

AD-A158 757

2

AFWAL-TR-85-3024



SPACECRAFT STRAIGHT-TUBE  
EVAPORATOR DESIGN

Robert E. Eastman

Environmental Control Branch  
Vehicle Equipment Division

May 1985

Final Report for Period 1 July 1983 - 31 March 1985

Approved for public release; distribution unlimited.

FLIGHT DYNAMICS LABORATORY  
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES  
AIR FORCE SYSTEMS COMMAND  
WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433-6553

DTIC  
ELECTE  
AUG 29 1985  
S D G

DTIC FILE COPY

85 2 18 06 5

NOTICE

When Government drawings, specifications, or other data are used for any purpose other than in connection with a definitely related Government procurement operation, the United States Government thereby incurs no responsibility nor any obligation whatsoever; and the fact that the government may have formulated, furnished, or in any way supplied the said drawings, specifications, or other data, is not to be regarded by implication or otherwise as in any manner licensing the holder or any other person or corporation, or conveying any rights or permission to manufacture use, or sell any patented invention that may in any way be related thereto.

This report has been reviewed by the Office of Public Affairs (ASD/PA) and is releasable to the National Technical Information Service (NTIS). At NTIS, it will be available to the general public, including foreign nations.

This technical report has been reviewed and is approved for publication.

Robert E. Eastman  
ROBERT E. EASTMAN, 1Lt, USAF  
Project Engineer

Paul D. Lindquist  
PAUL D. LINDQUIST  
Technical Area Manager  
Advanced Cryogenics Systems Group  
Environmental Control Branch

FOR THE COMMANDER

Solomon R. Metres  
SOLOMON R. METRES, Chief  
Vehicle Equipment Division  
Flight Dynamics Laboratory

|                    |                                     |
|--------------------|-------------------------------------|
| Accession For      |                                     |
| NTIS GRA&I         | <input checked="" type="checkbox"/> |
| DTIC TAB           | <input type="checkbox"/>            |
| Unannounced        | <input type="checkbox"/>            |
| Justification      |                                     |
| By _____           |                                     |
| Distribution/      |                                     |
| Availability Codes |                                     |
| Dist               | Avail and/or<br>Special             |
| A11                |                                     |



"If your address has changed, if you wish to be removed from our mailing list, or if the addressee is no longer employed by your organization please notify AFWAL/FIEE, W-PAFB, OH 45433 to help us maintain a current mailing list".

Copies of this report should not be returned unless return is required by security considerations, contractual obligations, or notice on a specific document.

UNCLASSIFIED

SECURITY CLASSIFICATION OF THIS PAGE

AD-A158757

## REPORT DOCUMENTATION PAGE

|  |   |   |   |
|--|---|---|---|
| 1a. REPORT SECURITY CLASSIFICATION<br><b>UNCLASSIFIED</b>  |   | 1b. RESTRICTIVE MARKINGS  |   |
| 2a. SECURITY CLASSIFICATION AUTHORITY  |   | 3. DISTRIBUTION/AVAILABILITY OF REPORT<br><br>Approved for public release;<br>distribution unlimited. |   |
| 2b. DECLASSIFICATION/DOWNGRADING SCHEDULE  |   | 4. PERFORMING ORGANIZATION REPORT NUMBER(S)<br><br>AFWAL-TR-85-3024                                   |   |
| 6a. NAME OF PERFORMING ORGANIZATION<br>Air Force Wright<br>Aeronautical Laboratories   |   | 6b. OFFICE SYMBOL<br>(If applicable)<br>AFWAL/FIEE  | 5. MONITORING ORGANIZATION REPORT NUMBER(S) |
| 6c. ADDRESS (City, State and ZIP Code)<br>Wright-Patterson Air Force Base,<br>Ohio 45433-6553  |   | 7a. NAME OF MONITORING ORGANIZATION   |   |
| 8a. NAME OF FUNDING/SPONSORING ORGANIZATION  |   | 8b. OFFICE SYMBOL<br>(If applicable)  | 7b. ADDRESS (City, State and ZIP Code)      |
| 8c. ADDRESS (City, State and ZIP Code)   |   | 9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER   |   |
| 11. TITLE (Include Security Classification)<br>Spacecraft Straight-Tube Evaporator Design  |   | 10. SOURCE OF FUNDING NOS.  |   |
|  |   | PROGRAM ELEMENT NO.<br>62201F   | PROJECT NO.<br>2402                         |
|  |   | TASK NO.<br>04  | WORK UNIT NO.<br>50                         |
| 12. PERSONAL AUTHOR(S)<br>Robert E. Eastman  |   | 14. DATE OF REPORT (Yr., Mo., Day)<br>May 85  |   |
| 13a. TYPE OF REPORT<br>Final   | 13b. TIME COVERED<br>FROM 7-1-83 TO 3-31-85 | 15. PAGE COUNT<br>85  |   |
| 16. SUPPLEMENTARY NOTATION   |   |   |   |
| 17. COSATI CODES   |   | 18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)                     |   |
| FIELD  | GROUP                                       | Forced convection boiling   |   |
| 20   | 04  | Boiling heat transfer   |   |
| 20   | 13  | Two-phase flow  |   |
| 19. ABSTRACT (Continue on reverse if necessary and identify by block number) Two-Phase (liquid-vapor) heat transfer systems are being considered for use on future spacecraft, operating at high energy levels. The design of straight-tube evaporators was studied for this application. The forced convective boiling phenomenon was examined to determine the characteristics of boiling flow and possible effects of gravitational acceleration. Experiments with straight-tube evaporators indicated that pressure fluctuations caused by slug flow may hamper efforts to build two-phase cooling systems. Upstream orifices and twisted tape inserts were investigated to overcome this problem. A method was also developed to size an evaporator based on the best available correlation equations. Some errors were corrected for flow-boiling heat transfer equations found in the literature. |   |   |   |
| 20. DISTRIBUTION/AVAILABILITY OF ABSTRACT<br>UNCLASSIFIED/UNLIMITED <input checked="" type="checkbox"/> SAME AS RPT. <input type="checkbox"/> DTIC USERS <input type="checkbox"/>  |   | 21. ABSTRACT SECURITY CLASSIFICATION<br><b>UNCLASSIFIED</b>   |   |
| 22a. NAME OF RESPONSIBLE INDIVIDUAL<br>WILLIAM HASKIN  |   | 22b. TELEPHONE NUMBER<br>(Include Area Code)<br>(513) 255-4853  | 22c. OFFICE SYMBOL<br>AFWAL/FIEE            |

## PREFACE

The analytical and experimental work described in this report was done in response to a need to develop more efficient and more effective heat transfer methods for future military space vehicles. The specific purpose of this study was to develop a method for designing straight-tube evaporators for use in spacecraft cooling systems. Although this study addressed the specific problems associated with designing an evaporator for operation in a weightless environment, the method of design is applicable to all straight-tube evaporators.

The work was conducted in the Environmental Control Branch of the Vehicle Equipment Division of the Flight Dynamics Laboratory at Wright-Patterson Air Force Base. This is the final technical report for work unit No. 24020450.

This report has also been submitted as a Master of Science thesis for the Air Force Institute of Technology, School of Engineering (AFIT/ENA) at Wright-Patterson Air Force Base. The thesis report number is AFIT/GA/AA/85M-1. The thesis is dated March 1985.

Many people provided assistance during the course of the work. Special appreciation is due to the thesis advisor, Dr. J.E. Hitchcock of the Air Force Institute of Technology. Additional thanks is extended to Mr. Larry Coulthard of the Flight Dynamics Laboratory for his assistance during the experimental portion of the work. A particular thanks is also due to Ms. Julie Roberts for her help and patience in typing the thesis.

## CONTENTS

| <u>Section</u>   | <u>Page</u> |
|--|-------------|
| List of Figures  | vii         |
| Tables   | ix          |
| Notation   | x           |
| I    Introduction  | 1           |
| II   Review of the Forced Convective Boiling Process           | 4           |
| General Discussion   | 4           |
| Subcooled Boiling  | 7           |
| Saturated, Nucleate Boiling                                    | 12          |
| Two-Phase, Forced Convection                                   | 14          |
| Liquid-Deficient Region  | 17          |
| III  Flow Visualization Experiment                             | 18          |
| Experimental Apparatus   | 18          |
| Instrumentation  | 18          |
| Evaporator   | 20          |
| Results of Flow Visualization                                  | 21          |
| IV  Two-Phase, Heat-Transfer Equations                         | 24          |
| V   Proposed Method for Design of Straight-Tube<br>Evaporators | 36          |
| Design Method  | 36          |
| Calculation of the Heat-Transfer Coefficient                   | 38          |
| Comparison with Data   | 42          |
| Flow Stabilization   | 44          |
| Pressure-Drop Calculation                                      | 52          |
| VI  Evaporator-Design Program                                  | 54          |

|     |                                 |    |
|-----|---------------------------------|----|
| VII | Conclusions                     | 60 |
|     | References                      | 61 |
|     | Appendix A: Sample Calculations | 65 |

## LIST OF FIGURES

| <u>Figure</u>   | <u>Page</u> |
|---|-------------|
| 1. Regions associated with forced-convective boiling in tubes . . . . .                               | 5           |
| 2. Variation of the heat-transfer coefficient along a tube during forced-convective boiling . . . . . | 6           |
| 3. Experimental apparatus . . . . .   | 19          |
| 4. Reynolds number factor $F$ versus the Lockhart-Martinelli parameter $X_{tt}$ . . . . .             | 31          |
| 5. Supression factor $S$ versus the Reynolds number .   | 31          |
| 6. Flow chart for calculating the length of an evaporator with a constant heat flux . . . . .         | 39          |
| 7. Flow chart for calculating the length of an evaporator with a constant wall temperature . .        | 40          |
| 8. Wall temperature versus heat flux for subcooled boiling . . . . .                                  | 43          |

|     |   |    |
|-----|---|----|
| 9.  | Difference between the wall temperature and the bulk-fluid temperature versus the heat flux for subcooled boiling . . . . . | 45 |
| 10. | Heat-transfer coefficient versus quality for low-heat-flux, low-quality boiling . . . . .                                   | 47 |
| 11. | Heat-transfer coefficient versus quality for low-heat-flux, high-quality boiling . . . . .                                  | 48 |
| 12. | Heat-transfer coefficient versus quality for high-heat-flux, low-quality boiling . . . . .                                  | 49 |
| 13. | Heat-transfer coefficient versus quality for high-heat-flux, high-quality boiling . . . . .                                 | 50 |

TABLE

1. Pressure readings across the test section and the orifice for various flow conditions.

23

## NOTATION

| <u>Symbols</u> |                                 | <u>Units</u>      |
|----------------|---------------------------------|-------------------|
| a              | Acceleration                    | $m/\varepsilon^2$ |
| $C_p$          | Specific Heat                   | $J/kgK$           |
| d              | Diameter                        | m                 |
| f              | Friction Factor                 |                   |
| G              | Mass Flux                       | $kg/m^2s$         |
| g              | Acceleration of Gravity         | $m/s^2$           |
| $g_c$          | Gravitational Constant          | $kgm/Nts^2$       |
| h              | Heat-Transfer Coefficient       | $W/m^2K$          |
| $h_{fg}$       | Latent Heat of Vaporization     | $J/kg$            |
| k              | Conductivity                    | $W/mK$            |
|                | Length                          | m                 |
| N              | Value Defined by Equation IV-33 |                   |
| p              | Pressure                        | $N/m^2$           |
| Pr             | Prandtl Number                  |                   |
| q              | Heat Flux                       | $W/m^2$           |
| Q              | Energy                          | W                 |
| r              | Radius                          | m                 |
| Re             | Reynolds Number                 |                   |
| T              | Temperature                     | K                 |
| u              | Velocity                        | $m/s$             |
| x              | Quality                         |                   |
| y              | Distance                        | m                 |

|   |   |                    |
|---|---|--------------------|
| Y | Number of Diameters per 180 Degree<br>Twist (Twist Ratio) |                    |
| Z | Distance along tube                                       | m                  |
|   | Value Defined by Equation IV-32                           | 1/K                |
|   | Viscosity   | kg/ms              |
|   | Surface Tension   | N/m                |
|   | Density   | kg/m <sup>3</sup>  |
|   | Specific Volume   | m <sup>3</sup> /kg |

Subscripts

|       |  |
|-------|--|
| a     | Axial  |
| b     | Bulk   |
| c     | Critical   |
| f     | Film   |
| FC    | Forced Convection                                |
| IB    | Incipient Boiling Point                          |
|       | Liquid   |
| L     | Liquid   |
| LO    | Liquid Alone                                     |
| LP    | Liquid Phase                                     |
| max   | Maximum Value                                    |
| NB    | Nucleate Boiling                                 |
| OTPFC | Onset of Two-Phase, Forced-Convection<br>Boiling |
| ONB   | Onset of Nucleate Boiling                        |

the radiator is quite large. Since all the heat must be radiated to space, the radiator also gets very big and heavy.

The final method presently available for transporting waste heat is the heat pipe. A heat pipe is nothing more than a tube, capped at both ends, that contains a small amount of liquid and a wick. When heat is added to one end (the evaporator) the liquid boils. Because of the increase in vapor pressure, the vapor is transported to the other end (the condenser) where it condenses as heat is taken out. The fluid is then wicked back to the evaporator to complete the cycle. The heat pipe is an extremely efficient and reliable method of transporting heat. The problem with heat pipes, however, is that they can only transport a limited amount of heat for a limited distance<sup>[1]</sup>. It is possible to put together many heat-pipe bundles in series to overcome this problem, but, again weight becomes a problem<sup>[2]</sup>.

Due to the shortcomings of the previous methods, it has been proposed to use two-phase-fluid loops<sup>[1]</sup>. Like a one-phase-fluid loop, these systems would involve pumping a fluid around a loop. Unlike one-phase systems, two-phase systems would use the latent heat of vaporization rather than the sensible heat of the circulating fluid. Since the latent heat is much greater than the sensible heat, this system can use a much smaller flow rate. In fact, preliminary calculations have shown that the pump power required will be approximately two orders of magnitude less than a comparable one-phase system. Furthermore, the entire system is essentially isothermal which will mean much smaller and lighter radiators. This system, however, has many design problems that

must be worked out before it can be used. Therefore, many research programs have been started to solve the problems associated with two-phase systems.

This study deals with the design of the evaporator. More specifically, the objects of this program are to do an in-depth review of the forced-convective boiling process to determine the effect that gravity will have on it, to set up an experimental evaporator to visualize the general performance and problems of straight-tube evaporators, to investigate methods of solving the problems associated with them, to review the methods available for calculating local heat-transfer coefficients, to develop a best possible approach, to develop a method for designing straight-tube evaporators, and to develop a computer program to design straight-tube evaporators.

## II REVIEW OF THE FORCED-CONVECTIVE BOILING PROCESS

In order to understand the relationship between the forced-convective boiling process and gravitational acceleration, a thorough study of the boiling process was conducted. The forced-convective boiling process is explained and the manner in which gravitational acceleration affects the heat-transfer process is examined in this section.

### GENERAL DISCUSSION

The forced-convective boiling process can be broken down into five regions: one-phase forced convection; subcooled boiling; saturated, nucleate boiling; two-phase forced convection; and the liquid-deficient region. This process is shown in Figure 1, and the variation of the heat-transfer coefficient associated with each region is shown in Figure 2. The one-phase, forced-convection region is the initial condition normally encountered in an evaporator. It is possible, however, that this region may be absent. In this region heat is conducted into and is swept away by the fluid.

As the temperature of the wall increases along the length, it reaches a point where bubble nucleation can begin. This is the start of the subcooled-boiling region. As the wall temperature increases further, the bubbles grow larger and begin to depart the surface. Since the bulk of the fluid is still subcooled, the bubbles condense upon leaving the warmer boundary layer. Although the net amount of vapor generated in this region is very small, the heat-transfer coefficient is very large.

