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MERCURY RANKINE PROGRAM (SNAP 2)

**BOILER CONDITIONING
PHASE I RESULTS**

RELEASED FOR ANNOUNCEMENT
IN NUCLEAR SCIENCE ABSTRACTS

AEC RESEARCH AND DEVELOPMENT REPORT

**PREPARED UNDER SUBCONTRACT N843 FS-101221
FOR**

ATOMICS INTERNATIONAL
A Division of North American Aviation, Inc.

TRW EQUIPMENT LABORATORIES

A DIVISION OF TRW INC. • CLEVELAND, OHIO 44117

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MERCURY RANKINE PROGRAM (SNAP 2)

BOILER CONDITIONING

PHASE I RESULTS

By
ALFRED KOESTEL

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NAA-SR-6322

TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	5
I. SUMMARY	6
II. INTRODUCTION	8
III. TEST PROCEDURE	10
IV. THE MECHANICS OF THE DECONDITIONED STATE	13
V. CAUSES OF DECONDITIONING	33
DEFINITION OF SYMBOLS	47
LIST OF REFERENCES	49
APPENDIX A SLUG FLOW MERCURY BOILING HEAT TRANSFER	50
APPENDIX B SPRAY-ANNULAR FILM FORCED CONVECTION MERCURY BOILING	69

LIST OF TABLES

<u>Table</u>	<u>Title</u>	<u>Page</u>
B-1	Test Conditions	79

LIST OF FIGURES

<u>Figure</u>	<u>Title</u>	<u>Page</u>
1	Schematic Diagram of the Boiler Deconditioning Test Rig	12
2	Computed Non-Wetting NaK Temperature Profile	14
3	Measured NaK Temperature Profile	15
4	Typical Measured NaK Temperature Profile from Phase I Deconditioning Tests	17
5	Flow Regimes for Non-Wetting Mercury Flowing with Nitrogen in Glass Tubes	18
6	Typical Measured NaK Temperature Profile from Phase I Deconditioning Tests Showing High Exit NaK Temperature	20
7	Comparison of NaK Temperatures with Computed Curve for Slug Flow . .	21
8	Comparison of Slug Flow Heat Transfer in Low Pitch Plug with SNAP 8 Test Data	23
9	Suggested Spiral Plug Design	24
10	SNAP 8 Single Tube Boiler NaK Temperature Profile	25
11	Computed NaK Temperature Profiles	26
12	Suggested Multichannel Plug Design	27
13	Helical Flow Boiling Mercury Heat Transfer Coefficients in an Electrically Heated Tube	29
14	Effect of Pressure Level and Flow Rate on Deconditioning	35
B-1	Spray-Annular Boiling	70
B-2	Experimental Dimensionless Mass-Transfer Coefficient Versus Reynolds Number	73
B-3	Comparison of Theoretical NaK Temperature Profile	78

ABSTRACT

The Phase I Boiler Conditioning Tests were conducted to investigate the effects of independent variables on mercury boiler deconditioning or conditioning. No extensive data reduction was performed, and only sample calculations were obtained. Results of the testing indicated that the causes of deconditioning are the result of subtle interconnections among various variables. Control of these variables, as a method of preventing deconditioning, would involve an extensive development program. In the boiler design, deconditioning can be controlled by minimizing the effect of these variables.

I. SUMMARY

This report summarizes the results obtained from the Phase I boiler conditioning test. No extensive data reduction has been performed, and only sample calculations have been made.

The effect on boiler deconditioning or conditioning of independent variables such as pressure and temperature levels, NaK and mercury flow rates, time, and air and argon injection, were assessed. Output quantities as indicators of the deconditioned state were:

- 1) Boiler inlet and out pressure-time traces
- 2) Boiling noise as recorded on tape
- 3) Film of two-phase flow pattern recorded from X-ray image amplifier
- 4) NaK temperature profile
- 5) Boiler pressure drop
- 6) Timing of liquid slugs
- 7) NaK and mercury outlet temperatures

The boiling heat transfer mechanisms defining the deconditioned state appear to be dictated by a two-phase liquid slug flow pattern. An analysis connecting the resultant deconditioned NaK temperature profile with the slug flow pattern is developed. A means for controlling this phenomenon, which also forms the basis for the proposed Phase II tests, is presented.

Possible causes of deconditioning are postulated based on observed and measured symptoms as defined by the period of the liquid slugs in the slug flow regime, the roughness of the pressure traces, the recording of cavitation sounds measured by vibration pickup, and the effects produced by argon and air injection.

NAA-SR-6322

The control of deconditioning does not necessarily depend on the complete understanding of the causes. A design approach based on this premise is suggested. This approach uses the slug flow heat transfer analysis and defines the type of plug insert required for deconditioning control.

The causes of deconditioning are believed to be the result of subtle interconnections among such phenomena as wetting, nucleation, liquid superheat, and aging of liquid-wall interface. These phenomena, result in the two-phase liquid slug flow pattern as observed by X-ray. The means of preventing deconditioning by control of these variables appears more complex and would involve a more extensive development program than would be involved in the development of the means of deconditioning control.

II. INTRODUCTION

The purpose of this report is to summarize the results from Phase I of the boiler deconditioning tests. Phase I covers the period between March 15 and May 18, 1965. The type of tests which are to be conducted in Phase II of the boiler deconditioning test depend on the interpretation of the Phase I results as summarized in this report.

To enable the reader to develop the proper perspective of boiler deconditioning, it is recommended that the following SNAP 2 references on the subject be consulted:

- 1) Page 163 of Reference 1, "Mercury Boiling Research", NAA-SR-6309, TRW Report No. ER-4833, covers flow establishment boiling which is a poor boiling heat transfer regime and is related to deconditioning. Pages 66, 70, and 71 of Reference 1 deal with the transition of the flow establishment regime to high heat flux vortex boiling.
- 2) Reference 2, "The Mercury Rankine Program Boiler Deconditioning Problems and Planned Investigations", NAA-SR-10860, describes the possible causes of boiler deconditioning and discusses the planned investigations to be conducted by TRW Inc. and Atomics International. Phase I of TRW's Boiler Deconditioning Test is a part of this program.
- 3) Reference 3, "Control Atmosphere, Mercury Drop Evaporation Experiment", TRW Report No. ER-6169, gives evaporation time rates of mercury drops on various heated surfaces (oxidized 316 SS, oxidized HS 25, clean HS 25 and oxidized copper). The curve of evaporation time versus temperature difference between heated surface and saturation temperature is interpreted to be indicative of the degree of wetting (contact angle) between drop and surface.

NAA-SR-6322

A mercury boiler which is deconditioned exhibits exceptionally low local heat transfer rates, on the order of several magnitudes less than a boiler having maximum conditioning. These low heat transfer rates are identified by the small slope of the NaK temperature profile.

The cause of deconditioning is related in some subtle way to both the wetting phenomenon and to an unfavorable boiling two-phase flow regime (slug flow). The purpose of TRW's deconditioning work is to examine the hydrodynamic aspects, while the purpose of AI's deconditioning work is to investigate the wetting-nonwetting influences. Eventually, these two aspects must be related to one another so that a complete understanding of deconditioning may be obtained.

The philosophy of the deconditioning investigations is to study the two-phase flow mechanics associated with deconditioning and to design a boiler configuration for minimizing its effect. It is not the intent of this program to prevent the causes of deconditioning, but to consider the effects of these causes in the boiler design. It is believed that the prevention would require a greater experimental effort than is now planned. Prevention of deconditioning would require the control of surface determinants such as wetting and nucleation as well as the control of the two-phase hydrodynamics. The control of deconditioning by minimizing its effect on heat transfer would require the control of the hydrodynamics only. This would improve the reliability of boiler performance, whereas the control of the surface factors (cleanliness, surface finish, etc.) might prove so complex that reliable operation might be difficult to achieve. These ideas are supported by the test observations which are summarized herein.

III. TEST PROCEDURE

The following independent variables were manipulated in the Phase I tests:

- 1) Time - The effect of time on deconditioning was examined by operation of the boiler at SNAP 2 design conditions for a period of two weeks. The object was to determine the natural direction of the phenomenon and to observe its mechanics by means of X-ray and image amplifier
- 2) Flow Rate (NaK and Hg), Pressure Level (Hg) and Temperature Level (Hg and NaK inlet)
- 3) Boiler Inlet Conditions - The effects of inlet velocity (small inlet tube 0.051-inch ID) and nucleation was examined by argon injection and electric heat addition at the boiler inlet
- 4) Argon and Air Wash - The effect on deconditioning of argon and laboratory air flow over the hot boiler surface prior to mercury boiling was examined

The following output or dependent quantities were observed and recorded as indicative of deconditioning or conditioning:

- 1) NaK temperature profile
- 2) Boiler inlet and outlet pressure pulsations
- 3) Two-phase flow pattern and wetting or nonwetting state of the mercury were observed by X-ray image amplifier and recorded on film
- 4) Boiling sounds were observed by means of vibration pickup at boiler inlet and recorded on tape

By means of these measurements, the nature of the mechanics of deconditioning is defined.

Figure 1 shows a schematic diagram of the boiler deconditioning test rig (as illustrated in Reference 2). The boiling test section is a single tube of the Model 7 boiler (19 feet long, 0.335-inch ID) but without the plug insert. The reason for not including the plug is to allow amplification of the deconditioning effect and to provide better X-ray observation. A spring type swirl device extends through the entire length of the boiler.

FIGURE 1. SCHEMATIC DIAGRAM OF THE BOILER DECONDITIONING TEST RIG

IV. THE MECHANICS OF THE DECONDITIONED STATE

The most important single observation made during the Phase I tests was that the low heat transfer rates associated with deconditioning can be attributed to a liquid slug type flow pattern. The agglomerated state of the liquid leaves relatively large areas of the heating surface in contact with dry vapor. To support this observation made with the X-ray image amplifier, a detailed mathematical analysis of the boiling heat transfer mechanics as defined by the liquid slug flow pattern was made. Appendix A presents the mathematical details. The object of the analysis was to determine the important variables that can be manipulated by the boiler designer, and to point out the methods available for controlling boiler deconditioning. The causes of deconditioning, or the reasons why extensive slug flow is formed under certain conditions and the reasons why it may not be formed under others are covered under the heading "Causes of Deconditioning".

SNAP 2 boilers are designed assuming that small non-wetting mercury drops are formed having a diameter equal to that of the swirl wire (0.041 inch). Reference 1 gives the basis for this assumption. Procedures are also available for designing a boiler when complete wetting is obtained. The analysis presented in Appendix B covers this method. To be conservative, a non-wetting state is usually assumed for design, and a computed NaK temperature profile, as shown in Figure 2, is obtained. When complete wetting occurs or when the mercury side heat transfer resistance is negligible, a NaK profile as shown in Figure 3 or Figure B-3 is measured. When a mercury film is formed on the tube wall, the major resistance to heat transfer becomes the tube wall and the NaK. This gives a

FIGURE 2. COMPUTED NON-WETTING NAK TEMPERATURE PROFILE
ACCORDING TO PROCEDURES ESTABLISHED IN
REFERENCE 1
NAA-SR-6322

FIGURE 3. MEASURED NAK TEMPERATURE PROFILE EXHIBITING
 NEGLIGIBLE MERCURY SIDE HEAT TRANSFER RESISTANCE

NAA-SR-6322

NaK temperature gradient of approximately 30 to 50° F/ft for the Model 7 or Model 5C boiler. For flowing qualities greater than about 85 percent, dry wall boiling occurs and the major resistance is due to eddy diffusion of mercury spray to the heated surface (see Appendix B).

