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**LA THÈSE A ÉTÉ
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AN ANALYTICAL AND EXPERIMENTAL INVESTIGATION
OF FORCED CONVECTIVE FILM BOILING

by

André Laperrière

A thesis submitted to the School of Graduate Studies and Research
of the University of Ottawa
in partial fulfillment of the requirements for the degree of
Master of Applied Science (Mechanical Engineering)

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ABSTRACT

Film boiling heat transfer was investigated analytically as well as experimentally. A heat transfer coefficient prediction method was developed in the low quality region; also referred to as inverted annular flow (IAF). The physical model will predict the heat transfer coefficient over a wide range of pressure for mass flux less than $500 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$.

In the experimental study, stable film boiling was established in a directly heated vertical tube with bottom flooding of light water. The data cover a wide range of mass flux ($1000 - 4500 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$) and inlet conditions (250 $\text{kJ} \cdot \text{kg}^{-1}$ subcooling to 0.15 positive thermodynamic equilibrium quality) at 9.6 MPa. This provides a range in local quality from -0.19 to 0.64.

The heat transfer coefficient data trends are examined qualitatively as is the influence of some parameters such as mass flux, inlet subcooling and heat flux. The data were compared with empirical correlations and poor agreement was generally found. Data of the true quench minimum film boiling temperature were also obtained.

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NOMENCLATURE

NOTE: The less frequently referenced symbols are defined when they appear in the text.

Symbol	Description	Units
A	Area	m^2
c_p	Specific heat	$J \cdot kg^{-1} \cdot ^\circ C^{-1}$
D	Diameter	m
f	Friction factor	-
F	View factor	-
g	Acceleration due to gravity	$m \cdot s^{-2}$
h	Heat transfer coefficient	$W \cdot m^{-2} \cdot ^\circ C^{-1}$
H	Enthalpy	$J \cdot kg^{-1}$
k	Thermal conductivity	$W \cdot m^{-1} \cdot ^\circ C^{-1}$
L	Length	m
m	Constant	-
Nu	Nusselt number (hD/k)	-
P, p	Pressure	MPa
P	Perimeter	m
Pr	Prandtl number ($c_p \mu / k$)	-
Q	Power	Watts
Q''	Heat flux	$Watts/m^2$
R, r	Radius	m
Re	Reynolds number (GD/μ)	-
S	Slip ratio (U_v/U_l)	-
T	Temperature	$^\circ C$
U, V	Velocity	m/s
W	Mass flow rate	kg/s
x	radial distance	m
X	Thermodynamic quality	-
y	distance normal to plate	m
z	axial distance	m

GREEK

Symbol	Description	Units
α	Void fraction	-
α	Thermal diffusivity ($k/(\rho c_p)$)	$m^2 \cdot s^{-1}$
γ	Absorptivity of the vapour	-
Γ	Evaporation rate	$kg \cdot m^{-1} \cdot s^{-1}$
δ	Film thickness	m
ϵ	Emissivity	-
μ	Dynamic viscosity	$kg \cdot m^{-1} \cdot s^{-1}$
ν	Kinematic viscosity	$m^2 \cdot s^{-1}$
ϕ	Surface heat flux	$W \cdot m^{-2}$
ρ	Density	$kg \cdot m^{-3}$
σ	Surface tension	$N \cdot m^{-1}$
σ	Stefan-Boltzman constant	$W \cdot m^{-2} \cdot ^\circ K^{-4}$
τ	Shear stress	$N \cdot m^{-2}$
τ	Vapour transmissivity	-

SUBSCRIPTS

Symbol	Description
cond	Conduction
conv	Convection
d	Diameter
e	Equilibrium
f	Saturated liquid
fg	Difference between saturated vapour and saturated liquid value.
g	Saturated vapour
i	Interfacial, inside
in	Inlet
l	Liquid
o	Initial value, outside
rad	Radiation
sat	Saturation
v	Vapour
w	Wall

SUPERScript

Symbol	Description
-	Mean value

I. INTRODUCTION

The coolant in the primary heat transport system of a water cooled nuclear reactor is normally kept at temperature just below saturation or is allowed to attain a small degree of superheat. The heating surface is kept at temperature at or slightly above saturation by the action of the very efficient nucleate boiling heat transfer mode. Therefore, nuclear reactors normally operate under conditions where dryout does not occur. If, due to unforeseen circumstances, the critical heat flux (CHF) is exceeded, the cooling of the fuel cladding proceeds by an inefficient boiling mode: film boiling. This may occur during the course of a loss-of-coolant accident (LOCA) caused by a rupture in the primary coolant system.

Reactor safety analysis requires the prediction of the heat transfer coefficient during the initial blowdown phase and in the subsequent emergency-core-cooling (ECC) phase. The critical heat flux could also be exceeded during a loss of regulation accident (LOR). In such an event, the reactor power accidentally increases and the fuel cladding wall temperatures can be predicted from empirical correlations or from theoretical models.

In this study, post-CHF data were obtained at 9.6 MPa over a wide range of mass flux ($1000 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$ - $4500 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$) and inlet quality. These data have some applicability to Candu reactor safety analysis. The hot patch technique was used in this experiment. A copper block is silvered soldered on a tube with its power controlled separately from the test section. This provides a heat flux sufficient to exceed the

V

critical heat flux (CHF). Consequently, the test section downstream of the hot patch is in the film boiling region.

A theoretical model based on the physical mechanisms of low quality film boiling (also referred as inverted annular) was also developed. This model predicts the surface temperature for a known uniform heat flux in a tube geometry. The physical model predicts reasonably well the surface temperature over a wide range of pressure for mass flux less than $500 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$. The proposed model adequately incorporates the effects of parameters such as pressure, inlet subcooling, void fraction and slip ratio.

II. TWO-PHASE FLOW HEAT TRANSFER

II.1. The boiling curve

The different stages of heat transfer from a surface to a liquid can be understood from the pool boiling curves (Figure II.1). Different regions are considered by reference to this curve in which heat flux is plotted against the surface temperature. The region A to B represents heat transfer without boiling for a single-phase liquid. The heat transfer to the liquid increases the near-wall liquid temperature and bubbles start to appear for a given liquid superheat at the beginning of region B. This is called the subcooled nucleate boiling region. As the subcooled bulk fluid temperature is increased, small bubbles are formed and grow in the cavities on the hot surface. As the surface temperature increases, some of the bubbles detach from the heated surface. More and more nucleation sites will become active resulting from an increase in the heat flux. Point D represents the critical heat flux (CHF) where the liquid supply to the heating surface is restricted by the interaction of liquid and vapour. As the heat flux is raised beyond the CHF, a sudden rise in wall temperature takes place. In region EFG, a vapour blanket covers the heating surface. This region is referred to as the film boiling region. Point E, the minimum film boiling, refers to the condition in which a fluid is insulated from the heated surface by a thin, self-generated vapour film [Snoek, 1972]. Below this temperature, stable film boiling cannot be maintained.

The transition boiling region is characterized by an unstable vapour blanket. Intermittent rewetting is believed to occur in

