EFFECT OF SUBCOOLING ON FILM BOILING

IN A HORIZONTAL TUBE

by

EDWARD FRANCIS DOYLE

B.S., Massachusetts Institute of Technology (1959)

SUBMITTED IN PARTIAL FULFILLMENT

OF THE REQUIREMENTS FOR THE

DEGREE OF MASTER OF

SCIENCE

at the

MASSACHUSETTS INSTITUTE OF

TECHNOLOGY

June, 1962

Signature of Author................................................
        Department of Mechanical Engineering, May 28, 1962

Certified by......................................................
                                   Thesis Supervisor

Accepted by......................................................
                   Chairman, Departmental Committee
                               on Graduate Students

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## ABSTRACT

An analytical and experimental study is presented for
stable film boiling of a liquid in forced convection inside
a horizontal tube when the entering liquid is subcooled.

Tube wall temperatures were measured for Freon-113
flowing inside an electrically heated stainless steel tube.
At the same heat transfer rate and flow rate, higher tube
wall temperatures were observed for subcooled liquid flow
than for saturated liquid flow. An analytical model is
presented describing this effect.

Thesis Supervisor:  Warren M. Rohsenow

Title:  Professor of Mechanical Engineering

## ACKNOWLEDGEMENTS

A grant from the National Science Foundation provided the financial support necessary for the investigation conducted and reported in this thesis.

The completion of this report was greatly aided by the following people. To them I wish to express my sincere thanks for the valuable assistance they rendered.

I am deeply grateful to my thesis advisor, Professor Warren M. Rohsenow, whose guidance and assistance were generously provided throughout the entire program.

Dr. Robert A. Kruger designed the equipment and his investigation of film boiling in horizontal tubes was the basis for this project.

Mr. Richard S. Dougall provided many valuable suggestions and assisted in operating the equipment and in performing the calculations.

My wife, Mary, was particularly helpful throughout the program, especially in the preparation of this report.

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# I  INTRODUCTION

In recent years, the process of evaporation has been subjected to careful study. Several distinct regions of boiling exist. The actual process of heat transfer is different in each region. Thorough discussions of the different regions of boiling have been presented by Rohsenow (ref. 1) and Westwater (ref. 2).

When the heating surface is at a temperature only slightly higher than the liquid saturation temperature, the liquid is superheated by natural convection and the heat transferred away at the surface by evaporation.

At somewhat higher values of temperature difference between the heating surface and the liquid, bubbles form at active nuclei on the heated surface and break away. The stirring action of the bubbles in the liquid results in much higher heat transfer rates than that due to natural convection. This nucleate boiling is the type encountered in most commercial applications.

In the transition region which occurs at higher temperature differences, the heat transfer rate drops with increasing temperature difference. Berenson (ref. 3) describes transition boiling as a combination of unstable nucleate and unstable film boiling.

Stable film boiling occurs at very large temperature differences. In film boiling, heat transfer rate again increases with increasing temperature difference. A thin vapor layer prevents the liquid from wetting the heated surface in stable film boiling. Radiation is important in film boiling due to the high temperature differences. Film boiling is the type of boiling that is discussed in this report.

Stable film boiling on external geometries has been observed and treated by several investigators. Bradfield (ref. 4) and Cess and Sparrow (ref. 5) analyzed laminar forced convection film boiling. Bromley discussed film boiling on a horizontal rod in natural convection (ref. 6) and in forced convection (ref. 7), and developed methods for predicting the heat transfer rates.

On vertical surfaces, Hsu and Westwater (ref. 8) found the film boiling coefficients were higher than those predicted by Bromley's laminar boundary layer analysis and explained the difference as resulting from turbulence in the boundary layer (ref. 9).

Graham (ref. 10) and von Glahn (ref. 11) investigated film boiling of cryogenic fluids inside vertical tubes.

Film boiling in horizontal tubes in forced convection without liquid subcooling was investigated by Kruger (ref. 12). He developed a method of predicting tube wall temperature distribution for constant heat transfer rates along the tube.

The purpose of this investigation was to study the effect of liquid subcooling on film boiling in horizontal tubes. This is a direct extension of the investigation conducted by Kruger.

In a horizontal tube, the vapor produced travels up the side of the tube in a thin film and collects at the top of the tube forming a stratified flow which increases in thickness as it travels down the tube. The nature of the flow can be seen in Figure 1, and the flow model as developed by Kruger is shown in Figures 7 through 9.

The object of the theoretical program was to extend the analysis by Kruger to include the effect of subcooling in the method for calculating tube wall temperatures for a given flow rate and heat transfer rate. Motte and Bromley (ref. 13) derived a relationship for film boiling of flowing subcooled liquids. However, it was not sufficiently similar to this case to be applied.

The object of the experimental program was to relate the tube wall temperature distribution to the heat transfer rate, flow velocity, and liquid subcooling. Only minor changes in the apparatus originally designed by Kruger were necessary for this program.

Both the theoretical and experimental programs reported here and by Kruger are limited to the case where the mass flow of vapor is small compared to the total mass flow in the tube.

## II THEORETICAL PROGRAM

### A. Introduction

The object of the theoretical program was to include the effect of subcooling of the liquid in the analysis of film boiling in a horizontal tube as developed by Kruger (ref. 12). The analysis is limited to the case of constant heat transfer rate along the tube and to the region in which the mass flow of vapor at the tube outlet is small compared to the total mass flow.

Visual observation of the flow showed that subcooling the liquid had no effect on the basic stratified flow pattern. The flow model, shown in Figures 7 through 9, remains the same as developed by Kruger. Vapor is formed in a thin film at the hot tube wall preventing the liquid from wetting the surface. Under the action of buoyancy forces, the vapor flows upward along the hot tube wall and collects in a vapor layer that flows along the top of the tube. The range in which this stratified flow model applies has not been determined.

Subcooling reduces the rate of vapor formation, and this effect must be introduced into the relations developed by Kruger. The exact nature of the subcooling process must be defined to determine the magnitude of this effect.

In order to determine the tube wall temperature distribution for a given heat transfer rate, flow velocity, and initial

liquid subcooling, five basic relations are combined. These relations are for the liquid subcooling heat transfer rate, heat transfer coefficient across the thin vapor film, the vapor layer heat transfer coefficient, the ratio of vapor area to liquid area at any cross section, and conduction around the tube wall.

## B. Liquid Subcooling Heat Transfer Rate

In order to determine the subcooling heat transfer rate, the liquid flow must be examined. Since the liquid is completely surrounded by vapor, the shearing forces on the liquid surface are entirely different than for a liquid flowing through a tube.

The vapor effectively lubricates the tube wall so that the liquid slips along the tube. In the absence of opposing shearing forces, the liquid would essentially have a uniform velocity profile in the flow direction. Actually, the vapor layer at the top of the tube, which moves faster than the liquid, tends to increase the liquid flow rate along the tube. The vapor flowing upward along the tube tends to introduce a secondary flow in the liquid. In this secondary flow the liquid at the vapor film interface moves upward along the vapor film and then mixes in the liquid core.

The relative liquid velocity with respect to vapor velocity at the interface is much lower than the absolute liquid velocity for both the main flow along the tube and for the secondary flow at the vapor film. For the liquid flow

velocities encountered in the experiment, turbulent flow would result if the liquid did not have a vapor layer around it. However, with the vapor layer, a laminar liquid flow could exist.

The controlling subcooling heat transfer mechanism could be either the main liquid flow down the tube or the secondary flow along the vapor film. Both cases were considered. The criterion to determine the more important mechanism is the ratio of tube length divided by axial liquid velocity $(L/V_o)$ to the tube diameter divided by liquid interface velocity $(D/L_{VF})$. This determines the time of boundary layer development in the liquid flow.

To evaluate this ratio, the liquid interface velocity must be determined. One method to determine this is from the factor $\beta'$ as defined by Kruger (ref. 12) and discussed in Appendix A. Kruger experimentally determined a value of 1.4 for $\beta'$. As shown in Appendix A, this corresponds to a value of liquid interface velocity equal to 1.085 times the average vapor velocity in the film. For the values encountered in the experiment, $D/L_{VF}$ would be much greater than $L/V_o$ indicating that the secondary flow is the important heat transfer mechanism. However, other considerations indicate that this may not be true. First, the value of $\beta'$ might be misleading since it is experimentally determined and could include effects not considered in the theory. Secondly, use of this value of $\beta'$ leads to values of subcooling heat transfer rates much higher than

encountered in the experiment. As discussed in more detail
in the section on experimental results, another effect was
encountered in the experiment which indicates that the primary
flow rather than the secondary flow might be the more important
mechanism. With liquid subcooling, the tube wall temperatures
near the tube outlet unexpectedly exceeded the highest tempera-
ture encountered anywhere on the tube for a saturated liquid
flow. As much as $100^\circ F$ higher tube wall temperatures were
encountered for $50^\circ F$ of subcooling.

A simple calculation shows that, for the heat transfer
rates and flow rates encountered in the experiments, a liquid
subcooled $50^\circ F$ will not rise greatly in temperature. This is
an important consideration. If the secondary flow was the main
subcooling heat transfer mechanism, the resulting heat transfer
coefficient would be approximately constant along the length
of the tube. With the amount of subcooling remaining constant
over the length of the tube, the subcooling heat transfer rate
would be approximately constant. When this effect is combined
with the other relationships, the tube wall temperature is
always lower with subcooling than for the corresponding case
for saturated liquid.

When the primary liquid flow is considered as the con-
trolling subcooling heat transfer mechanism, the heat transfer
coefficient decreases along the tube and the subcooling heat
transfer rate also decreases. This effect, when combined with

the other relationships, can result in the higher tube wall
temperatures observed with subcooling.

Since the liquid interface velocity could not be accurately
determined, and since the experiment indicated that the primary
flow might be the controlling subcooling mechanism, the primary
flow is considered as controlling in the theory. The effect
of the secondary flow will be neglected.

If a uniform velocity profile is assumed for the primary
liquid flow along the tube, the liquid can be considered as
a semi-infinite solid of uniform temperature, $T_f$. At the tube
entrance, it is exposed to a constant surface temperature, $T_s$,
equal to the saturation temperature. The liquid bulk tempera-
ture, $T_f$, will be considered constant along the tube. The
energy equation for this model is

$$\rho_L c_{p_L} \frac{dT_L}{dt} = K_L \frac{d^2 T_L}{dy^2} \tag{II-1}$$

where $T_L$ is the liquid temperature, y the distance from the
liquid vapor interface inward, and t the time.

For constant fluid properties and for constant surface
temperature, $T_s$, applied at y = 0 and t = 0, the solution of
equation (II-1) is given by (ref. 14)

$$\frac{T_L - T_s}{T_f - T_s} = erf \frac{y}{2\sqrt{\left(\frac{K}{\rho c_p}\right)_L t}} \tag{II-2}$$

where erf is the error function.

The subcooling heat transfer rate per unit area is given by

$$q/A_{sc} = -K_L \left(\frac{dT_L}{dy}\right)_{y=0} = -K_L \frac{(T_f - T_s)}{\sqrt{\pi}} \sqrt{\left(\frac{\rho c_p}{K}\right)_L} \sqrt{\frac{1}{t}} \tag{II-3}$$

The time in equation (II-3) is the time required for the liquid to travel from the entrance to a position x along the tube. This is given by

$$t = \frac{x}{\overline{V_L}} \qquad \text{(II-4)}$$

where $\overline{V}_L$ is the average liquid velocity to the position x.

The liquid velocity $V_L$ is

$$V_L = \frac{w_L}{\rho_L A_L} \qquad \text{(II-5)}$$

Since the mass of vapor formed is small compared to the total mass flow, $w_L$ is considered constant. The liquid area can be written as

$$A_L = \frac{\pi D^2}{4}\left[1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha\right] \qquad \text{(II-6)}$$

where D is the tube diameter and the angle $\alpha$ is defined in Figure 7.

At the tube entrance

$$w_L = \rho_L V_0 \frac{\pi D^2}{4} \qquad \text{(II-7)}$$

Combining these equations

$$V_L = \frac{V_0}{\left[1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha\right]} \qquad \text{(II-8)}$$

and

$$\overline{V}_L = \int_0^x \frac{V_0\, dx}{\left[1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha\right]} \qquad \text{(II-9)}$$

Combining this relation with (II-3)

$$q/A_{sc} = \frac{K(T_s - T_f)}{\sqrt{\pi}} \left(\frac{\rho c_p}{K}\right)_L^{1/2} \sqrt{\frac{V_o}{X}} \sqrt{\int_0^X \frac{dx}{\left[1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha\right]}} \qquad (II-10)$$

This heat transfer rate is based on the liquid surface

area given by

$$dA_{LS} = \pi D \left(1 - \frac{\alpha}{\pi} + \frac{1}{\pi}\sin\alpha\right)dx \qquad (II-11)$$

Introducing this relation

$$\frac{q_{sc}}{\pi D} = \frac{K(T_s-T_f)}{\sqrt{\pi}} \sqrt{\frac{V_o}{X}} \left(\frac{\rho c_p}{K}\right)_L^{1/2} \sqrt{\int_0^X \frac{dx}{\left(1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha\right)}} \int_0^X \left(1 - \frac{\alpha}{\pi} + \frac{\sin\alpha}{\pi}\right)dx \qquad (II-12)$$

$q_{sc}/\pi D$ is the average subcooling heat transfer rate

based on the tube area, $\overline{q/A}_{sc}$.

For $0 \leq \frac{\alpha}{\pi} \leq .50$ the combined integrals in equation

(II-12) are always approximately equal to 1.0 making this

simplification

$$\overline{q/A}_{sc} = \frac{K(T_s-T_f)}{\sqrt{\pi}} \left(\frac{\rho c_p}{K}\right)_L^{1/2} \sqrt{\frac{V_o}{X}} \qquad (II-13)$$

From equation (II-13) the assumption of a constant

liquid bulk temperature, $T_f$, can be verified. For the range

of the experiment $T_f$ increases a maximum of 10% at the tube

outlet.

Relation (II-13) can also be written as

$$\overline{q/A}_{sc} = \frac{K}{\sqrt{\pi}}(T_s-T_f)\left(\frac{\mu c_p}{K}\right)_L^{1/2} \sqrt{\frac{\rho V_o}{\mu X}} \qquad (II-14)$$

This is the same relation obtained by considering a uniform

velocity profile in the boundary layer over a flat plate.

The local average value of subcooled heat transfer is
given by

$$\overline{q/A_{sc}}_L = \frac{1}{\pi D} \frac{d}{dx} \left( \overline{q/A_{sc}} \, \pi D x \right) = \frac{1}{2} \overline{q/A_{sc}} \tag{II-15}$$

and then basing the local value on the liquid surface area

$$q/A_{sc_L} = \frac{\frac{1}{2} \overline{q/A_{sc}}}{\left( 1 - \frac{\alpha}{\pi} + \frac{1}{\pi} \sin \alpha \right)} \tag{II-16}$$

In considering the secondary flow, equation (II-3) can
be used, but in this case, t equals the distance the liquid
travels upward along the vapor film divided by the average
liquid interface velocity. Since this effect is not considered
as controlling, the complete derivation is not presented.

## C. Heat Transfer Coefficient Across the Thin Vapor Film

Subcooling reduces vapor formation, and the equations
developed by Kruger (ref. 12) must be revised to include this
effect. Heat flows into the liquid vapor interface from the
hot tube wall by radiation and convection while heat is re-
moved from the interface by convection into the subcooled
liquid. The rate of vapor formation per unit area at the
vapor film interface is given by

$$\frac{dw_{VF}}{d\theta} = \frac{D(q/A_T - q/A_{sc_L})}{2 \lambda'_{VF}} \tag{II-17}$$

where $q/A_T$ is the heat transfer rate from the tube wall by
both radiation and convection in the vapor film region, $q/A_{sc_L}$
is the subcooling heat transfer locally, and $\lambda'_{VF}$ is the average
heat content of the vapor in the film above that of saturated
liquid.

