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TRW REPORT No. ER-4833

THE SNAP II POWER CONVERSION SYSTEM

TOPICAL REPORT No. 17

MERCURY BOILING RESEARCH

MASTER

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SNAP II POWER CONVERSION SYSTEM
TOPICAL REPORT NO. 17
MERCURY BOILING RESEARCH
TRW REPORT NO. ER-4833

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1.0 INTRODUCTION

The SNAP II auxiliary power unit for a 3 KW electrical output consists of an Atomics International reactor heat source and heat rejector (condenser-radiator) and a TRW power conversion system. Thermal energy is transferred from the reactor to the power conversion system by a liquid metal (NaK) primary loop.

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, and the mercury pump which supplies condensed liquid to the boiler. The turbine is mounted on a single shaft with a mercury pump, a permanent magnet conduction NaK pump, and an alternator to generate 3 KW electrical power. The entire shaft assembly is enclosed in one housing and referred to as the combined rotating unit (CRU).

The work in the boiler development area up to September, 1960, is summarized in ER-4521, "Boiler Development" (Ref. 1-1). The boiler development program emphasized continuous development of primary fluid to mercury heat exchanger test sections and full-scale boilers to insure provision of vapor for systems operations. Research-type efforts such as those presented herein were minimized for the sake of expediency and until the overall boiler phenomena were better understood. The inherently simple and compact once-through boiler was selected as the design configuration for SNAP II to avoid complications of recirculating pumps and separators and minimize mercury inventory. A shell and tube arrangement of flow passages is used, with mercury flowing through the tubes and primary fluid through the shell. Large ratios of length to diameter are used for the mercury side to provide better assurance that boiling is complete and liquid is not carried over. A small number of tubes can also be used, thereby minimizing tube-to-tube sheet fabrication problems and permitting design of compact heat exchangers.

As a result of the boiler development efforts, certain basic concepts on the boiling mercury process were established. In the forced convection, high exit quality boiling of mercury, the dropwise flow regime extends over a major portion of the boiling length. The regimes in which the liquid phase is continuous cover an insignificant portion of the heating surface. The early breakup of the liquid mercury film is caused by its high surface tension and large ratio of liquid-to-vapor density. Low heat transfer coefficients result when heat must go from the tube wall to the vapor and then to the liquid particles. Swirl generators or swirl devices are therefore used to force the liquid component of the two-phase boiling mixture against the tube wall so that heat is transferred directly from the wall to the drop. The tendency of mercury to form drops would also significantly decrease the effect of "wetting" on the heat transfer to high vapor fractions flowing through tubes.

ER-4521 (Ref. 1-1) presents a mathematical development for the once-through forced convection vortex boiling of mercury based on the full-scale boiler tests and hot plate experiments reported therein. The complexity of the boiling process occurring in a full-scale boiler did not permit the independent study of temperature difference, pressure level, flow rate, and quality level. Consequently, the theory could not be verified and localized

effects of these parameters could not be established. Such information is needed to design minimum weight SNAP II and similar boilers. The need for this information is typical of the general sparsity of liquid metal boiling engineering data (Ref. 1-2 and 1-3). In particular, the high exit quality data and theory in vortex flow presented should prove useful. To provide some of the information needed, the boiling research program was initiated at TRW in May 1960.

The work in the boiling research program involved the building of a test rig, rig check-out, and obtaining of data for four different test section configurations. Three of the configurations tested had swirl wire inserts. One was a bare tube. A special effort was made during the testing of one of the swirl wire test sections to obtain data on the independent effect of temperature difference, pressure level, flow rate, and quality level on the heat transfer rate. The data, theory and correlations of this report form part of the foundation for the development of mercury boiling engineering.

2.0 SUMMARY

SNAP II mercury boiler design has advanced considerably from a cut-and-try development stage described in Reference 1-1. The present state of the art is the combined result of (1) the development of mathematical expression for the heat transfer and fluid mechanics of mercury in vortex forced convection boiling, (2) the local experimentation described in this report, and (3) full-scale SNAP II boiler testing.

The theoretical equations are based on a drop model for the forced convection boiling of mercury. Mercury tends to form drops as a result of its high surface tension and large liquid to vapor density ratio. The analysis of this report further establishes the validity of this model and shows that account must be made for drop deformation due to high "g" forces or lack of drop establishment due to low velocities. Initial drop size (δ) is assumed to equal the swirl wire size. It is further assumed that drops do not agglomerate as they are swirled through the tube. The drops may contact the tube wall surface, ride on a vapor film, or both, depending on the radial force tending to hold the drops against the wall and the vapor repulsive force due to the evaporation. When the mercury is in contact with the wall, Equation 7-16 of this report is used:

$$\frac{hD}{k_v} = \frac{1}{32} Nu_{\delta}^2 \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{\delta_0} \right)^2 \frac{(1-x)^{1/3}}{x} \frac{\rho_v}{\rho_f}$$

When film forced convection boiling prevails, Equation 7-17 is used:

$$\frac{hD}{k} = \frac{6^{1/3}}{4} \frac{(1-x)^{2/3}}{x^{1/3}} \frac{D}{\delta_0} \beta^{4/3} (\tan \alpha)^{2/3} \left(\frac{\rho_v}{\rho_f} \right)^{2/3} \left(\frac{DG_T}{\mu_v} \right)^{2/3} \left(\frac{h_{fv} \mu_v}{k_v \Delta T_m} \right)^{2/3}$$

The transitional equation for intermittent contact (Equation 7-25) was obtained empirically as a result of the data of this report:

$$\frac{hD}{k_v} = C^4 (234) \left(\frac{k_v}{k_f} \right)^4 (1-x)^{4/3} x \left(\frac{\delta_0}{D} \right) \left(\tan \alpha \frac{DG_T}{\mu_v} \frac{h_{fv} \mu_v}{k_v \Delta T_m} \right)^2$$

To determine which equation applies, checks must be made to determine which boiling regime exists. The limitations of applying the theory to design are described in Section 8.0.

Local tests were run on the configurations described in Table 2-1 to verify the heat transfer equations and provide empirical data on the heat transfer and pressure drop. The ranges over which data were obtained are summarized in Table 2-2. In obtaining the empirical data, the parametric ranges of interest to SNAP II were emphasized.

The test sections were 30 in. long and radiantly heated by concentric electrical heaters. Tube wall surface temperatures were measured to give temperature difference (wall minus saturation) and a heat balance was taken to give net heat flow. Quality level was controlled by means of the forced convection boiler upstream of the test section. Flow was provided by a pressurized bellows scheme. Analysis of the experimental techniques and data indicate the maximum error in net test section heat flow could be about $\pm 10\%$ and about $+27^\circ\text{F}/-13^\circ\text{F}$ in temperature difference. All of the data was correlated according to the heat transfer

TABLE 2-1
BOILER TUBE CONFIGURATIONS TESTED

<u>Designation</u>	<u>OD In, D_o</u>	<u>Wall, In.</u>	<u>In, D_i</u>	<u>Swirl Wire Dia, in. ϕ_o</u>	<u>Swirl Wire Pitch In, P</u>	<u>P/D_i</u>	<u>D_i/L_o</u>	<u>Length</u>	<u>Mat'l</u>	<u>Heat Transfer Means</u>	<u>No. of Data Points</u>
SN 1	5/8	0.065	0.495	0.054	1.8	3.64	9.2	30 in.	316 SS	Radiation	39
B	5/16		0.028	0.256	Bare Tube			30 in.	HS-25	Radiation	19
SN 2	5/16	0.028	0.256	0.054	1.0	3.9	4.7	30 in.	316 SS	Radiation	15
SN 3	5/16	0.028	0.256	0.028	1.0	3.9	9.2	30 in.	HS-25	Radiation & Conduction Through "Thermon" Cement	72
Rig Boiler	3/8	0.035	0.305	0.062	0.88	2.9	4.9	115 ft.	316 SS	Conduction Through Thermon	42

theory and substantiated the trends and magnitudes of the theory. The effect of drop deformation on heat transfer was detected.

Analysis of the test section data also showed that a considerable improvement over a bare tube is obtained by using a swirl device. An approximate factor of three times the bare tube heat transfer was obtained. This effect depends on swirl device geometry, mass velocity, boiling regime, etc., however, and could be greater or smaller depending on the application. It was found that the rig boiler swirl wire influenced the test section data. This was particularly noticeable with the SN 3 test data which had a 0.028 in. swirl wire as compared to the 0.0625 in. swirl wire used in the boiler.

Pressure drop correlations are presented for all test sections except the SN 1 for which the pressure drop was too low for measurement. It is shown that the drop flow of mercury results in greater pressure drop than the annular two-phase flow more commonly experienced. The mercury drops offer more surface area and therefore increased drag on the vapor. It is further shown that the wire swirl device resulted in less two-phase pressure drop effect than the bare tube. This is attributed to the wire shielding effect.

3.0 PROGRAM OBJECTIVES AND DESIGN OF EXPERIMENT

The boiling research program was established to provide data on heat transfer and pressure drop during vortex once-through forced convection boiling of mercury. Data obtained could then be correlated on the basis of the mathematical model briefly described in Section 7.0 as well as empirically to improve the design information available for SNAP II and similar boilers.

The specific objectives of the boiling research program were as follows:

1. Investigate the average local heat transfer coefficients and corresponding pressure drops for the forced convection boiling of mercury in various boiler tube configurations at different flow rates, pressure levels, quality levels and temperature differences.
2. Correlate the data on the basis of the mathematical drop model introduced in ER-4521 (Ref. 1-1), evaluate the theory, and incorporate refinements.
3. Correlate the data on an empirical basis.
4. Present the results of 2 and 3 above in a form useful to a designer of mercury boilers.

To provide data principally applicable to the SNAP II boiler, the following approximate design point parameter variations were target ranges for testing:

Quality, x	0 to 1.0
Flow, w	~ 3 lb/min-tube (1/4 in. ID tube)
Pressure, p	125 to 210 psia
Temperature difference, ΔT	0 to 300

Whenever possible, data were taken within these ranges. The known variables in the forced convection dropwise boiling of mercury in a vortex flow induced by tube inserts are listed below:

PRIMARY VARIABLES

FLOW, THERMAL, AND PROPERTY

1. Flow rate
2. Temperature difference
3. Pressure
4. Quality
5. Tube wall material

GEOMETRIC

6. Type of insert (spring or twisted ribbon)
7. Pitch
8. Wire diameter
9. Tube diameter
10. Tube wall thickness

VARIABLES DEPENDING ON THE HEAT TRANSFER MECHANISM IN DROPWISE BOILING (See Section 7.1 for discussion of dropwise heat transfer)

11. Film (drops)	14. Film (spheroids)
12. Intermittent contact (drops)	15. Intermittent contact (spheroids)
13. Contact (drops)	16. Contact (spheroids)

If spring inserts, tube wall material, and tube wall thickness are selected as fixed quantities, the number of variables is 13. For each variable a minimum of four data points is required for a complete correlation. As can be computed, the required number of test points is prohibitive. If the variables are further reduced by fixing the wire diameter, pitch, tube diameter, and reducing the minimum number of test points to three, 3^{10} test points are required. This is also prohibitive if one considers that the average time for each test point is nearly four hours. Some compromises were therefore made in the selection of configurations and parameter variations to be tested.

The test section configurations tested are presented in Table 2-1. Test section configuration SN 2 was selected as it was the SNAP II boiler configuration at the time of test. Test section configuration B, a bare tube with SN 2 dimensions, was used to measure the effect of a vortex type flow. The SN 1 test section configuration was selected to provide data on the effect of a larger tube-to-wire diameter ratio in a relatively large tube. Due to rig limitations, an adequate variation in significant parameters could not be obtained with the large SN 1 test section. Test section SN 3 with a large tube-to-wire diameter ratio was then tested. Test section SN 3 had the same tube diameter as SN 2 but a smaller wire. During all tests the rig boiler, as described in Table 2-1 and Section 4 and the system were identical.

Because of funding and time limitations, sections SN 1, SN 2, and B were tested in a random manner; that is, no intentional relationship between controllable inputs was maintained. The burden of correlation was left to the theory available in Reference 1-2. A supplemental funding allowed a more comprehensive study of test section SN 3. In particular, the variation of parameters was controlled such that the independent effect of flow (mass velocity), pressure level, quality level and temperature difference on heat flux could be observed. The variations were made in ranges applicable to the SNAP II system. Table 3-1 presents the test objectives for the SN 3 test section.

TABLE 3-1
SN 3 TEST SECTION PLAN*

$\dot{m}$ lb/hr	$\bar{p}$ psia	$\bar{x}$ %	$\dot{m}$ lb/hr	$\bar{p}$ psia	$\bar{x}$ %
150	150	10	240	150	20
150	150	20	150	200	50
150	150	50	150	60	50
150	150	75	150	35	50
90	150	20			

* Temperature difference, ΔT , was the remaining variable.

4.0 DESCRIPTION OF TEST RIG, TEST SECTION, AND INSTRUMENTATION

As explained in Section 3, the operating ranges and test section internal configurations used in the boiling research program were based on SNAP II boiler requirements. Thus, the basic operating ranges for the test rig were as follows:

1. maximum flow rate	4 lb/min
2. minimum flow rate	1.5 lb/min
3. maximum system pressure	500 psia
4. maximum boiler power	~15 kw
5. test section power	~2.8 kw
6. maximum test section or tube temperature level	~1400°F

The design value of 4 lb/min flow was chosen to adequately provide the 3 lb/min flow desired. The 500 psia pressure level is more than adequate considering that 200 psia was the pressure level desired to satisfy the SNAP II conditions. The boiler power was selected to give 0 to 1.0 variation in inlet quality to the test section for the flow ranges of interest. Test section power was selected to give a suitable variation of quality change (and therefore heat flux and temperature difference) across the test section for the flow ranges desired. The temperature level was based on the combination of maximum saturation temperature level of interest and the maximum ΔT expected for heat transfer. In general, one has to consider minimum values of operating parameters, but these were not significant in this test series.

The subsections below describe test rig design philosophy, major rig components, and the primary instrumentation.

4.1 GENERAL RIG DESCRIPTION

The SNAP II boiling research rig is described schematically in Figure 4-1. It is basically composed of a N₂-pressurized reservoir-accumulator mercury feed system, an electrically heated forced convection boiler, a 30-in. test section, an air-cooled condenser, and a combination collecting pot-mass weighing system.

Power input to the boiler controlled the quality of the mercury leaving the boiler. Guard heaters were controlled simultaneously to minimize boiler heat losses (see Section 4.2.2 for a detailed description of the boiler). Power input to the test section was controlled to vary total heat flux; variations of quality change and mean temperature difference were additional results. Guard heaters were also used in the test section to minimize heat losses.

The pressure level of the system was controlled by adjustment of heat rejection area and pressure drop at the condenser exit. This was accomplished by varying the cooling air flow rate and the condenser exit valving.

SCHEMATIC DIAGRAM OF SNAP II BOILING RESEARCH RIG

FIGURE 4-1

Mercury leaving the condenser was collected in a pot on a scale which allowed accurate measurement of flow. After collection at the condenser exit for weighing, the mercury was gravity fed into one of two stainless steel reservoirs. While one of the system reservoirs was collecting mercury, the other reservoir was pressurized with nitrogen which drove liquid mercury through the stainless steel bellows, thus serving as a pump. When one reservoir was emptied and the other had been filled, two four-way valves simultaneously depressurized the supplying reservoir and pressurized the receiving reservoir, reversing their duties. Discontinuities in flow were avoided by using an accumulator, similar in construction to the reservoir, to supply mercury to the system during the transition of the reservoirs from receiver to supplier. This insured that mercury was supplied from the reservoirs to the boiler at a constant rate and pressure head.

The reservoir-accumulator pumping scheme was selected because steady flow at constant pressure was possible without the use of seals. These advantages were felt to more than compensate for the large amount of mercury inventory required. At one time a separator had been used at the condenser exit and separate heat rejection paths were incorporated in the system, one for the liquid and one for the vapor. This scheme made rig operation difficult, particularly control of system pressure, so the separator scheme was eliminated. The separator system had also originally been intended to check test section quality level; however, the heat balances on the boiler and test section were found to be sufficiently accurate for this calculation.

4.2 DETAILED RIG COMPONENTS

The rig boiler, test section, and certain instrumentation are the major components of the rig; they have the most direct influence on the heat transfer data of interest. Accordingly, these components are described in detail below. Other components and instruments are less important and therefore do not require elaboration.

4.2.1 Test Section

The test section tube was 4 ft. long with a heated length of 30 in. Figure 4-2 is a diagram of the test section. Two 1-3/4 in. ID by 2-5/8 in. OD by 30 in. long, semi-cylindrical Hevi-Duty Electric Company heater halves enclosed the tube. The total rating of the two units was 3 kw on 230 volts. The tube was centered within the test heater by circular stainless steel baffle sheets. A sheet was placed between each thermocouple position.

The test heater was enclosed in stainless steel half shells that serve to mount guard thermocouples. The test heater was surrounded by a 5 in. OD x 3-3/4 in. ID x 32 in. long guard heater of the same type as the test heater. This guard heater was provided with stainless steel half shells, fitting snugly to the internal diameter of the heater, that served to mount additional guard thermocouples.

The space between the two heaters, i.e., the stainless steel half-shells, was filled with Babcock-Wilcox, 6 lb density, Kaowool thermal insulation.* The guard heater halves were strapped together and the unit was wrapped with additional insulation. The entire

* Insulation thermal conductivity for Kaowool varies $\pm 10\%$ around the nominal value.

TYPICAL TEST SECTION CONSTRUCTION (FOR TEST SECTION SN 3)

unit was then enclosed in a sheet metal housing. The ends of the test section were unguarded; however, the guard heater extended beyond the test heater thereby serving to guard the ends to some extent. In addition, the ends were thoroughly insulated with additional Kaowool to minimize losses to the ambient.

Test section instrumentation is described in Section 4.3.1. In the SN 1, SN 2 and B test series and the first SN 3 series (runs 6020 through 6260), the test section was constructed essentially as described. In the second SN 3 series (runs 6270 through 6591), the space between the tube and test heater was filled with "Thermon T-63"*; thus, heat was transferred to the tube primarily by radiation in the first instance and by conduction in the second. "Thermon" was utilized as a conductive medium to limit the thermal resistance between tube wall and heater in order to prevent excessive heater temperatures and premature failure under the high heat flux tests in the SN 3 investigations. Tests of the SN 1, SN 2 and B test sections were run at lower temperature levels and, therefore, premature failures were not experienced.

4.2.2 Rig Boiler

The boiler consisted of a 3/8 in. OD x 0.035 in. wall, Type 316 stainless steel tube 113 feet long. The tube was equipped with the swirl wire having a 1 in. pitch and 0.0625 in. diameter. The tube was wrapped around a 7-5/8 in. OD x 27 in. long special cartridge heater rated at 15 kw on 440 volts, 60 cycle AC as shown in Figure 4-3. The heater winding consisted of three equal resistance sections for use in a three-phase system.

A stainless steel sleeve was fitted over the boiler tubes, and the space between the tubes, sleeve and heater was filled with "Thermon T-63," a high-graphite heat transfer cement. The sleeve and both ends were surrounded with 1-1/2 in. thick Babcock-Wilcox, 6 lb. density, Kaowool thermal insulation. Special, stainless steel-enclosed, electrical resistance heaters surrounded the insulation to serve as guard heaters. The guard heaters consisted of four separate resistance heater elements, thus permitting guarding of four areas of the boiler. In addition, two ring heaters were used to guard the ends of the cartridge heater against heat loss to the ambient. The entire unit was enclosed in additional insulation to minimize heat loss from the guard heaters to the ambient.

The boiler instrumentation is described in Section 4.3. The boiler inlet is at the bottom of the coil and the discharge at the top in a horizontal take-off to the test section. The take-off section was 11 in. long and was equipped with a swirl wire identical to the wire in the boiler. The test section was coupled to the take-off by a compression fitting followed by a 15 in. long entrance section to the heated part of the test section. The entrance length was equipped with swirl wire identical to the wire in the test section. The entrance section included a 90 degree bend of 6 in. radius, approximately. The total tube length of 26 in. between boiler and test section was well-insulated to minimize heat loss to the ambient atmosphere.

* Thermon is a graphite-base high temperature heat transfer cement.

SNAP II RESEARCH RIG BOILER SCHEMATIC

4.3 RIG BOILER AND TEST SECTION INSTRUMENTATION

4.3.1 Temperature

Test Section

The stream temperature within the test tube was measured at the entrance and exit of the test section at points 1 and 18 shown in Figure 4-4. The internal thermocouple probes were Chromel P-Alumel in a 1/16 in. OD stainless steel tube with a junction welded to the tube at the end to provide maximum speed of response. Both thermocouples were installed parallel to the stream and immersed a length sufficient to eliminate conduction losses. The presence of the swirl wire maintained the liquid mercury near the tube wall and, consequently, the immersed couples measured vapor temperature.

Sixteen surface thermocouples fabricated from No. 20 AWG Chromel P-Alumel wire were welded to the tube wall. Thermocouples 2 and 3 of Figure 4-4 were outside the test section and the balance, thermocouples 4 to 17 were within the test section. The method of installing these thermocouples on the tube is shown in Figure 4-5. Initially, smaller gage wire was used; however, corrosion at the point of contact with the tube resulted in many failures and the decision to use a heavier gage wire. Thermocouple application error is discussed in Section 5.3.

Guard thermocouples, 19-24 and 13B to 24B, were installed on the stainless steel half-shells in the positions shown in Figure 4-4, in the same manner as on the test tube. During a test run, the average of the differences (19-20, 21-22, etc.) was computed and adjusted to as near zero as possible by adjusting guard heater input.

All thermocouples were fabricated from the same spool of standard wire for which the limit of error was $\pm 4^{\circ}\text{F}$ from 0°F to 530°F and $\pm 3/4\%$ above 530°F . These accuracies were checked as described in Section 5.3.2, "Correlation of Saturation, Wall, and Stream Temperature." Connections were made as required by means of Marlin connectors. A Lewis selector switch was used to connect any thermocouple to a Minimate Temperature Indicator, a product of the Thermo-Electric Company, Inc. This instrument is a null-balance potentiometer-type pyrometer having two ranges, $0-1250^{\circ}\text{F}$ and $1200-2400^{\circ}\text{F}$, with 5°F scale divisions, an accuracy of $1/4\%$ of full scale, and automatic cold-junction compensation.

Boiler

The boiler was instrumented with thirty Iron-constantan thermocouples installed on the boiler tube in the same manner as described in Figure 4-5. These surface thermocouples were installed in the positions given by the tables of Figures 6-5A and 6-5B on pages 62 and 63 of this report.

Boiler inlet and outlet stream thermocouples were contained in a 1/16 in. diameter stainless-steel sheath with the thermocouple welded to the end of the sheath in order to

TEST SECTION THERMOCOUPLE POSITIONS

Note: Tube surface thermocouples are in the vertical plane;
heater and insulation thermocouples are in the horizontal plane.

I - Immersed

FIGURE 4-4

THERMOCOUPLE INSTALLATION ON TEST SECTION TUBE

FIGURE 4-5

obtain maximum speed of response. To minimize conduction errors, these thermocouples were welded into the boiler tube with a substantial immersion length.

In addition, eight iron-constantan guard thermocouples were installed on the surface of the sleeve enclosing the boiler tubes and eight iron-constantan guard thermocouples were installed on the surface of the guard heater, as shown in Figure 4-3. The electrical input to the guard heaters was adjusted to provide minimum difference for each test run. All thermocouples were fabricated from the same spool of standard tolerance wire.

A Minneapolis-Honeywell strip chart indicator-recorder, calibrated for 0 to 1600^oF for iron-constantan, provided a readout for all thermocouples by means of an external switching arrangement. This instrument has cold junction compensations and an accuracy of 1/4% of span.

4.3.2 Power Input

Test Section

The heater used in the test section was connected to a 220 volt, 60 cycle, single-phase source. The power consumption in the heater was calculated from the product of current and voltage. The power factor for the heater was established as unity. No corrections were necessary for power consumption in the meters due to the large ratio of power dissipated in the heater to power consumed in the meters.

The ammeter used to obtain heater current was a Simpson, 10 amp full-scale, 2 amp graduation, iron-vane unit with a basic accuracy of $\pm 5\%$. The volt meter used to obtain voltage drop across the test section heater terminals was a Simpson, 200 volt full-scale, 5 volt graduation, iron-vane unit with a basic accuracy of $\pm 5\%$. Both instruments were calibrated in place against instruments having a basic accuracy of $\pm 1/4\%$.

Boiler

The heater used in the boiler was connected to a 440 volt, three-phase, 60 cycle source. Power consumption was calculated from the product of current and voltage. The power factor of the heater was established as unity. The ammeter used to obtain heater current in one phase was a Simpson 30 amp full scale, iron-vane unit with a basic accuracy of $\pm 5\%$. The voltmeter used to obtain voltage drop across one phase of the boiler heater was a Simpson, 500 volt, iron-vane unit with a basic accuracy of $\pm 5\%$. Both instruments were calibrated in place against instruments having a basic accuracy of $\pm 1/4\%$. The corrections found to be required were ± 0.5 volts and ± 0.05 amps.

4.3.3 Pressure

Test Section

The average pressure in the test section is defined as:

$$P_A = \frac{P_{in} + P_{out}}{2} = P_{in} - \frac{\Delta P}{2}$$

Helicoid, Bourdon tube, 0 to 600 psi and 0 to 100 psi pressure gages, accurate to $\pm 1/4\%$, were used to measure the pressure at the Inlet and outlet of the test section. These gages were used in the appropriate ranges to measure p_{in} and p_{out} . For more accurate measurement of the pressure drop across the test section, a Barton differential pressure gage was used.

The above pressure instrumentation was calibrated against a Heise standard accurate to 0.1% and it was noted that the correction required did not substantiate the manufacturer's claim of $\pm 1/4\%$ accuracy. Maximum corrections of +2 and -7 psi were required for the 0 to 100 and 0 to 600 gages, respectively.

Boiler

The Inlet pressure to the boiler was measured by Helicoid, Bourdon tube pressure gages with 0 to 150 psi and 0 to 800 psi ranges accurate to $1/4\%$. The gages were calibrated against a Heise standard accurate to 0.1% and corrections of +2 and +4 psi were required for the 0 to 150 and 0 to 800 psi gages respectively.

4.3.4 Flow Rate

Flow rate was determined by accumulating mercury on a weigh scale over a measured time increment. The maximum error in weight flow is estimated as $\pm 1.2\%$ on the basis that 20 lb of mercury are weighed to an accuracy of one scale division (1/10 lb) and that an error of 2 sec in approximately 300 sec may occur.

5.0 TEST RESULTS AND DATA REDUCTION

Tests were run to satisfy the general and specific program objectives defined in Section 3. The rig and test section configurations used were described in Section 4. Section 4 also described the instrumentation and operating procedures used to obtain data. An average of three to four hours per point was required to take equilibrium data.

Table 5-1 specifies the major parameters measured directly. This table also shows the corrections required of some of the data to account for calibrations and heat losses. (Heat losses were minimized by guard heaters.)

Heat loss depends on operating temperature levels in a manner too complex to account for during test, so a computer program was utilized to provide corrected values of heat transferred. The computer program was further utilized to reduce the data from the form described by Table 5-1 to the more useful parameters of average quality, $\bar{x}$; average pressure level, $\bar{p}$; total flow rate, $\dot{m}$, and mean heat transfer coefficient, $\bar{h}$. These results are presented in Section 5.1. Section 5.2 describes the data reduction procedures. Section 5.3 discusses the various instrument errors and estimates the accumulated effect thereof on major parameters such as heat flux and heat transfer coefficient. An analysis was also made of rig boiler performance, as discussed in Section 6.3. Data reduction for the theoretical correlations discussed in Section 7 was also performed by computer and the results are presented with the discussion.