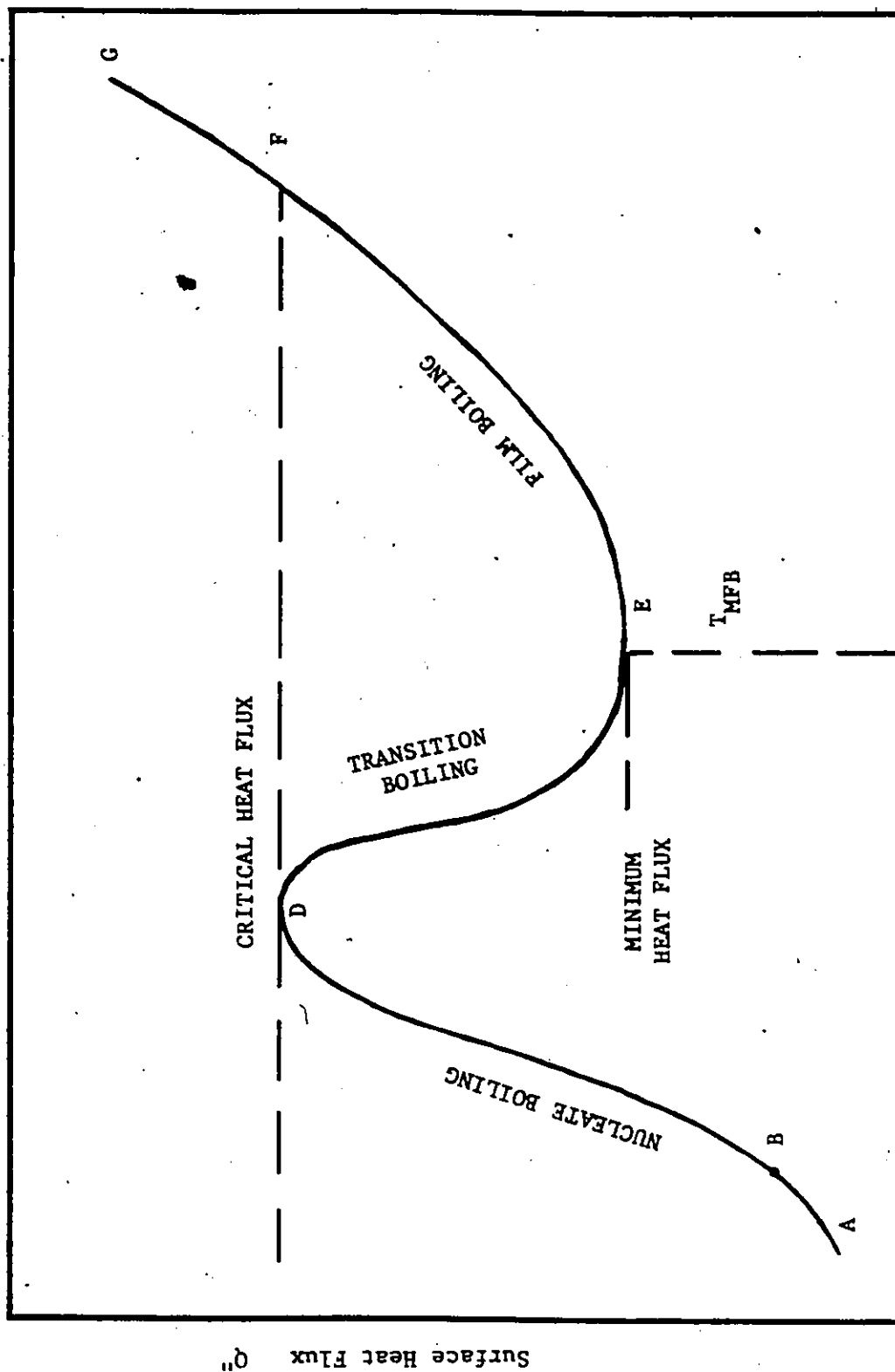


FIGURE II.1.1 Pool Boiling Curve

this region. This region cannot be studied in a constant heat flux control system but only under conditions approximating constant surface temperature. [Collier, 1972]

II.2. Two-phase flow pattern

The flow in a heated channel can have two configurations depending on the heat flux level [Hsu, 1972] as shown in Figure II.2. At low heat flux levels, prior to the film boiling region, the liquid is in contact with the wall. A part of this liquid is evaporated and some is entrained in the vapour core (annular flow). Eventually, liquid droplets are flowing in a vapour core (dispersed flow). For a high heat flux, a vapour blanket covers the heating surface (inverted annular flow). Eventually, a dispersed spray of droplets will flow in the tube. The flow regimes for the two heat flux levels can be summarized by the following regions:

- (1) Single-phase forced convection.
- (2) Bubbly flow. The heat is transferred by nucleation.
- (3) Slug or plug flow. In bubbly flow, the bubbles grow by coalescence and generate the typical bullet-shaped plugs [Ginoux, 1978].
- (4) Annular flow. The liquid is in the form of a liquid film and droplets dispersed in the vapour core of the flow.
- (5) Inverted annular flow (IAF). The liquid core is separated from the wall by a thin layer of vapour. This flow regime is thought to be in the low quality region in the post-dryout conditions.
- (6) Dispersed flow. A dispersed spray of droplets is flowing in the tube. This flow regime is often referred to as liquid-droplet deficient regime as insufficient liquid is available to maintain a wet wall. [Groeneveld, 1980].

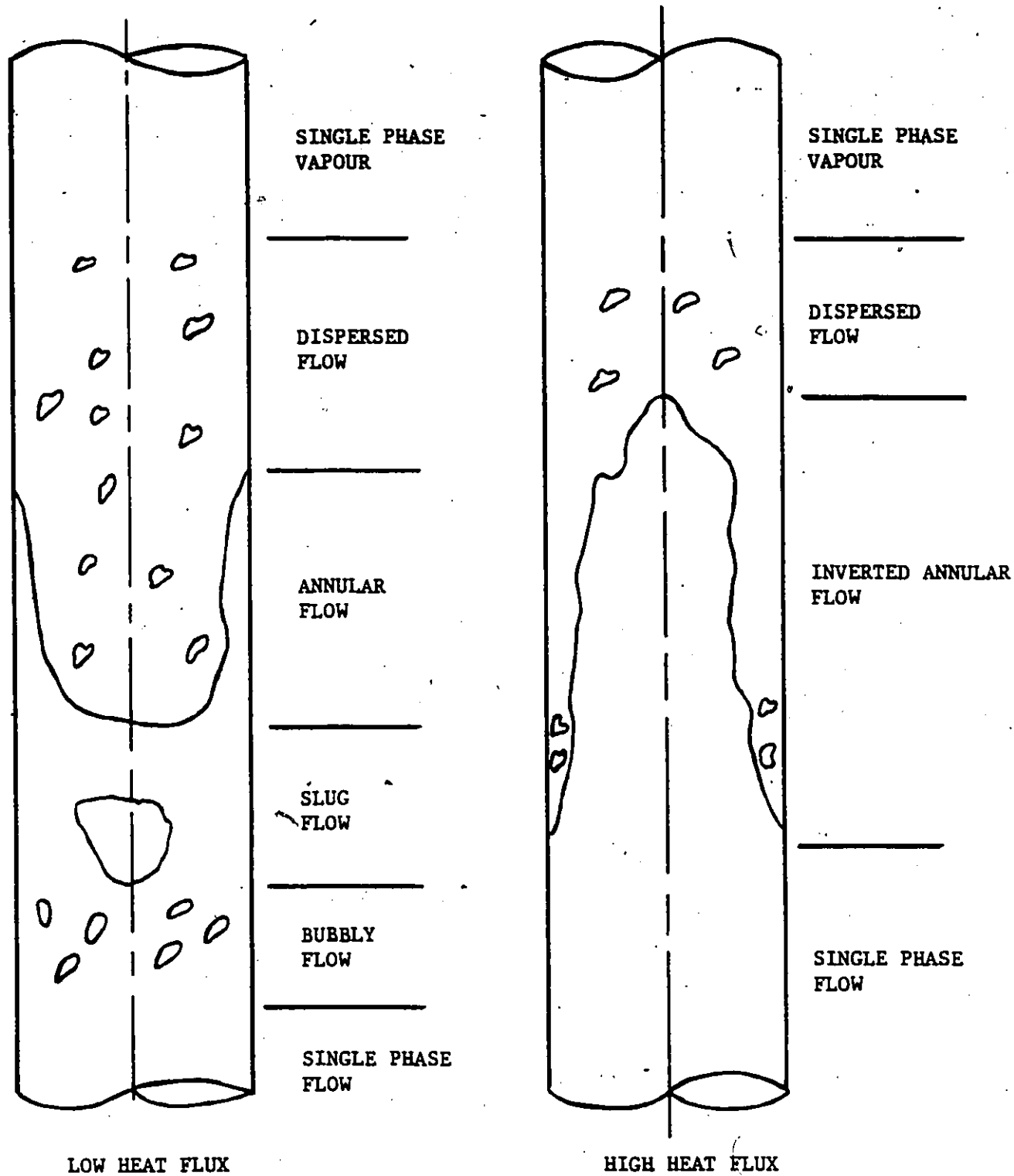


FIGURE II.2. Flow pattern in upward flow

II.3. Transition from inverted annular flow to dispersed flow

At a given point in the tube, a breakup of the liquid core occurs as the quality increases. Tong [1965] and Rankin [1961] derived two stability criteria. However, both are based on the wave number m , which is calculated using the wavelength, and can not be evaluated. Another method was proposed by Dougall and Rohsenow [1963]. The heat transfer coefficient in the high quality region is calculated using a Dittus-Boelter type of equation. The heat transfer in the low quality region is predicted using a proposed correlation. The quality at the intersection of both heat transfer regions is the transition quality (see Figure II.3). A value of 10% was obtained from low pressure experimental data. Dougall and Rohsenow mentioned that this value is approximate since the transition from the IAF regime to the dispersed flow regime is gradual. The transition based on thermodynamic equilibrium quality was also used by Shah [1980]. He proposed a value of 10% for a range of reduced pressure (0.01 to 0.97). However for low pressure, the void fraction is quite high at 10% quality. Since dispersed flow regime is expected to occur for high void, it therefore seems unreasonable to use the quality to predict the transition. As the pressure increases, the void fraction corresponding to the same quality is lower. Assuming that the transition occurs at the same void, this means that the transition occurs at higher quality for higher pressure.