Assuming laminar flow in the vapor film and neglecting momentum changes, the following equations can be written for the geometry shown in Figure 7.

For control volume A

$$\frac{\partial P}{\partial \theta} = -\rho_L g \frac{D}{2} \sin \theta \qquad (II-18a)$$

For control volume B

$$\frac{\partial P}{\partial \theta} = -\rho_V g \frac{D}{2} \sin \theta - \tau_{WT} \frac{D}{2b} \qquad (II-18b)$$

The total shearing stress on the vapor, $\tau_{WT}$, can be written as (see Appendix A)

$$\tau_{WT} = 2 (\beta')^3 \frac{\mu_V W_{VF}}{\rho_V b^2} \qquad (II-19)$$

where as shown by Kruger (ref. 12) and as derived in Appendix A

$$\frac{3}{2} \leq (\beta')^3 \leq 6 \qquad (II-20)$$

These two extremes represent the cases where the liquid is either completely rigid or completely frictionless.

Combining (II-18a), (II-18b), and (II-19), the vapor film thickness, b, can be obtained.

$$b = \beta' \left\{ \frac{\mu_V \left[ (q/A)_T - (q/A)_{SC_L} \right] D}{\rho_V (\rho_L - \rho_V) g \lambda'_{VF}} \right\}^{1/3} \left\{ \frac{\theta}{\sin \theta} \right\}^{1/3} \qquad (II-21)$$

Assuming a linear temperature distribution in the vapor, the local boiling heat transfer coefficient can be written as

$$h_{LB} = K/b \qquad (II-22)$$

Combining (II-21) and (II-22)

$$h_{LB} = \frac{1}{\beta'} \left\{ \frac{K_v^3 \rho_v (\rho_L - \rho_v) g \lambda'_{vF}}{D \mu_v (g/A_T - g/A_{sc_L})} \right\}^{1/3} \left\{ \frac{\sin\theta}{\theta} \right\}^{1/3} \quad \text{(II-23)}$$

The average heat transfer coefficient in the boiling region is

$$\bar{h}_{LB} = \frac{1}{\pi-\alpha} \int_0^{\pi-\alpha} h_{LB} \, d\theta \quad \text{(II-24)}$$

defining

$$\gamma \equiv \frac{1}{\pi-\alpha} \int_0^{\pi-\alpha} \left( \frac{\sin\theta}{\theta} \right)^{1/3} d\theta \quad \text{(II-25)}$$

The average coefficient can be written as

$$\bar{h}_{LB} = \frac{\gamma}{\beta'} \left\{ \frac{K_v^3 \rho_v (\rho_L - \rho_v) g \lambda'_{vF}}{D \mu_v (g/A_T - g/A_{sc_L})} \right\}^{1/3} \quad \text{(II-26)}$$

Kruger obtained $\gamma$ by numerical integration and the result is presented in Figure 10 as a function of $\alpha/\pi$. Kruger also determined experimentally that $\beta' = 1.4$.

The properties are evaluated at the mean film temperature. Kruger states that this introduces an error of less than 10%. Evaluation of the properties at this temperature is equally valid with subcooling. The average vapor heat content $\lambda'_{vF}$ is

$$\lambda'_{vF} = \lambda + \frac{1}{2} C_{\rho_v} (T_w - T_s) \quad \text{(II-27)}$$

D. **Ratio of Vapor Area to Liquid Area at Any Cross Section**

The ratio of vapor area to liquid area at any cross section can be obtained by applying the momentum equation to the control volumes shown in Figure 8.

For control volume A

$$\left( \frac{dp}{dx} \right)_v A_v - \tau_i D \sin\alpha - \tau_{wv} \alpha D = \frac{d}{dx}(w_{vb} V_v) - V_L \frac{dw_{vb}}{dx} \quad \text{(II-28)}$$

For control volume B

$$\left(\frac{\partial p}{\partial x}\right)_L A_L - \tau_{w_L}(\pi - \alpha) D + \tau_i D \sin \alpha = \frac{d}{dx}(w_L V_L) + V_L \frac{dw_{vb}}{dx} \tag{II-29}$$

When the mass flow of vapor is small compared to the total mass flow, the above equations can be greatly simplified. With this assumption, Kruger showed that all the frictional terms could be neglected, and that only the momentum change in the liquid, due to the area change, and the momentum change in the vapor, due to the addition of vapor, were important.

The vapor velocity, $V_V$, liquid velocity, $V_L$, and vapor area, $A_V$, can be eliminated from (II-28) and (II-29) with the following equations.

$$V_L = \frac{w_L}{\rho_L A_L} \tag{II-30a}$$

$$V_V = \frac{w_{vb}}{\rho_V A_V} \tag{II-30b}$$

$$A_V = A_T - A_L \tag{II-30c}$$

The pressure gradient in the vapor must equal the pressure gradient in the liquid to maintain continuity of pressure across the horizontal interface. The above equations then yield

$$\frac{(A_T - A_L)^2}{A_L^3} \frac{\partial A_L}{\partial x} = -\frac{2\rho_L w_{vb}}{\rho_V w_L^2} \frac{dw_{vb}}{dx} \tag{II-31}$$

The liquid flow rate can be considered constant as given by (II-7).

The vapor flow is

$$w_{vb} = \frac{(\overline{q/A} - \overline{q/A}_{sc})\pi D x}{\gamma_{vb}'} \tag{II-32}$$

where the average heat content of the vapor, $\lambda'_{Vb}$, is

$$\lambda'_{vb} = \lambda_{vg} + C_{P_{vb}}(T_{vb} - T_s) \qquad (II-33)$$

The average subcooling heat transfer rate, $\overline{q/A}_{sc}$, is given

by (II-13).

Since $w_{vb}$ is a function of $x$ only (II-31) can be integrated from 0 to $x$.

$$\left\{ \left(\frac{A_T}{A_L}\right)^2 - 4\frac{A_T}{A_L} + 2\,\ell n\,\frac{A_T}{A_L} + 3 \right\} = \frac{2\rho_L}{\rho_v}\,\frac{w_{vb}^2}{w_L^2} \qquad (II-34)$$

$\dfrac{A_L}{A_T}$ can be obtained from (II-6).

$$\frac{A_L}{A_T} = 1 - \frac{\alpha}{\pi} + \frac{1}{2\pi}\sin 2\alpha \qquad (II-35)$$

defining

$$X_M \equiv \sqrt{\frac{2\rho_L}{\rho_v}}\,\frac{w_{vb}}{w_L} \qquad (II-36)$$

combining the above relations

$$X_M = \frac{4\sqrt{2}\,(\overline{q/A} - \overline{q/A}_{sc})\,x}{\sqrt{\rho_L\rho_v}\,V_o\,\lambda'_{vb}\,D} = f\left(\frac{\alpha}{\pi}\right) \qquad (II-37)$$

This function is shown in Figure 11.

E.  Heat Transfer Coefficient for the Vapor Layer

In order to estimate the heat transferred directly to the
vapor layer from the top of the tube, equations for flow in a
round tube are used. The concept of hydraulic diameter is
introduced to account for the different flow geometry.

For the range of the experiments, the vapor Reynolds number
indicates that both laminar and turbulent flows can occur. For
this reason, both types of flow must be considered.

The hydraulic diameter is defined by

$$D_e = 4 \frac{\text{cross sectional area}}{\text{wetted perimeter}}$$ (II-38)

For the geometry of Figure 7

$$D_e = D \left\{ \frac{\alpha/\pi - \frac{1}{2\pi} \sin 2\alpha}{\alpha/\pi + \frac{1}{\pi} \sin \alpha} \right\}$$ (II-39)

For fully developed laminar flow and constant heat transfer rate (ref. 15)

(II-40)

$$\frac{h_{LV} D_e}{K_{vb}} = 4.36$$

Then for laminar flow in the vapor layer, the heat transfer coefficient is

$$h_{LV} = \frac{4.36 \, K_{vb}}{D} \left\{ \frac{\alpha/\pi + \frac{1}{\pi} \sin \alpha}{\alpha/\pi - \frac{1}{2\pi} \sin 2\alpha} \right\}$$ (II-41)

For fully developed turbulent flow (ref. 16)

$$\frac{h_{LV} D_e}{K_{vb}} = .023 \left( \frac{\rho_{vb} V_{vb} D_e}{\mu_{vb}} \right)^{.8} \left( \frac{\mu c_p}{K} \right)_{vb}^{.40}$$ (II-42)

Using (II-30b), (II-30c), (II-32) and (II-6)

$$V_{vb} = \frac{4 (q/A - q/A_{sc}) x}{\rho_{vb} \, \gamma_{vb}' \, D \left( \frac{\alpha}{\pi} - \frac{1}{2\pi} \sin 2\alpha \right)}$$ (II-43)

Then for turbulent flow in the vapor layer, the heat transfer coefficient is

$$h_{LV} = .023 \frac{K_{vb}}{D} \left( \frac{4(q/A - q/A_{sc}) x}{\mu_{vb} \, \gamma_{vb}'} \right)^{.8} \left( \frac{\mu c_p}{K} \right)_{vb}^{.4} \frac{\left( \frac{\alpha}{\pi} + \frac{\sin \alpha}{\pi} \right)^{.20}}{\left( \frac{\alpha}{\pi} - \frac{\sin 2\alpha}{2\pi} \right)}$$ (II-44)

In order to evaluate the film coefficient, the bulk temperature in the vapor layer must be determined. By using the condition at the horizontal liquid-vapor interface, an analysis was performed to obtain the bulk temperature in the

vapor layer. Calculations indicated that the bulk temperature was approximately half way between the tube wall temperature and the saturation temperature for both saturated and subcooled liquid flows. Using this approximation greatly simplifies the analysis.

The vapor flow is laminar near the tube inlet. At some point along the tube the flow becomes turbulent. The experiments strongly indicate that the transition from laminar to turbulent flow occurs at different vapor Reynolds numbers, depending on the amount of subcooling. With a saturated liquid flow, the experiment indicates the vapor quickly becomes turbulent. With subcooling, the flow appears to remain laminar at higher vapor Reynolds numbers.

At present, no method has been developed for determining the transition vapor Reynolds number as a function of subcooling. For this reason, the flow is assumed turbulent for saturated liquid flows and laminar for highly subcooled flows. No correlation has been determined for intermediate amounts of subcooling.

## F. Conduction in the Tube Wall

By using the model shown in Figure 9, Kruger developed an equation for the conduction around the tube. This relation is necessary because the heat transferred into the vapor film at the bottom of the tube can be significantly different from the heat transferred directly to the vapor at the top of the tube.

Since the tube is electrically heated, the heat generation per unit volume is essentially constant. The outside of the tube is assumed to be perfectly insulated. Radial temperature gradients are neglected. The heat transfer coefficients in the vapor film region and in the vapor layer region are taken as constant.

For a heat balance on an element of the tube wall, Kruger obtained

$$\frac{d^2 T}{d\theta^2} - \frac{hr^2}{K_s t} T = -\frac{hr^2}{K_s t} T_B - \frac{r^2}{K_s t} \left[ \overline{q/A} - \overline{q/A_{RAD}} \right] \qquad \text{(II-45)}$$

Equation (II-45) can be applied separately to the two regions of the tube.

In region 1
$$\left. \begin{array}{l} h = \hbar_{LB} \\ T_B = T_s \end{array} \right\} \quad 0 \le \theta \le (\pi - \alpha) \qquad \text{(II-46a)}$$

In region 2
$$\left. \begin{array}{l} h = \hbar_{LV} \\ T_B = T_{Vb} \end{array} \right\} \quad (\pi - \alpha) \le \theta \le \pi \qquad \text{(II-46b)}$$

The vapor is considered transparent to radiation from the tube wall. Kruger also assumed that the view factor between the tube wall and the liquid was constant for the values of $\alpha/\pi$ encountered in the experiments. He also showed that the radiation could be considered constant and evaluated at the mean tube wall temperature without introducing significant error. These assumptions are sufficient to linearize equation (II-45). With these assumptions, the heat transfer by radiation from the tube wall is

$$q/A_{RAD} = \epsilon_s \sigma \left( \overline{T}_w^4 - T_s^4 \right) \qquad \text{(II-47)}$$

Equation (II-45) can be applied separately to each region as described by (II-46). The boundary conditions to be used are

1. By symmetry

$$\frac{\partial T}{\partial \theta} = 0 \text{ at } \theta = 0 \tag{II-48a}$$

$$\frac{\partial T}{\partial \theta} = 0 \text{ at } \theta = \pi \tag{II-48b}$$

2. For continuity between the two regions

$$T_1 = T_2 \text{ at } \theta = (\pi - \alpha) \tag{II-48c}$$

$$\frac{\partial T_1}{\partial \theta} = \frac{\partial T_2}{\partial \theta} \text{ at } \theta = (\pi - \alpha) \tag{II-48d}$$

For these boundary conditions, Kruger obtained the following equations for the temperature distribution around the tube

$$\frac{T_1 - b_1}{b_2 - b_1} = \frac{1}{\xi} \frac{\cosh a_1 \theta}{\cosh a_1 (\pi - \alpha)} \text{ for } 0 \leq \theta \leq (\pi - \alpha) \tag{II-49a}$$

$$\frac{T_2 - b_2}{b_2 - b_1} = \left(\frac{1}{\xi} - 1\right) \frac{\cosh a_2 (\pi - \theta)}{\cosh a_2 \alpha} \text{ for } (\pi - \alpha) \leq \theta \leq \pi \tag{II-49b}$$

where

$$\xi = 1 + \frac{a_1}{a_2} \frac{\tanh a_1 (\pi - \alpha)}{\tanh a_2 \alpha} \tag{II-50a}$$

$$a_i = \sqrt{\frac{h_i r^2}{K_s t}} \tag{II-50b}$$

$$b_i = \left[\frac{1}{h_i} \left(q/A - q/A_{RAD}\right) + T_{si}\right] \tag{II-50c}$$

The subscripts refer to the section of the tube wall in which the quantities are to be evaluated. The regions are defined by equation (II-46).

## G. Method of Calculation

The temperature distribution in the tube wall can be determined from the equations developed in the theory. Since the properties of the fluid depend on temperature, it is necessary to use an iterative procedure. By assuming a mean tube wall temperature, the fluid properties and the radiation from the tube wall are calculated.

A second difficulty arises because more heat is transferred to the vapor film than to the vapor layer at the top of the tube. By calculating the angle $\alpha$ and the subcooling heat transfer rate, the heat transfer coefficient for the vapor layer can be determined. Based on the assumed mean tube wall temperature, the heat transferred from the tube wall to the vapor layer can be estimated. The corresponding heat that must be transferred from the tube wall at the vapor film region, $q/A_T$, is then calculated. The heat transfer coefficient for the vapor film can then be determined.

Using the relations for conduction in the tube wall, the tube wall temperature distribution can be calculated for a particular location along the tube. The mean tube wall temperature can then be compared to the assumed value. Since the equations are fairly insensitive to the assumed mean temperature, only a few iterations are required.

Using property values supplied by the manufacturer, a special set of figures was prepared for the groups of properties in the equations. This greatly reduces the time required for performing the calculations. It was necessary to extrapolate

the properties to obtain values at the temperatures encountered in the experiment.

