfer coefficients.

The specific determination of performance was difficult for this test as well as for Preliminary Test Section No. 1 (see Section 7.1.2.1). Nevertheless, the data presented in Table VIII were reduced, resulting in Table IX, and found to have roughly a 50 percent

TABLE VIII

PRELIMINARY TEST SECTION NO. 2 TEST DATA

Run	W _{Hg} lb/min	W _{Na} lb/min	T _{NaIn} °F	T _{NaOut} °F	P _{HgIn} psia	P _{HgOut} psia	T _{HgIn} °F	T _{x=0} [*] °F	T _{HgOut} °F
1	3.8	4.0	1125	925	72	71	600	860	855
2	3.8		1110	950	72	70	475	860	850
3	10.0		1112	840	79	70	670	875	850
4	21.8		1195	720	83	71	455	885	853
5	22.0		1412	782	93	72	358	900	853
6	10.2		1327	1000	75	70	610	870	853
7	4.2		1314	1110	71	71	559	865	860
8	16.0	36.0	1300	1238	98	71	590	910	853
9	16.0	35.4	1300	1238	89	55	594	890	816
10	4.2	33.7	1282	1272	32	33	602	760	758
11	16.2	34.0	1290	1238	118	103	557	935	903
12	10.0	11.0	1321	1166	54	35	660	825	760
13	10.0	29.5	1306	1257	57	36	657	840	766
14	10.3	4.0	1043	821	76	70	682	875	850
15	3.8	4.0	1038	874	71	71	573	868	852
16	22.0	4.3	1021	673	86	70	470	886	848
17	10.0	30.0	1108	985	155	85	670	980	868
18	4	30.0	1213	1194	36	34	562	780	757
19	9.0	32.0	1061	959	109	54	685	920	812
20	9.9	29.5	1246	1115	167	81	660	985	873
21	22.4	16.0	1087	973	62	38	497	840	772
22	20.0	29.0	1280	1191	86	52	514	886	812
23	22.2	20.5	1283	1156	96	62	484	910	826
24	19.6	10.5	1239	1022	76	52	491	872	819
25	10.0	10.4	1260	1100	56	35	650	830	762
26	10.0	19.8	1246	1163	56	35	651	830	762
27	18	4.0	1410	293	108	65	568	925	843

*This temperature was taken to be the saturation temperature at the inlet pressure.

TABLE IX
PRELIMINARY TEST SECTION NO. 2 REDUCED DATA

Run	Q_{Na} Btu/min	Q_{Hg}^* Btu/min	Q_{PH} Hg Btu/min	Q_B Hg Btu/min	X_E Wgt Fraction	ΔT_m °F	U Btu/Hr-Ft ² -°F	Nu	Re	$\frac{\mu h_{fg}}{k \Delta T_m}$
1	240	180	31	148	.310	185.6	20.1	106	5.82×10^4	17.83
2	192	132	47	84	.176	160.4	17.0	89.6	5.82	20.63
3	326	266	64	202	.161					
4	570	510	296	213	.0779					
5	756	696	176	520	.1881					
6	392	332	84	247	.1932	267.4	25.7	135.5	15.5	12.37
7	244	184	41	143	.272	339.8	11.3	59.6	6.45	9.74
8	669	609	163	446	.222	386.4	32.7	172	24.5	8.56
9	658	598	152	446	.222	412	30.1	163.4	25.2	8.03
10	101	41	23	18	.0352	606	1.42	8.4	7.2	5.46
11	530	470	196	274	.1347	340	28.6	142	23.5	9.74
12	511	451	52	399	.3133	440	21.2	121.5	17.3	7.52
13	433	373	57	316	.2519	482	16.0	93.9	17.2	6.86
14	266	206	61	144	.1116					
15	196	136	35	101	.2116	52.3	54.1	285	5.82	63.29
16	449	389	294	94	.034					
17	1107	1047	96	951	.7565	60.6	35.8	183.7	15.0	52.12
18	171	111	29	81	.1627	441	5.2	30.5	6.9	7.50
19	979	919	66	852	.752	113.5	167	807	14.2	29.63
20	1151	1091	99	992	.797	231	97.8	502	14.8	14.33
21	547	487	248	238	.0847	210	48.0	282	38.5	15.76
22	774	714	238	476	.1893	383	38.6	208	31.5	8.64
23	781	721	304	417	.149	340	43.9	234	34.6	9.73
24	683	623	239	384	.156	262	49.3	263	30.7	12.63
25	499	439	58	381	.3033	374	24.3	142	17.3	8.85
26	493	433	58	375	.298	406	22.0	140	17.3	8.15
27	740	680	203	477	.2108					

* $Q_{Hg} = Q_{Na} - 60$ Btu/Min Heat Loss

higher heat transfer coefficient than the bare tube of Section 7.1.2.1. Figures 30 and 31 were then made to compare the three sets of once-through forced convection mercury boiling data available. The SNAP I data were corrected by a factor of one-third and the twisted ribbon insert by a factor of two-thirds. This compilation of data was used to design the Thirteen-Tube Boiler discussed in Section 7.1.4.

The twisted ribbon insert induced mercury pressure drops to the extent that a pressure drop correlation was possible based on Section 6.3 and is presented in Figure 32. The gas-only pressure drop used to obtain Φ was determined by means of the friction factor obtained by flowing nitrogen through the boiler (see Section 7.2.3).

To supplement the overall data correlations, several runs were analyzed on a local basis where the boiler length is divided into increments and analyzed. Figures 35, 36, 37 and 38 present the results of analyzing the runs with the sodium temperature distributions of Figures 33 and 34. An attempt to obtain a general correlation is presented in Figure 39. The variation of coefficient with quality and length is as expected, but the various runs do not group for a general correlation.

7.1.3 SNAP I STE Boiler

As part of the SNAP I program, the Systems Test Enclosure (STE) boiler was designed and tested at TRW. This boiler replaced the two boilers discussed in Section 7.1.1. The data obtained in the SNAP I Boiler and Preliminary Test Sections 1 and 2 were used to design this boiler. The preliminary test section data were utilized to estimate the effect of mass velocity (Re) and ΔT_m , while the SNAP I swirl wire insert was used to give a three-fold increase in heat transfer coefficient over that of a bare tube. The resulting design was a stepped tube configuration 97 feet long having 4 feet of 1/4 inch tubing (0.049 inch wall), 15 feet of 3/8 inch tubing (0.049 inch wall) and 78 feet of 1/2 inch tubing (0.065 inch wall). Wire swirl devices were inserted inside the tubes. The resulting wire inserts were a 0.028 inch - 1/2 inch pitch in the 1/4 inch tube, a 0.051 inch - 1 inch pitch in the 3/8 inch tube, and a 0.081 inch - 1-1/2 inch pitch in the 1/2 inch tube.

The boiler consisted of 67 coils formed on a 4 inch helix diameter. Stepped tubes were employed to help eliminate the instability encountered in the SNAP I test. However, the adequacy of the stepped tubes in preventing instability could not be divorced from the stabilizing effect of the increased pressure drop (fL/D). The STE boiler had a total pressure drop of 35 to 40 psi while the SNAP I boiler had a pressure drop of 15 psi. Radiant heaters surrounding the coil provided the thermal energy.