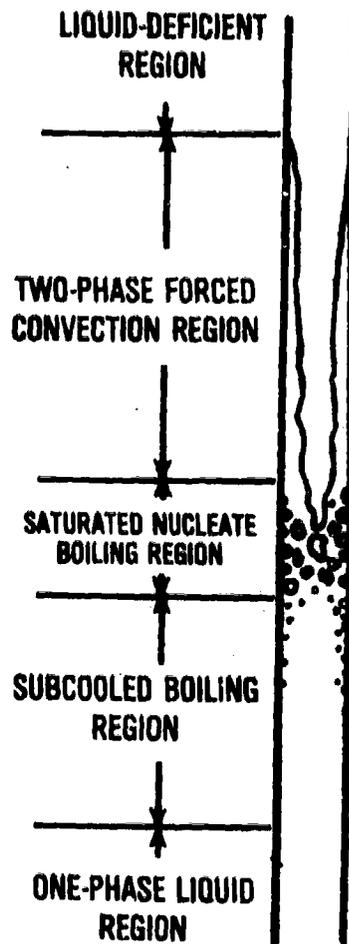


Figure 1: Regions associated with forced-convective boiling in tubes.

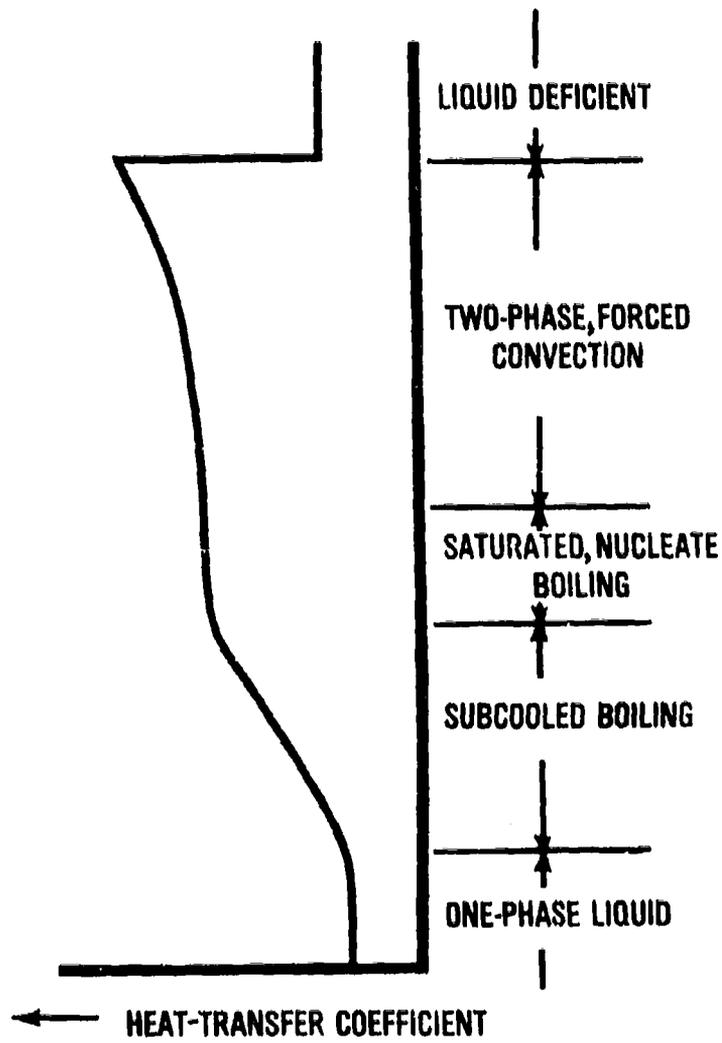


Figure 2: Variation of the heat-transfer coefficient along a tube during forced-convective boiling.

When the bulk fluid temperature reaches saturation conditions, saturated-nucleate boiling occurs. In this region, normal nucleate boiling occurs with a corresponding increase in vapor quality. Over this region the heat-transfer coefficient stays essentially constant.

As more vapor is produced, the liquid layer on the surface becomes thinner. Eventually, this liquid layer becomes thin enough to prevent the liquid immediately adjacent to the wall from superheating to the point of allowing bubble nucleation. At this point the mechanism of heat transfer switches from boiling to liquid-surface evaporation. Since bubble nucleation is completely suppressed and the heat is carried away by the vapor at the liquid/vapor interface, this region is called the two-phase, forced-convection region. This region has the highest heat-transfer coefficients.

The final region occurs when the fluid quality and velocity increase to a point where all of the liquid becomes entrained in the vapor. At this point the walls dry up and the heat-transfer coefficient decreases dramatically. This "burn out" point must be avoided at all costs in evaporator design. Heat transfer in the liquid-deficient region will not be considered in this study.

In the following text, two different boundary conditions will be used to further illustrate the boiling process in each region. The first boundary condition will be a constant heat flux at the evaporator wall. The second boundary condition will be a constant evaporator wall temperature. In reality a combination of both will be seen; but, for most design problems, one of the two boundary

conditions can be used with reasonable accuracy.

### SUBCOOLED BOILING

Subcooled boiling occurs when the liquid adjacent to the wall is superheated to a point where bubble growth can occur. The heat-transfer coefficient associated with this region is much larger than in the one-phase region. This heat-transfer coefficient increases until saturated-nucleate boiling is achieved.

The heat-transfer coefficient begins to increase even before bubble nucleation can be detected. A photographic study by Jiji and Clark<sup>[3]</sup> demonstrated this by showing that a considerably higher surface heat flux is required to permit visual detection of bubbles than to cause an improvement in the heat-transfer coefficient. This increase is caused by hot jets of fluid from the boundary layer being projected into the cooler core of the flow.

For the case of a constant heat flux, the onset of subcooled boiling occurs in the following manner. In the one-phase, forced-convection region the thermal-boundary layer next to the surface heats up as more heat is absorbed. Eventually a point is reached where bubble nucleation can occur. A bubble will grow when the pressure inside the bubble is larger than the pressure outside. The equilibrium condition for the spherical, vapor bubble of radius  $r$  is:

$$P_v - P_l = \frac{2\sigma}{r} \quad (\text{II-1})$$

where  $P_v$  is the pressure of the vapor,  $P_l$  is the pressure of the liquid, and  $\sigma$  is the surface tension of the fluid. The liquid superheat required to maintain this equilibrium can be related to the pressure difference with the use of the Clausius-Clapeyron thermodynamic-equilibrium relation:

$$P_v - P_l = \frac{\Delta T_{sat}}{T_{sat}} \frac{h_{fg}}{1/\rho_v - 1/\rho_l} \approx \frac{\Delta T_{sat} \rho_v h_{fg}}{T_{sat}} \quad (\text{II-2})$$

where  $\Delta T_{sat}$  is the superheat of the liquid at the wall ( $T_w - T_{sat}$ ).

On any surface there are minor imperfections that contain a trace of vapor or noncondensable gas. These imperfections determine the superheat required for bubble nucleation. Therefore, by combining equations (II-1) and (II-2):

$$\frac{2\sigma}{r} \approx \frac{\Delta T_{sat} \rho_v h_{fg}}{T_{sat}} \quad (\text{II-3})$$

it can be seen that the larger the size of the imperfection the lower the superheat that is required for nucleation.

Hsu<sup>[4]</sup> postulated that bubble nuclei on a heated wall will grow only if the lowest temperature on the bubble surface is greater than the superheat required for growth. Earlier work by this investigator<sup>[5]</sup> confirmed this. For a bubble of radius  $r_c$  to grow

in a uniformly-heated fluid of temperature  $T_b$ <sup>[4]</sup>:

$$T_b > \frac{P_l T_{sat}}{h_{fg} v_v} \ln \left( 1 + \frac{2\sigma}{P_l r_c} \right) + T_{sat} \quad (II-4)$$

For a heated surface the temperature gradient can be approximated in a linear form:<sup>[6]</sup>

$$T_l(y) = T_w - \left( \frac{qy}{k_l} \right) \quad (II-5)$$

where  $T_l(y)$  is the local, liquid temperature and  $y$  is the distance away from the wall. For a nucleus of radius  $r_c$  to begin to grow equations (II-4) and (II-5) must be tangent<sup>[6]</sup>. Therefore:

$$T_l(y) = T_b \quad \text{and} \quad \frac{dT_l(y)}{dy} = \frac{dT_b}{dr} \quad (II-6)$$

Bergles and Rohsenow<sup>[6]</sup> obtained a graphical solution of equations (II-4) and (II-5) subject to equations (II-6) for water over the pressure range of 15-2000 psia:

$$q_{ONB} = 15.6 P^{1.156} (\Delta T_{sat}) (2.3/P^{0.0234}) \quad (II-7)$$

Davis and Anderson<sup>[7]</sup> developed an analytical solution of equations (II-4), (II-5), and (II-6):

$$q_{ONB} = \frac{k_l h_{fg}}{8\sigma T_{sat} v_{fg}} (\Delta T_{sat})^2 \quad (II-8)$$

which works well for water.

Frost and Dzakovic<sup>[8]</sup> proceeded in the same manner as Davis and Anderson; but, they assumed that nucleation occurred when  $T_l(y)$  equaled  $T_b$  at a distance of  $r(\text{Pr})^2$  rather than  $r_c$ . Thus:

$$q_{\text{ONB}} = \frac{k_l h_{fg}}{8\sigma T_{\text{sat}} v_{fg}} \left( \frac{\Delta T_{\text{sat}}}{\text{Pr}} \right)^2 \quad (\text{II-9})$$

This equation compares favorably with experimental data for a variety of fluids.

As the temperature increases further, bubbles form on the surface, but condense before they detach. During this phase jets of warmer liquid continue to flow into the cooler core by thermocapillarity. Thermocapillarity is caused by the fact that the surface tension varies around a bubble growing in a temperature gradient. The resultant force pulls the interfacial liquid film outward (from hot to cold) which then drags adjacent warmer liquid with it. Cooler fluid then rushes in to replace the warmer fluid. Brown<sup>[9]</sup> suggested that this phenomenon can transfer a considerable fraction of the heat flux. The turbulence induced by thermocapillarity explains why the average temperature of the boundary layer is below saturation. Thermocapillarity also causes the temperature of the interface between the one-phase core and the bubble-laden boundary layer to fall with increasing heat flux because of increased turbulence.

In the final stage of subcooled boiling, when the temperature approaches saturation, the bubbles grow and detach from the

surface. However, they still condense as they migrate into the cooler core of the flow. The major forces affecting the flow involved in subcooled boiling are the force exerted by the fluid flowing through the tube and the thermocapillary force at the thermal-boundary-layer interface. Neither of these forces are effected by gravitational acceleration. (Buoyancy is less important than flow shear in this regime.) Furthermore, gravitational acceleration has no effect on bubble nucleation. For these reasons the heat-transfer coefficient measured in a 1-g environment will be the same as that measured in a 0-g environment for subcooled, forced-convective boiling.

In actual practice the equations presented in this section may predict  $q_{ONB}$  lower than what is observed, especially at lower pressures and low heat fluxes. This is due to the fact that the equations assumed a wide range of activation sites. On metal or glass tubes there may not be large enough nucleation sites for boiling to occur even though the equations predict it.

For the case where there is a constant wall temperature, subcooled boiling will begin immediately. However, the boiling process, discussed above, will be the same.

#### SATURATED-NUCLEATE BOILING

When the bulk-fluid temperature reaches saturation conditions, the vapor bubbles no longer condense. Instead they continue to grow and coalesce forming even larger bubbles. Therefore, the vapor quality begins to increase.

In this region the heat-transfer mechanism is very similar to nucleate-pool boiling. In fact, the flow rate of the fluid has

very little influence on the heat-transfer coefficient<sup>[6]</sup>. Assuming that the heat transfer is caused by two additive effects: forced-convection and nucleate boiling; the forced-convection component can be ignored. Therefore, an equation for the heat-transfer coefficient in nucleate, pool boiling can be used to calculate the heat-transfer coefficient for forced-convection, saturated, nucleate boiling. This has been done successfully by several investigators<sup>[10, 11]</sup>. However, one must be careful in this approach because subcooling greatly affects pool boiling, while having some subcooling (the average temperature is the saturation temperature) in the liquid core for forced convection has no effect.

The reasons for the large increase in the heat-transfer coefficients for nucleate boiling have been debated for years with no concise agreement<sup>[6]</sup>. The main mechanism, however, seems to be the turbulence caused by bubble generation and departure. Before the bubble grows, a thermal-boundary layer is built up. When the temperature near the surface gets high enough for bubble growth, the bubble grows outward, pushing the boundary layer with it. Furthermore, when the bubble detaches from the surface it then carries this thermal-boundary layer into the bulk of the fluid, thus, directly transporting energy outward. Cooler fluid then rushes back to the surface to complete the cycle.

The major factors affecting heat transfer during saturated-nucleate boiling<sup>[6]</sup> are the heat flux and the pressure. The flow rate, quality, and core subcooling have little effect in this region. Furthermore the heat-transfer coefficient remains constant

throughout the region because of the constant bulk temperature (equal to the saturation temperature). And, the boundary conditions at the wall have no effect.

In this region the major forces acting on the fluid<sup>[5]</sup> are surface tension, inertia, bouyancy, drag, and the Bernoulli effect. Surface tension holds the bubble to the wall while the inertia of the liquid surrounding a growing bubble attempts to pull it from the wall. The drag force caused by the fluid flowing around the bubble attempts to shear it from the wall. The bouyancy force pulls the bubble upward, pulling it from the wall or pushes it onto the wall depending on the surface orientation. Upon departure from the surface, bouyancy makes the bubble rise while the Bernoulli effect created by the velocity gradient within the tube causes the bubbles to migrate to the center. As long as the flow is not stratified, the differences between the heat-transfer coefficient in 1-g and 0-g for a round tube should be small. This is because, in 1-g, gravity helps on the bottom of the tube by helping to remove the vapor but hinders on the top by keeping the vapor near the surface. The average effect should, therefore, be similar to 0-g conditions.

#### TWO-PHASE, FORCED CONVECTION

As the vapor quality increases the vapor migrates towards the center creating, at first, a slug-flow pattern which is followed by an annular-flow pattern. A slug-flow pattern is where large bubbles or "slugs" of vapor are separated from one another by thin,

liquid regions. As the vapor quality increases the liquid regions, separating the slugs, disappear leaving an annular-flow pattern where the vapor core is separated from the wall by a thin liquid layer. Flow patterns and the conditions for their appearance are discussed extensively in Reference [5].

Eventually, a point is reached where the liquid layer adjacent to the wall is thin enough that all of the heat flux is conducted through the liquid without nucleate boiling. The mechanism of heat transfer then switches from nucleate boiling to evaporation at the liquid/vapor interface. This regime, therefore, is called the two-phase, forced-convection region.

In order to suppress bubble nucleation, the effective thermal conductivity of the liquid layer must be high enough so that the liquid cannot be superheated to a point where bubble growth occurs. The equations used earlier to predict the onset of subcooled, nucleate boiling are applicable here too. As mentioned earlier, Frost and Dzakovic [8] predicted the bubble growth would begin when:

$$q_{ONB} = \frac{k_l h_{fg}}{6 \sigma T_{sat}^3 v_{fg}} \left( \frac{\Delta T_{sat}}{Pr} \right)^2 \quad (II-9)$$

Combining this equation with Newton's law of cooling:

$$q_{SNB} = h_{OTPFC} \Delta T_{sat} \quad (II-10)$$

where  $h_{OTPFC}$  is the heat-transfer coefficient for the onset of two-phase, forced convection, and  $q_{SNB}$  is the heat flux for the suppression of nucleate boiling. Setting  $q_{ONB} = q_{SNB}$  an expression

for  $h_{OTPFC}$  for the onset of two-phase, forced convection can be found.

$$h_{OTPFC} = \frac{k_l h_{fg} \Delta T_{sat}}{8 \sigma T_{sat} v_{fg} Pr^2} \quad (II-11)$$

This equation for  $h_{OTPFC}$  is the point where bubble nucleation is suppressed. As in subcooled boiling, this number may be lower than what is actually found in practice due to the absence of large nucleation sites.