The calculation procedure is:

1.  Assume a mean tube wall temperature $\bar{T}_W$

2.  Calculate $q/A_{RAD}$ from (II-47)

3.  Calculate $q/A_{scL}$ from (II-16) and $\overline{q/A_{sc}}$ from (II-13)

4.  Calculate $\alpha/\eta$ from (II-37)

5.  Calculate $h_{LV}$ from (II-41) for a highly subcooled liquid flow, or from (II-44) for a saturated liquid flow.

6.  Using the assumed value of $\bar{T}_W$, estimate the heat transferred to the vapor and calculate the corresponding value of $q/A_T$.

7.  Calculate $\hbar_{LB}$ from (II-26)

8.  Calculate $a_i$, $b_i$, and $\xi$ from (II-50)

9.  Calculate $T_1$ and $T_2$ from (II-49)

10. Calculate the mean tube wall temperature, $\bar{T}_W$, and compare it to the assumed value.

Results for these calculations are discussed in Chapter IV, and are shown in Figures 12 through 22 for saturated liquid flows and for highly subcooled liquid flows.

## III EXPERIMENTAL PROGRAM

### A. Introduction

The object of the experimental program was to determine
the relation between tube wall temperature, heat transfer
rate, flow rate, liquid subcooling, and fluid properties for
film boiling in a horizontal tube. The investigation was
limited to the region where the mass flow of vapor is small
compared to the total mass flow in the tube.

After making a few minor alterations, the experiments
were performed on the apparatus originally designed by
Kruger (ref. 12). The equipment is designed for operation
with Freon-113 flowing through an electrically heated tube.
There are two test sections. The first can be used for visual
and photographic observation. A photograph of the stratified
flow pattern observed with this test section is shown in
Figure 1. The second test section has an electrically heated
stainless steel tube instrumented to measure temperatures at
the top and bottom of the tube. A series of guard heaters
are used on this test section to prevent heat losses to the
atmosphere.

### B. Description of Apparatus

The apparatus was designed to be used for both visual and
quantitative investigations of film boiling. A schematic
diagram showing the main components of the equipment is presented

in Figure 2. More detailed schematics of the visual and quantitative test sections are shown in Figures 3 and 4. Photographs of the actual equipment are shown in Figures 5 and 6.

The apparatus consists of two parallel test sections with one circulation system. The circulation system includes a pump, preheater, condenser, degasser, liquid reservoir, pressure and flow control valves, and rotameters for measuring the liquid flow rate.

The preheater has two heat exchangers made from 20 feet of $\frac{1}{2}$ inch copper tubing coiled in a tank. Temperature is controlled by varying the flow of hot and cold water into the tank. A similar heat exchanger is used for the condenser.

A degasser tank was added to the system to insure that no air was dissolved in the Freon-113. This was simply a brass tank surrounded with a water jacket. The tank had sufficient surface area so that air would rise to the top of the tank where it could be bled away. Hot water was circulated through the tank to drive the air out of the solution. Air is less dense than Freon-113 vapor, and rises to the top of the tank. Since the circulation system was sealed, the degasser could be used as a second condenser after all the air had been driven out of the flow.

Two rotameters were connected in parallel to measure the fluid flow. The rotameters were of different capacity to allow more accurate flow measurements.

A pressure gauge is connected to the system to measure the pressure at the test section inlet. A thermocouple was placed at the preheater outlet to measure the fluid temperature at the test section entrance. Another thermocouple was placed at the condenser outlet to insure that all the vapor was being condensed.

An electrically conducting glass tube with an inside diameter of .418 inches was used for the visual test section. The tube is coated with a semitransparent, electrically conducting material. An ammeter was used to measure the current passing through the tube. The tube was heated from a 220 volt power supply.

A stainless steel tube, .403 inch inside diameter and 15 inches long, was used for the quantitative test section. This test section has a high current d.c. power supply which gives the wide range of controlled power input necessary for the tests. In addition to insulation, there is an electrically powered guard heater ring around the tube to minimize heat losses to the atmosphere. The guard heater ring is at a diameter of 2.5 inches around the tube. The space between the tube and guard heater ring is filled with an insulation of known thermal conductivity.

Thermocouples are located at the top and bottom of the tube at six positions along the tube. There are six guard heaters in the guard heater ring and two end guard heaters. Each has independent controls so that the electrical heat input

to each can be adjusted separately. There is a thermocouple in each guard heater so that the losses to the atmosphere can be measured and adjusted. The thermocouples in the guard heater ring are located half way between the top and bottom tube wall thermocouples. The heat losses are minimized by maintaining these guard heaters at a temperature half way between the top and bottom tube wall temperatures.

A continuous recording Honeywell Potentiometer was installed. There are 16 channels on the recorder. By using a switching circuit, the recorder was operated with 32 channels. This reduced the time to record all the data for one run to thirty seconds. It originally required five minutes to record the data for one run with a simple potentiometer.

## C. Measurements and Accuracy

All temperature measurements were made with chromel-alumel thermocouples. Thermocouples were installed directly in the flow to measure the temperature of the fluid before it enters the test section and as it leaves the condenser.

The thermocouples in the guard heaters were insulated with ceramic tubing to separate them from the guard heater wires. The thermocouples on the tube were tied over thin mica sheets to prevent any electric contact with the tube. Kruger (ref. 12) showed that, for the heat transfer rate encountered in the experiment, the maximum difference between the inner and outer surfaces of the tube would not exceed $3^{\circ}F$.

The average heat transfer rate per unit surface area of the tube was determined from the following relation for the power input.

$$\left(\overline{q/A_s}\right) = \frac{1}{A_s}\left(\rho'\frac{L}{A_c}\right)I^2 \qquad\qquad \text{(III-1)}$$

where $A_s$ is the surface area on the inside of the tube, $\rho'$ the resistivity of the tube, L the length of the tube, $A_c$ the cross sectional area of the tube, and I the current flowing in the tube.

Values for resistivity of stainless steel were obtained from a materials handbook as a function of temperature. For the temperature variations along the tube, encountered in the experiments, the heat transfer rate would vary only a few percent. For this reason, the heat transfer rate was calculated for the value of resistivity at the average tube wall temperature.

The current was measured from the voltage drop across a shunt in the power line. The shunt resistance had been calibrated by the National Bureau of Standards to within 0.1%. The current had a tendency to wander over a period of time. This effect was minimized by installing the recorder to reduce the time required to record all the data for a run.

The voltage drop across the tube was measured by using a resistance bridge. This was used only as a check on the value determined for the heat transfer rate.

The guard heaters were maintained at a temperature between the temperatures measured at the top and bottom of the tube wall.

A temperature difference of $100^{\circ}F$ between the tube wall and the guard heater ring represents only a 1% heat loss. Since the guard heaters were easily maintained within $100^{\circ}F$ of the tube wall temperature, the heat losses are negligible. The end guard heaters were used to minimize losses along the electrical leads.

Gaskets 1/8 inch thick were used to separate the electrically heated section from the inlet and outlet lines. These gaskets were also necessary to reduce heat conduction axially along the tube.

Kruger found that the maximum error in the rotameter readings was less than 5% for the range of these tests.

The recorder and associated switching system could cause some error in the temperature and current readings. Tests were made to determine this error, but no error could be detected within the limits of the recorder scale. At lower recorder readings, where the percentage error for a reading is larger due to the number of scale divisions, the readings were checked with the potentiometer for added accuracy.

## D. Experimental Technique

Due to limitations in the equipment, it was necessary to use a certain amount of care in operating the apparatus. In order to prevent damage to the test section, a limit of $1000^{\circ}F$ was placed on the maximum allowable temperature. For this reason, the test section was heated to the minimum temperature at which stable film boiling could be maintained before the

fluid was started through the test section. For Freon-113 this is approximately 500°F. If the tube was heated while the fluid was flowing, there is a danger of exceeding 1000°F as the boiling process changes from the stable nucleate boiling region to the stable film boiling region.

Before the liquid flow is started, the test section is heated by turning on the guard heaters. The valves on the quantitative test section are closed and the valve opened on the visual test section. The system is then filled with Freon-113 and sealed. The flow is started and bypassed through the visual test section.

While the main test section is being heated to 500°F, there are other operations to perform. The preheater and condensers are put into operation. The preheater is used to heat the liquid to the saturation temperature to drive the air out of the solution. The degasser is also heated and the air is bled away. This procedure is maintained until all the air has been removed from the system.

The main power supply is turned on periodically in order to heat the tube at a faster rate. When the proper temperatures have been reached in the test section and the guard heaters balanced, the main power is turned on and set at the level for the first test. The valves are quickly opened to start the liquid flow in the test section before any overheating can result. The complete starting process takes approximately three hours.

When the run has been started, the preheater and condenser temperatures are adjusted to the proper level. The flow rate and power level are also set to the desired values. When the guard heaters have been balanced and the tube temperatures have reached equilibrium, the data can be recorded. All the data for a run can be recorded in thirty seconds. Approximately twenty minutes are required to reach equilibrium. The experimental conditions can then be changed and the second run started. Another twenty minutes are required to reach equilibrium again.

In operating the visual test section, the flow is bypassed through the quantitative test section. The proper temperature level to establish film boiling in the visual section is obtained by a trial and error process. Only a few minutes are required to heat the visual test section to the proper temperature since the 220 volt power supply is used.

If the flow is started before the proper temperature level is reached, the film boiling will change over to nucleate boiling and immediately cool the tube. In the visual test section, it was observed that there is a sharp interface between the nucleate and film boiling regions. This interface can be seen in Figure 1. The interface gradually moves downstream if the heat losses are high. This effect can also be observed in the main test section, when the tube wall temperatures are too low. When the interface travels downstream beyond a thermocouple location, the temperature drops rapidly back to a value typical of the nucleate region.

## E. Data Recorded

All experiments were conducted with Freon-113 as the test fluid. The pressure in the test section is essentially constant for a given run. The pressures recorded in the test section ranged from 3 to 7.5 psig. for the entire test series. This corresponds to liquid saturation temperatures of 125°F to 141°F.

The subcooling ranged from 6°F to 52°F for the tests.

The heat transfer rates per unit area ranged from 12,200 Btu/hr-ft$^2$ to 19,600 Btu/hr-ft$^2$.

The runs were made for two values of the liquid velocity at the entrance to the test section. These were 1.23 ft/sec. and 1.85 ft/sec.

The temperature of the liquid at the condenser outlet was recorded to be certain that all the vapor had been condensed.

Guard heater temperatures were recorded. These were always balanced at a value between the top and bottom tube wall temperatures.

Temperatures at the top and bottom of the tube wall were measured at six positions along the length of the tube. These points were: $x = 0.083$ ft., $x = 0.271$ ft., $x = 0.50$ ft., $x = 0.75$ ft., $x = 0.978$ ft., and $x = 1.17$ ft. Because of the high end losses near the outlet of the heated section, only the first five values are reported.

The important data is recorded in the Table of Experimental Results. These results are also shown in Figures 12 through 22.

## IV   RESULTS AND CONCLUSIONS

### A.  Discussion

The results of the experimental investigation are shown
in Figures 12 through 22.  Calculations from the analysis are
also shown for saturated liquid flows and for highly subcooled
liquid flows.  In these figures the temperature difference is
plotted as a function of the distance from the tube inlet.
The temperature difference is tube wall temperature minus fluid
saturation temperature.

Three basic differences were observed in the tube wall
temperature distribution when the liquid was subcooled.  They
are:

1.  Near the tube inlet, lower tube wall temperatures
    were observed with a subcooled liquid flow than
    those measured for a comparable saturated liquid
    flow.

2.  Near the tube outlet, higher tube wall temperatures
    were observed with a subcooled liquid flow than
    those measured at any location along the tube for a
    comparable saturated liquid flow.

3.  The difference between top and bottom tube wall
    temperature increased significantly with a sub-
    cooled liquid flow.

All these effects are shown in Figures 12, 13 and 14.  In these
figures the heat transfer rate and flow rate are constant and
the amount of subcooling is the variable.

Figures 15 and 16 show that as the subcooling is increased, the maximum tube wall temperature continues to rise. The average tube temperatures are shown in these figures for different amounts of subcooling. The heat transfer rate and flow rate are again the same for all runs.

A better correlation of the data is obtained by holding the subcooling and flow rate constant while the heat transfer rate is varied. Figures 17 through 22 show that this type of correlation does not exhibit the temperature profile crossover effect illustrated in Figures 12 through 16. For clarity, the mean tube wall temperature is used in these figures.

When the temperature profile crossover effect was first observed, some changes were made in the apparatus. First, a degasser was added to make certain that the results were not affected by air dissolved in the Freon-113. A pressure control was added so that the test section pressure could be held constant for both saturated and subcooled liquid flows. These changes had no effect on the observed temperature profile crossover.

The heat transfer rates were also carefully observed to be certain that there were no significant differences between saturated and subcooled runs. Runs were made where the liquid was gradually subcooled and the changing temperature profiles were continuously observed during this process.

Observation in the visual test section showed that no basic change occurred in the stratified flow as a result of subcooling.

An explanation of the effect can be obtained from the three basic differences observed with subcooling. First, higher tube wall temperatures indicate that the overall average heat transfer coefficient near the outlet of the tube is lower when the liquid is subcooled. The overall average heat transfer coefficient would include radiation, convection into the vapor layer and convection into the vapor film.

At the inlet, the film covers most of the tube, and convection into the vapor film is the most important method of heat transfer. With subcooling the observed tube wall temperatures were lower near the tube inlet. Since subcooling reduces the vapor in the film, the film heat transfer coefficient would be expected to increase. This would result in a lower tube wall temperature with subcooling. Therefore, the crossover effect does not appear to be the result of any unusual effects taking place in the vapor film.

There is no reason to believe the heat transferred from the wall by radiation would be significantly affected by subcooling the liquid flow. The only other method of heat transfer is convection into the vapor layer.

The effect of the heat transferred into the vapor increases along the tube. This is due to the larger flow area occupied by the vapor. Since the increase in tube wall temperatures for subcooled flows occurs downstream from the inlet, it seems reasonable to suspect that the heat transferred to the vapor layer is reduced by subcooling.

The increased difference between top and bottom tube wall temperatures means that the ratio of heat transferred to the vapor layer to heat transferred to the vapor film must be increasing with subcooling. Since the film heat transfer coefficient would be higher with subcooling, this effect would be expected. However, the difference was much larger than that observed in the saturated case. This also indicates that subcooling reduces the heat transferred to the vapor.

Since the temperatures at the outlet of the tube with subcooled flow greatly exceed the maximum temperature anywhere on a tube with saturated flow, the effect of subcooling on the heat transferred to the vapor layer must be very great. To account for this effect requires more than a small reduction in the vapor layer heat transfer coefficient.

When the vapor layer starts to grow near the tube inlet, the boundary layers developing at the tube wall and at the horizontal interface are probably laminar. However, as the vapor Reynolds number increases due to vapor formation, the boundary layers could become turbulent.

The rate of vapor formation is greater when the liquid flow is saturated. This means the disturbances in the vapor layer are greater than would occur with a subcooled liquid flow. Greater disturbances in the boundary layer can cause a transition to turbulent flow at lower vapor Reynolds numbers. Under these conditions, the vapor layer would become turbulent at much lower vapor Reynolds numbers.

Subcooling could result in condensation at the horizontal liquid-vapor interface. The boundary layer could remain stable to very high vapor Reynolds numbers under these conditions.

Heat transfer coefficients in the vapor layer would be many times higher for turbulent flow than for laminar flow at the same Reynolds number. This effect is large enough to account for the much higher tube wall temperatures observed with subcooled flow.