Figure 4 shows a typical NaK profile measured during the Phase I deconditioning tests and depicts a deconditioned heat transfer state for approximately the first 12 feet of boiling length measured from the boiler inlet. Beyond the 12-foot point, the NaK temperature gradient becomes very steep with a "wetting type" gradient of approximately 50° F/ft.

An important feature of the NaK temperature profile of Figure 4 is the existence of a transition point between a deconditioned boiling region and a high heat flux (conditioned) region. The transition is attributed to the "break-up" of the liquid slugs or "caterpillars" by the vapor velocity generated during slug flow boiling. The transition between slug or "caterpillar" flow and spray flow was first studied in the adiabatic mercury-nitrogen flow tests, and the flow regime map, shown in Figure 5, was constructed from these observations. A critical value of

$$\rho_v u_v^2 = 142 \frac{\text{lb (force)}}{\text{ft sec}^2}$$

is estimated to define the point of transition. If wetting beyond the critical point does not occur, the NaK profile temperature gradient may decrease to a point where moisture is ejected from the boiler, thus raising the NaK exit temperature at the boiler inlet. This situation is shown in Figure 6 which is a measured NaK profile obtained during the single-tube deconditioning tests.

FIGURE 4. TYPICAL MEASURED NAK TEMPERATURE PROFILE FROM PHASE I DECONDITIONING TESTS

FIGURE 5. FLOW REGIMES FOR NON-WETTING MERCURY
FLOWING WITH NITROGEN IN GLASS TUBES

Comparison between the NaK temperature profile for slug flow, as computed from equation (A-13), is compared with the measured profile in Figure 7. A NaK heat loss gradient of 1.3 °F/ft (measured during heat loss calibration) is also taken into account when comparing the computed with the measured NaK temperature profile. The computed transitional length (L_c) to the termination of slug flow regime using $\rho_v u_v^2 = 142$ (mercury-nitrogen data) is:

$$L_c = 13 \text{ ft} \quad (\text{See Appendix A for computations.})$$

which compares favorably with a measured length of approximately 15 feet. (See Figure 7.)

Based on this cursory analysis of the test results, the procedure to be followed when designing a boiler, if the deconditioning effect is to be included, can now be formulated. The first several weeks of operating the single-tube boiler at SNAP 2 design conditions with time as the independent variable indicated a natural tendency toward the deconditioned state. As a result, this natural phenomenon should be taken into account in the boiler design.

The plug insert which is used in SNAP 2 boilers at the present time was an improvement but its design is inadequate to do the job properly. By examining Equation (A-16), it can be seen that the required slug flow boiling length can be computed by introducing the conditions for the termination of slug flow. This flow passage length can be greatly shortened with respect to the tube length by means of a plug insert with a low pitch spiraled passage.

(Pitch is defined as the axial distance between turns.) In effect, this procedure amplifies the NaK temperature gradient by "accordioning" the required slug flow passage length.

The term α in Equation (A-17), considers this accordioning. To support this method for increasing the heat flux of the slug flow regime, the data obtained by Aerojet, on their low pitch plug insert (Reference 4) was analyzed. Figure 8 is a schematic of their plug insert

TEST NO 6. 4-6		DATE 4-8-65		SINGLE TUBE BOILER PER RECORD	
DATA POINT OBSERVER		EH, GK			
DATA POINT NUMBER		7		1	
DATA POINT TIME		1230		2	
ELAPSED HOURS FROM START		3		3	
TOTAL ELAPSED HOURS AT DESIGN				4	
TEMPERATURE	NaK HEATER ELECTRICAL POWER INPUT	KW	22	5	
	T34 NaK HEATER OUTLET LINE T/C 3-10 SKIN	°F	—	6	
	T4 NaK FLOWMETER T/C 4-5 SKIN	°F	—	7	
	T1 MERCURY INLET T/C 5-1	°F	694	8	
	T2 HG INLET TUBE SHEET SHELL T/C 5-2	°F	958	9	
	T3 NaK OUTLET LINE T/C 5-3	°F	1098	10	
	T22 NaK INLET LINE T/C 5-2 ND 11	°F	1200	11	
	T21 HG OUTLET TUBE SHEET T/C 5-2ND10	°F	1126	12	
TEMPERATURE	T23 HG OUTLET LINE IMM. T/C 5-2ND 12	°F	929	14	
	T12 CONDENSER HG VAPOR LINE T/C 1-12 IMM	°F	371	15	
FLOW	F1 BOILER MERCURY WEIGHT FLOW - RATE	% PPM	77 50	16	
	F2 BOILER NaK WEIGHT FLOW - RATE	MV PPM	175 175	17	
PRESSURE	P8 BOILER MERCURY INLET	PSIA	125	18	
	P9 BOILER MERCURY VAPOR DISCHARGE	PSIA	114	19	
	P11 BOILER NaK OUTLET	PSIG	112	20	
	P12 BOILER NaK INLET	PSIG	114	21	
	(B) BOILER STATIC NaK ΔP	PSI		22	
	(C) BOILER NaK ΔP PRESSURE	PSI		23	
	BOILER NaK FLOW ΔP = (B) - (C)	PSI		24	
	NOZZLE HG FLOW CHART	PPM	11	25

FIGURE 6. TYPICAL MEASURED NAK TEMPERATURE PROFILE FROM PHASE I DECONDITIONING TESTS SHOWING HIGH EXIT NAK TEMPERATURE DUE TO LIQUID DISCHARGE FROM BOILER

TEST NO 6.4-5		DATE 4-6-65		SINGLE TUBE BOILER PER RECORD	
DATA POINT OBSERVER		E H GK			
DATA POINT NUMBER		7		1	
DATA POINT TIME		1300		2	
ELAPSED HOURS FROM START		3		3	
TOTAL ELAPSED HOURS AT DESIGN				4	
TEMPERATURE	NaK HEATER ELECTRICAL POWER INPUT	KW	26	5	
	T34 NaK HEATER OUTLET LINE T/C 3-10. SKIN	°F	-	6	
	T41 NaK FLOWMETER T/C 4-5. SKIN	°F	-	7	
	T1 MERCURY INLET. T/C 5-1	°F	692	8	
	T2 HG INLET TUBE SHEET SHELL T/C 5-2	°F	858	9	
	T3 NaK OUTLET LINE T/C 5-3	°F	970	10	
	T22 NaK INLET LINE T/C 5-2 ND 11	°F	1185	11	
	T21 HG OUTLET TUBE SHEET T/C 5-2 ND 10	°F	1128	12	
	T23 HG OUTLET LINE IMM. T/C 5-2 ND 12	°F	930	14	
	T12 CONDENSER HG VAPOR LINE T/C 1-12. IMM.	°F	460	15	
FLOW	F1 BOILER MERCURY WEIGHT FLOW - RATE	% PPM	7.7 5.0	16	
	F2 BOILER NaK WEIGHT FLOW - RATE	MV PPM	17.5 17.5	17	
PRESSURE	P8 BOILER MERCURY INLET	PSIA	140	18	
	P9 BOILER MERCURY VAPOR DISCHARGE	PSIA	110	19	
	P11 BOILER NaK OUTLET	PSIG	11.5	20	
	P12 BOILER NaK INLET	PSIG	11.8	21	
	(B) BOILER STATIC NaK ΔP	PSI		22	
	(C) BOILER NaK ΔP PRESSURE	PSI		23	
	BOILER NaK FLOW ΔP = (B) - (C)	PSI		24	
	NOZZLE HG FLOW CHART	PPM	30	25

FIGURE 7. COMPARISON OF NAK TEMPERATURES WITH COMPUTED CURVE FOR SLUG FLOW ADJUSTED FOR HEAT LOSS AS DETERMINED BY CALIBRATION Eq. A-13

which by computation gives an $\alpha = 9.42$. The NaK temperature gradient computed by means of Equation (A-17), with $\alpha = 9.42$ is also compared with SNAP 8 test data in Figure 8. The results compare well. The slug flow length is "accordioned" by a factor of 9.42 which is the value for α . It is recommended that a tight fitting plug insert be designed to prevent short circuiting of the liquid and to provide a heat transfer path for conducting heat from the tube wall to the entire periphery of the spiral passage. Figure 9 depicts a possible design for a plug insert. Aerojet test data, obtained after several months of testing of a single-tube boiler with a high pitch plug insert (low value of α), indicated a deconditioned boiling length of approximately 30 feet. (See Figure 10.) This long length is attributed to a high flow rate per passage according to Equations (A-13) or (A-17). The momentary sharp rise in the NaK temperature at the plug exit is assumed to be a result of the spray evaporation which is caused by some break-up in liquid slugs. However, the vapor velocity formed is not adequate to continue the break-up of the remaining liquid slugs or to prevent liquid agglomeration.

Figure 11 compares computed NaK slug flow temperature rise profiles as a means of showing the increase in NaK temperature gradient obtainable with a low pitch plug and the very low gradient resulting from high flow rate per tube (high value of m/N in Equation (A-13)).

Figure 12 shows another possible plug design for accommodating the slug flow regime. By using multiple passages, the slug flow length can be decreased as predicted by Equation (A-13), when the value of m/N is reduced. A heat conducting porous plug would be a limiting case in which the number of passages are a maximum and the plug length a minimum.

FIGURE 8. COMPARISON OF SLUG FLOW HEAT TRANSFER IN LOW PITCH PLUG WITH SNAP 8 TEST DATA

NAA-SR-6322

FIGURE 9. SUGGESTED SPIRAL PLUG DESIGN FOR THE SLUG FLOW BOILING REGIME

NAA-SR-6322

Flow Rate NaK, W_N - 11, 100 LB/HR
 Temp. NaK Boiler In, T_{Nbi} 1329°F
 Temp. NaK Boiler Out, T_{Nbo} 1142°F
 Flow Rate Liquid Hg, W_{Hf} 2810 LB/HR
 Temp. Hg Boiler In, T_{Hbi} 468°F
 Press. Hg Boiler In, P_{Hbi} 331 ± 4.5 PSIA

Temp. Hg Boiler Out, T_{Hbo} = 1299°F
 Press. Hg Boiler Out, P_{Hbo} = 2551 ± 10 PSIA
 Hg Boiler Differential Press. ΔP_{Hb} = 85 PSI
 Press. Hg Vapor Nozzle In, P_o = 253 PSIA

Plug Insert
 Wire Diameter = .135"
 Pitch = 3", 4", 5"
 Length = 10 ft
 Data Point: 44 Time: 2230

FIGURE 10. SNAP 8 SINGLE TUBE BOILER NAK TEMPERATURE PROFILE

FIGURE 11. COMPUTED NAK TEMPERATURE PROFILES FOR THE SLUG FLOW REGIME

FIGURE 12. SUGGESTED MULTICHANNEL PLUG DESIGN FOR THE SLUG FLOW BOILING REGIME

Further verification of the critical conditions for the limit of the slug flow regime is desirable since this determines the length of the plug. The critical conditions of the unplugged tube should be introduced into Equation (A-13) or (A-17), if agglomeration into slugs is to be avoided when discharging the liquid-vapor mixture from the plug insert. Figure 13 shows helical flow boiling mercury data obtained by Geoscience Limited (Reference 5) compared with the slug flow curve computed by Equation (A-19). The termination of the slug flow regime using $\rho_v u_v^2 = 142 \text{ lb/ft}^2\text{-sec}^2$ is calculated to occur at a quality of 18.8 percent, since:

$$X_c = \frac{(142 \rho_v)^{1/2}}{G_o} = \frac{142 \times 0.83}{57.9} = 0.188 \quad \dots 1$$

which is not far from the minimum point of the curve (Figure 13) at about $X_c = 0.13$.