The design performance of the STE boiler follows:

$$W_{Hg} = 1.86 \text{ lb/min}$$

$$T_{Hg_{in}} = 830^{\circ}\text{F}$$

$$P_{Hg_{in}} = 252 \text{ psia}$$

$$T_{Hg_{out}} = 1360^{\circ}\text{F}$$

$$P_{Hg_{out}} = 215 \text{ psia}$$

$$X_{exit} = 100\%$$

The data obtained during the test are presented in Figure 40 along with the SNAP I boiler test results. No correction of data was made to account for the similar swirl devices. Figure 40 shows that the STE boiler data correlate along a straight line (1:1 slope) indicating a heat balance for complete evaporation. No adequate means of determining the actual boiling section length existed, although the correlation of Figure 40 indicates that the boiler was sufficiently large to provide complete evaporation over the range of flow rates and mean temperature differences that were tested. Figure 40 is effectively plotting

$\frac{h}{W^{0.8}}$ vs ΔT_m . A heat balance gives:

$$Q = hA_b \Delta T_m = Wh_{fg} \Delta X$$

$$\therefore \frac{h}{W^{0.8}} \approx X \frac{W^{0.2}}{A_b \Delta T_m}$$

Equation 56

The variation of flow during the test was not sufficient to have a pronounced effect and, as mentioned, the boiling area variation was not measurable, resulting in constant A_b . Thus Figure 40 is nothing more than Equation 56. However, since the actual boiling length required is less than the total, the coefficients are greater than those calculated from Equation 56, and Figure 40, therefore, represents minimum boiling coefficients. For similar reasons, the boiling section pressure drop friction factor based on Equation 54 could only be estimated as less than 0.15. Here again the boiling section length cannot be distinguished from the total length.

7.1.4 Thirteen Tube Boiler

The design objectives of the Thirteen Tube Boiler were basically:

- (a) Design a full-capacity SNAP II type boiler.
- (b) Obtain design data for more advanced high performance boilers.

It was important to obtain a full-capacity boiler (i.e., provide stable flow at the sodium and mercury conditions specified in Section 2.0, Table II) early in the SNAP II boiler development program and to demonstrate the feasibility of a multiple tube once-through forced convection boiler. Data for advanced designs were lacking at the time, because 100 percent quality once-through forced convection mercury boiling had not been reported in the literature. The resulting boiler is shown in Figure 41.

The counterflow boiler had 13 tubes of 5/16 inch diameter and 0.035 inch wall thickness. The six outer tubes were valved to permit a change in the number of active tubes from 13 to 7. By removing the header, blocking off the internal 7 tubes and reassembling, the number of active tubes could be reduced to one. Sodium passed inside a 1.900 inch O.D. tube (0.065 inch wall). The tubes in a shell were coiled 21-1/2 times on a 19-1/2 inch mean diameter. The total dry weight was 388 pounds while the sodium inventory was about 130 pounds and the mercury only 4-1/4 pounds at the design conditions with all 13 tubes in use. Total wet weight was 518 pounds. The counter-flow heat exchanger had mercury flowing through 13 (to 7) 5/16 inch tubes (0.035 inch wall).

The Thirteen Tube Boiler was designed to certain requirements not germane to currently conceived SNAP II prototype boilers. These included:

- (a) A meager amount of once-through forced convection mercury boiling data was available at the time of design.
- (b) An accurate method for accounting for the effect of coiling and flow obstructions on the sodium side pressure drop was lacking.
- (c) Available test rig dimensions dictated some of the overall dimensions of the boiler.

The heat transfer data available for use in the Thirteen Tube Boiler design were those shown in Figures 30 and 31. At the time, the theoretical analysis of Section 6.2.1 was not complete and the vapor-only correlation of Figure 30 was still under consideration. As shown in Figure 30, the vapor-only model did not correlate the SNAP I data reduced by a factor of three* to account for the swirl effect on heat transfer. The design curve of Figure 31 was thus favored as it correlated the Preliminary Test Section Data and the adjusted SNAP I data. Later, the theoretical analysis of Section 6.2.1 substantiated the correlating parameters of Figure 31.

The design curve assumed was conservative with respect to the data available to insure that full capacity (more simply, 100 percent vapor) was obtained.

*Gambill, W.R. et al, "Heat Transfer, Burnout, and Pressure Drop for Water in Swirl Flow Through Tubes with Internal Twisted Tapes," Oak Ridge National Laboratory Report 2911, March 28, 1960.

A swirl device geometrically similar to that used in the SNAP I boiler was used to increase the heat transfer coefficient. A three-fold increase in the coefficient was assumed over that of a bare tube. As a result, a minimum overall h ($\approx U$) of 12 Btu/hr-ft²-°F was expected with an estimated mean temperature difference of 118°F. The expected heat flux was thus 1420 Btu/hr-ft².

Pressure drop considerations were based on the similar spring swirl device data of the SNAP I boiler. Application of Equation 54 to the SNAP I data of Section 7.1.1 results in a $f = 0.16$. The Thirteen Tube Boiler was designed on the basis of this boiling friction factor and Equation 54. The expected pressure drop for the final design was about 85 psia. The sodium side pressure drop was estimated as about twice that based on frictional resistance alone to allow for non-calculable effects of, for example, the coiled multi-tube configuration. Due to fabrication and test rig dimension limitations an expected sodium side pressure drop of about 2.3 psi was used instead of the 2 psi required in Table II.

The coil diameter was retained at that of the preliminary test sections to expedite fabrication and installation into the test rig. The resulting 110 foot mean length of 1.900-inch O.D. shell coiled on a 19.5-inch diameter with 2-3/8 inch lead resulted. The heat transfer length based on the correlation of Figure 31 was computed with conservatism (lower part of the correlated data) to be approximately 112 feet. Thus the heat transfer requirements and rig limitations were considered satisfied.

The Thirteen-Tube Boiler was made from Type 304 stainless steel, which has adequate corrosion resistance for performance testing, was readily available and was suitable for use with sodium at high temperatures. More recent corrosion data has ruled out the use of Type 304 stainless for long-time use with mercury.

Six of the 13 tubes had valves to give the design the flexibility of greater mass flow variation. This feature added much to the usefulness of the Thirteen Tube boiler test.

The testing procedures of the Thirteen Tube Boiler were modified from those of the Preliminary Test Sections by the use of a choked nozzle at the boiler exit. Figure 42 shows the thermocouple and pressure instrumentation details for the Thirteen Tube Boiler. The nozzle allows the determination of an apparent vapor flow which would indicate the boiler exit quality from:

$$X (\%) = \frac{W_{\text{vapor out}}}{W_{\text{liquid in}}} \times 100$$

Thus the choked nozzle path of Figure 5 was used for the Thirteen Tube Boiler test.

Table X summarizes the Thirteen Tube Boiler design and actual performance. The expected performance is also compared with the actual performance. The actual performance was within the scatter of the data of Figure 31 used to design this boiler.

TABLE X
SUMMARY OF THIRTEEN TUBE BOILER DESIGN AND PERFORMANCE

	<u>Expected Performance</u>	<u>Actual Performance</u>
Number of Tubes	13	13
Hg Flow, lb/min	17.0	17.0
Hg Inlet Temperature, °F	425	425
Hg Outlet Temperature, °F	1150	1160
Hg Inlet Pressure, psia	190	206
Hg Outlet Pressure, psia	105	105
Hg Pressure Drop, psi	85	101
Na Flow, lb/min	42.5	46.0
Na Inlet Temperature, °F	1200	1200
Na Outlet Temperature, °F	1000	1000
Na Pressure Drop, psi	2.3	6.3
h_B , Btu/hr-ft ² -°F	12	45
ΔT_m , °F	118	135
q , Btu/hr-ft ²	1420	6000

The Thirteen Tube Boiler was tested in March, 1960 in the test rig described by Figure 5 (choked nozzle path). The test and instrumentation procedures of Section 5.1 were utilized. Table XI summarizes the steady state data points obtained. Surface temperature variations with length such as Figures 43, 44 and 45 were obtained for each of the points in Table XI.