TABLE 5-1
PARAMETERS MEASURED DIRECTLY ON SNAP II BOILING RESEARCH RIG

<u>Directly Measured Parameter</u>	<u>Test Result</u>
Hg Mass Collected + Time Taken to Collect Hg →	Hg Flow Rate
Boiler Heater Voltage* + Boiler Heater Current* →	Boiler Heat In**
Test Section Heater Voltage* + Test Section Heater Current* →	Test Section Heat In**
Test Section Inlet Pressure* + Test Section Outlet Pressure* →	Average Test Section Pressure
Test Section Wall Temperature →	T/S Wall Temperature

*Calibrations required.

**Correction required to account for heat losses. Heat losses were minimized by use of guard heaters.

5.1 TEST RESULTS

Tables 5-2 through 5-5 present the reduced test data. Runs 6270 through 6591 of Table 5-5 (SN 3 Test Section) were obtained using a graphite type conducting medium (Thermon) between the tube and heater. All other data were obtained using radiant heating.

As previously explained in the program objectives and as is evident by comparing Table 5-5 with Tables 5-2, 5-3, and 5-4, the parameter variations of the SN 3 data were controlled so that the separate effects of pressure level, flow, and quality on heat flux versus temperature difference could be distinguished. Further study of this data was felt to be appropriate. In particular, variation of test section heat flux and temperature difference with length was investigated.

Variation of temperature difference with length is plotted in Figures 5-1A through 5-1L. Actually, values for the wall temperature were not always available at all thermocouple positions due to thermocouple failures. The readings that were available are marked by a dot. A smooth profile for the temperature difference was drawn so that an integrated mean temperature difference (ΔT_m) for the section as a whole was available for correlation purposes. The variation of ΔT is presented in Table 5-5 according to thermocouple position. These temperature profiles are discussed in Section 6.1.

To give further insight into the phenomena occurring, estimates of variation of heat flux with test section length were made. These are presented in Figures 5-1A through 5-1L. These curves were obtained by proportioning the total test section heat flux according to the heater to tube wall temperature difference variations with length for the conductive tests (Runs 6270 through 6591 and according to the fourth power heater to tube wall temperature difference for the radiation tests (Runs 6020 through 6260).

The heat transfer data presented in Tables 5-2 through 5-5 are correlated in Section 7.2 on the basis of the forced convection vortex boiling theory explained in Section 7. The correlation in data obtained verifies the theory.

Empirical correlations of heat flux versus mean temperature difference for the various test sections are presented in Figures 5-2 through 5-11.

The SN 3 test section data were specifically taken to provide an empirical correlation; that is, heat flux versus tube wall to saturation temperature differences was obtained at carefully selected combinations of pressure level, mass velocity, and average quality. The independent effects of these parameters were then correlated and compared directly with theory in Figures 5-5 through 5-11. Section 6.2 discusses the comparison of the data with the theory which was developed in Section 7.1.

The data for test sections SN 1, SN 2 and B were not as amenable to empirical correlation as were the SN 3 data. Nevertheless, a crude curve was drawn through the points in Figures 5-2, 5-3, and 5-4 for which the flow rate, pressure and quality level were somewhat identical. Section 7.2 makes extensive use of the SN 1, SN 2, B and SN 3 data to evaluate the theoretical heat transfer equations.

TABLE 5-2
TEST RESULTS FOR TEST SECTION SN 1

Run No.	Reduced Data							
	$\dot{m}_T$	G_T	$\bar{h}$	$\bar{x}$	$\bar{p}$	ΔT_M	$\frac{\Delta P}{\Delta L}$	T_{sat}
	$\frac{Lb}{Hr}$	$\frac{Lb}{Sec-Ft^2}$	$\frac{Btu}{Hr-Ft^2-^{\circ}F}$		psia	$^{\circ}F$	$\frac{psi}{Ft}$	$^{\circ}F$
5010	182	37.8	100	0.0475	25.7	68	~ 0	732
5020	190	39.5	79.6	0.200	73.2	88	0.186	863
5040	206	42.8	39.9	0.088	73.9	167	0.353	864
5050	175	36.4	13.9	0.584	153	476	1.86	975
5070	183	38.0	91.4	0.699	161	74	2.68	983
5080	184	38.2	33.6	0.995	161	194	2.68	983
5150	265	55.1	138	0.256	68	71	~ 0	852
5152	264	54.9	125	0.201	70	74	~ 0	857
5160	269	55.9	130	0.416	103	72	~ 0	913
5170	266	55.3	252	0.681	105	38	0.037	916
5201	384	79.8	93.9	0.602	92	227	0.371	896
5202	371	77.1	104	0.0375	38	132	~ 0	776
5210	370	76.9	63.7	0.0525	51	335	~ 0	815
5230	367	76.3	72	0.118	64	188	~ 0	844
5250	362	75.2	127	0.707	69	108	0.595	854

SN 1: 5/8 in. OD with 0.065 in. Wall; 30 in. long, Type 316 SS;
0.054 in. Diameter Swirl wire with 1.8 in. pitch.

TABLE 5-3

TEST RESULTS FOR TEST SECTION SN 2

Run No.	Reduced Data							
	$\dot{m}_T$	G_T	$\bar{h}$	$\bar{x}$	$\bar{p}$	ΔT_M	$\frac{\Delta P}{\Delta L}$	T_{sat}
	Lb Hr	Lb Sec-Ft ²	Btu Hr-Ft ² -°F		psia	°F	psi Ft	°F
2010	175	136	301	0.194	26.9	16.2	3.31	736
2020	153	119	321	0.299	28.8	15.3	3.02	743
2030	172	134	249	0.661	58.3	33.0	7.49	832
2040	174	135	255	0.678	51.5	32.2	7.04	839
2050	168	131	196	0.992	71.7	55.0	7.36	894
2060	181	141	161	0.923	92.4	94.6	6.11	895
2070	222	173	624	0.0436	18.7	11.2	0.487	697
2080	262	204	313	0.345	51.2	37.1	5.71	815
2090	188	146	269	0.211	25.2	40.8	3.16	728
2100	201	156	238	0.368	36.0	46.7	6.32	770
2110	199	155	214	0.407	37.2	57.4	6.65	773
2120	201	156	184	0.416	38.0	82.7	4.78	776
2130	197	153	184	0.517	55.2	47.7	6.89	824
2140	206	160	197	0.522	52.6	56.7	8.11	818
2150	201	156	210	0.560	54.6	66.0	8.27	823
2160	194	151	120	0.852	76.2	68.1	7.78	869
2170	199	155	81.2	0.852	79.7	178	8.92	875
2180	176	137	49.8	0.999	102	153	8.12	908
2190	110	85.6	136	0.482	30.3	32.2	4.70	749
2200	123	95.5	164	0.510	28.8	45.0	4.54	743
2210	110	85.8	169	0.624	32.8	77.1	4.38	758
2220	111	86.4	195	0.151	21.9	19.0	0.649	713
2230	118	91.8	183	0.147	18.4	42.2	0.811	696
2240	113	88.1	224	0.173	18.6	52.3	0.973	697
2250	206	160	166	0.0305	16.1	27.9	0.486	683
2260	203	158	151	0.0392	15.8	60.4	0.632	681
2271	202	157	286	0.0498	16.5	46.5	0.568	685
2272	262	204	82.2	0.0608	17.6	65.6	1.22	692
2280	265	206	259	0.0712	18.1	43.3	1.38	694
2290	264	205	282	0.103	20.3	88.6	2.19	705
4010	53.4	41.5	83.2	0.371	24.2	40.5	1x10 ⁻⁵	725
4020	52.5	40.8	77.7	0.729	25.2	45.6	1.20	730
4040	181	141	104	0.242	58.7	171	0.941	832
4060	97	75.4	348	0.237	23.7	38.5	0.843	723
4070	95.7	74.4	287	0.411	24.2	47.0	1.85	725
4090	96.8	75.2	191	0.736	64.7	70.5	2.21	846
4100	107	83.1	300	0.579	41	43.0	2.43	961
4110	94.4	73.4	37.4	0.990	45.1	333	6.71	800
4120	92.6	72.0	34.4	0.990	88.6	366	2.50	890

SN 2: 5/16 in. OD with 0.028 in. Wall; 30 in. long, Type 316 SS;
0.054 in. Diameter Swirl wire with 1 in. pitch.

TABLE 5-4
TEST RESULTS FOR TEST SECTION B

Run No.	Reduced Data							
	$\dot{m}_T$	G_T	$\bar{h}$	$\bar{x}$	$\bar{p}$	ΔT_m	$\frac{\Delta P}{\Delta L}$	T_{SAT}
	$\frac{Lb}{Hr}$	$\frac{Lb}{Sec-Ft^2}$	$\frac{Btu}{Hr-Ft^2-^{\circ}F}$		Psia	$^{\circ}F$	$\frac{Psi}{Ft}$	$^{\circ}F$
3020	97.1	75.5	9.01	0.150	17.7	155	0.227	692
3031	99	76.9	5.80	0.154	17.6	493	0.260	692
3032	101	78.8	20.0	0.345	32.8	74.5	0.389	761
3042	108	83.87	23.1	0.598	49.3	136	1.01	810
3050	165	128	211	0.0787	16.4	15.1	0.227	684
3060	166	129	263	0.0592	16.8	22.8	0.227	687
3070	168	131	240	0.0445	17.0	36.6	0.0973	688
3080	159	124	232	0.0842	18.5	39.4	0.308	1.00
3120	161	125	314	0.347	26.6	18.8	1.78	736
3130	167	130	395	0.379	29.3	30.4	1.99	147
3150	181	141	24.8	0.591	13.6	329	1.46	956
3171	181	141	267	0.761	152	11.8	2.92	974
3172	185	144	57.6	0.761	159	160	2.85	981
3180	228	177	284	0.0463	22.8	16.5	0.035	719
3200	230	179	352	0.0491	23.8	33.5	0.162	724
3220	230	179	15.6	0.0741	21.4	799	0.292	712
3230	48.8	38.0	373	0.414	25.0	5.12	0.0811	729
3240	53.3	41.4	241	0.387	25.6	11.7	0.714	730
3250	53.3	41.4	163	0.396	25.8	24.4	0.0328	732

B: 5/16 in. OD with 0.028 in. Wall; Bare Tube, 30 in. long, Haynes 25.

HEAT FLUX AND TEMPERATURE DIFFERENCE
DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6020	146	129.05	.1352	5360	102
6041	147.5	146.7	.1251	5880	68
6042	148.5	147.8	.1165	5780	64
6050	153	148.35	.1228	6090	22
6061	149	148.85	.1103	11,000	29
6062	148.7	148.9	.1070	11,000	31
6071	152.3	148.75	.0897	22,200	88
6072	151.7	148.70	.1096	21,800	95

FIGURE 5-1A

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6031	153	148.35	.4779	11,000	26.6
6180	148.5	148.1	.5268	6,250	15.7
6190	150.6	148.1	.5193	11,800	21.5
6200	151	147.95	.5072	16,900	31.5
6210	150.5	148.05	.5001	22,400	43
6220	151	144.95	.4687	31,600	66

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6081	152.5	149.9	.2058	22,400	84
6082	151.3	150.4	.2075	22,400	70
6091	151.7	148.35	.1944	11,200	50
6092	153.5	148.35	.1944	11,400	46
6100	149	147.6	.2041	24,400	82

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6111	152.5	148.65	.1517	23,400	166
6112	151	148.60	.1579	24,200	138
6121	200	149.5	.2822	18,100	569
6122	204	149.55	.2714	18,000	571

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6130	94.3	149.55	.2066	6600	317
6140	93.1	152.05	.1830	3650	117
6151	89.10	149.05	.2015	5100	233
6152	90.0	149	.2053	5170	226
6160	88.5	150.1	.1857	9100	336
6170	88.2	150.35	.2166	8100	23.3

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6230	148	151.55	.7700	5300	20
6240	150	147.55	.7428	10,500	27
6251	151.2	151.60	.7354	13,100	31
6252	154.2	148.7	.7195	13,400	33.2
6260	146.7	149.45	.7329	21,100	51

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6300	145	145	.548	33,100	275

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6311	141.5	146	.7908	22,600	217
6312	151.5	145.9	.7221	21,900	99
6570	147	151	.758	17,700	221

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	P, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6330	154	151.5	.2160	21,500	116
6320	153	150.15	.2053	11,800	36

FIGURE 5-11

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

HEAT FLUX AND TEMPERATURE DIFFERENCE
DISTRIBUTION ALONG TEST SECTION SN 3

Run No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	Q/A , Btu/hr-ft ²	ΔT
6420	241.0	152.35	.2104	16,700	341
6430	237.97	150.10	.2004	26,400	426
6440	239.97	151.80	.2003	8,300	13
6450	240.0	146.85	.1888	12,700	27
6460	241.79	147.75	.1810	15,600	38
6470	240.07	149.4	.2119	21,700	35
6480	238.34	149.55	.2087	28,200	31

FIGURE 5-1K

HEAT FLUX AND TEMPERATURE DIFFERENCE DISTRIBUTION ALONG TEST SECTION SN 3

Kun No.	$\dot{m}$, lb/hr	$\bar{P}$, psia	$\bar{x}$	$Q/A, \text{Btu/hr-ft}^2$	ΔT
6490	150.35	198.9	.4964	5,570	4.35
6500	141.58	196.4	.5496	11,200	11.4
6501	151.20	193.9	.5021	10,200	17.8
6510	149.53	199.9	.4983	16,700	25.0
6520	151.14	194	.5122	27,200	34.4
6530	152.88	192.9	.4973	28,700	60.9
6540	147.47	198.8	.4974	31,700	256.3

APPROXIMATE TEST SECTION SN 1 HEAT FLUX VARIATION WITH TEMPERATURE DIFFERENCE
AT VARIOUS COMBINATIONS OF FLOW RATE, PRESSURE AND QUALITY

FIGURE 5-2

APPROXIMATE TEST SECTION SN 2 HEAT FLUX VARIATION WITH TEMPERATURE

DIFFERENCE, FLOW RATE, PRESSURE AND QUALITY LEVELS

FIGURE 5-3

APPROXIMATE TEST SECTION B HEAT FLUX VARIATION WITH TEMPERATURE DIFFERENCE

AT VARIOUS COMBINATIONS OF FLOW RATE, PRESSURE AND QUALITY

FIGURE 5-4

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

EFFECT OF MASS VELOCITY ON Q/A VS ΔT_m ($p \approx 150$ PSIA; $x \approx 0.20$)

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

EFFECT OF PRESSURE LEVEL ON Q/A VS ΔT_m ($\bar{x} \approx 0.5$; $G_T \approx 116 \text{ LB/SEC-FT}^2$)

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

EFFECT OF QUALITY LEVEL ON Q/A VS ΔT_m ($p \approx 150$ PSIA; $G_T \approx 116$ LB/SEC-FT²)

FIGURE 5-7

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

VARIAION OF Q/A VS ΔT_m ($\bar{x} = 0.1$; $\bar{p} = 150$ PSIA; $G_T = 116$ LB/SEC-FT²)

4.3

FIGURE 5-8

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

VARIAION OF Q/A VS ΔT_m ($\bar{x} = 0.2$; $\bar{p} \approx 150$ PSIA; $G_T \approx 116$ LB/SEC-FT²)

FIGURE 5-7

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

VARIATION OF Q/A VS ΔT_m ($\bar{x} = 0.5$; $\bar{p} \approx 150$ PSIA; $G_T \approx 116$ LB/SEC-FT²)

FIGURE 5-10

COMPARISON OF THEORY WITH SN 3 DATA FOR THE FORCED CONVECTION BOILING OF MERCURY

VARIATION OF Q/A VS ΔT_m ($\bar{x} = 0.75$; $\bar{p} \approx 150$ PSIA; $G_T \approx 116$ LB/SEC-FT²)

FIGURE 5-11

5.2 DATA REDUCTION PROCEDURES

The basic data reduction procedures used to produce the data presented in Tables 5-2 through 5-5 were very straightforward. A computer program was used to make the basic calculations as well as to account for the heat losses that could not be completely accounted for during testing.

The heat input to the test section was computed from the wattage applied to the test section as measured by a voltage and current meter. A correction for heat loss in the test section was required. This calculation was performed using the computer program on the basis that heat loss was purely radial from the test section. As a result, the net heat transferred into the 30 inches of test section was obtained. Heat flux or heat transfer per unit of area was then readily obtained.

The temperature difference ΔT in the test section was taken as the difference between the measured wall temperature and the estimated saturation temperature at the corresponding axial location. The wall temperatures were measured by thermocouples attached to the outside of the test section. The saturation temperature was determined by assuming a linear pressure profile through the test section based on the inlet and outlet pressures.

The heat transfer coefficient in the test section was determined by calculating the total or net heat transfer to the test section and dividing it by the known area in the 30 in. line and the integrated mean temperature difference as determined by temperature profiles within the test section. Figures 5-1A through 5-1L present typical values of these temperature profiles for test section SN 3. Since a curve was required to provide a mean temperature difference, these plots were made on the basis of the available temperatures as shown.

Heat input in the rig boiler was also obtained on the basis of voltage and current as measured, and a heat loss correction was determined by the computer program. The net heat transfer into the boiler in conjunction with the measured flow rate of mercury and the thermodynamic properties of mercury allowed the calculation of the quality entering the test section. This inlet quality in conjunction with the quality change across the test section (as determined from net heat transferred into the test section) allowed a calculation of the average quality in the test section.

5.3 ESTIMATION OF MAXIMUM ERRORS

An exact error analysis that holds for, or is calculated for, each data point would be very time-consuming and usually quite impractical. It is the purpose of the discussion following to identify and bracket the errors of significance and thereby establish the approximate accuracy of data obtained. To simplify the analysis, only the SN 3 test section data were used where needed. These data are conservative relative to the other test sections because temperature levels and therefore heat losses were higher. Furthermore, where a specific SN 3 test data point was required, a conservative point was chosen.

The error analysis is first applied to the calculation of test section heat flow. Since the procedures used neglected end heat losses, these are then estimated. Temperature difference measurement is also investigated as is the expected error in test section quality level. No attempt is made to combine the effect of heat flow rate and mean temperature difference to obtain the accuracy of a heat transfer coefficient calculation.

5.3.1 Errors in Test Section Heat Transfer Rate

An error analysis was performed to give the error in net heat flow received by the test section as computed by the data reduction procedures. The sources of error were (1) the maximum instrumentation errors as determined from calibration (see Section 4.3) and (2) the accuracy of the manufacturer's value of insulation conductivity used in the computer program (see Section 4.2.1). The thermocouple errors as they affect the heat loss calculation were negligible because the thermocouples were from the same spool and application errors were assumed equal. The total error in heat input was found to be the function of heater current:

$$\frac{\Delta Q}{Q} = \frac{0.131}{I} + 0.0462 \quad \text{Eq. 5-1}$$

where I is the current in amps.

Figure 5-12 plots this equation against the measured Q ($Q \propto I^2$).

The error analysis which resulted in Figure 5-12 was necessarily simplified. In particular, it was necessary to assume that the end heat losses through the insulation and tube wall were negligible. The insulation end heat losses could have been calculated on the basis of the simplifying assumption that the heat flowed in the axial direction. For a maximum temperature gradient, the end heat losses were only about 10 Btu/hr. This loss represents only about 2% of the minimum test section heat flow rate and therefore was justifiably neglected.

Another end heat loss that must be considered is the axial heat flow out of the test section via the test section tube walls. The possible error in heat flow was calculated to be negligible. Correspondingly, the error in effective heat transfer area was nil. These conclusions were established by solving the differential equation for the tube wall with a heated and unheated outside boundary and a constant temperature sink inside. Inside the tube, a heat transfer coefficient of about 11 Btu/hr-ft²-°F was assumed for the heated length and about 350 Btu/hr-ft²-°F for the external or unheated length. The situation assumed was representative as well as conservative. The result was that the maximum heat loss was of the order of 4% of the total heat transferred. This loss would be expected to apply at the higher heat fluxes. It was also estimated that 99% of the temperature potential would be lost in only about 0.4 in. and therefore have only a small effect on effective heating area.

In summary then, the maximum error in heat flow due to end heat losses could be greater than that of Figure 5-12 by 2% at low heat flow rates and 4% at high heat flow rates. On an overall basis, 10% maximum error could not be easily disputed. Since the end heat loss errors are too difficult to account for in each test, the error curve of Figure 5-12 is suggested as representative of the variation of error in heat flow.

5.3.2 Temperature Difference Errors

The temperature potential for heat transfer in the test section was the wall minus the fluid saturation temperature. The accuracy of this measurement was established in several steps:

LIMIT OF ERROR IN TEST SECTION Q/A AS A FUNCTION OF Q/A

FIGURE 5-12

1. A calibration was made of the measured wall versus saturation temperature as determined by pressure for saturated mercury.
2. The application errors for the surface thermocouples were estimated. Included was the effect of heat transferred to the thermocouple junction from the heat source.
3. Thermocouple and instrumentation errors were determined.

These steps are discussed below.

Correlation of Saturation, Wall and Stream Temperatures

As noted before, the variable $t_{\text{wall}} - t_{\text{fluid}}$ is determined by subtracting the saturation temperature of the mercury vapor, corresponding to the measured pressure, from the measured wall temperature. Outside of the test section, or within the test section if the heaters are disconnected, a heat loss to the ambient will occur thereby maintaining the tube wall and stream thermocouples at the saturation temperature of the mercury by condensation on these surfaces. Thus it is possible to check the measured wall and stream temperature against the saturation temperature of the mercury at the measured pressure. Figures 5-13 and 5-14 compare the inlet and exit stream thermocouple indications with the saturation temperature of the mercury at the measured pressure. Maximum differences are in the order of $\pm 20^\circ\text{F}$ in the region 700° to 900°F .

Inspection of the data for tube-wall thermocouples outside of the test section, No. 2 and 3 in Figure 4-4, indicate the same variability as the stream temperature. Thus, the limit of error for temperature difference is approximately $\pm 20^\circ\text{F}$. This value is consistent with the combined errors expected in measuring wall temperatures and saturation temperatures (as determined from pressure.)

Lead Conduction

The ceramic divider utilized in applying the test section surface thermocouples (see Figure 4-5) also minimized the heat gain to the thermocouple junction from lead conduction. A calculation was made to determine the temperature difference between the wire and the tube at the thermocouple junction. Conservative values of temperature were assumed for the case of the wire acting as a fin with the ceramic tube as an environment. The error due to lead conduction was negligible.

Radiation Transfer to the Tube

The ceramic divider was also analyzed to determine its effect on the immediate surface temperature (see Figure 4-5). The divider was calculated to act as a fin causing a maximum error of $+7^\circ\text{F}$ in measurement of surface temperature which corresponded to 11% of the measured ΔT for Run 6220. Run 6220 was chosen because the fin effect of the ceramic divider was maximum.

COMPARISON OF TEST SECTION INLET MEASURED
TEMPERATURE AND SATURATION TEMPERATURE

FIGURE 5-13

COMPARISON OF TEST SECTION EXIT MEASURED
TEMPERATURE AND SATURATION TEMPERATURE

FIGURE 5-14

Conduction Transfer to Tube

For high heat flux tests it was necessary to transfer heat to the tube by solid conduction rather than radiation in order to avoid excessive heater temperatures. A heat transfer cement, "Thermon," was used to fill the space between tube and heater.

To establish the application error of the thermocouples with the "Thermon" fill, the temperature of the tube surface under the ceramic, assuming the ceramic to be "smeared" over the tube circumference, was compared to the bare tube surface temperature. Using the data for Run 6410 of the SN 3 tests, a 3°F error or 7.7% of total tube wall-to-saturation ΔT was calculated. This error is less than that calculated for radiation heat transfer due to the lateral conduction of the isotropic "Thermon" fill. As was the case for radiation heat transfer to the test section, the relative error of the measurement increases for increasing inside coefficients. Also, in both cases, the magnitude of the temperature error is considerably less than the permissible error for temperature measurement.

Thermocouple Error and Instrumentation Error

The last source of error to be considered is the thermocouple calibration tolerance, drift and instrumentation error.

The Chromel P-Alumel wire used in the test section thermocouples carried the standard tolerance of $\pm 4^{\circ}\text{F}$ to 530°F and $\pm 3/4\%$ above 530°F . In the region of tube wall temperature (1200°F), the error in thermocouple calibration could be as large as $\pm 9^{\circ}\text{F}$. The maximum error in the Minimate indicator is $1/4\%$ of full scale or $\pm 6^{\circ}\text{F}$ in the region of wall temperature. The maximum error in measuring wall temperature consequently could be as large as $\pm 15^{\circ}\text{F}$.

The drift in thermocouple calibration at tube wall temperature was studied by reference to the manufacturer's catalogs as well as an ORNL report on base-metal thermocouples (Ref. 5-1). Both sources indicate negligible drift will occur at 1200°F . However, there is ample evidence that a serious drift in calibration of Chromel will occur if chromium carbide is formed. To minimize carburization of the Chromel wire in tests in which Thermon was used as a heat transfer medium between heater and tube, all bare thermocouple leads were coated with Sauereisen insulating cement.

5.3.3 Summary of Wall to Saturated Temperature Difference Errors

Based on the above discussion, the maximum temperature difference error when heat is transferred to the tube by radiation could be $\pm 27^{\circ}\text{F}$. For conduction heat transfer, $\pm 23^{\circ}\text{F}$ is the maximum temperature difference error. These values are the result of adding the saturated-wall temperature correlation ($\pm 20^{\circ}\text{F}$) and the radiation ($\pm 7^{\circ}\text{F}$) and conduction ($\pm 3^{\circ}\text{F}$) application errors.

5.3.4 Error in Test Section Quality Level

An important parameter in the analysis of the test section data is the rig boiler exit quality or test section inlet quality, for the quality level in the test section is thus determined. To determine the accuracy of this quantity, the effects of errors in calculating the rig boiler heat flow and flow rate were considered.

The net boiler heat input error due to power measuring instrument errors (as determined by calibration) and heat losses not accounted for by the computer program were found to be a function of current.

Incorporating the effect of a $\pm 1.2\%$ error in the flow rate (see Section 4.1) results in the following equation:

$$X_{B \text{ exit}} = \frac{Q}{\dot{m} h_{fV}} \left[\frac{2.09}{\sqrt{Q}} + 0.0464 \right] \quad \text{Eq. 5-2}$$

where Q , the heat input into the boiler, is in watts.

The maximum error in quality is plotted in Figure 5-15 for several mercury flow rates and test section pressure levels. As shown in Figure 5-15, the maximum deviation in test section quality level as discussed in this report will be ± 0.02 at 0.10 quality and ± 0.07 at 0.75 quality.

LIMIT OF ERROR IN QUALITY AS A FUNCTION OF QUALITY

FIGURE 5-15

6.0 DISCUSSION OF TEST RESULTS

The data obtained have been presented in the following forms:

1. Tables 5-2, 5-3, 5-4, and 5-5 present the flow rate, average pressure, average quality, mean temperature difference, and heat flux for all test sections.
2. Figures 5-1A through 5-1L show the variation of temperature difference with test section length for most of the SN 3 test data.
3. Figures 5-1A through 5-1L also show the estimated variation of heat flux with length based on plots of the corresponding variation of temperature difference.
4. Figures 5-2 through 5-11 present plots of total test section heat flux versus mean temperature difference.

This section discusses these results with emphasis on the temperature difference variation and heat flux plots. Further discussion of the test results is presented in Sections 7.2 and 7.3 where the heat transfer and pressure drop data are correlated according to the forced convection vortex mercury drop boiling theory developed therein. Section 6.3 discusses an analysis of the rig boiler performed during this program.

6.1 TEST SECTION TEMPERATURE DIFFERENCE PROFILES

To provide a suitable mean temperature difference for the heat transfer occurring in the test section, variation of ΔT with length was necessary. This variation was provided by surface thermocouples and by saturation temperatures based on the pressures in and out (see Figure 4-4). The variations in temperature difference for the SN 3 test section are presented in Figures 5-1A through 5-1L. Runs 6020 through 6270 used radiation heat transfer while the other runs were accomplished using Therman (see Section 4.2.1 for elaboration). The radiation-heated runs may be considered representative of the SN 1, SN 2, and B test section results.

