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AEC RESEARCH &
DEVELOPMENT REPORT

ORGANIC WORKING FLUID MASTER BOILER INVESTIGATION —

AEC ORGANIC RANKINE CYCLE TECHNOLOGY PROGRAM TOPICAL REPORT

**AVIATION DIVISION
SUNDSTRAND CORPORATION**

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ORGANIC WORKING FLUID
BOILER INVESTIGATION-

A.E.C. RANKINE CYCLE TECHNOLOGY
PROGRAM

TOPICAL REPORT

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CONTRACT AT(04-3)-651
FOR THE
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FOREWORD

This final report was prepared by Sundstrand Aviation, a Division of Sundstrand Corporation, in accordance with the requirements of Contract AT(04-3)-651. Mr. Ronald Anderson, Division of Space Nuclear Systems, USAEC, Washington, D. C., presently administers the program. During the majority of this program Mr. Gerald S. Leighton was the USAEC Project Administrator. Mr. R. E. Niggemann serves as Advisory Engineer at Sundstrand. Mr. J. Petersen is the Contract Program Manager for Sundstrand. In the early design and synthesis stages of the Boiler Rig Program, Messrs. J. C. Newcomb, R. Taylor, and R. E. Niggemann were the chief contributors. In the actual testing and data reduction of the data recorded Mr. J. L. Martz was the chief contributor.

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1.0 INTRODUCTION AND SUMMARY

This report presents the analytical and experimental work associated with the design of a boiler loop and oncethrough boiler tubes with various boiler tube internal geometries.

This effort is one of the tasks funded by the A.E.C., under contract no. AT(04-3)-651, the Organic Rankine Cycle Technology Program.

The purpose of the heat transfer and stability tests was to establish satisfactory analytical equations for predicting boiler heat transfer performance and establish hydrodynamic stability design criteria for once through Organic Rankine Cycle boiler systems.

This report presents a discussion of the test hardware, instrumentation, test procedures, operating difficulties and test results of a oncethrough Dowtherm-A vaporizer heated by thermal radiation from electrical heaters.

In predicting boiler heat transfer performance the several boiling regions were considered. The Dittus-Boelter equation was used in the subcooled single phase flow region. An original correlation of the form $(Q/A)_b = C\Delta T^n$ was used in the nucleate boiling regime.

The Dengler-Addoms correlation was employed to predict film coefficients in the two phase flow region.

It was found that the analytical coefficients were underestimated by a factor of 1.4, with the superficial F factor equal to 1.

In the bare tube runs the Dittus-Boelter equation predicted actual results within $\pm 25\%$. Film coefficients in two-phase flow were within $\pm 10\%$ with $F = 1.0$ in Runs 1A and 1B and $F = 1.4$ in Run 1C. With the addition of a .25" concentric tube in the subcooled liquid region, orifices in the low quality end, and a spiral wire in the high quality end, film coefficients were found to be twice as high in the subcooled liquid region, since h is dependent on velocity in the Dittus-Boelter equation. The film coefficient profile in the quality region had a smaller band of high h 's over a given length of boiler tube as compared to the bare tube runs. The Dengler-Addoms correlation did not predict actual results as well; analytical and experimental results were as much as $\pm 30\%$ apart.

Boiler stability tests were also conducted. Initially, with a bare tube, stable operation was observed above 450°F inlet. With the insertion of a .25" concentric tube in the subcooled liquid region, orifices in the low quality end and spiral wires in the high quality end, stable boiler operation was observed under all inlet conditions and under all flow conditions. With removal of the orifices, the boiler still ran stable, indicating that the concentric inner tube in the subcooled and low quality region followed by a swirl wire tube insert is all that is required to make the boiler tube hydrodynamically stable. Inlet throttling of the liquid to the boiler had no apparent effect on stability in any of the tests performed.

2.0 TECHNICAL DISCUSSION

In the following sections is a discussion of the equipment used, the test plan and procedures, data processing, experimental results, and a final discussion and conclusions.

2.1 EQUIPMENT

2.1.1 Test Apparatus

2.1.1.1 Boiler

The A.E.C. boiler loop test section is contractually specified to be heated by thermal radiation from electrical heaters capable of providing controllable thermal flux levels to the test section.

The vaporizer consists of two 16 foot straight sections of .5" O.D. AISI 302 Stainless Steel tubing with .049" thick tube walls. These two 16 foot straight sections are connected by a U-tube made of the same material. Sixteen 2 foot long clam-shell heaters provide a radiant heat flux to the boiler tubes. The heaters are insulated to minimize heat loss to the vacuum chamber wall. To decrease heat losses to the

environment by convection the boiler tube is housed in a vacuum chamber made of AISI 302 Stainless Steel. The chamber is 17 feet long, 18" in diameter and .25" thick. The chamber is operated at a vacuum of 100 microns Hg under test conditions. The maximum temperature of the chamber wall is 250°F. See Figures 2.1.1.1-1 and 2.1.1.1-2 for a boiler tube closeup and layout of the rig.

2.1.1.2 Pump

The pump used in the AEC boiler rig is a two-stage regenerative type pump with an available head of over 400 feet of water. At design conditions the pump has 360 feet of head. Maximum operating temperature of the pump should be kept under 300°F. Glycol coolant flows within the pump to keep it running cool. See Figure 2.1.1.1-2 for a layout of the pump.

2.1.1.3 Condenser and Regenerator

The condenser-regenerator assembly consists of the following:

- a. The vapor inlet ducting
- b. The regenerator, including all baffles and side plates
- c. The regenerator ducting
- d. The condenser-subcooler reservoir, including the palladium diffusion section.

Figure 2.1.1.1-1 Boiler Tube - Close Up

Figure 2.1.1.3-1 shows the assembly prior to final cleaning and Figure 2.1.1.3-2 shows the palladium diffusion section welded in place on the condenser.

2.1.1.4 Cavitating Venturi

The cavitating venturi allows the overall system flow rate to be completely independent of boiler pressure characteristics. If the venturi inlet pressure (feed pump outlet pressure) is at a constant level, then the venturi outlet velocity and hence, system flow, will be constant.

The venturi performed this function satisfactorily up to a back pressure of approximately 70% of the venturi inlet pressure. This indicates a venturi diffuser pressure recovery of about 75%. The total pressure recovery across the venturi was expected to be 80 to 85%, based upon information in the literature.

Although the venturi was specified by the vendor to flow .35 gpm (170 lb/hr) at 150 psia inlet pressure, preliminary tests indicated a pressure of 167 psia was necessary to achieve this flowrate. One possible explanation for this 17 psi deviation is that the vendor sized the venturi for 150 psi gage instead of absolute.

Figure 2.1.1.2-2 Rig Layout

Figure 2.1.1.3-1 Regenerator-Condenser Assembly

Figure 2.1.1.3-2 Palladium Diffusion Section

Figure 2.1.1.4-1 shows the venturi flow characteristics over a wide range of operating conditions.