The transition from the inverted annular flow regime to the dispersed flow regime based on the void fraction is shown in Figure II.4 [Fung, 1981]. For a void fraction less than 0.3, the flow regime is inverted annular. Beyond 0.8, dispersed flow regime occurs. Between these two values, transition flow exists in the form of slugs [Groeneveld, 1980].

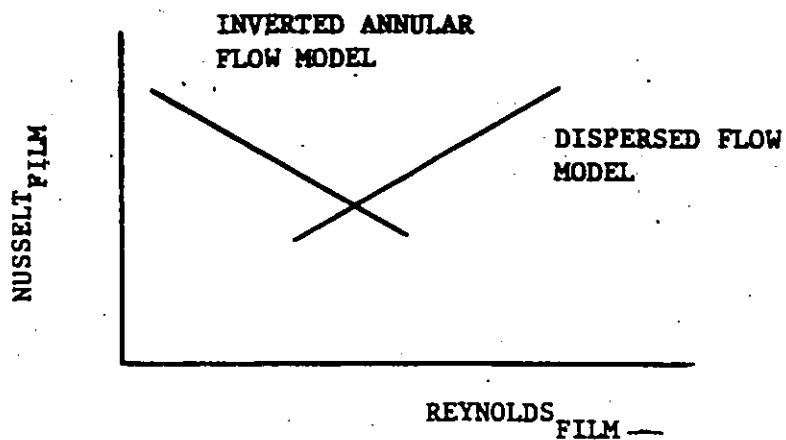


FIGURE II.3. Transition from inverted annular flow model to dispersed flow model

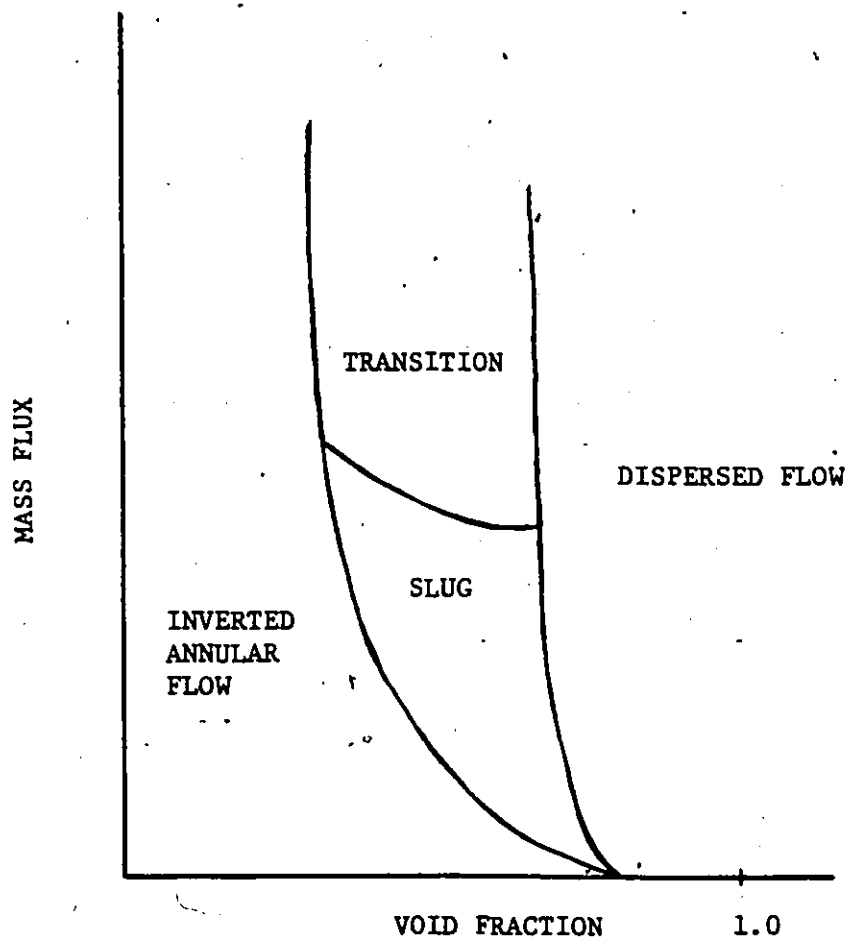


FIGURE II.4. Transition from inverted annular flow model to dispersed flow model based on void fraction

III. LITERATURE REVIEW

III.1. Introduction

Post-Dryout Heat-transfer prediction methods can be divided into four categories:

- (a) Correlations applicable to pool boiling and low flow.
- (b) Empirical correlations using Dittus-Boelter type equation (h based on $T_w - T_{sat}$).
- (c) Empirical correlations which recognize the departure from a thermodynamic equilibrium condition and attempt to calculate the "true" vapour quality and vapour temperature.
- (d) Semi-theoretical models in which attempts have been made to account for the physical mechanism.

A review of these prediction methods is presented in the following sections.

III.2. Pool boiling correlations

Bromley [1950] published one of the earliest heat transfer correlations for film boiling. His correlation was limited to pool boiling and was based on the following assumptions:

- 1) The liquid is separated from the hot wall by a continuous vapour blanket.
- 2) Heat transfer through a vapour film is by conduction and radiation.
- 3) Most of the heat is used for evaporation (superheating of the vapour being small).
- 4) The kinetic energy of vapour is negligible.
- 5) Vapour-liquid interface is smooth and continuous.
- 6) The liquid is at the boiling point at the vapour-liquid interface.
- 7) The vapour temperature is assumed to be the average of the saturation and inside wall temperatures.

The assumptions are similar to Nusselt's equations for condensation with ρ_l , cp_l , $(T_v - T_w)$ and Pr replaced by ρ_v , cp_v , $(T_w - T_{sat})$ and Pr_v . Bromley [1950] derived the following equation for the outside of a horizontal tube:

$$h_{co} = 0.62 \left[\frac{k_v^2 \rho_v (\rho_l - \rho_v) g h'_{fg} cp_v}{D(T_w - T_{sat}) Pr_v} \right]^{1/4} \quad \dots 3.1$$

h'_{fg} : difference in heat content between vapour and liquid at its boiling point.

h_{co} : heat transfer coefficient without radiation.

The radiation effect can be large for high wall temperature and must be considered. Bromley suggested the following expression to include the effect of radiation.

$$h = h_{co} + 0.75 h_{rad} \quad \dots 3.2$$

where

$$h_{rad} = \left[\frac{\sigma}{\frac{1}{\epsilon_w} + \frac{1}{\epsilon_l} - 1} \right] \left[\frac{T_w^4 - T_{sat}^4}{T_w - T_{sat}} \right] \quad \dots 3.3$$

Other pool boiling correlations of this type are listed in Table 1. The constants, 0.62 for Bromley and 0.425 for Berenson [1961], are evaluated empirically from experimental data. Although these correlations were initially derived for pool boiling, they are often used to predict heat transfer coefficients for low flows.

III.3 Empirical correlation assuming thermodynamic equilibrium

Most of the empirical correlations are similar to the Dittus-Boelter equation which was initially derived for fully developed heat transfer

$$Nu_d = C Re_d^m Pr^n F \quad \dots 3.4$$

where

F: function of quality and sheath temperature.

C, m and n: constants.

The Prandtl number is the ratio of thermal diffusivity to momentum diffusivity. It has been shown from physical reasoning [Holman, 1976] that the Reynolds number and the Prandtl number are the relevant groups in the dimensionless heat transfer coefficient, Nu. This type of equation is used in the dispersed flow region but occasionally, it predicts the heat transfer coefficient in the inverted annular flow regime e.g. Kalinin's equation [1977] shown in Table 1.