The "hilly" appearance of the temperature profile in Figures 5-1A through 5-1L is attributed to the oscillatory nature of the mercury drops in a direction normal to the heating surface (see Appendix A for analysis). When the mixture of mercury liquid and vapor is passed from the adiabatic section (between the rig boiler and local test section) into the heated tube, a great deal of vapor is suddenly generated under the liquid drops. The repulsive force produced by the vapor ejects the liquid away from the heated surface, thereby raising the wall temperature. The liquid eventually returns to the wall as a result of a decrease in the vapor repulsive force and the process is repeated.

During the film pool boiling of a liquid, the waves generated on the liquid-vapor interface experience a similar action. Vapor is formed in bursts when the liquid approaches the heated surface. A slug of vapor is generated with explosive suddenness, and this forces the liquid away from the heated surface.

In a forced convection film boiling system these liquid oscillations manifest themselves as temperature profiles in the direction of flow, whereas in a non-flowing system (pool boiling) the temperature profiles appear as a function of time. It might be conjectured that these temperature profiles could provide local-local data, that is, local heat transfer coefficients along the test section which in itself should provide over-all localized data. This would require an assumption of uniform heat flux which should be plausible for an electrically heated test section. However, the fact that the conditions are not monotonically changing along the tube when undergoing these oscillations negates the value of such a correlation. Since the oscillations of ΔT occur about some average ΔT , a mean heat transfer coefficient computed from the mean temperature difference has some significance because it is expected that this would more nearly represent the heat transfer coefficient after the oscillations are damped out and the conditions relevant to the heat transfer rate begin to change in monotonous fashion. It should be kept in mind that the dynamics of this oscillatory phenomenon will probably be different if the heating surface is isothermal instead of one of constant heat flux.

The dynamic analysis of a mercury drop in a heated surface presented in Appendix A can be extended to give the spatial temperature profile resulting if a velocity parallel to the wall is imparted to the liquid drop. This is done in Appendix E. The analysis is restricted to small amplitude oscillations and small disturbing forces and can hardly be expected to apply accurately to the case wherein a sudden burst of vapor is generated when the liquid passes from an adiabatic surface onto a heated surface. However, the analysis might reveal the pertinent variables. Equation E-3 in Appendix E gives the wave length of the temperature profiles, and Equation E-4 is the same equation with all the constants of test section SN 3 substituted.

Figure 6-1 shows some of the measured wave lengths plotted versus the variable of Equation E-4. The data can be correlated reasonably well on a negative 45 degree slope on a log-log paper as shown in Figure 6-1 and suggested by Equation E-4. The absolute value of the correlation, however, in no way agrees with that predicted by Equation E-4. This could be attributed to the assumption of small oscillations in the analysis.

6.2 EMPIRICAL HEAT FLUX CORRELATIONS

Figures 5-2 through 5-11 present the over-all heat flux versus mean temperature difference for the SN 3 test section data. The curves are grouped so that the separate effects of pressure level, quality level, and mass velocity may be investigated. Ninety-five per cent of the points obtained, including both Thermon and radiation heat transfer test data, are plotted.

MEASURED WAVE LENGTHS OF TEST SECTION SN 3 TEMPERATURE

DIFFERENCE PROFILE

FIGURE 6-1

The shortage of points made it difficult to correlate the data for a particular combination of mass velocity, pressure and quality level. To give insight into what the correlation might be, curves are drawn based on the theory for the forced convection vortex boiling of mercury drop developed in Section 7. In particular, Equations 7-16, 7-17 and 7-25 were used to predict the contact, intermittent contact and film heat flux versus temperature difference relationships respectively. From Figures 5-8 through 5-11, an appreciation for the accuracy to be expected from these equations may be obtained. The presently known extremes for each boiling regime are given. In Figures 5-5 through 5-7, the equations are plotted to give a nominal predicted value for comparison with the data. In plotting Equations 7-16, 7-17 and 7-25, the boiler rig wire geometry (0.062 in. wire diameter and 1 in. pitch) was used because this most probably determined the drop size entering the test section (Section 7.1.2 describes the effect of this geometry). In any event, the curves drawn provide a reference as to the possible correlation of the data.

The various forced convection drop boiling regimes are similar to the more common pool boiling regimes and are compared in Table 6-1. This similarity is further apparent when the typical heat flux curves for surface boiling, mercury drop evaporation, and water drop evaporation presented in Figures 6-2, 6-3 and 6-4, respectively, are compared.

TABLE 6-1

COMPARISON BETWEEN POOL TYPE BOILING AND DROP BOILING REGIMES

<u>POOL BOILING</u>	<u>DROP BOILING</u>	<u>SIMILARITY</u>
Nucleate	Contact	Liquid In Contact
Transitional or Partial Film	Intermittent Film - Contact	Liquid in Partial Contact
Film	Film	No Liquid Contact

The phenomena of surface pool boiling exhibit three separate regimes as shown in Figure 6-2. These modes are nucleate, transitional, and film boiling. Nucleate boiling (AB) is characterized by the generation of vapor bubbles at selective locations on the surface; the remainder of the surface remains in contact with the liquid. The partial film boiling (BC) is characterized by the existence of an unstable vapor blanket that releases great patches of vapor at more or less regular frequencies. At (C) a stable vapor film covers the entire surface and film boiling exists.

6.3 RIG BOILER PERFORMANCE

The rig boiler supplied the local test section with mercury at various pressures, flow rates, and qualities. Boiler tube dimensions were 3/8 in. OD by 0.035 in. wall thickness by 114.78 ft. A swirl device running along the entire tube length had a pitch of 0.88 in. and a wire diameter of 0.0625 in. The boiler was heated electrically and the heat losses were reduced to a minimum by means of guard heaters. Thirty thermocouples were installed

TYPICAL SURFACE BOILING CURVE

q VS ΔT FOR MERCURY DROPS ON A HOT PLATE IN AIR
(REF 1-1)

FIGURE 6-3

HEAT TRANSFER TO WATER DROPS ON A HOT PLATE OPEN TO ATMOSPHERE (REF 6-1)

FIGURE 6-4

along the outer tube surface. These thermocouples not only measured the tube wall temperature profile but also indicated the position of the liquid leg in the tube, thereby allowing one to determine the boiling and preheat length. The liquid leg position in the tube corresponded to the point where the surface temperature was a maximum, indicating the completion of preheating.

Figure 6-5 shows typical boiler tube temperatures as a function of length. Tube surface temperatures increase as the mercury is preheated to the saturation temperature. At saturation temperature a maximum value is reached after which the temperatures decrease as a result of the decrease in saturation temperature with pressure drop. In some cases the wall temperatures may again increase when the superheat region is approached. The saturation temperature where boiling begins as computed from the inlet pressure and liquid leg hydrostatic head is compared with the measured maximum temperature in Figure 6-6.

The rig boiler was sufficiently instrumented to provide heat transfer data during its operation in conjunction with the test section. The computed heat transfer coefficient as determined from the boiling length and the mean temperature difference represents a mean over a quality range of 0 to X_E (boiler exit quality). Table 6-2 is a tabulation of the results.

An evaluation of the data provided a better understanding of the heat transfer mechanisms involved during the establishment of flow in vortex boiling (see Figure 7-9). The boiling length required to break up the liquid and induce a swirling mixture is of practical interest to the boiler designer. To analytically synthesize such phenomena, three parameters were devised, namely:

$$\Gamma_{vf} \equiv \frac{h(\text{MEASURED})}{h(\text{VORTEX FILM BOILING})} \quad \text{Eq. 6-1}$$

where $h(\text{measured})$ is the mean heat transfer coefficient as measured in the rig boiler between $X = 0$ and $X = X_E$, and $h(\text{vortex film boiling})$ is computed by means of Equation 7-17 (film boiling coefficient integrated from 0 to X_E) or

$$\Gamma_{vf} = \frac{(h_m D / k_v)_{\text{MEASURED}}}{\frac{G^{1/3} D}{4 \delta_o} \theta^{4/3} \left(\tan \alpha \frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \frac{\alpha_E}{\int_0^{\alpha_E} \frac{\alpha^{1/3} d\alpha}{(1-\alpha)^{4/3}}} } \quad \text{Eq. 6-2}$$

$$\Gamma_{vc} \equiv \frac{h(\text{MEASURED})}{h(\text{VORTEX CONTACT BOILING})} \quad \text{Eq. 6-3}$$

where $h(\text{measured})$ is the mean heat transfer coefficient as measured in the rig boiler and $h(\text{vortex contact boiling})$ is computed by means of Equation 7-16 (contact boiling coefficient integrated from 0 to X_E) or

$$\Gamma_{vc} = \frac{(h_m D / k_v)_{\text{MEASURED}}}{\frac{Nu_b^2}{32} \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{\delta_o} \right)^2 \frac{\rho_v}{\rho_f} \frac{\alpha_E}{\int_0^{\alpha_E} \frac{\alpha d\alpha}{(1-\alpha)^{4/3}}} } \quad \text{Eq. 6-3}$$

TYPICAL SURFACE TEMPERATURE VERSUS LENGTH IN TEST RIG
ONCE-THROUGH FORCED CONVECTION BOILER WITH SWIRL DEVICE

Thermocouple Location Along Boiler Length	
Thermocouple Number	Boiler Length, Ft.
1	0.1
2	0.9
3	3.2
4	5.3
5	7.4
6	9.5
7	11.6
8	17.8
9	24.1
10	30.4
11	36.7
12	43.0
13	49.2
14	55.5
15	61.8
16	68.1
17	74.4
18	78.6
19	82.7
20	86.9
21	91.1
22	95.3
23	99.5
24	103.7
25	105.8
26	109.9
27	110
28	112
29	113.6
30	114.7

TYPICAL SURFACE TEMPERATURE VERSUS LENGTH IN TEST RIG
ONCE-THROUGH FORCED CONVECTION BOILER WITH SWIRL DEVICE

Thermocouple Location Along Boiler Length	
Thermocouple Number	Boiler Length, Ft.
1	0.1
2	0.9
3	3.2
4	5.3
5	7.4
6	9.5
7	11.6
8	17.8
9	24.1
10	30.4
11	36.7
12	43.0
13	49.2
14	55.5
15	61.8
16	68.1
17	74.4
18	78.6
19	82.7
20	86.9
21	91.1
22	95.3
23	99.5
24	103.7
25	105.8
26	109.9
27	110
28	112
29	113.6
30	114.7

SNAP II BOILING RESEARCH RIG BOILER COMPARISON BETWEEN SATURATED
LIQUID TEMPERATURES BASED ON ESTIMATED AND MEASURED PRESSURES
RESPECTIVELY. CORRECTION MADE FOR LIQUID LEG HYDROSTATIC HEAD.

FIGURE 6-6

TABLE 6-2
RIG BOILER TEST DATA

Run No.	Test Results						Calculating Parameters		Correlating Parameters			
	P, psia	$\rho_v, \text{lb/ft}^3$	$\frac{D G_T}{\mu_v}$	$\Delta T_m, ^\circ\text{F}$	x_E	$\left(\frac{h_m D}{k_v}\right) \text{ MEAS.}$	$\frac{x_E}{\int_0^1 \frac{x_E}{(1-x)^{1/2}} dx}$	$\frac{x_E}{\int_0^1 \frac{x_E}{(1-x)^{1/2}} dx}$	Γ_{FILM}	Γ_{CONTACT}	Γ_{POOL}	$\left(\frac{D G_T}{\mu_v}\right) \left(\frac{x_E}{\rho_v^{1/4}}\right)$
3020	23.02	0.370	30,756	14.9	0.141	100.9	12	2.35	0.123	0.0278	1.09	7,130
3031	22.77	0.365	31,344	12.0	0.136	123.3	12.1	2.40	0.127	0.0346	1.14	7,060
3032	45.32	0.674	29,010	18.3	0.336	132.51	5.45	1.62	0.1845	0.0505	2.5	11,900
3041	52.47	0.769	25,938	18.3	0.356	126.78	5.10	1.58	0.179	0.0468	2.24	10,500
3042	76.88	1.08	28,306	17.2	0.58	205.74	3.0	1.17	0.28	0.1003	5.38	15,800
3050	20.1	0.323	53,305	3.4	0.0682	660.84	21.5	2.90	0.184	0.113	3.89	6,400
3060	20.3	0.325	53,666	6.7	0.0369	160.75	34.50	3.30	0.0613	0.0172	0.946	3,480
3070	20.1	0.323	54,365	13.8	0.027	57.08	44.0	3.47	0.0333	0.0048	0.336	2,590
3100	22.2	0.353	51,204	4.0	0.0411	392.0	32.0	3.20	0.1064	0.0423	2.3	3,540
3120	50.2	0.738	45,096	21.2	0.322	166.94	5.70	1.65	0.1763	0.0571	2.95	16,900
3130	53.8	0.787	46,418	21.5	0.332	175.21	5.5	1.64	0.177	0.059	3.06	17,400
3150	172.9	2.23	41,457	5.4	0.562	945.93	3.1	1.19	0.277	0.267	5.78	15,700
3171	198.4	2.52	40,664	10.3	0.744	650.04	2.12	0.93	0.349	0.246	8.9	19,000
3180	26.4	0.410	69,605	9.2	0.0341	195.12	37.0	3.33	0.0652	0.016	1.15	3,710
3200	27.2	0.422	70,320	7.1	0.0184	129.93	58.5	3.70	0.0321	0.00674	0.765	2,000
3220	27.5	0.425	70,058	5.6	0.0415	309.49	32.0	3.20	0.075	0.0293	1.82	4,470
3240	31.25	0.482	16,120	17.4	0.354	80.60	5.1	1.58	0.2065	0.0425	2.19	8,240
3250	31.65	0.488	16,113	22.6	0.351	61.24	5.3	1.60	0.1831	0.031	1.9	8,100
4010	30.98	0.479	16,004	25.6	0.333	53.45	5.5	1.65	0.273	0.0263	1.7	7,700
4020	41.18	0.618	15,012	29.3	0.684	70.74	2.4	1.02	0.352	0.0656	6.21	13,100
4040	78.03	1.10	46,915	21.12	0.180	111.47	9.6	2.15	0.0672	0.0173	0.656	8,050
4060	32.78	0.503	28,813	26.5	0.151	43.98	11.2	2.30	0.0678	0.0102	0.532	6,150
4070	41.08	0.617	27,431	29.7	0.321	75.22	5.7	1.66	0.156	0.0294	1.95	11,200
4090	91.77	1.27	24,323	23.4	0.648	135.6	2.6	1.07	0.2496	0.0683	4.53	14,000
4100	163.02	2.12	24,438	16.2	0.499	152.9	3.6	1.30	0.1255	0.0385	1.52	8,340
4110	93.5	1.29	23,724	86.8	1.093	56.79	1.11	0.38	0.699	0.066	22,700	19,000
4120	123.3	1.65	22,173	76.8	1.10	60.44	1.11	0.38	0.608	0.059	19,000	6,500
5020	81.0	1.13	45,250	7.6	0.152	371.4	11.0	2.27	0.107	0.0477	2.18	2,100
5040	84.04	1.17	48,763	9.4	0.0464	72.92	29.0	3.15	0.01625	0.00348	0.428	12,800
5050	181.7	2.33	36,546	12.2	0.535	369.5	3.3	1.25	0.1873	0.0952	3.16	15,100
5070	203.8	2.58	37,384	11.4	0.651	441.9	2.55	1.07	0.229	0.137	4.72	21,800
5080	209.3	2.64	37,501	12.1	0.948	612.7	1.33	0.56	0.631	0.357	20.4	12,100
5150	85.02	1.18	62,667	11.9	0.21	327.1	8.5	2.0	0.113	0.0527	1.93	12,100
5152	87.57	1.22	62,198	11.6	0.156	239.2	10.8	2.25	0.0712	0.0298	1.4	8,810
5160	143.5	1.89	58,273	19.0	0.371	278.9	4.95	1.55	0.1293	0.0555	2.27	15,800
5170	171.2	2.21	56,011	31.3	0.635	939.3	2.65	1.080	0.804	0.31	20.8	23,800
5190	202.9	2.57	55,643	30.2	0.802	314.09	1.90	0.84	0.306	0.132	9.7	27,700
5201	182.8	2.34	79,987	41.9	0.531	224.44	3.38	1.24	0.154	0.0559	3.94	27,800
5230	73.8	1.04	91,253	4.6	0.0706	731.25	21.0	2.9	0.0783	0.0523	4.3	6,320
5246	120.3	1.61	80,953	23.1	0.326	299.16	5.7	1.65	0.1328	0.0576	2.67	20,800
5250	178.6	2.30	75,850	57.2	0.657	199.64	2.55	1.06	0.2065	0.0665	6.53	32,800

Rig Boiler: 3/8 in. OD with 0.035 in. Wall; 115 feet, Type 316 SS;
0.062 in. Swirl Wire diameter, 1 in. pitch.

$$\Gamma_p \equiv \frac{h_{\text{(MEASURED)}}}{h_{\text{(POOL BOILING)}}} \quad \text{Eq. 6-4}$$

where $h_{\text{(measured)}}$ is the mean heat transfer coefficient as measured in the rig boiler and $h_{\text{(pool boiling)}}$ is computed by means of Equation G-12 or G-14 in Appendix G; thus,

$$\Gamma_p = \frac{(h_m D / k_v)_{\text{MEASURED}}}{\left(606 \frac{D}{k_v}\right)^2 \frac{1}{\Delta T^{\circ F}} \frac{\rho_v}{\rho_f} \frac{1}{\pi 12} \frac{\alpha_e}{\int_0^{\alpha_e} \frac{\alpha_e}{1-\alpha} d\alpha}} \quad \text{Eq. 6-5}$$

When ΔT approaches zero, the resistance of the vapor film becomes negligible and the resistance of the liquid mercury becomes important. Under these conditions the denominator of Equation 6-5 should be made equal to $2/3 \frac{1}{\pi} \frac{k_e}{k_v}$ (see Equation G-14, Appendix G). This quantity represents the upper limit of the Nusselt number for forced convection pool boiling in a 1 g environment.

Table 6-2 lists values of the above parameter ($\Gamma_v, \Gamma_c, \Gamma_p$). Note that the upper limit of the computed pool boiling Nusselt number is 170 as determined by Equation G-14. The results have been plotted in Figures 6-7, 6-8, and 6-9. The abscissa, $\frac{DG_T}{\mu_v} \frac{\alpha_e}{\rho_v^{1/2}}$ represents a quantity proportional to $\rho_v u_{ve}^2$. This quantity was selected because according to the literature the liquid breakup is a function of the Weber number.* It should be noted that when $\Gamma_p > 1$, Γ_c and Γ_v approach 1; conversely, when Γ_c and $\Gamma_v < 1$, Γ_p approaches 1.0. When Γ_c and $\Gamma_v < 1$ and $\Gamma_p > 1$, the boiling between $X = 0$ and $X = X_E$ is partially in vortex boiling and partially in forced convection pool boiling. When $\Gamma_p = 1$, the boiling regime over the entire boiling length is in forced convection pool boiling. At this point according to Figure 6-9,

$$\frac{DG_T}{\mu_v} \frac{\alpha_e}{\rho_v^{1/2}} = 5500 \left(\frac{\text{ft}^3}{\text{lbm}} \right)^{1/2} \quad \left(\text{Note: } \frac{DG_T}{\mu_v} \frac{\alpha_e}{\rho_v^{1/2}} = \frac{D}{\mu_v} \rho_v^{1/2} u_{ve} \right)$$

For $D = 0.306 \text{ in.}$ in (rig boiler)

$$\mu_v = 0.55 \times 10^{-4} \text{ lb/ft-sec.}$$

We find that this results in $\rho_v u_{ve}^2 = 142 \text{ lb/ft-sec}^2$ which must be the critical value for liquid breakup into vortex flow. This value can be compared with results from the mercury-nitrogen adiabatic flow tests.

Figure 6-10 is a flow regime map based on mercury-nitrogen adiabatic two-phase flow tests reported in Reference 7-19. In these tests the average nitrogen density was about 0.11 lbm/ft^3 . From Figure 6-10 it may be seen that the critical velocity required to convert "caterpillar" flow to "drop" flow was about 36 ft/sec . A value for $\rho_v u^2$ of 143 lb/ft-sec^2 is thus obtained as the critical value for liquid breakup for the mercury-nitrogen tests and

* Weber number = $\frac{\rho u^2}{\sigma}$

Γ_{vf} AS DEFINED BY EQUATION 6-2 PLOTTED
AGAINST QUANTITY PROPORTIONAL TO $e_v u_{ve}^2$

FIGURE 6-7

Γ_{vc} AS DEFINED BY EQUATION 6-3 PLOTTED
AGAINST QUANTITY PROPORTIONAL TO $\rho_v u_{vE}^2$

Γ_p AS DEFINED BY EQUATION 6-5 PLOTTED
AGAINST QUANTITY PROPORTIONAL TO $\rho_v u_{vE}^2$

FIGURE 6-9

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

FIGURE 6-10

compares favorably with the rig boiler performance herein reported. It is then concluded that in the design of mercury boilers, the quantity

$$\rho_v u_{ve}^2 = 142 \frac{\text{lb}}{\text{ft}^2 \cdot \text{sec}^2} \quad \text{Eq. 6-6}$$

can be considered a critical value below which a forced convection pool boiling Nusselt number should be applied (Equations G-12 and G-14), and above which the vortex boiling Nusselt numbers should be used.

7.0 CORRELATION OF DATA ACCORDING TO FORCED CONVECTION VORTEX BOILING THEORY

Practically all possible two-phase flow regimes, as illustrated in Figure 7-1, are present in a forced convection boiling system. As flow progresses along the heated pipe, i.e., as the quality increases, the flow regime changes from bubble flow to annular flow and finally to dropwise flow. In the first five regimes shown in Figure 7-1, liquid appears as the continuous phase in contact with the heating surface. As the quality increases, the vapor velocity in the annular flow regime increases, generating a capillary wave at the interface. At a critical value of the vapor velocity these capillary waves become unstable and break up into liquid drops. As a result, the liquid film is destroyed by breakup and by evaporation and the heated wall is exposed to the vapor which in turn becomes the continuous phase in the dropwise flow regime. This change is accompanied by a decrease of the heat transfer coefficient and by a marked increase of the wall temperature if a heat flux limited heat source is used. Pressure oscillations and flow instabilities can also cause the formation of large vapor patches in contact with the heating surface and the subsequent "burnout."

In the bubble, slug, and annular flow regimes, energy is transferred from the wall through the liquid to the vapor. Because the motion of the liquid is different for different flow regimes, a characteristic heat transfer coefficient corresponds to each regime. At present, the demarcation between the different flow regimes is not well established.

The essential data required to design heat transfer equipment utilizing this phenomenon are

1. heat transfer coefficient, pressure drop, and inventory in the regimes where the liquid is in the continuous phase in contact with the heating surface (bubble, slug, and annular flow regimes),
2. conditions leading to the destruction of the liquid film adjacent to the heating surface, i.e., the change from annular to dropwise flow, and
3. heat transfer coefficient, pressure drop, and inventory in the dropwise flow regime.

Enough published data and correlations are available to estimate the boiling convective heat transfer coefficients for the case when a liquid film is in contact with the heating surface. The superposition method proposed by Rohsenow and Clark (Ref 7-1) is usually used to compute the forced convection boiling heat transfer coefficient when liquid is in the continuous phase. This method is an attempt to simplify the phenomenon by considering the total heat flux to be composed of two components: one a "boiling flux" and the other a "convective flux". By superposing the two effects, one can use existing heat transfer equations to predict the heat transfer characteristics of a forced convection boiling system as long as a stable liquid film remains in contact with the heating surface. Rohsenow found good experimental substantiation of the superposition technique. The "boiling flux" is usually computed from the many available "pool boiling" correlations.

FLOW REGIMES IN FORCED CONVECTION BOILING (WITH NO EXTERNAL TRANSVERSE FORCE FIELD)

- (1) Small vapor bubbles dispersed in liquid phase (liquid phase continuous) - bubble flow
- (2) Large vapor bubbles dispersed in liquid phase (liquid phase continuous) - slug flow
- (3) Smooth liquid film on heating surface (both phases continuous) - annular flow
- (4) Wavy liquid - liquid interface (both phases continuous) - annular flow
- (5) Liquid film breakup (liquid phase becoming discontinuous) - misty annular flow
- (6) Liquid droplets dispersed in vapor phase (vapor phase continuous) - dropwise flow

FIGURE 7-1

Among the more frequently used correlations are correlations of Rohsenow (Ref 7-2), Forster and Zuber (Ref 7-3), Kutateladze (Ref 7-4), Kruzhilin (Ref 7-5), and Labountzov (Ref 7-6). The importance of nucleating characteristics in boiling has been shown by Courty and Foust (Ref 7-7), Kurihara and Meyers (Ref 7-9), and Berenson (Ref 7-9). As discussed by Griffith and Wallis (Ref 7-10) and Zuber and Tribus (Ref 7-11), no correlation of nucleate boiling can be general unless it takes into account the nucleating characteristics of the surface. The film boiling region in pool boiling has been investigated by Bromley (Ref 7-12) and Berenson (Ref 7-9), who also derived equations which permit the prediction of the heat transfer rates.

An equation for predicting low quality boiling heat transfer in forced convection has also been proposed by Stermann (Ref 7-13, 7-14). This equation can be used for predicting heat transfer rates to water and to various organic liquids. More recently, an equation was published by Labountov (Ref 7-15) which predicts boiling heat transfer to liquids at saturation or to liquids subcooled in forced convection. This equation is also used for liquid metals such as mercury and sodium after modifying for the low Prandtl number effect. The equation contains characteristic lengths not of the tube dimensions but of the bubbles, i.e., the bubble Nusselt and Reynolds number as is formulated by Zuber (Ref 7-16). The dynamic similarity for different boiling liquids is thereby maintained.

Hydrodynamic stability, the physics of flow pattern formation, analytically treats the instabilities of the interface between the liquid and vapor phases. However, the mathematical analyses are usually profound, sometimes requiring oversimplification which can make the results impractical. Experimentation is the most fruitful method for obtaining data on two-phase flow patterns in most cases.

Any comprehensive treatment on hydrodynamic stability of a liquid-vapor interface will contain all of the usual dimensionless flow parameters and the ratios of the properties of the two phases, namely:

$$S = f(R_e, W_e, Fr, \frac{\rho_v}{\rho_l}, \frac{\mu_v}{\mu_l}, \alpha, R_1, R_2) \quad \text{Eq. 7-1}$$

where S is a dimensionless wave amplitude growth-rate number; R_e, W_e, Fr are Reynolds, Weber, and Froude numbers respectively; $\rho_v/\rho_l, \mu_v/\mu_l$ and α are vapor-liquid density, viscosity ratio, and quality respectively; and R_1, R_2 are geometric ratios required for geometric similarity ratios of drop size or film thickness to tube diameter and insert geometry (springs and twisted ribbons).

When $S \leq 0$ the wave patterns are stable. Most experimental correlations do not contain all of the dimensionless parameters shown in Equation 7-1, but may give one or two of the more important ones. Many references are available in the technical literature which treat both the analytical and experimental aspects of this subject and the reader is referred to these for details (Ref 7-17 and 7-18).

In the forced convection boiling of pure mercury, the dropwise flow regime extends over

a major portion of the boiling length. The regimes in which the liquid phase is continuous cover an insignificant portion of the heating surface. Early film breakup is caused by the destabilizing effect on pure mercury films of high surface tension, large ratio of liquid-to-vapor density, and non-wetting. The exact point of film breakup is not predictable. Experiments show that most of the heat is transferred to dropwise flow (Ref 1-1), however, so prediction of the point of film breakup and inventory considerations are of little importance in predicting the heat transfer coefficients in dropwise flow.

To design forced convection boilers for pure mercury, adequate correlations are required for dropwise heat transfer and pressure drop. Regions of flow stability for dropwise flow must also be predictable. To obtain this information, which was not available in the literature, Thompson Ramo Wooldridge initiated the boiling research program. This program had two immediate goals:

1. establish local heat transfer coefficients for once-through vortex forced convection boiling of mercury as a function of temperature difference, pressure, flow rate, and quality,
2. correlate these data with the mathematical model postulated for the once-through vortex forced convection boiling of mercury to substantiate the theory.