Tables XII, XIII and XIV present the reduced data according to the procedures of Section 5.2, "Data Reduction Procedures."

All of the steady state operating points corresponded to an exit quality of 100 percent, although this value was not precisely established on the basis of heat balance and a comparison of the indicated vapor exit flow through the choked nozzle and the inlet liquid flow. Table XII presents these values. Indications that the exit quality was 100 percent are based on Figures 46 and 47. Figure 46 plots the sodium heat transfer versus mercury flow and compares the data to the heat transfer required to completely preheat, boil and evaporate the flow. It may be seen that the data vary linearly, thereby indicating the generation of 100 percent vapor. The data points would show a decrease in quality with increasing mercury flow rate beyond that at which a quality of one would occur at the boiler exit.

Figure 47 plots the measured choked nozzle inlet pressure with measured boiler liquid inlet flow. Also shown is the ideal choked nozzle curve. The fact that the data vary linearly also indicates that 100 percent vapor was being passed. If the moisture content increased with flow, which is to be expected, then the data would show a shift into the wet region.

The pressure drop distributions throughout the boiler length (e.g., see Figures 43, 44 and 45) were determined by means of the one-phase friction factors correlated in Figure 48 and the measured inlet and outlet pressures (see Equation 7). Figure 48 was determined from tests performed on the boiler using nitrogen. Application of the boiling section pressure drops to the correlating procedures of Section 6.3 resulted in Figure 49. The data is compared with other two-phase correlations in Figure 20. Due to data scatter, a positive correlation was not possible. It is expected, however, that the data has a negative slope similar to the condenser data of Figure 20.

It is also expected that a minimum value of $\Phi \approx 1.5$ (fog flow) exists for once-through forced convection boiling as it does for the other two-phase experiments of Figure 20. Care should thus be exercised in utilizing Figure 49 for internal configurations and

$$Re \left(\frac{v_g}{10} \right)^{1.25}$$

much different from those of the Thirteen Tube Boiler tests.

TABLE XI

THIRTEEN TUBE MERCURY BOILER TEST DATA

Run No.	No. Tubes	MERCURY PARAMETERS							SODIUM PARAMETERS			
		W lb/min	W/Tube lb/min	T _{in} °F	T _{out} °F	P _{in} psia	P _{out} psia	ΔP psi	W lb/min	T _{in} °F	T _{out} °F	ΔP psi
1	11	17.0	1.546	709	1173	250	106	144	46.0	1205	1015	6.2
2	11	17.2	1.563	690	1174	250	111	139	43.0	1209	1025	5.7
3	11	17.0	1.546	687	1212	271	116	155	45.8	1240	1041	6.4
4	7	17.0	2.43	656	1183	259	109	150	38.0	1211	1003	4.8
5	7	18.6	2.66	622	1174	244	117	127	40.9	1200	990	5.6
6	13	17.6	1.353	683	1171	239	116	123	46.0	1198	1016	6.3
7	13	17.0	1.308	800	1180	212	110	102	46.4	1214	1060	6.5
8	13	19.0	1.460	715	1189	229	116	113	46.0	1218	1045	6.3
9	13	20.0	1.540	683	1190	246	119	127	46.0	1215	1031	6.4
10	10	15.0	1.500	690	1189	225	108	117	40.8	1228	1067	5.6
11	7	15.0	2.14	722	1191	244	106	138	38.5	1233	1073	5.0
12	13	13.0	1.00	615	1169	184	104	80	38.5	1209	1066	5.1
13	13	18.6	1.430	505	1190	219	97	122	46.0	1224	1048	6.4
14	7	12.0	1.715	464	1152	237	70	167	30.5	1207	995	3.2
15	7	14.0	2.00	474	1156	240	85	155	35.8	1201	988	4.1
16	7	17.0	2.43	445	1159	248	101	147	40.7	1161	988	5.6
17	7	19.0	2.718	442	1155	231	108	123	42.5	1190	978	5.7
18	7	20.0	2.86	440	1162	233	112	121	45.7	1196	981	6.5
19	7	14.0	2.00	787	1060	141	81	60	20.7	1095	895	1.6

NOTE: To obtain boiling lengths see Table XIV.

TABLE XII

REDUCED DATA OF THIRTEEN TUBE BOILER TEST

MERCURY PARAMETERS

Run No.	W/Tube lb/min	T _{in} °F	T _{x=0} °F	T _{Sat. Out} °F	T _{out} °F	ΔT _{SH} * °F	Q _{Hg} ** Btu/min	P _{in} psia	P _{x=0} psia	P _{x=1} psia	P _{out} psia	ΔP _{tot} psi	ΔP _{ESH} psi	ΔP _B psi	W _{Hg} lb/min	W _{Nozz} lb/min	Q _{Na} Btu/min
1	1.546	709	1054	914	1173	259	2350	250	248	226	106	144	120	22	17.0	17.4	2550
2	1.563	690	1054	922	1174	251	2390	250	247	224	111	139	113	23	17.2	18.2	2490
3	1.546	687	1067	928	1212	284	2380	271	269	231	116	155	115	38	17.0	18.8	2770
4	2.43	656	1060	920	1183	263	2400	259	249	225	109	150	116	24	17.0	17.6	2325
5	2.66	622	1034	928	1174	246	2630	244	226	187	117	127	70	39	18.6	18.9	2610
6	1.353	683	1046	928	1171	243	2450	239	235	216	116	123	100	19	17.6	18.7	2610
7	1.308	800	1023	921	1180	259	2290	212	212	213	110	102	103	-1	17.0	17.7	2220
8	1.460	715	1038	930	1189	259	2640	229	229	210	116	113	94	19	19.0	NA	2490
9	1.540	683	1050	932	1190	258	2800	246	246	201	119	127	82	45	20.0	NA	2540
10	1.500	690	1034	918	1189	271	2090	225	225	195	108	117	87	30	15.0	17.4	2000
11	2.142	722	1050	916	1191	275	2070	244	244	262	106	138	156	-18	15.0	NA	1850
12	1.000	615	1000	912	1169	257	1830	184	184	195	104	80	91	-11	13.0	16.7	1652
13	1.430	505	1030	902	1190	288	2690	219	219	204	97	122	107	15	18.6	NA	2430
14	1.715	464	1044	858	1152	294	1750	237	230	206	70	167	136	24	12.0	11.3	1940
15	2.00	474	1046	884	1156	272	2040	240	225	170	85	155	85	55	14.0	NA	2290
16	2.43	445	1050	908	1159	251	2485	248	235	204	101	147	103	31	17.0	16.4	2130
17	2.72	442	1040	918	1155	237	2780	231	214	169	108	123	61	45	19.0	NA	2700
18	2.86	440	1040	924	1162	238	2930	231	216	176	112	121	64	40	20.0	18.1	2970
19	2.00	787	958	878	1060	182	2000	141	122	99	81	60	18	23	14.0	13.9	1930