III.4 Non-equilibrium method

In an experimental study by Parker and Grosh [1962] it was determined that even at an equilibrium mixture quality beyond 1.0, liquid drops can still exist in the flow, thereby indicating that non equilibrium could exist during film boiling. Nijhawan [1967], Gottula [1983] and Laperriere [1983] measured the superheated steam temperature and observed the following trend: an increase of flow decreases the steam temperature,

and a heat flux rise increases the steam temperature. Correlations which predict the non-equilibrium (based on $T_w - T_{sat}$) during dispersed flow film boiling were developed by Tong and Young [1974], Groeneveld and Delorme [1976] and Chen, Ozkaynak and Sundaram [1979]. Many of these authors have emphasized that the correct prediction of the superheated vapour temperature is a necessary step in the prediction of the wall temperature. As shown by Groeneveld and Delorme [1976], the assumption of thermodynamic equilibrium ($T_v = T_{sat}$) will result in a lower predicted wall temperature.

III.5 Semi-theoretical and theoretical models

III.5.1 Models using boundary layer analysis

Using the concept of the boundary layer, an analysis has been made by Cess and Sparrow [1961] to determine the heat transfer in forced convective film boiling on a flat plate. The basic principles of mass, momentum and energy conservation govern the velocity and temperature distribution

$$\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0 \quad \dots 3.5$$

$$U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} = \nu \frac{\partial^2 U}{\partial y^2} \quad \dots 3.6$$

$$U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y} = \alpha \frac{\partial^2 T}{\partial y^2} \quad \dots 3.7$$

where x is the distance along the plate from leading edge (Elias [1980] neglected the velocity profile in the y direction (V is constant)).

The boundary conditions are as follows:

- 1) The tangential velocity interface is the same for the liquid and the vapour ($U_{l1} = U_{v1}$).
- 2) The tangential shear at interface is the same for the liquid and the vapour.

$$\mu_l \left(\frac{\partial U}{\partial y} \right)_{l1} = \mu_v \left(\frac{\partial U}{\partial y} \right)_{v1} \quad \dots 3.8$$

- 3) Far away from the plate, the liquid velocity is U_∞ .
- 4) The mass flow crossing the interface must be preserved:

$$k_v \frac{\partial T}{\partial y} = \dot{m} H_{fg} \quad \dot{m}: \text{mass flow across the interface} \quad \dots 3.9$$

This last equation indicates that the principal mode of heat transfer at the vapour side of the interface is by conduction.

The application of the boundary layer concept was also used by Ito and Nishikawa [1966] to study the heat transfer in film boiling on a flat plate. However, as mentioned in their conclusions, the following conditions should be satisfied:

- 1) the vapour-liquid interface, must be smooth;
- 2) the liquid must not contain bubbles.

When discussing the paper of Cess and Sparrow [1961], Rohsenow wrote: "It should be remembered that the analysis presented requires that all of the vapour generated remains in the film. Visual experiments currently being conducted in our laboratory on forced convection film boiling inside horizontal and vertical tubes indicates that much of the vapor formed actually appears in the core, which becomes a mixture of liquid and vapour. The film thickness may in fact, be nearly uniform

along the tubes. Experimental heat-transfer results are of the order of four times those obtained by this type of analysis." [Cess and Sparrow, 1961].

Cess and Sparrow [1961], Ito and Nishikawa [1966] and Koh [1962] all developed theoretical models using the boundary layer theory. It appears that the heat transfer coefficient obtained from these models is in good agreement with the value obtained by the Bromley equation [Koh, 1962]. However, Rohsenow [Cess and Sparrow, 1961] and Hsu and Westwater [1959] showed that Bromley's equation for film boiling may predict heat transfer coefficients which are too small, particularly if turbulence develops.

Boundary layer models and Bromley's equation [1950] are theoretical and limited to pool boiling. However, they can be used to determine the heat transfer for low flows.

III.5.2 Hsu and Westwater's model

Hsu and Westwater [1959] developed an approximate theory for film boiling on a vertical surface for saturated liquids. The assumptions are:

- 1) The vapour flow in the upstream part of the heating surface is laminar and the heat transfer can be correlated by the Bromley equation.
- 2) Radiation heat transfer is neglected.
- 3) When the local Reynolds number reaches a critical Reynolds number, a laminar sublayer next to the wall exists and it is assumed that this sublayer constitutes the entire resistance to the heat transfer.

The local Reynolds number is defined as:

$$Re^* = \frac{y^* u^* \bar{\rho}_v}{\mu_v} \quad \dots 3.10$$

y^* : thickness of the vapour layer at the critical height.

u^* : maximum velocity of the vapour layer at the critical height.

$\bar{\rho}_v$: average density of the vapour in the laminar film.

The model can be represented by Figure III.1. When the critical Reynolds number is reached, the transition between laminar and turbulent flow is indicated. The choice of $Re^* = 100$ as critical Reynolds number is based on the boundary layer theory. The laminar sublayer heat transfer is assumed to take place by conduction only and is given by:

$$h = \frac{k_v}{\delta_s} \quad \dots 3.11$$

δ_s : laminar sublayer thickness.

A theoretical prediction of the heat transfer from Hsu and Westwater is shown in Figure III.2. The increase of heat transfer with length can be explained by the fact that the laminar sublayer thickness decreases with length, causing an increase in the heat transfer coefficient.

3.4.3 Kalinin's model

A mathematical model of inverted annular flow film boiling was developed by Kalinin et al. [1977]. The assumptions are:

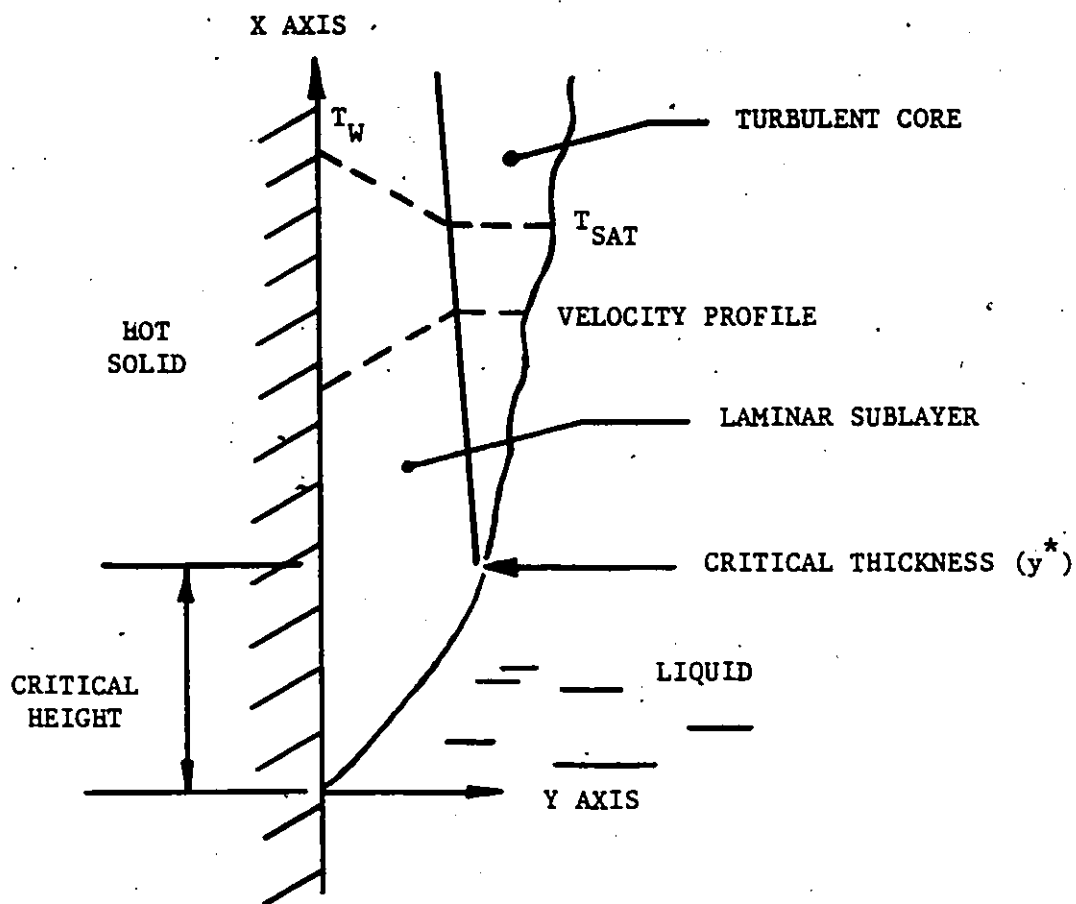


FIGURE III.1. Hsu's model

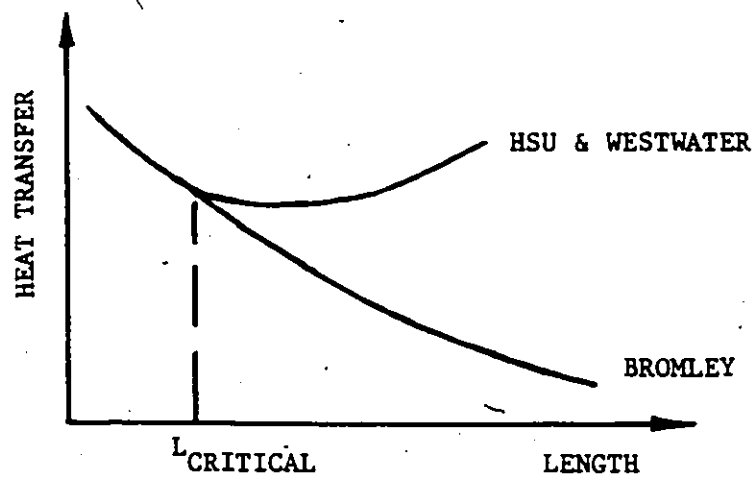


FIGURE III.2. A comparison between the Bromley's model and the Hsu's model

- 1) The liquid moves in the form of a turbulent jet which is separated from the heated surface by a turbulent vapour film.
- 2) The interfacial temperature is equal to the saturation temperature ($T_i = T_{sat}$).
- 3) The vapour and the liquid are moving at the same velocity ($U_v = U_l$).

The model used two basic equations. The first equation represents the continuity equation. The total mass flow rate is of the vapour flow rate and the liquid flow rate:

$$W_{total} = W_v + W_l \quad \dots 3.12$$

In expanded form:

$$\rho_l U_o \frac{\pi D^2}{4} = \rho_v U \pi (D-\delta) \delta + \rho_l U \frac{\pi}{4} (D-2\delta)^2 \quad \dots 3.13$$

$$U = \frac{U_o D^2}{\left[4 \left(\frac{\rho_v}{\rho_l} \right) (D-\delta) \delta + D^2 \left(1 - \frac{2\delta}{D} \right)^2 \right]} \quad \dots 3.14$$

$$U = \frac{U_o}{\left[\left(\frac{\rho_v}{\rho_l} \right) + \left(1 - \frac{\rho_v}{\rho_l} \right) \left(1 - \frac{2\delta}{D} \right)^2 \right]} \quad \dots 3.15$$

The second equation represents the energy equation. For unit length, this equation can be written as:

$$\pi D Q''_{total} = \pi (D-2\delta) Q''_l + Q''_{vaporization} \quad \dots 3.16$$

The total heat from the wall into the vapour film must be equal to the sum of the heat to the liquid and the heat used to generate the vapour. The convection heat transfer mode is used in the model for the vapour and the liquid:

$$Nu_v = A Re_v^m \quad \dots 3.17$$

$$Nu_l = B Re_l^n \quad \dots 3.18$$

where

$$A = 0.023 Pr_l^{0.4}$$

$$B = 0.0198 Pr_v^{0.4}$$

m,n: constants.

The energy equation (3.16) is then given by:

$$\begin{aligned} \pi D \frac{k_v}{2\delta} (T_w - T_s) A \left(\frac{U 2\delta}{v_v} \right)^m &= \frac{k_l}{(D-2\delta)} (T_s - T_l) B \left(U \frac{(D-2\delta)}{v_l} \right)^n \pi (D-2\delta) \\ &+ H_{fg} \frac{d}{dz} [\rho_v U \pi (D-\delta) \delta] \end{aligned} \quad \dots 3.19$$

The differential equation of the vapour layer thickness can be written by using both the mass conservation equation and the energy equation (see Appendix A for derivation):

$$\pi \rho_v U_o H_{fg} \frac{D^2}{4} \frac{db}{dz} =$$

$$\frac{\pi k_v (T_w - T_s) A (U_o D / \nu_v)^m \left[(1-b)^2 + \frac{\rho_v}{\rho_l} (2-b)b \right]^{2-m}}{2 b^{1-m} (1-b)}$$

$$- \frac{\pi k_l (T_s - T_l) B (U_o D / \nu_l)^n \left[(1-b)^2 + \frac{\rho_v}{\rho_l} (2-b)b \right]^{2-n}}{2(1-b)^{1-n}}$$

...3.20

$$\text{with } b = \frac{2\delta}{D}.$$

Using equations 3.17 and 3.18, Kalinin found poor agreement with his experimental data. Subsequently, using the empirical film boiling heat transfer coefficient given in equations 3.21 to 3.26, he succeeded in obtaining a much better agreement between the experimental data and the model predictions.

$$Nu_v = 0.0078 \left[\frac{\rho_v U_v 2\delta_v}{\mu_v} \right]^{0.75} Pr_v F \quad \dots 3.21$$

$$F = [1 + (3 \times 10^{-6}) \phi + 1.3 \text{Exp}(-4 \times 10^{-6} \phi)] \quad \dots 3.22$$

$$\phi = \frac{cp_v \Delta T_s}{H_{fg}} \left[\frac{\rho_v \sigma D}{\mu_v^2} \right]^{0.7} \quad \dots 3.23$$

$$Nu_l = B Re_l = 0.0012 Re_l^{0.4} Pr_l f_1(z/D) \quad \dots 3.24$$

$$f_1(z/D) = 1 + 1.22 \exp(-0.038 z/D) \quad \dots 3.25$$

$$Re_l = U_l (D-2\delta)/\nu_l \quad \dots 3.26$$

IV. FILM BOILING PREDICTION METHOD

IV.1. General

This section describes a model proposed for the inverted annular film boiling region. This model is a separated two fluid model and is used to predict the heat transfer coefficient in the low quality region. It is more rigorous than Kalinin's model [1977] which assumed no slip between the two phases. Kalinin used a turbulent vapour film with the heat transfer from the wall to the interface being described by a convection equation of the form $Nu_v = A Re_v^m$. Fung [1981] instead used a laminar film at low Reynolds number and a turbulent film above a critical Reynolds number. The method of Fung was initially derived by Hsu and Westwater [1959]. Hsu assumed a resistance only through the laminar sublayer but he found an increase in heat transfer with quality due to the decrease of the thickness of this sublayer. Fung instead defined a thermal sublayer which is thicker than the laminar sublayer. In the proposed model, the heat is transferred from the wall to the interface by conduction and radiation and from the wall to the vapour by convection. Strictly speaking the conduction heat transfer and the convection heat transfer can not be both included. However, the vapour velocity is initially low and the vapour film is thin. This means a high conduction and a low convection (low Reynolds number). As the vapour velocity increases and the film gets thicker, the conduction is reduced and the convection increases (higher Reynolds number). This will lead to a situation similar to Kalinin's for which a turbulent vapour film is assumed. This also permits a smooth transition from laminar flow to turbulent flow.

A laminar velocity profile was assumed and this limits the range of application of mass flux. Assuming a laminar velocity profile will mean a lower gradient, du/dx , at the wall and an underprediction of shear stress at high Reynolds number. However, it was convenient to use in order to obtain an interfacial velocity equal to the average of the mean velocity of the vapour and the liquid velocity. This is reasonable due to the lack of measurement of interfacial velocity.

The proposed model is a modification of Chan and Yadigaroglu's model [1980]. The heat transfer modes are shown in Figure IV.1. The total heat flux from the wall is transferred by conduction, convection and radiation. At the interface, the total heat flux from the wall (Q'') removes the subcooling (Q''_l) and generates the vapour (Q''_v) at a rate Γ_v . All the vapour generated is assumed to end up in the vapour film. (It is assumed that no vapour is entrained in the liquid core.) This assumption is different than Chan and Yadigaroglu's model where a fraction of the total vapour generated was assumed to be present in the film. Our assumptions are:

- 1) The heat flux from the wall takes place by conduction, convection and radiation ($Q = Q_{cond} + Q_{conv} + Q_{rad}$).
- 2) The vapour properties in the film are calculated at an arithmetic average of the wall temperature and saturation temperature ($T_v = (T_w + T_{sat})/2$).
- 3) Interfacial shear stress is assumed at the vapour-liquid interface.
- 4) Subcooled liquid is present in the liquid core and properties of the liquid are calculated at the subcooled temperature ($T_l < T_{sat}$).
- 5) Radiation heat transfer is calculated assuming an exchange between two infinite gray plates.