In this two-phase, forced-convection region the heat is transferred across the liquid layer and evaporates liquid at the liquid/vapor interface. Therefore, the heat-transfer coefficient is very dependent on the thickness of the liquid layer. This thickness is governed by the quality. Therefore,  $h_{TPFC}$  is very dependent on quality. Furthermore, as the thickness of the liquid layer decreases, the temperature drop across the liquid layer decreases. Because of this, extremely high heat-transfer coefficients can be achieved.

Another factor that affects the heat-transfer coefficient is the mass flux<sup>[6]</sup>. Higher mass fluxes increase the vapor velocity which increases the evaporation at the liquid/vapor interface. Unlike the saturated-nucleate-boiling regime, the heat flux has no effect on the heat-transfer coefficient.

Since the heat-transfer coefficient increases along the tube with increasing quality, the wall temperature of the tube will decrease along the tube for the case of a constant heat flux. For the case of a constant wall temperature, the heat flux will

increase along the tube.

In this regime, the only contribution of gravity to the heat transfer process is in the distribution of film thickness around the circumference of the tube. However, the average value will be the same as the constant value found under 0-g conditions<sup>[5]</sup>, if the fluid wets the tube. Therefore, heat transfer in this regime should be relatively insensitive to gravity. Furthermore, all analytical treatments of this regime have used a constant, fluid thickness around the perimeter (0-g condition) successfully in past analyses<sup>[5]</sup>.

#### LIQUID DEFICIENT REGION

As the quality increases in the two-phase, forced-convection region the liquid layer gets thinner, while the vapor velocity increases. Eventually a point is reached where the liquid becomes totally entrained in the vapor core, thus causing the walls to dryout. When dryout conditions are reached, the heat-transfer coefficient decreases drastically. With a constant heat flux, this causes the wall temperature to increase tremendously. Often it will increase above the melting point of the evaporator walls, thus "burning out" the evaporator. For the case of a constant wall temperature, the heat flux decreases drastically. This causes the liquid deficient region of the evaporator to be essentially useless. This condition, therefore, must be avoided.

### III FLOW-VISUALIZATION EXPERIMENT

In order to observe the general operation of and problems associated with straight-tube evaporators, a flow-visualization experiment was set up. The experimental apparatus and the results of the experiment are discussed below.

#### EXPERIMENTAL APPARATUS

The flow-visualization loop consisted of: an evaporator, a condenser/accumulator, a preheater, a pump, a flow meter, and an adjustable orifice. This set up is shown in Figure 3. The evaporator consisted of a long, small-diameter, glass tube and will be discussed in greater detail later. The condenser/accumulator consisted of a copper coil (condenser) submerged in a bucket of water (accumulator). The steam and water mixture from the evaporator flowed through the copper coil and emptied into the bottom of the bucket. The copper coil thus served as a condenser and as a heater for the water supply in the accumulator. The accumulator was open to the atmosphere. The preheater was a hot plate beneath the accumulator. It was used to help raise the temperature of the water to near saturation. The flow meter, a rotameter, consisted of a small, vertical, glass tube. A steel and a rubber ball rose within the tube with an increase in the flow rate. The flow meter also contained a valve that could be used to regulate the flow. The purpose of the adjustable orifice will be discussed later.

#### Instrumentation

Total instrumentation consisted of three pressure meters,

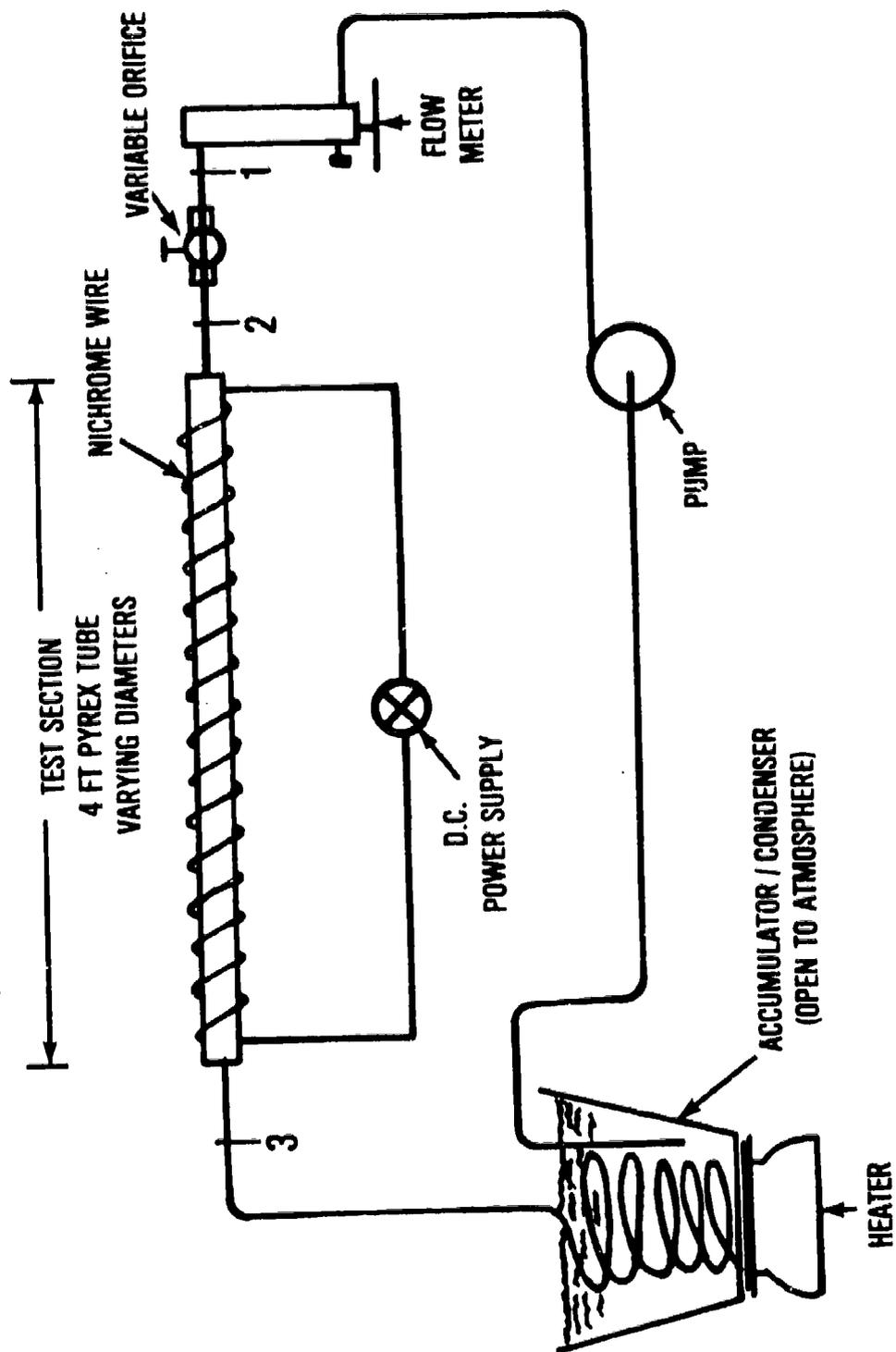


Figure 3: Experimental apparatus.

three thermocouples, an ammeter, and a voltmeter. Two of the pressure meters were set up to measure the pressure drop across the evaporator and across the adjustable orifice. The third pressure gage measured the pressure of the flow prior to entering the orifice. The three thermocouples were located before and after the evaporator and in the accumulator. The ammeter and voltmeter were used to measure the power input to the evaporator.

### Evaporator

The evaporator consisted of a long, thin, glass tube. Heat was supplied to the tube by nichrome heating wire which was coiled around the tube. The heat from the wire was conducted through the glass tube which caused the water to boil as it passed through the tube.

The purpose of experiment was to visualize the boiling process in a spacecraft evaporator. To do this it was necessary to minimize the influence of gravity. On earth, gravity tends to dominate two-phase flow through tubes, while in space, surface tension tends to dominate. However, in very small-diameter tubes, surface tension dominates over gravity. For this experiment, this point was assumed to be the point where the liquid is able to wick itself completely around the tube so that a stratified-flow pattern cannot exist. From earlier work<sup>[5]</sup> this point is:

$$r = \sqrt{\frac{2\sigma}{\rho_l g}} \quad (\text{III-1})$$

Using water properties:

$$r = \left[ \frac{2(5.89 \times 10^{-2} \text{ NT/m})}{(955 \text{ kg/m}^3)(9.8 \text{ m/s}^2)} \right]^{1/2}$$

$$r = 0.0035 \text{ m} = 0.14 \text{ ft} = 1/8 \text{ in}$$

Therefore, tubes with  $\frac{1}{4}$  inch diameters or less were used for the evaporator.

#### RESULTS OF FLOW VISUALIZATION

The flow entered the evaporator subcooled. Almost immediately very small bubbles would appear at the evaporator wall (subcooled boiling). Although these bubbles remained small, those generated further along the tube did grow as the bulk temperature increased. They also departed from the surface and recondensed in the flow. As the bulk temperature of the flow approached saturation, the bubbles began to grow larger and coalesce. When the bulk temperature reached saturation the flow pattern changed from bubbly flow to slug flow. In this slug-flow regime the liquid layer between the vapor core and the wall was thin enough to suppress nucleate boiling. Still further down the tube, the slugs expanded into an annular-flow pattern. This pattern established in the evaporator was always the same regardless of power input or flow rate. The only difference being whether enough power was being supplied to reach the flow quality necessary for annular flow.

Although the flow pattern established in the evaporator was consistent, it was not stable. This instability was caused by the extreme pressure oscillations associated with slug flow. In a spacecraft evaporator these pressure oscillations cannot be

allowed. They would cause vibration, disruption of flow uniformity in parallel-flow networks, wear on pumps, and instabilities in the condenser. Therefore, before straight-tube evaporators can be used, a means of stabilizing the flow must be devised.

In an effort to solve this problem, an orifice was installed immediately in front of the evaporator tube. If the orifice could be sized so that the pressure drop through the evaporator is much smaller than the pressure drop through the orifice, the flow should be stable in front of the evaporator. This would eliminate the problems of parallel-flow instability and pump wear. To test this theory, an adjustable orifice was placed ahead of the evaporator. The pressure was measured at point 1 (see Figure 3), and the pressure drop was measured between points 1 and 2 and between 2 and 3. The results of this test were very conclusive - this method will not work! The orifice in this experiment was varied until the pressure drop across the orifice was approximately an order of magnitude greater than the pressure drop through the evaporator. And still, the flow oscillations could not be damped out prior to the orifice. The pressure readings are shown in Table I for various mass flow rates. An investigation of the accumulator showed the flow to be steady to the pump, therefore the only cause of these instabilities could be the slug flow. Any further increase in the pressure drop across the orifice was deemed unacceptable because the corresponding increase in pump power required would eliminate the major advantage of the two-phase systems having a low pump-power requirement.

TABLE I. Pressure measurements of evaporator test section for  $Q = 1.620$  watts

| MASS VELOCITY<br>(kg/s) | TEST SECTION<br>$P_2 - P_3^*$ (in $H_2O$ ) | PRESSURE ORIFICE<br>$P_1 - P_2^*$ (in $H_2O$ ) | INLET PRESS<br>$P_1^*$ (psia) |
|-------------------------|--|--|-------------------------------|
| 6.53                    | 3 - 6                                      | 0 - 15   | 1 - 1.5                       |
| 6.57                    | 3 - 8                                      | 20 - 65  | 2.6 - 3.2                     |
| 6.35                    | 3.5 - 10                                   | 100+   | 6.6 - 6.75                    |
| 4.00                    | 2 - 10                                     | 71 - 79  | 3.65 - 3.7                    |
| 3.95                    | 2 - 8                                      | 100+   | 5.7 - 5.72                    |

\* See Figure 3 for location of pressure measurements

#### IV TWO-PHASE, HEAT-TRANSFER EQUATIONS

In order to determine a method of calculating the heat-transfer coefficients for forced-convective boiling, a review of past work is presented here. The information obtained in this section is then used to develop a better method in Section V.

Many empirical equations have been developed to predict the heat-transfer coefficients in forced-convection-boiling heat transfer. Most of the equations were developed for only one region of boiling, while others use weight factors with several equations to make a general equation. The weight given to a certain equation depends on the local flow conditions. This section describes some of the newer and/or more popular equations.

One of the easiest equations to use is that of McAdams et al. [12] They developed an empirical equation for the heat flux for water during subcooled, nucleate boiling:

$$q = C' \Delta T_{sat}^{3.86} \quad (IV-1)$$

where  $C'$  is a dimensionless constant that ranges from 0.190 to 0.074 depending on the extent of degassing of the water and  $T_{sat}$  is the difference between the wall temperature and the saturation temperature. They found  $q$  to be independent of water velocity, pressure, degree of subcooling, and diameter. The equation is good for pressures of 30 psia to 90 psia. The main problem with this equation is that it was developed solely for water and is not applicable to other fluids.

As mentioned previously, pool-boiling equations have been used

successfully for calculating the heat-transfer coefficients in the subcooled and saturated-nucleate-boiling regions. Specifically, Lavin and Young<sup>[11]</sup>, and Myers and Katz<sup>[10]</sup> have successfully used McNelly's equation<sup>[13]</sup> for the nucleate, pool-boiling heat-transfer coefficient to correlate their saturated, forced-convection nucleate-boiling data. McNelly's equation is:

$$\frac{hd}{k_l} = C \left( \frac{qd}{\mu h_{fg}} \right)^{0.69} \left( \frac{C_{p_l} \mu_l}{k_l} \right)^{0.69} \left( \frac{\rho_l}{\rho_v} - 1 \right)^{0.31} \left( \frac{Pd}{\sigma} \right)^{0.31} \quad (IV-2)$$

where C was given as .225 by McNelly but modified to .25 for the inside of a plain tube by Lavin and Young. And:

$$q = h\Delta T_{sat} \quad (IV-3)$$

McNelly developed this equation by first looking very closely at the boiling process, and then doing a dimensional analysis. He also used the heat flux as the driving mechanism rather than the temperature difference. It should be noted that like the McAdams equation, the diameter has no effect (divides out) and the mass flux has no effect (developed for pool boiling). This equation is good for all fluids.

Lavin and Young also went on to develop an equation for the two-phase, forced-convection region. To correlate their data they developed the equation:

$$\frac{h_{TP}}{h_{LP}} \left( \frac{Gh_{fg}}{q} \right)^{0.1} = 3.79 \left( \frac{1+x}{1-x} \right)^{1.16} \quad (IV-4)$$

where  $h_{TP}$  is the two-phase, forced-convection, heat-transfer coefficient;  $h_{LP}$  is the one-phase, heat-transfer coefficient for the liquid phase;  $G$  is the mass flux; and  $x$  is the quality. For  $h_{LP}$  they used the Sieder-Tate equation:

$$h_{LP} = 0.023 \frac{k}{d} Re_{\ell}^{0.8} Pr^{0.33} \left( \frac{\mu}{\mu_w} \right)^{0.14} \quad (IV-5)$$

where the Reynolds number is defined as:

$$Re_{\ell} = \frac{Gd(1-x)}{\mu} \quad (IV-6)$$

Since this equation was developed for the two-phase, forced-convection region, it is very dependent on the quality and relatively independent of the heat flux.