NA - Not Available

$$*\Delta T_{SH} = (T_{out} - T_{Sat. out})$$

$$**Q_{Hg} = Q_{PH} + Q_B + Q_{SH} \quad (\text{Assuming 100\% Quality})$$

TABLE XIII
REDUCED DATA OF THIRTEEN TUBE BOILER TEST

Run No.	WT/N lb/min	q_B Btu/hr-ft ² x 10 ⁻³	ΔT_{MB} °F	U_B Btu/hr-ft ² -°F	Nu_B	$\left(\frac{\bar{\mu}_v h_{fg}}{\bar{k}_v \Delta T_m} \right)_B$
1	1.546	8.37	84	99.7	180.6	33.4
2	1.563	8.48	79	107.3	194.7	35.4
3	1.546	7.37	138	53.4	94.6	20.3
4	2.43	11.69	60	194.8	352	46.6
5	2.66	28.8	88	327	594	31.8
6	1.323	9.85	99	99.5	181.1	28.2
7	1.308	6.26	~153	40.9	75.0	~18.24
8	1.460	3.97	138	28.8	52.1	20.3
9	1.540	3.23	150	21.5	40.1	18.48
10	1.500	3.10	183	16.94	30.4	15.33
11	2.142	7.10	~165	43.0	77.0	~17.06
12	1.00	6.67	~184	36.2	61.4	~13.93
13	1.430	4.28	161	26.6	48.2	17.43
14	1.715	15.76	102	154.5	282	27.4
15	2.00	19.89	98	203	373	28.5
16	2.43	15.26	97	157.3	284	28.9
17	2.718	14.74	65	227	419	42.9
18	2.86	16.28	54	301	554	51.7
19	2.00	10.03	88	114.0	228	31.2

TABLE XIV

DISTRIBUTION OF THIRTEEN TUBE BOILER LENGTH

RUN NO.	L _{PH} FT	L _{SB} FT	L _B FT	L _{ESH} FT
1		8	22	83
2		15	22	75
3		8	25	79
4		44	25	44
5		80	11	21
6		11	16	85
7		0	25	86
8	LESS THAN A FOOT	0	44	68
9		0	58	54
10		0	58	54
11		0	36	75
12		0	18	94
13		0	40	72
14		32	13	67
15		68	12	32
16		58	19	35
17		73	22	17
18		74	21	17
19		83	24	5

123 FT = OUTER LENGTH OF SHELL

112 FT = MEAN LENGTH OF SHELL

The reduced boiling heat transfer data of Table XIII was correlated according to the procedures of Section 6.2.3 as shown in Figure 50. The exponent of the parameter,

$$\frac{\mu_v h_{fg}}{k_v \Delta T_m}$$

was found to be about 1.75. The heat transfer coefficient is thus inversely proportional to $\Delta T_m^{1.75}$ or the heat flux is

$$q \propto \frac{1}{\Delta T_m^{1.75}} \quad \text{Equation 57}$$

Figure 51 is presented as another instance where film boiling heat flux is inversely proportional to ΔT_m . The theory of Section 6.2.1 substantiates the order of magnitude of this correlation.

The heat transfer correlation of Figure 50 should be applied with prudence to internal configurations and mass velocities different from those of the Thirteen Tube Boiler.

The SNAP I boiler test points of Section 7.1.1 are also plotted on Figure 50 for comparison. These points were for ΔT_m of about 400°F and with a nearly constant temperature difference distribution.

The primary fluid side pressure drop is also an important consideration. Figure 52 presents Thirteen Tube boiler data obtained on sodium pressure drop. This curve may then be applied to the design of boilers with different geometries. Table 1 describes the Thirteen Tube boiler geometry in detail.

In general, the boiler was easily controlled and stable operation was obtainable. A wide variation in operating conditions was possible, as is seen from the steady-state points of Table XII. Transient runs were performed and it was found that careful control of the sodium heater power allowed the attainment of near equilibrium in about one hour. For example, Figures 53 a, b, c and d show the variation of parameters in a full-flow startup of the Thirteen Tube Boiler.

The data presented were taken on March 6, 1960, by using seven tubes of the Thirteen Tube Boiler. Previous to step introduction of full mercury flow (17 lb/min), the sodium flow was set at 41 lb/min, which was the steady state value previously obtained for similar conditions. The sodium inlet temperature was 1168°F and the outlet 1140°F. Previous to the startup it was established that the following conditions would be held during the start:

1. The sodium flow would be constant at 41 lb/min.
2. Mercury preheater heat input would not be modulated from its setting at the beginning of the startup.

3. Power input to the sodium heaters would be controlled to get a final value of 70 kw (the previously experienced steady state value) quickly without excessive sodium temperatures. Thus the sodium power variation of Figure 53a resulted in the sodium inlet and outlet temperatures of the same figure ($W_{Na} = \text{const}$).
4. Mercury flow would be changed from 0 to 17 lb/min as quickly as possible and held at 17 lb/min.

The startup was begun by an increase in the sodium heater power from 20 to 30 kw (20 kw being the steady value for no mercury flow). Seventeen pounds per minute of Hg was allowed to flow and reached the boiler in about 32 seconds. This flow could not be maintained throughout the startup, however, since the boiler inlet pressure approached that of the rig pump (about 300 psia) after about 16 minutes. The pump output was thus limited to about 15 lb/min until the boiler inlet pressure ($P_{Hg, in}$) dropped off. The variation of mercury parameters with time is shown in Figures 53a, b, c and d. These curves show that approximately one hour was required to attain equilibrium.

The large equilibrium time can be attributed primarily to the insulation equilibrium time. The boiler filling time is also significant as the rate of filling the 7 tubes at 17 lb/min is only 9 ft/min. The insulation equilibrium may be observed in Figure 53a, which shows the sodium inlet and outlet temperature variation. From 23 minutes on it may be seen that for constant heater power input and mercury and sodium flows the sodium temperatures are drifting downward as the insulation is reaching equilibrium.

The decrease in sodium outlet temperature is reflected in the increase of the "suppressed boiling length" as shown in Figure 53c. The initial portion of the startup time can be attributed to the heater power variation, filling and insulation temperature variation.

It may be concluded that a decrease in the filling requirements ("suppressed boiling length") and elimination of insulation temperature equilibrium effects (e.g., by decreasing the amount of insulation and/or having a more precise heater power input control) would significantly decrease the startup time.

The specific startup test and the various transients normally experienced during testing conclusively showed that controlled transients such as those represented by Figures 53a, b and c are possible with proper modulation of sodium heater input and mercury flow.

7.2 Supplemental Tests

7.2.1 Non-Wetting Porous Bed Experiments

Twenty hours of testing were logged on a mercury non-wetting porous bed boiler. To expedite the experimentation and to keep costs down, a standard porous filter was selected ("Selas," micro-porous porcelain, bacteriological filter, pore size less than 2 microns)

and tested at temperatures between 500 to 615°F (see Figure 54). The bed dimensions are

Figures 55 and 56 show the test apparatus and the Instrumentation used to measure the porous bed performance. The vertical liquid mercury column on the right of Figure 55 was used to pressurize the liquid mercury in the heat exchanger section. The pressure should be equal to, or greater than, the saturation pressure corresponding to the lead bath temperature. The maximum allowable pressure difference across the bed was set at 25 psi by the manufacturer. This is a structural limitation and could be increased by proper reinforcement. Surface tension of the liquid also limits pressure. The maximum allowable pressure difference across the interface can be estimated by:

$$P_f - P_v = \frac{4 \sigma \cos \theta}{D}$$

$$\text{For } D = 0.787 \times 10^{-4} \text{ inches (2 microns)}$$

$$\text{Note: } 10^{-4} \text{ cm} = 1 \text{ micron}$$

$$\sigma = 0.032 \text{ lb}_f/\text{ft}$$

$$\text{Assume } \cos \theta = 1$$

$$P_f - P_v = \frac{4 \times 0.032 \times 12}{0.787 \times 10^{-4} \times 144} = 135 \text{ psia}$$

The boiler pressure was measured by the height of the mercury column in the condenser, and the flow rate was measured by the volume of mercury collected per unit time. The true boiler pressure is greater than the boiler pressure measured by the pressure drop in the glass tubing. The glass tubing at the boiler exit was used to visually detect the presence of liquid due to a possible leak in the porous bed. The temperature of the lead bath and the exit boiler temperature were also measured.