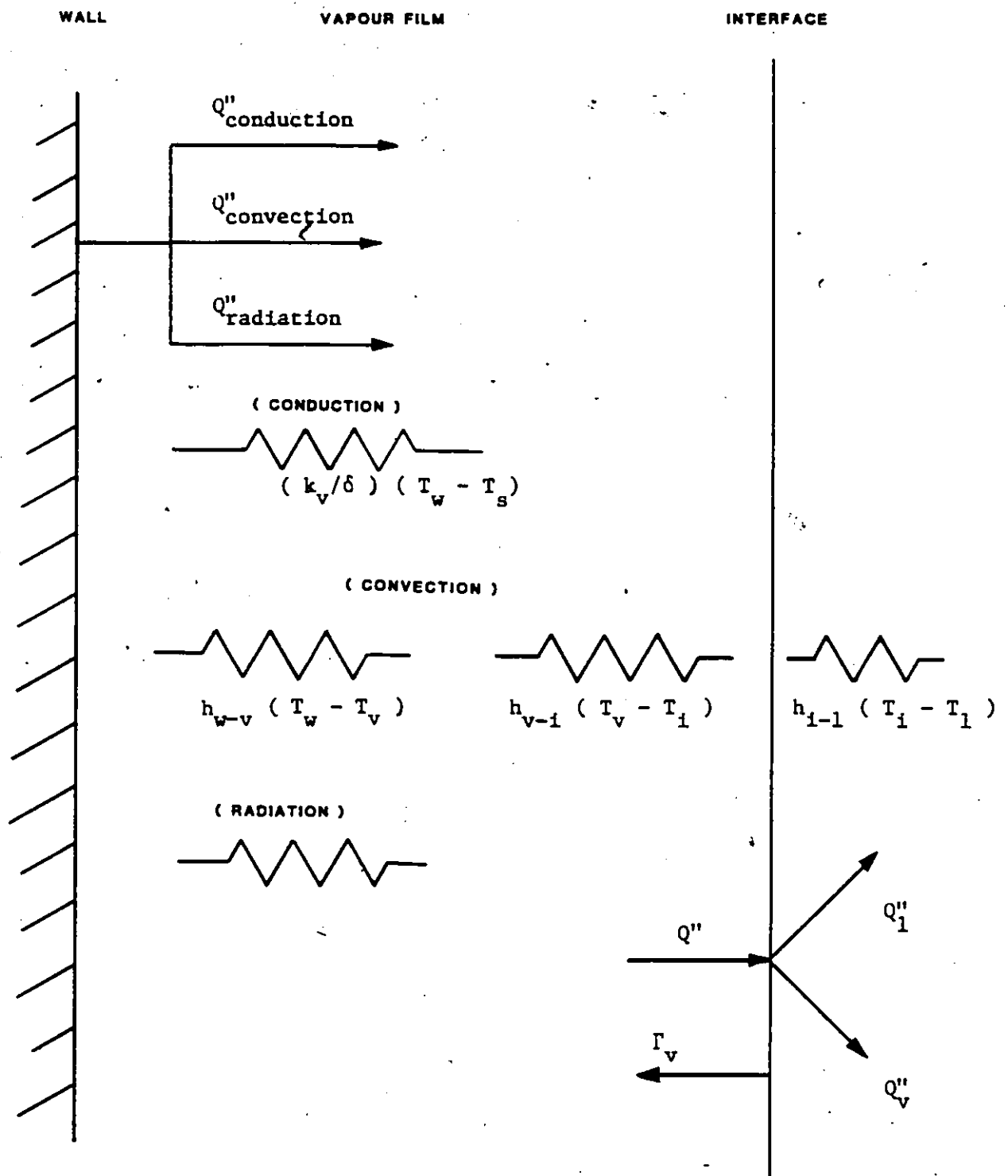


FIGURE IV.1. Heat transfer modes

- 6) At the quench front, the film vapour layer thickness is calculated using a pool boiling equation (Berenson). This method was also used by Hsu and Westwater [1959].
- 7) The interface velocity is the average of the liquid core velocity and the vapour film velocity.
- 8) No vapour is assumed in the core and there is no entrainment of liquid in the vapour film.

The proposed model is presented in Appendix B. Several simplifications were made and these are explained in the same appendix.

IV.2. Discussion of the model

IV.2.1 General

In this section, the proposed model is compared with data obtained by Stewart [1981]. Since the model can not predict the heat transfer coefficient for a test section positive quality inlet, the positive inlet data will not be used. It will be shown that the proposed model adequately incorporates the effect of parameters such as pressure and subcooling. The effects of parameters such as void fraction, slip ratio will also be discussed.

IV.2.2 The effect of pressure

The effect of pressure is shown in Figure IV.2 for two experimental runs of Stewart [1981]. The physical properties (such as the Prandtl number and the vapour conductivity) change following an increase in pressure which in turn will affect the heat transfer coefficient. The

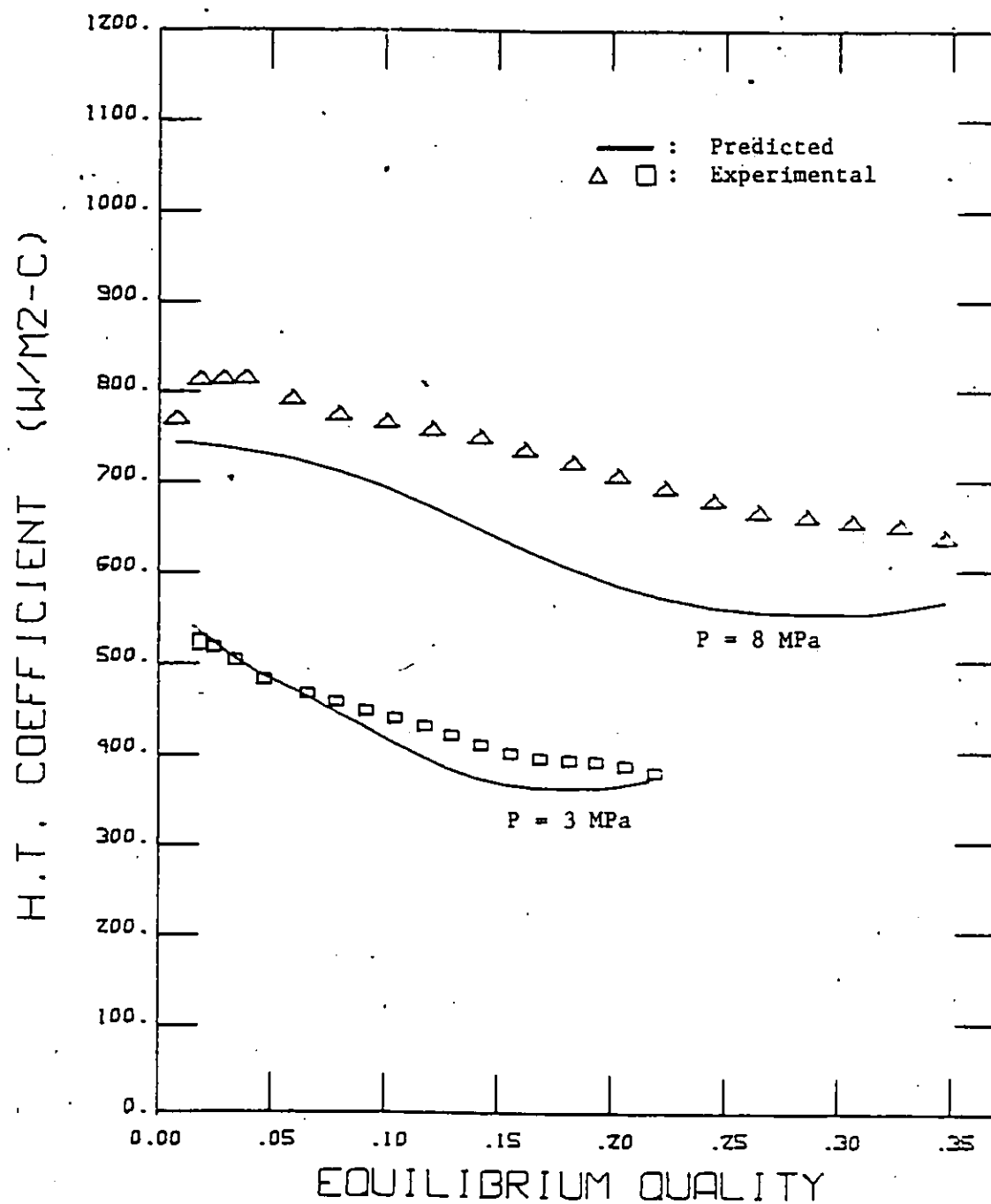


FIGURE IV.2. Effect of pressure on the heat transfer coefficient
 ($P = 8 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$,
 $\phi = 243 \text{ kW.m}^{-2}$ and $P = 3 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$,
 $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 180 \text{ kW.m}^{-2}$)

effect of any given property can not be easily evaluated due to the complexity of the differential equations presented in Appendix D. The void fraction is also lower at higher pressure which will result in a thinner vapour film and a higher conduction heat transfer.