Lockhart and Martinelli devised a parameter,  $X$ , to correlate the pressure drop for two-phase flow in tubes. This parameter is also used to calculate the liquid-film thickness in two-phase flows. Since the heat-transfer coefficient is very dependent upon the liquid layer adjacent to the surface in two-phase, forced convection, many investigators have used  $X$  to successfully correlate their data. They used an equation of the form:

$$\frac{h_{TP}}{h_L} = f \left( \frac{1}{X_{tt}} \right)^n \quad (IV-7)$$

where  $X_{tt}$  is the Lockhart-Martinelli parameter for turbulent liquid, turbulent vapor:

$$X_{tt} = \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \left( \frac{u_l}{u_v} \right)^{0.1} \quad (\text{IV-8})$$

Dengler and Addoms<sup>[14]</sup> found that:

$$\frac{h_{TP}}{h_{LO}} = 3.5 \left( \frac{1}{X_{tt}} \right)^{0.5} \quad (\text{IV-9})$$

where  $h_{LO}$  is the heat-transfer coefficient for liquid alone found by setting  $x=0.0$  and using equation (IV-5). Guerrieri and Talty<sup>[15]</sup> found that:

$$\frac{h_{TP}}{h_{LP}} = 3.5 \left( \frac{1}{X_{tt}} \right)^{0.45} \quad (\text{IV-10})$$

where  $h_{LP}$  is the one-phase, heat-transfer coefficient. Bennett et al<sup>[16]</sup> found that:

$$\frac{h_{TP}}{h_{LO}} = 0.64 \left( \frac{1}{X_{tt}} \right)^{0.74} q^{0.11} \quad (\text{IV-11})$$

All of the above correlations were compared by Chen<sup>[17]</sup>. He found that Dengler and Addoms' equation predicted values too large when compared with other experimenters data. Guerrieri and Talty's equation predicted lower coefficients than Dengler and Addoms' but still lacked accuracy, and Bennet's predicts  $h_{TP}$  well for water but not for other fluids. For these reasons Chen developed his own method which covers both the saturated, nucleate-boiling region and the two-phase, forced-convection region. He assumed that the two mechanisms, nucleate boiling and forced convection, are present to some extent in both regions. Therefore, he devised a correlation using superposition:

$$h_{TP} = h_{NB} + h_{FC} \quad (IV-12)$$

Where  $h_{NB}$  is the contribution due to nucleate boiling and  $h_{FC}$  is the contribution due to forced convection. A Dittus-Boelter type equation was used for  $h_{FC}$ :

$$h_{FC} = 0.023 Re_{TP}^{0.8} Pr_{TP}^{0.4} \frac{k_{TP}}{d} \quad (IV-13)$$

Chen argued that  $k_{TP}$  and  $Pr_{TP}$  can be equated with the liquid values. Furthermore, he defined a parameter  $F$  as:

$$F = \left( \frac{Re_{TP}}{Re_l} \right)^{0.8} = \left( \frac{Re_{TP}}{G(1-x)d/\mu_l} \right)^{0.8} \quad (IV-14)$$

The only unknown in  $h_{FC}$  then is  $F$ . However, since  $F$  is a flow parameter it can be related to  $X_{tt}$ . Chen used Foster and Zuber's analysis of nucleate boiling for  $h_{NB}$ :

$$h_{NB} = 0.0022 \frac{k_l^{0.79} C_{p_l}^{0.45} \rho_l^{0.45} g_c^{0.25}}{\sigma^{0.5} \mu_l^{0.29} h_{fg}^{0.24} \rho_v^{0.24}} \Delta T^{0.24} \Delta P^{0.75} \quad (IV-15)$$

He also defined a nucleate-boiling, suppression factor,  $S$ , which relates the mean superheat,  $\Delta T$ , to the wall superheat,  $\Delta T_{sat}$ :

$$S = \left( \frac{\Delta T}{\Delta T_{sat}} \right)^{0.99} \quad (IV-16)$$

using the Clausius-Clapeyron equation:

$$\Delta P = \frac{\Delta T h_{fg}}{T_{sat} v_{fg}} \quad (IV-17)$$

Combining equations: (IV-15), (IV-16) and (IV-17);  $h_{NB}$  equals:

$$h_{NB} = 0.00122 \frac{k_l^{0.79} C_{p_l}^{0.45} \rho_l^{0.49} g_c^{0.25} h_{fg}^{0.49}}{\sigma^{0.5} \mu_l^{0.29} \rho_v^{0.24} T_{sat}^{0.75} v_{fg}^{0.75}} \Delta T S \quad (IV-18)$$

and

$$h = h_{NB} + Fh_{FC} \quad (IV-19)$$

The Parameters S and F are graphed in Figures 4 and 5. The Chen equation has been correlated with the data of many investigators with 85% of the data being within - 15.1%. The problem with the Chen equation is that it is not easy to program on a computer due to the use of charts to find S and F.

A final method for calculating  $h_{TP}$  was proposed recently by Bjorge, Hall, and Rohsenow. [18] They also used a superposition formula where:

$$q_{TP} = q_{FC} + q_{NB} - q_{IB} \quad (IV-20)$$

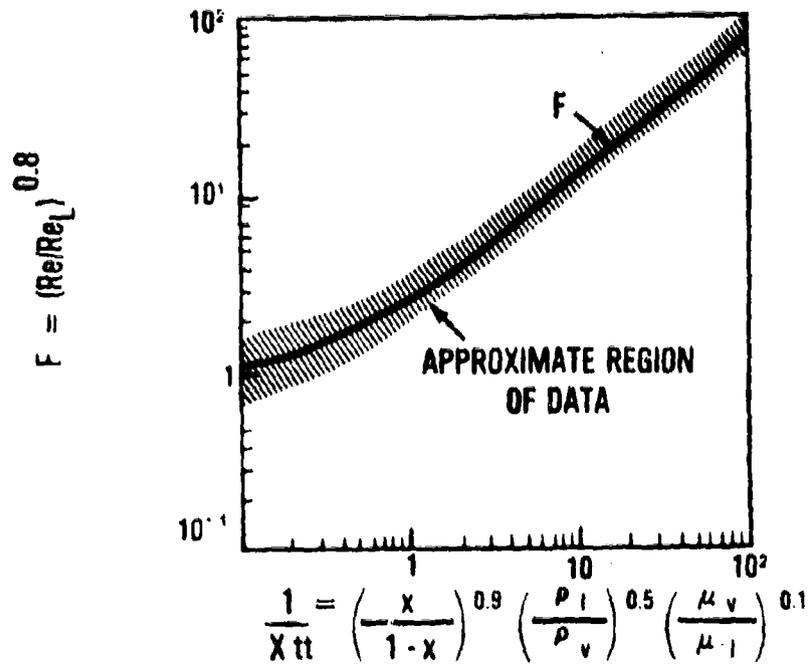
for the high quality region ( $x > 0.05$ ), and

$$q_{TP} = \left[ q_{FC}^2 + (q_{NB} - q_{IB})^2 \right]^{1/2} \quad (IV-21)$$

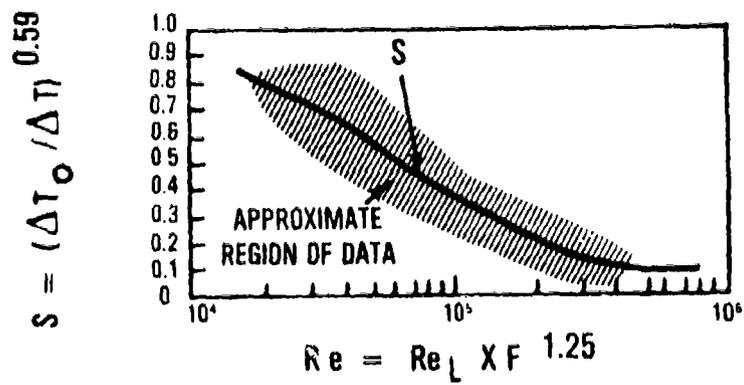
for the low quality region.  $q_{IB}$  is the heat flux at the incipient boiling point. Since  $q_{NB}$  and  $q_{IB}$  are proportional to  $T^3$ :

$$q_{TP} = q_{FC} + q_{NB} \left[ 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^3 \right] \quad (IV-22)$$

for the high quality region, and for the low quality region:



**Figure 4:** Reynolds number factor  $F$  versus the Lockhart-Martinelli parameter  $X_{tt}$ .



**Figure 5:** Suppression factor  $S$  versus the Reynolds number.

$$q_{TP} = \left\{ q_{FC}^2 + q_{NB}^2 \left[ 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^{3.2} \right]^{\frac{1}{2}} \right\} \quad (IV-23)$$

For the high quality region, the equation developed by Travis et al<sup>[19]</sup>, and modified by Hall<sup>[20]</sup> was used for  $q_{FC}$ :

$$q_{FC} = \frac{Re_l^{0.9} Pr_l F(X_{tt}) k_l}{F_2 d} \Delta T_{sat} \quad (IV-24)$$

where:

$$F(X_{tt}) = 0.15 \left[ \frac{1}{X_{tt}} + 2.0 \left( \frac{1}{X_{tt}} \right)^{0.32} \right] \quad (IV-25)$$

$$Re_l = \frac{Gd(1-x)}{\mu_l} \quad (IV-6)$$

and

$$F_2 = 5Pr_l + 5 \ln(1 + 5Pr_l) + 2.5 \ln(0.0031 Re_l^{0.812}) \quad (Re > 1125) \quad (IV-26a)$$

$$F_2 = 5Pr_l + 5 \ln \left[ 1 + Pr_l (0.0964 Re_l^{0.585} - 1) \right] \quad (50 < Re < 1125) \quad (IV-26b)$$

$$F_2 = 0.0707 Pr_l Re_l^{0.5} \quad (Re > 50) \quad (IV-26c)$$

The equation was developed by Traviss et al. as a correlation for condensation in annular flow. Since two-phase, forced-convection evaporation is the opposite of forced-convection condensation, this equation should work for two-phase evaporation. Hall found this to be true, with a slight modification of the coefficient.

For the subcooled and low-quality region, Colburn's equation was used:

$$h_{FC} = 0.023 \frac{k_b (Gd)^{0.8}}{d (u_f)} \left( \frac{u_f C_{p_f}}{k_f} \right)^{0.333} \quad (IV-27)$$

where the subscript f means that the property should be evaluated at the film temperature,  $(T_w + T_b)/2$ , the subscript b refers to the bulk temperature, and  $q_{FC}$  can be found with equation (IV-3)

The nucleate boiling heat flux for both regions is calculated by using the Mikic-Rohsenow correlation:

$$\frac{q_{NB}}{h_{fg}} \left( \frac{g_c \sigma}{g(\rho_l - \rho_v)} \right)^{1/2} = B_m \frac{k_l^{1/2} \rho_l^{17/8} C_{p_l}^{19/8} \rho_v^{1/8}}{h_{fg}^{7/8} (\rho_l - \rho_v)^{9/8} \sigma^{5/8} T_{sat}^{1/8}} \Delta T_{sat}^3 \quad (IV-28)$$

where  $B_m$  is  $1.89 \times 10^{-14}$  in SI units for forced-convection boiling of water.

For the wall superheat at the incipience of boiling, they used the equation derived by Davis and Anderson for the high quality region:

$$\Delta T_{sat, IB} = \frac{8 \sigma T_{sat} \rho_v h_{fg}}{k_l h_{fg}} \quad (IV-29)$$

In the subcooled and low quality region, this equation is modified to account for the fact that there may not be large nucleation sites:

$$\Delta T_{\text{sat,IB}} = \frac{1}{1-N} \left( \frac{1}{4\Gamma N} - N\Delta T_{\text{sc}} \right) \quad (r_{\text{tang}} > r_{\text{max}}) \quad (\text{IV-30})$$

where  $r_{\text{tang}}$  is the radius of tangency between equation II-4 and II-5 in the subcooled-boiling section or:

$$r_{\text{tang}} = \frac{4\sigma T_{\text{sat}} v_{fG}}{h_{fG} \Delta T_{\text{sat,IB}}} \quad (\text{IV-31})$$

Furthermore:

$$\Gamma = \frac{k_l h_{fG}}{8\sigma T_{\text{sat}} v_{fG} h_{FC}} \quad (\text{IV-32})$$

and

$$N = \frac{h_{FC} r_{\text{max}}}{k_l} \quad (\text{IV-33})$$

Where  $r_{\text{max}} = 10^{-6}$  m.

It should be noted that, in the method devised by Bjorge, Hall and Rohsenow,  $h_{FC}$  for the high-quality region is completely different from  $h_{FC}$  in the low-quality and subcooled regions. In

the low-quality and subcooled region  $h_{FC}$  represents the one-phase, forced-convection, heat-transfer coefficient, equation (IV-27); while in the high-quality region,  $h_{FC}$  represents the heat transfer coefficient for forced convection with a phase change at the boundary, equation (IV-24).

## V PROPOSED METHOD FOR DESIGN OF STRAIGHT-TUBE EVAPORATORS

In this section the information that was presented in the preceding three sections is combined to develop a new approach to designing straight-tube evaporators, and a new method for sizing the length of the evaporator tube is developed. Next, a new way of calculating the local heat-transfer coefficients and how to stabilize the flow is presented. This is followed by a way to calculate the pressure drop. And finally all of this information was used to develop a computer design program which is also explained in this section.

### DESIGN METHOD

Presently, the length of the tube necessary for a certain size of evaporator is calculated by treating the one-phase and two-phase sections separately. First the length of the one-phase section is calculated by using the one-phase heat-transfer coefficient, the energy it takes to reach saturation conditions, and the desired wall temperature. This, of course, gives a certain heat flux in the one-phase section.

After the calculations are finished for the one-phase section, the length of the two-phase section is calculated in the same way, using appropriate two-phase equations. For this calculation it is assumed that boiling begins when the bulk temperature reaches saturation. The total length can then be calculated by adding the lengths and multiplying by a safety factor.

The problem with this approach is that by breaking up the tube into two separate parts, the heat flux is considerably different in

the two sections. In fact, since  $Q/A$  is proportional to  $h$  and  $h_{OP}$  is less than  $h_{TP}$ ,  $Q/A_{OP}$  will be much less than  $Q/A_{TP}$ . This, however, will not be the case in an actual evaporator. Instead, there will nearly be a constant heat flux, such as with electronics cooling, or a constant wall temperature, such as with exchanging heat with a heat pipe. Furthermore, with a constant wall temperature, subcooled boiling will occur rather than one-phase heat transfer. With a constant heat flux, the wall temperature will vary with the heat-transfer coefficient,  $h$ , while with a constant wall temperature, the local heat flux will vary with  $h$ .

Because of the inaccuracies of the past approach, the following two methods of "sizing" an evaporator have been developed. These methods are depicted in Figures 6 and 7 for a constant heat flux, and for a constant wall temperature, respectively. For a constant heat flux, first calculate the incipient-boiling point. This is the point where the wall temperature is high enough above the saturation temperature that boiling can begin. Next, calculate  $h_{OP}$  and  $h_{TP}$ ,  $h_{TP}$  will vary along the length of the two-phase section. Using the allowable  $\Delta T_{sat} = T_w - T_{sat}$ , calculate the length using  $Q_{total}$ , average  $h_{TP}$ ,  $d$  and  $T_{sat}$ . This is the length of tube necessary for the evaporator. The length of the subcooled section is the length necessary to get  $\Delta T_{total} - \Delta T_{SC} = \Delta T_{IB}$ , where  $\Delta T_{total} = T_w - T_b$ ,  $\Delta T_{sc} = T_{sat} - T_b$ , and  $\Delta T_{IB} = T_{IB} - T_{sat}$ . From the incipient boiling point to the saturation point, subcooled boiling will take place and  $h_{TP}$  must be used. Prior to that point  $h_{OP}$  can be used to determine the wall temperature.

For a constant wall temperature, the heat flux will vary along the tube. Since the wall temperature will be higher than the incipient-boiling point, boiling will start immediately. Therefore, using appropriate two-phase equations,  $h_{TP}$  and the heat flux can be calculated. And finally, the length can be found.