2.1.1.5 Preheaters

The preheaters consist of five clam-shell heaters arranged in a pyramid type position, with three heaters on the bottom and two on top. Refer to Figure 2.1.1.5-1. The working fluid is heated by thermal radiation from electrical preheaters as in the test section. A temperature readout device provides a signal to an SCR power modulator which maintains the desired temperature of the fluid. The inlet temperature was increased gradually to allow for temperature equilibrium to occur. Temperature measurements of the preheaters are determined by chromel-alumel thermocouples. One thermocouple measures the temperature of the three bottom preheaters, and the other thermocouple measures the temperature of the two top preheaters. Refer to Figure 2.1.1.5-1.

Figure 2.1.1.4 - 1 Venturi Flow Characteristics

2.1.1.6 Inlet-Outlet Throttling Valves Used in the AEC Test Section

The inlet throttling valve has a maximum operating pressure rating 4000-5000 psig and is manufactured by the Hoke Valve Company. Pressure and temperature measurements are taken on each side of the valve.

The valve and piping connecting the valve is insulated with Fibrefrax. The outside of the insulation is wrapped in aluminum foil to reduce radiant heat transfer to the heat sink (assumed black body at 150°F). Initially, this valve was thought to affect boiler stability.

Pressure level of the boiler outlet valve is measured by a throttling calorimeter valve. This valve is well insulated and located near the boiler outlet near the vacuum chamber wall. Fluid temperature and pressure instrumentation is located on both sides of the valve. The valve is essentially a throttling calorimeter for determining boiler outlet quality. Refer to Figure 2.1.1.5-1.

Figure 2.1.1.5-1 Boiler Test Loop Components

2.1.2 Instrumentation

Temperature measurement is accomplished by use of chromel-alumel thermocouples. There are 32 wall temperature thermocouples placed approximately 1 foot apart. Fluid temperatures are measured at 16 points along the 32 foot long boiler tube. Temperatures of the regenerator, venturi, condenser, and all other system components are determined in the same way.

The following discussion concerns installation of the test section thermocouples. The boiler tube was slotted to accept the tube wall thermocouples. These slots were produced by the electrical milling process (EDM) and are 3 inches long by .026 by .028 inches deep and wide. Thus, the .025 inch thermocouples had a close fit and were subsequently brazed in place with silver solder as shown in Figure 2.1.1.1-1. Due to the 3 inch submergence length in the slot, axial conduction errors were negligible. With the thermocouple located as above and at a typical heat flux, the thermocouple read about 3.5 degrees higher than the inner tube wall temperature. The uncertainty of the thermal resistance of the tube wall-thermocouple combination is sufficient to result in an error of less than one degree. However small, this error can cause some spreading of the test data, so great care was taken to assure a uniform braze.

As can be seen in Figure 2.1.1.1-1, the boiler tube was painted with Pyromark high temperature resistant black paint. Thus, the absorbtivity of the boiler tube was substantially increased. This reduced the operating temperature of the clam shell heaters, prolonging their life and minimizing thermal growth of the heater shells.

Boiler pressures are determined by using Wianko reluctance type pressure transducers. The recording system is designed to permit strip chart recordings (8 channels) and meter indication of pressures as sensed by Wianko variable reluctance transducers. The ΔP transducers are capable of reading up to ± 20 psi and the inlet and outlet transducers can register up to 200 psig.

In all, there are 12 pressure transducers in the boiler test seciton. Initially, all transducers were referenced to the inlet side of the boiler. Further testing required larger ΔP 's and thus appropriate transducers were replumbed to the inlet and outlet. For a complete schematic of all thermocouple and transducer instrumentation refer to Figure 2.1.2-1.

Figure 2.1.2-1 Test Section Instrumentation

Pressures of all other components except chamber pressure were read by bourdon tube gages. The transducers appear in the test loop as in Figure 2.1.1.5-1.

Additional instrumentation consisted of a Dana digital voltmeter which read m.v. outputs of all thermocouples and pressure transducers, a Burrough's dial indicator which specified temperatures in °F, and three multipoint patch recorders that indicated temperatures in °F of all wall stations, fluid stations and heaters. Refer to Figure 2.1.2-2.

2.2 TEST PLAN

The objective of these tests was to provide experimental data necessary to 1) establish satisfactory analytical equations for predicting boiler performance and 2) formulate design criteria for Organic Rankine Cycle boiler systems.

2.2.1 Description of Test Conditions

Three boiler tube configurations, as described in the following paragraph, were tested at the design point. In addition, each configuration was tested over a wide range of operating conditions. The design point and range of test conditions are presented in Table I.

Figure 2.1.2-2 Boiler Test Loop and Instrumentation

TABLE I
RANGE OF TEST CONDITIONS

<u>Parameter</u>	<u>Design Point</u>	<u>Range</u>
Flow Rate (lb/hr)	170	85-200
Boiler Exit Pressure (psia)	71	45-107
Boiler Inlet Temperature (°F)	460	350-650
Subcooled Liquid ΔP (psid)	51.85	As required for steady state
Maximum Tube Wall Temp. (°F)*	700	650-700
Exit Quality (%)	100	0-100

*Fixes heat flux input. At design conditions this heat flux is approximately 12,000 BTU/hr-ft².

FIGURE 2.1.2-3

2.2.2 Boiler Tube Test Configurations

Three different boiler tube internal configurations were tested, while the external configuration of all tubes consists of a bare 1/2" O.D. 302 stainless steel tube with .049" wall. The internal configurations are as follows:

a. Configuration A

No internal devices---bare tube

b. Configuration B

Swirl wire and tubular insert (low quality region)

c. Configuration C

Swirl wire, flashing orifice

In the actual tests configurations b and c were combined. Also, swirl wire of various pitches and diameters were tested. Pitches consisted of 1/2", 1", and 2" with .054" diameter and a 2" pitch with .085" diameter.

2.2.3 Scope of Performance Tests

Each boiler tube configuration was tested to evaluate its performance with respect to its stability, heat transfer and pressure drop characteristics.

2.2.4 Description of Test Loop

Refer to Figure 2.2.4-1 for the discussion which follows.

Subcooled liquid is fed to the boiler feed pump under a condenser pressure of approximately 5 psia (Point 1). The liquid is pressurized by the pump. The outlet pressure (Point 2) and the system flow rate are controlled by adjusting the pump bypass valve. From the feed pump the liquid enters the flowmeter, then the cavitating venturi. Here the flow is accelerated to a velocity sufficient to lower the static pressure below the saturation pressure. The resulting cavitation allows for a constant flow rate, independent of pressure oscillations that may exist in the boiler test section. The venturi will function as mentioned as long as the back pressure is less than or equal to the pressure recovery of the venturi. For

this particular venturi, pressure recovery is approximately 80%. When flow requirements are other than the design flow rate, a venturi bypass valve is used. From the venturi the flow enters the regenerator, being preheated by the superheated vapors exiting from the boiler. The amount of system flow through the regenerator is controlled by the regenerator inlet and bypass valves. The temperature of the liquid entering the boiler is regulated by electrical preheaters (Point 5). The inlet temperature is readily maintained by this type of temperature control. An inlet throttling valve is regulated to control the degree of inlet subcooling (Point 6). The liquid then enters the boiler and the rate of vaporization and outlet temperature (Point 7) is regulated by controlling the heat input of 16 clam-shell heaters. The vapor exits at 100% quality and is expanded to a superheated state through a throttling calorimeter valve (Point 8). Boiler pressure is controlled by throttling the boiler outlet valve. From Point 8 the low pressure vapor passes through the regenerator, exiting at a temperature (Point 9) commensurate with the amount of heat transferred to the boiler feed flow. The vapor is further desuperheated and condensed in the condenser (Point 10). Control of condenser pressure and temperature is accomplished by regulating the ethylene glycol coolant flow through the condenser.