IV.2.3. The effect of subcooling

As expected, the wall temperature is lower for a higher subcooling. This can be explained using equation B.44.

$$P_v Q'' = P_v Q_v'' + P_l Q_l'' \quad \dots 4.1$$

For a high subcooling, a greater part of the total heat flux from the wall (Q'') is required to remove the subcooling (Q_l'') and less vapour is generated (Q_v''). This results in a higher conductivity through the thinner vapour film. Each term of the previous equation is plotted in Figure IV.3 for a specific run. The subcooling value $P_l Q_l''$ is high initially and then decreases as the subcooling of the water is removed. The vapour generation rate and consequently the value of Q_v'' is low for a low quality and then increases with quality as more vapour is generated.

IV.2.4 Void fraction and slip ratio

The model's void fraction is compared with the void fraction calculated from the following equation:

$$\alpha = \frac{X}{\left[X + \frac{\rho_g}{\rho_f} (1-X) S \right]} \quad \dots 4.2$$

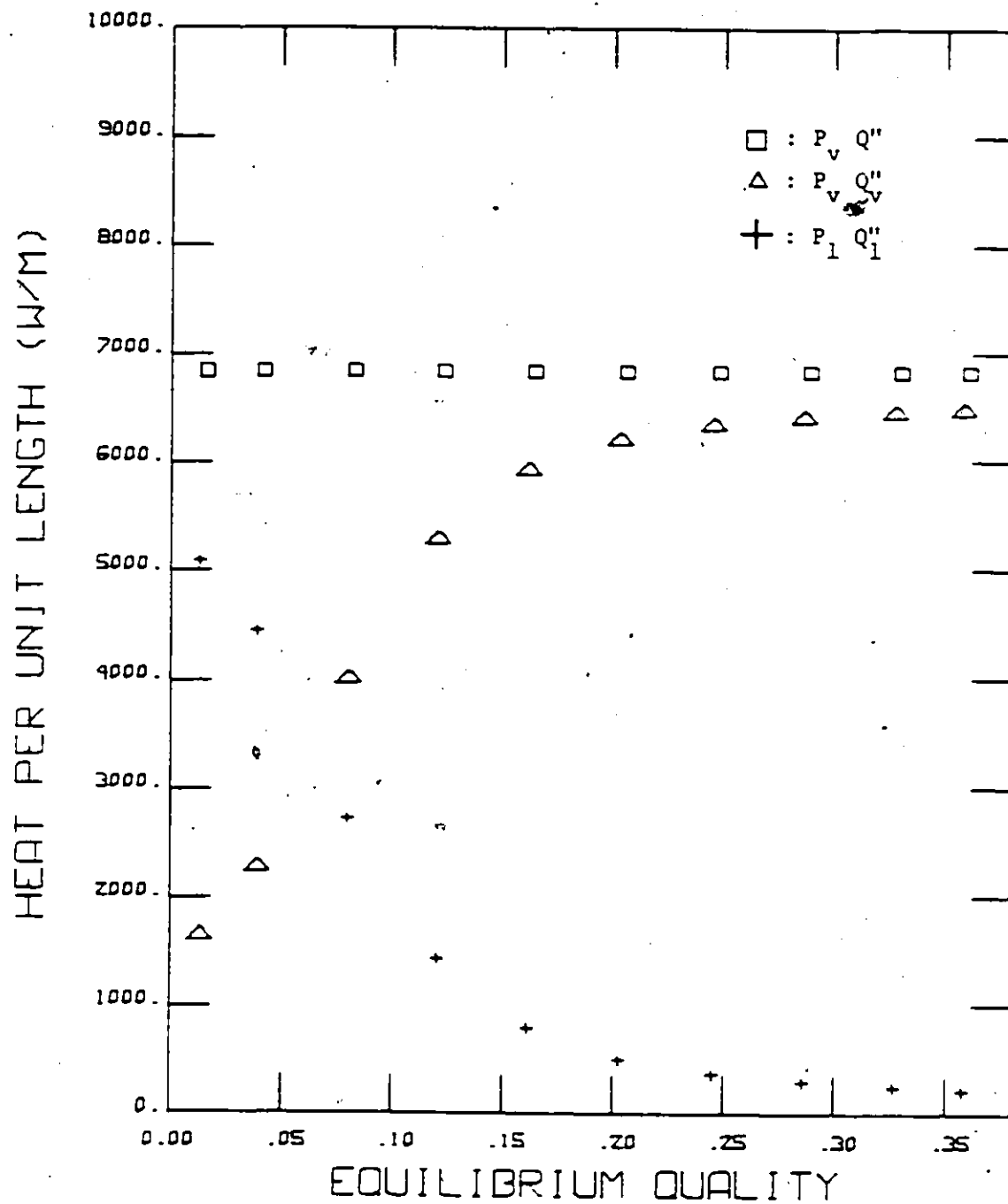


FIGURE IV.3. heat per unit length ($P = 8 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$,
 $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 243 \text{ kW.m}^{-2}$)

A slip ratio (S) was derived by Stewart [1981] using the experimental values of void fraction of Fung [1981] obtained at atmospheric pressure. (The slip ratio was also corrected for the effect of pressure.) The equation is

$$S = \frac{8.00}{4.36} \left[\frac{\rho_f}{\rho_g} \right]^{0.2} \quad \dots 4.3$$

This implies that the slip ratio is constant for a given pressure which is unrealistic. The model's slip ratio increases with axial direction until it reaches a maximum value. It then follows a decrease of the slip ratio approaching the dispersed flow region.

The effect of pressure on the slip ratio may be understood from the equation:

$$S = \left[\frac{X}{1-X} \right] \left[\frac{1-\alpha}{\alpha} \right] \left(\frac{\rho_f}{\rho_g} \right) \quad \dots 4.4$$

From this equation opposing effects of pressure on slip may be observed i.e. decreasing slip with increasing pressure through the density ratio and increasing slip with increasing pressure through the void fraction ratio. Figures IV.4 and IV.5 show the comparison of void fraction and slip ratio with model's prediction for a specific run. The void fraction shows reasonable agreement as does the slip ratio. However, even if it shows the correct trend with changing pressure, the model under-estimates the void fraction. It was found that this model predicts a more realistic slip ratio than Fung [1981].