The development of the mathematical model for boiling mercury was described in Reference 1-1. This section correlates test results of the boiling research program with the theory of heat transfer in dropwise flow postulated by A. Koestel of TRW.

7.1 THEORY OF DROPWISE HEAT TRANSFER

The mechanics of dropwise flow was first investigated at TRW under adiabatic conditions by atomizing mercury droplets into flowing nitrogen gas. Pressure drop and liquid hold-up measurements provided the data necessary for defining the essential flow mechanics. In other tests single mercury droplets were injected into glass tubes and their behavior in a gravity-flow field was studied using a microscope. Moving drop sizes were also measured by high-speed photography. The effect of spring inserts on altering the mercury droplet flow pattern was also studied under adiabatic conditions in glass tubes with flowing nitrogen. Reference 7-19 covers the details of this work and treats the condensation phenomena of dropwise flow.

To obtain high heat transfer rates during the evaporation of mercury in through flow, the drops must be held against the heating surface so that most of the heat is directly transferred to the drops. To accomplish this, a vortex flow must be induced by swirl devices. Spring or twisted ribbon inserts can be used. Based on prototype boiler tests a factor of 3 improvement results over the heat transfer in bare tubes (Ref 1-1).

The experimental and analytical approach used in resolving the complex heat transfer phenomena of dropwise evaporation in a vortex flow is analogous to the superposition

method proposed by Rohsenow and Clark (Ref 7-1). Their method is applicable to a bubble flow regime (see Figure 7-1) since a "pool boiling flux" consisting of bubbles is added to a "convective flux". The analogous situation for dropwise boiling is the evaporation of stationary drops on a hot plate with suitable transformations applied for the case when the drops are moving. In the hot plate experiments described in Reference 1-1, the time to completely vaporize mercury droplets of a given initial size is measured. These data formed a basis for theoretically predicting heat transfer coefficients to a single droplet. When applying the "hot plate" measurements to a forced convection system with vortex flow, the time for evaporation of the droplet becomes the residence time in a tube. The residence time θ is related to the droplet velocity and the tube length as follows:

$$d\theta = \frac{dL}{u_v \epsilon} \quad \text{Eq. 7-2}$$

where u_v is the axial vapor velocity, ϵ is the slip ratio u_e/u_v , L is the tube length, and θ is the residence time of the droplet in the tube.

Equation 7-2 can be integrated to give the length of tube required to completely vaporize the drops. Initial droplet sizes can be estimated from liquid-vapor interfacial stability considerations. The slip ratio is determined from fluid dynamics.

7.1.1 Dropwise Boiling Mechanism

Vortex flow induced by swirl devices has been established by test as necessary to give high heat transfer coefficients (Ref 1-1). The liquid is thus forced against the wall and the transfer of heat is from the wall directly to the liquid. When boiling pure mercury (non-wetting), the liquid will be in the form of spheres or flat spheroids, depending on their size. Body forces induced by swirling action will tend to flatten spherical drops.

The liquid spheres and spheroids in dropwise flow can be (1) supported by a vapor film, (2) in direct contact with the heated wall, (3) intermittently in contact and on a film during the heat transfer process. These heat transfer regimes depend on the magnitudes of the temperature differences and the body forces, and they are similar to the regimes encountered in pool boiling. Figures 7-2 through 7-5 illustrate these various dropwise heat transfer mechanisms. The appropriate equation for each dropwise heat transfer regime is given in these figures together with references. References are not given when the equation can be readily derived. Note that the heat transfer coefficients computed from these equations are based on the projected area of the liquid sphere or spheroid on the heating surface. To apply these equations to boiler design, appropriate transformations are required to convert to heat transfer coefficients based on the tube surface area.

The film heat transfer equation of Figure 7-2 is derived in Reference 1-1 by applying the momentum, continuity, energy, and the heat conduction equation to the supporting vapor film. This derivation is presented in Appendix B. The various contact Nusselt numbers presented in Figures 7-4 and 7-5 were estimated by sketching orthogonal trajectories for two-dimensional heat transfer. This, of course, is approximate since a sphere resting on

FILM BOILING OF A SPHERICAL DROP

Boiling Mechanism: Film (Spheres)

$$\text{Equation: } h_s = \beta \left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{s} \right)^{1/4} \left(\frac{4 k_v h_{fv}}{\Delta T} \right)^{1/2} \quad \text{EQ 7-3}$$

$$\text{or } Nu_s = \beta \frac{s^{3/4}}{k_f} \left(\rho_v \rho_f g_c \frac{2}{3} n \right)^{1/4} \left(\frac{4 k_v h_{fv}}{\Delta T} \right)^{1/2}$$

Reference: Equation is derived analytically in Reference 1-1.
See Appendix B, Equation B-7B.

where h_s is defined by $Q \equiv h_s \frac{\pi s^2}{4} \cdot \Delta T$.

β is defined as:

$$\beta = \frac{(\text{heat transfer coefficient})}{(\text{heat transfer coefficient for spherical drop supported by a vapor film})}$$

$$\beta = 1$$

* Integration with respect to drop diameter gives $\bar{h}_s/h_{s_0} = 4/3$ and with respect to time $\bar{h}_s/h_{s_0} = 5/4$. The former factor was applied to Equation 7-3 to give the spherical drop curve in Figure 7-6.

INTERMITTENT CONTACT AND FILM BOILING OF A DROP

Boiling Mechanism: Intermittent Contact and Film (Spheres)

$$\text{Equation: } h_s = \beta \left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{s} \right)^{1/4} \left(\frac{4 k_v h_{fv}}{\Delta T} \right)^{1/2} \quad \text{EQ 7-4}$$

$$\text{or } Nu_s = \beta \frac{s^{3/4}}{k_f} \left(\rho_v \rho_f g_c \frac{2}{3} n \right)^{1/4} \left(\frac{4 k_v h_{fv}}{\Delta T} \right)^{1/2}$$

Reference: Equation is derived analytically in Reference 1-1.
See Appendix B, Equation B-7B.

$$\beta > 1$$

FIGURE 7-3

CONTACT BOILING OF SPHERICAL DROPS

FIGURE 7-4

BOILING OF FLAT SPHEROIDS

* Can also use pool boiling data for pure mercury as an approximation

FIGURE 7-5

a plane presents a three-dimensional temperature field.

The transition zones from one heat transfer mechanism to another are not easily computable. A theoretical transition between film and contact boiling can be computed by equating Equation 7-3, Figure 7-2, to Equation 7-7, Figure 7-4, for a drop resting on a relatively thin wall. This approximates the case of a SNAP II boiler in which the stainless steel or Haynes 25 walls have a conductivity close to that of liquid mercury ($\cong 10$ Btu/hr-ft-°F, and the wall thickness is estimated to be one-half of the maximum drop size. This will be discussed in more detail in subsequent sections. Combining the two equations we have:

$$Nu_{\delta(\text{CONTACT})} = \left[\frac{\delta^{3/4}}{k_f} \left(e_e e_c g_c \frac{2}{3} n \right)^{1/4} \left(\frac{4 k_v h_{fv}}{\Delta T} \right)^{1/2} \right]_{(\text{TRANSITION})} \quad \text{Eq. 7-11}$$

For a given vapor pressure the critical values causing transition are the drop size δ , the radial or vortex force field acceleration n , and the temperature difference ΔT , as can be seen. Thus both the temperature and the flow field affect the transition. The physical interpretation of this transition is simply that as the vapor film is reduced to very small values, the vapor film heat transfer resistance can become the order of the resistances in the liquid and metal walls. As a result, these resistances begin to play the predominant role.

To complicate this picture for the actual case another transitional region must be considered, namely, intermittent contact. The liquid drop supported by a vapor film is an oscillatory system with little damping. Appendix A presents a dynamic analysis of a drop supported by a vapor film which yields stability criteria. As the vapor film becomes small, the amplitude of vibration approaches the thickness of the vapor film and this causes the intermittent contact phenomenon which is a transitional dropwise boiling regime occurring between contact and film boiling. This regime is similar to partial film boiling observed during pool boiling experiments.

Figure 7-6 shows heat transfer data obtained from tests made on spherical mercury drops placed on a stainless steel ($k \cong 12$) hot plate (Ref 1-1). The plotted heat transfer coefficient based on the drop projected area represents the time mean value for complete vaporization. The initial drop diameters as shown in Figure 7-6 are about 0.1 in. which is in the range of interest to the SNAP II boiler*; therefore the transitional regions indicated in Figure 7-6 should realistically portray prototype conditions for a one-g environment. The heat transfer coefficients during intermittent contact fall, as expected, between those for contact and film. Various equations presented in Figures 7-2 through 7-5 are presented as curves in Figure 7-6. Equation 7-3 as presented in Figure 7-2 is multiplied by $4/3$ to obtain the mean heat transfer coefficient for a drop with an initial diameter of δ_0 and a final diameter of zero (See Eq. B-8). The average drop diameter equal to $5/9 \delta_0$ is used with a contact Nusselt number of 1 for computing the mean heat transfer coefficient in the contact regime.

* As explained in Section 4, initial drop sizes are probably near the wire insert size which actually varied from about 0.060 to 0.028 in.

OPEN STAINLESS STEEL HOT PLATE MERCURY DROP EVAPORATION DATA (REF. 1-1)
SHOWING VARIOUS BOILING REGIMES AND VALUES FOR β AND ψ .
(DATA CORRECTED FOR RADIATION, CONVECTION, & DIFFUSION EFFECTS)

FIGURE 7-6

The intermittent contact phenomenon, as is evident by the hot plate data of Figure 7-6, can be characterized by defining a term Θ (see Figure 7-3) as follows:

$$\Theta \equiv \frac{Nu_{\delta} \text{ (ACTUAL)}}{Nu_{\delta} \text{ (FILM)}} \quad \text{Eq. 7-12}$$

where

$$Nu_{\delta \text{ (ACTUAL)}} = \frac{\int_0^{\Theta_{TOT}} Nu_{\delta \text{ FILM}} d\Theta + \int_0^{\Theta_{TOT}} Nu_{\delta \text{ CONTACT}} d\Theta}{\Theta_{TOT}} \quad \text{Eq. 7-13}$$

In the hot plate vaporization test of mercury drops, Θ_{TOT} represents the total vaporization time. Figure 7-7 illustrates graphically the integration of Equation 7-13. The film Nusselt number is shown as decreasing with time since the drop size becomes smaller due to vaporization (see Equations 7-4 and 7-11). The contact Nusselt number remains constant (see Figure 7-4).

The effect of acceleration and droplet size on the droplet Nusselt number is depicted in Figure 7-8. As shown, increasing both the drop size and the acceleration increases the film Nusselt number. The contact Nusselt number is shown to vary somewhat with acceleration due to the distortion of the liquid from a sphere to a flat spheroid (also see Figures 7-4 and 7-5). Droplets moving in a vortex flow would experience these effects. The zone in which the vaporizing drops experience intermittent contact is shown as increasing with acceleration and drop size (n, δ). This is due to the destabilizing effect of these quantities. Equation A-12 in Appendix A, derived from a transient momentum analysis, shows this as follows:

$$\frac{1}{\theta_c} \left(\frac{k_v}{k_f} \right) \left(\frac{g \mu_v}{\theta_c} \right)^2 > n \delta^3 Nu_{\delta} \quad \text{Eq. 7-14}$$

This equation establishes the conditions for which the drop supported on a vapor film is no longer an oscillatory system. In other words, as n & δ increase, the maximum Nu_{δ} for film boiling decreases.

7.1.2 Drop Size Establishment

In the hot plate experiment mercury drops of predetermined size were placed on the heated surface and the evaporation time recorded. In a prototype boiling system the drop sizes are determined by interfacial hydrodynamic stability considerations. Reference 1-1 presents a comprehensive summary of this subject. To properly apply the heat transfer data obtained from hot plate measurements to forced convection boiling in a tube, the sizes of drops must be known since the characteristic dimension in the equations for the drop heat transfer coefficient (see Equations 7-3 through 7-10) is the drop diameter or the thickness of a flat spheroid.

In both the hot plate test and the forced convection boiling systems, the initial drop size is required for purposes of analysis. A spray nozzle between the preheating and boiling section of a boiler can be used to produce drops of a given size. By proper preheating,

VARIATION OF DROP NUSSELT NUMBER, Nu_δ ,
WITH TIME FOR INTERMITTENT CONTACT

FIGURE 7-7

EFFECT OF RADIAL ACCELERATION (n) AND DROP SIZE (δ) ON DROP
BOILING NUSSELT NUMBER UNDER STATIC CONDITIONS IN 1 "g"*

FIGURE 7-8

vapor flashing in addition to the liquid pressure drop can be utilized for atomization. Figure 7-9A shows this scheme. Good atomization producing small drops requires a short residence time for vaporization and thus gives short boiling lengths. The agglomeration phenomenon and its effect on determining the maximum stable drop size after atomization should be considered since the boiler tube should be long enough to completely evaporate the larger drop sizes. Spray nozzles are not used in full-scale boilers due to plugging possibilities and other complexities.

Initial drop sizes formed in boilers without spray nozzles are of importance to the SNAP II system since this is the scheme under consideration due to practical considerations. Fig. 7-9B shows the processes assumed in establishing the initial drop size δ_0 . The liquid interface is heated to saturation, and the preheater-boiler demarcation interface becomes statically unstable and the liquid interface "falls" apart forming spheroids. These large globules are carried along the tube by the vapor, and since they are unstable to the body forces and interfacial shear stresses caused by the relative velocity of the vapor, they break up until they are of stable size. This size is postulated to be equal to the wire diameter. If a drop were greater than the wire diameter, it is assumed that it would project out into the flowing vapor and thus be sheared down to the wire size. This is, no doubt, a first approximation since various flow parameters (Weber, Reynolds, and Froude numbers) may be involved in addition to the geometric parameters (wire diameter to tube diameter ratio and pitch).

The drop sizes entering the test section can be computed from:

$$\delta = \delta_0 (1 - \alpha)^{1/3} \quad (\text{see Equation B -17, Appendix B}) \quad \text{Eq 7-15}$$

which assumes that the quality is near zero at the point where the initial drop size, δ_0 , is established and that these droplets decrease all equally by evaporation only without changing in number. Equation 7-15 implies that the initial drop size is established outside the test section, as would be the case for any mixing device, and that the drop size entering the test section depends on the quality or on the previous history of vaporization in the boiler before the test section. (This is entirely realistic since it conforms to prototype conditions).

7.1.3 Dropwise Heat Transfer In Forced Convection

To obtain high heat transfer coefficients during dropwise boiling, the liquid drops should be held against the heated surface. In forced convection this requires the formation of vortex flow induced by swirl devices such as springs or twisted ribbons. Under these conditions the drops experience an environment which is different from that on a hot plate, that is, (1) the radial acceleration is a variable along the tube and not equal to one-g as on a hot plate, (2) the drops are in transit, thus presenting a moving heat sink to the heated surface, (3) the vaporization time is now related to tube length, and (4) the flow parameters are such as vapor velocity and slip (see Equation 7-2).

DROP SIZE ESTABLISHMENT

(a) Initial drop size formation by spray nozzle

(b) Initial drop size formation by preheater interface breakup and swirl wire

(c) Schematic showing drop size entering research test section as determined by previous boiling history.

Appendix B presents the development of a theoretical expression for the heat transfer during vortex forced convection dropwise boiling. A brief summary of this development is in order.

In making the transformation from the hot plate to a tube with flow, the following manipulations and assumptions are required:

1. Convert residence time to tube length by combining:

$$h_s = 2 h_{ev} \frac{\rho_v}{\Delta T} \frac{d\delta}{d\Theta}$$

and

$$d\Theta = \frac{dL}{u\epsilon}$$

2. Evaluate the slip ratio, ϵ , by dynamic analysis of a moving drop.
3. Tie the radial acceleration acting on a drop to the vortex flow by:

$$\eta = \left(\frac{\epsilon G \alpha \tan \alpha}{\rho_v} \right)^2 \frac{2}{D g_c} .$$

A liquid drop senses the presence of other drops by means of the accelerations and tube residence time as produced by the mass flow rates and quality. The drop population is determined by these quantities.

4. Include the equations for the various boiling mechanisms (see Figures 7-2 to 7-5).
5. Convert from a droplet heat transfer coefficient to a heat transfer coefficient based on the tube heating surface by substituting the following:

$$h = G \frac{d\alpha}{dL} \frac{D}{4} \frac{h_{ev}}{\Delta T} .$$

6. Assume quasi-static conditions for the wall temperatures as depicted in Figure 7-4, that is, a moving drop has essentially the same temperature field as a drop resting on a hot plate.
7. Establish the size of the drops from thermodynamic and hydrodynamic stability considerations (see Figure 7-9 namely, $\delta = \delta_o (1 - \alpha)^{1/3}$). The initial drop size, δ_o , at the point of two-phase flow establishment is postulated to be equal to the wire diameter.

The details of analysis and of mathematical procedures required for converting the hot plate phenomena to a forced convection boiling system are presented in Appendix B.

The following equations are the results of analysis.

Equation 7-16 is derived as Equation 26-B in Appendix B:

$$\frac{hD}{k_v} = \frac{1}{32} Nu_b^2 \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{\delta_o} \right)^2 \frac{(1-\alpha)^{1/3}}{\alpha} \frac{e_v}{e_f} \quad \text{Eq. 7-16}$$

Equation 7-16 gives the local heat transfer coefficient during forced convection boiling. The boiling mechanism is contact vaporization. The values to be substituted for Nu_b depend on the shape of the liquid spheroid and on the wall thickness and thermal conductivity of the heated surface as shown in Figures 7-4 and 7-5. Equation 7-16 can be used for correlation purposes by experimentally determining Nu_b for a local section of a forced convection boiling system, as is done in Sections 7.2.1 and 7.2.2, Heat Transfer Correlations.

Equation 7-17 is derived as Equation 19-B in Appendix B:

$$\frac{hD}{k_v} = \frac{6^{1/3}}{4} \frac{(1-\alpha)^{2/3}}{\alpha^{1/3}} \frac{D}{\delta_o} \mathfrak{E}^{1/3} (\tan \alpha)^{2/3} \left(\frac{e_v}{e_f} \right)^{2/3} \left(\frac{DG}{\mu_v} \right)^{2/3} \left(\frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \quad \text{Eq. 7-17}$$

Equation 7-17 gives the local heat transfer coefficient during forced convection boiling when the boiling mechanism is film or intermittent contact vaporization. The values to be substituted for $\mathfrak{E}$ depend on the shape of the liquid spheroid and degree of intermittent contact (see Figures 7-2, 7-3 and 7-5). Equation 7-17 can be used for correlation purposes by experimentally determining $\mathfrak{E}$ for a local section of a forced convection boiling system, as is done in Section 7.2.1 and 7.2.2, Heat Transfer Correlations.

The problem of identifying the boiling regime or mechanism from test measurements obtained from a local section of a forced convection boiling system is by no means a simple task. Empirically it can be done by holding all variables constant except the temperature difference and by plotting the local heat transfer coefficient versus the temperature difference (see Figure 7-10). The point of maximum heat transfer coefficient, or the point at which the heat transfer coefficient no longer varies with the boiling mechanism, changes from intermittent contact to contact.

Point A in Figure 7-10 can be theoretically determined for a forced convection boiling process in terms of flow parameters by equating the Nusselt number of Equation 7-17 with the Nusselt number of Equation 7-16. The result is:

$$\left(\frac{h_{fv} \mu_v}{\Delta T k_v} \right)_{\text{CRIT.}} = \frac{Nu_b^2 \left(\frac{k_f}{k_v} \right)^3 \left(\frac{D}{\delta_o} \right)^{3/2} \left(\frac{e_v}{e_f} \right)^{1/2}}{8^{3/2} G^{1/2} (1-\alpha)^{1/2} \alpha \mathfrak{E}_{\text{CRIT.}}^2 \left(\frac{DG}{\mu_v} \right) \tan \alpha} \quad \text{Eq. 7-18}$$

where the temperature difference at point A in Figure 7-10 establishes the magnitude of the parameter $\left(\frac{h_{fv} \mu_v}{\Delta T k_v} \right)$. Equation 7-11 was derived by considerations similar to those of Equation 7-18. $\mathfrak{E}_{\text{crit.}}$ in Equation 7-18 cannot be determined by theory, at least not at the time of this compilation. As a result the value for $\mathfrak{E}_{\text{crit.}}$ as determined

EMPIRICAL METHOD FOR DETERMINING THE VARIOUS BOILING REGIMES
IN A FORCED CONVECTION BOILING SYSTEM. ALL PARAMETERS HELD
CONSTANT EXCEPT THE TEMPERATURE DIFFERENCE.

FIGURE 7-10

by hot plate measurements can be used in Equation 7-18 pending determination of $\Theta_{crit.}$ by empirical methods, that is:

$$\Theta_{crit.} = \frac{h @ A}{h @ B} \quad \text{Eq. 7-19}$$

according to Figure 7-10 $\Theta_{crit.} = 3$ for drop as determined by the hot plate data of Figure 7-6. In applying Equation 7-18 to test data, compute $\left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)_{crit.}$ by means of Equation 7-18 and compare with $\left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)$ as measured, as follows:

$$\text{When: } \left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)_{crit.} > \left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)_{measured} \quad \text{Eq. 7-20}$$

$$\text{or } \Delta T_{measured} > \Delta T_{crit.}$$

we have film or intermittent contact heat transfer.

$$\text{When: } \left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)_{crit.} < \left(\frac{h_{fv} \mu_v}{k_v \Delta T} \right)_{measured} \quad \text{Eq. 7-21}$$

$$\Delta T_{measured} < \Delta T_{crit.}$$

we have contact heat transfer.

Since each boiling regime requires its own unique correlating parameters as is evident by examining Equations 7-16 and 7-17, Equation 7-18 can be used to sort or classify measured data as to their proper boiling regime. This is in contrast to the empirical approach illustrated in Figure 7-10. It should be kept in mind that both Nu_b and $\Theta_{crit.}$ in Equation 7-18 may require adjustment, the amount depending on the deviation of the shape of the liquid spheroid from a sphere as illustrated in Figure 7-5. This is of importance in low quality mixtures where agglomeration may produce large flat spheroids.

7.2 HEAT TRANSFER CORRELATIONS

The sections above discuss the field of boiling heat transfer in general and dropwise boiling of mercury specifically. In this section, the physical interpretation of the dropwise boiling of mercury is used in conjunction with hot plate and other experimental observations to correlate the data obtained for the forced convection boiling of mercury. The data can be distinguished because the SN 3 data were obtained for empirical correlation and the SN 1, SN 2 and B data were obtained in a random manner. The results are therefore analyzed and discussed separately.

In the discussion to follow the randomly obtained data are correlated with reference to the hot plate correlations in the intermittent regime because a detailed analytical expression was not available for this regime. Such expressions were available for the con-

tact and film regimes in the form of Equations 7-17 and 7-16. The intermittent regime was considered to be similar to the film regime but different by a variable factor Θ ($\Theta = Nu / Nu_{\text{FILM-DROP}}$). For comparison, the hot plate Θ (as a function of ΔT) was utilized for reference. In the case of the SN 3 data, however, it was possible to determine the intermittent regime Nusselt number on an empirical basis. This expression is derived in Appendix F and reviewed in Section 8.

7.2.1 Dropwise Heat Transfer Correlation of Test Sections SN 1, SN 2, and B

Randomly obtained test results* for SN 1, SN 2, and B tests are tabulated in Table 7-1. (The test sections are described in Table 2-1.) Pressure drop data are included, but their correlation is discussed in Section 7.3.1, "Pressure Drop Correlation". Due to the random nature of the data taken, correlation by means of the parameters in the theoretical equations (Equations 7-16 and 7-17) is necessary. Equation 7-18 is used to sort the data as to the boiling mechanism, namely, intermittent contact or contact. The implication is that the forced convection boiling process in a vortex flow is similar to the hot plate drop evaporation phenomena and the data obtained should substantiate this.

The correlation procedures were essentially carried out as follows:

1. For each data point compute the equivalent Nu_g by means of Equation 7-16 by substituting experimentally determined quantities such as quality and the heat transfer coefficient.
2. For each data point compute Θ by means of Equation 7-17 substituting experimentally determined quantities such as quality, heat transfer coefficient, flow rate, and temperature difference
3. For each data point compute the quantity $(h_{fg} \mu_v / \Delta T k_v)_{\text{CRIT}}$ by means of Equation 7-18 by substituting experimentally determined quantities such as quality and flow rate. However, for Nu_g and Θ_{CRIT} theoretical quantities should be used. Since the conditions portrayed in Figure 7-4C most nearly fit the local test sections (see Table 2-1), a value of 0.8 for Nu_g was used in Equation 7-18. Θ_{CRIT} equal to 3, as determined from the static hot plate tests (see Figure 7-6) was substituted.
4. Compare the computed quantity $(h_{fg} \mu_v / \Delta T k_v)_{\text{CRIT}}$ with the experimentally determined quantity $(h_{fg} \mu_v / \Delta T k_v)$ and decide by the criteria defined by Equations 7-20 and 7-21 whether the boiling regime is contact or film intermittent contact. If the regime is contact, a symbol C is used in Table 7-1, and if film-intermittent contact, a symbol F is used.

* No intentional relationship between the input variables such as flow rate, quality, temperature difference, and pressure level.