In order to predict the tube wall temperature distribution, it would be necessary to develop a criterion to determine the heat transfer coefficient in the region of transition from laminar to turbulent flow. Such a relation must include the effect of subcooling on transition. No method has been developed as yet to predict this transition.

Since a criterion was not developed to predict transition, only the limiting cases can be determined from the theory. With saturated liquid flow, the transition would take place close to the tube inlet. It was assumed that the vapor layer was turbulent for a saturated liquid flow.

With high subcooling, $50^\circ$F for these experiments, the vapor layer was assumed to remain laminar over the entire length of the tube.

It was necessary to assume a bulk temperature in the vapor layer to make these calculations. An analysis was made to

determine a value for the bulk temperature by considering the conditions at the interface. This analysis showed that the bulk temperature was approximately half way between the tube wall temperature and saturation temperature for both subcooled and saturated liquid flow. This value was used in the calculations.

The bulk temperature effect, although in the right direction, was not large enough to cause the increased temperatures with subcooling. Since the predominant effect appears to be the transition from laminar to turbulent flow, the effect of changes in the bulk temperature in the vapor layer were neglected. To determine temperature distribution for intermediate amounts of subcooling, this effect might have to be included in the analysis.

The model chosen for subcooling was discussed completely in the theoretical section. The effects of other assumptions in the theory were discussed in detail by Kruger (ref. 12).

The results of the analysis made with these assumptions are shown in Figures 12 through 22. For the limiting cases, the analysis shows good agreement with the experiment.

The calculations compare favorably with experimental results with respect to:

1. Temperature level,

2. Temperature distribution around the tube,

3. Rate of change of temperature along the tube, and

4. Relative level of temperatures along the tube for subcooled flows with respect to saturated flows.

B. <u>Summary of Results and Conclusions</u>

1. Experimental results are presented for film boiling with subcooling of the liquid flow for Freon-113 flowing inside a horizontal tube.

2. A theoretical model was developed which explains the effects observed with subcooled liquid flow. Calculations are presented for saturated and highly subcooled liquid flows which appear to represent, with a certain degree of accuracy, the results observed in the experiments. The analysis cannot accurately predict the effects of intermediate values of subcooling.

3. Both the theory and experimental investigations were limited to the region in which the mass flow of vapor is small compared to the total mass flow.

4. The region in which the theoretical model applies was not determined.

5. Visual observation confirmed many of the basic assumptions made in the flow model.

6. Near the tube inlet subcooling the liquid decreases the tube wall temperature, while it increases the tube wall temperature downstream.

# NOMENCLATURE

## English Symbols

| | |
|---|---|
| $A$ | = Area |
| $a_1$, $b_1$ | = Non-dimensional parameters |
| $b$ | = Thickness of the vapor film |
| $C$ | = Constant |
| $C_p$ | = Specific heat at constant pressure |
| $D$ | = Diameter |
| $De$ | = Equivalent diameter |
| $f$ | = Friction factor |
| $g$ | = Acceleration of gravity |
| $h$ | = Heat transfer coefficient |
| $I$ | = Electric current |
| $K$ | = Thermal conductivity |
| $K_s$ | = Thermal conductivity of the steel tube |
| $L$ | = Length of the tube |
| $P$ | = Pressure |
| $q$ | = Heat transfer rate |
| $(q/A)$ | = Heat transfer rate per unit area |
| $\overline{(q/A)}$ | = Mean heat transfer rate per unit area |
| $r$ | = Radius |
| $T$ | = Temperature |
| $t$ | = Thickness of the tube wall |
| $V$ | = Velocity |
| $V_o$ | = Velocity of the liquid at the test section inlet |

w   = Mass flow rate

x   = Distance from the tube inlet

Xm  = .Non-dimensional tube length-flow rate factor

y   = Distance from interface to the tube wall in the vapor
      film

Greek Symbols

$\alpha$   = Interface angle shown in Figure 7

$\beta'$   = Experimentally determined coefficient related to
      velocity at liquid vapor interface in film region

$\gamma$   = Averaging factor for the local boiling heat transfer
      coefficient

$\epsilon_s$ = Emissivity of the tube surface

$\theta$   = Angle from the bottom of the tube shown in Figure 7

$\lambda$  = Latent heat of the liquid

$\lambda'$ = Average heat content of the vapor

$\mu$   = Viscosity

$\xi$   = Non-dimensional number

$\rho$   = Density

$\rho'$  = Electrical resistivity

$\sigma$   = Stefan-Boltsman constant

Subscripts

B   = Refers to a temperature in tube wall conduction equations

c   = Convection

co  = Convection in the absence of radiation

| | | |
|---|---|---|
| f | = | Temperature of liquid at tube inlet |
| i | = | Horizontal liquid-vapor interface |
| L | = | Liquid |
| LB | = | Local conditions in the boiling region |
| LV | = | Local conditions in the vapor region |
| r | = | Radial |
| RAD | = | Radiation |
| s | = | Saturation temperature |
| S | = | Refers to surface area |
| sc | = | Refers to subcooling |
| T | = | Total |
| V | = | Vapor |
| Vb | = | Refers to vapor layer at top of tube |
| VF | = | Refers to vapor film region |
| W | = | Conditions at the tube wall |
| 1 | = | Evaluated in the region |
| 2 | = | Evaluated in the region |

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## APPENDIX A

## Velocity Profile in the Vapor Film

A parabolic velocity profile was assumed in the vapor film. The general form of the equation is

$$\frac{V}{\bar{V}} = C_1 + C_2 \frac{y}{b} + C_3 \left(\frac{y}{b}\right)^2 \qquad \text{(A-1)}$$

where $\bar{V}$ is the average vapor velocity in the film, y is the distance from the interface to the tube wall, and b is the film thickness.

The velocity, V, at the tube wall must be zero.

$$V = 0 \quad \text{at } y = b \qquad \text{(A-2)}$$

The average velocity must satisfy the condition

$$\int_0^1 \frac{V}{\bar{V}} d\left(\frac{y}{b}\right) = 1 \qquad \text{(A-3)}$$

A third condition can be obtained for the total shearing stresses on the vapor

$$\tau_{W_T} = \mu_v \left[ \left(\frac{dV}{dy}\right)_{y=0} - \left(\frac{dV}{dy}\right)_{y=b} \right] \qquad \text{(A-4)}$$

The constants in equation (A-1) can be determined from the boundary condition given by (A-2), (A-3), and (A-4). The equation that results is

$$\frac{V}{\bar{V}} = 3\left(1 - \frac{y}{b}\right) - \frac{b\tau_{W_T}}{6\mu_v \bar{V}} \left(1 - 4\frac{y}{b} + 3\left(\frac{y}{b}\right)^2\right) \qquad \text{(A-5)}$$

The limits on the shearing stress can be determined. If the liquid at the interface is rigid, then

$$\left(\frac{dV}{dy}\right)_{y=0} = \left(\frac{dV}{dy}\right)_{y=b} \qquad \text{(A-6)}$$

and the maximum shearing stress is

$$\tau_{w_{T\,max}} = \frac{12\,\mu_v\,\bar{V}}{b} \tag{A-7}$$

If the liquid is considered frictionless at the inter-face

$$\left(\frac{dV}{dy}\right)_{y=0} = 0 \tag{A-8}$$

and the minimum shearing stress is

$$\tau_{w_T} = \frac{3\,\mu_v\,\bar{V}}{b} \tag{A-9}$$

The coefficient $\beta'$ is defined as

$$\beta'^3 \equiv \frac{\tau_{w_T}\,b}{2\,\mu_v\,\bar{V}} \tag{A-10}$$

The limits on $\beta'$ are

$$\frac{3}{2} \leq \beta'^3 \leq 6 \tag{A-11}$$

Kruger (ref. 12) determined that the best experimental correlation is obtained when $\beta' = 1.4$.

Substituting (A-10) into (A-5) yields

$$\frac{V}{\bar{V}} = 2\left(1 - \frac{y}{b}\right) - \frac{1}{3}\beta'^3\left(1 - 4\frac{y}{b} + 3\left(\frac{y}{b}\right)^2\right) \tag{A-12}$$

The velocity at the liquid-vapor interface is

$$V = \left(2 - \frac{1}{3}\beta'^3\right)\bar{V} \tag{A-13}$$

## TABLE OF EXPERIMENTAL RESULTS

The double subscripts in the following table indicate the location of the tube wall temperature measurements in the following manner.

A.  The first subscript refers to the position of the thermo-couple along the tube as shown below.

| Subscript | x - ft. |
|-----------|---------|
| 1 | 0.083 |
| 2 | 0.27 |
| 3 | 0.50 |
| 4 | 0.75 |
| 5 | 0.98 |

B.  The second subscript refers to the position of the thermo-couple around the tube as shown below.

b - Bottom of tube wall

t - Top of tube wall

m - Arithmetic mean of top and bottom tube wall temperature

# TABLE OF EXPERIMENTAL RESULTS

| Run No. | 1 | 2 | 3 | 4 | 5 | 6 |
|---|---|---|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 |
| I - amps. | 253 | 247 | 247 | 253 | 253 | 256 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 13680 | 13100 | 13150 | 13800 | 13820 | 14150 |
| P - psig. | 6.5 | 5.5 | 5.5 | 5.5 | 5.5 | 4.0 |
| $T_s$ - $^oF$ | 139 | 136 | 136 | 136 | 136 | 131 |
| $T_f$ - $^oF$ | 126 | 119 | 117 | 117 | 116 | 103 |
| $(T_s-T_f)$ - $^oF$ | 13 | 17 | 19 | 19 | 20 | 28 |
| $(T_{1b}-T_s)$ - $^oF$ | 458 | 457 | 472 | 475 | 479 | 455 |
| $(T_{1t}-T_s)$ - $^oF$ | 510 | 526 | 526 | 530 | 532 | 505 |
| $(T_{1m}-T_s)$ - $^oF$ | 484 | 492 | 499 | 503 | 506 | 480 |
| $(T_{2b}-T_s)$ - $^oF$ | 540 | 530 | 547 | 547 | 552 | 531 |
| $(T_{2t}-T_s)$ - $^oF$ | 596 | 608 | 608 | 612 | 612 | 591 |
| $(T_{2m}-T_s)$ - $^oF$ | 568 | 569 | 578 | 580 | 582 | 561 |
| $(T_{3b}-T_s)$ - $^oF$ | 544 | 539 | 556 | 560 | 565 | 567 |
| $(T_{3t}-T_s)$ - $^oF$ | 600 | 618 | 618 | 622 | 624 | 634 |
| $(T_{3m}-T_s)$ - $^oF$ | 572 | 579 | 587 | 591 | 595 | 601 |
| $(T_{4b}-T_s)$ - $^oF$ | 540 | 539 | 556 | 560 | 565 | 578 |
| $(T_{4t}-T_s)$ - $^oF$ | 598 | 622 | 622 | 629 | 633 | 657 |
| $(T_{4m}-T_s)$ - $^oF$ | 569 | 581 | 589 | 595 | 599 | 618 |
| $(T_{5b}-T_s)$ - $^oF$ | 544 | 549 | 565 | 573 | 577 | 587 |
| $(T_{5t}-T_s)$ - $^oF$ | 598 | 631 | 631 | 639 | 641 | 668 |
| $(T_{5m}-T_s)$ - $^oF$ | 571 | 590 | 598 | 606 | 609 | 628 |

## TABLE OF EXPERIMENTAL RESULTS

| Run No. | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 |
| $I$ - amps. | 265 | 271 | 274 | 268 | 265 | 268 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 15420 | 16200 | 16620 | 15900 | 15400 | 15900 |
| $P$ - psig. | 6.5 | 6.0 | 6.0 | 6.0 | 6.0 | 5.0 |
| $T_s$ - $^{\circ}F$ | 139 | 137 | 137 | 137 | 137 | 134 |
| $T_f$ - $^{\circ}F$ | 122 | 117 | 117 | 117 | 117 | 104 |
| $(T_s-T_f)$ - $^{\circ}F$ | 17 | 20 | 20 | 20 | 20 | 30 |
| $(T_{1b}-T_s)$ - $^{\circ}F$ | 519 | 536 | 551 | 546 | 512 | 532 |
| $(T_{1t}-T_s)$ - $^{\circ}F$ | 579 | 594 | 611 | 604 | 572 | 592 |
| $(T_{1m}-T_s)$ - $^{\circ}F$ | 549 | 565 | 581 | 575 | 542 | 562 |
| $(T_{2b}-T_s)$ - $^{\circ}F$ | 600 | 611 | 626 | 617 | 585 | 593 |
| $(T_{2t}-T_s)$ - $^{\circ}F$ | 660 | 670 | 687 | 679 | 645 | 656 |
| $(T_{2m}-T_s)$ - $^{\circ}F$ | 630 | 641 | 657 | 648 | 615 | 625 |
| $(T_{3b}-T_s)$ - $^{\circ}F$ | 605 | 617 | 630 | 623 | 594 | 622 |
| $(T_{3t}-T_s)$ - $^{\circ}F$ | 664 | 679 | 691 | 685 | 653 | 694 |
| $(T_{3m}-T_s)$ - $^{\circ}F$ | 635 | 648 | 661 | 654 | 624 | 658 |
| $(T_{4b}-T_s)$ - $^{\circ}F$ | 605 | 617 | 630 | 623 | 598 | 639 |
| $(T_{4t}-T_s)$ - $^{\circ}F$ | 664 | 687 | 700 | 691 | 666 | 720 |
| $(T_{4m}-T_s)$ - $^{\circ}F$ | 635 | 652 | 665 | 657 | 632 | 680 |
| $(T_{5b}-T_s)$ - $^{\circ}F$ | 605 | 625 | 641 | 632 | 611 | 650 |
| $(T_{5t}-T_s)$ - $^{\circ}F$ | 664 | 691 | 704 | 700 | 679 | 732 |
| $(T_{5m}-T_s)$ - $^{\circ}F$ | 635 | 658 | 673 | 666 | 645 | 691 |

(Con't)

## TABLE OF EXPERIMENTAL RESULTS

| Run No. | 13 | 14 | 15 | 16 | 17 | 18 |
|---|---|---|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 |
| I - amps. | 265 | 265 | 277 | 280 | 283 | 277 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 15500 | 15420 | 17270 | 17820 | 18150 | 17300 |
| P - psig. | 4.5 | 3.5 | 6.0 | 5.0 | 5.0 | 5.0 |
| $T_s$ - °F | 133 | 130 | 137 | 134 | 134 | 134 |
| $T_f$ - °F | 102 | 84 | 112 | 102 | 98 | 100 |
| $(T_s - T_f)$ - °F | 31 | 46 | 25 | 32 | 36 | 34 |
| $(T_{1b} - T_s)$ - °F | 516 | 485 | 598 | 605 | 597 | 584 |
| $(T_{1t} - T_s)$ - °F | 574 | 536 | 657 | 665 | 656 | 648 |
| $(T_{1m} - T_s)$ - °F | 545 | 511 | 628 | 635 | 627 | 616 |
| $(T_{2b} - T_s)$ - °F | 585 | 566 | 662 | 656 | 650 | 643 |
| $(T_{2t} - T_s)$ - °F | 644 | 635 | 725 | 720 | 716 | 707 |
| $(T_{2m} - T_s)$ - °F | 615 | 601 | 694 | 688 | 683 | 675 |
| $(T_{3b} - T_s)$ - °F | 623 | 609 | 666 | 686 | 688 | 682 |
| $(T_{3t} - T_s)$ - °F | 696 | 681 | 729 | 760 | 764 | 758 |
| $(T_{3m} - T_s)$ - °F | 660 | 645 | 698 | 723 | 726 | 720 |
| $(T_{4b} - T_s)$ - °F | 636 | 626 | 666 | 701 | 701 | 695 |
| $(T_{4t} - T_s)$ - °F | 720 | 711 | 743 | 791 | 791 | 787 |
| $(T_{4m} - T_s)$ - °F | 678 | 669 | 705 | 746 | 746 | 741 |
| $(T_{5b} - T_s)$ - °F | 644 | 643 | 683 | 715 | 713 | 711 |
| $(T_{5t} - T_s)$ - °F | 729 | 732 | 755 | 800 | 805 | 796 |
| $(T_{5m} - T_s)$ - °F | 687 | 688 | 719 | 758 | 759 | 754 |