To predict the heat transfer performance in the prototype range of boiler operation, one must extrapolate the data from the temperature range of 500 to 615°F by means of correlating parameters based on the proper mechanisms of heat transfer (see Section 6.2.2). Figure 57 shows the overall heat transfer coefficient plotted versus the parameter derived by analyses (Equation 47) and based on the flow resistance of the porous bed. Extrapolation of these data into the prototype range reveals that the predicted overall heat transfer

coefficient is above 300. In this range the resistances of the lead, container, and liquid mercury become important. The following approximate calculations indicate the conductance (excluding the resistance of the porous bed) of the test apparatus,

$$h \text{ (liquid mercury)} = \frac{k}{\delta} \simeq \frac{8 \times 12}{0.05} = 1930 \text{ Btu/hr/ft}^2\text{-}^\circ\text{F}$$

0.5 Inch is the estimated thickness of the liquid film.

$$h \text{ (steel container)} = \frac{12 \times 12}{0.1} = 1440$$

0.1 Inch is the estimated thickness of the stainless steel container.

$$h \text{ (lead bath)} \simeq 2000$$

$$\therefore \frac{1}{C} = \frac{1}{2000} + \frac{1}{1440} + \frac{1}{1930}$$

$$C = 584 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

In Figure 57, C equal to 584 represents the upper limit for U. The temperature difference used for computing U is the lead bath temperature minus the saturation temperature corresponding to the boiler pressure. Since the boiler pressure is measured downstream of the glass tubing venting the vapor to the condenser-manometer (see Figure 56), the measured pressure is less than the boiler pressure due to a pressure drop. This will make the measured U less than the true U based on the saturation pressure within the porous-plug-vapor space. Figure 58 shows the dependence of the correlating parameter of Figure 57 on the lead bath temperature. The overall heat transfer coefficient U versus the lead bath temperature is shown in Figure 59. This plot again shows an increasing U value with increasing hot fluid temperature. When this curve is extrapolated into the prototype range (1100°F), U's in the hundreds can be anticipated.

The experimental results presented have established the feasibility of obtaining heat transfer coefficients in the range 100 to 1000. Further testing in the prototype range is required to verify this. Improvements in performance could possibly be obtained from thinner and higher porosity beds.

Several serious difficulties exist with the application of the porous bed with a SNAP II type system. Some of the most serious include:

- (a) The maintenance of non-wetting properties over long periods of time.
- (b) Determination of the effect of clogging of the pores for extensive periods of time.
- (c) Structural limitations of the porous bed at the environmental condition encountered during launch and normal operations.

A considerable amount of fabrication, development and testing would have been required to provide a porous bed boiler with sufficient reliability for use in a SNAP II system. Therefore, the porous bed approach was deferred in favor of once-through forced convection which was in a more advanced developmental state.

7.2.2 Two-Phase Flow Tests

Simple two-phase adiabatic isothermal experiments were performed to aid in the understanding of the forced-convection boiling and condensing of mercury. Figure 60 shows the test rig. The tests used liquid mercury and nitrogen gas in a horizontal 0.394 inch I.D. "Pyrex" glass tubing 4 feet long. The tests yielded basic pressure drop and inventory data for the bare tube with various internal configurations. Visual observations of flow patterns and distributions indicated that the liquid phase (mercury) was always in drop form (the mercury never formed a continuous liquid film or layer). These observations led to the forced-convection drop heat transfer model analyzed in Section 6.2.1.

Since it appeared that swirl devices would improve the heat transfer performance during boiling, these tube inserts were investigated in the two-phase rigs. The insertion of swirl devices increased the total pressure drop considerably. At qualities (gas-to-total flow ratios) above 35 percent this increase was apparently due to the added resistance to gas flow in the tube rather than a new two-phase flow phenomenon caused by the swirling motion. The induced centrifugal forces were observed to decrease the drop size and cause the drops to follow the swirl. Pressure fluctuations were appreciably reduced. At qualities below 35 percent, however, all the drops did not follow the swirl wire and were "spilled" into the main gas stream. This resulted in turbulence and an associated sharp increase in pressure drop with more pronounced pressure fluctuations.

A comparison between swirl devices and bare tube data is shown in Figures 61 and 62. More elaborate two-phase flow experiments with swirl devices are being conducted at the present time. In particular, the influence of wire diameters and pitch on pressure fluctuations and the resulting flow stability are being investigated.

7.2.3 Sodium to Nitrogen Tests

Preliminary Test Section No. 2, previously described, was utilized in a sodium to nitrogen test to obtain greater insight into the characteristics of the sodium to mercury boilers of interest. Nitrogen was passed on the mercury side of the test section. Instrumentation was identical to that described in Section 7.1.2.2 except for the installation of a nitrogen flow meter. The test was run in October, 1959.

Analysis of the data resulted in the following conclusions:

1. The sodium side coefficients have a small effect on the overall heat transfer coefficient (U) for inside (N_2 and Hg) heat transfer coefficients of about 100 Btu/hr-ft²-°F or less.

2. The primary fluid Inlet and outlet (Na) thermocouple readings were within an accuracy of 0.75%.

Conclusion (1) is based on a Wilson reciprocal plot of the test results in which the wide variation of sodium flow (5.5 to 30 lb/min) did not affect the overall coefficient for U ($\approx h_i$) of 100 or less. Conclusion (2) is based on a heat balance in which the heat lost by the sodium was compared to that gained by the nitrogen (heat loss being negligible). The sodium and nitrogen temperatures had to be within 0.75 percent accurate as both heat rates depended on these temperatures.

7.2.4 Experiments on the Heat Transfer to Mercury Drops

These experiments were initiated to define more clearly the "drop-wise" or "dry-wall" boiling that apparently explained the low magnitude of coefficient experienced in early forced convection boiling tests. The results of these experiments were utilized in the theoretical development of Section 6.2.1

EXPERIMENTAL SETUP TO DETERMINE HEAT TRANSFER TO MERCURY DROPS

The sketch depicts the experimental setup used. (The lead bath gives the drop heating surface a uniform temperature). A correlation was first obtained between the mercury drop weight and diameter as observed by a microscope from above. Figure 63 presents

the correlation. Drops having 50, 100 and 150 mg with corresponding diameters of 0.080 inch, 0.102 inch and 0.114 inch were then evaporated on the heated surface at different plate (lead) temperatures. The time required to completely evaporate the drop was measured (0.74 to 1.97 min). The results are presented in Figure 64.

The local instantaneous heat transfer coefficient for drops based on the projected area is given by:

$$h_{\delta} = \frac{2 \rho_f h_{fg} d \delta}{\Delta T dt} \quad \text{Equation 58}$$

Equation 58 is derived from a heat balance. The mean h_{δ} can be computed by substituting

$$\frac{d \delta}{dt} = \frac{\delta_o}{t_o}$$

This can be done since the δ vs t curve is flat and the error was computed to be only 7 percent. Therefore

$$\bar{h}_{\delta} = \frac{2 \rho_f h_{fg} \delta_o}{\Delta T t_o}$$

A mercury droplet evaporated on an open hot plate exposed to air has a greater heat transfer rate than a droplet surrounded by mercury vapor and hot surfaces as would be the case when confined in a boiler tube. As a result, a correction should be applied to the drop heat transfer coefficient as computed from Equation 58 if it is to be applicable to boiler design. The difference in heat transfer coefficients is attributed primarily to the mass transfer of mercury vapor into the surrounding air and secondarily to radiation and convection effects due to the low temperature surroundings. In a heated boiler tube the radiation and convection heat flow is reversed. The sketch shows the heat flow paths from a drop on an open hot plate.