TABLE 7-1
HEAT TRANSFER AND PRESSURE DROP CORRELATING PARAMETERS FOR TEST SECTION SN 1

Run No.	Reduced Data								Properties				Heat Transfer Parameters													Type of Flow	Pressure Drop Correlating Parameters	
	$\dot{m}_T$ $\frac{\text{Lb}}{\text{Hr}}$	G_T $\frac{\text{Lb}}{\text{Sec-Ft}^2}$	$\bar{h}$ $\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$	$\bar{x}$	$\bar{p}$ psia	ΔT_m °F	$\frac{\Delta P}{\Delta L}$ $\frac{\text{psi}}{\text{Ft}}$	T_{sat} °F	k_v $\frac{\text{Btu}}{\text{Hr-Ft-°F}}$	k_f	R $\frac{\text{Lb}}{\text{Ft}^3}$	P_f	μ_v $\frac{\text{Lb}}{\text{Ft-Hr}}$	μ_f	h_{fv} $\frac{\text{Btu}}{\text{Lb}}$	$\frac{k_f}{k_v}$	$\frac{P_f}{P_f}$	$\frac{hD}{k_v}$	$\frac{DG_T}{\mu_v}$	$\frac{h_{fv}\mu_v}{k_v\Delta T_m}$	Nu_s	β	$\frac{h_{fv}\mu_v}{k_v\Delta T_m} \left(\frac{DG_T x}{\mu_v} \right) \Psi$ CRITICAL				M^*	Φ
5010	182	37.8	100	0.0475	25.7	68	~ 0	732	0.00742	8.53	0.403	786	0.159	1.92	127	1149	0.000513	556	35,300	39.8	0.123	0.752	1,350	1,679	0.154	F	3,054	0
5020	190	39.5	79.6	0.200	73.2	88	0.186	863	0.00868	8.83	1.04	769	0.187	1.82	125	1016	0.00135	378	31,400	30.7	0.149	0.657	107,400	6,273	0.186	F	20,800	14.8
5040	206	42.8	39.9	0.088	73.9	167	0.353	864	0.00869	8.83	1.04	769	0.187	1.82	125	1015	0.00136	189	34,000	16.2	0.068	0.393	870	2,989	0.085	F	4,413	124
5050	175	36.4	13.9	0.584	153	476	1.86	975	0.00981	8.97	1.99	753	0.212	1.78	125	915	0.00265	584	25,500	5.65	0.089	0.541	264	14,900	0.111	F	137,800	39.3
5070	183	38.0	91.4	0.699	161	74	2.68	983	0.00990	8.98	2.09	751	0.214	1.77	124	908	0.00278	381	26,400	36.4	0.257	1.03	251	18,500	0.321	F	260,900	37.8
5080	184	38.2	33.6	0.995	161	194	2.68	983	0.00990	8.98	2.09	751	0.214	1.77	124	908	0.00278	140	26,600	13.9	0.368	6.64	1,358	26,400	0.46	F	2.98(10) ⁷	18.4
5150	265	55.1	138	0.256	68	71	~ 0	852	0.00858	8.81	0.970	771	0.185	1.83	126	1027	0.00126	664	44,300	38.1	0.232	0.864	253	11,300	0.29	F	48,500	0
5152	264	54.9	125	0.201	70	74	~ 0	857	0.00862	8.81	0.994	770	0.186	1.82	125	1022	0.00129	598	43,900	36.5	0.191	0.735	313	8,822	0.239	F	28,500	0
5160	269	55.9	130	0.416	103	72	~ 0	913	0.00918	8.90	1.40	762	0.198	1.79	125	970	0.00184	584	41,900	37.5	0.253	0.857	189	17,400	0.317	F	105,600	0
5170	266	55.3	252	0.681	105	38	0.037	916	0.00921	8.91	1.43	762	0.199	1.79	125	967	0.00187	1128	41,300	71.0	0.494	1.56	159	28,100	0.618	F	457,800	0.178
5201	384	79.8	93.9	0.602	92	227	0.371	896	0.00901	8.88	1.27	765	0.194	1.80	125	985	0.00166	430	61,000	11.9	0.288	1.40	108	36,700	0.36	F	440,900	0.975
5202	371	77.1	104	0.0375	38	132	~ 0	776	0.00784	8.64	0.575	781	0.168	1.88	126	1102	0.000737	546	68,100	20.5	0.0939	0.583	935	2,553	0.117	F	2,513	0
5210	370	76.9	63.7	0.0525	51	335	~ 0	815	0.00821	8.73	0.748	776	0.177	1.85	126	1063	0.000965	320	64,700	8.07	0.0773	0.612	727	3,397	0.0965	F	3,661	0
5230	367	76.3	72	0.118	64	188	~ 0	844	0.00850	8.79	0.918	772	0.183	1.83	126	1035	0.00119	350	62,000	14.4	0.113	0.572	358	7,312	0.141	F	13,900	0
5250	362	75.2	127	0.707	69	108	0.595	854	0.00860	8.81	0.982	771	0.185	1.82	126	1025	0.00127	609	60,400	25.0	0.430	1.75	107	42,700	0.537	F	971,700	0.988

$$* M = \left(\frac{DG_v}{\mu_v} \right)^{0.8} \frac{\mu_v}{\mu_f} \frac{P_f}{P_v} \frac{x}{(1-x)}$$

SN 1: 5/8 in. OD with 0.065 in. Wall; 30 in. long, Type 316 SS;
0.054 in. Diameter Swirl wire with 1.8 in. pitch.

TABLE 7-1 (Cont.)

HEAT TRANSFER AND PRESSURE DROP CORRELATING PARAMETERS FOR TEST SECTION SN 2

Run No.	Reduced Data								Properties							Heat Transfer Parameters										Type of Flow	Pressure Drop Correlating Parameter	
	$\dot{m}_T$	G_T	$\bar{h}$	$\bar{x}$	$\bar{p}$	ΔT_M	$\frac{\Delta P}{\Delta L}$	T_{sat}	k_v	k_f	ρ_v	ρ_f	μ_v	μ_f	h_{fv}	$\frac{k_f}{k_v}$	$\frac{\rho_v}{\rho_f}$	$\frac{hD}{k}$	$\frac{D G_T}{\mu_v}$	$\frac{h_{fv} \mu_v}{k_v \Delta T_M}$	Nu_s	β	$\left(\frac{h_{fv} \mu_v}{k_v \Delta T_M}\right) \left(\frac{D G_T x}{\mu_v}\right)$	ψ	M*		ϕ	
	$\frac{Lb}{Hr}$	$\frac{Lb}{Sec-Ft^2}$	$\frac{Btu}{Hr-Ft^2-^{\circ}F}$		psia	$^{\circ}F$	$\frac{psi}{Ft}$	$^{\circ}F$	$\frac{Btu}{Hr-Ft-^{\circ}F}$		$\frac{Lb}{Ft^3}$		$\frac{Lb}{Ft-Hr}$		$\frac{Btu}{Lb}$													
2010	175	136	301	0.194	26.9	16.2	3.31	736	0.00746	8.54	0.421	785	0.160	1.92	126	1145	0.000536	861	65,700	167	0.603	0.994	83.8	12,800	0.377	C	72,200	4.14
2020	153	119	321	0.299	28.8	15.3	3.02	743	0.00753	8.56	0.448	785	0.161	1.91	126	1137	0.000570	912	56,600	177	0.768	1.26	68.6	16,900	0.480	C	152,000	2.24
2030	172	134	249	0.661	58.3	33.0	7.49	832	0.00838	8.77	0.844	773	0.180	1.84	126	1046	0.00109	635	57,200	82.0	0.845	1.78	47.5	37,800	0.530	C	803,000	1.68
2040	174	135	255	0.678	61.5	32.2	7.04	839	0.00845	8.78	0.886	773	0.182	1.83	126	1039	0.00115	646	57,200	83.8	0.855	1.79	47.8	38,700	0.535	C	851,000	1.55
2050	168	131	196	0.992	91.7	55.0	7.36	894	0.00899	8.88	1.27	765	0.194	1.80	125	987	0.00166	466	52,000	49.1	1.45	11.6	244	51,600	0.905	F	5.13(10) ⁷	1.15
2060	181	141	161	0.923	92.4	94.6	6.11	895	0.00900	8.88	1.28	765	0.194	1.80	125	986	0.00167	383	56,000	28.5	0.851	3.83	75.4	51,600	0.532	F	4.53(10) ⁶	0.960
2070	222	173	624	0.0436	18.7	11.2	0.487	697	0.00710	8.43	0.302	790	0.152	1.96	127	1187	0.000383	1878	87,900	242	0.467	0.958	242	3,830	0.292	C	6,775	5.38
2080	262	204	313	0.345	51.2	37.1	5.71	815	0.00821	8.73	0.751	776	0.177	1.85	126	1063	0.000968	813	88,700	72.8	0.647	1.18	41.8	30,600	0.405	C	200,800	1.82
2090	188	146	269	0.211	25.2	40.8	3.16	728	0.00739	8.52	0.397	786	0.158	1.93	127	1153	0.000505	777	71,300	66.4	0.612	1.49	71.4	15,000	0.382	C	95,200	2.75
2100	201	156	238	0.368	36.0	46.7	6.32	770	0.00778	8.63	0.548	781	0.167	1.89	126	1109	0.000701	652	71,700	58.0	0.678	1.52	47.6	26,400	0.423	C	252,600	2.21
2110	199	155	214	0.407	37.2	57.4	6.65	773	0.00781	8.64	0.564	781	0.168	1.88	126	1106	0.000722	586	70,900	47.1	0.676	1.63	45.1	28,900	0.422	C	313,600	1.97
2120	201	156	184	0.416	38.0	82.7	4.78	776	0.00784	8.64	0.575	781	0.168	1.88	126	1102	0.000737	502	71,300	32.7	0.629	1.74	44.4	29,700	0.393	C	326,600	1.37
2130	197	153	184	0.517	55.2	47.7	6.89	824	0.00830	8.75	0.805	775	0.178	1.84	126	1054	0.00104	475	66,200	56.7	0.620	1.29	43.9	34,200	0.387	C	422,200	1.84
2140	206	160	197	0.522	52.6	56.7	8.11	818	0.00824	8.74	0.770	775	0.177	1.85	126	1060	0.000993	511	69,300	47.7	0.658	1.50	41.5	36,200	0.411	C	466,700	1.88
2150	201	156	210	0.560	54.6	66.0	8.27	823	0.00829	8.75	0.796	775	0.178	1.84	126	1055	0.00103	541	67,300	40.9	0.703	1.78	41.7	37,700	0.439	C	548,000	1.80
2160	194	151	120	0.852	76.2	68.1	7.78	869	0.00874	8.84	1.07	768	0.188	1.82	125	1011	0.00140	293	61,800	39.7	0.685	1.96	52.6	52,600	0.427	C	2.55(10) ⁶	1.05
2170	199	155	81.2	0.852	79.7	178	8.92	875	0.00880	8.85	1.12	768	0.189	1.81	125	1005	0.00145	197	63,000	15.1	0.554	2.29	51.9	53,600	0.346	F	2.51(10) ⁶	1.19
2180	176	137	49.8	0.999	102	153	8.12	908	0.00913	8.90	1.39	763	0.197	1.80	125	974	0.00182	116	53,700	17.6	0.977	17.5	644	53,600	0.610	F	3.65(10) ⁸	1.25
2190	110	85.6	136	0.482	30.3	32.2	4.70	749	0.00758	8.58	0.469	784	0.162	1.91	126	1131	0.000598	382	40,500	84.2	0.653	1.44	69.5	19,600	0.408	C	359,900	2.70
2200	123	95.5	164	0.510	28.8	45.0	4.54	743	0.00753	8.56	0.448	785	0.161	1.91	126	1137	0.000571	467	45,600	60.2	0.762	1.99	59.7	23,200	0.476	C	478,900	1.79
2210	110	85.8	169	0.624	32.8	77.1	4.38	758	0.00767	8.60	0.504	783	0.164	1.90	126	1121	0.000644	470	40,200	35.1	0.844	3.15	64.3	25,100	0.527	C	237,500	1.61
2220	111	86.4	195	0.151	21.9	19.0	0.649	713	0.00725	8.48	0.349	788	0.155	1.94	127	1169	0.000443	574	42,900	143	0.463	0.988	156	6,500	0.289	C	35,800	2.78
2230	118	91.8	183	0.147	18.4	42.2	0.811	696	0.00709	8.43	0.297	790	0.151	1.96	127	1188	0.000377	551	46,600	64.2	0.477	1.47	142	6,843	0.298	F	41,300	2.77
2240	113	88.1	224	0.173	18.6	52.3	0.973	697	0.00710	8.43	0.301	790	0.152	1.96	127	1188	0.000382	676	44,700	51.8	0.573	2.05	129	7,747	0.358	F	54,900	2.63
2250	206	160	166	0.0305	16.1	27.9	0.486	683	0.00697	8.39	0.263	791	0.149	1.98	127	1203	0.000333	509	82,700	97.2	0.215	0.570	353	2,525	0.134	F	3,753	1.2
2260	203	158	151	0.0392	15.8	60.4	0.632	681	0.00695	8.38	0.258	792	0.148	1.98	127	1205	0.000326	464	81,800	44.9	0.235	0.852	278	3,206	0.147	F	5,981	8.92
2271	202	157	286	0.0498	16.5	46.5	0.568	685	0.00699	8.39	0.269	791	0.149	1.97	127	1201	0.000340	873	81,100	58.3	0.358	1.26	224	4,036	0.224	F	8,938	5.20
2272	262	204	82.2	0.0608	17.6	65.6	1.22	692	0.00705	8.41	0.286	790	0.151	1.97	127	1193	0.000362	249	104,000	41.3	0.207	0.529	146	6,326	0.129	F	15,100	4.73
2280	265	206	259	0.0712	18.1	43.3	1.38	694	0.00707	8.42	0.293	790	0.151	1.96	127	1191	0.000371	783	104,900	62.6	0.394	1.04	125	7,470	0.246	F	19,950	3.92
2290	264	205	282	0.103	20.3	88.6	2.19	705	0.00717	8.45	0.326	789	0.153	1.95	127	1178	0.000414	839	102,800	30.6	0.471	1.68	91.5	10,600	0.294	F	36,400	3.33
4010	53.4	41.5	83.2	0.371	24.2	40.5	1x10 ⁻⁵	725	0.00736	8.51	0.381	787	0.158	1.93	127	1156	5.04x10 ⁻⁴	242	20,300	66.4	0.479	1.53	158	7,530	0.299	F	1.25(10) ⁵	3.35(10) ⁻⁵
4020	52.5	40.8	77.7	0.729	25.2	45.6	1.20																					

$$* M = \left(\frac{D G_v}{\mu_v} \right)^{0.8} \frac{\mu_v}{\mu_f} \frac{\rho_f}{\rho_v} \frac{x}{(1-x)}$$

SN 2: 5/16 in. OD with 0.028 in. Wall; 30 in. long, Type 316 SS;
0.054 in. Diameter Swirl wire with 1 in. pitch.

NOTE: A "vapor only" friction factor of 0.095 is used to compute Φ for Test Section SN 2
Runs 2050, 2180, 4110, 4120 indicate that the average friction factor is 0.106
assuming the effect of the liquid phase on the pressure drop for $x > 99\%$ is negligible.

TABLE 7-1 (Cont.)

HEAT TRANSFER AND PRESSURE DROP CORRELATING PARAMETERS FOR TEST SECTION B

Run No.	Reduced Data								Properties					Heat Transfer Parameters											Pressure Drop Correlating Parameters		
	$\dot{m}_T$	G_T	$\bar{h}$	$\bar{x}$	$\bar{p}$	ΔT_m	$\frac{\Delta P}{\Delta L}$	T_{SAT}	k_v	k_f	ρ_v	ρ_f	μ_v	μ_f	h_{fv}	$\frac{k_f}{k_v}$	$\frac{\rho_v}{\rho_f}$	$\frac{hD}{k}$	$\frac{DG_T}{\mu_v}$	$\frac{h_{fv}\mu_v}{k_v\Delta T_m}$	Nu_g	β	$\left(\frac{h_{fv}\mu_v}{k_v\Delta T_m}\right)\left(\frac{DG_Tx}{\mu_v}\right)$	ψ	Φ	M^*	
	$\frac{\text{Lb}}{\text{Hr}}$	$\frac{\text{Lb}}{\text{Sec-Ft}^2}$	$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$		Psia	°F	$\frac{\text{Psi}}{\text{Ft}}$	°F	$\frac{\text{Btu}}{\text{Hr-Ft}^2-\text{°F}}$		$\frac{\text{Lb}}{\text{Ft}^3}$		$\frac{\text{Lb}}{\text{Ft-Hr}}$		$\frac{\text{Btu}}{\text{Lb}}$			k	μ_v	$k_v\Delta T_m$			CRITICAL				
3020	97.1	75.5	9.01	0.150	17.7	155	0.227	692	0.00705	8.41	0.287	790	0.151	1.97	127	1193	3.63(10) ⁻⁴	27.3	38,600	17.4	0.126	0.310	0.0380	5,786	0.079	2.79	3.81(10) ⁴
3031	99	76.9	5.80	0.154	17.6	493	0.260	692	0.00705	8.41	0.286	790	0.151	1.97	127	1193	3.62(10) ⁻⁴	17.6	39,300	5.50	0.103	0.398	0.0364	6,048	0.064	2.94	4.08(10) ⁴
3032	101	78.8	20.0	0.345	32.8	74.5	0.389	761	0.00769	8.61	0.502	783	0.165	1.90	126	1119	6.42(10) ⁻⁴	55.7	36,800	36.3	0.229	0.396	0.0216	12,700	0.143	1.76	1.37(10) ⁵
3042	108	83.87	23.1	0.598	49.3	136	1.01	810	0.00816	8.72	0.727	776	0.175	1.85	126	1068	9.36(10) ⁻⁴	60.4	36,800	19.9	0.295	0.69	0.0167	22,000	0.185	2.22	4.48(10) ⁵
3050	165	128	211	0.0787	16.4	15.1	0.227	684	0.00698	8.39	0.268	791	0.149	1.97	127	1201	3.39(10) ⁻⁴	647	66,200	208	0.45	0.624	0.0400	5,213	0.281	3.18	1.79(10) ⁴
3060	166	129	263	0.0592	16.8	22.8	0.227	687	0.00701	8.40	0.274	791	0.150	1.97	127	1199	3.47(10) ⁻⁴	802	65,200	119	0.429	0.88	0.0526	3,935	0.268	5.29	1.03(10) ⁴
3070	168	131	240	0.0445	17.0	36.6	0.0973	688	0.00702	8.40	0.277	791	0.150	1.97	127	1197	3.50(10) ⁻⁴	730	67,200	74.1	0.352	0.95	0.0688	2,990	0.220	32.3	6098
3080	159	124	232	0.0842	18.5	39.4	0.308	1.00	0.00163	4.49	0.753	852	0.0283	3.54	132	2755	8.84(10) ⁻⁴	3051	336,700	58.3	0.273	1.06	0.143	28,400	0.171	0.025	3032
3120	161	125	314	0.347	26.6	18.8	1.78	736	0.00746	8.54	0.416	785	0.160	1.92	126	1145	5.29(10) ⁻⁴	900	60,100	144	0.994	1.39	0.0128	20,800	0.620	2.99	2.38(10) ⁵
3130	167	130	395	0.379	29.3	30.4	1.99	147	0.00262	5.62	0.905	840	0.0508	3.07	131	2145	0.00108	3223	196,100	83.9	0.739	1.92	0.0347	74,200	0.462	7.79	73,700
3150	181	141	24.8	0.591	13.6	329	1.46	956	0.00961	8.96	1.79	756	0.208	1.78	125	932	0.00237	55.2	52,100	8.16	0.201	0.525	0.0125	30,800	0.125	3.14	2.77(10) ⁵
3171	181	141	267	0.761	152	11.8	2.92	974	0.00980	8.97	1.98	753	0.212	1.78	125	916	0.00263	583	51,300	228	0.785	0.778	0.0129	39,100	0.491	4.41	6.83(10) ⁵
3172	185	144	57.6	0.761	159	160	2.85	981	0.00987	8.98	2.06	752	0.213	1.77	124	909	0.00274	125	51,800	16.8	0.358	0.876	0.0128	39,400	0.224	4.32	6.64(10) ⁵
3180	228	177	284	0.0463	22.8	16.5	0.035	719	0.0073	8.49	0.361	787	0.156	1.94	127	1163	4.59(10) ⁻⁴	830	87,000	164	0.345	0.548	0.0537	4,025	0.216	0.946	6525
3200	230	179	352	0.0491	23.8	33.5	0.162	724	0.00735	8.51	0.376	787	0.157	1.93	127	1158	4.77(10) ⁻⁴	1024	87,700	80.9	0.389	0.904	0.0506	4,302	0.243	4.01	7103
3220	230	179	15.6	0.0741	21.4	799	0.292	712	0.00724	8.47	0.341	788	0.155	1.94	127	1170	4.32(10) ⁻⁴	45.9	88,800	3.39	0.106	0.504	0.0330	6,579	0.665	3.2	16,700
3230	48.8	38.0	373	0.414	25.0	5.12	0.0811	729	0.00740	8.52	0.393	786	0.158	1.93	127	1152	0.0005	1078	18,500	529	1.23	1.7	0.0366	7,648	0.770	0.755	1.49(10) ⁵
3240	53.3	41.4	241	0.387	25.6	11.7	0.714	730	0.00741	8.53	0.403	786	0.159	1.92	127	1150	5.12(10) ⁻⁴	694	20,100	232	0.941	1.68	0.0355	7,764	0.590	6.59	1.31(10) ⁵
3250	53.3	41.4	163	0.396	25.8	24.4	0.0328	732	0.00743	8.53	0.404	786	0.159	1.92	127	1148	0.000514	470	20,000	111	0.785	1.83	0.0349	7,938	0.491	0.291	1.39(10) ⁵

$$* M = \left(\frac{DG_v}{\mu_v} \right)^{0.8} \frac{\mu_v}{\mu_f} \frac{\rho_f}{\rho_v} \frac{x}{(1-x)}$$

NOTE: A "vapor only" friction factor equal to $0.316/Re_v^{1/4}$ is used to compute for test section B.

B: 5/16 in. OD with 0.028 in. Wall; Bare Tube, 30 in. long, Haynes 25.

Table 7-1 lists the above-computed quantities together with other quantities of interest. Five of the tests shown in Table 7-1 show qualities equal to or greater than 99%. These have unrealistically high values of Θ and Nu_δ because the portion of the heat transferred to the vapor directly is large compared to the heat flowing to the mercury drops. In the formulation of the theory, only the heat flowing directly from the heated wall to the liquid spheroids was considered. These five test points are required for determining the "vapor only" friction factor for the tube with its spring insert.

The remaining data for which the qualities are less than 99% indicate values of Θ and Nu_δ which compare favorably in magnitude to those obtained by the hot plate experiments (see Figure 7-6). To present the results of Table 7-1 in a form more directly comparable to those obtained by the hot plate, a method of plotting shown in Figure 7-11 was devised. Here Θ is defined as:

$$\Theta \equiv \frac{Nu_\delta \text{ (ACTUAL)}}{Nu_\delta \text{ (FILM SPHERICAL DROPS)}}$$

and is plotted versus

$$\psi \equiv \frac{Nu_\delta \text{ (ACTUAL)}}{Nu_\delta \text{ (CONTACT SPHERICAL DROPS)}}$$

The shape of the curves shown in Figure 7-11 is drawn to conform to the hot plate boiling phenomena, that is, when $\psi \ll 1$, Θ should equal one since the spherical drop will be stably supported by a vapor film. When $\psi \rightarrow 1$, oscillations of the drop cause intermittent contact and $\Theta > 1$. The hot plate data indicate that

$$\frac{\Theta}{\left(\frac{h_{ev} \mu_v}{\Delta T k_v}\right)^{1/2}} = \text{CONSTANT} \quad \text{Eq. 7-22}$$

In the intermittent contact regime and that Θ attains a maximum value of about 3 which is defined as $\Theta_{\text{CRIT.}}$ in Equation 7-18. Since the curve for the film Nusselt number intersects the curve for the contact Nusselt number as shown in Figure 7-10, the values for Θ should decrease from 3 to 1 at the point of intersection. Beyond this intersection point, values for Θ are less than one, and values for ψ should be equal to one. These trends are all illustrated in Figure 7-11. Included in Figure 7-11 are the data obtained from the hot plate tests.

The point at which transition from film to intermittent contact occurs (See Figure 7-11) is near a ψ of 0.1 according to the hot plate data plotted in Figure 7-6. This may not be the case in forced convection boiling where accelerations and drop sizes are different from those experienced on the hot plate. Equation 7-14 bears this out. Combining Equation 7-14 with Equation 7-16, eliminating the drop size, and substituting

$$Nu_\delta = \psi Nu_\delta \text{ (CONTACT SPHERICAL DROP)}$$

METHOD OF PORTRAYING HOT PLATE BOILING REGIMES.
DATA SYMBOLS ARE FROM HOT PLATE TEST (SEE FIGURE 7-6).

$$\beta \equiv \frac{Nu_6 \text{ (actual)}}{Nu_6 \text{ (film; spherical drops)}}$$

$$\psi \equiv \frac{Nu_6 \text{ (actual)}}{Nu_6 \text{ (contact; spherical drops)}}$$

FIGURE 7-11A

HOT PLATE DATA CORRELATION (ALSO SEE FIGURES 7-6, 7-11A)

FIGURE 7-11B

we have

$$\frac{1}{q_c Nu_{\delta}(\text{CONTACT}) \delta_o^3} \left(\frac{k_v}{k_f} \right) \left(\frac{g \mu_v}{\rho_f} \right)^2 > n (1 - \alpha) \psi \quad \text{Eq. 7-23}$$

On the left side of Equation 7-23 are grouped the near constants encountered during the tests. The initial drop size is assumed to be established near the preheater interface in the boiler section before the test section. For a given test section the acceleration, n , is proportional to the vapor Reynolds number if the pressure level is held constant. When one examines the data in Table 7-1, one will detect a trend between the quality and the vapor Reynolds number. The low Reynolds numbers correspond to low qualities. This was not intentional since the data were taken randomly with no apparent connection between variables. However, the effect on Equation 7-23 is to reduce the variation of the term $n (1 - \alpha)$. As a result, if one were to plot the forced convection data of Table 7-1 in accordance with the method described in Figure 7-11, one would expect to find the point of incipient intermittent contact (ψ_{CRIT} in Figure 7-11) to be nearly constant. Such is the case as will be subsequently shown. The dashed lines in Figure 7-11 are for flat spheroids. During film boiling of flat spheroids the data of Figure 7-6 show that values for ϕ are less than one and are near 0.43. During contact boiling of flat spheroids, ψ should have a value of

$$\psi = \frac{2/3}{0.8} = 0.83$$

according to the conditions defined in Figure 7-4C and Figure 7-5D.

Figure 7-12 compares data from Table 7-1 which have been identified as film or intermittent contact boiling by means of Equation 7-18 with the hot plate boiling curves of Figure 7-11. Indications are that practically all of the points are in intermittent contact. It would have been desirable to have covered a greater range in temperature difference so that more data would lie in the stable film region. Since the data fall below the curve for spherical drops in Figure 7-12, it might be postulated that the mercury spheroids are deformed from the spherical shape by the accelerations of the vortex flow, and thus have drop Nusselt numbers lower than that of a spherical drop.

Figure 7-13 compares data from Table 7-1, which have been identified as contact boiling by means of Equation 7-18, with the hot plate boiling curves of Figure 7-11. Again, indications are that the mercury spheroids are flat since the data groups about the dotted curve. The data of Figure 7-13 are also plotted in Figure 7-14 as the measured drop Nusselt number versus quality. The data show a downward trend in drop Nusselt number with decreasing quality which would be indicative of flatter drops at lower qualities as would be intuitively expected.

The data of Table 7-1 as plotted in Figures 7-12, 7-13 and 7-14 substantiate the dropwise forced convection boiling theory as constructed from the hot plate experiment, since the magnitudes and trends of the experimentally determined quantities of ϕ and ψ compare

FORCED CONVECTION DATA CORRELATED TO IDENTIFY BOILING REGIMES
AS EITHER FILM OR INTERMITTENT CONTACT. COMPARISON IS MADE WITH
CURVES FOR STATIC BOILING OF DROPS OR SPHEROIDS (SEE FIGURE 7-11)

FIGURE 7-12

DATA IDENTIFIED AS CONTACT BOILING COMPARED WITH
CURVES FROM HOT PLATE TESTS (SEE FIGURES 7-6, 7-11)

FORCED CONVECTION CONTACT BOILING DATA COMPARED
WITH THEORETICAL DROP NUSSELT NUMBERS AS OBTAINED
FROM FIGURES 7-4C AND 7-4D

FIGURE 7-14

favorably with those obtained from the hot plate experiment. Minor adjustments can be made to the values for $\mathcal{Q}$ and ψ to account for their deviation from the curves constructed for spherical drops when using these correlations for design purposes.

It can be concluded by inspecting Figures 7-12 and 7-13 that:

1. transition from film to intermittent contact boiling occurs near $\psi = 0.1^*$ ($\psi_{crit.}$),
2. transition from intermittent contact to contact boiling occurs near $\mathcal{Q} = 3^*$ ($\mathcal{Q}_{crit.}$),
3. The use of Equation 7-18 for sorting data as to film or intermittent contact and contact boiling appears valid since the data symbols fall into their corresponding regions of Figures 7-12 and 7-13.
4. $\mathcal{Q}$, ψ , and Nu_{δ} values in Figures 7-12, 7-13 and 7-14 respectively appear somewhat low when compared to the theoretical curves as constructed from the hot plate tests. This could possibly be attributed to the difference between the swirl wire diameter in the boiler section ($\delta_o = 0.0624$ in.) and the local test section ($\delta_o = 0.054$ in. for both SN 1 and SN 2). See Table 2-1. Since the initial drop diameter, δ_o , is formed near the interface of the boiler section, the drops entering the local test section will be larger than anticipated since the local test section wire diameter has been used in Equations 7-16, and 7-18 for correlation. The values for $\mathcal{Q}$, ψ , and Nu_{δ} could be adjusted by multiplying by the following factors:

$$\text{for } \mathcal{Q}: \left[\frac{\delta_o (\text{BOILER})}{\delta_o (\text{TEST SECTION})} \right]^{3/4} = \left(\frac{0.0624}{0.054} \right)^{3/4} = 1.115 \quad \text{See Equation 7-17}$$

$$\text{for } \psi \text{ and } Nu_{\delta}: \left(\frac{0.0624}{0.054} \right) = 1.156 \quad \text{See Equation 7-16}$$

The bare tube data of Table 7-1 is plotted in Figure 7-15 in the same manner as in Figures 7-12 and 7-13. Equations 7-16 and 7-17 were used to compute $\mathcal{Q}$ and Nu_{δ} ; however, the values for δ_o and $\tan \alpha$ which were substituted were those of the boiler upstream of the test section. It was assumed that the initial droplet size was established by the wire size in the boiler and that some residual swirl still existed in the flow entering the local test section. When one compares the bare tube data of Figure 7-15 with the swirl tube data of Figures 7-12 and 7-13, one quickly detects the effect of swirl device. The greatest effect occurs toward the film boiling regime, and the effect diminishes as contact boiling is approached. This is to be expected since the swirl wire effect is included in the film boiling equation,

* In forced convection boiling conditions much different from those of these tests, the above statements may not be true. See discussion associated with Equation 7-23.