(Con't)

## TABLE OF EXPERIMENTAL RESULTS

| Run No. | 19 | 20 | 21 | 22 | 23 | 24 |
|---|---|---|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 |
| $I$ - amps. | 277 | 295 | 295 | 250 | 262 | 265 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 17070 | 19780 | 19600 | 13220 | 14880 | 15290 |
| $P$ - psig. | 3.0 | 7.5 | 7.5 | 6.0 | 6.5 | 6.5 |
| $T_s$ - $^oF$ | 127 | 141 | 141 | 137 | 138 | 138 |
| $T_f$ - $^oF$ | 77 | 121 | 121 | 123 | 123 | 123 |
| $(T_s-T_f)$ - $^oF$ | 50 | 20 | 20 | 14 | 15 | 15 |
| $(T_{1b}-T_s)$ - $^oF$ | 509 | 619 | 603 | 460 | 503 | 507 |
| $(T_{1t}-T_s)$ - $^oF$ | 552 | 679 | 658 | 512 | 563 | 567 |
| $(T_{1m}-T_s)$ - $^oF$ | 531 | 649 | 631 | 486 | 533 | 537 |
| $(T_{2b}-T_s)$ - $^oF$ | 608 | 698 | 687 | 514 | 563 | 567 |
| $(T_{2t}-T_s)$ - $^oF$ | 680 | 764 | 753 | 576 | 627 | 635 |
| $(T_{2m}-T_s)$ - $^oF$ | 644 | 731 | 720 | 545 | 595 | 601 |
| $(T_{3b}-T_s)$ - $^oF$ | 659 | 692 | 675 | 506 | 563 | 567 |
| $(T_{3t}-T_s)$ - $^oF$ | 739 | 746 | 725 | 564 | 627 | 635 |
| $(T_{3m}-T_s)$ - $^oF$ | 699 | 719 | 700 | 535 | 595 | 601 |
| $(T_{4b}-T_s)$ - $^oF$ | 684 | 687 | 666 | 512 | 576 | 580 |
| $(T_{4t}-T_s)$ - $^oF$ | 773 | 746 | 721 | 568 | 635 | 644 |
| $(T_{4m}-T_s)$ - $^oF$ | 729 | 717 | 694 | 540 | 606 | 612 |
| $(T_{5b}-T_s)$ - $^oF$ | 706 | 683 | 658 | 521 | 582 | 588 |
| $(T_{5t}-T_s)$ - $^oF$ | 798 | 742 | 713 | 572 | 640 | 646 |
| $(T_{5m}-T_s)$ - $^oF$ | 752 | 713 | 686 | 547 | 611 | 617 |

(Con't)

## TABLE OF EXPERIMENTAL RESULTS

| Run No. | 25 | 26 | 27 | 28 | 29 | 30 |
|---|---|---|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 | 1.85 |
| I - amps. | 265 | 271 | 265 | 271 | 241 | 289 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 15210 | 16020 | 15250 | 16210 | 12220 | 18800 |
| P - psig. | 6.5 | 6.5 | 6.5 | 6.5 | 3.0 | 3.0 |
| $T_s$ - $^oF$ | 138 | 138 | 138 | 138 | 125 | 125 |
| $T_f$ - $^oF$ | 108 | 104 | 102 | 123 | 110 | 73 |
| $(T_s-T_f)$ - $^oF$ | 30 | 34 | 36 | 15 | 15 | 52 |
| $(T_{1b}-T_s)$ - $^oF$ | 455 | 473 | 446 | 539 | 433 | 529 |
| $(T_{1t}-T_s)$ - $^oF$ | 528 | 528 | 503 | 597 | 485 | 593 |
| $(T_{1m}-T_s)$ - $^oF$ | 492 | 501 | 475 | 568 | 459 | 561 |
| $(T_{2b}-T_s)$ - $^oF$ | 520 | 533 | 524 | 595 | 503 | 644 |
| $(T_{2t}-T_s)$ - $^oF$ | 593 | 610 | 593 | 661 | 556 | 716 |
| $(T_{2m}-T_s)$ - $^oF$ | 557 | 572 | 559 | 628 | 530 | 680 |
| $(T_{3b}-T_s)$ - $^oF$ | 554 | 576 | 563 | 608 | 511 | 682 |
| $(T_{3t}-T_s)$ - $^oF$ | 635 | 656 | 644 | 673 | 556 | 758 |
| $(T_{3m}-T_s)$ - $^oF$ | 595 | 616 | 604 | 641 | 534 | 720 |
| $(T_{4b}-T_s)$ - $^oF$ | 588 | 610 | 597 | 622 | 515 | 712 |
| $(T_{4t}-T_s)$ - $^oF$ | 669 | 690 | 682 | 690 | 571 | 800 |
| $(T_{4m}-T_s)$ - $^oF$ | 629 | 650 | 640 | 656 | 543 | 756 |
| $(T_{5b}-T_s)$ - $^oF$ | 610 | 631 | 618 | 637 | 535 | 746 |
| $(T_{5t}-T_s)$ - $^oF$ | 694 | 716 | 707 | 703 | 593 | 839 |
| $(T_{5m}-T_s)$ - $^oF$ | 652 | 674 | 663 | 670 | 564 | 793 |

## TABLE OF EXPERIMENTAL RESULTS

| Run No. | 31 | 32 | 33 |
|---|---|---|---|
| $V_o$ - ft./sec. | 1.85 | 1.23 | 1.23 |
| $I$ - amps. | 289 | 283 | 283 |
| $\overline{(q/A)}$ - Btu./hr.ft.$^2$ | 19050 | 18000 | 17660 |
| $P$ - psig. | 3.0 | 5.5 | 5.5 |
| $T_s$ - °F | 125 | 136 | 136 |
| $T_f$ - °F | 102 | 130 | 90 |
| $(T_s - T_f)$ - °F | 23 | 6 | 46 |
| $(T_{1b} - T_s)$ - °F | 593 | 586 | 461 |
| $(T_{1t} - T_s)$ - °F | 657 | 637 | 509 |
| $(T_{1m} - T_s)$ - °F | 625 | 612 | 485 |
| $(T_{2b} - T_s)$ - °F | 686 | 684 | 556 |
| $(T_{2t} - T_s)$ - °F | 750 | 751 | 635 |
| $(T_{2m} - T_s)$ - °F | 718 | 718 | 596 |
| $(T_{3b} - T_s)$ - °F | 720 | 692 | 603 |
| $(T_{3t} - T_s)$ - °F | 784 | 751 | 686 |
| $(T_{3m} - T_s)$ - °F | 752 | 722 | 645 |
| $(T_{4b} - T_s)$ - °F | 741 | 671 | 650 |
| $(T_{4t} - T_s)$ - °F | 822 | 730 | 747 |
| $(T_{4m} - T_s)$ - °F | 782 | 701 | 699 |
| $(T_{5b} - T_s)$ - °F | 767 | 646 | 701 |
| $(T_{5t} - T_s)$ - °F | 847 | 701 | 803 |
| $(T_{5m} - T_s)$ - °F | 807 | 674 | 752 |

Figure 1 - Film boiling of Freon-113 in a
horizontal tube with $V_O$ = 1.85 ft/sec.
The flow is from left to right.

heating water supply

control valves

pressure gage

preheater

to drain

thermocouple

rotameter

control valve

control valves

quantitative
and visual
test sections

control
valves

pump

motor

relief valve

coolant supply

control valves

thermocouple

pressure
control valve

degasser

condenser

liquid reservoir

to drain
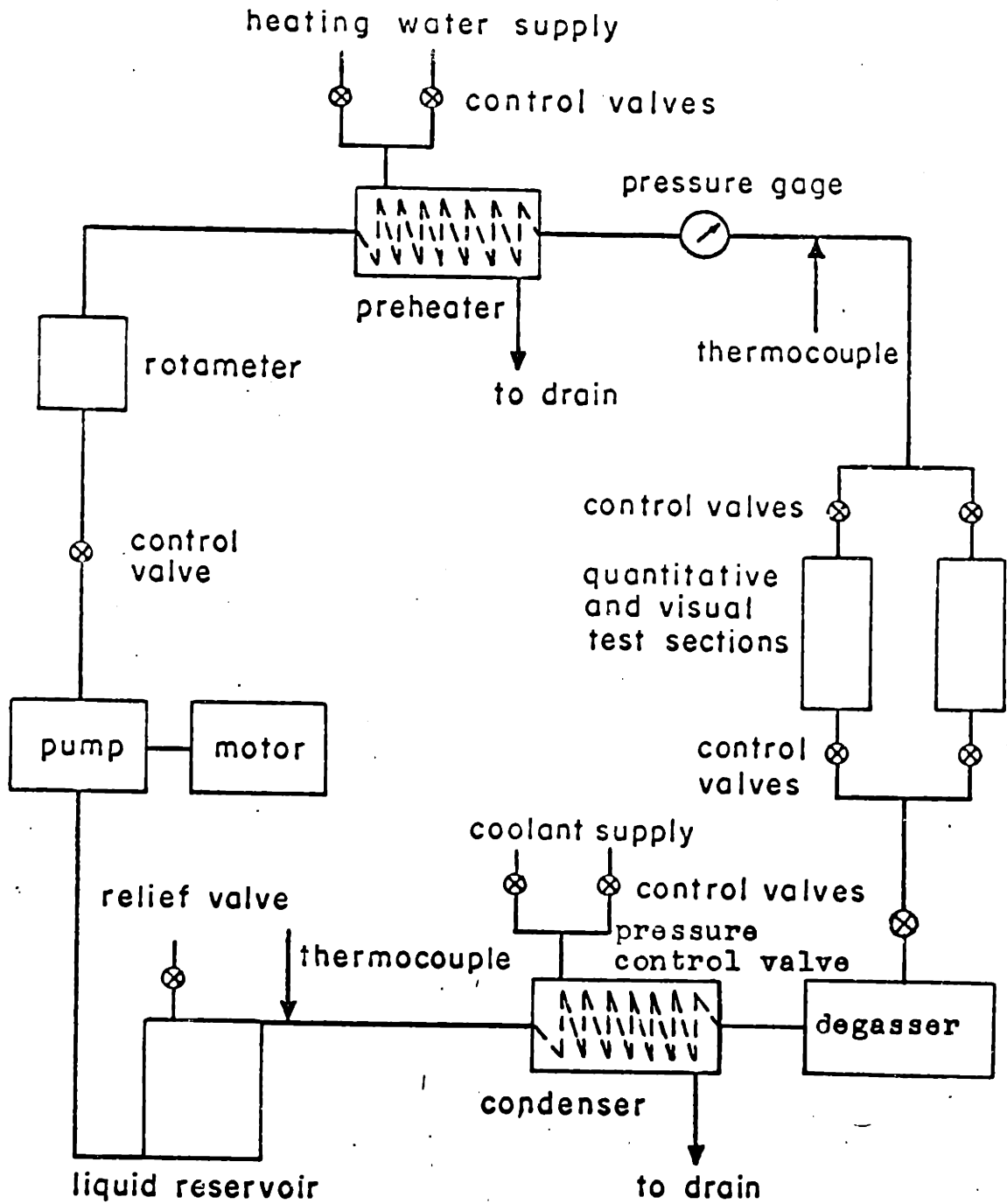
FIG. 2 SCHEMATIC DIAGRAM OF THE CIRCULATION
SYSTEM

9"  — heated section

liquid supply

liquid return

voltmeter   ammeter

+ power supply

power regulator

−

conducting glass tubing
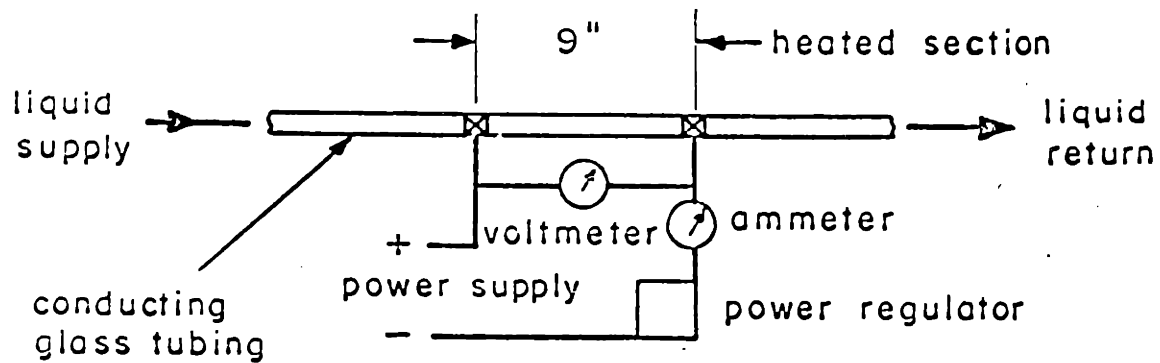
Tube I. D. = 0.418 inches,  L = 48 inches

Fig. 3  SCHEMATIC DIAGRAM OF THE VISUAL TEST SECTION



thermocouples

insulation

insulation

liquid supply

liquid return

Guard heater sections

N.B.S. shunt

power regulator

+   −

+   power supply   −
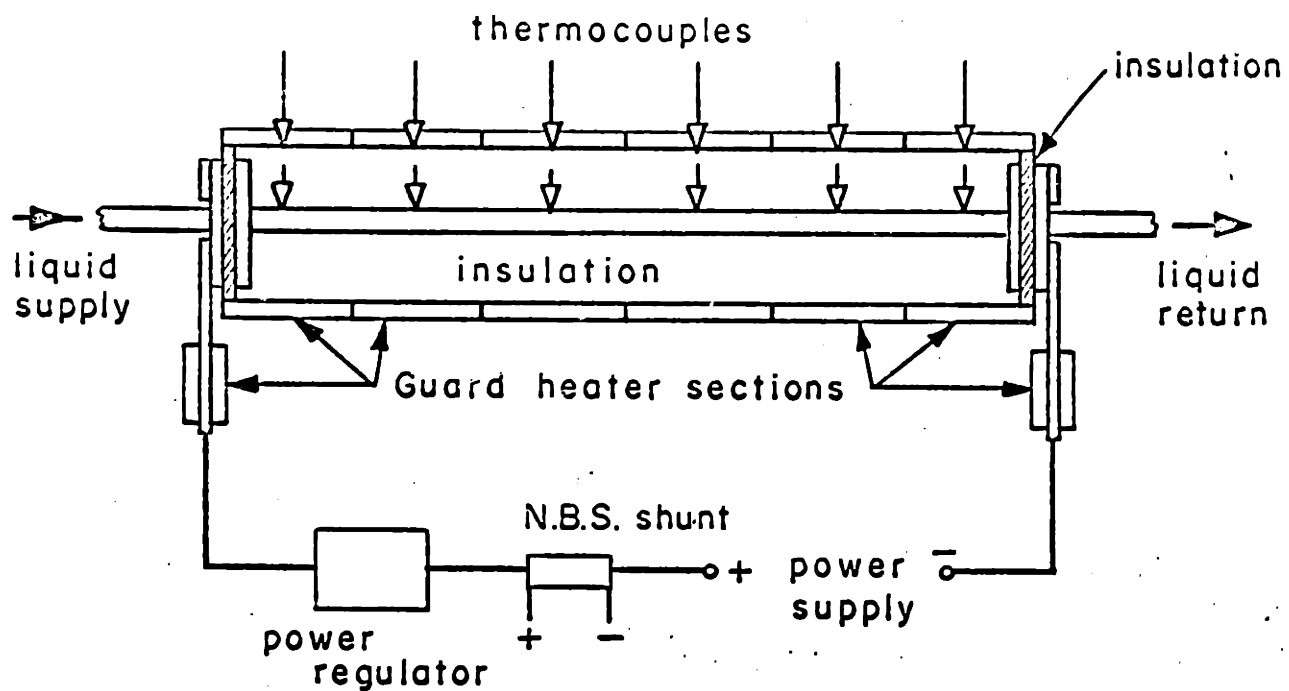
FIG. 4 SCHEMATIC DIAGRAM OF THE QUANTITATIVE TEST SECTION