If the heat flow from the drop surface by mass transfer, radiation, and convection was reduced to zero for the same total heat flow to the drop, more vapor would be ejected

from the lower surface of the drop. As a result this increase in momentum produces a greater distance between the heating surface and the drop, resulting in a decrease in the heat transfer coefficient.

In order for the open hot plate data to be applicable to boiler design and to be comparable with the theoretical drop heat transfer coefficient as computed in Section 6.2.1 and shown in Figure 12, the measured drop heat transfer coefficients should be corrected (reduced) for the various heat flows, namely,

$$\bar{h}_{\delta_{\text{corrected}}} = C \frac{2 \rho_f h_{fg} \delta_o}{\Delta T t_o} \quad \text{Equation 59}$$

The correction factor C was computed to be approximately 0.6 when mass transfer, convection, and radiation effects were computed for the open hot plate tests.

Equation 59 was used to reduce the experimental drop data. The ΔT used in Equation 59 was determined from:

$$\Delta T = T_w - 640$$

An "effective" mercury saturation temperature of 640°F was used instead of 670°F ($T_{\text{sat}} @ 14.7 \text{ psia}$). This "effective" saturation temperature was determined by the asymptotic temperature approached by the data when the evaporation time θ_o was plotted versus the plate temperature.

The reduced data is plotted in Figures 65 and 66. These data proved very useful in revealing the mechanics of dropwise heat transfer without convection and in developing the theoretical analysis of once-through forced convection boiling of mercury presented in Section 6.2.1.

Visual observation of the evaporating mercury drops indicated the existence of some of the phenomena noted by Borishansky* with water and other wetting fluids. Since mercury is non-wetting, no spreading of the liquid was expected; however, at small temperature differences between the saturation temperature and the temperature of the heating surface ($\Delta T < 70^\circ\text{F}$), a tendency towards spreading of the drop was noted. This was probably the stable contact region in which the drop rested on the heating surface (see Section 6.2.1). As soon as this narrow temperature range was passed, the drops resumed their spherical shape.

As an extension to the data thus far obtained, hot plate tests are continuing to determine the effect of wetting which may occur with time or with additives in an actual boiler. The tests will be improved to exclude the effects of air around the drop.

*Borishansky, V.M., "Heat Transfer to a Liquid Freely Flowing over a Surface Heated to a Temperature above the Boiling Point," from Problems of Heat Transfer During a Change of State: A Collection of Articles by S.S. Kutateladze, AEC TR 3405, Leningrad, 1953.

8.0 RECOMMENDED DESIGN PROCEDURES

Section 7.0 discusses the tests performed on the SNAP II program and the heat transfer and pressure correlations obtained based on the theoretical analyses and correlating procedures of Section 6.0. A general boiler heat transfer and pressure drop design procedure has resulted. This section summarizes these design procedures in the form of Figures 67a to 67c which show the variation of heat flux with mass velocity, mean temperature difference and pressure drop for the PCS I boiler described in Table I. The SNAP II design conditions of Table II were used. Conservative values of the correlating parameters were used to insure complete evaporation and thereby avoid liquid carryover which could be highly detrimental to turbine operation. The basis for these curves is discussed below.

The most extensive data available for the design of a SNAP II once-through forced convection mercury boiler have resulted from the testing of the Thirteen Tube Boiler discussed in Section 7.1.4. These data provide the foundation for advanced designs whose tests will result in additional design data. The Boiling Research Program described in Section 9.0 will make available refinements in design procedures. Extrapolation of the design procedures to conditions grossly different from those under which the data were obtained should be made with care to insure that proper accounting is made for the differences.

Mercury side heat transfer characteristics were estimated by means of the correlation of Figure 50, which is a correlation of the Thirteen Tube Boiler data. The use of the correlation as well as the order of magnitude of the heat transfer coefficients is verified in Section 6.2.1 (see Figure 12).

A wire swirl device was used to obtain the Thirteen Tube Boiler data and should be used in designs based on the data. To maintain similar flow patterns a pitch-to-diameter ratio of 4 and tube-to-wire diameter ratio of about 5 should be used. This pitch-to-diameter ratio appears to be nearly an optimum based on recent experiments.* By the same token, extrapolation to Reynolds numbers much different from those of the test ($1 - 6 \times 10^4$) should be made with care.

The mean temperature difference (wall minus mercury saturation temperature) required to utilize Figure 50 requires some discussion. It may be seen from Figures 43 through 45 that a certain temperature distribution is associated with the Thirteen Tube Counter-flow Boiler mean temperature difference during boiling. This temperature distribution is characterized by low-temperature differences in the low quality region and high values in the high quality region. The SNAP I data points are also shown on Figure 50, but these points were for a

* Gambill's report referred to in Section 7.1.1.

constant temperature difference distribution of about 400°F. Also, Preliminary Test Sections 1 and 2 had temperature difference distributions of great variety. It thus appears that the correlation is relatively independent of temperature difference distribution.

Pressure drop in the boiling section may be predicted by means of the Thirteen Tube Boiler data of Figure 49 and Equation 54. The curves of this section are computed for a ϕ of 4.0.

For the PCS I boiler, Figures 67a through 67c summarize the boiling section's design curves that result from these heat transfer and pressure drop considerations. The decrease in heat flux with ΔT_m is not common, but can be explained by the dropwise heat transfer model. Figure 51 shows that Benzene behaves in a manner similar to that of mercury. As mentioned previously, extrapolation of the once-through forced convection boiling curves should be performed prudently pending further data from the Boiling Research Program described in Section 9.0.

The mercury side boiling heat transfer and pressure drop are the major boiler design considerations. Other considerations that must be made include:

1. Mercury preheat
2. Mercury superheat
3. Boiler stability
4. NaK pressure drop
5. Mechanical design, fabrication and inspection limitations

Mercury preheat has been studied and many references are available.* Typical heat transfer curves are presented in Figures 67a and 67b. Since it will generally be so small (velocities are about 1 ft/sec), the preheat pressure drop is not considered in Figure 67c. Figures 67a through 67c present heat transfer and pressure drop design curves for the superheating of mercury vapor. The heat transfer calculations are based on the conventional Dittus-Boelter equation. To be conservative, no improvement for the swirl device was incorporated. The pressure drop is based on the nitrogen test results performed on tubes with such inserts. Figure 48 plots the friction factor obtained.

*Lubarsky, B. and Kaufmann, S.J., "Review of Experimental Investigations of Liquid Metal Heat Transfer," NACA Report 1270, 1956.

The actual area needed for superheating has been found to be small in comparison to that required for boiling in counter-flow heat exchangers. The curves presented do not allow for the superheating that has been observed to occur simultaneously with the boiling.

Generally, the operational characteristics of the Thirteen Tube Boiler indicated that stable flow exists at design conditions. It may then be concluded, on the basis of the stability considerations of Section 6.4, that future SNAP II boilers with $f L/D$ and inlet pressure levels approximately equal to or greater than those established as stable in the Thirteen Tube Boiler will be stable.

NaK pressure drop estimates may be made by means of Figure 52, which correlates the primary fluid pressure drop for the Thirteen Tube Boiler test. Changes in configuration should be accounted for when this curve is used.

9.0 BOILING RESEARCH PROGRAM

9.1 Objectives

Although it will be possible to design satisfactory lightweight mercury boilers with the correlations established by the current boiler development program, it is still necessary and desirable to investigate the boiling phenomena of mercury. Mercury does not behave as other Rankine cycle fluids, principally because of its non-wetting characteristics. A program has therefore been initiated to establish local boiling mercury heat transfer coefficients as a function of local temperature difference, pressure, mass velocity, local quality and wetting. The research test rig has provisions for controlling all these variables, whereas in the boiler development tests only the mean temperature difference and the mass velocity were input variables. This program is expected to provide the basic thermodynamic data required for advanced boiler designs capable of high performance in a zero-gravity environment.