Figure 2.2.4-1

Due to air inleakage in the low pressure plumbing, it is necessary to pump to maintain a vacuum on the system. Subcooler outlet temperature (Point 11) is controlled by adjusting an ethylene glycol coolant valve similar to the one at the condenser.

2.3 CALIBRATION

2.3.1 Thermocouple Calibration

Calibration of all thermocouples was accomplished with the boiler fully assembled and all probes in place. Five points from 450-650°F in 50°F increments, were used to establish the emf versus temperature relationship for each thermocouple. A comparison to a standard chromel-alumel table from the National Bureau of Standards was used. Each thermocouple was aged at 1000°F for 100 hours to stabilize the junctions. Cooling water was used to maintain the thermocouple terminal strips at a constant temperature regardless of the power level being tested to preclude errors at these intermediate junctions. Figure 2.3.1-1 depicts the relationship between the thermocouple terminal strips and the aluminum support beam and

Figure 2.3.1-1 Cooling Duct In Boiler Test Sections

cooling duct. To further safeguard accuracy of the thermocouple readings, calibration runs preceded each major series of tests.

During the calibration runs heat input to the boiler was necessary to maintain temperature equilibrium. However, heat transfer may occur between an individual heater and the tube section it surrounds. Thus, heater temperature and power was carefully regulated to eliminate any net heat transfer from the heater to the boiler tube. Also, extraneous heat transfer to the thermocouple junction was minimized by embedding the thermocouple into an axial slot in the tube wall for approximately 3 inches. This configuration allowed the heat being conducted axially down the thermocouple assembly to be dissipated into the tube wall. (Figure 2.3.1-2). With a net zero heat flux, the bulk and tube wall thermocouples see the same temperature throughout.

In order to use the platinum probes as a calibrating standard, the probes were calibrated at three points bracketing the expected range of conditions. The points were the ice point at 32°F, a 400°F point in a hot, agitated, silicon oil while both were compared to a primary standard resistance from NBS. The last point was at the freezing point of zinc which has been validated and is regarded as a standard as is the ice bath. The platinum resistance probe was selected due

.025" O. D. Stainless Steel Sheathed Chromel-Alumel Thermocouple

Figure 2.3.1 - 2 Tube Wall Thermocouple Installation

to its superior ability and accuracy in the range of temperatures that were encountered in these tests.

In order to calibrate the thermocouples in place, two platinum resistance probes were used. One probe is at the inlet side and the other at the boiler outlet side. These probes reach into the insulated area within the vacuum chamber thus reading a bulk liquid temperature at the beginning of the first heated segment and at the end of the last. The loop pump was used to circulate the design flow rate (171 lb/hr) to the boiler tube such that the probe temperatures were within $\pm 1^{\circ}\text{F}$. A linear interpolation was employed to determine the temperature of a thermocouple at any given station.

Resistances of the platinum probes were read on a resistance bridge circuit made especially for this purpose and the thermocouple outputs were read on a Dana Digital Millivolt meter. An ice bath reference was coupled to the thermocouples and all connections made in the thermocouple circuits were held to within $\pm 5^{\circ}\text{F}$. This last precaution is important in reducing errors due to voltages which can be generated at the connection of two wires or connectors at which a slight composition change occurs. Each thermocouple was checked for possible effects due to compositional changes along

its length. Variations in the chemical composition of the thermocouple wire could change the net output voltage if the gradient along the wire should differ from the gradient during calibration. Those wires indicating this effect were discarded.

Exclusive of errors common to all thermocouple applications, (initial calibration, drift, reference junction, etc.) there are three sources of error inherent in the steady state measurement of wall temperatures.

A list of the three sources of error is as follows:

- a. heat transfer to or from the thermocouple junction
- b. change in surface heat transfer coefficient in the area of attachment
- c. change in internal thermal resistance of the wall

The first error has been minimized by insertion of the thermocouple in the tube wall. Refer to Figure 2.3.1-2.

The change in the surface heat transfer coefficient is strictly a function of the surface emissivity, since radiation is the only mode of heat transfer. When

the thermocouple lead wires were attached to the tube wall, care was taken to 1) select a braze material with approximately the same emissivity as the tube 2) confine the braze material to the area beneath the thermocouple sheath.

The change in internal thermal resistance becomes a potential source of error when the thermocouple diameter becomes a significant percentage of the wall thickness.

Although sheathed thermocouples as small as .01" are available, a .025" diameter thermocouple was specified when considering such factors as lead time, cost, reliability, ease of handling and installing. Fortunately, the magnesium oxide (MgO) that insulates the thermocouple wires has approximately the same thermal conductivity as the stainless steel tube wall over the temperature range expected.

Therefore, the thermal resistance of the thermocouple is reduced to the contact resistance between the sheath, MgO and the tube wall. If the thermocouple is brazed into place, the increase in thermal conductivity of the braze material may compensate for the contact resistance between the MgO and the sheath.

2.3.2 Pressure Transducer Calibration

Calibration was accomplished in the calibration lab. A special weight was placed on the unit and a given m.v. reading recorded. Thus, a curve of psig versus milli-volts was generated for each of the twelve transducers.

2.3.3 Clam Shell Heater Loss

Prior to conducting a heat loss calibration it was necessary to establish a vacuum chamber pressure level that will be maintained for all tests. This pressure level was selected on the basis of heat loss vs vacuum pressure curves of Figure 2.3.3-1, the performance characteristics of the vacuum chamber and vacuum pump, and the required or desirable accuracy of the calibration data.

Referring to Figure 2.3.3-1 the heat loss is only a weak function of the vacuum pressure over the range of pressures investigated (25 microns being the lowest pressure attainable from this particular vacuum chamber and pump combination). This plateau is to be expected because the natural convection mode of heat transfer is essentially non-existent at these pressures. Also, the thermal conductivity of air is not affected by pressures above 50 microns. On the basis of these

Figure 2.3.3 - 1 Typical Test Section Heat Loss Vs. Vacuum Level

factors, 100 microns was somewhat arbitrarily selected as a convenient pressure to conduct all testing.

The heat loss calibration was determined by setting each of the 16 heater sections at a preset power input level, and then measuring the resulting equilibrium temperature of the ceramic heater surface. From these data, a curve of heater power loss versus heater temperature can be constructed. Since all heater temperature data fell within a ± 12 watt band width, it was decided to use the same heat loss calibration curve for all 16 heater sections. The error in net heat flux as a result of this assumption was less than 1/2% for each heater. Figure 2.3.3-2 shows a general heat loss curve derived from all the calibration data. These tests were conducted without the boiler tube in place to prevent damage to the tube and instrumentation at high temperature levels. To insure thermal equilibrium, data was not recorded until a minimum of 12 hours had elapsed after the previous power adjustment. Only a limited amount of testing was conducted above 1200°F so as not to compromise system reliability.