Figure 5 – Apparatus control panel and the
visual test section
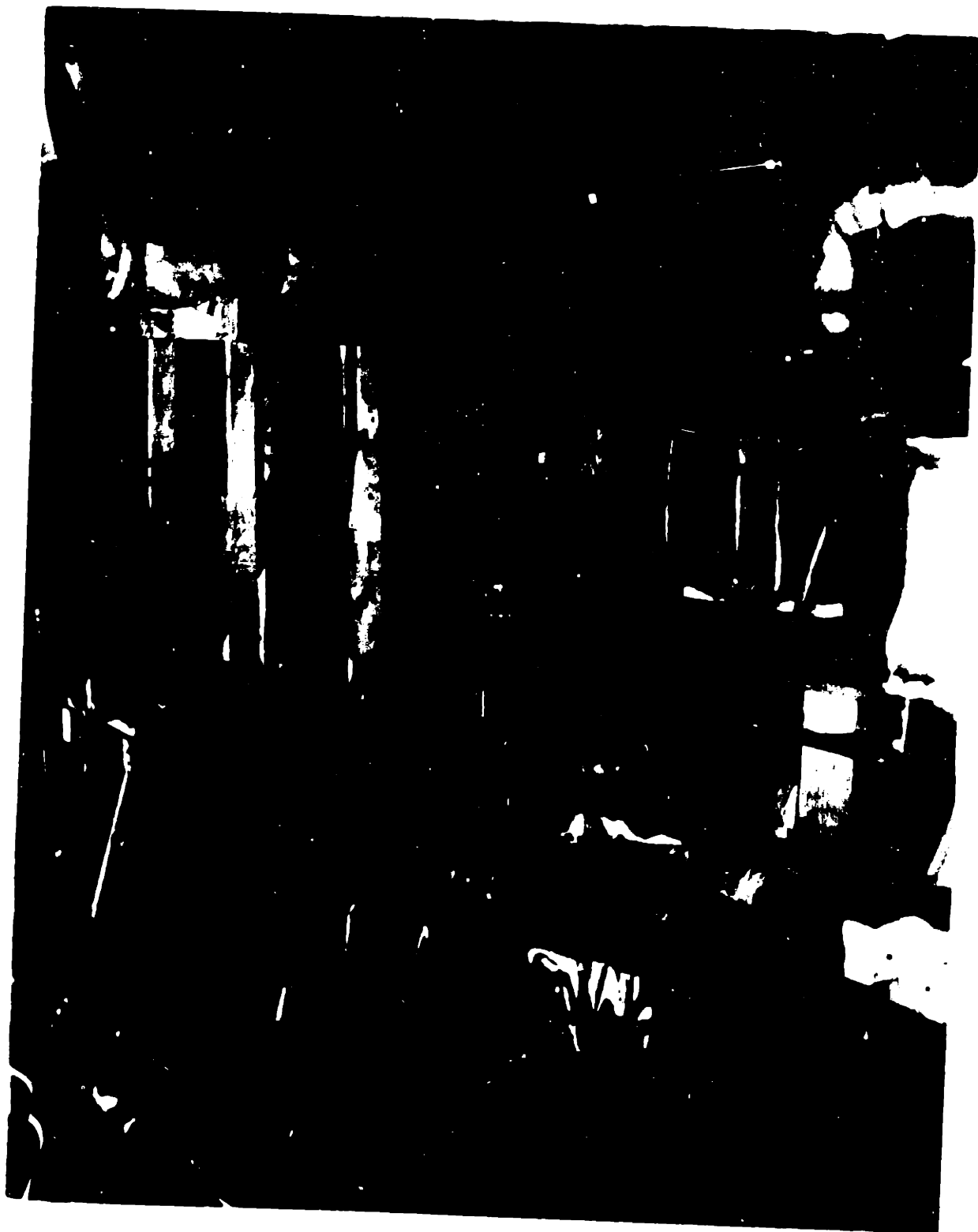
Figure 6 - Quantitative test section and parts
of the control and circulation systems.
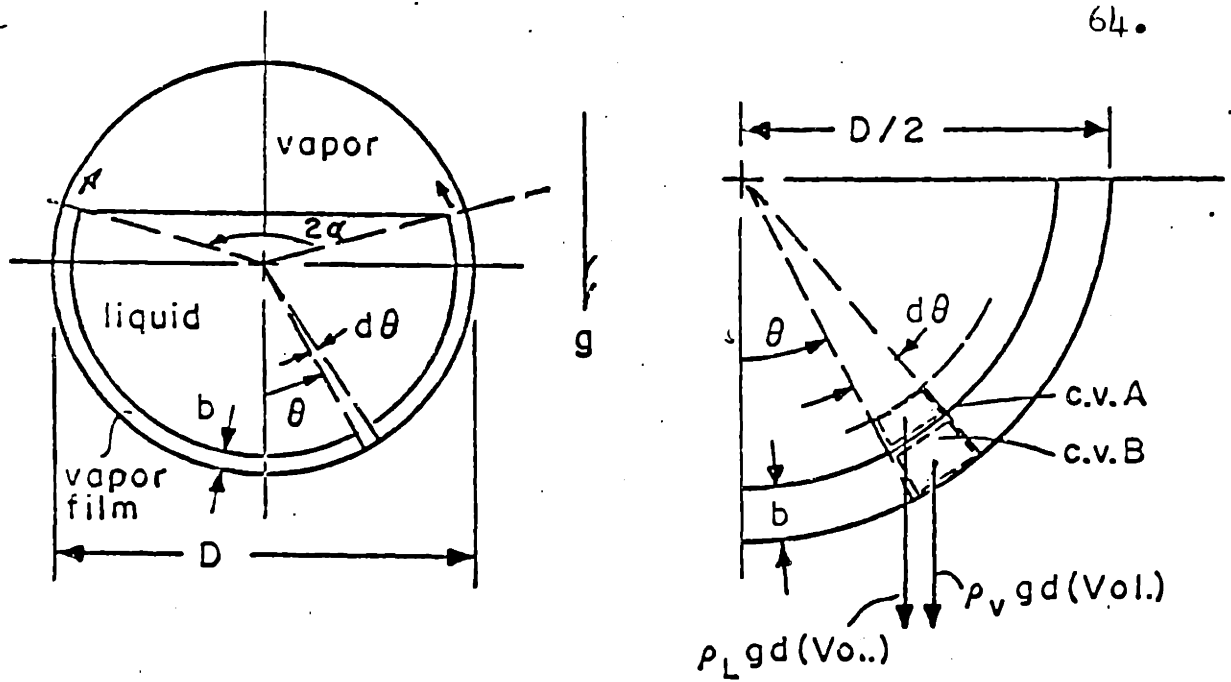
FIG. 7 CROSS SECTION OF IDEALIZED FLOW MODEL
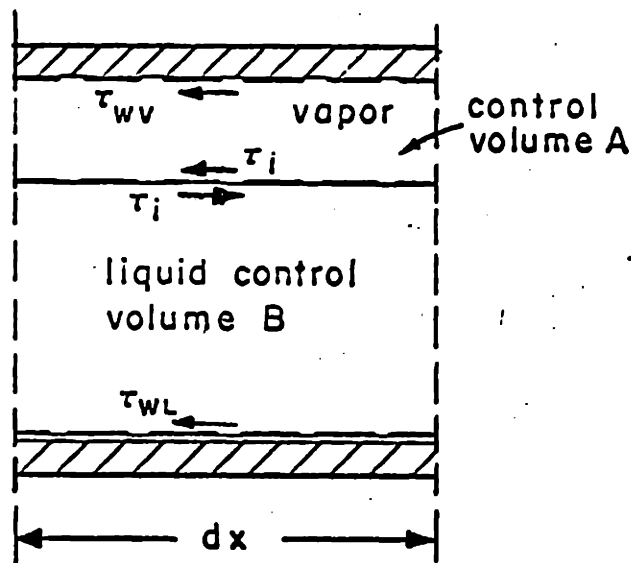


FIG. 8 GEOMETRY FOR
MOMENTUM
ANALYSIS
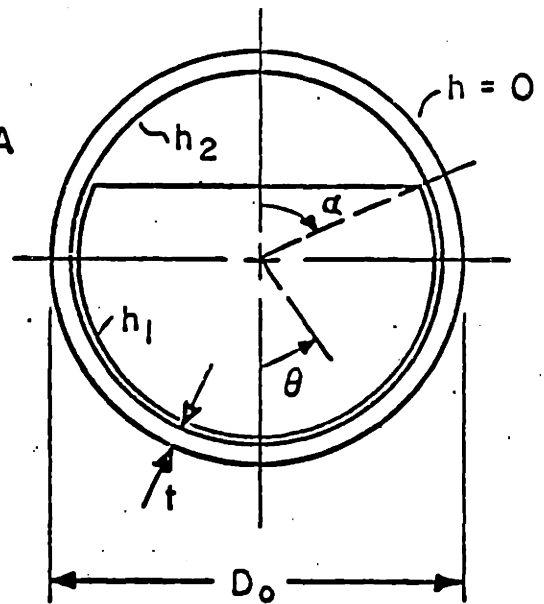
FIG. 9 GEOMETRY FOR
CONDUCTION
ANALYSIS

FIG. II  TUBE LENGTH - FLOW
RATE PARAMETER
vs. $\alpha/\pi$
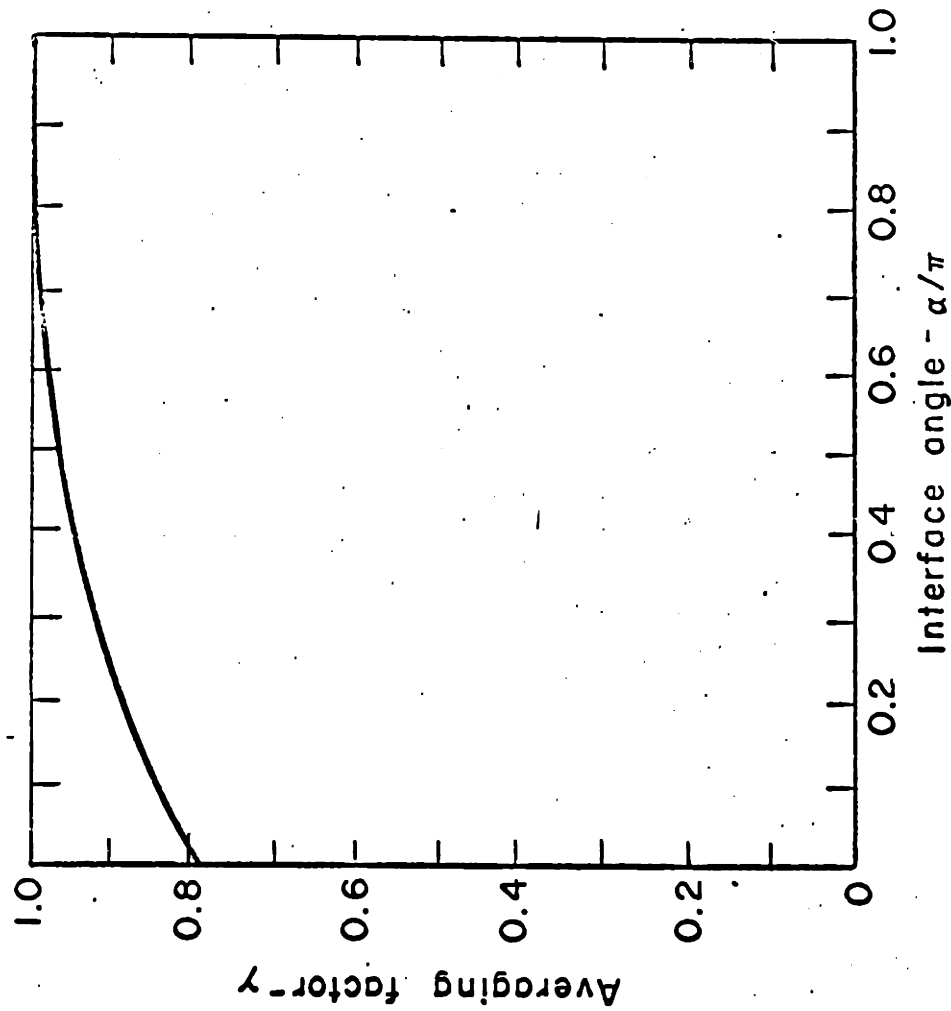


FIG. 10  BOILING HEAT TRANSFER COEFFICIENT
MULTIPLIER, $\gamma$, vs. $\alpha/\pi$

**Top and Bottom Tube Wall Temperatures**

$$V_o = 1.85 \text{ ft/sec}$$

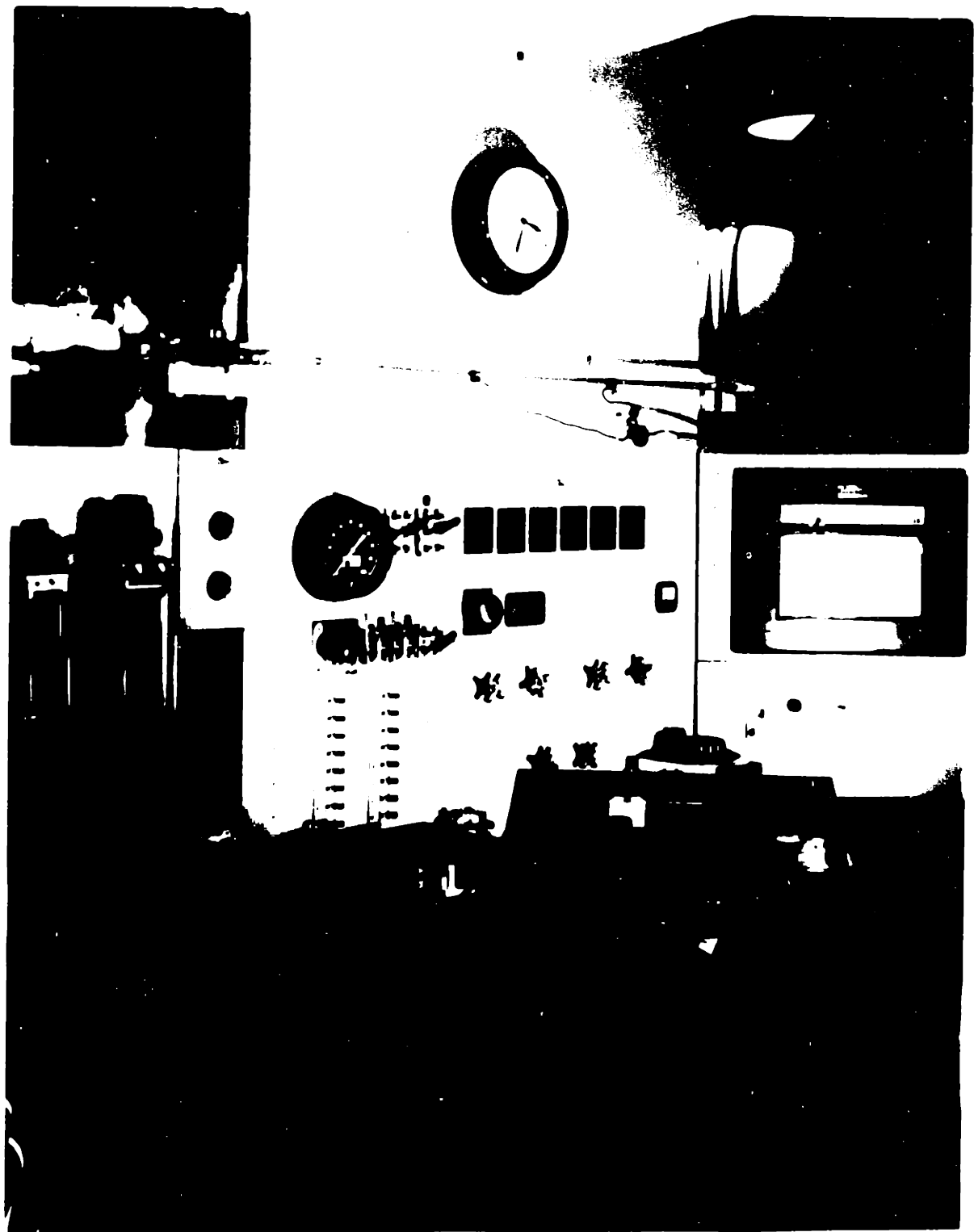$$\overline{q/A} = 15420 \text{ Btu/hr-ft}^2$$