CALCULATION OF THE HEAT-TRANSFER COEFFICIENT:

To calculate the heat-transfer coefficient, the superposition method of Rohsenow<sup>[18]</sup> was employed. This method was used because of its success in the past and its adaptability to a computer solution. This method assumes that the forced-convection boiling heat flux can be found by summing the heat fluxes for pool boiling and forced-convection. More specifically, for subcooled boiling flow ( $x=0$ ):

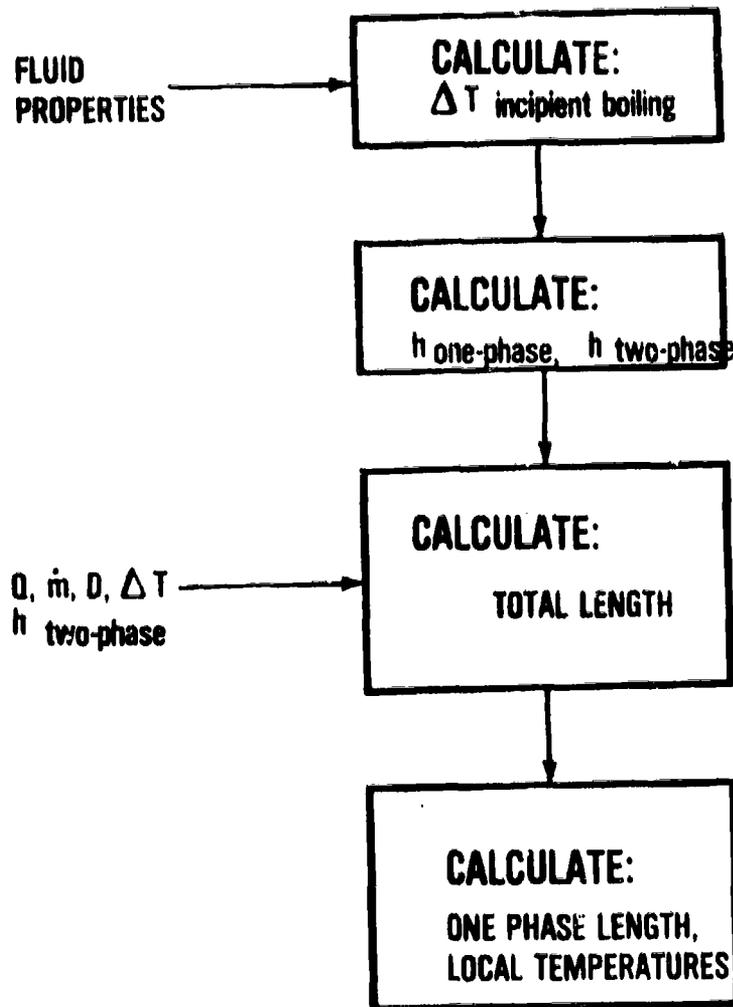
$$q_{TP} = \left\{ q_{FC}^2 + q_{NB}^2 \left[ 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^3 \right]^2 \right\}^{1/2} \quad (IV-23)$$

and for saturated-boiling flow ( $x > 0$ ):

$$q_{TP} = q_{FC} + q_{NB} \left[ 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^3 \right] \quad (IV-22)$$

and for high-quality flows where nucleate boiling is suppressed

$$q_{TP} = q_{FC} \quad (V-1)$$



**Figure 6:** Flow chart for calculating the length of an evaporator with a constant heat flux.

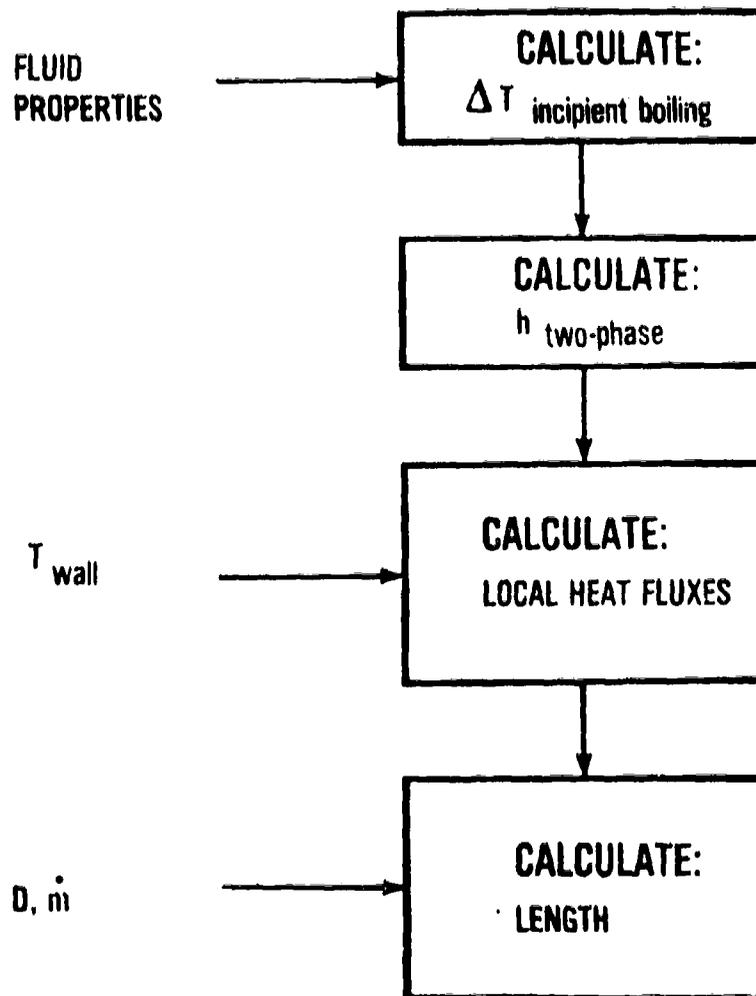


Figure 7: Flow chart for calculating the length of an evaporator with a constant wall temperature.

where  $q_{FC}$  for the subcooled regions is the one-phase, forced-convection heat flux, equations (IV-27) and (V-2), and  $q_{FC}$  is the forced-convection, evaporative heat flux, equation (IV-24), for the saturated regions. The force-convective heat flux for the subcooled region can be calculated by Colburn's equation:

$$h_{FC} = 0.023 \frac{k_f (Gd)^{0.8}}{d (u_f)} \left( \frac{u_f C_{p_f}}{k_f} \right)^{0.333} \quad (IV-27)$$

with:

$$q_{FC} = h_{FC} (\Delta T_{sat} + \Delta T_{sc}) \quad (V-2)$$

$q_{FC}$  for the high-quality region can be found by Traviss' equation:

$$q_{FC} = \frac{Re_\ell^{0.9} Pr_\ell F(x_{tt}) k_f \Delta T_{sat}}{F_2 d} \quad (IV-24)$$

$$F(x_{tt}) = 0.15 \left[ \frac{1}{x_{tt}} + 2.0 \left( \frac{1}{x_{tt}} \right)^{0.32} \right] \quad (IV-25)$$

$$Re_\ell = \frac{Gd(1-x)}{u_\ell} \quad (IV-6)$$

$$F_2 = 5Pr_\ell + 5 \ln(1 + 5Pr_\ell) + 2.5 \ln(0.0031 Re_\ell^{0.812}) \quad (Re > 1125) \quad (IV-26a)$$

$$F_2 = 5Pr_\ell + 5 \ln \left[ 1 + Pr_\ell (0.0964 Re_\ell^{0.585} - 1) \right] \quad (50 < Re < 1125) \quad (IV-26b)$$

$$F_2 = 0.0707 Pr_L Re_L^{0.5} \quad (IV-27c)$$

$q_{NB}$  for both equations can be found by using McNelly's equation:

$$\frac{h_B^d}{k_L} = C \left( \frac{q_{TP}^d}{\mu h_{fg}} \right)^{0.69} \left( \frac{Cp_L \mu_L}{k_L} \right)^{0.69} \left( \frac{\rho_L}{\rho_V} - 1 \right)^{0.31} \left( \frac{Pd}{\sigma} \right)^{0.31} \quad (IV-2)$$

where  $q_{TP}$  is the total heat flux and:

$$q_{NB} = h_B \Delta T_{sat} \quad (V-3)$$

And finally,  $\Delta T_{sat,IB}$  can be found with Frost and Dzakovic's equation:

$$\Delta T_{sat,IB} = \frac{8 \sigma T_{sat}^3 \nu_{fg} h_{FC}}{k_L h_{fg}} Pr^2 \quad (V-4)$$

where  $h_{FC}$  is found from equation (IV-27).

#### COMPARISON WITH DATA

The method outlined above was compared with data from Cheng, et.al.[21]; Latsh, et.al.[22] and Wright[23]. Cheng's data was for subcooled boiling water and is shown in Figure 8. In this figure both the method outlined above and the original method by Bjorge, Hall and Rohsenow is shown. Notice that there is very little

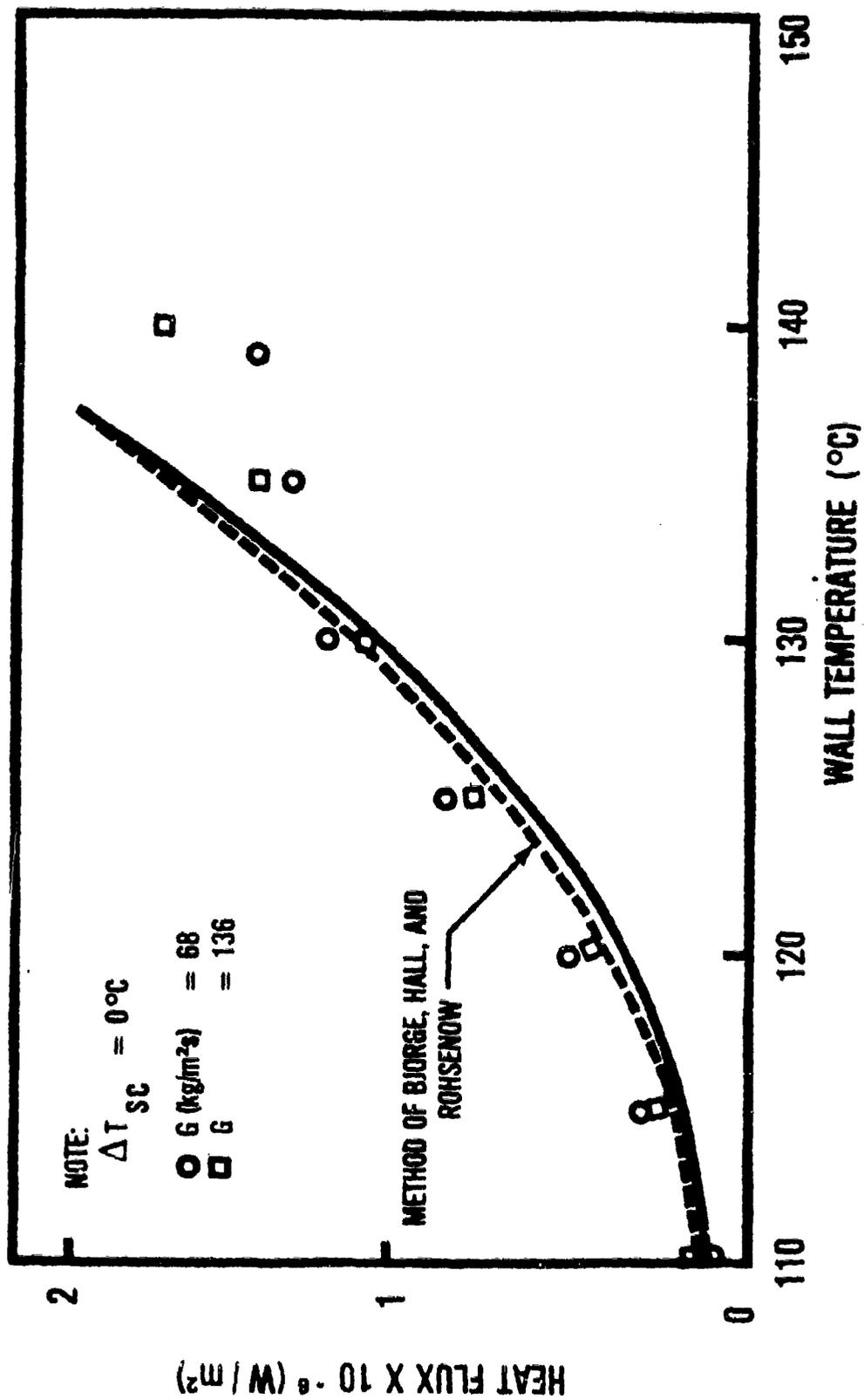


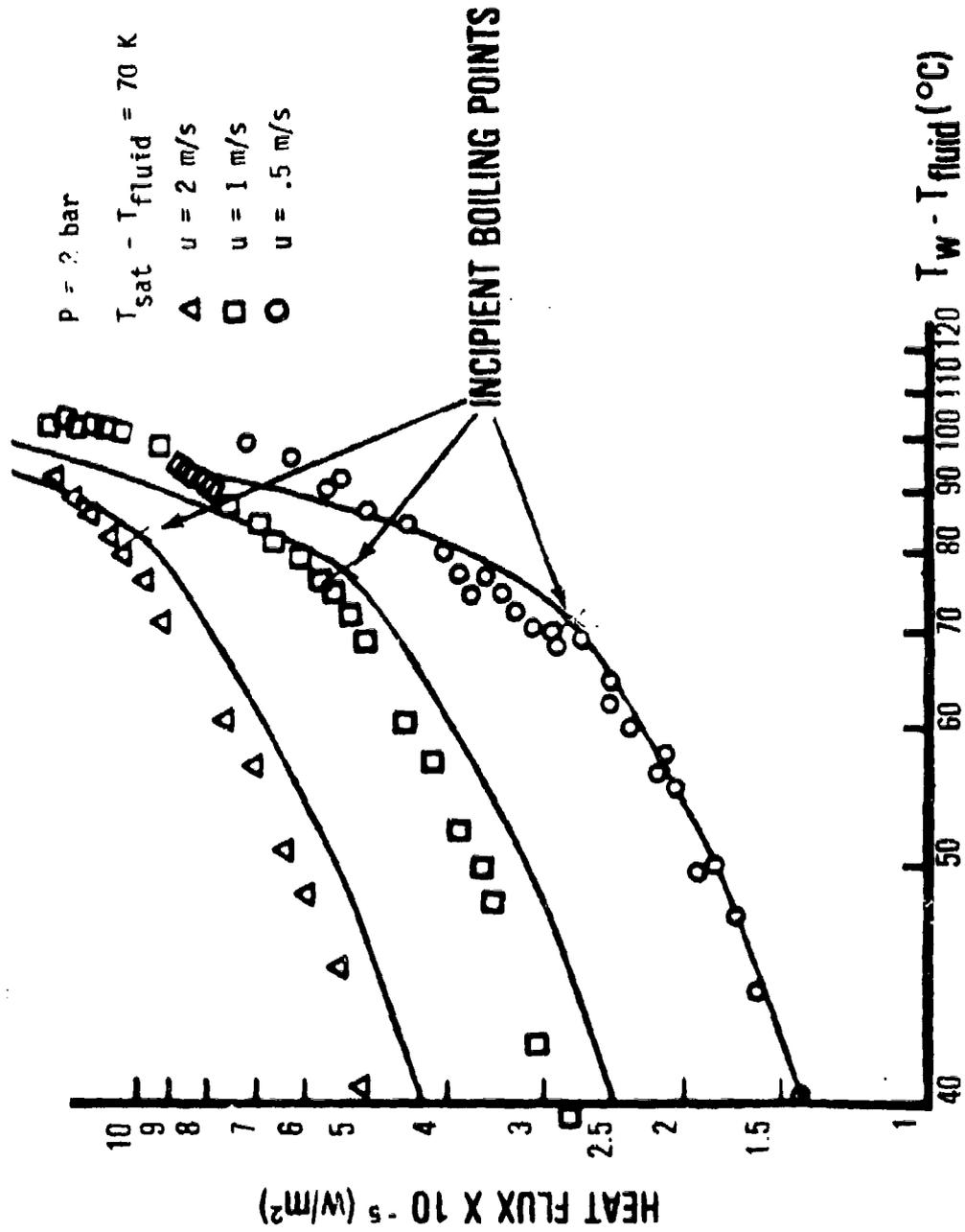
Figure 8: Wall temperature versus heat flux for subcooled boiling. Data from Reference 21.

difference between the two. This is due to the fact that both methods are the same except for the equation used to calculate  $q_{NB}$ . For water both these equations give essentially the same answers. However, the equation used by Bjorge, Hall, and Rohsenow is only applicable to water, where the new method developed in this study is applicable to any fluid.

Latsh's data for one-phase, forced-convection and subcooled boiling is shown in Figure 9. This figure shows where the calculated incipient boiling point is. Notice that this point is where the data makes a bend upward. This upward bend is also evident in the computer generated lines.

Wright's data for low-quality and high-quality flow is shown in Figures 10 through 13. The accuracy of the low-quality predictions clearly shows that convective evaporation is taking place under low-quality conditions. In fact, in Appendix A where a sample calculation is shown, forced-convective evaporation comprises all of the heat transfer at qualities lower than five percent under the flow condition.