Since the quantity $\rho_v u_v^2$ probably originates from a Weber number type parameter, other effects, such as surface tension and tube dimensions, should be included if the critical quality X_c is to be accurately determined. It should be noted that the critical conditions computed from the many NaK temperature profiles measured in Phase I of the deconditioning tests are approximated by the quantity $\rho_v u_v^2 = 142 \text{ lb/ft}^2\text{-sec}^2$.

The measured slug flow heat transfer coefficients in Figure 13 are somewhat greater than predicted by the slug flow equation, Equation (A-19). This is plausible when one realizes that slug flow heat transfer is a limiting case for the drop wise boiling heat transfer equations presented in Reference 1. For contact boiling of mercury drops, Equation B-23 of Reference 1 is presented as follows:

$$\frac{h D}{K_v} = \frac{3}{8} Nu_\delta \frac{K_f}{K_v} \left(\frac{D}{\delta_o} \right)^2 \frac{1}{\epsilon} \frac{(1-x)^{1/3}}{x} \frac{\rho_v}{\rho_f} \quad \dots 2$$

NAA-SR-6322

FIGURE 13. HELICAL FLOW BOILING MERCURY HEAT TRANSFER COEFFICIENTS IN AN ELECTRICALLY HEATED TUBE
 $D = 0.190$ INCH ID, 41 LB Hg PER HR; $p = 56.6$ PSIA (INLET)
 DATA FROM REFERENCE 5, COMPARISON OF DATA WITH CURVE FOR SLUG FLOW BOILING

Normally it is suggested that the initial drop size δ_o be made equal to the wire size (0.041 inch). However, if for some reason drops approaching the tube diameter are formed ($\delta_o \rightarrow D$), then the ratio of the heat transfer coefficients becomes:

$$\frac{h'}{h} = \left(\frac{\delta_o}{D} \right)^2 \quad . . . 3$$

where h' is for $\delta_o \rightarrow D$ and h is for δ_o equal to the wire size. For typical SNAP 2 values,

$$\frac{h'}{h} = \left(\frac{0.041}{0.335} \right)^2 = 1.5 \times 10^{-2}$$

This is sufficient for decreasing the NaK temperature gradient from a maximum value obtained for wetting of

$$\frac{dT}{dL} = 50 \text{ F/ft}$$

to

$$\frac{dT}{dL} = 50 \times 1.5 \times 10^{-2} = 0.75 \text{ F/ft}$$

which is a sufficient reduction in heat flux to give a deconditioning effect.

Phase I of the deconditioning tests included a test series in which boiling was initiated by electrical heat in a two-foot length of 0.051-inch ID tube. The liquid-vapor mixture was then ejected into a very short plug with a swirl wire extending about 6 inches into the boiler. The purpose of this short plug was to induce vortex flow without providing a long passage length whereby other factors might be introduced. From heat balance calculations of data point 7 (5/11/65), it was computed that a flowing quality of 1.93 percent was obtained with

a superficial vapor velocity of 150 ft/sec. The density of the vapor was estimated to be 0.6 lb/ft³; thus giving a value for,

$$\rho_v u_v^2 = 0.6 \times 150^2 = 13.5 \times 10^3$$

which should be sufficient for atomizing large liquid drops. It was observed on the X-ray image amplifier that a high velocity vortex flow was formed at the plug discharge.

However, the mixture appeared to have relatively large swirling drops which were probably formed by liquid agglomeration of the low quality mixture. The NaK temperature gradient corresponding to this flow pattern was small enough to consider this region deconditioned.

Based on this observation it can be postulated that $\rho_v u_v^2 = 142$ is a necessary condition, but not sufficient for defining the termination of slug type flow. Reference 6 gives conditions for avoiding liquid spray recombination. No increase in average drop size was observed if the distance between drops was greater than 14 drop diameters. In terms of volumetric flow rate this becomes:

$$\frac{L''}{\delta} = \left(\frac{\pi}{6} \frac{\dot{V}_v}{\dot{V}_f} \right)^{1/3} \quad \dots 4$$

where $\dot{V}_v$ and $\dot{V}_f$ are the vapor and liquid volumetric flow rates and L'' is the distance between drops of diameter δ . In terms of flowing quality Equation 4 becomes:

$$\frac{L''}{\delta} = \left(\frac{\pi}{6} \frac{x}{1-x} \frac{\rho_f}{\rho_v} \right)^{1/3} \quad \dots 5$$

Equation 5 gives qualities near 90 percent for SNAP 2 conditions, if $L''/\delta = 14$.

However, $L''/\delta = 3$ to 4 gives values which correspond to measured critical qualities at the slug flow transition in mercury boilers. Further work is required in this area for a more exact definition. However, it should be noted with caution that the design of an

inlet spray nozzle for the purpose of avoiding slug flow may require some experimental development. To avoid large liquid slugs, a high velocity is required for atomization and a low liquid drop population density for avoiding liquid recombination. Later in this report, the section on causes of deconditioning points out that pulsations resulting from nucleation and liquid superheat can be a factor contributing to liquid agglomeration and the formation of slug flow. Actual critical values for slug flow termination for SNAP 2 design conditions can be determined from available NaK temperature profiles, and require no extrapolation into unknown areas. These values are directly available to a SNAP 2 boiler designer for determining the length of either a spiral flow or a multiple channel flow plug insert.

Figure 7 compares typical NaK temperature measurements with the curve computed for slug flow boiling, Equation (A-13), and adjusted for insulation heat loss. The termination of the slug flow regime as computed from $W_c = 142 \text{ lb/ft-sec}^2$, Equation (A-16), based on N_2 -Hg adiabatic data is also indicated. Test data from the Model 7 boiler gave a similar slug flow length (L_c) when deconditioned and when operating at conditions similar to those given in Figure 7.

It is recommended that the objective of Phase II of the deconditioning tests be the investigation of the means for controlling the slug flow boiling regime as described in this section. In developing plug inserts for increasing the heat transfer rate in the slug flow regime (deconditioned state), the level of the mixture velocity should be established by erosion considerations. If conditioning should occur within the plug, mixture velocities would approach a maximum and this value should be within tolerable limits.

NAA-SR-6322

V. CAUSES OF DECONDITIONING

Assuming that deconditioning is a natural state, it should be considered in boiler design.

The designer of mercury boilers has two alternatives for accommodation of the deconditioning phenomenon: (1) control and (2) prevention. The control of deconditioning requires a knowledge of its mechanics which have been discussed in Section IV. The prevention of deconditioning is dependent on the complete definition of the causes of deconditioning and may prove to be the more difficult task since its causes are still subject to conjecture.

In this section an attempt is made to define the causes of deconditioning by analyzing the deconditioning symptoms as determined by various measurements. To aid the definition of the causes, certain factors suspected of influencing deconditioning were also investigated.

These factors were:

- 1) Heated surface at a temperature of 1200° F exposed to argon gas flow prior to boiling
- 2) Heated surface at a temperature of 1200° F exposed to laboratory air flow prior to boiling
- 3) Argon injection during boiling
- 4) Manipulation of boiler pressure and flow levels
- 5) Time if considered an independent variable
- 6) High velocity mixture (vapor and liquid) injection into NaK heated boiler section

Based on the Phase I deconditioning test results, the following symptoms were found to be associated with a deconditioned boiler:

- 1) Pressure pulses of relatively high frequency and amplitude were measured.
(See Figure 14 for typical pressure trace.)
- 2) By X-ray observation, liquid slugs were noted; and, by count, periods ranging from 1.5 to 3 seconds were measured. The liquid slugs appeared to be made of a number of caterpillar type globules.
- 3) By means of a vibration pick-up at the boiler inlet, boiling sounds were detected. These sounds were similar in nature to those detected by Oak Ridge National Laboratories (ORNL) during bulk boiling of water and potassium. These sounds resembled sharp hammer blows or pings, like small steel balls falling on a steel plate. ORNL identified these sounds as being implosions resulting from liquid superheat "explosions". These sounds are similar in tone quality to those produced in pump cavitation.
- 4) Prior to deconditioning, low frequency large amplitude pressure pulses occurred (approximately 25 seconds/cycle), which were identified as being caused by fluctuations between a conditioned and a deconditioned state. Large amounts of vapor are generated near the boiler inlet, when conditioned boiling occurs, but near the boiler outlet, when deconditioned boiling occurs. This varies the pressure drop and causes the pressure oscillations. (See Figure 14 for a typical pressure trace.) Boiling sound characteristics substantiated these observations since fluctuations between "pinging" and "hissing" sounds were noted to correspond to the pressure pulses.

NAA-SR-6322

NAA-SR-6322

FIGURE 14. EFFECT OF PRESSURE LEVEL AND FLOW RATE ON DECONDITIONING (Sheet 1 of 3)

FIGURE 14. EFFECT OF PRESSURE LEVEL AND FLOW RATE ON DECONDITIONING (Sheet 2 of 3)

NAA-SR-6322

FIGURE 14. EFFECT OF PRESSURE LEVEL AND FLOW RATE ON DECONDITIONING (Sheet 3 of 3)

NAA-SR-6322

- 5) A deconditioned boiler has a much lower pressure drop than a conditioned boiler. This is attributed to the spatial distribution of the vapor generation rate along the tube.

The results of introducing variables suspected of influencing deconditioning were:

- 1) Time itself will decondition an initially bright annealed Haynes-25 boiler tube. (Some prototype MRP boilers which have been tested to date have not substantiated this statement.)
- 2) Exposure of the boiler surface at 1200° F to argon flow for approximately 1.5 hours prior to boiling has no appreciable effect in conditioning a deconditioned boiler.
- 3) Exposure of the boiler surface at 1200° F to laboratory air flow from 40 minutes to 6 hours prior to boiling conditioned a deconditioned boiler for several days. (Exposure of the 5C boiler to air does not appear to cause the same effect.)
- 4) Argon injection into the mercury passing through a 0.051-inch ID electrically heated tube immediately conditioned a deconditioned boiler. When the argon flow was stopped, it took about one hour for deconditioning symptoms to re-appear.
- 5) Argon injection into mercury passing through a 1/4-inch electrically heated tube slowly conditioned a deconditioned boiler. The degree of conditioning depended on the amount of argon flow.
- 6) A boiler that was in the process of deconditioning could be instantaneously conditioned by decreasing the boiler pressure level. This effect was reproducible for several hours after which the boiler remained deconditioned when the pressure level was decreased. Then by decreasing both the NaK

and mercury flow by one-half, instantaneous conditioning was obtained.

By returning the NaK and mercury to full flow, the boiler immediately deconditioned. The boiler pressure traces are shown in Figure 14. Time appeared to be a factor which eventually induced irreversibility.