Experiment

⬡ ○ - Top - $(T_s - T_f) = 46°F$
⬡ - Bottom - $(T_s - T_f) = 46°F$
□ - Top - $(T_s - T_f) = 17°F$
△ - Bottom - $(T_s - T_f) = 17°F$

Theory

A - Top - $(T_s - T_f) = 50°F$
B - Bottom - $(T_s - T_f) = 50°F$
C - Top - $(T_s - T_f) = 0°F$
D - Bottom - $(T_s - T_f) = 0°F$

FIG. 12 TEMPERATURE DISTRIBUTION WITH X/L

FIG. 13 TEMPERATURE DISTRIBUTION WITH X/L

Top

Bottom

Top

Bottom

Top and Bottom Tube Wall Temperatures

$V_o = 1.23$ ft/sec

Experiment

Broken Line - $\overline{q/A} = 17660$ Btu/hr-ft$^2$
 $(T_s - T_f) = 46°F$

Solid Line - $\overline{q/A} = 18000$ Btu/hr-ft$^2$
 $(T_s - T_f) = 6°F$

Temperature difference - $(T_{wall} - T_{sat})$ - $°F$

Distance from inlet end - X/L
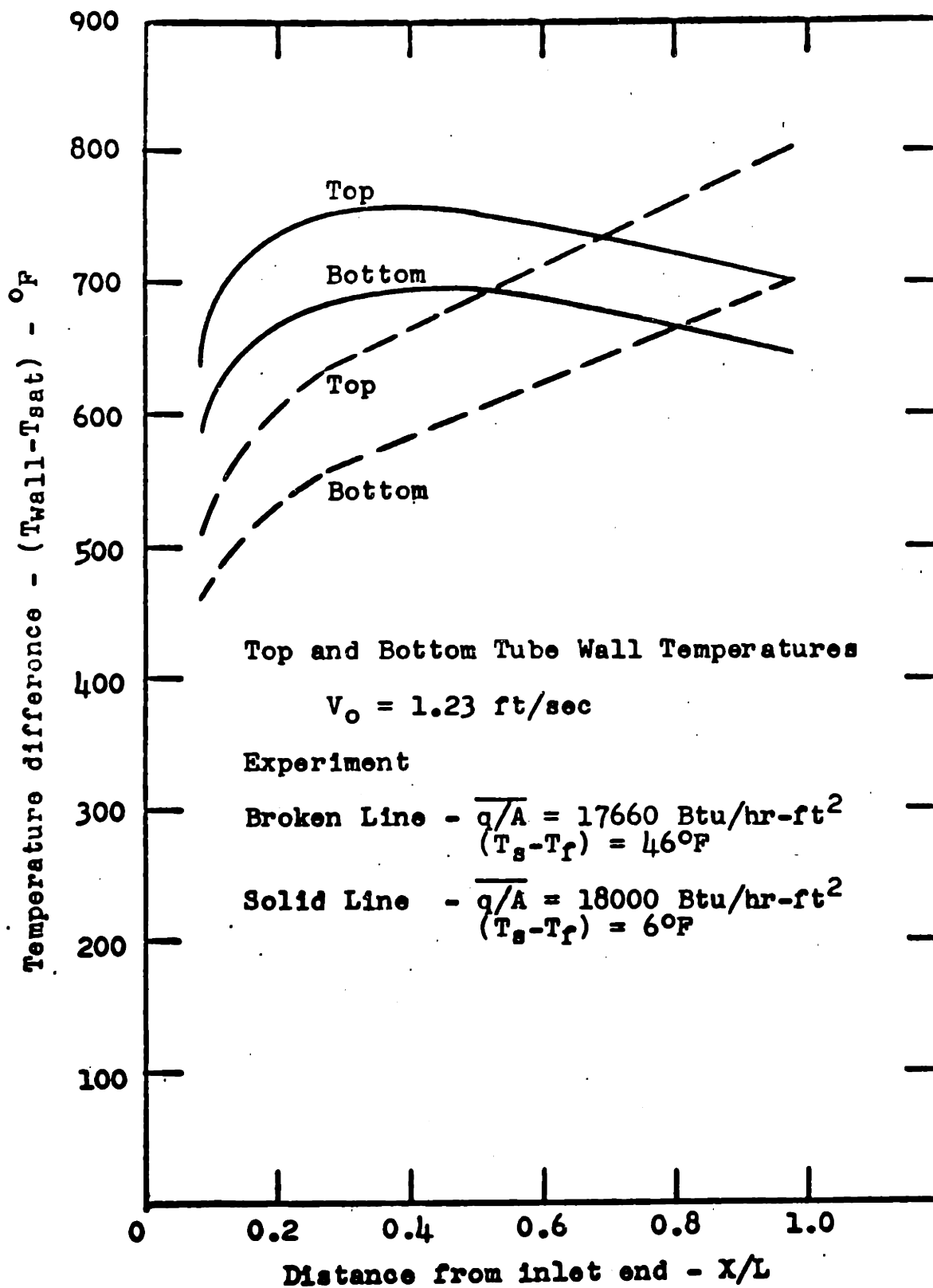
FIG. 14 TEMPERATURE DISTRIBUTION WITH X/L

FIG. 15 TEMPERATURE DISTRIBUTION WITH X/L

Temperature difference - $(T_{wall}-T_{sat})$ - °F

900

800 — A

700 — B
— C
— D
— E

600

**Mean Tube Wall Temperatures**

$V_o = 1.85$ ft/sec

$\overline{q/A}$ approximately 15420 Btu/hr-ft$^2$

500

**Broken Line - Theory**

A - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
$(T_s-T_f) = 50$°F

400

C - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
$(T_s-T_f) = 0$°F

300

**Solid Line - Experiment**

B - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
$(T_s-T_f) = 46$°F

200

D - $\overline{q/A} = 15400$ Btu/hr-ft$^2$
$(T_s-T_f) = 20$°F

100

E - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
$(T_s-T_f) = 17$°F

0    0.2    0.4    0.6    0.8    1.0

Distance from inlet end - X/L

**FIG. 16  TEMPERATURE DISTRIBUTION WITH X/L**

Temperature difference - ($T_{wall}-T_{sat}$) - °F

900

800

700 — — — — — — — — — — A
                                    B

— — — — — — — — — — C
                                    D
600                                 E
                                    F
                                    G
                                    H
500                                 I
                                    J
                                    K

Mean Tube Wall Temperatures

$V_o = 1.85$ ft/sec

($T_s-T_f$) approximately 15°F

|  |  | $\overline{q/A}$ - Btu/hr-ft$^2$ | ($T_s-T_f$)-°F |
|---|---|---|---|
| Theory | A | 17170 | 0 |
|  | C | 15420 | 0 |
|  | K | 12000 | 0 |
| Experiment | B | 16210 | 15 |
|  | D | 15420 | 17 |
|  | E | 15290 | 15 |
|  | F | 14880 | 15 |
|  | G | 13100 | 17 |
|  | H | 13680 | 13 |
|  | I | 12220 | 15 |
|  | J | 13220 | 14 |

400

300

200

100

0    0.2    0.4    0.6    0.8    1.0

Distance from inlet end - X/L
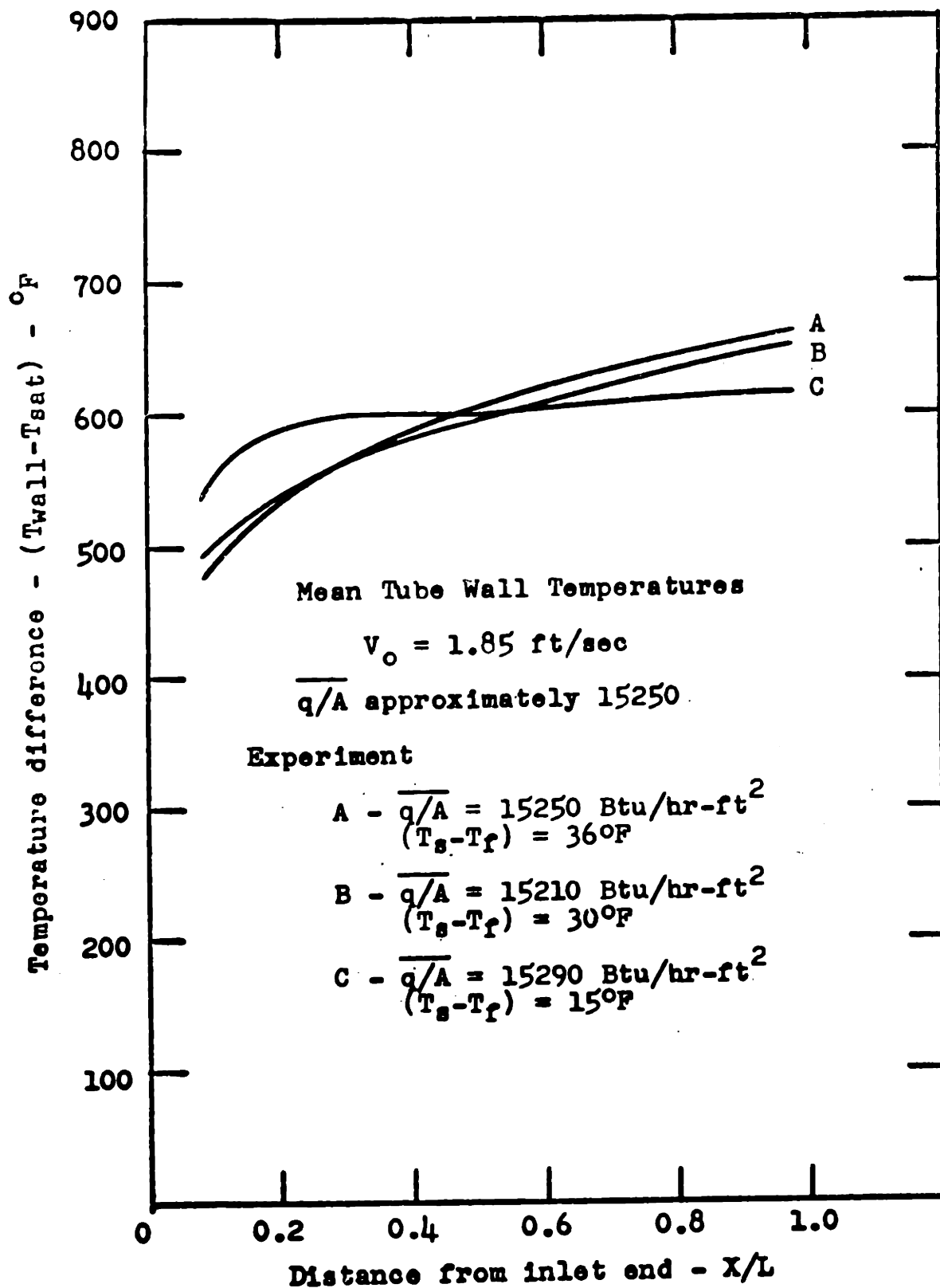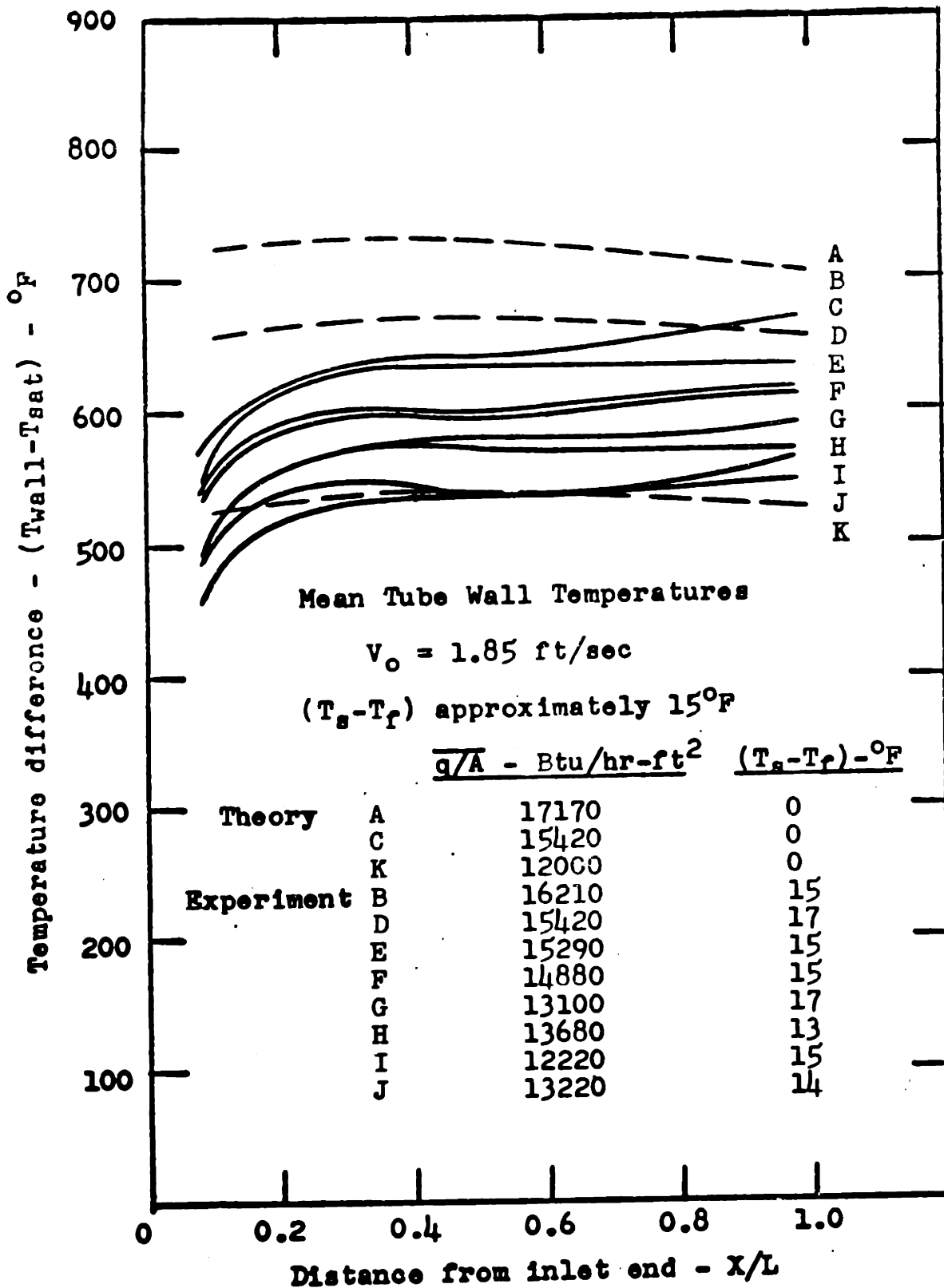
FIG. 17  TEMPERATURE DISTRIBUTION WITH X/L