2.3.4 Flow Meter Calibration

A calibration curve was established by the calibration laboratory. The calibration curve consists of GPM versus frequency in CPS. The frequency in CPS is the frequency of the vane propeller. By passing a known amount of fluid through the flowmeter and recording the frequency, a calibration curve of flow in GPM versus CPS can be constructed.

2.4 TEST PROCEDURE

2.4.1 Boiler Startup Procedure

Refer to Figure 2.4.1-1 for explanation of symbols.

1. Open valves V_{10} , V_{12} , V_9 . These are the glycol coolant valves to the condenser, subcooler and pump respectively.
2. Open valves V_1 , V_2 . These valves are the boiler inlet and outlet respectively.
3. Open valve V_4 and close valve V_3 . These valves are the regenerator inlet and bypass respectively.
4. Close valve V_5 and open Valve V_6 . These valves are the venturi bypass and inlet respectively.
5. Open valve V_7 and close valve V_8 . These valves are the boiler feed pump outlet and bypass.

6. Open valve V₁₁ and V₁₃ and draw vacuum on condenser. When the vacuum is about -10" Hg, close valve V₁₃.
7. Turn feed pump switch on. Check flow on strip recording. The flow should read .35 GPM or equivalently 171 lb/hr. Adjust by opening V₈.
8. Turn on SCR panel switch and variac switch. Adjust heat accordingly.

NOTE: An open valve is opened all the way. Conversely for a closed valve. Required amount of Dowtherm A to fill system is 22 lbm (commercial grade).

2.4.2 Shut Down Procedure

1. Turn off SCR and variac switches
2. Open venturi bypass valve, V₅
3. Open V₁ and V₂
4. Open V₃ and close V₄
5. Close V₈
6. Periodically, check multipoint recorder for heater temperatures. When heater temperatures are below 500°F, system flow rate can be shut off.

<u>Symbol</u>	<u>Function</u>
V ₁	Boiler Inlet Valve
V ₂	Boiler Outlet Valve
V ₃	Regnerator Bypass Valve
V ₄	Regenerator Inlet Valve
V ₅	Venturi Bypass Valve
V ₆	Venturi Inlet Valve
V ₇	Pump Outlet Valve
V ₈	Pump Bypass Valve
V ₉	Pump Glycol Coolant Inlet Valve
V ₁₀	Condenser Glycol Coolant Inlet Valve
V ₁₁	Condenser Vacuum Valve
V ₁₂	Subcooler Glycol Coolant Valve
V ₁₃	Condenser Vacuum Valve

Figure 2.4.1-1

2.5 EXPERIMENTAL RESULTS

2.5.1 Bare Tube Heat Transfer and Stability Tests

The first of a series of heat transfer tests were conducted at the following conditions (bare tube):

SYSTEM PARAMETERS FOR RUNS 1A, 1B AND 1C

<u>Parameter</u>	<u>Run 1A</u>	<u>Run 1B</u>	<u>Run 1C</u>
Flow Rate (lb/hr)	171	171	200
Heat Flux (BTU/hr-ft ²)	12,500	12,500	12,500
Boiler Exit Pressure (psia)	71	105	77.5
Boiler Inlet Temperature (°F)	461	498	465
Inlet Subcooling (°F)	195	202	195
Inlet Subcooled ΔP (psia)*	43	13	3
Exit Quality	100	100	100

FIGURE 2.5.1-1

*ΔP drop across inlet valve to boiler.

Refer to figures 2.5.1-2 to 2.5.1-4 for corrected temperature profiles for the above runs.

40

Figure 2.5.1-2

17

Figure 2.5.1-3

Figure 2.5.1 - 4 Boiler Temperatures As A Function of Boiler Tube Length

Stability tests were conducted with the bare tube configuration. A relation was obtained between the boiler inlet temperature and flow rate for a constant outlet pressure. By increasing the flow rate, it was also found necessary to increase the boiler inlet temperature in order to obtain stable boiler operation, that is, no pressure spikes in the boiler inlet pressure. These tests were conducted for two different outlet pressures corresponding to saturation temperatures of 650°F and 680°F. Refer to Figure 2.5.1-5.

Another series of bare tube stability tests were conducted in which the heat flux and flow rate were fixed and the boiler outlet pressure (saturation temperature) was varied. Figure 2.5.1-6 depicts a relation between boiler inlet temperature for stable operation (no pressure spikes) and boiler outlet pressure. The top curve represents the boundary for stable operation. Temperatures above this line resulted in stable boiler operation. The bottom curve represents the boundary for unstable operation. Temperatures below this line resulted in unstable boiler operation.

In order to fully appreciate the meaning of boiler instability, it is suggested the reader refer to Appendix A, Figure A-1.

Figure 2.5.1 - 5 Boiler Inlet Temperature Required for Stability as a Function of Flow Rate

Figure 2.5.1 - 6 Boiler Inlet Temperature Required for Stability as a Function of T_{SAT}

2.5.2 Swirl Wire Heat Transfer and Stability Tests

Efforts were made to build a flashing orifice which would preclude the complicated flow regimes present in two-phase boiling. These boiling regimes, i.e., bubbly and slug-plug, probably account for the instabilities present in the boiler. If these flow regimes could be prevented from occurring and if it were possible to go from a state of subcooled liquid to quality instantaneously at a particular cross-section, then the system should have no noticeable instabilities.

The following is an explanation of the process the boiler theoretically should have undergone with the installation of the flashing orifice. Subcooled liquid would enter the boiler at $T = 460^{\circ}\text{F}$ and $P = 160$ psia. The reason for this high inlet pressure is to make the saturation temperature higher than any local wall temperature. The liquid then travels through an annular cross-section for the first length of boiler tube. The purpose of having the annular cross-section is to increase the heat transfer coefficient and thereby decrease the temperature difference between the wall and fluid. By starting with a high inlet pressure, the local pressure anywhere along the tube is always higher than the saturation pressure corresponding to the local wall temperature. This results in no bubble generation for the first 16 feet of heated length.

The subcooled liquid travels down the first 16 feet of boiler tube with a corresponding 1.18 psia pressure drop. The liquid is at 680°F upstream of the orifice. By passing through the orifice, the liquid is flashed down to 71 psia and 15 percent quality. For the last 16 feet of boiler tube, the fluid travels through swirl wire turbulators. It is proposed that this type of internal configuration should allow for stable boiler operation (no pressure spikes in boiler inlet pressure).

The following is the development of a mathematical model for the concentric tube.