The characteristics of a boiler in a conditioned state were observed to be as follows:

- 1) Pressure levels were steady, and there was no visual appearance of slugging.
- 2) Appearance of large quantities of fast moving drops gave the impression of rain.
- 3) Splotches of liquid mercury occurred near the boiler inlet and appeared to stick to the tube wall giving the impression of partial wetting.
- 4) Boiler sounds resembled hissing, and no pinging was detected.

The possible cause of deconditioning can be hypothesized from the results listed above. Wetting, surface tension, nucleation characteristic of the surface, and ability of liquid metal to sustain liquid superheat, all blend together in subtle relationships which affect the two-phase hydrodynamics characteristic of the deconditioned state. The prevention of deconditioning appears difficult since a development program which in effect would control these factors would be required to prevent deconditioning. Long-term reliability of a SNAP 2 boiler may be affected if the causes of deconditioning are not completely understood. The control of deconditioning, not the prevention of it, appears to be a possible alternative.

Other tests for detecting causes of boiler deconditioning have been or are being conducted on the SNAP 2 program. Reference 3 reports mercury drop evaporation time on bright annealed and oxidized surfaces. A.I. is conducting pool boiling tests on various surfaces. The nature of the type of boiling can be inferred by means of the heat flux-temperature potential curve. In summary, the pool boiling tests have shown that clean or oxidized Haynes-25 surfaces give nucleate boiling curves characteristic of a wetted surface. Wetting appears to be a natural tendency. However, oil contamination of a Haynes 25 surface adversely affected the wetting and pool boiling heat transfer. The drop evaporation tests of Reference 3 have shown that:

- 1) For drop sizes equal to 0.0826 inch in diameter the evaporation time is between 1 and 1.5 minutes when the plate temperature difference with respect to saturation temperature is approximately 100°F. The evaporation time remains the same regardless of the type of surface since for at this temperature difference the drop is supported on a film of vapor.
- 2) Boiler droplet resident times, of about one minute would require exceptionally long tubes if the acceleration fields are near 1 g. This indicates the difficulty encountered in evaporating large liquid drops, globules, or slugs in high performance, short tube boilers. Small, unagglomerating liquid droplets must be formed if the heat flux is to be maintained at a high level.
- 3) For plate temperature differences less than 100°F, the droplet evaporation time depends on the surface treatment. The contact angle or degree of wetting thus becomes important in the heat transfer since the liquid is now

supported directly by the heating surface and not by a vapor film. In this range of plate temperature difference, the evaporation time on the oxidized surface was less than that of the unoxidized surface indicating a greater degree of wetting. The oxide layer was obtained by exposing the high temperature surface to room air.

To obtain high heat fluxes it is apparent that wetting must occur and/or small drops must be formed. A thin liquid film resulting from complete wetting (spreading conditions) with no liquid entrainment gives the maximum possible heat flux and NaK temperature gradient. The controlling thermal resistance is then in the NaK and tube wall. In any event, agglomerated liquid reduces the area of the heat flow path necessary for evaporation. The factors determining the formation of relatively large chunks of liquid can therefore be defined as possible causes of deconditioning. Reference 7 attributes the unstable liquid-metal boiling phenomenon in natural-and-forced-circulation loops as being caused by high superheat and high thermal conductivity which are peculiar to liquid metals. The oscillatory nature of the flow can cause liquid agglomeration and superheating of the incoming liquid can result in vapor formation behind the liquid-vapor interface, thus ejecting large chunks of liquid into the two-phase boiling region. In this manner the slug flow pattern is connected to liquid superheating. In addition, liquid superheating tendencies are related to wetting and the "washing out" of nucleation sites. Reference 8 gives visual confirmation of this concept in a boiling water loop. Reference 8 also notes that there is an interaction of the heat transfer and the hydrodynamics in two-phase flow with addition of heat. Differences were also noted in the flow regime resulting from an active heating surface (large number of active nucleation sites) and a passive heating surface (few active sites). The observation demonstrates the importance of surface condition in establishing the two-phase pattern.

According to Reference 8, if a tube surface lacks a large number of effective nucleation sites, slug flow results from the rapid formation of a Taylor bubble resembling the cavitation phenomenon.

Reference 9 is another important reference relating to this phenomenon. By heating sub-cooled water or alcohol, in a pool, on a horizontal heated surface a significant overshoot in liquid temperature, above the saturation level was noted. This overshoot was followed by a significant undershoot. After these transient observations, normal nucleate boiling of saturated liquid was established. The overshoot resulted in the formation of a large vapor slug which developed explosively as in some cavitation phenomenon. It was concluded that the development of the vapor slug is a necessary transitional step to the onset of nucleate boiling. In a forced convection boiler in which sub-cooled liquid is introduced, this phenomenon can become repetitive and result in a continuous formation of liquid slugs from the "explosions". It is possible that the implosions or cavitation sounds one hears in boilers are a result of the cooling effect on the vapor of the temperature "undershoot" following the "overshoot". Reference 9 also notes that surface condition is also a factor.

In light of the foregoing discussion interpretation of some of the Phase I test observations can now be made. The conditioning effect of the "hot air wash" prior to boiling may be due to the combined effect of wetting and the introduction of active gas nucleation sites which temporarily conditioned the boiler for several days until the gaseous sites or possibly the wetting effect was eliminated. The drop evaporation test (Reference 3) also showed a wetting tendency for an air oxidized plate. The "hot argon wash" apparently did not produce this combined effect and resulted in no appreciable conditioning.

NAA-SR-6322

The injection of a high velocity mixture of vapor and liquid (150 ft/sec; 1.93% flowing quality) into the NaK heated section of the boiler did not appreciably increase the heat flux at the boiler inlet. This is attributed to liquid agglomeration as interpreted by X-ray observation. This agglomeration could have been caused by the high droplet density of the low quality mixture, or possibly by instabilities resulting from poor nucleation in the electrically heated injection tube. The flow mixture did not exhibit the severe slugging normally observed in deconditioning. The mixture swirled and followed the pitch of the swirl wire. However, the injection of argon into the same electrically heated tube produced startling results in conditioning. The superficial argon velocity had a minimum value of 15 feet per second, and the superficial liquid velocity was 10 feet per second, giving a value of $\rho_v u_v^2$ in excess of $142 \frac{\text{lb (force)}}{\text{ft}^2 \text{ sec}^2}$. The resulting mixture velocity would be, of course greater than either of these quantities. If the mixture existed as a fog, the velocity is estimated to be 21 feet per second. Conditioning can be attributed to three possible reasons: (1) atomization, (2) no liquid superheat resulting from the presence of the gas, and (3) the 7 ppm of oxygen in the welding grade argon which might have affected the wetting. The purity of the argon was given by the supplier as 99.996%. It should be noted in contrast that the argon "hot wash" prior to boiling had no appreciable influence on conditioning; therefore, its effect on wetting could be negligible. Good nucleation (elimination of superheat instabilities and liquid agglomeration) is the remaining major difference between the high velocity vapor-liquid mixture and the high velocity vapor-liquid-argon mixture. Argon injection into liquid flowing in the large nucleation tube conditioned the boiler in gradual amounts depending on the argon flow. This indicates that the hydrodynamics at the boiler inlet has some influence on conditioning. It should be cautioned that this reasoning in causality is highly speculative.

NAA-SR-6322

In order to support the hypothesis that the liquid slugging in a deconditioned boiler can be attributed to liquid superheat at the boiler inlet an analysis of the phenomenon can be made. The slugging frequency computed by this means can then be compared with the measured quantity. Consider a given quantity of flowing liquid at the boiler inlet being heated by a temperature reservoir. A heat balance results in,

$$dQ = \frac{\pi D^2}{4} dL \rho_f C_f \frac{dT}{d\theta} \quad . . . 6$$

and in,

$$dQ = h \pi D dL \Delta T \quad . . . 7$$

Combining results in;

$$d\theta = \frac{D}{4} \frac{\rho_f C_f}{h \Delta T} dT \quad . . . 8$$

And also introduce the Nusselt number,

$$\frac{hD}{K_f} \equiv Nu \quad . . . 9$$

$$d\theta = \frac{D^2}{4} \frac{\rho_f C_f dT}{\Delta T Nu K_f} \quad . . . 10$$

During the superheating process consider the fluid temperature to rise from the saturation temperature to the reservoir temperature and the average temperature difference between the fluid and the reservoir to be one-half of the total rise. As a result, the temperatures can be eliminated from the equation; since

$$\theta = \frac{D^2 \rho_f C_f \Delta T_{SH}}{4 \frac{\Delta T_{SH}}{2} Nu K_f} = \frac{D^2 \rho_f C_f}{2 Nu K_f} \quad . . . 11$$

By substituting values from test conditions we have

$$\theta = \left(\frac{0.335}{12} \right)^2 \frac{\times 800 \times 0.034 \times 3600}{2 \times Nu \times 8} = \frac{4.75}{Nu} \text{ seconds per pulse}$$

For liquid metal flowing at low velocity (low Peclet number) values for Nu range from 5 to 1.5 based on theory and experiment. Therefore, the period for liquid slug formation will have values of,

$$\theta = \frac{4.75}{1.5} = 3.16 \text{ seconds}$$

$$\theta = \frac{4.75}{5} = 0.95 \text{ seconds}$$

These values compare favorably with the periods measured by visually counting liquid slugs which gave periods of 1.5 to 3 seconds.

The concept that the given quantity of liquid under consideration is being superheated and that eventually this heat is converted by a nucleation site into a relatively large volume of saturated vapor suggests that a fixed point at the boiler inlet is used for "triggering" this conversion. If flow rates and pressure levels are altered, nucleation will take place at a different point which may have different nucleating characteristics than the original one. This explanation can account for the instantaneous conditioning and deconditioning observed when flow rates and pressure levels were varied during the Phase I deconditioning tests. Each new active nucleation site eventually became deactivated by wetting or "washing out" and resulted in time wise deconditioning regardless of which point within the boiler was utilized for the inception of boiling. The design of stable nucleation sites which could

not be washed out by the wetting liquid was experimentally investigated by ORNL (Reference 7). To incorporate the findings of this investigation into a SNAP 2 boiler design would require extensive development work.

Further work in data reduction of Phase I test results is recommended. A comprehensive correlation of the test data will yield more precise values needed for the design of insert devices for the control of deconditioning.

DEFINITION OF SYMBOLS

C_f	liquid heat capacity
D	tube diameter
G_o	mass velocity (both phases)
h	heat transfer coefficient
K_v	vapor thermal conductivity
K_f	liquid thermal conductivity
ℓ''	distance between drops
L	tube length from boiler inlet to point in question
Nu_δ	drop Nusselt number
Nu	Nusselt number
Q	heat transfer rate
T_N	NaK temperature
ΔT	temperature difference
ΔT_{SH}	liquid superheat temperature difference
U_v	vapor velocity
$\dot{V}_v$	vapor volumetric flow rate
$\dot{V}_f$	liquid volumetric flow rate
$W \equiv$	$\rho_v U_v^2$
X	flowing quality
X_c	flowing quality at termination of slug flow
δ	drop diameter
δ_o	initial drop diameter

NAA-SR-6322

DEFINITION OF SYMBOLS (Continued)

ϵ ratio of liquid to vapor velocity

θ time

ρ_v vapor density

ρ_f liquid density

LIST OF REFERENCES

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APPENDIX A

SLUG FLOW MERCURY BOILING HEAT TRANSFER

The purpose of the following analysis is to develop equations for computing slug flow heat transfer coefficients, slug flow NaK temperature profiles, and length of tubing required for termination of the slug flow regime. An equation for designing a plug insert for the slug flow regime is also formulated.