9.2 Approach

To examine the variables of mercury boiling outlined above, a test section will be designed and instrumented so that mercury of a known quality, mass flow, etc., will be introduced into the section. After heat has been added under controlled conditions, the exit quality of the mercury and other thermodynamic conditions will be accurately determined.

The exit quality state is measured by passing the vapor leaving the test section through a liquid-vapor separator. The liquid will be measured in a weigh tank while the vapor will pass to a condenser and then to a separate weigh tank. Liquid-vapor fractions can thus be determined to establish a check on the heat added in the test section.

9.3 Expected Results

A comprehensive compilation of a maximum number of mercury boiling variables is the goal of the boiling research program. The data resulting from this program will provide correlations and design curves which will permit the designer to produce a boiler of minimum weight and size with a minimum amount of uncertainty caused by extrapolation into untested conditions.

The effect of changing the temperature field on heat transfer, the relationship between exit quality and mean heat transfer coefficient for recirculation boilers, and the lower limit of mean temperature difference for improving the heat transfer flux will be determined from data made available in the boiling research program.

10.0 CONCLUSIONS

1. A successful once-through mercury boiler has been developed which provides the SNAP II design pressures, temperatures and flow rate under stable and reliable conditions.
2. The helically coiled shell and tube(s) configuration presents no serious fabrication or inspection limitations and minimizes mechanical stress problems.
3. Shell surface temperature measurements are adequate parameters of boiler performance and are useful in the reduction of data.
4. Ground operational procedures have been developed for current SNAP II boiler design.
5. A theoretical model has been developed for the once through forced convection boiling of mercury. This model also provides the required correlating parameters.
6. Procedures have been established for the complete design of SNAP II forced convection boilers.
7. A non-wetting porous bed boiler could generate vapor at SNAP II conditions with coefficients in the 100's. The fabrication problems of such a boiler are too severe, however, and further investigation is not justified.

11.0 RECOMMENDATIONS

1. Long duration tests of SNAP II boilers are required to establish time effects on boiler performance.
2. Advanced high performance SNAP II boilers which utilize existing boiler data for decreasing boiler weight should be designed and tested.

12.0 NOMENCLATURE

A	=	area
A_o	=	cross sectional area of pore
a	=	velocity of sound propagation
C	=	constant
C_D	=	coefficient of drag
C_p	=	specific heat at constant pressure
D	=	diameter
D_e	=	equivalent diameter
d	=	differential operator
F_o	=	force
F	=	function
f	=	friction factor, frequency
G	=	total mass velocity
g	=	gravitational constant
h	=	heat transfer coefficient; enthalpy
$\bar{h}$	=	area mean heat transfer coefficient
k	=	thermal conductivity
K	=	$\frac{\mu_v h_{fg}}{k_v \Delta T_m}$ once-through forced convection mercury boiling parameter
L	=	length
$\dot{m}$	=	mass flow rate
N	=	number of "g's" inertial force induced in droplets by swirl devices

Nu	=	$\frac{h D}{k}$ = Nusselt number
$\bar{Nu}$	=	time average Nusselt number
P	=	pressure
Q	=	heat transferred
q	=	heat flux
R	=	Universal gas constant
Re	=	$\frac{G D}{\mu}$ = Reynolds number
Re_o	=	Reynolds number at exit conditions
T	=	temperature
t	=	time
t_o	=	total resident time of drop
U	=	overall heat transfer coefficient
u	=	velocity
u'	=	vapor velocity escaping from under drop
v	=	specific volume
$\bar{v}$	=	mean specific volume
W	=	weight flow
X	=	quality $\left(\frac{W_{liquid}}{W_{total}} \right)$
y	=	distance from tube
α	=	swirl device angle
β	=	factor to account for Hg drop contact with heating surface
Δ	=	incremental change
δ	=	diameter of drop or droplet
δ'	=	mean film thickness under Hg drop

δ_o	=	initial diameter of drop
ϵ	=	slip ratio u_l/u_v
θ	=	contact angle
μ	=	viscosity
ρ	=	density
σ	=	surface tension
Φ	=	pressure drop ratio P_{Tp}/P_v - ref. to section 6.3
ϕ	=	area ratio A_o/A , pore area to total area
$\propto$	=	proportionality symbol

Superscripts

m	=	exponential power
n	=	exponential power

Subscripts

B	=	boiling
d	=	pertaining to the drop or droplet
e	=	exit
eff	=	effective
ESH	=	excess superheat
f	=	liquid
fg	=	change of state
g	=	gas or vapor
l	=	liquid
m	=	mean

PH	=	preheat
sat	=	saturation
SH	=	superheat
T	=	tangential; total
TP	=	two phase
TOT	=	total
v	=	vapor
w	=	wall
$x = 0$	=	saturated liquid
$x = 1$	=	saturated vapor
δ	=	pertaining to the drop or droplet

APPENDIX I: FIGURES

SIMPLIFIED IDEAL FLUID TEMPERATURE SCHEMATIC IN COUNTER-FLOW SNAP II BOILER

FIGURE 1

COMPARISON OF EXPERIMENTAL BOILING HEAT TRANSFER COEFFICIENTS FOR MERCURY WITH ADDITIVES

FIGURE 2

**VARIATION OF MAGNESIUM-AMALGAM BOILING COEFFICIENTS
WITH QUALITY IN A VERTICAL TUBE**

Ref: Korneev, M.I., "Investigation
of Heat Exchange of Mercury and
Magnesium Amalgam in Natural
Convection," Teplo Energetica
No. 7, 1955

FIGURE 3

THE EFFECT OF STAGNANT ZONE ON TEMPERATURE DIFFERENCE AT MERCURY INLET

FIGURE 4

SNAP II BOILER TEST LOOP SCHEMATIC

FIGURE 5

PRELIMINARY TEST SECTION NO. 2 INSTALLED
WITH SIX INCH SURROUNDING LAYER OF INSULATION

FIGURE 6

"VAPOR ONLY" MODEL OF ONCE-THROUGH FORCED CONVECTION BOILING

FIGURE 7

PROCESSES ASSUMED FOR COMPUTING THE EFFECTIVE SPECIFIC HEAT
OF A VAPORIZING MIXTURE

FIGURE 8

RELATIONSHIP BETWEEN MEAN HEAT TRANSFER COEFFICIENT AND OUTLET STEAM QUALITY

FIGURE 9

RELATIONSHIP BETWEEN HEAT TRANSFER COEFFICIENT AND VAPOR QUALITY FOR SODIUM AMALGAM

FIGURE 10

FILM BOILING DROP MODEL

FIGURE 11

COMPARISON BETWEEN THEORETICAL AND EXPERIMENTAL HEAT TRANSFER COEFFICIENT FOR MERCURY ON A HOT PLATE IN AIR

FIGURE 12

FILM VAPORIZATION OF A DROP BY FORCED CONVECTION

FIGURE 13

COMPARISON OF TEST DATA WITH EQUATION FOR FORCED
CONVECTION VAPORIZATION OF DROPS CORRECTED FOR
CONTACT FACTOR OBTAINED FROM HOT PLATE TESTS

FIGURE 14

HEAT TRANSFER TO HEMISPHERICAL LIQUID MERCURY DROPLET (POROUS BED BOILER)