FIG. 18 TEMPERATURE DISTRIBUTION WITH X/L

Mean Tube Wall Temperatures

$V_o = 1.85$ ft/sec

$(T_s - T_f)$ approximately $25°F$

Experiment

A - $\overline{q/A} = 19050$ Btu/hr-ft$^2$
$(T_s - T_f) = 23°F$

B - $\overline{q/A} = 17270$ Btu/hr-ft$^2$
$(T_s - T_f) = 25°F$

C - $\overline{q/A} = 14150$ Btu/hr-ft$^2$
$(T_s - T_f) = 28°F$
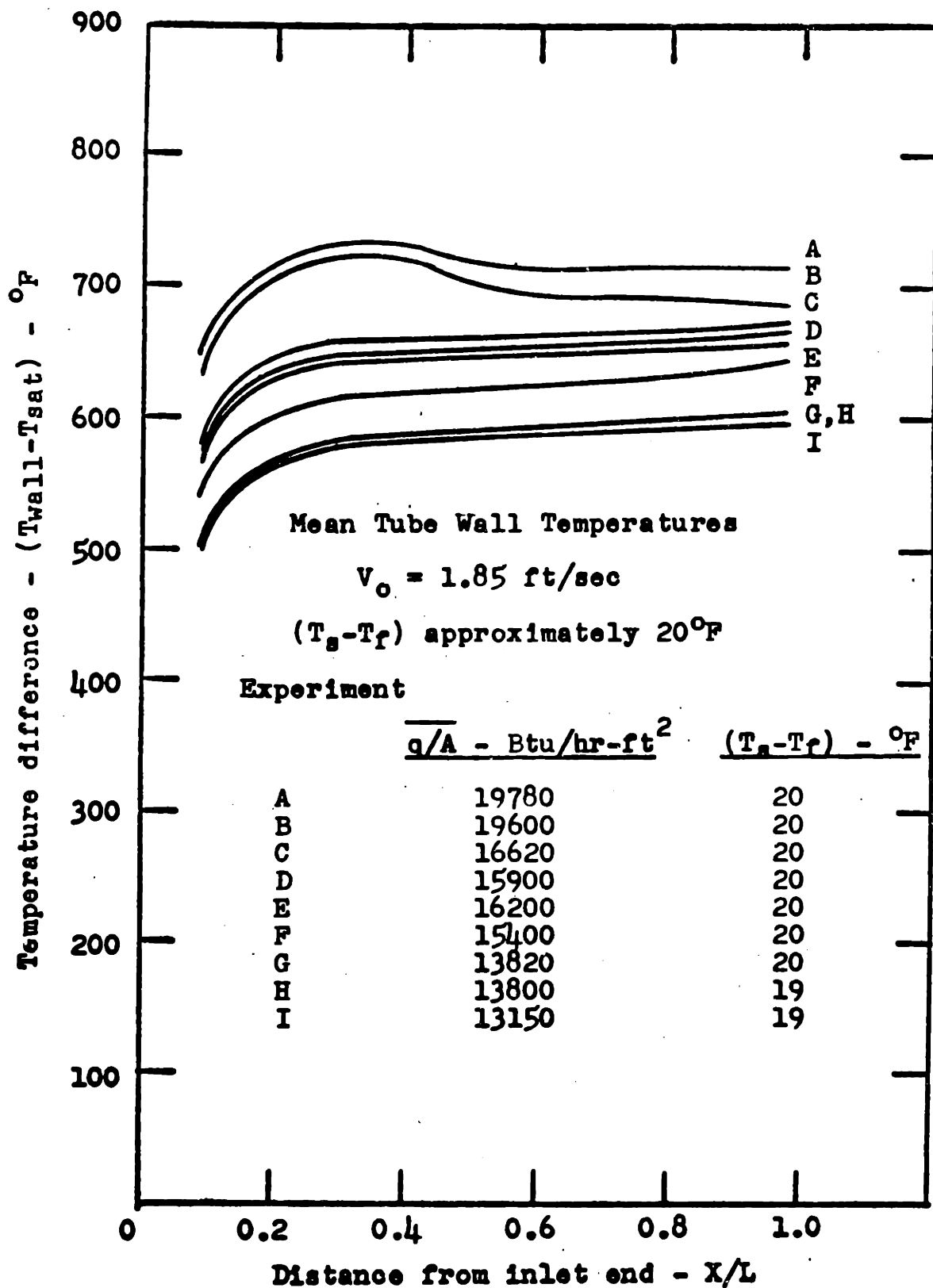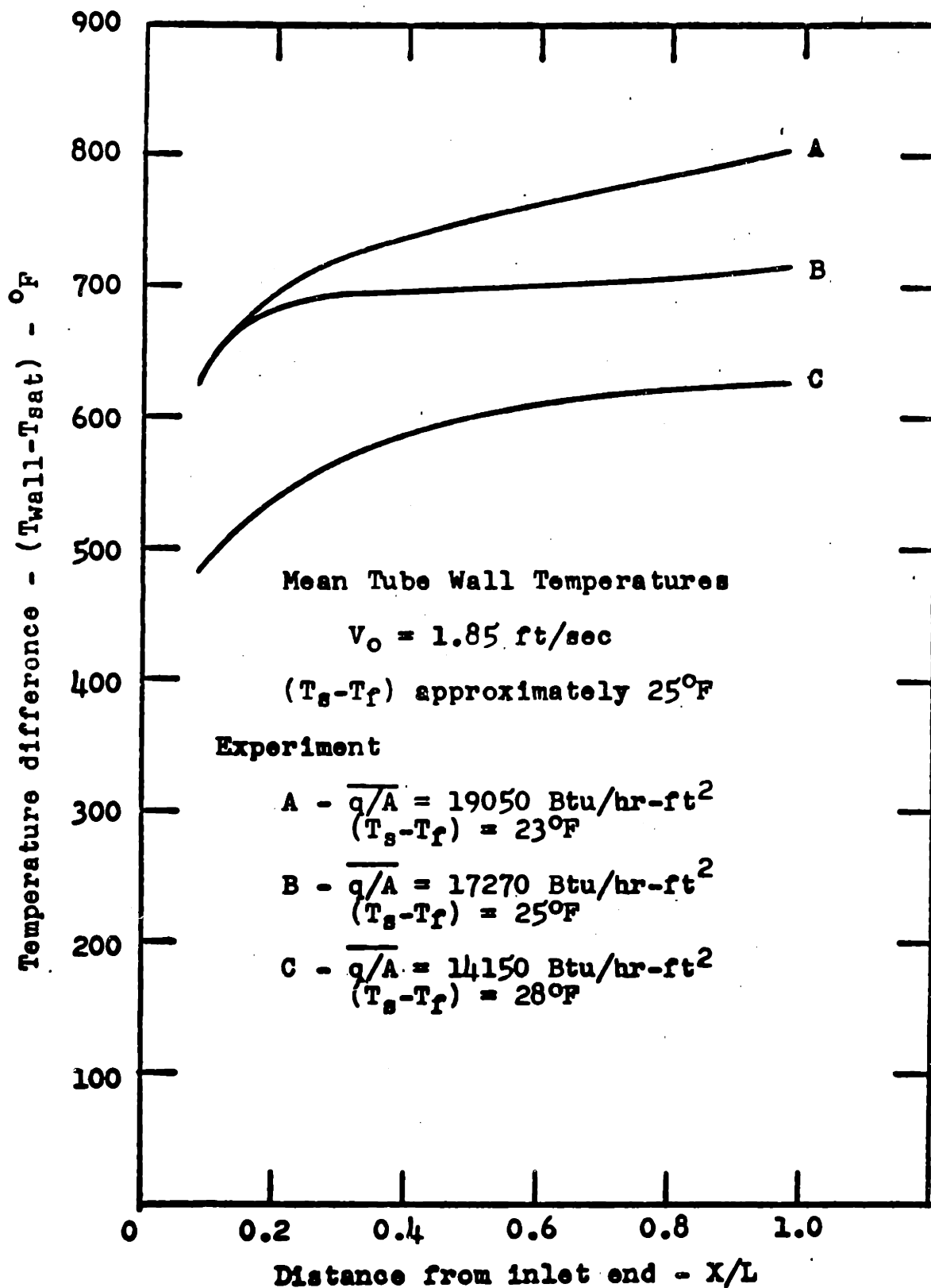
FIG. 19 TEMPERATURE DISTRIBUTION WITH X/L

FIG. 20 TEMPERATURE DISTRIBUTION WITH X/L

Mean Tube Wall Temperatures

$V_0 = 1.85$ ft/sec

$(T_s - T_f)$ approximately $30°F$

Experiment

A − $\overline{q/A} = 17820$ Btu/hr-ft$^2$
$(T_s - T_f) = 32°F$

B − $\overline{q/A} = 15900$ Btu/hr-ft$^2$
$(T_s - T_f) = 30°F$

C − $\overline{q/A} = 15500$ Btu/hr-ft$^2$
$(T_s - T_f) = 31°F$

D − $\overline{q/A} = 15210$ Btu/hr-ft$^2$
$(T_s - T_f) = 30°F$

Mean Tube Wall Temperatures

$V_o$ = 1.85 ft/sec

$(T_s - T_f)$ approximately 35°F

Experiment

A - $\overline{q/A}$ = 18150 Btu/hr-ft$^2$
$(T_s - T_f)$ = 36°F

B - $\overline{q/A}$ = 17300 Btu/hr-ft$^2$
$(T_s - T_f)$ = 34°F

C - $\overline{q/A}$ = 16020 Btu/hr-ft$^2$
$(T_s - T_f)$ = 34°F

D - $\overline{q/A}$ = 15250 Btu/hr-ft$^2$
$(T_s - T_f)$ = 36°F

Distance from inlet end - X/L

FIG. 21 TEMPERATURE DISTRIBUTION WITH X/L

Temperature difference - $(T_{wall}-T_{sat})$ - °F

900

800 — A

— B
— C

700
— D

600

500

**Mean Tube Wall Temperatures**

$V_o = 1.85$ ft/sec

$(T_s-T_f)$ approximately 50°F

400

**Broken Line - Theory**

A - $\overline{q/A} = 17170$ Btu/hr-ft$^2$
     $(T_s-T_f) = 50°F$

300

B - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
     $(T_s-T_f) = 50°F$

**Solid Line - Experiment**

200

C - $\overline{q/A} = 17070$ Btu/hr-ft$^2$
     $(T_s-T_f) = 50°F$

D - $\overline{q/A} = 15420$ Btu/hr-ft$^2$
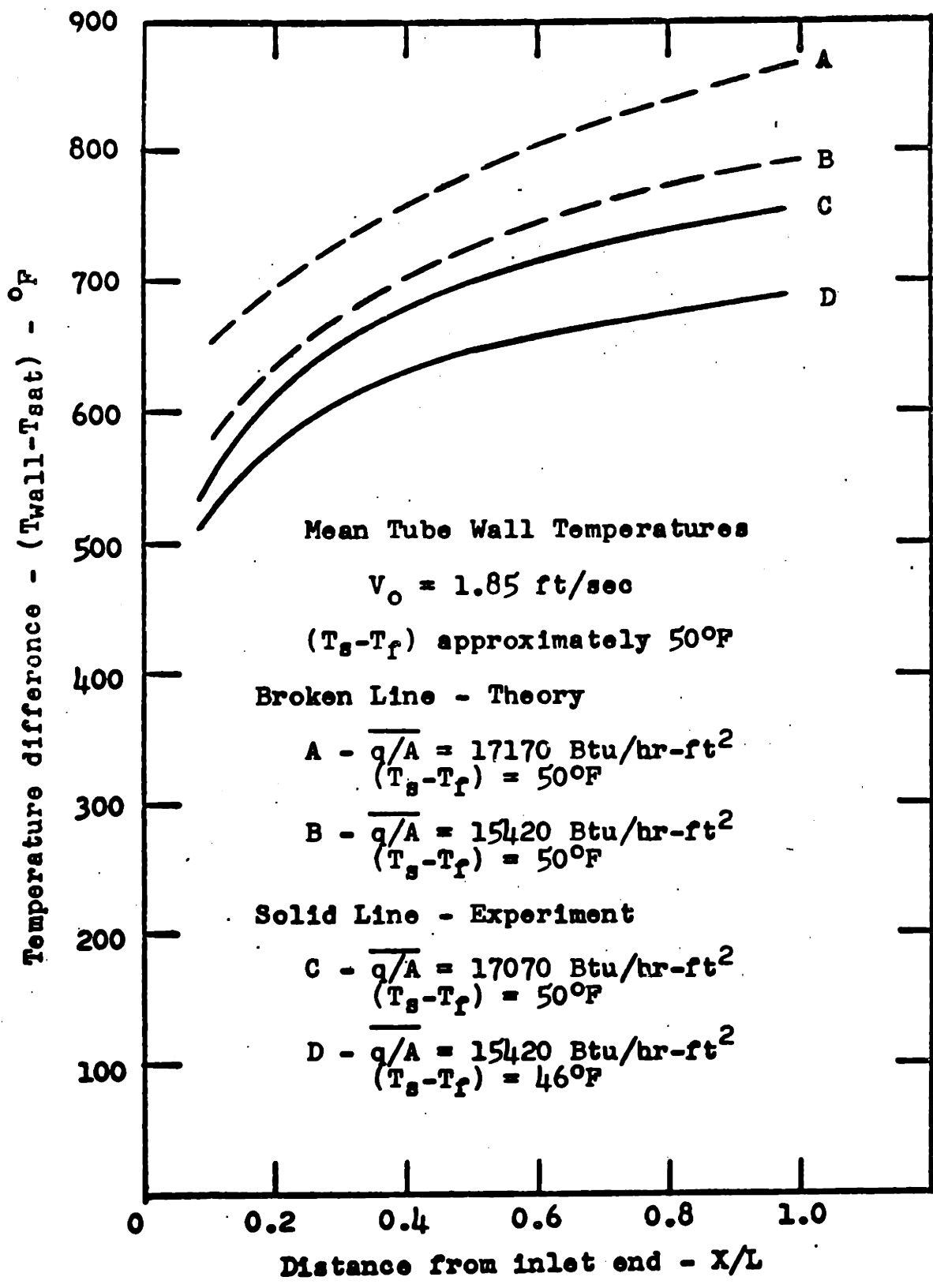100  $(T_s-T_f) = 46°F$

0    0.2    0.4    0.6    0.8    1.0

Distance from inlet end - X/L

**FIG. 22  TEMPERATURE DISTRIBUTION WITH X/L**