The equations developed herein can be directly applied to SNAP 2 mercury boiler design for the control of boiler deconditioning. These equations form the basis for Phase II of the boiler deconditioning tests. Sample calculations are submitted for verification of the equations with experimental evidence.

The heat transfer through a differential tube length is:

$$U \pi D N (T_N - T_{HG}) dL = m_N C_N dT_N \quad \dots A-1$$

NAA-SR-6322

and by heat balance:

$$m h_{vf} dX = m_N C_N dT_N$$

Integrating,

$$m h_{vf} X = m_N C_N T_N + C$$

Where C is a constant of integration,

$$\text{when } X = 0 \quad T_N = T_{NE} \quad \therefore C = -m_N C_N T_{NE}$$

$$\therefore X = \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) \quad \dots A-2$$

Or,

$$\text{when } X = 1 \quad T_N = T_{Ni} \quad \therefore C = m h_{vf} - m_N C_N T_{Ni}$$

$$\therefore m h_{vf} (X - 1) = m_N C_N (T_N - T_{Ni})$$

Or,

$$m h_{vf} (1 - X) = m_N C_N (T_{Ni} - T_N)$$

$$1 - X = \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_N)$$

$$1 - \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_N) = X \quad \dots A-2a$$

The following also holds true:

$$T_{Ni} - T_{NE} = \frac{m h_{vf} X_E}{m_N C_N} \quad \dots A-2b$$

if $X = X_E$ at boiler discharge

Heat transfer in slug flow regime:

NAA-SR-6322

The mean heat transfer coefficient over the length ℓ' is

$$h = \frac{h_{\ell}}{\ell'} + h_v \frac{(\ell' - \ell)}{\ell'} \quad \dots A-3$$

where h_{ℓ} and h_v are the heat transfer coefficients for the liquid and vapor slugs respectively.

Assume that the vapor slug acts as a "heat dam" containing superheated vapor; that is, neglect the heat transfer to the vapor slug. Therefore,

$$h = h_{\ell} \frac{\ell}{\ell'} \quad \dots A-4$$

Since there is no slip between the vapor and liquid phase for this flow pattern, the static quality equals the flowing quality. Therefore,

$$X = \frac{M_v}{M_T} = \frac{\rho_v (\ell' - \ell)}{\rho_v (\ell' - \ell) + \rho_f \ell} = \frac{\rho_v / \rho_f (\ell' / \ell - 1)}{\rho_v / \rho_f (\ell' / \ell - 1) + 1} \quad \dots A-5$$

Substitute Equation A-4 into Equation A-5 eliminating ℓ' / ℓ .

$$X = \frac{\rho_v / \rho_f \left(\frac{h_{\ell}}{h} - 1 \right)}{\rho_v / \rho_f \left(\frac{h_{\ell}}{h} - 1 \right) + 1}$$

Solving for h gives,

$$h = \frac{h_{\ell} (1 - X)}{1 - X (1 - \rho_f / \rho_v)} \quad \dots A-6$$

Combining Equation A-6 and Equation A-2 eliminating X ,

$$h = \frac{h_{\ell} \left[1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) \right]}{1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) (1 - \rho_f / \rho_v)} \quad \dots A-7$$

NAA-SR-6322

Assuming the plug flow regime offers the major resistance to the heat transfer, then

$$h \approx U$$

and combining Equation A-7 and Equation A-1 we obtain,

$$\frac{h_x \left[1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) \right] dL}{\left[1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) (1 - \rho_f / \rho_v) \right]} = \frac{m_N C_N dT_N}{\pi D N (T_N - T_{HG})}$$

$$\frac{h_x \pi D N dL}{m_N C_N} = \frac{\left[1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) (1 - \rho_f / \rho_v) \right] dT_N}{\left[1 - \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE}) \right] (T_N - T_{HG})} \quad \dots A-8$$

Equation (A-8) contains the condition that $X = 0$ at $T_N = T_{NE}$

An alternate form of Equation (A-8) containing X_E and T_{Ni} in place of T_{NE} at $X = 0$, can be obtained by use of the heat balance Equation (A-2b);

$$T_{NE} = T_{Ni} - \frac{m h_{vf} X_E}{m_N C_N}$$

Therefore Equation A-8 becomes,

$$\frac{h_x \pi D N dL}{m_N C_N} = \frac{\left[1 - \frac{m_N C_N}{m h_{vf}} \left(T_N - T_{Ni} + \frac{m h_{vf} X_E}{m_N C_N} \right) (1 - \rho_f / \rho_v) \right] dT_N}{\left[1 - \frac{m_N C_N}{m h_{vf}} \left(T_N - T_{Ni} + \frac{m h_{vf} X_E}{m_N C_N} \right) \right] (T_N - T_{HG})}$$

$$\frac{h_x \pi D N dL}{m_N C_N} = \frac{\left[1 - (1 - \rho_f / \rho_v) \frac{m_N C_N}{m h_{vf}} (T_N - T_{Ni}) - X_E (1 - \rho_f / \rho_v) \right] dT_N}{\left[1 + \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_N) - X_E \right] (T_N - T_{HG})} \quad \dots A-9$$

If the following conditions are valid Equation (A-9) can be simplified and easily integrated:

$$X_E = 1 \quad \text{and} \quad \rho_f / \rho_v \gg 1$$

then by rearranging terms and integrating we obtain the following:

$$\frac{h_L \pi D N dL}{m_N C_N} = \frac{\left[\frac{\rho_f}{\rho_v} \frac{m_N C_N}{m h_{vf}} (T_N - T_{Ni}) + \rho_f / \rho_v \right] dT_N}{\frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_N) (T_N - T_{HG})}$$

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} dL = \frac{\left[\frac{m_N C_N}{m h_{vf}} (T_N - T_{Ni}) + 1 \right] dT_N}{(T_{Ni} - T_N) (T_N - T_{HG})}$$

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} dL = \frac{dT_N}{(T_{Ni} - T_N) (T_N - T_{HG})} - \frac{m_N C_N}{m h_{vf}} \frac{dT_N}{(T_N - T_{HG})}$$

integrating

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} \int_0^L dL = \int_{T_{NE}}^{T_N} \frac{dT_N}{(T_{Ni} - T_N) (-T_{HG} + T_N)} - \frac{m_N C_N}{m h_{vf}} \int_{T_{NE}}^{T_N} \frac{dT_N}{(-T_{HG} + T_N)}$$

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} L = \frac{1}{T_{Ni} \times 1 - (-T_{HG})(-1)} \ln \left[\frac{(-T_{HG} + T_N)}{(T_{Ni} - T_N)} \right]_{T_{NE}}^{T_N} - \frac{m_N C_N}{m h_{vf}} \ln (-T_{HG} + T_N) \Big|_{T_{NE}}^{T_N}$$

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} L = \frac{1}{(T_{Ni} - T_{HG})} \left[\ln \frac{(T_N - T_{HG})}{(T_{Ni} - T_N)} - \ln \frac{(T_{NE} - T_{HG})}{(T_{Ni} - T_{NE})} \right] \\ - \frac{m_N C_N}{m h_{vf}} \left[\ln (T_N - T_{HG}) - \ln (T_{NE} - T_{HG}) \right]$$

$$\frac{h_L \pi D N \rho_v}{m h_{vf} \rho_f} L = \frac{1}{T_{Ni} - T_{HG}} \ln \frac{(T_N - T_{HG})(T_{Ni} - T_{NE})}{(T_{Ni} - T_N)(T_{NE} - T_{HG})} \\ - \frac{m_N C_N}{m h_{vf}} \ln \frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})} \quad \dots A-10$$

The liquid slug heat transfer coefficient (h_L) can be estimated by means of the classical viscous flow equation for uniform wall heat addition;

$$h_L = \frac{48}{11} \frac{K_L}{D} \quad \dots A-11$$

Note: This equation was also used by Poppendiek in Reference 5

Substituting Equation (A-11) into Equation (A-10)

$$\frac{48 K_L \pi N \rho_v}{11 m h_{vf} \rho_f} L = \frac{1}{T_{Ni} - T_{HG}} \ln \frac{(T_N - T_{HG})(T_{Ni} - T_{NE})}{(T_{Ni} - T_N)(T_{NE} - T_{HG})} \\ - \frac{m_N C_N}{m h_{vf}} \ln \frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})} \quad \dots A-12$$

or since $\frac{m h_{vf}}{m_N C_N} = T_{Ni} - T_{NE}$ and if $X_E = 1$, we obtain an equation by which the NaK temperature profile can be determined,

$$\frac{48 \times K_g \pi N \rho_v L}{11 m h_{vf} \rho_f} = \frac{1}{(T_{Ni} - T_{HG})} \ln \frac{(T_N - T_{HG})}{(T_{Ni} - T_N)} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})}$$

$$- \frac{1}{T_{Ni} - T_{NE}} \ln \frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})} \quad \dots \text{A-13}$$

another form can be obtained as follows:

$$\frac{48 K_g \pi N \rho_v (T_{Ni} - T_{HG}) L}{11 m h_{vf} \rho_f} = \ln \frac{(T_N - T_{HG}) (T_{Ni} - T_{NE})}{(T_{Ni} - T_N) (T_{NE} - T_{HG})}$$

$$- \frac{(T_{Ni} - T_{HG})}{T_{Ni} - T_{NE}} \ln \frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})}$$

$$\frac{48 K_g \pi N \rho_v (T_{Ni} - T_{HG})}{11 m h_{vf} \rho_f} L = \ln \frac{(T_N - T_{HG}) (T_{Ni} - T_{NE})}{(T_{Ni} - T_N) (T_{NE} - T_{HG})} \left(\frac{T_N - T_{HG}}{T_{NE} - T_{HG}} \right)^{-\frac{(T_{Ni} - T_{HG})}{(T_{Ni} - T_{NE})}}$$

$$\frac{48 K_g \pi N \rho_v (T_{Ni} - T_{HG}) L}{11 m h_{vf} \rho_f} = \ln \frac{(T_N - T_{HG}) (T_{Ni} - T_{NE})}{(T_{Ni} - T_N) (T_{NE} - T_{HG})} \left[\frac{(T_{NE} - T_{HG})}{(T_N - T_{HG})} \right]^{\frac{(T_{Ni} - T_{HG})}{(T_{Ni} - T_{NE})}}$$

$$\frac{48 K_g \pi N \rho_v (T_{Ni} - T_{HG}) L}{11 m h_{vf} \rho_f} = \ln \left[\frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})} \right] \left(1 - \frac{T_{Ni} - T_{HG}}{T_{Ni} - T_{HG}} \right) \frac{(T_{Ni} - T_{NE})}{(T_{Ni} - T_N)}$$

Note: See sample calculations, included in this appendix, for determining NaK temperature profile.