Constraints

Inlet temperature to orifice = 680°F

Pressure at orifice inlet = 91 psia

Boiler inlet temperature = 460°F

Reynold's Number:

$$Re = \frac{\rho V d_h}{\mu} = \rho \left[\frac{W}{\rho A} \right] \frac{d_h}{\mu} = \frac{W d_h}{A \mu}$$

d_h = hydraulic diameter (ft)

A = cross-sectional area (ft²)

C = wetted perimeter (ft)

D_o = .0334 (ft)

Evaluating properties @ 680°F, the Reynold's Number becomes:

$$Re = \frac{300}{D_o + D_i}$$

$$Pr = \frac{c_p \mu}{k} = \frac{(.585)(.726)}{.0725} = 5.85$$

$$Nu = \frac{hdh}{k} = \frac{h(D_o - D_i)}{7.25 \times 10^{-2}} = 13.8h (D_o - D_i)$$

Substituting into the Dittus-Boelter correlation for single phase flow one obtains,

$$h = \frac{.33}{(D_o - D_i)^{.8} (D_o - D_i)}$$

Frictional pressure drop considerations

$$\Delta P_f = \frac{f \rho v^2}{2g} \frac{L}{dh}$$

with

$$f = .3164 Re^{-.25}$$

$$L = 16 \text{ feet}$$

$$v = \frac{w}{\rho A}$$

$$\rho = \frac{\rho_{460^\circ F} + \rho_{680^\circ F}}{2} = 50 \text{ lbm/ft}^3$$

One obtains:

$$\Delta P_f = 13.6 \times 10^{-7} \frac{(D_o + D_i)^{.25}}{(D_o^2 - D_i^2)^2 (D_o - D_i)} \text{ (psia)}$$

Assuming that $\frac{dP}{dT} \approx 1$, then $\Delta P \approx \Delta T$

Hence, for a $\Delta T = T_w - T_F$ of 10°F

this would correspond to a $\Delta P \approx 10 \text{ psia}$

Thus, from Newton's Law of Cooling:

$$\frac{(Q/A)}{h} = T_w - T_F = \Delta T \approx \Delta P$$

Assuming $(Q/A) = 12,500 \text{ BTU/hr.ft}^2$

Thus,

$$\frac{(Q/A)}{h} = \frac{12,500 (D_o + D_i)^{.8} (D_o - D_i)}{.33}$$

$$\frac{(Q/A)}{h} = 37,500 (D_o + D_i)^{.8} (D_o - D_i)^*$$

*This expression must be included in the equation for the boiler inlet pressure to preclude the possibility of $T_w > T_{\text{sat}}$.

The expression for the boiler inlet pressure as a function of concentric tube diameter becomes:

$$P_{\text{boiler inlet}} = \frac{13.6 \times 10^{-7} (D_o + D_i)^{.25}}{(D_o^2 - D_i^2)^2 (D_o - D_i)} + 91.3 + \frac{37,500 (D_o + D_i)^{.8} (D_o - D_i)}{}$$

This expression is plotted in Figure 2.5.2-1.

Obviously, the lowest inlet pressure would be desirable since this implies the lowest operating temperatures. However, it would be quite difficult to insert a .33 in. O.D. concentric tube in a .402" I.D. boiler tube. Several butt welds penetrated the tube wall and also there would be considerable difficulty in measuring temperatures with the thermocouple probes available. Thus, it was decided to use a .25" O.D. concentric tube.

Figure 2.5.2 - 1 Boiler Inlet Pressure As a Function of Concentric Tube Diameter

The original orifice design consisted of 4 holes drilled at an angle with diameters of .005". This idea was discarded because of the possibility of orifice malfunction. Another design allowed for a .03" hole drilled at a 45° angle with the horizontal. When the tests were conducted, the orifice clogged. Finally, it was decided to sacrifice the high pressures in the subcooled region, and use an orifice with a larger hole. The final design used in the testing consisted of one hole drilled perpendicular to the orifice surface with a diameter of .093". Two of these orifices were used to allow for a total pressure drop of 12 psia across them. It is realized that with the final orifice design, less subcooling in the first length of tube could result in net vapor generation of the fluid before reaching the orifice. In actual testing this was noticed. Refer to Figures 2.5.2-2 and 2.5.2-3 for orifice designs and location of them in the test loop.

Stability tests were conducted with the 1" pitch, .054 diameter spiral wire insert, the .25" O.D. concentric tube and orifices. The following table depicts the results of using two orifices with .093" diameter holes located at the U-tube connecting the two straight sections of boiler tube. Similar tests were conducted at 650°F and 700°F saturated conditions at the outlet. In all cases stable operation was observed.

Figure 2.5.2-2 Boiler Orifice Designs

NOTE: THE ORIFICES USED IN THE TEST LOOP WERE THE FINAL DESIGN

Figure 2.5.2-3 Orifice Location In Test Loop

STABILITY TEST RESULTS WITH 1" PITCH .054" DIAMETER
SPIRAL WIRE INSERT

T_m °F (Subcooled)	T_{out} (Saturated)	Effect
400; 80 psia	680	Stable Operation
450; 80 psia	680	Stable Operation
500; 80 psia	680	Stable Operation
550; 80 psia	680	Stable Operation
600; 80 psia	680	Stable Operation

FIGURE 2.5.2-4

Also, a stability test was conducted at the 450°F inlet condition with the orifices removed. Stable operation was observed. Refer to Appendix C, Figures C-1 through C-5 for strip chart traces of all swirl wire stability tests.

Heat transfer tests were conducted with various pitches and diameters of swirl wires. The following tables list the runs.

The following table lists system parameters and their values:

Run 1D   .054 dia. - 1" pitch swirl wire
 Run 1E   .054 dia. - 2" pitch swirl wire
 Run 1F   .054 dia. - 1/2" pitch swirl wire

SYSTEM PARAMETERS WITH VARIOUS DIAMETERS AND PITCHES
 OF SPIRAL WIRE INSERTS

<u>Parameter</u>	<u>Run 1D</u>	<u>Run 1E</u>	<u>Run 1F</u>
Flow Rate (lb/hr)	171	171	171
Heat Flux (BTU/hr-ft ²)	12,000	12,000	12,000
Boiler Exit Pressure (psia)	71	71	71
Boiler Inlet Temperature (°F)	460	460	460
Inlet Subcooling (°F)	190	190	190
Inlet Subcooled ΔP (psia)	2	2	2
Exit Quality	---	---	100%

Run 1G   .085 dia. - 2" peak to peak

<u>Parameter</u>	<u>Run 1G</u>
Flow Rate (lb/hr)	171
Heat Flux (BTU/hr-ft ²)	12,000
Boiler Exit Pressure (psia)	71
Boiler Inlet Temperature (°F)	460
Inlet Subcooling (°F)	190
Inlet Subcooled ΔP (psia)	2
Exit Quality	100%

FIGURE 2.5.2-5

Noticeable conclusions are that Runs 1A through 1C are unstable and 1D through 1G are stable. The definition of stability can be understood and appreciated by referring to Appendix A. Notice the pressure spikes in the boiler inlet pressure in Runs 1A - 1C. Corrected wall and fluid temperature profiles are shown in Figures 2.5.2-6 through 2.5.2-9 for the various pitches and diameters of wire used.