BARE TUBE DATA COMPARED WITH CURVES FROM HOT PLATE TESTS (SEE FIGURES 7-6, 7-11)

FIGURE 7-15

Equation 7-17, but not in the contact boiling equation (Equation 7-16). It is estimated from Figure 7-15 that the swirl wire increases the heat transfer coefficient in the film boiling region by a factor of

$$\left[\frac{\Theta_{\text{(SWIRL)}}}{\Theta_{\text{(BARE)}}} \right]^{4/3} = \left(\frac{5.7}{3.0} \right)^{4/3} = 2.36.$$

Without the residual swirl effect produced by the boiler, the above factor might exceed a factor of 3 which was the initial estimate of the swirl wire effect applied during SNAP II boiler development (see Ref 1-1). Figure 7-15 also indicates a $\psi_{\text{crit.}}$ value less than 0.1 (transition from film to intermittent contact). This is also to be expected since acceleration is destabilizing according to Equation 7-23 and $n_{\text{(swirl)}} > n_{\text{(bare tube with residual swirl)}}$.

7.2.2 Heat Transfer Correlation of Test Section SN 3

Tabulated results for test section SN 3 are presented in Table 7-2. Two values for Θ and ψ are shown: Θ' and ψ' are computed by means of Equations 7-16 and 7-17 using the rig boiler wire diameter ($\delta_o = 0.0625$ in.). Θ and ψ are determined by the same equations with the test section wire diameter ($\delta_o = 0.028$ in.) substituted. This was done to detect the effect of rig boiler or test section wire diameter in establishing the liquid mercury drop sizes.

Figure 7-16 shows the data of Table 7-2 plotted using the same coordinates used in Figures 7-11, 7-12, and 7-13. Since the correlation of test section SN 3 (Figure 7-16) agrees reasonably well with the correlation of test sections SN 1 and SN 2 (Figures 7-12 and 7-13) when Θ' and ψ' are used, it can be postulated that the data of test section SN 3 were rig sensitive and that the test section wire diameter was relatively ineffective. It can be argued that the rig boiler established the initial flow pattern and that the test section swirl device played a negligible role in altering it. This was, no doubt, also true for test sections SN 1 and SN 2; however, the swirl wire diameters in test sections SN 1 and SN 2 were not much different from that of the rig boiler. Any further testing should require that the swirl device and tube diameter for the rig boiler be made geometrically the same as the test section, thus producing the proper previous history before entering the local test section.

Appendix F presents an empirical derivation of an equation for the forced convection, dropwise boiling of mercury in intermittent contact. The bases of the derivation are the plotted data shown in Figures 7-12 and 7-16. These data in the intermittent contact regime can be represented empirically by:

$$\Theta = C \psi^n \quad \text{Eq. 7-24}$$

where the exponent n is approximately equal to 1.0 and C has a value between 2.3 and 3.0. Equation 7-24 results in

$$\frac{hD}{k_v} = C^4 234 \left(\frac{k_v}{k_f} \right)^4 (1 - \alpha)^{4/3} \alpha \left(\frac{\delta_o}{D} \right) \left(\tan \alpha \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v} \right)^2 \quad \text{Eq. 7-25}$$

TABLE 7-2

HEAT TRANSFER AND PRESSURE DROP CORRELATING PARAMETERS FOR TEST SECTION SN 3

Run No.	Reduced Data		General		Parameters For Comparison With Theory										Type Of Flow	Pressure Drop Parameters		
	$\bar{x}$	$\bar{P}_f$ psia	$\bar{h}_f$ $\frac{Btu}{hr-ft^2-F}$	$\frac{hD}{k}$	$\frac{DG}{\mu}$	$\left(\frac{h_{fv}\mu_v}{k_v\Delta T_m}\right)^*$	$\left(\frac{h_{fv}\mu_v}{k_v\Delta T_m}\right)^*$	Nu_s	ψ	ψ'	β	β'	Film Or Contact	$f\phi$	ϕ $f_{0.053}$	M		
6020	0.1552	129.05	52.33	117.7	42,427	26.30	4620	263	0.0562	0.0703	0.157	0.1767	0.322	F	0.463	8.74	8.07×10^3	
6031	0.4779	148.35	416.0	915.9	43,499	101.3	1654	94.2	0.3065	0.383	0.855	0.6855	1.25	C	0.149	2.81	1.19×10^5	
6041	0.1251	146.7	86.22	190.0	41,981	39.49	5044	287	0.0658	0.0824	0.184	0.1909	0.348	F	0.396	7.47	6.26×10^3	
6042	0.1165	147.8	90.17	198.5	42,220	42.01	5363	305	0.0647	0.081	0.071	0.1857	0.339	F	0.453	8.55	5.44×10^3	
6050	0.1228	148.35	272.0	598.7	43,500	120.1	4965	283	0.1152	0.144	0.322	0.2514	0.458	F	1.22	23.0	6.15×10^3	
6061	0.1103	148.85	377.4	830.8	42,362	92.05	5646	322	0.1281	0.16	0.358	0.3590	0.655	F	0.759	14.3	4.88×10^3	
6062	0.1070	148.9	358.5	789.3	42,277	87.59	5822	331	0.1229	0.154	0.343	0.3511	0.641	F	0.720	13.6	4.59×10^3	
6071	0.0897	148.75	251.4	553.4	43,300	30.48	6712	382	0.0939	0.117	0.262	0.4274	0.781	F	3.411	64.4	3.34×10^2	
6072	0.1096	148.7	228.6	503.3	43,130	28.19	5576	318	0.0994	0.124	0.277	0.4408	0.804	F	0.987	18.7	4.89×10^2	
6081	0.2058	149.9	267.0	585.9	43,216	32.01	3121	178	0.1497	0.187	0.417	0.5722	1.045	F	0.558	10.6	1.70×10^4	
6082	0.2075	150.4	320.1	702.4	42,876	38.51	3129	178	0.1644	0.206	0.458	0.6009	1.10	F	0.326	6.15	1.72×10^4	
6091	0.1944	148.35	223.9	493.0	43,130	53.57	3301	186	0.1334	0.167	0.372	0.3825	0.698	F	0.444	8.38	1.52×10^4	
6092	0.1944	148.35	244.3	537.9	43,642	57.60	3262	186	0.1393	0.174	0.389	0.3914	0.715	F	0.434	8.19	1.54×10^4	
6100	0.2041	147.6	0.2970	0.6539	42,362	32.80	3219	183	0.00499	-	-	0.00349	-	F	0.294	5.55	1.66×10^4	
6111	0.1512	148.65	140.9	310.2	43,357	16.20	4105	234	0.0926	0.116	0.258	0.4484	0.818	F	0.467	8.83	9.26×10^2	
6112	0.1579	148.60	174.5	384.2	42,931	19.43	3997	228	0.1052	0.131	0.294	0.4898	0.894	F	0.480	9.06	9.95×10^2	
6121	0.2822	149.5	31.84	70.03	56,800	4.728	1831	104	0.0616	0.077	0.172	0.3008	0.549	F	0.100	1.89	4.14×10^4	
6122	0.2714	149.55	31.48	69.24	57,936	4.712	1853	105	0.0599	0.0749	0.167	0.2907	0.531	F	0.0970	1.83	3.86×10^4	
6130	0.2066	149.55	20.74	45.61	26,781	8.483	5045	287	0.0418	0.0524	0.117	0.2085	0.381	F	0.181	3.42	1.17×10^4	
6140	0.1830	152.05	31.26	68.53	26,355	23.05	5698	324	0.0478	0.0596	0.134	0.1642	0.300	F	0.240	4.53	8.91×10^2	
6151	0.2015	149.05	21.72	47.77	25,305	11.54	5448	310	0.0423	0.0529	0.118	0.1889	0.345	F	0.212	4.0	1.07×10^4	
6152	0.2053	149.0	22.86	50.28	25,560	11.91	5305	302	0.0438	0.0548	0.122	0.1937	0.353	F	0.200	3.78	1.12×10^4	
6160	0.1857	150.1	10.95	24.04	25,107	3.221	5902	336	0.0286	0.0358	0.0798	0.2075	0.379	F	0.170	3.21	8.91×10^2	
6170	0.2166	150.35	345.7	759.4	25,022	115.4	5181	295	0.1748	0.218	0.488	0.4898	0.894	F	0.252	4.76	1.22×10^4	
6180	0.5268	148.1	397.87	875.91	42,135	171.22	1625.5	92.5	0.3201	0.400	0.894	0.5581	1.02	C	0.0818	1.55	1.53×10^5	
6190	0.5193	148.1	546.72	1203.6	45,376	125.14	151.92	92.5	0.3716	0.464	1.036	0.2496	0.455	C	0.000726	-	9.84×10^3	
6200	0.5072	147.95	450.84	992.52	42,931	71.720	1622.9	92.5	0.3323	0.416	0.928	0.9111	1.66	F	0.0965	1.82	1.40×10^5	
6210	0.5001	148.05	520.8	1147	42,790	62.64	1640	93.4	0.3537	0.442	0.986	1.076	1.96	F	0.0920	1.74	1.34×10^5	
6220	0.4687	144.95	481.4	1062	43,024	41.03	1681	95.6	0.3289	0.411	0.917	1.207	2.20	F	0.111	2.1	1.14×10^5	
6230	0.7700	151.55	261.0	572.1	41,900	132.4	1603	91.2	0.3504	0.438	0.978	0.7217	1.32	C	0.0588	1.11	6.11×10^5	
6240	0.7420	147.55	383.0	843.2	42,646	98.45	1542	87.8	0.4136	0.518	1.155	1.053	1.92	C	0.0600	1.13	5.3×10^5	
6251	0.7354	151.60	427.1	936.3	42,802	87.35	1532	87.2	0.4279	0.536	1.195	1.172	2.12	C	0.0598	1.13	4.97×10^5	
6252	0.7195	148.7	404.0	887.5	43,746	80.97	1483	84.4	0.4110	0.514	1.045	1.127	2.05	F	0.0628	1.185	4.66×10^5	
6260	0.7329	149.45	410.8	902.5	41,618	52.36	1572	89.5	0.4207	0.526	1.17	1.495	2.73	F	0.0632	1.192	4.84×10^5	
6270	0.1897	148.0	273.1	600.6	25,645	229.3	5667	323	0.1452	0.182	0.405	0.2750	0.502	F	0.621	11.7	9.52×10^3	
6280	0.1985	151.50	33.15	72.74	25,646	16.49	5457	311	0.0515	0.0644	0.144	0.2124	0.388	F	0.430	8.11	1.03×10^4	
6290	0.2386	152.0	40.17	88.15	24,966	13.95	4793	273	0.0626	0.0782	0.1745	0.2900	0.529	F	0.315	5.94	1.47×10^4	
6300	0.548	144.75	120.6	266.1	41,315	9.796	1622	92.4	0.1830	0.229	0.511	1.007	1.84	F	0.193	3.64	1.73×10^5	
6311	0.7908	146.0	104.1	229.5	40,273	12.41	1699	96.6	0.2317	0.289	0.646	1.305	2.38	F	0.0819	1.545	7.08×10^5	
6312	0.7221	145.9	220.4	485.7	43,120	27.12	1506	85.6	0.3069	0.384	0.856	1.267	2.31	F	0.0914	1.725	4.75×10^5	
6320	0.2053	150.15	457.9	1006	43,405	104.8	3127	178	0.1956	0.244	0.546	0.4724	0.861	F	0.349	6.59	1.70×10^4	
6330	0.2160	151.5	184.6	405.2	43,642	23.10	2980	170	0.1272	0.159	0.355	0.5150	0.940	F	0.334	6.30	1.88×10^4	
6340	0.5080	60.75	863.6	2190	49,179	473.3	1348	76.6	0.6602	0.825	1.84	0.9053	1.65	C	0.122	2.3	2.97×10^5	
6350	0.5181	60.05	530.2	1345	49,343	141.9	1323	75.4	0.5273	0.659	1.47	1.169	2.13	C	0.122	2.3	3.19×10^5	
6360	0.5186	60.30	646.8	1640	49,606	101.0	1318	75.0	0.5816	0.728	1.62	1.602	2.93	C	0.129	2.44	3.20×10^5	
6370	0.5038	56.2	653.2	1676	49,874	75.76	1324	75.4	0.5898	0.736	1.65	1.893	3.45	C	0.134	2.53	3.11×10^5	
6380	0.5113	58.75	246.0	633.4	48,785	21.50	1387	78.9	0.3568	0.446	0.996	1.712	3.12	F	0.145	2.74	3.02×10^5	
6420	0.2104	152.35	49.09	107.6	68,222	78.97	1949	111	0.0645	0.0806	0.180	0.2574	0.47	F	0.136	2.57	2.53×10^4	
6430	0.2004	150.10	62.12	136.6	67,584	63.16	2057	117	0.0711	0.0889	0.198	0.3418	0.624	F	0.202	3.81	2.30×10^4	
6440	0.2033	151.80	621.7	1364	68,004	201.5	2018	115	0.2256	0.282	0.629	0.3391	0.618	C	0.292	5.51	2.36×10^4	
6450	0.1888	146.85	470.2	1038	68,457	99.96	2143	122	0.1909	0.239	0.533	0.3861	0.705	F	0.2960	5.59	2.08×10^4	
6460	0.1810	147.75	406.9	895.7	68,743	69.95	2202	125	0.1734	0.217	0.484	0.4050	0.740	F	0.437	8.25	1.92×10^4	
6470	0.2119	149.4	613.4	1349	68,180	76.18	1938	110	0.2307	0.288	0.644	0.5588	1.02	F	0.300	5.66	2.61×10^4	
6480	0.2087	148.0	909.1	2001	67,762	86.80	1972	112	0.280	0.35	0.781	0.7050	1.29	F	0.284	5.36	2.53×10^4	
6490	0.4964	198.9	281	2683	40,612	617.6	1733	98.6	0.4917	0.615	1.37	0.5777	1.06	C	0.0846	1.60	1.01×10^5	
6500	0.5496	196.4	574.7	1206	38,324	138.4	1754	100.0	0.5488									

SN 3 DATA IDENTIFIED AS FILM OR INTERMITTENT CONTACT BOILING

(OPEN SYMBOL) AND CONTACT BOILING (CLOSED SYMBOL)

COMPARED WITH CURVES FROM HOT PLATE TESTS

FIGURE 7-16

which is also Equation 5-F in Appendix F. Figures 7-17 to 7-20 compare the data of Table 7-2 with Equation 7-25 (intermittent contact) in addition to Equation 7-16 (contact) and Equation 7-17 (film). As can be seen, more data carefully spaced would be desirable for a better comparison with the curves of Equations 7-16, 7-17, and 7-25. The boiler swirl wire geometry was used in the theoretical expressions.

Curves of θ and ψ as determined from the hot plate test (see Figure 7-6) are shown in Figure 7-21 as a function of $h_{fv} \mu_v / \Delta T k_v$. Figures 7-22 to 7-27 present the forced convection boiling data. The similarity between the curves of the static tests made on the hot plate (Figure 7-21) and those from the forced convection boiling tests is readily apparent. At qualities below 20%, θ approaches the value of a flat spheroid whereas at the higher qualities it approaches the value of a sphere. On each of the plots the computed value for $(h_{fv} \mu_v / \Delta T k_v)_{crit.}$ is also indicated. Again more data closely spaced would be desirable to yield better definition.

7.3 DROPWISE PRESSURE DROP CORRELATIONS

Pressure drop results from the SNAP II Boiling Research tests were subordinate to heat transfer data. As a result, a short (30 in.) test section length was utilized and relatively inaccurate pressure drop data were obtained. Nevertheless, this section presents the data obtained and compares it with the conventional Martinelli correlation and an analytical correlation herein derived and supplemented with experimental constants.

Martinelli and others (Ref 7-21, 7-22, 7-23 and 7-24) have developed the most general correlation available for calculating the pressure drop and density in a two-phase flowing system. Martinelli found in his original experiments that the two-phase flow pressure drop is essentially independent of the details of the flow as long as (1) the flow is steady and (2) the static pressure drop in the liquid is equal to the static pressure drop in the vapor, i.e., there are no radial pressure gradients in the flow. Essentially, it is assumed that an annular flow pattern exists, that is, a cylindrical core of vapor surrounded by an annular film of liquid on the wall.

Several years ago Thompson Ramo Wooldridge made a rigorous analysis of the annular type flow as applied to a film of mercury. Particular effort was expended to obtain a proper term for the interfacial shear stress at the vapor-liquid interface. The differential equations were solved on a digital computer and results were found to agree quite closely with the Martinelli correlation.

Intuitively, a greater pressure drop than predicted by the Martinelli correlation can be expected for a dropwise flow pattern due to the relatively large liquid surface exposed to the flowing vapor. This was found to be the case. A detailed mathematical and experimental treatment for computing pressure drop in dropwise flow as applied to mercury condensers can be found in Reference 7-25. In this reference a ratio defined as

$$\Phi \equiv \frac{\Delta p_{TP} \text{ (friction)}}{\Delta p \text{ (vapor only)}} \equiv \frac{\Delta p_{TP} \text{ (measured)} + \Delta p_{mom}}{\Delta p \text{ (vapor only)}} \quad \text{Eq. 7-26}$$

is used for correlation purposes.

COMPARISON OF SN 3 TEST DATA WITH EQUATIONS 7-16, 7-17, 7-25

COMPARISON OF SN 3 TEST DATA WITH EQUATIONS 7-16, 7-17, AND 7-25

FIGURE 7-18

COMPARISON OF SN 3 TEST DATA WITH EQUATIONS 7-16, 7-17, AND 7-25

FIGURE 7-19

COMPARISON OF SN 3 TEST DATA WITH EQUATIONS 7-16, 7-17, 7-25

HOT PLATE VALUES OF β AND γ AS DETERMINED FROM FIGURE 7-6

AND γ FOR SN 3 FORCED CONVECTION BOILING TESTS
(COMPARE WITH STATIC HOT PLATE BOILING TESTS OF FIGURE 7-21)

β AND γ FOR SN 3 FORCED CONVECTION BOILING TESTS.

COMPARE WITH STATIC HOT PLATE BOILING TESTS OF FIGURE 7-21

FIGURE 7-23

β AND ψ FOR SN 3 FORCED CONVECTION BOILING TESTS.
COMPARE WITH STATIC HOT PLATE BOILING TESTS OF FIGURE 7-22.

FIGURE 7-24

β AND ψ FOR SN 3 FORCED CONVECTION BOILING TESTS.
COMPARE WITH STATIC HOT PLATE BOILING TESTS OF FIGURE 7-21

FIGURE 7-25

β AND ψ FOR SN 3 FORCED CONVECTION BOILING TESTS.
COMPARE WITH STATIC HOT PLATE BOILING TEST OF FIGURE 7-21

Run No. Used
6230 6260
6240 6311
6251 6312
6252
 $\bar{X} = 74\%$
 $\bar{P} = 150$ psia
SN 3 Test Section $\frac{DG}{\mu_v} = 42,000$

Tube: 5/16 in. OD with 0.028 in. Wall

Wire: Pitch = 1.0 in.; Diameter = 0.028 in.

Material: Haynes 25

FIGURE 7-26

β AND ψ FOR SN 3 FORCED CONVECTION BOILING TESTS.
COMPARE WITH STATIC HOT PLATE BOILING TESTS OF FIGURE 7-21

$\Delta P_{\text{vapor only}}$ is the pressure drop as either computed or measured by test for the case of vapor only flowing in the test section. $\Delta P_{\text{TP friction}}$ is the measured pressure drop corrected for momentum changes for the case when two phases are flowing in the test section.

Table 7-1 contains pressure drop data for the test sections SN 1, SN 2 and B. The magnitude of the pressure drop for test section SN 1 was too small to measure accurately and is, therefore, not tabulated. In Figure 7-28, Φ for test section SN 2 is plotted versus the parameter

$$\left(\frac{DG_v}{\mu_v}\right)^{0.8} \frac{\mu_v}{\mu_t} \frac{c_t}{c_v} \frac{\alpha}{(1-\alpha)} \quad \text{Eq. 7-27}$$

This parameter is composed of the same variables found in the Martinelli correlating parameter for viscous-turbulent flow. If viscosity and liquid density effects are neglected for a given fluid and if the change in quality is one (once-through boilers or condensers), this parameter can be reduced to

$$\frac{DG_v}{\mu_v} \left(\frac{\mu_v}{10}\right)^{1.25} \quad \text{Eq. 7-28}$$

which has been used as the correlating parameter for "once-through" boilers and condensers in References 1-1, 7-19 and 7-25. Φ as measured for "once-through" boilers and condensers is a mean value defined as

$$\Phi \equiv \frac{\Delta P_{\text{TP}} \text{ (overall friction pressure drop)}}{\int_{Q_{(\alpha=0)}}^{L_{(\alpha=1)}} f c_v \frac{u_v^2}{2g_c} \frac{dL}{D}} \quad \text{Eq. 7-29}$$

whereas Φ as defined by Equation 7-26 is a local quantity.

Φ for the bare tube test section is correlated in Figure 7-29. The approximate value of the correlating parameter for the SNAP II prototype boiler and condenser is indicated. An average quality of 0.5 is used for "once-through" boilers and condensers when computing the parameter of Equation 7-27. The Φ value as estimated from Figure 7-29 for the indicated range of the SNAP II prototype condenser ($\Phi = 2.5$ to 3.0) agrees with the Φ values measured for "once-through" condensers (see Reference 7-19).

In Figure 7-28 is plotted a point computed from pressure drop data obtained during SNAP II prototype boiler tests. This point falls in with the data obtained from local test section SN 2, which is to be expected since the test section is geometrically the same as that used in the prototype boiler. In general, the Φ values for the bare tube (Figure 7-29) are greater than those for the SN 2 tube (Figure 7-28). This is attributed to the "shielding" effect produced by the swirl wire. When a spherical drop is adjacent to a cylindrical body (wire), its effective drag is less than its drag when all solid objects are removed. Figure 7-30 clearly illustrates this phenomenon by showing that the total drag of two wires in close proximity to one another always is considerably less than twice the drag of

FORCED CONVECTION BOILING PRESSURE DROP CORRELATION FOR TEST

SECTION SN 2 AND PROTOTYPE SNAP II BOILER USING $\bar{X} = 0.5$

FIGURE 7-28

FORCED CONVECTION BOILING PRESSURE DROP CORRELATION FOR BARE TUBE TEST SECTION (B)

RANGE OF ABSCISSA FOR SNAP II BOILER AND CONDENSER INDICATED

FIGURE 7-29

INTERFERENCE DRAG OF TWO WIRES (REF 7-28)

FIGURE 7-30

a single wire due to mutual interference. Since Φ is a measure of the contribution of the liquid phase to the pressure drop, the wire drag is not included in this quantity. The wire drag effect is included in the "vapor only" pressure drop term in Equation 7-26. Thus the influence of the wire is to lower the value of Φ .

The Martinelli correlation curve for two-phase, viscous-turbulent flow in a bare tube is plotted in Figure 7-29. As can be seen, the data indicate higher Φ values than the Martinelli curve. This bears out what is intuitively expected, namely, that droplets dispersed in vapor exert a greater drag than if the liquid existed as an annular film.

Figure 7-34 presents all of the SN 3 test section results of Table 7-2 in terms of $f\Phi$ versus the correlating parameter. Figure 7-35 shows the same data in terms of Φ . When computing Φ , the friction factor, f , was taken as a constant having a value of 0.053 for this test section (0.2565 in. ID, 1 in. pitch, and 0.028 in. wire diameter). The correlation of Figure 7-35 agrees well with that of Figure 7-28 which applies to test section SN 2. Figure 7-36 shows the influence of quality on Φ for a pressure level of 150 psia and a Reynolds number of 45,000 based on the total mass velocity. Since Φ represents the influence of the liquid phase on the pressure drop, it is to be expected that Φ should increase as the quality is decreased and the total flow rate and pressure are held constant.

Figures such as Figure 7-35 are empirical correlations which have their limitations; hence, care must be taken not to extrapolate into extreme conditions not encountered in the tests. A theoretical pressure drop analysis is desirable since it will enhance our understanding of the two-phase flow processes associated with liquid spheroids in vapor, thus increasing our capabilities in judiciously applying the correlations to design. The physical model as constructed for the heat transfer processes can be extended to include the pressure drop. In a dropwise flow regime the major contribution of the liquid phase to the pressure drop is through the drag exerted by the liquid spheroids on the flowing vapor (Ref. 7-25). If a swirl device is used, the liquid spheroids can be assumed to be held against the wall. Appendix C contains the mathematical analysis of such a flow pattern. Equation C-14, Appendix C, is presented as follows:

$$Cd_s = \frac{f(\Phi-1)}{\frac{1}{8} \frac{\rho_v}{\rho_l} \frac{k_s D}{k_v \delta_o} \frac{(1-\alpha)^{2/3}}{\alpha} Nu_s} \quad \text{Eq. 7-30}$$

Equation 7-30 affords a means of computing the drag coefficient for an individual liquid spheroid by substituting measured Φ values. Table 7-1 shows the drag coefficients for test section SN 2 as computed by means of Equation 7-30.* The effect of mutual interference caused by the closeness of other liquid spheroids on the drag coefficient can be expected to be a complicating factor. Reference 7-26 presents data dealing with this situation. Spheres having diameters of 0.41 and 0.21 cm are arranged in varying degree of packing on a wall. The individual drag coefficient is measured by passing air over the spheres. Figure 7-31 shows the data in terms of the drag coefficient versus the fraction of

* For simplicity, contact boiling is assumed for all data and $Nu_s = 0.8$ is substituted into Equation 7-30.

DRAG COEFFICIENT OF MULTIPLE SPHERES ON A WALL AS A FUNCTION
OF FRACTION OF TOTAL AREA COVERED BY SPHERES. CURVE SHOWS
MUTUAL INTERFERENCE EFFECT (REF 7-26)

FIGURE 7-31

DRAG COEFFICIENTS OF MERCURY DROPS AS COMPUTED BY EQUATION 7-30
COMPARED TO CURVE BASED ON DATA SHOWN IN FIGURE 7-19. DATA ARE FOR $x \leq 99\%$.

FIGURE 7-32

DRAG COEFFICIENTS OF SINGLE SPHERES, DISKS, AND DROPS
(REF 7-27)

FIGURE 7-33

VORTEX FORCED CONVECTION BOILING PRESSURE

DROP CORRELATION FOR TEST SECTION SN 3

FIGURE 7-34

VORTEX FORCED CONVECTION BOILING PRESSURE
DROP CORRELATION FOR TEST SECTION SN 3

FIGURE 7-35

SN 3 $\bar{h}$ PLOTTED AS A FUNCTION OF QUALITY FOR A PRESSURE OF 150 PSIA
AND A REYNOLDS NUMBER OF 45,000 BASED ON THE TOTAL MASS VELOCITY

FIGURE 7-36

the total surface area covered by the projected area of the spheres. Equation C-18, Appendix C, shown as follows:

$$TF = \frac{1}{32} \frac{(1-x)^{2/3}}{x} \quad Nu_6 \frac{D}{\delta_6} \frac{k_s}{k_v} \frac{\rho_v}{\rho_s} \quad \text{Eq. 7-31}$$

allows one to compute the fraction of the total surface area covered by the projected area of flowing liquid drops. Since,

$$Cd_6 \text{ (solid spheres)} = F(TF) \quad \text{(curve of Figure 7-31)} \quad \text{Eq. 7-32}$$

and

$$TF \text{ (liquid spheres)} = F' \left(\alpha, \frac{\rho_v}{\rho_s} \right) \quad \text{Eq. 7-33}$$

(Equation 7-31 assuming $\frac{k_s}{k_v}, \frac{D}{\delta_6}$ and Nu_6 to be constants).