Assume transition between slug flow and vortex-drop wise flow is given by,

$$\rho_v u_v^2 = W_c \text{ subst. } G_v/G = X, \text{ and we obtain,}$$

$$\frac{G_v^2}{\rho_v} = W_c$$

$$\frac{G^2 X_c^2}{\rho_v} = W_c$$

$$\therefore X_c = \frac{(W_c \rho_v)^{1/2}}{G}$$

Note: W_c is evaluated from $Hg - N_2$

adiabatic data - Reference 1,

pages 70 and 71.

. . . A-14

From heat balance (Equation A-2a):

$$1 - X = \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_N) \text{ for } X = 1 \text{ at } T_N = T_{Ni}$$

and at $X = 0$ $T_N = T_{NE}$

$$\therefore 1 = \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_{NE})$$

$$1 - x = \frac{T_{Ni} - T_N}{T_{Ni} - T_{NE}} \quad (\text{for } X = 1 \text{ at } T_N = T_{Ni})$$

if $X = X_E$ at boiler discharge then heat balance equation becomes

$$X_E = \frac{m_N C_N}{m h_{vf}} (T_{Ni} - T_{NE})$$

$$\text{and } X = \frac{m_N C_N}{m h_{vf}} (T_N - T_{NE})$$

combining

$$\frac{X}{X_E} = \frac{(T_N - T_{NE})}{(T_{Ni} - T_{NE})}$$

$$\frac{X}{X_E} (T_{Ni} - T_{NE}) + T_{NE} = T_N$$

. . . A-15

Substituting Equation (A-15) into Equation (A-13) eliminating T_N gives,

$$\frac{48 K_L \pi N \rho_v L}{11 m h_{vf} \rho_f} = \frac{1}{T_{Ni} - T_{HG}} \ln \left(\frac{\left(\frac{X}{X_E} (T_{Ni} - T_{NE}) + T_{NE} - T_{HG} \right) (T_{Ni} - T_{NE})}{\left[T_{Ni} - \frac{X}{X_E} (T_{Ni} - T_{NE}) - T_{NE} \right] (T_{NE} - T_{HG})} \right) - \frac{1}{T_{Ni} - T_{NE}} \ln \left(\frac{\left(\frac{X}{X_E} (T_{Ni} - T_{NE}) + T_{NE} - T_{HG} \right)}{(T_{NE} - T_{HG})} \right)$$

$$\frac{48 K_L \pi N \rho_v L}{11 m h_{vf} \rho_f} = \frac{1}{T_{Ni} - T_{HG}} \ln \left[\frac{\frac{X}{X_E} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1}{(1 - X/X_E)} \right] - \frac{1}{T_{Ni} - T_{NE}} \ln \left[\frac{X}{X_E} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right]$$

By substituting Equation (A-14) and letting $L = L_c$ we get an equation by which we can complete the tube length to the termination of the slug flow regime:

$$\frac{48 K_L \pi N \rho_v L_c}{11 m h_{vf} \rho_f} = \frac{1}{T_{Ni} - T_{HG}} \ln \left[\frac{\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{T_{NE} - T_{HG}} + 1}{1 - \frac{(W_c \rho_v)^{1/2}}{X_E G}} \right] - \frac{1}{T_{Ni} - T_{NE}} \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right]$$

$$\frac{48 K_g \pi N \rho_v L_c}{11 m h_{vf} \rho_f} = \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right]^{\frac{1}{T_{Ni} - T_{HG}}} - \ln \left[1 - \frac{(W_c \rho_v)^{1/2}}{X_E G} \right]^{\frac{1}{T_{Ni} - T_{HG}}} - \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right]^{\frac{1}{T_{Ni} - T_{NE}}}$$

$$\frac{48 K_g \pi N \rho_v L_c}{11 m h_{vf} \rho_f} = \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right]^{\left(\frac{1}{T_{Ni} - T_{HG}} - \frac{1}{T_{Ni} - T_{NE}} \right)} - \ln \left[1 - \frac{(W_c \rho_v)^{1/2}}{X_E G} \right]^{\frac{1}{T_{Ni} - T_{HG}}}$$

$$\frac{48 K_g \pi N \rho_v L_c}{11 m h_{vf} \rho_f} = \left(\frac{1}{T_{Ni} - T_{HG}} - \frac{1}{T_{Ni} - T_{NE}} \right) \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right] - \frac{1}{(T_{Ni} - T_{HG})} \ln \left[1 - \frac{(W_c \rho_v)^{1/2}}{X_E G} \right]$$

$$\frac{48 K_g \pi N \rho_v (T_{Ni} - T_{HG}) L_c}{11 m h_{vf} \rho_f} = \left(1 - \frac{T_{Ni} - T_{HG}}{T_{Ni} - T_{NE}} \right) \ln \left[\frac{(W_c \rho_v)^{1/2}}{X_E G} \frac{(T_{Ni} - T_{NE})}{(T_{NE} - T_{HG})} + 1 \right] - \ln \left[1 - \frac{(W_c \rho_v)^{1/2}}{X_E G} \right]$$

... A-16

which equation gives the slug flow boiling length, L_c

Note: see sample calculations, included in this appendix, for computing the length L_c

The following equation gives the effect of swirl passage pitch and plug insert on the NaK temperature profile:

$$\frac{48 K_L \pi N \rho_v L \alpha}{11 m h_{vf} \rho_f} = \frac{1}{T_{Ni} - T_{HG}} \ln \frac{(T_N - T_{HG})(T_{Ni} - T_{NE})}{(T_{Ni} - T_N)(T_{NE} - T_{HG})} - \frac{1}{T_{Ni} - T_{NE}} \ln \frac{(T_N - T_{HG})}{(T_{NE} - T_{HG})} \quad \dots A-17$$

where $\alpha = \frac{L}{P}$

heat conducting material

$$\alpha = \sqrt{1 + \left(\frac{\pi D}{P}\right)^2}$$

and $L\alpha$ in Equation A-17 represents the length of travel along the swirl flow passage, whereas L is measured along the axis of the tube. Confirmation of above equation with Aerojet 1/8 inch pitch plug data in AN-SNAP-65-240, 26 April 1965 (Reference 4).

See Figure 8.

Run No. 5-2 (4-8-65)

$$P_{\text{plug in}} = 382$$

$$K_L = 9.7 \text{ Btu/hr-ft F}$$

$$P_{\text{plug out}} = 314 \text{ psi}$$

$$h_{vf} = 123.5 \text{ Btu/lb}$$

$$P_{\text{ave}} = 348 \text{ psia}, 1110 = \bar{T}_{HG}$$

$$\rho_v = 4.5 \text{ lb/ft}^3$$

$$m = 481 \text{ lb/hr (8.01 lb/min)}$$

$$\alpha = 9.42$$

$$T_{Ni} = 1300 \text{ F}$$

$$T_{NE} = 1120 \text{ F}$$

$$T_{Ni} - T_{HG} = 1300 - 1110 = 190 \text{ F}$$

$$T_{Ni} - T_{NE} = 1300 - 1120 = 180 \text{ F}$$

$$T_{NE} - T_{HG} = 1120 - 1110 = 10 \text{ F}$$

NAA-SR-6322

$$\frac{48 \times 9.7 \times \pi \times 1 \times 4.5 \times 9.42 \text{ L}}{11 \times 481 \times 123.5 \times 800} = 1.187 \times 10^{-4} \text{ L} =$$

$$= \frac{1}{190} \ln \frac{(T_N - 1110) \times 180}{(1300 - T_N) 10} - \frac{1}{180} \ln \frac{(T_N - 1110)}{10}$$

Let $T_N = 1160$

$$1.187 \times 10^{-4} \text{ L} = \frac{1}{190} \ln \frac{(1160 - 1110)}{1300 - 1160} \times 18 - \frac{1}{180} \ln \frac{1160 - 1110}{10}$$

$$= \frac{1}{190} \ln \frac{50 \times 18}{140} - \frac{1}{180} \ln 5$$

$$= \frac{1}{190} \ln 6.42 - \frac{1}{180} \ln 5 = \frac{1.86}{190} - \frac{1.61}{180}$$

$$= 0.978 \times 10^{-2} - 0.894 \times 10^{-2} = 0.084 \times 10^{-2}$$

$$L = \frac{8.4}{1.187} \times 10^4 = 7.07 \text{ ft} \quad \frac{\Delta T_N}{\Delta L} = \frac{40}{7.05} = 5.65 \text{ F/ft}$$

Let $T_N = 1140$

$$1.187 \times 10^{-4} \text{ L} = \frac{1}{190} \ln \left(\frac{1140 - 1110}{1300 - 1140} \right) 18 - \frac{1}{180} \ln \left(\frac{1140 - 1110}{10} \right)$$

$$= \frac{1}{190} \ln \frac{30 \times 18}{160} - \frac{1}{180} \ln \frac{30}{10}$$

$$= \frac{1.22}{190} - \frac{1.1}{180} = 0.642 \times 10^{-2} - 0.612 \times 10^{-2} = 0.03 \times 10^{-2}$$

$$L = \frac{3 \times 10^{-4}}{1.187} \times 10^4 = 2.53 \text{ ft} \quad \frac{\Delta T_N}{\Delta L} = \frac{2.0}{2.53} = 7.9 \text{ F/ft}$$

Let $T_N = 1150$

$$\begin{aligned}
 1.187 \times 10^{-4} L &= \frac{1}{190} \ln \frac{(1150 - 1110)}{1300 - 1150} 18 - \frac{1}{180} \ln \frac{(1150 - 1110)}{10} \\
 &= \frac{1}{190} \ln \frac{40 \times 18}{150} - \frac{1}{180} \ln \frac{40}{10} \\
 &= \frac{1}{190} \ln 4.8 - \frac{1}{180} \ln 4 \\
 &= \frac{1.57}{190} - \frac{1.387}{180} = 0.826 \times 10^{-2} - 0.771 \times 10^{-2}
 \end{aligned}$$

$$1.187 \times 10^{-4} L = 0.055 \times 10^{-2} = 5.5 \times 10^{-4}$$

$$L = \frac{5.5}{1.187} = \underline{\underline{4.64}}$$

Slug flow heat transfer coefficient

$$h = \frac{h_g (1 - X)}{1 - X (1 - \rho_f / \rho_v)} \quad . . . A-18$$

where,

$$h_g = \frac{48}{11} \frac{K_g}{D}$$

$$@ p = 56.6 \text{ psia} \quad \rho_v = 0.83 \quad \rho_f = 800 \quad K_g = 9 \quad D = 0.19 \text{ in ID}$$

$$h_g = \frac{48}{11} \times \frac{9}{0.19} \times 12 = 2480$$

$$h = \frac{2480 (1 - X)}{1 - X \left(1 - \frac{800}{0.83}\right)} = \frac{2480 (1 - X)}{1 - X (1 - 963)}$$

$$h = \frac{2480 (1 - X)}{1 + X 962} \quad . . . A-19$$

NAA-SR-6322

$$X = 0.10 \quad h = \frac{2480 \times 0.9}{1 + 0.1 \times 962} = \frac{2480 \times 0.9}{97.2} = 23.0$$

$$X = 0.15 \quad h = \frac{2480 \times 0.85}{1 + 0.15 \times 962} = \frac{2480 \times 0.85}{145} = 14.5$$

$$X = 0.05 \quad h = \frac{2480 \times 0.95}{1 + 0.05 \times 962} = \frac{2480 \times 0.95}{49.1} = 48.2$$