It should be noted that this method is similar to that developed by Bjorge, et. al. They chose  $X = 0.05$  as the boundary between equation (IV-23) and (IV-22) and did not use equation (V-1). However, the flow visualization experiment done in this study and a comparison of the equations with Wright's<sup>[23]</sup> data showed that the boundary between equation (IV-23) and (IV-22) should be between zero and one percent quality. For computational ease, this value was chosen as zero percent quality. Furthermore, sample calculations (see Appendix A) showed that nucleate boiling is



**Figure 9:** Difference between the wall temperature and the bulk-fluid temperature versus the heat flux for subcooled boiling. Data from Reference 22.

suppressed at relatively low qualities. If equation (IV-22) continues to be used for suppressed, nucleate-boiling conditions the nucleate-boiling term is actually being subtracted off. This, of course, is inaccurate. Therefore equation (V-1) must be used under suppressed, nucleate-boiling conditions.

#### FLOW STABILIZATION

A method of stabilizing the flow is to prevent slug flow from occurring. This can be done by inducing an artificial, body force so that stratified flow occurs instead of slug flow. The simplest way of accomplishing this is to induce swirl in the flow so that there is a large centrifugal force. To induce swirl a twisted tape inside the tube can be used. If the wicking height of the fluid is used as a transition point between slug and stratified flow<sup>[5]</sup>:

$$\frac{r}{2} = \sqrt{\frac{2\sigma}{\rho_l g}} \quad (V-5)$$

or:

$$g = \frac{8\sigma}{\rho_l r^2} \quad (V-6)$$

The tangential acceleration due to fluid swirl is:

$$a_{\text{tang}} = \frac{u_{tw}^2}{r} \quad (V-7)$$

$q = 43525 \text{ W/m}^2$   
 $p = 21 \text{ psia}$   
 $\dot{m} = .21 \text{ kg/s}$

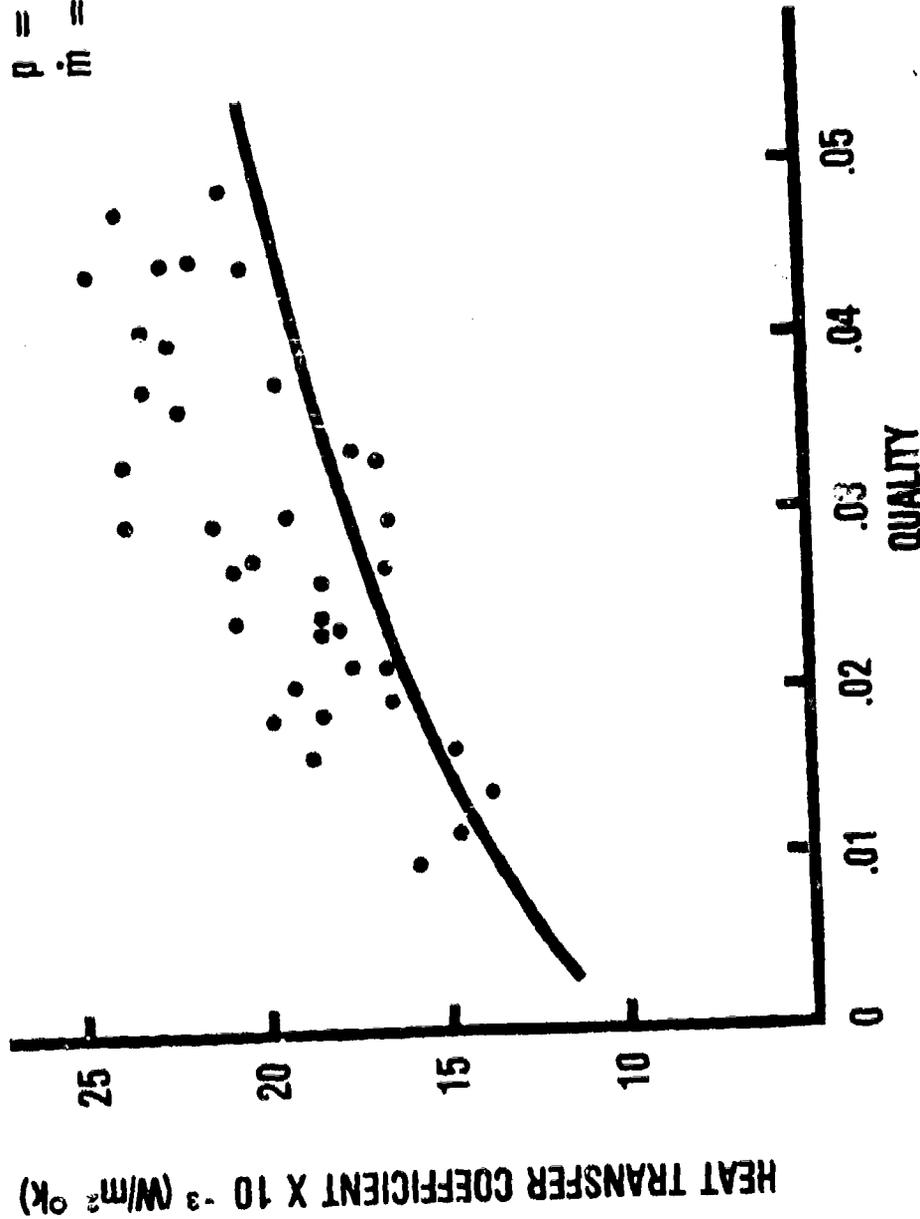


Figure 10: Heat-transfer coefficient versus quality for low heat-flux, low-quality boiling. Data from Reference 23

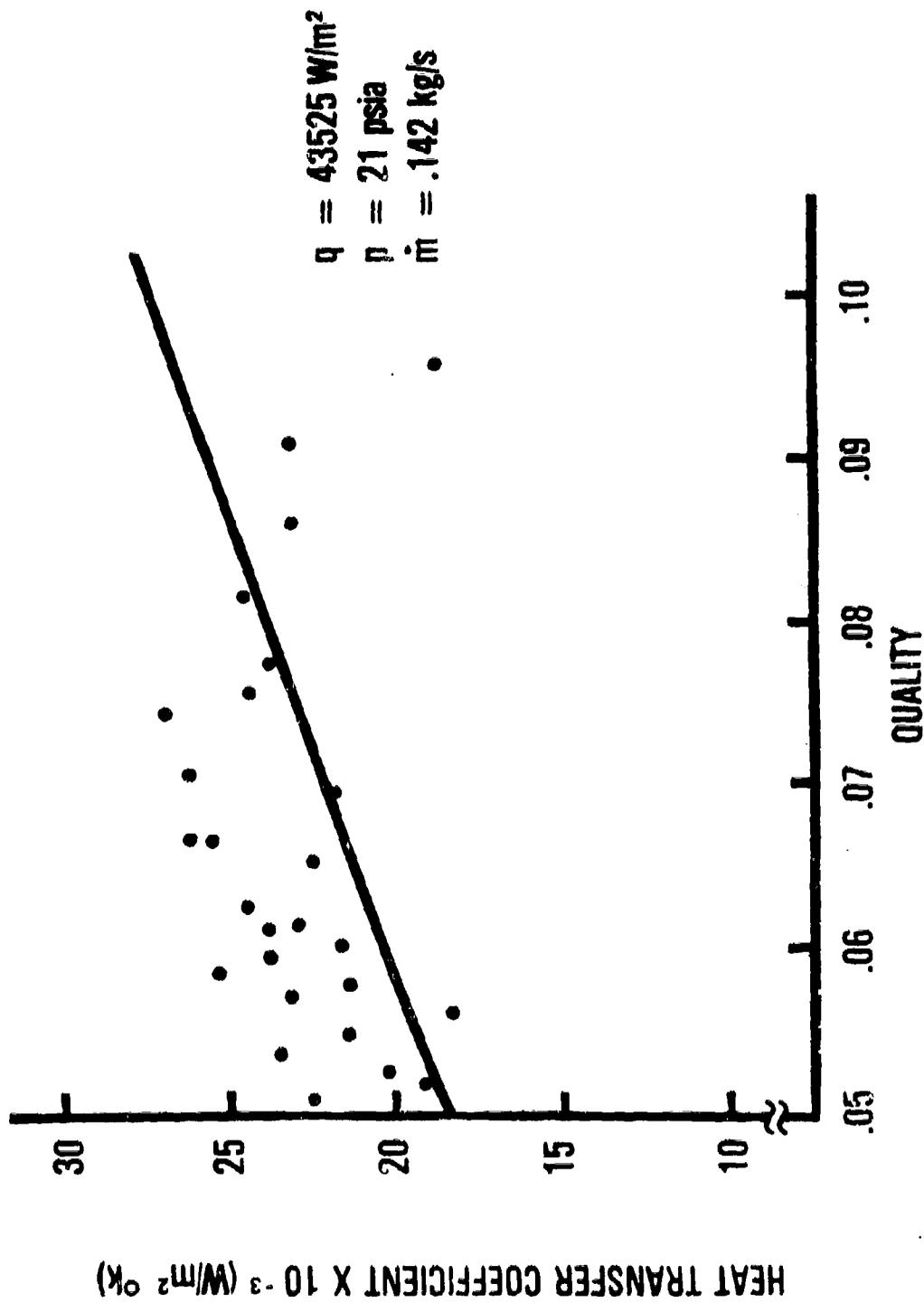
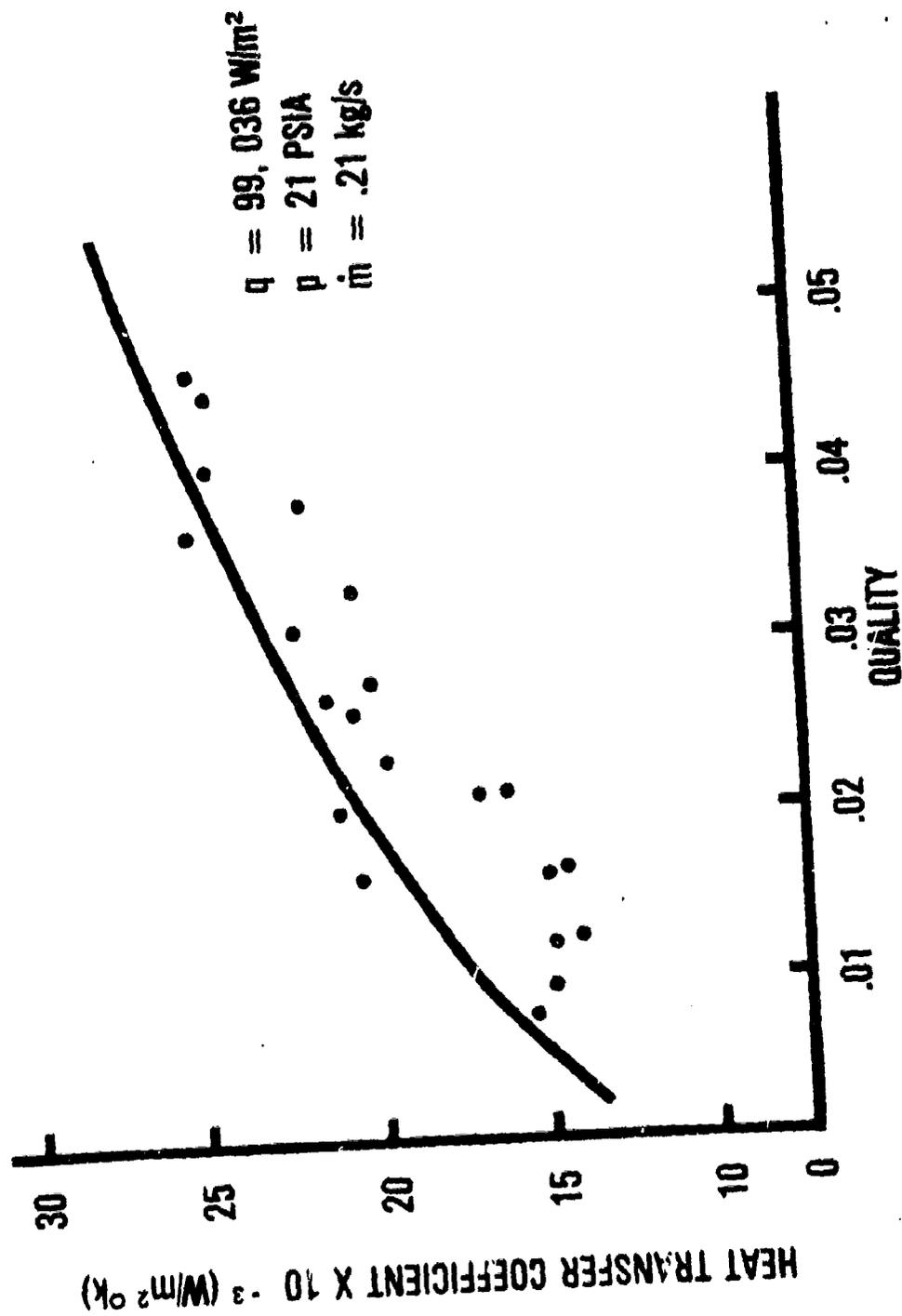


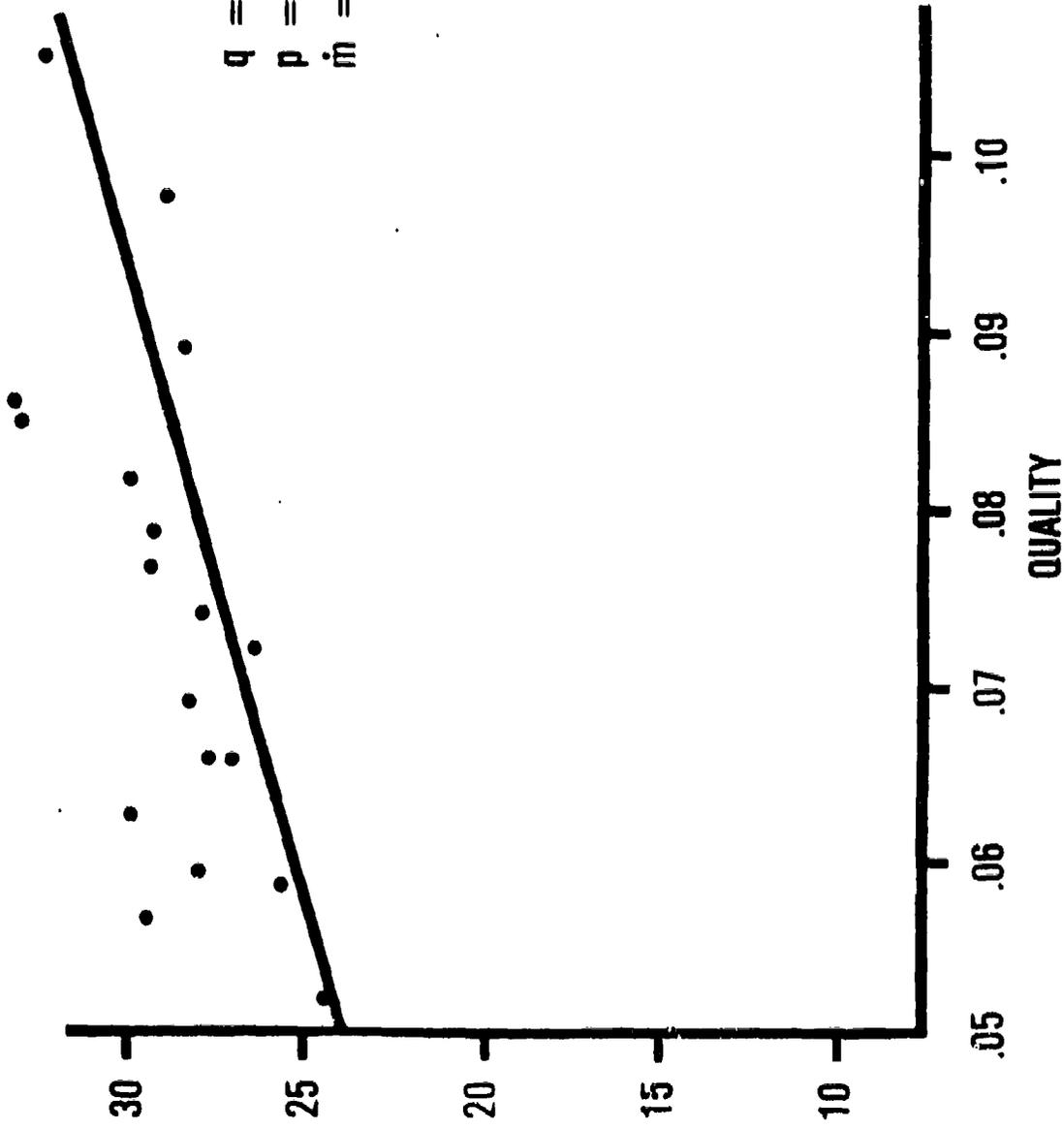
Figure 11: Heat-transfer coefficient versus quality for low heat-flux, high-quality boiling. Data from Reference 23



**Figure 12:** Heat-transfer coefficient versus quality for high

heat-flux, low-quality boiling. Data from Reference 23.