Therefore,

$$Cd_6 \text{ (liquid spheres)} = F'' \left(\alpha, \frac{\rho_v}{\rho_s} \right) \quad \text{Eq. 7-34}$$

by combining Equations 7-32 and 7-33 and assuming Cd_6 (solid spheres) equals Cd_6 (liquid spheres). The functional relationship between Cd_6 and TF as defined by Equation 7-32 and Figure 7-31 but in terms of x , and ρ_v/ρ_s is plotted in Figure 7-32 together with measured drag coefficients of liquid drops flowing in test section SN 2 (see Table 2-1). As can be seen, the magnitudes of the measured drag coefficients agree with those measured in Reference 7-26 and shown in Figure 7-32 which indicates an understanding to a certain degree of dropwise flow mechanics. The scatter of the data in Figure 7-32 is, of course, either due to inaccuracies in the postulated physical model or in the physical measurements; the cause of scatter is at present subject to speculation.

Figure 7-33 shows drag coefficients for individual liquid spheroids (Ref. 7-27) without mutual interference but indicating the effect of spheroidal deformation. Comparing the magnitudes of the drag coefficients in Figure 7-33 with those in Figure 7-32, one readily sees that the effect of mutual interference of drops and swirl wire is to reduce the drag coefficient by a factor of 3 or 4.

7.4 STATISTICAL COMPARISON BETWEEN HEAT TRANSFER THEORY AND DATA

Section 7.2 correlated test results on the basis of the mathematical model postulated for the vortex forced convection boiling of mercury. To give this comparison more meaning, it should be verified on the basis of a heat flux (Q/A) versus mean temperature difference (ΔT_m) plot.

Heat flux and temperature difference curves for SN 3 test results could be plotted with various combinations of pressure level, flow rate (or mass velocity), and average quality level. These data are plotted in Figures 7-17, 7-18, 7-19, and 7-20. The theoretical prediction for the heat flux versus temperature difference curves at the same combinations of average quality, pressure level, and mass flow rate are also plotted in these figures. The theoretical curves vary in these figures because various values were chosen for the constants used in the theoretical equations. Section 8 discusses the ranges of variation of these constants as they would be applied for determination of heat flux.

For determining the theoretical heat flux curves, an initial drop size of 0.0625 in. was assumed as this was the wire size in the test rig boiler and therefore is the determining factor in formation of initial drop size. Note that relatively good correlation was obtained in the contact and film regions. This was expected because of the more uniform and steady state aspects of these two boiling regimes. Quite a bit of scatter was found to exist in the intermittent contact regimes because this is an inherently unstable and unsteady regime. The "goodness of fit" between the theoretical and observed values of Q/A at a given ΔT_m for Figures 7-17, 7-18, 7-19, and 7-20 was statistically analyzed. A χ^2 test was performed on the hypothesis that the relative error between the theory and measurement was normally distributed. On the basis of this test it was shown that the errors of measurement described in Section 5 and summarized by Figure 5-1 accounted for very little of the variation between theory and observation.

The following additional conclusions about the ability of the theory to predict heat flow rates were reached:

1. The probability that the theory and data differ by more than $\pm 10\%$ is 82%.
2. The probability that they differ by more than $\pm 20\%$ is 65%.
3. The relative error was less than $\pm 72.5\%$ for 90% of the data.

For comparison purposes, the accuracy of correlations obtained in other heat transfer experiments was $+38\%$, -28% for NaK in pool boiling and $\pm 20\%$ for low quality wetting boiling of mercury (Ref 1-2). The accuracy obtained is quite reasonable when one considers that three boiling regimes were covered. Modification of the theory on the basis of more thorough testing would greatly improve the theoretical accuracy.

8.0 PRECAUTIONS IN APPLICATION OF THEORY

Little information is available in the literature about heat transfer to evaporating mercury in dropwise flow. The material covered in this report advances the state of the art in this area of heat transfer. The macroscopic heat transfer characteristics of a multitude of vaporizing mercury drops in swirling motion in a tube with flowing vapor are similar to heat transfer characteristics of a single drop or of a pool of liquid on a heated surface. In a vaporizing drop, however, vapor is ventilated radially outward from underneath the drop whereas in a vaporizing pool of mercury the vapor nucleates and passes through the liquid. As should be expected, the magnitude of the heat transfer rate depends on the flow parameters such as the strength of the acceleration field and the vapor mass velocity, as well as on thermodynamic properties such as density and quality.

Figure 8-1 shows schematically a typical boiling curve for the vortex forced convection boiling of mercury in dropwise flow. The curve applies to a given set of conditions as noted. Four boiling regimes are identified, three of which apply to vortex dropwise boiling, namely contact, intermittent contact, and film which are analogous to nucleate, partial film and film boiling in a pool of liquid. The appropriate analytical equation developed for each regime is identified by equation number. A boiling curve for any set of flow and thermodynamic conditions can be constructed by means of these equations. The intersections of the equations define the termination of a regime or the beginning of a new boiling regime. The detailed method of applying these curves to design is presented below.

The dotted curve applies to dropwise boiling in 1 g. Here the strength of the acceleration field due to swirl and the size of the liquid globules do not produce vortex boiling. These conditions are found near the liquid interface (preheater-boiler demarcation) when the quality and vapor velocities are low in magnitude. The transition from this regime into vortex boiling is computed by means of a Weber number type parameter which defines the stability of a liquid globule moving along the bottom of a tube in the presence of interfacial shear caused by flowing vapor. Equation 6-6 defines the point of transition. Figure 8-1 shows the progressive flow patterns starting at the point of incipient breakup of the liquid emerging from the preheat section of the boiler.

Since the heat transfer rate depends on both the centrifugal force due to swirl and the temperature difference, a boiler design procedure requires an adequate pressure drop correlation so that the available pressure drop for increasing the heat transfer rate is fully utilized. Pressure measurements were always recorded concurrently with the temperature measurements in the tests reported. Figure 7-35 presented a typical pressure drop correlation in terms of the appropriate flow and thermodynamic parameters. The Φ term represents the ratio of the two-phase frictional pressure drop to the pressure drop that would result if the vapor phase alone were flowing in the tube. The "vapor phase alone" pressure drop is readily computable using conventional, well-established methods.

Heat transfer results were presented in a theoretical as well as in an empirical fashion, thus affording a choice of a design approach. The theoretical approach consists of verifying or modifying theoretical equations by means of test. The empirical method consists of

TYPICAL MERCURY FORCED CONVECTION VORTEX BOILING HEAT FLUX
VERSUS ΔT CURVE (FOR A GIVEN MASS VELOCITY, PRESSURE, AND QUALITY)

FIGURE 8-1

curves of Q/A versus ΔT for which the various flow and thermodynamic properties are held constant. These curves limit the ability to extrapolate to unknown conditions.

As is usually the case, there are limitations to the extent of application of both the test results and the theory. These are summarized as follows:

1. In the low quality section of the boiler where the liquid mercury is broken up and the vortex flow is established, the 1 g environment plays an important role since the dynamic forces $\frac{1}{2} \rho u^2$ are not $\gg \rho g D$. The behavior in zero gravity is thus difficult to assess. The lack of gravity may appreciably lengthen this region. The flow establishment region can be minimized by stepping the tube and producing higher vapor velocities.
2. Normally, the major portion of the heat transfer takes place directly between the liquid drops and the heated wall. The heat flow from the wall to the vapor and then to the liquid plays a minor role except in the high quality region where the droplets are very small. Equations 7-16 and 7-18 both indicate that the Nusselt number is zero when the quality is one. It seems reasonable to believe that near a quality of one, the heat flow from the wall to the vapor and then to the liquid becomes important. This heat flow path is ignored in the formulation of Equations 7-16 and 7-17; hence these equations become inaccurate as the forced convection boiling process approaches completion. It should be noted, however, that these equations give conservative results and that the actual Nusselt number is greater than the calculated one. Analyses show that when the quality approaches 98%, the heat transfer rates are about equal for the two paths.
3. Non-wetting between the liquid and the wall is tacitly postulated in the formulation of Equations 7-16 and 7-17 as well as in the interpretation of the data. No visual verification of this could be made during the test runs. During dropwise film boiling the wetting phenomenon will have no effect since the drop is separated from the wall by a supporting vapor film. However, dropwise contact boiling, which is envisioned as a spherical drop in point contact with the wall, will not exist during wetting. A liquid film in contact with the wall vaporizing at the liquid-vapor interface is a more probable hypothesis. If the liquid film were detached from the wall by a vapor film generated at the wall, it would roll into drops. It is expected that heat transfer rates will be greater for a continuous film in contact with the wall than for scattered droplets in contact boiling.

4. The role of the swirl wire insert in establishing the initial drop size during forced convection boiling is not firmly established. Although the size of the wire insert was changed in the local test section, the rig boiler wire insert was not changed. Since it is believed that the initial drop sizes are formed in the rig boiler near the preheater interface and not in the local test section, no effect in heat transfer should be detected. No effect in heat transfer was detected. Equations 7-16 and 7-17 contain the initial drop sizes which are assumed equal to the wire diameter based on stability considerations. This is still a tentative supposition.

9.0 RECOMMENDATIONS

To obtain refined space mercury Rankine cycle boiler design information, the following additional work is recommended:

1. The theoretical and empirical equations should be tested by further forced convection local type testing. In particular, higher mass velocities and pressure levels and variations in swirl device geometry should be investigated. The phenomena are fairly complex, however, and many data points are required.
2. In obtaining further data, care must be taken in the control of drop size. The rig boiler should therefore have the same swirl wire geometry as the test section.
3. Small-scale (probably adiabatic) experiments are needed to establish the effect of "g" force on drop shape, to study drop size establishment and means of control, and to define more accurately the flow conditions at which the globules are broken up.
4. Hot plate experiments should be performed with mercury drops in a mercury vapor environment. The detailed mechanics of the phenomena should be observed, with particular attention to the intermittent contact regime.

10.0 NOMENCLATURE

a	Wall thickness
B	Constant in Equations A-9, A-10
c_p, c_v	Specific heat at constant pressure and constant volume, respectively
C	Constant
C_d	Drag coefficient
d	Wire diameter (see Figure 7-30) and differential
D	Tube diameter
f	Friction factor $\left(\text{from } \Delta P = f \frac{\Delta L}{D} \frac{G^2}{2g_c \rho_v}\right)$
F	Function
Fr	Froude number, $\frac{u^2}{gL}$
G	Mass velocity (ρu)
g	Acceleration
g_c	Standard acceleration of gravity at sea level
h	Heat transfer coefficient based on tube wall area
h_{ev}	Latent heat
$\bar{h}$	Mean heat transfer coefficient
K	Constant in Equations A-9, A-10
k_f	Liquid conductivity
k_m	Wall conductivity
k_v	Vapor conductivity
L	Length
M	$\left(\frac{DG_v}{\mu_v}\right)^{0.8} \frac{\mu_v}{\mu_f} \frac{\rho_f}{\rho_v} \frac{n}{(1-n)}$ or constant in Equations A-9 and A-10
$\dot{m}$	Mass flow rate (usually total)

m	Mass and constant in Equation A-9
n	g/g_c
N	Number of spheres per unit of surface area
Nu	Nusselt number
p, P	Pressure, period
Q	Heat transfer
q	Wall thickness, heat flux
R_i, R_t	Geometric ratios
Re	Reynolds number, $\frac{GD}{\mu}$
S	Spacing (see Figure 7-30)
S	Wave amplitude growth rate number, dimensionless
t	Wall thickness, tangential
T	Temperature
u'	Velocity of vapor ejected from under drop (see Figure A-1)
u	Velocity
v	Vapor specific volume
W	Velocity of drop normal to wall (see Figure A-1)
We	Weber number $\frac{\rho u^2 L}{\sigma g_c}$
x	Quality, vapor weight/total weight
y	Distance of drop from wall (see Figure A-1)

GREEK SYMBOLS

α Swirl wire angle

$\theta = \frac{Nu_\delta \text{ (actual)}}{Nu_\delta \text{ (film - spheres)}}$ for definition of θ' see Section 7.2.2

TF	Fraction of total area covered by spheres
λ	Wave length
Γ_v	$\frac{h \text{ measured}}{h \text{ vortex film}}$
Γ_c	$\frac{h \text{ Measured}}{h \text{ vortex contact}}$
Γ_p	$\frac{h \text{ measured}}{h \text{ pool}}$
Δ	Differential (e.g. $\Delta T =$ temperature difference)
δ	Drop diameter
δ_0	Initial drop diameter and swirl wire size (see Section 7.1.2)
ϵ	Slip ratio equal to u_p/u_v
Θ	Time
μ	Viscosity
ρ	Density
σ	Surface tension
γ_s	Shear force exerted by wall on drop divided by $\frac{\pi}{4} \delta^2$
γ_v	Shear stress exerted by wall on vapor
$\bar{f}$	$\frac{\Delta p_{TP} \text{ (friction)}}{\Delta p_v}$
γ	$\frac{Nu_{\delta} \text{ (actual)}}{Nu_{\delta} \text{ (contact)}}$ (for definition of γ' see Section 7.2.2)

SUBSCRIPTS

δ	Drop
v	Vapor
f	Liquid
o	Initial

FV	Liquid to vapor
SAT.	Saturated
W	Wall
m	Metal, mean
mom	Momentum
CRIT.	Critical
TP	Two-phase
E	Exit
t	Total

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APPENDICES

A. DYNAMIC ANALYSIS OF A DROP IN FILM EVAPORATION ON HOT PLATE*

The momentum equation applied to a spherical drop supported by its vapor film and acted upon by an externally induced forced field n times that of gravity is:

$$\frac{\Delta P \pi \delta^2}{4} - \frac{\pi \delta^3 \rho_f n}{6} - C_{d\delta} \frac{\pi \delta^2 \rho_v}{4} \frac{W^2}{2g_c} = \frac{\pi \delta^3 \rho_f}{6} \frac{dW}{g_c d\Theta} \quad \text{Eq. A-1}$$

Equation A-1 applies in the y direction. The transient term is assumed negligible which in effect assumes the mass of the drop, m_s , to remain constant with time.

Also, from the momentum equation in the x -direction,

$$\Delta P \pi \delta y = \frac{\dot{m}_v u'}{g_c} \quad \text{Eq. A-2}$$

(See Eq. 1-B)

Continuity gives,

$$\pi \delta y u' \rho_v = \dot{m}_v \quad \text{Eq. A-3}$$

A heat balance assuming heat is conducted across the film only gives,

$$\dot{m}_v h_{fv} = \frac{k_v}{y} \frac{\pi \delta^2}{4} \Delta T \quad \text{Eq. A-4}$$

(See Eq. 3-B)

Assume:

$$C_{d\delta} = \frac{24}{\left(\frac{\delta \rho_v W}{\mu_v} \right)} \quad \text{Eq. A-5}$$

Note that:

$$W = \frac{dy}{d\Theta}, \quad \frac{dW}{d\Theta} = \frac{d^2y}{d\Theta^2}$$

Combining Equations A-1, A-2, A-3, A-4, and A-5 we obtain:

$$\frac{\delta}{\rho_v g_c 4} \left(\frac{k_v \Delta T}{4 h_{fv}} \right)^2 \frac{1}{y^3} = \frac{\rho_f n y}{6} + \frac{3 \mu_v y}{\delta^2 g_c} \frac{dy}{d\Theta} + \frac{\rho_f y}{6 g_c} \frac{d^2y}{d\Theta^2} \quad \text{Eq. A-6}$$

which is a non-linear, second order differential equation.

Equation A-6 can be converted to an equation for a "spring-mass-damping" system if the

*(See Figure A-1)

coefficients are considered constant by assuming small perturbations of y and

$$y = \frac{k_v}{h_\delta} = \frac{k_v}{\left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{\delta}\right)^{1/4} \left(\frac{k_v 4 h_{ev}}{\Delta T}\right)^{1/2}} \quad \text{Eq. A-7}$$

where h_δ is given by Equation B-7 for a drop evaporating on a hot plate. Substituting into Equation A-6 and rearranging terms we have:

$$n g_c = \frac{g_c n}{k_v} \left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{\delta}\right)^{1/4} \left(\frac{k_v 4 h_{ev}}{\Delta T}\right)^{1/2} y + \frac{18 \mu_v}{\rho_f \delta^2} \frac{dy}{d\theta} + \frac{d^2 y}{d\theta^2} \quad \text{Eq. A-8}$$

Equation A-8 is an equation for an oscillatory element of the form:

$$m = k y + B \frac{dy}{d\theta} + M \frac{d^2 y}{d\theta^2} \quad \text{Eq. A-9}$$

Equation A-9 is no longer oscillatory when:

$$B^2 > 4 k M \quad \text{Eq. A-10}$$

Using Equation A-10 as a stability criterion, one can predict the conditions under which the drop will oscillate and is therefore likely to intermittently touch the wall. For intermittent contact boiling then:

$$\left(\frac{18 \mu_v}{\rho_f \delta^2}\right)^2 > 4 \frac{g_c n}{k_v} \left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{\delta}\right)^{1/4} \left(\frac{k_v 4 h_{ev}}{\Delta T}\right)^{1/2}$$

or

$$\left(\frac{18 \mu_v}{\rho_f \delta^2}\right)^2 > 4 \frac{g_c n}{k_v} h_\delta \quad \text{Eq. A-11}$$

Substituting

$$Nu_\delta = \frac{h_\delta \delta}{k_f}$$

into Equation A-11 we have:

$$\frac{1}{g_c} \frac{k_v}{k_f} \left(\frac{g \mu_v}{\rho_f}\right)^2 > \delta^3 n Nu_\delta \quad \text{Eq. A-12}$$

The terms on the left are essentially constant; thus increases in the imposed force field or drop diameter, δ , reduce the stable drop Nusselt number. (See Equations 7-14 and 7-23 in the main text for discussion.)

SKETCH FOR DYNAMIC ANALYSIS OF DROP
SUPPORTED ON VAPOR FILM

Figure A-1

B. DERIVATION OF LOCAL HEAT TRANSFER COEFFICIENTS FOR VORTEX FORCED CONVECTION DROPWISE BOILING

Mechanics of Drop Heat Transfer

The following analysis is restricted to small drops resting on a heated plate. Small mercury drops are spherical in shape and are relatively stable to interfacial disturbance. As a result, a hydrodynamic wave type stability and heat transfer analysis is not required. Large drops are not spherical in shape and are relatively unstable due to the large ratios of body to surface tension forces to which they may be subjected. Since the analysis is for (1) a stationary drop, (2) on an open hot plate, (3) in a one- g gravity field, the results must be extrapolated into (a) a variable body force field of greater than one- g , (b) a moving drop, (c) an environment in which the drop is surrounded by its own vapor. These are the conditions to which a non-wetting mercury droplet is subjected in a forced convection mercury boiler with a swirl insert for forcing the mercury droplets toward the heated wall.

The assumption is made that a vapor layer separates the liquid drop from the heating surface. The momentum of the ejected vapor from the underside of the drop "builds up" a pressure which supports it. This action is similar to the "air car" which rides on a layer of air. The application of the momentum, energy, and continuity equations follows:

DROP ON A HOT PLATE
SUPPORTED BY A VAPOR FILM

Figure B-1

Momentum:

$$\Delta P \pi \delta y = \frac{\dot{m}_v u'}{g_c} \quad \text{Eq. B-1}$$

$$\frac{\Delta P \pi \delta^2}{4} = \frac{\pi \delta^3}{6} \rho_f n \quad \text{Eq. B-2}$$

where $\Delta P = (\text{mean pressure under drop} - \text{ambient pressure}).$

Energy:

$$\dot{m}_v h_{fv} = \frac{k_v}{y} \frac{\pi \delta^2}{4} \Delta T \quad \text{Eq. B-3 A}$$

$$\frac{k_v}{y} = h_\delta \quad \text{Eq. B-3 B}$$

where $\Delta T = (\text{wall temperature} - \text{saturation temperature})$.

Continuity:

$$\pi \delta y u' \rho_v = \dot{m}_v \quad \text{Eq. B-4}$$

Combining Equations B-1 and B-2 results in:

$$\left(\frac{\rho_c}{\rho_v} \frac{2}{3} \delta \rho_f \eta \right)^{1/2} = u' \quad \text{Eq. B-5}$$

Combining Equations B-3 and B-4 results in:

$$(\pi \delta y u' \rho_v) h_{fv} = \frac{k_v}{y} \frac{\pi \delta^2}{4} \Delta T \quad \text{Eq. B-6}$$

Combining Equations B-5 and B-6 finally give:

$$h_\delta = \left(\rho_v \rho_f \rho_c \frac{2}{3} \frac{\eta}{\delta} \right)^{1/4} \left(\frac{k_v 4 h_{fv}}{\Delta T} \right)^{1/2} \quad \text{Eq. B-7 A}$$

To account for intermittent contact of the drop with the wall a factor β can be utilized.

Thus:

$$h_\delta = \beta \left(\rho_v \rho_f \rho_c \frac{2}{3} \frac{\eta}{\delta} \right)^{1/4} \left(\frac{k_v 4 h_{fv}}{\Delta T} \right)^{1/2} \quad \text{Eq. B-7 B}$$

The mean heat transfer coefficient to completely vaporize a drop is:

$$h_{\delta} = \frac{\text{CONST.}}{\delta^{1/4}} \quad , \quad (\Delta T = \text{CONST.})$$

INSTANTANEOUS DROP HEAT
TRANSFER COEFFICIENT
VERSUS DROP DIAMETER

Figure B-2

$$\bar{h}_{\delta} = \int_0^{\delta_0} \frac{h_{\delta} d(\delta)}{\delta_0}$$

$$\bar{h}_{\delta} = \frac{\text{CONST.}}{\delta_0} \int_0^{\delta_0} \frac{d(\delta)}{\delta^{1/4}}$$

$$\bar{h}_{\delta} = \frac{\text{CONST.}}{\delta_0} \frac{4}{3} \delta_0^{3/4}$$

$$\bar{h}_{\delta} = \text{CONST.} \frac{4}{3} \frac{1}{\delta_0^{1/4}}$$

and using Equation B-7a

$$\bar{h}_{\delta} = \frac{4}{3} \left(\rho_v \rho_l g_c \frac{2}{3} \frac{n}{\delta_0} \right)^{1/4} \left(\frac{k_v 4 h_{fv}}{\Delta T} \right)^{1/2} *$$

Eq. B-8 A

Also

$$h_{\delta_0} = \frac{\text{CONST.}}{\delta_0^{1/4}}$$

$$\bar{h}_{\delta} = \frac{\text{CONST.}}{\delta_0^{1/4}} \frac{4}{3}$$

$$\frac{\bar{h}_{\delta}}{h_{\delta_0}} = \frac{4}{3}$$

Eq. B-8 B

* Integrating with respect to time gives a constant of 5/4 instead of 4/3 as shown above.

Vaporization of Small Drops by Forced Convection

$$h_{fv} \frac{dm}{d\theta} = h_s \frac{\pi \delta^2}{4} \Delta T \quad (\text{heat balance})$$

$$m = \frac{\pi \delta^3}{6} \rho_f$$

$$dm = \frac{\pi}{2} \rho_f \delta^2 d\delta$$

$$h_{fv} \frac{\pi}{2} \rho_f \delta^2 \frac{d\delta}{d\theta} = h_s \frac{\pi \delta^2}{4} \Delta T$$

$$h_s = -2 h_{fv} \frac{\rho_f}{\Delta T} \frac{d\delta}{d\theta} \quad \text{Eq. B-8C}$$

In a tube in which the drop is moved by the vapor, the resident time is related to the tube length by,

SLIP FLOW OF DROP IN A TUBE
WITH VAPOR FLOWING

$$d\theta = \frac{dL}{u\epsilon}$$

ϵ is the slip ratio which
is less than one.

Figure B-3

$$h_s = -2 h_{fv} u \epsilon \frac{\rho_f}{\Delta T} \frac{d\delta}{dL} \quad \text{Eq. B-9}$$

The over-all heat transfer coefficient of the tube is,

$$G \propto \frac{\pi D^2}{4} h_{fv} \equiv h \pi D L_t \Delta T$$

For exit quality of one,

$$h = \frac{G}{4} \frac{D}{L_t} \frac{h_{fv}}{\Delta T} \quad \text{Eq. B-10A}$$

Combining Equation B-9 and B-7A

$$-\frac{2 h_{fv} \rho_t u \epsilon}{\Delta T} \frac{d\delta}{dL} = \beta \left(\rho_v \rho_t g_c \frac{2}{3} \frac{n}{\delta} \right)^{1/4} \left(\frac{k_v 4 h_{fv}}{\Delta T} \right)^{1/2} \quad \text{Eq. B-10B}$$

where β is the ratio of measured heat transfer coefficient to drops to theoretical heat transfer coefficient.