See Figure 13 for comparison with data of Reference 5

Sample Calculations of NaK Temperature Profile for SNAP 2 Design Conditions

Assumed quantities

$$\rho_f = 800 \text{ lb/ft}^3$$

$$m_N = 76 \text{ lb/min}$$

$$\rho_v = 1.9 \text{ lb/ft}^3$$

$$m = 20.8 \text{ lb/min 4 tubes}$$

$$K_f = 9.2 \text{ Btu/hr-ft F}$$

$$P_{in} = 181 \text{ psia}$$

$$h_{vf} = 124.4 \text{ Btu/lb}$$

$$P_{out} = 121 \text{ psia}$$

$$C_N = 0.21 \text{ Btu/lb F}$$

$$T_{HG} (\text{sat. ave}) = 970 \text{ F @ 151 psia}$$

$$T_{Ni} = 1203 \text{ F}$$

Compute T_{NE} @ $X = 0 \text{ sat.}$

$$m h_{vf} = m_N C_N (T_{Ni} - T_{NE})$$

$$\frac{20.8 \times 124.4}{76 \times 0.21} = 1203 - T_{NE}$$

$$T_{NE} = 1041 \text{ F}$$

Applying Equation (A-13),

$$\frac{48 \times 9.2 \times \pi \times 4 \times 1.9 \text{ L}}{11 \times 20.8 \times 60 \times 124.4 \times 800} = \frac{1}{1203 - 970} \ln \frac{(T_N - 970)(1203 - 1041)}{(1203 - T_N)(1041 - 970)} \\ - \frac{1}{1203 - 1041} \ln \frac{(T_N - 970)}{(1041 - 970)}$$

For $T_N = 1050 \text{ F}$ $L = 1.8 \text{ ft}$

$T_N = 1070 \text{ F}$ $L = 24 \text{ ft}$

$T_N = 1060 \text{ F}$ $L = 9 \text{ ft}$

$T_N = 1080 \text{ F}$ $L = 45 \text{ ft}$

Sample calculations of NaK Temperature Profile for SNAP 8 Single Tube Test Conditions

(Run 42)

Assumed quantities:

$$m = 2810 \text{ lb/hr}$$

$$h_{vf} = 124 \text{ Btu/lb}$$

$$P_{in} = 295 \text{ psia}$$

$$\rho_v = 3.5 \text{ lb/ft}^3$$

$$T_{Ni} = 1284 \text{ F}$$

$$\rho_f = 800 \text{ lb/ft}^3$$

$$T_{HG}(\text{sat.}) = 1085 \text{ F}$$

$$K_g = 9.5 \text{ Btu/hr-ft F}$$

$$T_{NE} = 1123 \text{ F}$$

Applying Equation (A-13)

$$\frac{48 \times 9.5 \times \pi \times 1 \times 3.5 \text{ L}}{11 \times 2810 \times 124 \times 800} = \frac{1}{1284 - 1085} \ln \frac{(T_N - 1085)(1284 - 1123)}{(1284 - T_N)(1123 - 1085)} \\ - \frac{1}{1284 - 1123} \ln \frac{(T_N - 1085)}{(1123 - 1085)}$$

NAA-SR-6322

$$T_N = 1130 \text{ F} \quad L = 22 \text{ ft}$$

$$T_N = 1127 \text{ F} \quad L = 6 \text{ ft}$$

$$T_N = 1128 \text{ F} \quad L = 12 \text{ ft}$$

$$T_N = 1124 \text{ F} \quad L = 5.2 \text{ ft}$$

$$T_N = 1137 \text{ F} \quad L = 138 \text{ F}$$

Sample Calculations in Computing L_c (length of Tube to Termination of Slug Flow Regime)
from Conditioning Tests

Mass Velocity

$$G = \frac{5}{60} \frac{144 \times 4}{0.335^2 \pi} = 136 \text{ lb/sec-ft}^2$$

$$\rho_v = 1.6 \text{ lb/ft}^3 \quad X_E = 1 \quad \therefore \frac{(W_c \rho_v)^{1/2}}{X_E G} = \frac{(142 \times 1.6)^{1/2}}{1 \times 136} = 0.111$$

$$W_c = 142 \text{ lb/ft-sec}^2 \quad (\text{from adiabatic } N_2\text{-Hg tests})$$

$$T_{HG} = 970 \text{ F} \quad h_{vf} = 124.4 \text{ Btu/lb}$$

$$T_{N_i} = 1200 \text{ F} \quad K_\ell = 9.2 \text{ Btu/hr-ft F}$$

$$T_{N_i} - T_{NE} = 162 \text{ F} \quad \rho_f = 800 \text{ lb/ft}^3$$

$$m = 5 \text{ lb/min}$$

Applying Equation (A-16)

$$\frac{48 \times 9.2 \times \pi \times 1.6 \times 230 L_c}{11 \times 5 \times 60 \times 124.4 \times 800} = \left(1 - \frac{230}{162}\right) \ln \left[0.111 \times \frac{162}{68} + 1\right] - \ln [1 - 0.111]$$

$$L_c = 13 \text{ ft} \quad \text{from test } L_c \approx 12 \text{ to } 15 \text{ ft}$$

Computation for L_c with 1/2 NaK-Hg flow at reduced pressure level 5-12-65 test data

$$p = 18 \text{ psia} \quad \rho_v = 0.3 \text{ lb/ft}^3 \quad T_{HG} = 695 \text{ F} \quad G = 68 \text{ lb/sec-ft}^2; \quad T_{N_i} = 1192 \text{ F};$$

$$T_{NE} = 955 \text{ F}$$

$$\frac{48 \times 8.5 \times \pi \times 1 \times 0.3 \times 497 L_c}{11 \times 2.5 \times 60 \times 124.4 \times 800} = \left(1 - \frac{497}{237}\right) \ln \left[0.096 \times \frac{237}{260} + 1\right] - \ln [1-0.096]$$

$$\text{where} \quad \frac{(W_c \rho_v)^{1/2}}{X_E G} = \frac{(142 \times 0.3)^{1/2}}{1 \times 68} = 0.096$$

$$L_c = 7.25 \text{ ft}$$

$$\text{From test } L_c \approx 10 \text{ ft}$$

APPENDIX A

DEFINITION OF SYMBOLS

T_N	- NaK temperature
T_{Ni}	- Inlet NaK temperature
T_{NE}	- Exit NaK temperature for inlet mercury at saturated conditions
T_{HG}	- Mercury boiling temperature
m	- Mercury flow rate
m_N	- NaK flow rate
C_N	- NaK specific heat
h_{vf}	- Mercury latent heat of evaporation
D	- Tube diameter
L	- Tube length
U	- Overall heat transfer coefficient
h	- Heat transfer coefficient
h_l	- Liquid heat transfer coefficient
h_v	- Vapor heat transfer coefficient
ℓ	- Liquid slug length, passage length per pitch
ℓ'	- Distance between slugs
M_v	- Vapor weight
M_T	- Total mixture weight
X, X_c	- Flowing quality, flowing quality at termination of slug flow regime
X_E	- Exit quality

NAA-SR-6322

DEFINITION OF SYMBOLS (Continued)

- ρ_v - Vapor density
- ρ_f - Liquid density
- N - Number of tubes
- L_c - Tube length measured to termination of slug flow regime
- U_v - Vapor velocity
- G, G_v, G_{vc} - Mass velocity, vapor mass velocity, vapor mass velocity at termination of slug flow regime, respectively
- $W_c = \rho_v U_v^2$
- $\alpha \equiv l/P$ (See Equation A-17)
- P - Pitch, pressure
- K - Liquid thermal conductivity

APPENDIX B

SPRAY - ANNULAR FILM FORCED CONVECTION MERCURY BOILING

The purpose of this analysis is to formulate the basis for determining the NaK temperature profile when complete wetting results and the two-phase flow pattern is of the spray-annular film type. (See Figure B-1.) This mode of heat transfer represents a limiting case for SNAP 2 mercury boilers since the internal heat transfer resistance of the boiling mercury is at a minimum. The NaK temperature profile obtained by this method is comparable to those measured on some boiler tests in which exceptionally long superheat lengths were obtained.

Although the spring insert imparts a swirl to the flowing mixture which tends to drive the liquid to the wall, the intense turbulence level created by the high mass velocities can be expected to counteract this swirl and, thereby, form some entrained liquid spray. This same phenomenon occurs in an overloaded centrifugal separator which shows a drop-off in efficiency as the inlet flow exceeds the design point.

Integrity of the Liquid Film During Spray-Annular Flow

During spray -annular flow, most of the liquid can be carried in droplet form by the vapor, although under isothermal conditions a thin liquid film can be observed at the wall. During boiling or evaporation, the liquid film evaporates and must be replenished by droplets from the main stream. This replenishment is accomplished by eddy diffusion of droplets. The rate of droplet diffusion to the film is limited by the turbulent diffusion process. It can be postulated that the film will disappear when the evaporation rate of the film becomes greater

NAA-SR-6322

FIGURE B-1. SPRAY-ANNULAR BOILING

than the rate needed for the complete evaporation of all liquid droplets which diffuse to the wall. This condition is also referred to as "burnout" in Reference B-1. Another effect which can destroy the liquid film may be the formation of vapor in the wall-liquid interface by too great a temperature difference. In essence, the film is lifted from the surface by the vapor and the liquid rolled into drops, resulting in what is known as spheroidal boiling. When the film is on the wall, the heat transfer can be expected to be large. If the film disappears, the heat transfer rates can be considerably lower. Therefore, this critical point (burnout) is of importance in boiler design. The following analysis will derive equations for computing the point of film disappearance.