HEAT TRANSFER COEFFICIENT  $\cdot X 10^{-3} (W/m^2 \text{ } ^\circ K)$



$q = 99,036 \text{ W/m}^2$   
 $p = 21 \text{ psia}$   
 $\dot{m} = .142 \text{ kg/s}$

Figure 13: Heat-transfer coefficient versus quality for high

heat-flux, high-quality boiling. Data from Reference 23.

where  $u_{tw}$  is the tangential velocity at the tube wall.  $u_{tw}$  can be calculated from:

$$u_{tw} = \frac{2\pi}{60} r(\text{RPM}) \quad (\text{V-8})$$

where:

$$\text{RPM} = 15 \frac{u_a}{Yr} \quad (\text{V-9})$$

$u_a$  is the axial velocity and  $Y$  is the number of diameters per 180-degree twist (twist ratio). Therefore:

$$a_{\text{tang}} = \frac{2.47}{r} \left( \frac{u_a}{Y} \right)^2 \quad (\text{V-10})$$

Setting  $a_{\text{tang}}$  equal to  $g$ :

$$\frac{\rho \sigma}{\rho_l r^2} = \frac{2.47}{r} \left( \frac{u_a}{Y} \right)^2$$

or

$$Y = \frac{0.31 \rho_l r}{\sigma} u_a \quad (\text{V-11})$$

The Y value calculated for the experiment with a mass-flow rate of  $5 \times 10^{-3}$  kg/s and an evaporator diameter of 0.25 inches was 0.66. All attempts to make a twisted ribbon with this twist ratio, with the facilities available, failed. Therefore, this method could not be experimentally verified.

#### PRESSURE-DROP CALCULATION

In a report published by Bae et al. [24] the pressure gradient in a tube for two-phase flow is:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_{\text{frict}} + \left(\frac{dP}{dz}\right)_{\text{mom}} + \left(\frac{dP}{dz}\right)_{\text{grav}} \quad (\text{V-12})$$

where  $dP/dz)_{\text{frict}}$  is the frictional-pressure gradient,  $dP/dz)_{\text{mom}}$  is the pressure gradient due to a change in momentum as the liquid changes to a vapor, and  $dP/dz)_{\text{grav}}$  is the gravitational-pressure gradient. The frictional-pressure gradient can be calculated with:

$$\begin{aligned} \left(\frac{dP}{dz}\right)_{\text{frict}} = & \frac{2G^2}{\rho_v d g_c} \left\{ 0.045 \left(\frac{Gd}{\mu_v}\right)^{-0.2} \left[ x^{1.8} + 5.7 \left(\frac{\mu_l}{\mu_v}\right)^{0.0523} \right. \right. \\ & \left. \left. x (1-x)^{0.47} x^{1.33} \left(\frac{\rho_v}{\rho_l}\right) + 8.11 \left(\frac{\mu_l}{\mu_v}\right) \right. \right. \\ & \left. \left. x (1-x)^{0.94} x^{0.86} \left(\frac{\rho_v}{\rho_l}\right)^{0.522} \right] \right\} \quad (\text{V-13}) \end{aligned}$$

The momentum-pressure gradient can be found with:

$$\begin{aligned} \left(\frac{dP}{dz}\right)_{\text{mom}} = & \frac{2G^2}{\rho_v d g_c} \left\{ \frac{1}{2} \frac{dx}{dz} \left[ 2(1-x) \left(\frac{\rho_v}{\rho_l}\right)^{2/3} + \left(\frac{1}{x} - 3 + 2x\right) \right. \right. \\ & x \left(\frac{\rho_v}{\rho_l}\right)^{4/3} + (2x-1-8x) \left(\frac{\rho_v}{\rho_l}\right)^{1/3} \\ & \left. \left. + \left(28\frac{\beta}{x} - 8x\right) \left(\frac{\rho_v}{\rho_l}\right) \right] \right\} \end{aligned} \quad (V-14)$$

And the gravitational-pressure gradient can be found with:

$$\left(\frac{dP}{dz}\right)_{\text{grav}} = g(\rho_l - \rho_v) \frac{dy}{dz} \quad (V-15)$$

where  $dy$  is the differential vertical drop.

These equations were programmed into a computer routine and compared with data from the experimental apparatus. The equations predicted a good, average, pressure drop. However, the accuracy of the equations could not be measured because of the severe pressure fluctuations encountered in the test section.

## VI EVAPORATOR DESIGN PROGRAM

The methods outlined in section V were used to develop a program to design straight-tube evaporators for electronics cooling at 10°C using either ammonia or freon-11 or at 100°C using water. Other fluids can be used, but require the addition of their fluid properties to the program. The user of the program chooses a working fluid, the total input power, diameter of the tube, inlet temperature, allowable maximum wall temperature, and the exit quality or the mass-flow rate. The program performs the following tasks. It calculates the mass-flow rate needed or the exit quality; whether it is possible to design an evaporator with the given inputs; the lengths of the one-phase, subcooled, and two-phase sections and the total length; the wall-temperature differences for each section; the average heat-transfer coefficient for each section; the pressure drop for each section and the total pressure drop.

To calculate the length of an evaporator, the following methods can be used. For an evaporator subjected to a constant heat flux, the incipient-boiling point must be calculated first using equation V-4. Then  $h_{op}$  and  $h_{tp}$  are calculated using equations IV-27, IV-23, IV-24 and V-1. And finally, using the allowable  $\Delta T$ , average  $h_{tp}$ ,  $Q$ , and  $d$ , the length can be found. For an evaporator subjected to a constant wall temperature, calculate the average two-phase heat-transfer coefficient using equations IV-23, IV-24, and V-1. Since one-phase heat-transfer cannot occur, this will be the average heat-transfer coefficient for the entire tube. Next, calculate the heat flux using the given wall temperature. And finally, calculate the length. The pressure drop can be calculated using equation V-17.

Several sample runs are shown on the following pages, and sample calculations are shown in the appendix.

Example 1: Water

ENTER FLUID: WATER(1),AMMONIA(2),F-11(3)1  
ENTER POWER (W) >>1000  
ENTER DIAMETER (M) >>1.27E-2  
ENTER INLET TEMPERATURE (K) >>363.0  
ENTER DELTA T (K) >>10  
DO YOU WANT TO ENTER EXIT QUALITY (1) OR FLOW RATE (2)?>>1  
EXIT QUALITY FRACTION >>.8

THE MASS FLOW RATE IS 0.5408E-03(kg/s)

INITIAL FLOW IS LAMINAR

BOILING STARTS IMMEDIATELY

|                        | ONE-PHASE  | SUBCOOL BOILING | TWO-PHASE  | TOTAL      |
|------------------------|------------|-----------------|------------|------------|
| LENGTH (m)             | 0.0000E+00 | 0.4014E-02      | 0.1697     | 0.1737     |
| DELTA T (K)            | 0.0000E+00 | 18.74           | 10.00      |            |
| h (W/m <sup>2</sup> K) | 233.6      | 7703.           | 0.1443E+05 |            |
| DELTA P (PSI)          | 0.0000E+00 | 0.1441E-06      | 0.3694E-02 | 0.3694E-02 |

Example 2: Ammonia

ENTER FLUID: WATER(1),AMMONIA(2),F-11(3)2  
ENTER POWER (W) >>1000  
ENTER DIAMETER (M) >>1.27E-2  
ENTER INLET TEMPERATURE (K) >>273  
ENTER DELTA T (K) >>10  
DO YOU WANT TO ENTER EXIT QUALITY (1) OR FLOW RATE (2)?>>1  
EXIT QUALITY FRACTION >>.8

THE MASS FLOW RATE IS 0.9731E-03(kg/s)

INITIAL FLOW IS LAMINAR

BOILING STARTS IMMEDIATELY

|                        | ONE-PHASE  | SUBCOOL BOILING | TWO-PHASE  | TOTAL      |
|------------------------|------------|-----------------|------------|------------|
| LENGTH (m)             | 0.0000E+00 | 0.1626E-01      | 0.3433     | 0.3596     |
| DELTA T (K)            | 0.0000E+00 | 10.93           | 10.00      |            |
| h (W/m <sup>2</sup> K) | 100.3      | 6377.           | 6970.      |            |
| DELTA P (PSI)          | 0.0000E+00 | 0.1352E-05      | 0.1591E-02 | 0.1593E-02 |

Example 3: Freon-11

ENTER FLUID: WATER(1),AMMONIA(2),F-11(3)3  
ENTER POWER (W) >>1000  
ENTER DIAMETER (M) >>1.27E-2  
ENTER INLET TEMPERATURE (K) >>273  
ENTER DELTA T (K) >>15  
DO YOU WANT TO ENTER EXIT QUALITY (1) OR  
EXIT QUALITY FRACTION >>.8

FLOW RATE (2)?>>1

THE MASS FLOW RATE IS 0.6377E-02(kg/s)

INITIAL FLOW IS LAMINAR

BOILING STARTS IMMEDIATELY

|                        | ONE-PHASE  | SURCOOL BOILING | TWO-PHASE  | TOTAL      |
|------------------------|------------|-----------------|------------|------------|
| LENGTH (m)             | 0.0000E+00 | 0.1967E-01      | 0.3397     | 0.3593     |
| DELTA T (K)            | 0.0000E+00 | 17.98           | 15.00      |            |
| h (W/m <sup>2</sup> K) | 35.73      | 3880.           | 4650.      |            |
| DELTA P (PSI)          | 0.0000E+00 | 0.9424E-05      | 0.8642E-01 | 0.8643E-01 |

Example 4: Subcooled Water

ENTER FLUID: WATER(1),AMMONIA(2),F-11(3)1  
ENTER POWER (W) >>500  
ENTER DIAMETER (M) >>1.27E-2  
ENTER INLET TEMPERATURE (K) >>330  
ENTER DELTA T (K) >>15  
DO YOU WANT TO ENTER EXIT QUALITY (1) OR  
ENTER MASS FLOW RATE (kg/s) >>6.0E-3

FLOW RATE (2)?>>2

ENTIRE SECTION IS SUBCOOLED

## VII CONCLUSIONS

1. From the review of the forced-convective boiling process:
  - a. The gravitational acceleration has very little effect in the subcooled-boiling and two-phase, forced-convective regions. Therefore, the lack of a gravitational acceleration (0-g) should not affect the calculation of local heat-transfer coefficients.
  - b. The gravitational acceleration does affect the saturated nucleate-boiling region. However, since the gravitational effect tends to balance out around the circumference of the tube and since this region is generally very short, the use of equations developed for 1-g accelerations can be used to calculate the local heat-transfer coefficient without much loss of accuracy.
2. From the flow-visualization study, placing a twisted-tape insert into the evaporator should stabilize the flow by inducing a large centrifugal acceleration which prevents slug flow from occurring. The twist ratio must be large enough to ensure stratified flow under all conditions. This twist ratio can be calculated using equation V-11.
3. The length of a straight-tube evaporator can be calculated for a constant heat flux or a constant wall temperature by using the methods developed in this study and summarized in Section VI.

## REFERENCES

1. Military Space Systems Technology Model, 2nd ed., SD-TR-83-01
2. Mahefkey, T., "Military Spacecraft Thermal Management; The Evolving Requirements and Challenges", AIAA 82-0827
3. Jiji, L.M., and Clark, J.A., "Incipient Boiling in Forced Convection Channel Flow" ASME Paper 62-WA-202 (1962).
4. Hsu, S.T., and Ing. P.W., "Experiments in Forced Convection Subcooled Nucleate Boiling Heat Transfer", ASME Paper 62-HT-38, (1962).
5. Eastman, R.E., Feldmanis, C.J., Haskin, W.L., and Weaver, K.L., "Two-Phase Fluid Thermal Transport for Spacecraft", AFWAL TR 84-3028 (1984).
6. Collier, J.G., Convective Boiling and Condensation, McGraw Hill Book Co. (1972).
7. Davis, E.J., and Anderson, G.H., "The Incipience of Nucleate Boiling in Forced Convection Flow", A.I. Ch. E. Journal, 12, (4) 774-780, (July 1966).
8. Frost, W., and Dzakowic, G.S., "An Extension of the Method of Predicting Incipient Boiling on Commercially Finished Surfaces", ASME Paper 67-HT-61 (1967).

9. Brown, W.T., "A Study of Flow Surface Boiling", Ph. D. Thesis, M.I.T. Mechanical Engineering Department (Jan 1967).
10. Myers, J.E., and Katz, D.B., Chemical Engineering Program Symposium, Ser. No. 5., 49, 107, (1953).
11. Lavin, J.G., and Young, E.H., "Heat Transfer to Evaporating Refrigerants in Two-Phase Flow". A.I. Ch. E. Journal, VOL. II, No. 6. pp. 1124-1132, (Nov 1965).
12. McAdams, W.H., "Heat Transmission", McGraw-Hill Publishing Co. (1954).
13. McNelly, M.J., "A Correlation of the Rates of Heat Transfer to Nucleate Boiling Liquids," J. Imp. Coll. Chem. Eng. Soc., 7, 19 (1953).
14. Dengler, C.E., and Addoms, J.N., "Heat Transfer Mechanism for Vaporization of Water in a Vertical Tube", Chemical Engineering Program Symposium Series, No. 18, VOL. 52 (18), pp. 95-103 (1956).
15. Guerrieri, S.A., and Talty, R.D., "A Study of Heat Transfer to Organic Liquids in Single Tube, Natural Circulation Vertical Tube Boilers", Chemical Engineering Program Symposium Series: Heat Transfer, No 18. VOL. 52, pp. 69-77 (1956).

16. Bennett, J.A.R., Collier, J.G., Pratt, H.R.C., and Thorton, J.D., "Heat Transfer to Two-Phase Gas-Liquid Systems, Part I: Steam/Water Mixtures in the Liquid Dispersed Region in an Annulus", Trans. Inst. Chemical Engineers, 39, 113 (1961).
17. Chen, J.C., "A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow", ASME Paper 63-HT-34 (1963).
18. Bjorge, R.W., Hall, G.R., and Rohsenow, W.M., "Correlations of Forced Convection Boiling Heat Transfer Data", Int. J. of Heat and Mass Transfer, VOL. 25, No. 6 Pages 753-758 (June 1982).
19. Traviss, D.P., Rohsenow, W.M., and Baron, A.B., "Forced Convection Condensation Inside Tubes: A Heat Transfer Equation for Design", ASHRAE Reprint No. 2272 RP-63 (1972).
20. Hall, G.R., "Method of Correlating Forced Convection Boiling Heat Transfer Data", Ocean Engineering and SM Thesis in Mechanical Engineering, Massachusetts Institute of Technology (1977).
21. Cheng, S.C., Ng, W.W.L., and Heng, K.T., "Measurements of Boiling Curves of Subcooled Water Under Forced Convective Conditions", Int. J. Heat Mass Transfer, VOL. 21, pp. 1385 (1978).

22. Latsch, K., Morell, F., and Rampg, H., "Subcooled Forced Convection Boiling Heat Transfer at Subatmospheric Pressure", Proc. Sixth Int. Heat Transfer Conference, VOL. I, pp. 287-292, (1978).
  