Also,

$$G \propto u \rho_v$$

$$n = \left(\frac{\epsilon G \propto}{\rho_v} \tan \alpha \right)^2 \frac{2}{D g_c}$$

SWIRL WIRE FOR ACCELERATING DROP

Figure B-4

Combining:

$$-\frac{2 h_{fv} \rho_t G \propto}{\Delta T \rho_v} \frac{d\delta}{dL} = \beta \left(\rho_v \rho_t g_c \frac{2}{3} \frac{2}{\delta D g_c} \right)^{1/4} \left(\frac{\epsilon G \propto \tan \alpha k_v 4 h_{fv}}{\rho_v \Delta T} \right)^{1/2}$$

$$-\left(\frac{h_{fv}}{\Delta T k_v} \right)^{1/2} \left(\frac{G \propto}{\tan \alpha} \right)^{1/2} \left(\frac{\rho_t}{\rho_v} \right)^{3/4} \frac{d\delta}{dL} = \beta \left(\frac{4}{3} \frac{1}{\delta D} \right)^{1/4} \quad \text{Eq. B-11}$$

The slip ratio can be estimated as follows:

Assume laminar shear and number. $C_d = \frac{24}{Re}$ for small spheres and low Reynolds

MOVING DROP SHOWING
ASSUMED VELOCITY PROFILE

$$D_{RAG} = \mu_v \frac{du}{dy} \text{ (AREA)}$$

(Drag = Shear)

Figure B-5

$$\frac{\pi \delta^2}{4} C_d \rho_v \frac{(u - u\epsilon)^2}{2g_c} = \frac{\mu_v}{g_c} \frac{u\epsilon}{k_v/h_\delta} \frac{\pi \delta^2}{4}$$

The reference area for the drag force and the shear force is selected as $\frac{\pi \delta^2}{4}$

$$C_d \rho_v \frac{(u - u\epsilon)^2}{2} = \mu_v \frac{u\epsilon}{k_v/h_\delta}$$

$$\frac{24 \rho_v}{\delta \rho_v \frac{(u - u\epsilon)}{\mu_v}} \frac{(u - u\epsilon)^2}{2} = \mu_v \frac{u\epsilon}{k_v/h_\delta}$$

$$\frac{12}{\delta} (u - u\epsilon) = \frac{u\epsilon}{k_v/h_\delta}$$

$$\frac{1 - \epsilon}{\epsilon} = \frac{1}{12} \frac{h_\delta \delta}{k_v}$$

$$\frac{1}{\epsilon} - 1 = \frac{1}{12} \frac{h_\delta \delta}{k_v}$$

If the slip ratio $\epsilon \ll 1$ then

$$\epsilon = \frac{12 k_v}{h_\delta \delta}$$

Eq. B-12A

Combining Equations B-12A and B-7A

$$\epsilon = \left(\frac{12 k_v}{\delta \rho} \right) \left(\rho_v \rho_f g_c \frac{2}{3} \frac{n}{\delta} \right)^{-1/4} \left(\frac{k_v 4 h_{fv}}{\Delta T} \right)^{-1/2} \quad \text{Eq. B-12 B}$$

Substituting

$$n = \left(\frac{\epsilon G \alpha \tan \alpha}{\rho_v} \right)^2 \frac{2}{D g_c} \quad \text{Eq. B-13 A}$$

$$\epsilon^{1/2} = \frac{G^{1/3}}{\rho^{1/3} \delta^{1/4}} \left(\frac{\rho_v 3 D}{\rho_f 4} \right)^{1/2} \left(\frac{k_v \Delta T}{G \alpha \tan \alpha h_{fv}} \right)^{1/6} \quad \text{Eq. B-13 B}$$

Substituting Equation B-13B into B-11, we have,

$$\begin{aligned} - \left(\frac{h_{fv}}{\Delta T k_v} \right)^{1/2} \left(\frac{G \alpha}{\tan \alpha} \right)^{1/2} \left(\frac{\rho_f}{\rho_v} \right)^{3/4} \frac{d\delta}{dL} \left(\frac{G^{1/3}}{\rho^{1/3} \delta^{1/4}} \right) \left(\frac{\rho_v 3 D}{\rho_f 4} \right)^{1/2} \left(\frac{k_v \Delta T}{G \alpha \tan \alpha h_{fv}} \right)^{1/6} &= \rho \left(\frac{4}{3 \delta D} \right)^{1/4} \\ - \left(\frac{h_{fv} G \alpha}{\Delta T k_v} \right)^{1/3} \left(\frac{\rho_f}{\tan \alpha \rho_v} \right)^{2/3} \left(\frac{9 D}{2} \right)^{1/3} \frac{1}{\rho^{4/3}} \frac{d\delta}{dL} &= 1 \end{aligned} \quad \text{Eq. B-14}$$

By definition the local heat transfer coefficient is equal to:

$$h \equiv \frac{GD}{4} \frac{h_{fv}}{\Delta T} \frac{d\alpha}{dL} \quad \text{Eq. B-15}$$

Also:

$$\alpha \equiv \frac{\frac{\pi \delta_o^3}{6} - \frac{\pi \delta^3}{6}}{\frac{\pi \delta_o^3}{6}} = 1 - \left(\frac{\delta}{\delta_o} \right)^3 \quad \text{Eq. B-16}$$

and

$$\delta = \delta_o (1 - \alpha)^{1/3} \quad \text{Eq. B-17}$$

$$\frac{d\alpha}{d\delta} = - \frac{3}{\delta_o} (1 - \alpha)^{2/3} \quad \text{Eq. B-18}$$

Combining Equations B-14, B-15, B-17, and B-18 and constructing the Reynolds and Nusselt numbers we obtain:

$$\frac{hD}{k_v} = \frac{6^{1/3} (1-\alpha)^{2/3}}{4 \alpha^{1/3}} \frac{D}{\delta_0} \beta^{4/3} \left(\frac{\tan \alpha \rho_v}{\rho_f} \right)^{2/3} \left(\frac{DG}{\mu_v} \right)^{2/3} \left(\frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \quad \text{Eq. B-19}$$

Equation B-19 gives the local Nusselt number in terms of local temperature difference, ΔT , local contact factor, β , and local quality, α . All other quantities are assumed to be approximately independent of tube length.

For contact dropwise boiling the drop Nusselt number is constant (See Figures 7-4 and 7-5). By definition:

$$Nu_\delta \equiv \frac{h_\delta \delta}{k_f} \quad \text{Eq. B-20}$$

Also by heat balance the drop heat transfer coefficient is:

$$-2 h_{fv} u \in \frac{\rho_f}{\Delta T} \frac{d\delta}{dL} = h_\delta \quad \text{Eq. B-21}$$

(See Equations 7-29 and 7-31
Reference 1-1)

Also

$$u \rho_v = G \alpha \quad \text{Eq. B-22}$$

Combining Equations B-20, B-21, B-22, B-15, B-17, and B-18 we obtain:

$$\frac{hD}{k_v} = \frac{3}{8} Nu_\delta \frac{k_f}{k_v} \left(\frac{D}{\delta_0} \right)^2 \frac{1}{\epsilon} \frac{(1-\alpha)^{1/3}}{\alpha} \frac{\rho_v}{\rho_f} \quad \text{Eq. B-23}$$

The slip ratio for contact boiling can be evaluated if we assume that

$$\epsilon_{\text{CONTACT}} = \epsilon_{\text{FILM}} \quad \text{at the point where}$$

$$h_{\delta \text{CONTACT}} = h_{\delta \text{FILM}}$$

For film boiling:

$$\epsilon = \frac{12 k_v}{h_\delta \delta} \quad (\text{See Equation 7-32, Ref. 1-1}) \quad \text{Eq. B-24}$$

Combining Equations B-24 and B-20 as applied to contact boiling we have:

$$\epsilon_{\text{CONTACT}} = \frac{12 k_v}{k_f \text{Nu}_{\delta \text{CONTACT}}} \quad \text{Eq. B-25}$$

Substituting Equation B-25 into Equation B-23 and eliminating the slip ratio we have:

$$\frac{hD}{k_v} = \frac{1}{32} \text{Nu}_{\delta \text{CONTACT}}^2 \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{\delta_o} \right)^2 \frac{(1-x)^{1/3}}{x} \frac{\rho_v}{\rho_f} \quad \text{Eq. B-26}$$

Equation B-26 gives the local Nusselt number in terms of local quality, x . All other quantities are assumed to be approximately independent of tube length.

C. PRESSURE DROP DURING VORTEX FORCED CONVECTION DROPWISE BOILING

The frictional pressure drop across the element dL in Figure C-1 is due to the wall shear stress induced by both the flowing droplets and vapor. All of the liquid droplets are assumed to be held against the wall by a swirl device. A force balance gives:

$$dP_{fp} \frac{\pi D^2}{4} = \tau_v \pi D dL + \tau_f N \pi D dL \frac{\pi \delta^2}{4} \quad \text{Eq. C-1}$$

The total projected area of liquid on the wall is assumed small compared to the area in contact with the vapor. τ_v is the wall shear stress due to the flowing vapor and includes the drag effect of any wire insert. τ_f is the wall shear stress due to the presence of the liquid phase (drops), and N is the drop population in terms of number of drops per unit of surface area.

By definition

$$dP_v \frac{\pi D^2}{4} \equiv \tau_v \pi D dL \quad \text{Eq. C-2}$$

and

$$\Phi \equiv \frac{dP_{fp}}{dP_v} \quad \text{Eq. C-3}$$

also

$$\rho_f \frac{\pi \delta^3}{6} N u_v \pi D = \dot{m}_f \quad \text{Eq. C-4}$$

SCHEMATIC DIAGRAM SHOWING WALL SHEAR STRESS DUE TO FLOWING VAPOR AND LIQUID DROPS

Figure C-1

and

$$dP_v = f \frac{dL}{D} \frac{G_v^2}{2g_c \rho_v} \quad \text{Eq. C-5}$$

$$\dot{m}_v = G_v \frac{\pi D^2}{4} \quad \text{Eq. C-6}$$

$$\frac{\dot{m}_s}{\dot{m}_v} = \frac{1-\alpha}{\alpha} \quad \text{Eq. C-7}$$

Combining Equations C-1, C-2, C-3, C-4, C-5, C-6, and C-7, we have:

$$\Phi = 1 + \frac{3}{8} \gamma_f \frac{\rho_v}{\rho_f} \frac{(1-\alpha)}{\alpha} \frac{\pi D^3 2 g_c 4}{8 \epsilon u_v f G_v \pi D^2} \quad \text{Eq. C-8}$$

Substitute the following quantities into Equation C-8:

$$u_v = \frac{G_v}{\rho_v} \quad \text{Eq. C-9}$$

$$\epsilon = \frac{12 k_v}{h_s \delta} \quad (\text{See Equation 7-32, Ref. 1-1}) \quad \text{Eq. C-10}$$

$$\delta = \delta_o (1-\alpha)^{1/3} \quad \text{Eq. C-11}$$

$$Nu_\delta \equiv \frac{h_s \delta}{k_f} \quad \text{Eq. C-12}$$

Assuming that the slip is large, that is, $u_v \gg u_\epsilon$ and that the drop is moving at constant velocity we have:

$$\gamma_f = C_d \frac{G_v^2}{2g_c \rho_v} \quad \text{Eq. C-13}$$

The above equations substituted in Equation C-8 result in:

$$\Phi = 1 + \frac{1}{8} \frac{1}{f} C_{d_s} \left(\frac{\rho_v}{\rho_f} \right) \left(\frac{k_t}{k_v} \right) \frac{D}{\delta_o} \frac{(1-\alpha)^{2/3}}{\alpha} Nu_s$$

and

$$C_{d_s} = \frac{f (\Phi - 1)}{\frac{1}{8} \left(\frac{\rho_v}{\rho_f} \right) \left(\frac{k_t}{k_v} \right) \frac{D}{\delta_o} \frac{(1-\alpha)^{2/3}}{\alpha} Nu_s} \quad \text{Eq. C-14}$$

The drag coefficient is expected to be a function of the drop Reynolds number:

$$Re_s = \frac{\delta \rho_v u_v}{\mu_v} = \frac{D G_v}{\mu_v} \frac{\delta_o}{D} (1-\alpha)^{1/3} \quad \text{Eq. C-15}$$

and the effect of mutual interference of nearby drops (see discussion accompanying Equation 7-30.) According to Reference 7-26, the coefficient of drag of groups of spheres placed on a wall is a function of the fraction of the total area covered by the projected area of the spheres, namely:

$$C_{d_s} = f(\mathbb{F}) \quad \text{Eq. C-16}$$

where $\mathbb{F}$ is defined by

$$\mathbb{F} \equiv N \frac{\pi \delta^2}{4} \quad \text{Eq. C-17}$$

combining Equations C-4, C-7, C-9, C-10, C-11, C-12 with C-17, we obtain:

$$\mathbb{F} = \frac{1}{32} \frac{(1-\alpha)^{2/3}}{\alpha} Nu_s \frac{D}{\delta_o} \frac{k_t}{k_v} \frac{\rho_v}{\rho_f} \quad \text{Eq. C-18}$$

This relationship between the fraction of the total area covered by the projected area of the spheres, quality and density ratio is used in Section 7.3.

D. EVALUATION OF MEAN HEAT TRANSFER COEFFICIENTS

Film or Intermittent Contact

The equation for the local Nusselt number is:

$$\frac{hD}{k_v} = \frac{G^{1/3}}{4} \frac{(1-\alpha)^{2/3}}{\alpha^{1/3}} \frac{D}{\delta_o} \beta^{4/3} \left(\tan \alpha \frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \quad \text{Eq. D-1}$$

(see Equation B-19)

Substitute Into Equation D-1:

$$h \equiv G \frac{d\alpha}{dL} \frac{D}{4} \frac{h_{fv}}{\Delta T} \quad \text{Eq. D-2}$$

The mean heat transfer coefficient over the length L is defined as:

$$h_m \equiv \frac{G}{4} \frac{D}{L} \frac{h_{fv}}{\Delta T} \alpha_e \quad \text{Eq. D-3}$$

Equations D-1 and D-2 may be combined to give Equation D-4 the assumptions (1) inlet quality to length L is zero, (2) ΔT is constant, and (3) β is constant.

$$\int_0^{\alpha_e} \frac{\alpha^{1/3} d\alpha}{(1-\alpha)^{2/3}} = \frac{\Delta T}{h_{fv} D^2 G} \frac{G^{1/3}}{4} \frac{D}{\delta_o} \beta^{4/3} \left(\tan \alpha \frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \int_0^L dL \quad \text{Eq. D-4}$$

Combining Equations D-4 and D-3 we have:

$$\left(\frac{h_m D}{k_v} \right) \frac{1}{\alpha_e} \int_0^{\alpha_e} \frac{\alpha^{1/3} d\alpha}{(1-\alpha)^{2/3}} = \frac{G^{1/3}}{4} \frac{D}{\delta_o} \beta^{4/3} \left(\tan \alpha \frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v} \right)^{2/3} \quad \text{Eq. D-5}$$

The results of graphical integration of the integral in Equation D-5 are listed below:

For $X_E = 1$,

$$\frac{1}{\alpha_e} \int_0^{\alpha_e} \frac{\alpha^{1/3} d\alpha}{(1-\alpha)^{2/3}} = 2.67$$

α_e	$\int_0^{\alpha_e} \frac{\alpha^{1/3} d\alpha}{(1-\alpha)^{2/3}}$
0.1	0.038
0.2	0.097
0.3	0.173
0.4	0.267
0.5	0.382
0.6	0.521
0.7	0.698
0.8	0.928
0.9	1.268
1.0	2.670

Contact

The equation for the local contact Nusselt number is:

$$\frac{hD}{k_v} = \frac{1}{32} Nu_b^2 \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{b_o} \right)^2 \frac{(1-\alpha)^{1/3}}{\alpha} \frac{\rho_v}{\rho_f} \quad \text{Eq. D-6}$$

(See Equation B-13)

Combining D-6 and D-2 and integrating:

$$\int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} \frac{GD^2 h_{fv}}{4 k_v \Delta T} = \frac{Nu_b^2}{32} \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{b_o} \right)^2 \frac{\rho_v}{\rho_f} \int_0^L dL$$

Substituting Equation D-3, we have:

$$\left(\frac{h_m D}{k_v} \right)_{\alpha_e} \frac{1}{\alpha_e} \int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} = \frac{Nu_b^2}{32} \left(\frac{k_f}{k_v} \right)^2 \left(\frac{D}{b_o} \right)^2 \frac{\rho_v}{\rho_f} \quad \text{Eq. D-7}$$

From integral tables,

$$\int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} = \frac{3}{5} (1-\alpha_e)^{5/3} - \frac{3}{2} (1-\alpha_e)^{2/3} + \frac{9}{10}$$

For

$$\begin{aligned} \alpha_e = 0.5 & \quad \int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} = 0.143 \\ \alpha_e = 0.7 & \quad \int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} = 0.305 \\ \alpha_e = 1.0 & \quad \int_0^{\alpha_e} \frac{\alpha d\alpha}{(1-\alpha)^{1/3}} = \frac{9}{10} \end{aligned}$$

Of interest is the value of the mean heat transfer coefficient for the case of a "once through" boiler ($X_E = 1$) in which contact boiling occurs over the entire length. This represents an upper limit in the heat transfer coefficient for dropwise, forced convection boiling. Assume SN 2 configuration, namely $D/\xi_0 = 4.74$. For mercury assume $\rho_v = 2.0 \text{ lb/ft}^3$, $K_f = 8 \text{ Btu/hr-ft-}^\circ\text{F}$, $K_v = 0.0098 \text{ Btu/hr-ft-}^\circ\text{F}$, $Nu_g = 0.8$, $D = 0.256 \text{ in.}$,

$$h_m = \frac{10}{9} \times \frac{.0098}{0.256} \times 12 \times \frac{1}{32} \times (0.8)^2 \left(\frac{8}{.0098} \right)^2 (4.74)^2 \times \frac{2}{800} = 382 \quad \text{Btu/hr-ft}^2\text{-}^\circ\text{F}$$

E. DYNAMIC ANALYSIS OF A DROP IN FILM EVAPORATION IN FORCED CONVECTION VORTEX BOILING

The "hilly" temperature profiles (Figures 5-1A to 5-1L) measured along the heat transfer test section are attributed to the oscillatory nature of flowing mercury drops in vortex flow in the intermittent contact boiling regime. The relationship between the period P and the wave length $\mathcal{L}$ of the oscillations is given by,

$$P = \frac{\mathcal{L}}{u \epsilon} = \frac{\mathcal{L} \rho_v}{G \alpha \epsilon} \quad \text{Eq. E-1}$$

where $\mathcal{L}$ is defined in Figure E-1.

Figure E-1

For a system with a small damping ratio, Equation E-1 becomes

$$\mathcal{L} = \frac{G \alpha \epsilon}{\rho_v} \sqrt{\frac{M}{K}} \quad \text{Eq. E-2}$$

where M and K are coefficients as defined in Equation A-9. Equation E-2 can be applied to forced convection vortex boiling if the following equations are substituted:

$$K = \frac{q_c \eta}{k_v} \theta \left(\rho_v \rho_f q_c \frac{2}{3} \frac{\eta}{\delta} \right)^{1/4} \left(\frac{4 k_w h_{fv}}{\Delta T} \right)^{1/2} = \frac{q_c \eta h_\delta}{k_v} \quad (\text{see Eq. A-8})$$

$$\eta = \left(\frac{\epsilon G \alpha}{\rho_v} \tan \alpha \right)^2 \frac{2}{D q_c} \quad (\text{see Eq. B-13A})$$

$$\epsilon = \frac{12 k_v}{h_\delta \delta} \quad (\text{see Eq. B-12})$$

$$\delta = \delta_o (1 - \alpha)^{1/3} \quad (\text{see Equation B-17})$$

$$\epsilon = \frac{6^{2/3}}{\theta^{1/3} \delta^{1/2}} \left(\frac{\rho_v 3 D}{\rho_f 4} \right)^{1/6} \left(\frac{k_v \Delta T}{G \alpha \tan \alpha h_{fv}} \right)^{1/3} \quad (\text{see Equation B-13 B})$$

Combining the above equations in the following steps,

$$K = \frac{g_c}{k_v} \left(\frac{G \alpha \tan \alpha}{\rho_v} \right)^2 \frac{2}{D g_c} \epsilon^2 h_g$$

$$K = \frac{1}{k_v} \left(\frac{G \alpha \tan \alpha}{\rho_v} \right)^2 \frac{2}{D} \frac{12 k_v}{\delta} \epsilon$$

and including Equation E-2 ($M = 1$, see Equation A-9)

$$\left(\frac{G \alpha \epsilon}{\rho_v} \right)^2 \left(\frac{2 \pi}{L} \right)^2 = \left(\frac{G \alpha \tan \alpha}{\rho_v} \right)^2 \frac{2}{D} \frac{12}{\delta} \epsilon$$

$$\epsilon \left(\frac{2 \pi}{L} \right)^2 = (\tan \alpha)^2 \frac{24}{D \delta}$$

$$\frac{6^{2/3}}{\theta^{1/3} \delta^{1/2}} \left(\frac{3}{4} \frac{\rho_v D}{\rho_f} \right)^{1/6} \left(\frac{k_v \Delta T}{G \alpha \tan \alpha h_{fv}} \right)^{1/3} \left(\frac{2 \pi}{L} \right)^2 = (\tan \alpha)^2 \frac{24}{D \delta}$$

$$\frac{D}{L} = \frac{3^{1/2}}{\pi} \frac{2^{1/3}}{\theta^{1/3}} (\tan \alpha)^{7/6} \theta^{1/3} \left(\frac{D}{\delta_o} \right)^{1/4} \frac{\alpha^{1/6}}{(1 - \alpha)^{1/12}} \left(\frac{\rho_f}{\rho_v} \right)^{1/2} \left(\frac{D G h_{fv} \mu_v}{k_v \Delta T} \right)^{1/6} \quad \text{Eq. E-3}$$

For a given test section, Equation E-3 reduces to:

$$\frac{1}{L} = \text{CONST.} \left(\frac{\dot{m} \alpha}{\Delta T} \right)^{1/6} \frac{1}{[(1 - \alpha) \rho_v]^{1/12}} \quad \text{Eq. E-4}$$

The variables in Equation E-4 can be used for correlation purposes as shown in Figure 6-1.

F. EMPIRICAL DERIVATION OF EQUATION FOR THE FORCED CONVECTION DROPSIZE BOILING OF MERCURY IN INTERMITTENT CONTACT

A large portion of the data so far accumulated lies in the intermittent contact regime as evidenced by Figures 7-12 and 7-16. If one assumes that an equation of the form,

$$\beta = C \psi^n \quad \text{Eq. F-1}$$

empirically correlates the data in this regime, one can derive the corresponding heat transfer equation for computing the Nusselt number. Expressing Equation 7-16 in terms of ψ we have:

$$\frac{32 \frac{hD}{k_v}}{(0.8)^2 \left(\frac{k_f}{k_v}\right)^2 \left(\frac{D}{\delta_o}\right)^2 \frac{(1-\alpha)^{1/3}}{\alpha} \frac{\rho_v}{\rho_f}} = \psi^2 \quad \text{Eq. F-2}$$

where

$$\psi = \frac{Nu_b}{Nu_{b(\text{CONTACT})}} = \frac{Nu_b}{0.8}$$

Similarly Equation 7-17 results in,

$$\frac{\frac{hD}{k_v}}{\frac{G^{1/3}}{4} \frac{(1-\alpha)^{1/3}}{\alpha^{1/3}} \frac{D}{\delta_o} (\tan \alpha)^{1/3} \left(\frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v}\right)^{1/3}} = \beta^{1/3} \quad \text{Eq. F-3}$$

Combining Equations F-1, F-2, and F-3 eliminating ψ and β we have:

$$\left(\frac{hD}{k_v}\right)^{3/4 - n/2} = C \frac{G^{1/4}}{4^{1/4}} \frac{(1-\alpha)^{1/2}}{\alpha^{1/4}} \left(\frac{D}{\delta_o}\right)^{1/4} (\tan \alpha)^{1/2} \left(\frac{\rho_v}{\rho_f} \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v}\right)^{1/2} \frac{32^{n/2}}{(0.8)^n \left(\frac{k_f}{k_v}\right)^n \left(\frac{D}{\delta_o}\right)^n \frac{(1-\alpha)^{n/6}}{\alpha^{n/2}} \left(\frac{\rho_v}{\rho_f}\right)^{n/2}} \quad \text{Eq. F-4}$$

By inspecting Figures 7-12 and 7-16, $n = 1$ appears to fit the data. Equation F-4 then results in,

$$\frac{hD}{k_v} = C^4 234 \left(\frac{k_v}{k_f}\right)^4 (1-\alpha)^{1/3} \alpha \frac{\delta_o}{D} \left(\tan \alpha \frac{DG}{\mu_v} \frac{h_{fv} \mu_v}{\Delta T k_v}\right)^2 \quad \text{Eq. F-5}$$

The constant C has a value somewhere between 2.3 and 3.0 according to Figures 7-12 and 7-16.

G. FLOW ESTABLISHMENT BOILING

The condition assumed prior to the establishment of vortex boiling is pool boiling in a one-g environment with the entire liquid phase agglomerated in the bottom of the tube. This is depicted in Figure G-1. The liquid phase is transported along the tube by the flowing vapor until the vapor velocity becomes sufficiently high to make the liquid-vapor interface unstable. At this point the liquid breaks up into drops small enough to be swirled around the tube periphery by the vapor and the spring insert. The heat transfer coefficient of the agglomerated liquid based on its projected area can be assumed the same as that for pool boiling. Figure G-2 shows a pool boiling curve for pure mercury. An approximation to this curve is represented by Equation G-1, namely,

$$h_g = \frac{606}{\Delta T^{0.8}} \quad \text{Eq. G-1}$$

Equation G-1 is accurate for temperature differences below 100°F which is satisfactory for application to the data of Table 6-2.

Since h_g is based on the liquid area, to convert to a coefficient based on the tube wall area combine the following two equations,

$$\begin{aligned} h_g \delta L \Delta T &= Q \\ h \pi D L \Delta T &= Q \\ h_g \frac{\delta}{\pi D} &= h \end{aligned} \quad \text{Eq. G-2}$$

Continuity results in,

$$\frac{\pi \delta^2}{4} \rho_f u_v = \dot{m}_f$$

Since

$$\frac{1-\alpha}{\alpha} = \frac{\dot{m}_g}{\dot{m}_f}$$

we have

$$\begin{aligned} \frac{\pi \delta^2}{4} \rho_f u_v &= \frac{1-\alpha}{\alpha} \frac{\pi D^2}{4} \rho_g u_g \\ \left(\frac{\delta}{D}\right)^2 &= \frac{1-\alpha}{\alpha} \frac{\rho_g}{\rho_f} \frac{1}{\epsilon} \end{aligned} \quad \text{Eq. G-3}$$

FORCED CONVECTION POOL BOILING IN A ONE "G" ENVIRONMENT OF PURE NON-WETTING MERCURY

Figure G-1

POOL BOILING CURVE OF PURE MERCURY

Figure G-2

Assume the slip ratio ϵ is the same as for a drop, namely,

$$\epsilon = \frac{12 k_v}{h_s \delta} \quad \text{Eq. G-4}$$

(see Equation B-12)

Combining Equations G-1 and G-2 we have,

$$\frac{60G}{\Delta T^{0.8}} \frac{\delta}{D} \frac{1}{\pi} = h \quad \text{Eq. G-5}$$

Combining Equations G-4 and G-2 we have ,

$$\frac{12 k_v}{\epsilon} \frac{1}{\pi D} = h \quad \text{Eq. G-6}$$

Combining Equations G-5 and G-3 we have ,

$$\frac{60G}{\Delta T^{0.8}} \left[\frac{(1-\alpha)}{\alpha} \frac{c_v}{c_s} \frac{1}{\epsilon} \right]^{1/2} \frac{1}{\pi} = h \quad \text{Eq. G-7}$$

Combining Equation G-1 with Equation G-6 we have,

$$\frac{1}{\Delta T^{0.8}} \left(\frac{60G D}{k_v} \right)^2 \left[\frac{(1-\alpha)}{\alpha} \frac{c_v}{c_s} \frac{1}{12\pi} \right] = \frac{h D}{k_v} \quad \text{Eq. G-8}$$

The mean heat transfer coefficient between $X=0$ and $X = X_E$ is defined as,

$$h_m \pi D L \Delta T \equiv \dot{m} h_{ev} \alpha_s \quad \text{Eq. G-9}$$

and the local heat transfer coefficient as,

$$h \pi D dL \Delta T = \dot{m} h_{ev} d\alpha \quad \text{Eq. G-10}$$

Combining Equations G-9 and G-10 results in,

$$h_m \frac{L}{\alpha_e} d\alpha = h dL \quad \text{Eq. G-11}$$

Substituting Equation G-3 into Equation G-11 and integrating we have,

$$\frac{h_m D}{k_v} \frac{L}{\alpha_e} \int_0^{\alpha_e} \frac{\alpha d\alpha}{1-\alpha} = \frac{1}{\Delta T^{0.8}} \frac{(GOG)^2}{12\pi} \frac{\rho_v}{\rho_e} \left(\frac{D}{K}\right)^2 \int_0^L dL$$

$$\frac{h_m D}{k_v} = \frac{\alpha_e}{\int_0^{\alpha_e} \frac{\alpha d\alpha}{1-\alpha}} \frac{1}{\Delta T^{0.8}} \left(\frac{GOG D}{k_v}\right)^2 \frac{\rho_v}{\rho_e} \frac{1}{12\pi} \quad \text{Eq. G-12}$$

The value of the integral as a function of X_E is plotted in Figure G-3.

When ΔT approaches zero, the resistance of the vapor film becomes negligible and the resistance of the liquid mercury becomes important. Under these conditions

$$\frac{h_s \delta}{k_e} = \frac{2}{3} \quad \text{Eq. G-13}$$

(see Equation 7-10, Figure 7-5)

Combining Equation G-13 with Equation G-2 we have,

$$\frac{hD}{k_v} = \frac{2}{3} \frac{1}{\pi} \frac{k_e}{k_v} \quad \text{Eq. G-14}$$

For mercury in the range of interest the Nusselt number has a value of,

$$\frac{hD}{k_v} = 170 \quad \text{Eq. G-15}$$

Which represents an upper limit for forced convection pool boiling of pure non-wetting mercury in a one "g" environment.

VALUE OF INTEGRAL AS A FUNCTION OF x_E
SEE EQUATION G-12

Figure G-3

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