Diffusion Limitation

In Reference B-1, the problem of mass transfer of liquid droplets from a turbulent stream to a solid boundary is derived mathematically in a manner analogous to the treatment of turbulent momentum exchange and heat transfer. For the condition that all the liquid reaching the wall is evaporated, Reference B-1 gives;

$$Q/A = K_G S (1 - x) G_o h_{vf} \quad . . . B-1$$

Or in terms of the heat transfer coefficient,

$$h_G = \frac{K_G S(1 - x) G_o h_{vf}}{T_w - T_{HG}} \quad . . . B-2$$

Based on the analogy between mass and momentum transfer,

$$K_G = \frac{0.023}{Re_o^{0.2}} \quad \text{for } S = 1 \quad . . . B-3$$

Actual values of K_G as measured from Reference B-1, are compared with Equation B-3 in Figure B-2. In view of the oversimplifying assumptions in the derivation of Equation B-3, the order of magnitude agreement is remarkable. Assuming that the liquid film will disappear when the evaporation rate exceeds the diffusion rate, the condition for transition becomes,

$$h_G = h_{film} \quad . . . B-4$$

Where

$$h_{film} = K_f/a \quad . . . B-5$$

The combined heat transfer coefficient for the tube wall and NaK is estimated to be 2.8×10^3 . Therefore, the liquid evaporating film is usually of negligible resistance, and, as a result, the critical quality X_c , defining the disappearance of the liquid film, can be computed in a simple manner by means of Equation B-1,

$$U_{N-W} (T_{NC} - T_{HG}) = K_G S (1 - x_c) G_o h_{vf} \quad . . . B-6$$

Where

$$U_{N-W} \approx U_o$$

For a NaK heated boiler T_{NC} is a function of x_c and can be determined from a heat balance which gives;

$$T_{NC} = \frac{m_{HG} h_{vf}}{m_N C_{pN}} (X_c - 1) + T_{Ni} \quad . . . B-7$$

Combining Equations B-6 and B-7 results in,

$$U_{N-W} \left[\frac{m_{HG} h_{vf}}{m_N C_{pN}} (X_c - 1) + (T_{Ni} - T_{HG}) \right] = K_G S (1 - X_c) G_o h_{vf} \quad . . . B-8$$

* This is the small friction factor, multiply by 4 to obtain large friction factor

FIGURE B-2. EXPERIMENTAL DIMENSIONLESS MASS-TRANSFER COEFFICIENT VERSUS REYNOLDS NUMBER (REFERENCE B-1)

NAA-SR-6322

Substituting the test conditions of Table B-1 into Equation B-8 and selecting

$$\begin{aligned} K_{GS=1} &= 2 \times 10^{-2} \quad \text{for} \quad Re_o = 75,000 \text{ in Figure B-2. We have,} \\ &= 2800 \left[173 (x_c - 1) + (1240 - 1085) \right] \\ &= 2 \times 10^{-2} (1 - x_c) 0.65 \times 10^6 \times 129 \end{aligned}$$

and

$$x_c = 0.8$$

which corresponds to a computed value of

$$T_{NC} = 1205^\circ\text{F}$$

Boiler Analysis

It is a simple procedure to apply this technology to boiler design. One first computes the heating length, L_c , up to the point where "dry wall" boiling occurs as defined by Equation B-6 in terms of x_c . One then computes the final length based on the "dry wall" condition. This will be illustrated in the subsequent analysis.

The overall heat transfer coefficient U_o over the length, L_c , is primarily determined by the tube wall and NaK resistance.

Therefore,

$$U_{N-W} = \frac{m_N C_{pN} d T_N}{\pi D N (T_N - T_{HG}) d L} \quad \dots B-9$$

$$\text{Integrating} \quad \left[L \right]_0^{L_c} = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \ln (T_N - T_{HG}) \left[\right]_{T_{No}}^{T_{Nc}}$$

$$L_c = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \ln \frac{(T_{NC} - T_{HG})}{(T_{NO} - T_{HG})} \quad \dots B-10$$

Or, if the temperature profile is to be computed,

$$L = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \ln \frac{(T_N - T_{HG})}{(T_{NO} - T_{HG})} \quad \dots B-11$$

Substituting values from Table B-1,

$$L = \frac{71.1 \times 6.0 \times 0.21 \times 12}{\pi \times 0.335 \times 3 \times 2800} \ln \frac{(T_N - T_{HG})}{(T_{NO} - T_{HG})}$$

$$L = 1.215 \ln \frac{(T_N - T_{HG})}{(T_{NO} - T_{HG})} \quad \dots B-12$$

Since "suppressed boiling" conditions occurred near the mercury inlet, $T_{NO} - T_{HG}$ is theoretically equal to zero, however, by examining pool boiling data for mercury (Liquid Metals Handbook) approximately 10°F of superheat is required to initiate boiling. This value is approximate and is dependent on many variables, such as surface finish, cleanliness, liquid pressure etc. Therefore, let $T_{NO} - T_{HG} = 10$ and

$$L = 1.215 \ln \frac{(T_N - 1085)}{10} \quad \dots B-13$$

And, if $T_{NC} = 1205^\circ\text{F}$ at X_c as computed by Equation B-8, then

$$L_c = 1.215 \times \ln \frac{(1205 - 1085)}{10} = 3.02 \text{ ft}$$

To compute the NaK temperature profile and boiling length for $L > L_c$ one proceeds as follows:

$$1/U_o = 1/U_{N-W} + 1/h_G \quad \dots B-14$$

Thus,

$$\left(\frac{1}{U_{N-W}} + \frac{1}{h_G} \right)^{-1} = \frac{m_N C_{pN} d T_N}{\pi D N (T_N - T_{HG}) dL} \quad \dots B-15$$

Substituting Equation B-2,

$$\left[\frac{1}{U_{N-W}} + \frac{(T_W - T_{HG})}{K_G S (1-x) G_o h_{fv}} \right]^{-1} = \frac{m_N C_{pN} d T_N}{\pi D N (T_N - T_{HG}) dL}$$

$$dL = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \frac{D T_N}{(T_N - T_{HG})} + \frac{m_N C_{pN}}{\pi D N K_G S (1-x) G_o h_{fv}} \frac{(T_W - T_{HG})}{(T_N - T_{HG})} dT_N$$

Substitute,

$$1 - x = \frac{m_N C_{pN}}{m_{HG} h_{fv}} (T_{Ni} - T)$$

which is the heat balance

$$\text{and } G_o = \frac{m_{HG} 4}{\pi D^2 N}$$

we obtain

$$dL = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \frac{d T_N}{(T_N - T_{HG})} + \frac{D}{K_G S 4} \frac{(T_W - T_{HG})}{(T_N - T_{HG})} \frac{d T_N}{(T_{Ni} - T_N)} \quad \dots B-16$$

For conservatism and simplicity let $T_W = T_N$ then Equation B-16 becomes

$$dL = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \frac{d T_N}{(T_N - T_{HG})} + \frac{D}{K_G S 4} \frac{d T_N}{(T_{Ni} - T_N)} \quad \dots B-17$$

Integrating,

$$\left[L \right]_{L_c}^L = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \ln (T_N - T_{HG}) \Bigg|_{T_{N_c}}^{T_N} + \frac{D (-1)}{K_G S_4} \ln (T_{N_i} - T_N) \Bigg|_{T_{N_c}}^{T_N}$$

$$L - L_c = \frac{m_N C_{pN}}{\pi D N U_{N-W}} \ln \frac{(T_N - T_{HG})}{(T_{N_c} - T_{HG})} + \frac{D}{K_G S_4} \ln \frac{(T_{N_i} - T_{N_c})}{(T_{N_i} - T_N)}$$

. . . B-18

Substituting the values from Table B-1 and letting $K_{GS} = 1 = 2 \times 10^{-2}$, $R_{e_o} = 75,000$

(Figure B-2)

$$L - L_c = \frac{71.1 \times 60 \times 0.21 \times 12}{\pi 0.335 \times 3 \times 2800} \ln \frac{(T_N - 1085)}{(T_{N_c} - 1085)} + \frac{0.335}{12 \times 2 \times 10^{-2} \times 4} \ln \frac{(1240 - T_{N_c})}{(1240 - T_N)}$$

$$T_{N_c} = 1205^\circ\text{F at } X_c \text{ (see Equation B-8 and example)}$$

$$L - L_c = 1.215 \ln \frac{(T_N - 1085)}{120.4} + 0.349 \ln \left(\frac{34.6}{1240 - T_N} \right) \quad . . . \text{ B-19}$$

Figure B-3 compares the NaK temperature profile as computed by means of Equations B-18 or B-19 and B-11 or B-12 with NaK temperature measurements. As can be seen, the agreement is good. However, further comparison should be made with other tests for definite substantiation of the equations if they are to be used for design purposes. Large values of $T_{N_o} - T_{HG}$ at the mercury inlet of the boiler may result in spheroidal boiling which would necessitate the application drop wise boiling equations. (See Reference B-2).

FIGURE B-3. COMPARISON OF THEORETICAL NAK TEMPERATURE PROFILE AS COMPARED BY EQUATIONS B-11 & B-13 WITH TEST DATA RUN 1952 (5C-2) 30° LOGGER TEMPERATURE CORRECTION

TABLE B-1. TEST CONDITIONS

$$C_{PN} = 0.21 \text{ Btu/lb F}$$

$$m_N = 71.1 \text{ lb/min}$$

$$m_{HG} = 20 \text{ lb/min}$$

$$C_{pf} = 0.0326 \text{ Btu/lb F (liquid mercury)}$$

$$G_o = 0.65 \times 10^6 \text{ lb/hr ft}^2; 181 \text{ lb/sec ft}^2$$

$$N = 3 \text{ tubes (one of four tubes blocked)}$$

$$\mu_v = 0.242 \text{ lb/ft-hr}$$

$$D = 0.335 \text{ in.}$$

$$\frac{D G_o}{\mu_v} = 75,000 \text{ (Reynolds number)}$$

$$T_{Ni} = 1240^\circ \text{ F}$$

$$T_{HG} = 1085^\circ \text{ F (300 psia)}$$

$$\rho_v = 3.5 \text{ lb/ft}^3$$

$$\rho_f = 800 \text{ lb/ft}^3$$

$$\tan \alpha = 0.785$$

$$\frac{m_{HG} h_{vf}}{m_N C_{PN}} = 173^\circ \text{ F}$$

$$h_{vf} = 129 \text{ Btu/lb}$$

$$D_h \text{ (NaK side)} = 0.314 \text{ in. (hydraulic diameter)}$$

$$K_f \text{ (NaK)} = 15 \text{ Btu/hr ft F}$$

$$R_e \text{ (NaK)} = 12,900$$

$$P_r \text{ (NaK)} = 0.0055 \text{ (Prandtl number)}$$

$$Nu \text{ (NaK)} = 8 \text{ (estimate) (Nusselt number)}$$

$$h \text{ (NaK)} = 4600 \text{ Btu/hr ft}^2 \text{ F}$$

$$h \text{ (tube wall)} = 7200(\text{wall thickness}) = 0.020 \text{ in.}$$

$$K = 12 \text{ Btu/hr ft F}$$

$$U_{N-W} \text{ (tube wall and NaK)} = 2800 \text{ Btu/hr ft}^2 \text{ F}$$

"Suppressed boiling conditions existed near the mercury inlet."

APPENDIX B

LIST OF REFERENCES

1. K. Goldman, H. Firstenberg, C. Lombardi, Burnout in Turbulent Flow - A Droplet Diffusion Model. Trans. ASME May 1961.
2. The SNAP II Power Conversion System Topical Report No. 17, Mercury Boiling Research, TRW ER-4833.

APPENDIX B

DEFINITION OF SYMBOLS

a	- film thickness
g_s	- standard sea level value for gravity
D	- tube diameter
g	- gravity acceleration
x	- quality
x_c	- critical quality (dry wall condition)
L	- boiler length
L_c	- boiler length to critical quality
K_G	- dimensionless mass-transfer coefficient
Re_o	- total flow rate Reynolds number
$h_G; h_{film}$	- heat transfer coefficients
K_f	- liquid conductivity
h_{f_v}	- latent heat of evaporation
T_N	- NaK temperature (see Figure B-1 for additional T_N definitions)
T_W	- Wall temperature
T_{HG}	- mercury temperature
G_o	- total flow mass velocity
ΔT	- temperature difference
U_{N-W}	- heat transfer coefficient of tube and NaK
U_o	- overall heat transfer coefficient
m_N	- NaK flow rate

NAA-SR-6322

DEFINITION OF SYMBOLS (Continued)

m_{HG}	- mercury flow rate
C_{pN}	- NaK specific heat
N	- number of tubes
S	- ratio of vapor to liquid velocity
Q	- heat transfer rate
A	- heat transfer area
ρ_s	- partial density
ρ_f	- liquid density
ρ_v	- vapor density
μ_v	- vapor viscosity