FIGURE 15

MERCURY POROUS BED BOILER SETUP

FIGURE 16

SINGLE CYLINDRICAL PORE

FIGURE 17

HEAT TRANSFER COEFFICIENTS TO BOILING MERCURY IN SLOPED TUBES (FOG FLOW)

FIGURE 18

EFFECT OF REYNOLDS NUMBER ON RATIO OF MEAN VORTEX NUSELT NUMBER TO MEAN AXIAL NUSELT NUMBER

FIGURE 19

COMPARISON OF TWO-PHASE PRESSURE DROP DATA

FIGURE 20

EFFECT OF RESTRICTION PRESSURE DROP IN CORRECTING
UNSTABLE PRESSURE DROP CHARACTERISTICS
(a) TO STABLE FORM (b)

FIGURE 21

EFFECT OF INLET SUBCOOLING
AND BOILER PRESSURE ON
STABILITY

FIGURE 22

HYDRODYNAMIC STABILITY SURFACE FOR A ONCE-THROUGH BOILER
WITH EXIT QUALITY OF ONE

FIGURE 23

SNAP I BOILER

PRELIMINARY TEST SECTION NO. 1 INSTALLED

VARIATION OF T_{No} WITH DISTANCE FOR PRELIMINARY TEST SECTION NO. 1
0.402" I.D. x 34' LONG

FIGURE 26

PRELIMINARY TEST SECTION NO. 1 (PLAIN TUBE) TEST DATA
LIQUID AND BOILING SECTIONS

FIGURE 27

PRELIMINARY TEST SECTION NO. 1 (PLAIN TUBE) TEST DATA
BOILING SECTION ONLY

FIGURE 28

PRELIMINARY TEST SECTION NO. 1 (PLAIN TUBE) TEST DATA
BOILING SECTION ONLY

FIGURE 29

COMBINED ONCE-THROUGH FORCED CONVECTION MERCURY BOILING DATA
(FOR PREHEAT AND BOILING SECTION)

- Preliminary Test Section No. 1
- Preliminary Test Section No. 2 (Adjusted by 2/3 to account for effect of twisted ribbon swirl device with 1 foot pitch)
- △ "SNAP I" Boiler Data (Adjusted by 1/3 to account for effect of wire swirl device with 1 inch pitch).

FIGURE 31

TWO-PHASE BOILING PRESSURE DROP IN PRELIMINARY TEST SECTION NO. 2

FIGURE 32

VARIATION OF T_{Na} WITH DISTANCE FOR RUN NO. 5 OF PRELIMINARY TEST SECTION NO. 2

FIGURE 33

FIGURE 34

HEAT TRANSFER AS A FUNCTION OF QUALITY FOR RUN NO. 5
OF PRELIMINARY TEST SECTION NO. 2 BASED ON LOCAL DATA

FIGURE 35

PRELIMINARY TEST SECTION NO. 2
VARIATION OF LOCAL PARAMETERS
FOR RUN NO. 5

FIGURE 36

HEAT TRANSFER AS A FUNCTION OF QUALITY FOR RIJN NO. 20
OF PRELIMINARY TEST SECTION NO. 2 BASED ON LOCAL DATA

FIGURE 37

PRELIMINARY TEST SECTION NO. 2
VARIATION OF LOCAL PARAMETERS
FOR RUN NO. 20

FIGURE 38

COMPOSITE OF LOCAL CORRELATIONS FOR PRELIMINARY TEST SECTION NO. 2

FIGURE 39

STE BOILER TEST RESULTS
(SEE SECTION 7.1.3)

FIGURE 40

THIRTEEN TUBE BOILER

SURFACE AND IMMERSED THERMOCOUPLE AND PRESSURE MEASUREMENT
LOCATIONS FOR THIRTEEN TUBE BOILER

FIGURE 42

SODIUM AND ESTIMATED MERCURY EQUILIBRIUM TEMPERATURE DISTRIBUTION IN THIRTEEN TUBE BOILER

FIGURE 43

SODIUM AND ESTIMATED MERCURY EQUILIBRIUM TEMPERATURE DISTRIBUTION IN THIRTEEN TUBE BOILER

FIGURE 44

SODIUM AND ESTIMATED MERCURY EQUILIBRIUM TEMPERATURE DISTRIBUTION IN THIRTEEN TUBE BOILER

FIGURE 45

SODIUM HEAT LOSS VS. MERCURY FLOW FOR THIRTEEN TUBE BOILER

FIGURE 46

BOILER EXIT PRESSURE VS. INLET MERCURY FLOW COMPARED WITH IDEAL NOZZLE EQUATION
FOR THIRTEEN TUBE BOILER

FIGURE 47

ONE-PHASE THIRTEEN TUBE BOILER FRICTION FACTOR OBTAINED USING NITROGEN

FIGURE 48

THIRTEEN TUBE BOILER BOILING SECTION PRESSURE DROP DATA PLOT

FIGURE 49

THIRTEEN TUBE BOILER HEAT TRANSFER CORRELATION FOR BOILING SECTION

FIGURE 50

AVERAGE BOILING SECTION HEAT FLUX FOR BENZENE WITH 100% EXIT QUALITY

FIGURE 51

FIGURE 52

THIRTEEN TUBE BOILER STARTUP USING SEVEN TUBES
(ID = 0.2425")

FIGURE 53a

FIGURE 53b

THIRTEEN TUBE BOILER STARTUP USING SEVEN TUBES
(LENGTH OF LIQUID VS. TIME)
ID = 0.2425"

FIGURE 53c

FIGURE 53d

NON-WETTING POROUS BED AND LEAD CONTAINER

MERCURY POROUS BED BOILER EXPERIMENTAL SETUP

FIGURE 55

NON-WETTING POROUS BED EXPERIMENTAL RIG

OVERALL HEAT TRANSFER COEFFICIENT OF NON-WETTING POROUS BED
 PLOTTED ACCORDING TO EQUATION 44
 (Micro-porous Porcelain "SELAS" - Pore Size Less Than 2 μ Microns)

FIGURE 57

NON-WETTING POROUS BED PARAMETER VS. LEAD TEMPERATURE
ACCORDING TO EQUATION 47

FIGURE 58

NON-WETTING POROUS BED OVERALL HEAT TRANSFER COEFFICIENT VS. LEAD BATH TEMPERATURE

FIGURE 59

TWO PHASE FLOW RIG

COMPARISON OF SWIRL DEVICES WITH BARE TUBE IN TWO-PHASE FLOW

FIGURE 61

FIGURE 62

CORRELATION BETWEEN MERCURY DROP WEIGHT AND DIAMETER

FIGURE 63

RESULTS OF MERCURY HOT PLATE DROP EXPERIMENTS IN AIR

FIGURE 64

$\bar{h}_g$ VS. ΔT FOR MERCURY DROPS ON A HOT PLATE IN AIR

FIGURE 65

q VS. ΔT FOR MERCURY DROPS ON A HOT PLATE IN AIR

FIGURE 66

GENERAL DESIGN CURVES FOR SNAP II PCS I ONCE-THROUGH
FORCED CONVECTION PREHEATER, BOILER AND SUPERHEATER
(See Table II for Specifications and Table I for Geometry)

FIGURE 67a

GENERAL DESIGN CURVES FOR SNAP II PCS I ONCE-THROUGH
FORCED CONVECTION PREHEATER, BOILER AND SUPERHEATER
(See Table II for Specifications and Table I for Geometry)

FIGURE 67b

GENERAL DESIGN CURVES FOR SNAP II PCS I ONCE-THROUGH
FORCED CONVECTION PREHEATER, BOILER AND SUPERHEATER
(See Table II for Specifications and Table I for Geometry)

FIGURE 67c

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