WALL AND FLUID TEMPERATURE PROFILE
WITH SPIRAL WIRE & CONCENTRIC TUBE

Figure 2.5.2 - 6

WALL AND FLUID TEMPERATURE PROFILE
WITH SPIRAL WIRE + CONCENTRIC TUBE

Figure 2.5.2-7

Figure 2.5.2-8

T9

Figure 2.5.2-9

2.6 DATA REDUCTION

One of the primary objectives in conducting these boiler tests is to develop certain empirical equations to predict boiler performance analytically.

Initially, all wall and fluid temperatures were corrected allowing for calibration errors. Next, all inner wall temperatures were determined by assuming a linear $\frac{dT}{dr}$ across the tube wall.

Experimental values of film coefficients were calculated using Newton's Law of Cooling:

Where,

$$h_{exp} = (Q_{net}/A)/(T_{IW}-T_{TF})$$

$$\text{and } Q_{net} = Q_{in} - Q_{loss}$$

A = surface area of boiler tube covered by
clam shell heater

T_{IW} = inner wall temperature

T_{TF} = fluid temperature at inner wall location

In calculating analytical film coefficients the various boiling regions were considered. The discussion to follow is for the bare tube analysis.

Until the point at which wall superheat in the liquid begins, $T_w \geq T_{sat}$, the Dittus-Boelter equation was employed (Region I).

$$(Nu)_{dh} = .023 (Re)_l^{.8} (Pr)_l^{.4}$$

Properties are based on the liquid bulk temperature. In the subcooled nucleate boiling region, that is from the point at which wall superheat in the liquid begins to quality or vapor formation, a superposition of forced convection and nucleate boiling heat fluxes were used (Region II). An original correlation was used to predict nucleate boiling heat fluxes (Region II):

$$(Q/A)_{boil} = C \Delta T_{sat}^n, \text{ where } C = 1.35 \text{ and } n = 2.6$$

based on experimental data.

$$(Q/A)_{\text{forced convection}} = h_{DB} (T_w - T_F)$$

Thus, $(Q/A)_{total} = (Q/A)_{boil} + (Q/A)_{\text{forced convection}}$

$$\text{or, } h_{exp} = \frac{C \Delta T_{sat}^n}{(T_w - T_F)} + h_{DB}$$

Throughout the quality region up to where critical quality was formed, the Dengler-Addoms correlation was used (Region III):

$$\text{Where, } h_B = h_l F(3.5/X_{tt}^{.5})$$

h_l = Dittus-Boelter based liquid flow and properties

$$F = .67[\Delta T_w - \Delta T_1] \left(\frac{dP}{dT} \right) \left(\frac{D_1}{\sigma} \right)^{.1} \quad 1.0$$

$$V_{avg} = \text{average volume}; \quad \frac{1}{V_{avg}} = \rho_l R_l + (1 + R_l) \rho_v \quad \frac{\text{lbm}}{\text{ft}^3}$$

R_l = liquid fraction

$$\Delta T_w = T_w - T_F \text{ (}^\circ\text{F)}$$

$$\Delta T_1 = 10 \left[\frac{W V_{avg}}{3600 A_F} \right]^{.3} \text{ (}^\circ\text{F)}$$

$$\begin{aligned} A_F &= \text{flow area} = \frac{\pi D_1^2}{4} \text{ (bare tube) ft}^2 \\ &= \frac{\pi}{4} (D_o^2 - D_1^2) \text{ (swirl wire)} \end{aligned}$$

D_o = I.D. of boiler tube

D_1 = wire diameter

$$\frac{dP}{dT} = \frac{778 \text{ hfg}}{t_{sat} (V_v - V_l)} \quad \frac{\text{lb}}{\text{ft}^2}$$

$$X_{tt}^2 = \frac{\frac{dP}{dl}_l}{\frac{dP}{dl}_v} \quad \text{(Martinelli Parameter)}$$

The above correlation was in general lower by a factor of 1.4 with $F = 1.0$.

In the 200 lbm/hr heat transfer run an experimental and analytical critical quality was determined. Therefore, the following correlation was used in the data reduction for this case (Region IV):

$$h_B = h_{DBV} \left(\frac{h_{B, Y_t}}{h_{DBV}} \right)^{\left[\frac{1-y}{1-y_t} \right]} \quad (\text{BTU/hr-ft}^2\text{°F})$$

Where,

h_{DBV} = Dittus-Boelter based on vapor properties ($F = 1.4$)

h_B, Y_t = Dengler-Addoms correlation calculated @ the critical quality

y = quality

Finally, in the mist flow regime the Dittus-Boelter equation was used (Region V):

$$(Nu)_{dh} = .023 (Re)_v^{.8} (Pr)_v^{.4}$$

In most of the runs where quality existed at the boiler outlet, region V did not exist. Region IV probably existed in all runs, however, it was analyzed only in Run 1C. In Runs 1D - 1G the heat flux was constant up to a quality less than the critical quality. Therefore, it was decided not to correlate Region IV in Runs 1D - 1G, the mist flow regime. Region IV consists of two phase flow ranging from the critical quality

to 100% quality. Region V consists of superheated vapor which also was analyzed only for Run 1C.

In all runs the Dengler-Addoms correlation was used in the quality region. In addition another correlation was used in the quality region:

$$(Nu)_{dh} = .023 (Re)_v^{.8} (Pr)_l^{.4}$$

The above correlation fit the data quite well. This suggests that the film coefficient in the quality region is a strong function of the quality. Film coefficient profiles for Runs 1A - 1G are shown in Figures 2.6-1 through 2.6-7. Stable and unstable operation is illustrated in the strip chart traces located in Appendix A. Note the pressure spikes in Runs 1A - 1C and steady state pressures in Runs 1D - 1G.

Figure 2.6-1

Figure 2.6-2

HEAT TRANSFER COEFFICIENT PROFILE RUN IF

Figure 2.6-3

HEAT TRANSFER COEFFICIENT PROFILE RUN IG

Figure 2.6-4

HEAT TRANSFER COEFFICIENT PROFILE BARE TUBE

RUN 1A

○ EXPERIMENTAL RESULTS

× ANALYTICAL PREDICTIONS

HEATERS 1-13 @ 1000
HEATERS 14-16 @ 30

Figure 2.6-5

Figure 2.6 - 6 Heat Transfer Coefficient Profile Bare Tube

Figure 2.6-6

Figure 2.6 - 7 Heat Transfer Coefficient as a Function of Boiler Tube Length

Pressure drop studies were also undertaken. The results indicate that in order to achieve stable operation, approximately twice the pressure drop in the second 16 feet of boiler tube is required. Two phase pressure drops were determined analytically and were lower by a factor of two. Analytical two phase pressure drops were calculated as follows:

Superficial Reynold's numbers were first calculated:

$$Re_l = \frac{\bar{U}_l dh}{\mu_l v_l}, \quad Re_v = \frac{\bar{U} dh}{\mu_v v_v}$$

$$\bar{U}_l = \frac{4 w_l}{(D_2^2 - D_1^2)}, \quad \bar{U}_v = \frac{4 w_v}{(D_2^2 - D_1^2)}$$

where,

D_2 = tube diameter

D_1 = wire diameter

$\bar{U}$ = superficial velocity of all liquid or vapor flow

If $Re_l \geq 2000$ then the friction factor becomes:

$$f_l = .3164/Re_l^{.25}; \quad Re_l \leq 2000 \text{ then } f_l = 64/Re_l$$

Also, $Re_v \geq 2000$; $f_v = .3164/Re_v^{.25}$

and $Re_v \leq 2000$; $f_v = 64/Re_v$

$$\text{Now, } \frac{\Delta P}{\Delta L}_l = \frac{f_l \rho \bar{U}_L^2}{2g_c D_H} \quad \text{and} \quad \frac{\Delta P}{\Delta L}_v = \frac{f_v \rho \bar{U}_v^2}{2g_c D_H}$$

Then the Martinelli Parameter becomes:

$$X_{tt} = \sqrt{\frac{(\Delta P/\Delta L)_l}{(\Delta P/\Delta L)_v}}$$

In order to find the two phase pressure gradient the ϕ function must be known:

Re _v	Re _l	ϕ
<2000	<2000	ϕ_{vv}
≥2000	<2000	ϕ_{vt}
<2000	≥2000	ϕ_{tv}
≥2000	≥2000	ϕ_{tt}

By checking the values of Re_v and Re_l the proper value of ϕ will be determined:

$$\text{Then, } \frac{\Delta P}{\Delta L}_{\text{two phase}} = \phi^2 \frac{\Delta P}{\Delta L}_l$$

In addition to two phase frictional pressure drop considerations, momentum pressure drops must be added.

$$\Delta P_{MOM} = w_l(0^+) \bar{U}_l(0^+) + w_v(0^+) \bar{U}_v(0^+) - w_l(0^-) \bar{U}_l(0^-) - w_v(0^-) \bar{U}_v(0^-)$$

where 0^+ indicates an infinitesimal distance preceeding the point of interest and 0^- indicates an infinitesimal distance proceeding the point of interest.

The total pressure drop over a given length becomes:

$$\Delta P_{total} = \int_1^2 \frac{dP}{dL} \text{ two phase } dx + \Delta P \int_1^2 \text{ momentum.}$$

Using the above equation it was found that predicted pressure drops in the boiling region (second length of boiler tube) were a factor of two under the actual values with the spiral wire inserts. Plots of $\Delta P_{\text{two phase}}$ versus wire pitch and $\Delta P_{\text{two phase}}$ versus wire diameter are shown in Figure 2.6-8. Experimental and analytical results for Runs 1A and 1B are shown in Figures 2.6-9 and 2.6-10.

EXPERIMENTAL AND ANALYTICAL PRESSURE
DROPS AS A FUNCTION OF WIRE PITCH AND
DIAMETER

Figure 2.6-8

TWO PHASE PRESSURE DROP
($\Delta P/D$) $\sim$ P.S.I./ FT.

Figure 2.6-9 Two Phase Pressure Drop as a Function
of Heated Length

Figure 2.6-10 Two Phase Pressure Drop as a Function of Heated Length

2.7 DISCUSSION AND CONCLUSIONS

2.7.1 Malfunction of System Components During Loop Operation

With the original orifice design, the hole was so small that clogging had occurred. There are two major operational problems with the rig. If there is a pump power failure or flow failure, the thermal inertia of the clam shell heaters causes a significant over-temperature (above 750°F) in the boiler. Hence, the Dowtherm degrades and results in carbon particles suspended in the flow. Since the modified orifice had a small hole drilled at an angle, over-temperaturing the boiler resulted in a clogged orifice. As a result, a new orifice design was initiated. With the new design, the ΔP across the orifice was not as large but the problem of orifice clogging was eliminated. In several runs the cavitating venturi clogged (due to overheating the Dowtherm). To minimize this problem a flow filter was installed downstream of the flow meter.

Other minor problems included buckling of the regenerator housing that was not initially reinforced, faulty thermocouples, poorly wired clam shell heaters that frequently shorted out, expanded "o" ring used in vacuum chamber connections and periodic degradation (because of overheating above 750°F) and consequent changing of the working fluid.

2.7.2 Conclusions

Through extensive testing, it was shown that empirical correlations could predict boiler performance. By insertion of spiral wires in the high quality region and a concentric tube in the subcooled and low quality region, stable boiler operation could be achieved. Also, with these inserts, higher film coefficients could be attained as compared to bare tube runs. This is to be expected, since higher velocities in the subcooled region result in larger film coefficients according to the Dittus-Boelter correlation. In the quality region the swirl wire increased turbulence thereby decreasing the resistance to heat transfer or increasing the film coefficient. Also, it was found that an acceptable correlation predicted the nucleate boiling regime for all cases (1A - 1G) tested. This correlation was of the form $(Q/A)_b = 1.35 \Delta T_{sat}^{2.6}$. Also, no dry wall conditions were observed with the swirl wire inserts in the test section. By use of the swirl wire and concentric tube, the boiler ran stable in Runs (1D - 1G), although unstable operation was observed in the bare tube runs Run (1A - 1C).

Stability tests indicated that inlet throttling had no effect on boiler stability in either the bare tube testing or with the tube inserts. Also, it was discovered that orifice installation had no effect,

other than higher total pressures drops, on
stable boiler operation.

APPENDICES

APPENDIX A

STRIP CHART TRACES OF HEAT
TRANSFER RUNS 1A - 1G

Appendix A consists of the Heat Transfer Runs 1A - 1G. Figure A-1 is a strip chart trace of (bare tube) Run 1A. The boiler inlet temperature was 450°F. Notice that with this inlet temperature, even 40 psid inlet throttling was ineffective in preventing pressure oscillations in the first few feet of boiler tube. As the fluid reaches the quality region the oscillations damp out indicating that annular flow is established and the vapor greatly damps the pressure pulses. The outlet pressure is nearly constant at 56 psig.

Run 1B (Figure A-2) is another bare tube heat transfer run at 500°F boiler inlet. Pressure oscillations were nearly damped out. In comparison to Run 1A notice that the pressure level was greater, the boiler temperature was also greater and inlet throttling was only 10 psid.

Figure A-2 Strip Chart Recording of Data, Run 1 B

Run 1C (Figure A-3) was the last of a series of bare tube heat transfer runs. In this run the flow was increased from .35 gpm to .45 gpm. With the increased flow rate and approximately the same inlet temperature and pressure as Run 1A, the pressure oscillations were considerably greater in amplitude and frequency. This is so since with more flow there is more momentum change and thus larger pressure spikes. No inlet throttling was used. Also, it should be noted that because of a shift in inlet temperature the pressure oscillations damped out independent of inlet throttling. The transition to stable operation occurred at a boiler inlet temperature of 500°F.

Figure A-3

Strip Chart Data, Run 1 C

In Runs 1D - 1G (Figures A-4 thru A-7) stable boiler operation was observed with no inlet throttling. In each run a different pitch spiral wire was inserted in the quality region. In all cases the boiler temperature was fixed at 450°F. Also, the pressure level was fixed at 56 psig. The concentric tube insert tended to prevent slug-plug flow in the subcooled region. The spiral wire resulted in about twice the pressure drop in the boiling region as in the bare tube runs.