23. Wright, R.M., "Down Flow Forced Convection Boiling of Water in Uniformly Heated Tubes", UCRL-9744 (1961).
  
24. Bae, S., Maulbetsch, J.S., Rohsenow, W.M., "Refrigerant Force-Convection Condensation Inside Horizontal Tubes". Report No. DSR 79760-59, Contract No. ASHRAE RP63 (1 Nov 68).

## APPENDIX A

This appendix shows how the calculations were done to obtain the solid lines on the graphs. More specifically, sample calculations are shown for Figures 8, 9, and 10. In addition, a sample twist ratio calculation is done.

SAMPLE CALCULATION: Subcooled Boiling, Figure 8 (for water)

Data:

|   |  |
|---|--|
| $k = 0.68 \text{ W/m K}$                  | $\mu = 2.8 \times 10^{-4} \text{ kg/m s}$  |
| $\rho_l = 958.0 \text{ kg/m}^3$           | $\sigma = 5.89 \times 10^{-2} \text{ N/m}$ |
| $\rho_v = 0.6 \text{ kg/m}^3$             | $d = 1.27 \times 10^{-2} \text{ m}$        |
| $h_{fg} = 2.258 \times 10^6 \text{ J/kg}$ | $C_p = 4.21 \times 10^3 \text{ J/kg K}$    |
| $P = 1.014 \times 10^5 \text{ N/m}^2$     | $\Delta T_{\text{sat}} = 30 \text{ C}$     |

Equation:

$$h_{\text{NB}} = 0.25 \left( \frac{k}{d} \right) \left( \frac{q d}{h_{fg} \mu} \right)^{0.69} \left( \frac{C_p \mu}{k} \right)^{0.69} \left( \frac{P a}{\sigma} \right)^{0.31} \\ \times \left( \frac{\rho_l}{\rho_v} - 1 \right)^{0.31}$$

Calculation:

$$h_{\text{NB}} = 0.25 \left( \frac{0.68 \text{ W/m K}}{1.27 \times 10^{-2} \text{ m}} \right) \left( \frac{q (1.27 \times 10^{-2} \text{ m})}{(2.258 \times 10^6 \text{ J/kg}) (2.8 \times 10^{-4} \text{ kg/ms})} \right)^{0.69} \\ \times \left( \frac{(4.21 \times 10^3 \text{ J/kgK}) (2.8 \times 10^{-4} \text{ kg/ms})}{0.68 \text{ W/mK}} \right)^{0.69} \\ \times \left( \frac{(1.014 \times 10^5 \text{ N/m}^2) (1.27 \times 10^{-2} \text{ m})}{5.89 \times 10^{-2} \text{ N/m}} \right)^{0.31} \\ \times \left( \frac{958.0 \text{ kg/m}^3}{0.6 \text{ kg/m}^3} - 1 \right)^{0.31}$$

$$h_{\text{NB}} = 2.45 q^{0.69} \text{ W/m}^2 \text{ K}$$

$$q = h_{\text{NB}} \Delta T_{\text{sat}}$$

$$q = 18.07(30)^{3.23}$$

$$q = 1.06 \times 10^6 \text{ W/m}^2$$

SAMPLE CALCULATION: Subcooled Boiling, Figure 9

One-Phase Forced Convection:

|       |                                    |  |
|-------|------------------------------------|--|
| Data: | $T_w - T_{FL} = 60 \text{ K}$      | $\rho_l = 983.3 \text{ kg/m}^3$            |
|       | $T_{FL} = 30 \text{ C}$            | $\mu = 4.71 \times 10^{-4} \text{ kg/m s}$ |
|       | $u = 0.5 \text{ m/s}$              | $k = 0.654 \text{ W/m K}$                  |
|       | $d = 1.2 \times 10^{-2} \text{ m}$ | $Pr = 3.01$                                |

Equation:

$$h_{FC} = 0.23 Re^{0.8} Pr^{0.4} \frac{k}{d}$$

Calculation:

$$T_{ref} = \frac{T_w + T_{FL}}{2} = \frac{90 + 30}{2} = 60 \text{ C}$$

$$Re = \frac{\rho u d}{\mu} = \frac{(983.3 \text{ kg/m}^3)(0.5 \text{ m/s})(1.2 \times 10^{-2} \text{ m})}{4.71 \times 10^{-4} \text{ kg/m s}}$$

$$Re = 1.253 \times 10^4$$

$$h_{FC} = 0.023 (1.253 \times 10^4)^{0.8} (3.01)^{0.4} \frac{0.654 \text{ W/mK}}{1.2 \times 10^{-2} \text{ m}}$$

$$h_{FC} = 3697 \text{ W/m}^2 \text{ K}$$

$$q = h_{FC} \Delta T$$

$$q = (3697 \text{ W/m}^2 \text{ K})(60 \text{ K}) = 2.22 \times 10^5 \text{ W/m}^2$$

Incipient Boiling Point:

Data:  $k_l = 0.68 \text{ W/m K}$        $\sigma = 5.89 \times 10^{-2} \text{ N/m}$   
 $\rho_l = 958.0 \text{ kg/m}^3$        $Pr = 1.66$   
 $h_{fg} = 2.258 \times 10^6 \text{ J/kg}$        $v_{fg} = 1.6719 \text{ m}^3/\text{kg}$   
 $\mu = 2.8 \times 10^{-4} \text{ kg/m s}$        $d = 1.2 \times 10^{-2} \text{ m}$

Equation:

$$\Delta T_{\text{sat, IB}} = \frac{8\sigma T_{\text{sat}} v_{fg} h_{FC}}{k_l h_{fg}} (Pr)^2$$

Calculation:

$$Re = \frac{\rho_l u d}{\mu} = \frac{(958 \text{ kg/m}^3)(0.5 \text{ m/s})(1.2 \times 10^{-2} \text{ m})}{2.8 \times 10^{-4} \text{ kg/m s}}$$

$$Re = 2.053 \times 10^4$$

$$h_{FC} = 0.023 Re^{0.8} Pr^{0.4} \frac{k}{d}$$

$$h_{FC} = 0.023 (2.053 \times 10^4)^{0.8} (1.66)^{0.4} \frac{0.68 \text{ W/mK}}{1.2 \times 10^{-2} \text{ m}}$$

$$h_{FC} = 4498 \text{ W/m}^2\text{K}$$

$$\Delta T_{\text{sat, IB}} = \frac{(8)(5.89 \times 10^{-2} \text{ N/m})(373 \text{ K})(1.6719 \text{ m}^3/\text{kg})}{(0.68 \text{ W/mK})(2.258 \times 10^6 \text{ J/kg})}$$

$$\times (4447 \text{ W/m}^2\text{K})(1.66)^2$$

$$\Delta T_{\text{sat, IB}} = 2.37 \text{ K}$$

Subcooled Boiling:

Data:  $\rho_v = 0.6 \text{ kg/m}^3$        $P = 2 \times 10^5 \text{ N/m}^2$   
 $C_p = 4.21 \times 10^3 \text{ J/kg K}$        $\Delta T_{\text{sat}} = 20 \text{ K}$

Equation:

$$h_{NB} = 0.25 \frac{k}{d} \left( \frac{qd}{h_{fg}\mu} \right)^{0.69} \left( \frac{Cp\mu}{k} \right)^{0.69} \left( \frac{Pd}{\sigma} \right)^{0.31} \\ \times \left( \frac{\rho_l}{\rho_v} - 1 \right)^{0.31}$$

$$h_{NB} = 0.25 \left( \frac{0.68 \text{ W/mK} \cdot q (1.2 \times 10^{-2} \text{ m})}{1.2 \times 10^{-2} \text{ m} (2.258 \times 10^6 \text{ J/kg}) (2.8 \times 10^{-4} \text{ kg/ms})} \right)^{0.69} \\ \times \left( \frac{(4.21 \times 10^3 \text{ J/kgK}) (2.8 \times 10^{-4} \text{ kg/ms})}{0.68 \text{ W/mK}} \right)^{0.69} \\ \times \left( \frac{(2 \times 10^5 \text{ N/m}^2) (1.2 \times 10^{-2} \text{ m})}{5.89 \times 10^{-2} \text{ N/m}} \right)^{0.31} \left( \frac{958 \text{ kg/m}^3}{0.6 \text{ kg/m}^3} - 1 \right)^{0.31}$$

$$h_{NB} = 3.02q^{0.69}$$

$$q = h_{NB} \Delta T_{sat}$$

$$q = 3.02q^{0.69} \Delta T_{sat}$$

$$q = 35.4 \Delta T_{sat}^{3.23}$$

$$q = 35.4 (20 \text{ K})^{3.23} = 5.65 \times 10^5 \text{ W/m}^2$$

SAMPLE CALCULATION: Forced-Convective Boiling Figure 10

Data:

$$k = 0.68 \text{ W/m K}$$

$$h_{fg} = 2.258 \times 10^6 \text{ J/kg}$$

$$\rho_l = 958 \text{ kg/m}^3$$

$$Cp = 4.21 \times 10^3 \text{ J/kg K}$$

$$\rho_v = 0.6 \text{ kg/m}^3$$

$$\sigma = 5.89 \times 10^{-2} \text{ N/m}$$

$$\mu_l = 2.8 \times 10^{-4} \text{ kg/m s}$$

$$\mu_v = 1.27 \times 10^{-5} \text{ kg/m s}$$

$$d = 1.91 \times 10^{-2} \text{ m} \quad q = 43525 \text{ W/m}^2$$

$$P = 1.45 \times 10^5 \text{ N/m}^2 \quad Pr = 1.66$$

$$v_{fg} = 1.67 \text{ m}^3/\text{kg} \quad x = 0.03$$

Equations:

$$h_{NB} = 0.25 \frac{k}{d} \left( \frac{qd}{h_{fg}\mu} \right)^{0.69} \left( \frac{Cp\mu}{k} \right)^{0.69} \left( \frac{Pd}{\sigma} \right)^{0.31} \left( \frac{\rho_l}{\rho_v} - 1 \right)^{0.31}$$

$$\Delta T_{sat, IB} = \frac{8\sigma T_{sat} v_{fg} h_{FC}}{k_l h_{fg}} (Pr)^2$$

$$h_{FC} = 0.023 Re^{0.8} Pr^{0.4} \frac{k}{d} \quad (\text{one-phase})$$

$$q_{FC} = \frac{Re_l^{0.9} Pr_l F(X_{tt}) k_l}{F_2 d} \Delta T_{sat} \quad (\text{two-phase})$$

$$F(X_{tt}) = 0.15 \left( \frac{1}{X_{tt}} + 2.0 \frac{1}{X_{tt}} \right)^{0.32}$$

$$q = q_{FC} + q_{NB} \left( 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^3 \right)$$

$$q = q_{FC} + Cq^{0.69} \Delta T_{sat} \left( 1 - \left( \frac{\Delta T_{sat, IB}}{\Delta T_{sat}} \right)^3 \right)$$

Calculations:

$$Re = \frac{\dot{m}d}{A\mu} = \frac{(0.21 \text{ kg/s})(1.91 \times 10^{-2} \text{ m})}{\pi/4 (1.91 \times 10^{-2} \text{ m})^2 (2.8 \times 10^{-4} \text{ kg/m}\cdot\text{s})}$$

$$Re = 5 \times 10^4$$

$$h_{FC} = 0.023 (5 \times 10^4)^{0.8} (1.66)^{0.4} \frac{0.68 \text{ W/m}\cdot\text{K}}{1.91 \times 10^{-2} \text{ m}}$$

$$h_{FC} = 5760 \text{ W/m}^2\cdot\text{K}$$

$$\Delta T_{\text{sat, IB}} = \frac{(8)(5.89 \times 10^{-2} \text{ N/m})(373 \text{ K})(1.67 \text{ m}^3/\text{kg})}{(0.68 \text{ W/mK})(2.758 \times 10^6 \text{ J/kg})}$$

$$\times (5760 \text{ W/m}^2 \text{ K})(1.66)^2 = 2.71 \text{ K}$$

$$h_{\text{NB}} = 0.25 \left( \frac{(q)(0.68 \text{ W/mK})}{(2.258 \times 10^6 \text{ J/kg})(2.8 \times 10^{-4} \text{ kg/ms})} \right)^{0.69}$$

$$\times \left( \frac{(4.21 \times 10^3 \text{ J/kgK})(2.8 \times 10^{-4} \text{ kg/ms})}{0.68 \text{ W/mK}} \right)^{0.69}$$

$$\times \left( \frac{(1.45 \times 10^5 \text{ N/m}^2)(1.91 \times 10^{-2} \text{ m})}{5.89 \times 10^{-2} \text{ N/m}} \right)^{0.31}$$

$$\times \left( \frac{958 \text{ kg/m}^3}{0.6 \text{ kg/m}^3} - 1 \right)^{0.31}$$

$$h_{\text{NB}} = 2.735 q^{0.69} = 2.735 (43525 \text{ W/m}^2)^{0.69}$$

$$h_{\text{NB}} = 4341 \text{ W/m}^2 \text{ K}$$

$$X_{\text{tt}} = \left( \frac{0.6 \text{ kg/m}^3}{958 \text{ kg/m}^3} \right)^{0.5} \left( \frac{2.8 \times 10^{-4} \text{ kg/ms}}{1.27 \times 10^{-3} \text{ kg/ms}} \right)^{0.1} \left( \frac{1-0.03}{0.03} \right)^{0.9}$$

$$X_{\text{tt}} = 0.7788$$

$$\text{Re}_\ell = \text{Re}(1-x) = 4.85 \times 10^5$$

$$F(X_{\text{tt}}) = 0.15 \left( \frac{1}{.7788} + 2.0 \frac{1}{.7788} \right)^{0.32}$$

$$F(X_{\text{tt}}) = 0.5176$$

$$F_2 = 5(1.66) + 5 \ln(1 + 5(1.66)) + 2.5 \ln(0.0031)$$

$$\times (4.85 \times 10^4)^{0.812}$$

$$F_2 = 26.91$$

$$q_{FC} = \frac{(4.85 \times 10^4)^{0.9} (1.66) (.5176) (.068 \text{ W/mK}) \Delta T_{\text{sat}}}{(26.91) (1.91 \times 10^{-2} \text{ m})}$$

$$q_{FC} = 18743 \Delta T_{\text{sat}}$$

$$q = 43525 = 18743 \Delta T_{\text{sat}} + 4341 \Delta T_{\text{sat}} \left(1 - \frac{2.71}{\Delta T_{\text{sat}}}\right)^3$$

$$\Delta T_{\text{sat}} = 2.49 \text{ K}$$

since  $\Delta T_{\text{sat, IB}} = 2.71 > \Delta T_{\text{sat}}$ , boiling is suppressed and  $q = q_{FC}$ .

Therefore:

$$h = 18743 \text{ W/m}^2\text{K}$$

SAMPLE CALCULATION: Twist Ratio

|       |   |   |
|-------|---|---|
| Data: | $\dot{m} = 5 \times 10^{-3} \text{ kg/s}$ | $d = 0.25 \text{ in} = 6.35 \times 10^{-3} \text{ m}$ |
|       | $\rho_L = 955 \text{ kg/m}^3$             | $\sigma = 5.89 \times 10^{-2} \text{ N/m}$            |

Equation:

$$Y = \left(\frac{0.31 \rho_L r}{\sigma}\right)^{\frac{1}{2}} u_a$$

Calculation:

$$u_a = \frac{\dot{m}}{\rho A} = \frac{5 \times 10^{-3} \text{ kg/s}}{(955 \text{ kg/m}^3) (\pi/4 (6.35 \times 10^{-3} \text{ m})^2)}$$

$$u_a = 1.65 \times 10^{-1} \text{ m/s}$$

$$\gamma = \left( \frac{(0.31)(955 \text{ kg/m}^3)(6.35 \times 10^{-3} \text{ m})}{(2)(5.89 \times 10^{-2} \text{ N/m})} \right)^{1/2} 1.65 \times 10^{-1} \text{ m/s}$$

$\gamma = 0.66$  diameters/180 degree twist