Figure A-4 Strip Chart Data, Run 1 D

page blank

Figure A-6 Strip Chart Data, Run 1 F

Figure A-7

Strip Chart Data, Run 1 G

APPENDIX B

STRIP CHART TRACES OF BARE TUBE VARIABLE HEAT FLUX STABILITY TESTS

Figure B-1 Variable Heat Flux Stability Tests, Case I $\dot{Q} = .175$ GPM

Figure B-2 Variable Heat Flux Stability Tests, Case II $\dot{Q} = .2$ GPM

Figure B-3 Variable Heat Flux Stability Test, Case III $\dot{Q} = .25 \text{ GPM}$

Figure B-4 Variable Heat Flux Stability Tests, Case IV $\dot{Q} = .3$ GPM

APPENDIX C

STRIP CHART TRACES OF
STABILITY TEST WITH
SWIRL WIRE INSERTS OF
VARIOUS PITCHES AND DIAMETERS

Appendix C (Figures C-1 thru C-5) consisted of spiral wire stability tests. In each case tested the inlet subcooling was increased 50°F. Thus, Case I consisted of a boiler inlet temperature of 400°F and Case V consisted of a 600°F inlet. In all tests the outlet was fixed at 680°F saturated conditions. Case I thru Case IV show no gage pressure at the inlet to the inlet throttling valve. The pressure transducer at this station was not functional. In all cases the boiler showed negligible pressure oscillations.

Figure C-1 Stability Tests with Swirl Wire (1" pitch, .054 dia.)
 Case I

Figure C-2

Stability Tests with 1" Pitch - .054 dia Spiral Wire
 Case II

Figure C-3 Stability Tests with 1" Pitch - 0.54 dia Swirl Wire
 Case III

Figure C-4 | Stability Tests with 1" Pitch - .054 Dia Swirl Wire
 Case IV

Figure C-5 Stability Tests with 1" Pitch - .054 Dia Swirl Wire
Case V

APPENDIX D

COMPUTER OUTPUTS OF FILM
COEFFICIENTS OF RUN 1A AND RUN 1E

RUN 1A

DITTUS-BOELTER	STATION	ANALYTICAL FILM COEFFICIENT		HEATED LENGTH
0.1331003E 03	2			
0.1346615E 03	3			
0.1365189E 03	4			
0.1382937E 03	5			
DITTUS-BOELTER	QOAPB			
0.1400197E 03	0.5209E 02	0.1404349E 03		54.6000
0.1415465E 03	0.2443E 04	0.1612623E 03		66.1000
0.1437877E 03	0.7258E 04	0.2068166E 03		79.0000
0.1466121E 03	0.5650E 04	0.2058408E 03		90.6000
0.1504334E 03	0.5102E 04	0.2170290E 03		103.4000
0.1543398E 03	0.4728E 04	0.2315464E 03		114.9000
0.1592817E 03	0.4718E 04	0.2636033E 03		127.8000
0.1637594E 03	0.6744E 04	0.3571683E 03		139.3000
0.1666026E 03	0.5505E 04	0.4018353E 03		152.0000
0.1674360E 03	0.5437E 04	0.4380797E 03		163.6000
0.70307E 03	203.3000	0.2984E 00	0.1558E 00	0.2197E-02
0.72216E 03	214.8000	0.2252E 00	0.1773E 00	0.1712E-02
0.72627E 03	227.7000	0.1649E 00	0.1989E 00	0.1234E-02
0.71448E 03	239.1000	0.1241E 00	0.2151E 00	0.8728E-03
0.68096E 03	252.0000	0.8662E-01	0.2266E 00	0.5298E-03
0.62647E 03	263.6000	0.5819E-01	0.2308E 00	0.2858E-03
0.00000E 00	276.3000	0.0000E 00	0.1210E 00	0.0000E 00
0.00000E 00	288.1000	0.0000E 00	0.1210E 00	0.0000E 00
0.00000E 00	300.6000	0.0000E 00	0.1210E 00	0.0000E 00
0.00000E 00	312.3000	0.0000E 00	0.1211E 00	0.0000E 00
0.00000E 00	325.1000	0.0000E 00	0.1211E 00	0.0000E 00
0.00000E 00	336.5000	0.0000E 00	0.1212E 00	0.0000E 00
0.00000E 00	349.3000	0.0000E 00	0.1221E 00	0.0000E 00
0.00000E 00	361.1000	0.0000E 00	0.1225E 00	0.0000E 00
0.00000E 00	373.7000	0.0000E 00	0.1228E 00	0.0000E 00
0.00000E 00	385.3000	0.0000E 00	0.1232E 00	0.0000E 00

Figure D-1 Computer Output of Run 1 A

DITTUS-BOELTER	STATION	ANALYTICAL FILM COEFFICIENT		HEATED LENGTH
0.2395588E 03	2			
0.2435508E 03	3			
0.2465447E 03	4			
0.2482089E 03	5			
0.2501960E 03	6			
0.2523215E 03	7			
0.2549927E 03	8			
0.2578107E 03	9			
0.2617776E 03	10			
0.2663871E 03	11			
0.2713328E 03	12			
DITTUS-BOELTER	QQAPB			
0.2713328E 03	0.5103E 03	0.2812614E 03		127.8000
0.2747897E 03	0.2227E 03	0.2784653E 03		139.3000
0.2795218E 03	0.9233E 03	0.2959471E 03		152.0000
0.2847683E 03	0.5877E 03	0.2979669E 03		163.6000
0.2915429E 03	0.9767E 03	0.3195838E 03		176.4000
0.2982255E 03	0.5822E 04	0.4644810E 03		188.0000
0.71681E 03	203.3000	0.3550E 00	0.1707E 00	0.3027E-02
0.74474E 03	214.8000	0.2649E 00	0.1975E 00	0.2393E-02
0.75794E 03	227.7000	0.1883E 00	0.2278E 00	0.1722E-02
0.75153E 03	239.1000	0.1410E 00	0.2493E 00	0.1234E-02
0.72326E 03	252.0000	0.9872E-01	0.2667E 00	0.7696E-03
0.67342E 03	263.6000	0.6722E-01	0.2753E 00	0.4333E-03
0.62917E 03	276.3000	0.5082E-01	0.2760E 00	0.2756E-03
0.61951E 03	288.1000	0.4787E-01	0.2764E 00	0.2500E-03
0.60815E 03	300.6000	0.4465E-01	0.2769E 00	0.2231E-03
0.59744E 03	312.3000	0.4186E-01	0.2771E 00	0.2004E-03
0.64319E 03	325.1000	0.3223E-01	0.2063E 00	0.1216E-03
0.62210E 03	336.5000	0.3088E-01	0.2093E 00	0.1139E-03
0.59728E 03	349.3000	0.2935E-01	0.2126E 00	0.1053E-03
0.57325E 03	361.1000	0.2791E-01	0.2157E 00	0.9731E-04
0.54768E 03	373.7000	0.2644E-01	0.2187E 00	0.8916E-04
0.52270E 03	385.3000	0.2505E-01	0.2217E 00	0.8158E-04

Run 1E

Figure D-2 Computer Output of Run 1 E