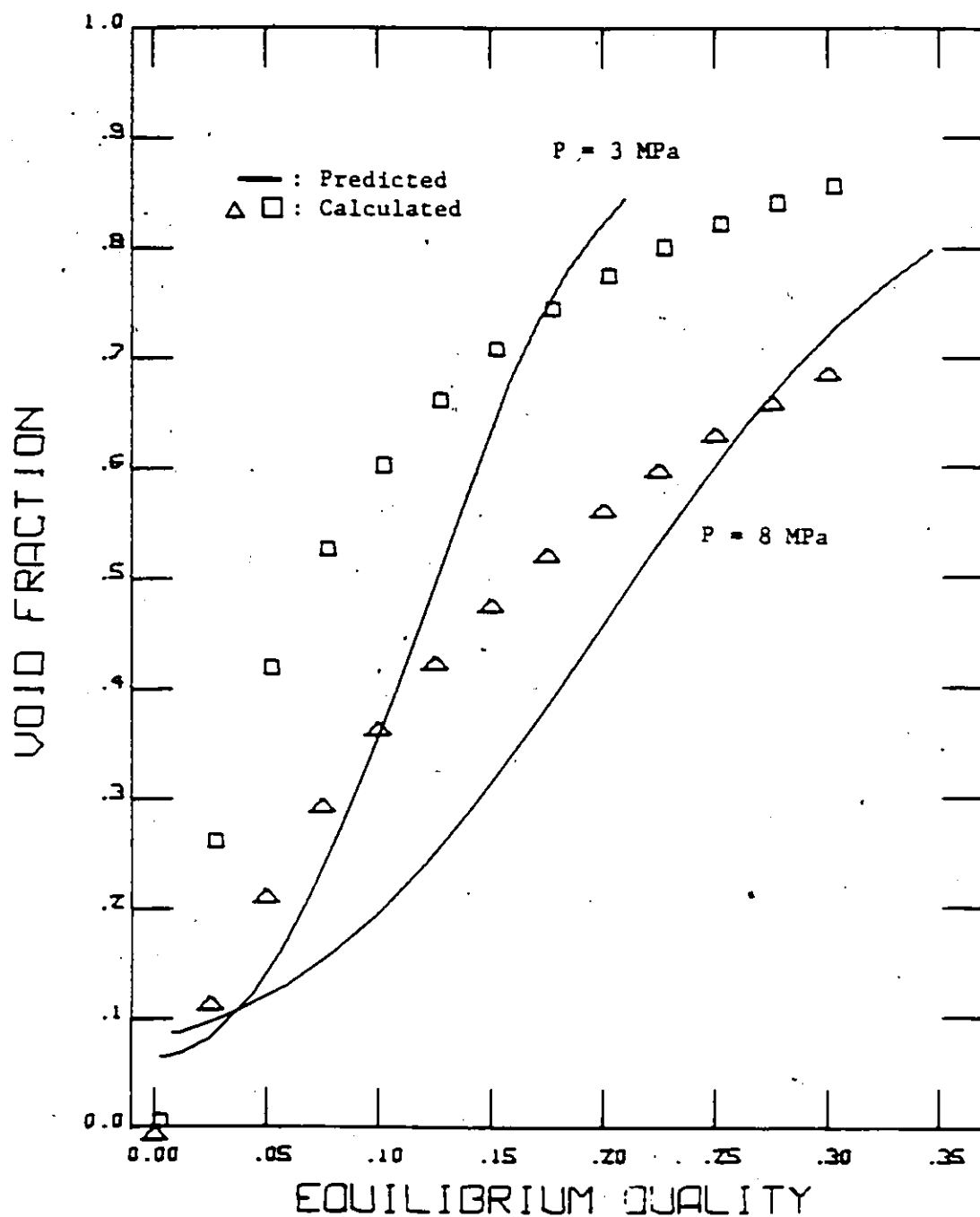


FIGURE IV.4. Calculated (equation 4.2) and predicted void fraction.
 ($P = 8 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$,
 $\phi = 243 \text{ kW.m}^{-2}$ and $P = 3 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$,
 $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 180 \text{ kW.m}^{-2}$)

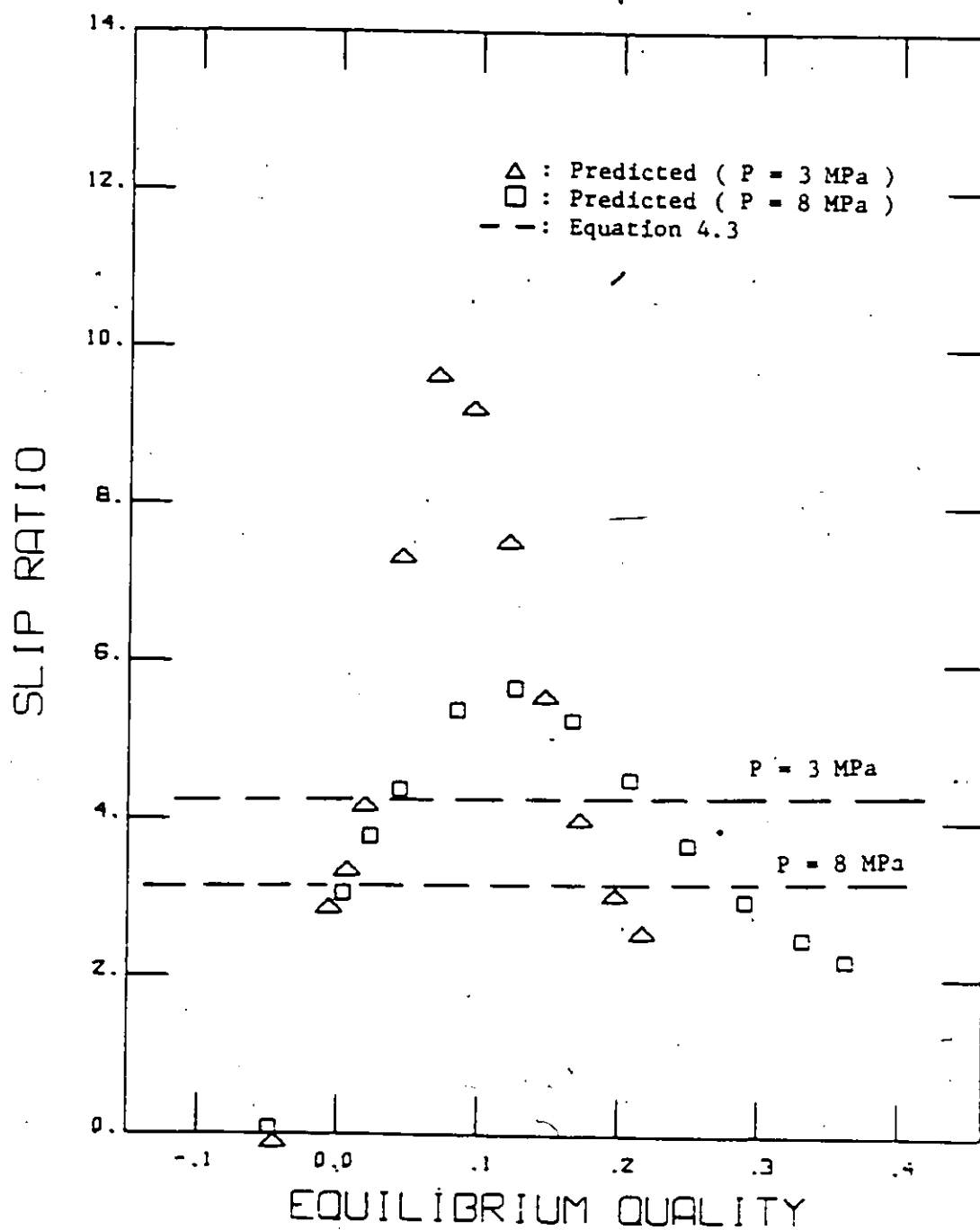


FIGURE IV.5. Predicted slip ratio ($P = 8 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 243 \text{ kW.m}^{-2}$ and $P = 3 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 180 \text{ kW.m}^{-2}$)

IV.2.5 Vapour film thickness

The predicted vapour film thickness as a function of quality is shown in Figure IV.6. The vapour film at the lower pressure ($P = 3$ MPa) is thicker than the vapour film at the higher pressure for the same quality. However, at low quality (less than .05) the vapour film thickness is almost identical in both cases. The higher heat transfer coefficient at higher pressure may be explained by better vapour conductivity. As expected, the vapour film is thicker at the lower pressure which explains the higher void fraction for the same quality.

IV.3 Comparison with experimental data

In this section, the model is compared with a few experimental data over a range of pressure (2 MPa - 8 MPa), inlet subcooling ($112 \text{ kJ} \cdot \text{kg}^{-1} - 228 \text{ kJ} \cdot \text{kg}^{-1}$) and mass flux ($219 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1} - 539 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$). It was found that the prediction method gives reasonable results in this range of data. (See Figures IV.7 - IV.12). However, the increase of the experimental heat transfer coefficient in the low quality region (i.e. the upstream part of the test section) may be explained by the axial conduction of the hot patch.

The effect of inlet subcooling is shown by comparing Figure IV.7 and Figure IV.8. The model's heat transfer coefficient is initially higher for the high subcooling run (Figure IV.8).

However, the inlet subcooling effect disappears in the high quality region (i.e. the downstream part of the test section) as discussed in the section IV.2.3.

Figures IV.9 and IV.10 present the effect of mass flux. The heat transfer coefficients can not be compared as a function of quality since the equilibrium quality is a function of the mass flux. The experimental initial value of the heat transfer coefficient (i.e. the upstream part of the test section) is higher for the high mass flux run but this effect is not present in the model since the model's heat transfer coefficient is calculated in the quench front region by using a pool boiling correlation which is independent of mass flux. However, it was found that the model predicted a higher heat transfer coefficient for the $523 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ mass flux than the $369 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ mass flux. This increase might be explained by the higher vapour Reynolds number and also by a better mixing in the vapour layer, which consequently prevents a large degree of vapour superheating.

The effect of pressure is also shown in Figures IV.11 and IV.12. The model's heat transfer coefficient as well as the experimental heat transfer coefficient is higher for the high pressure run as explained previously in section IV.2.2.

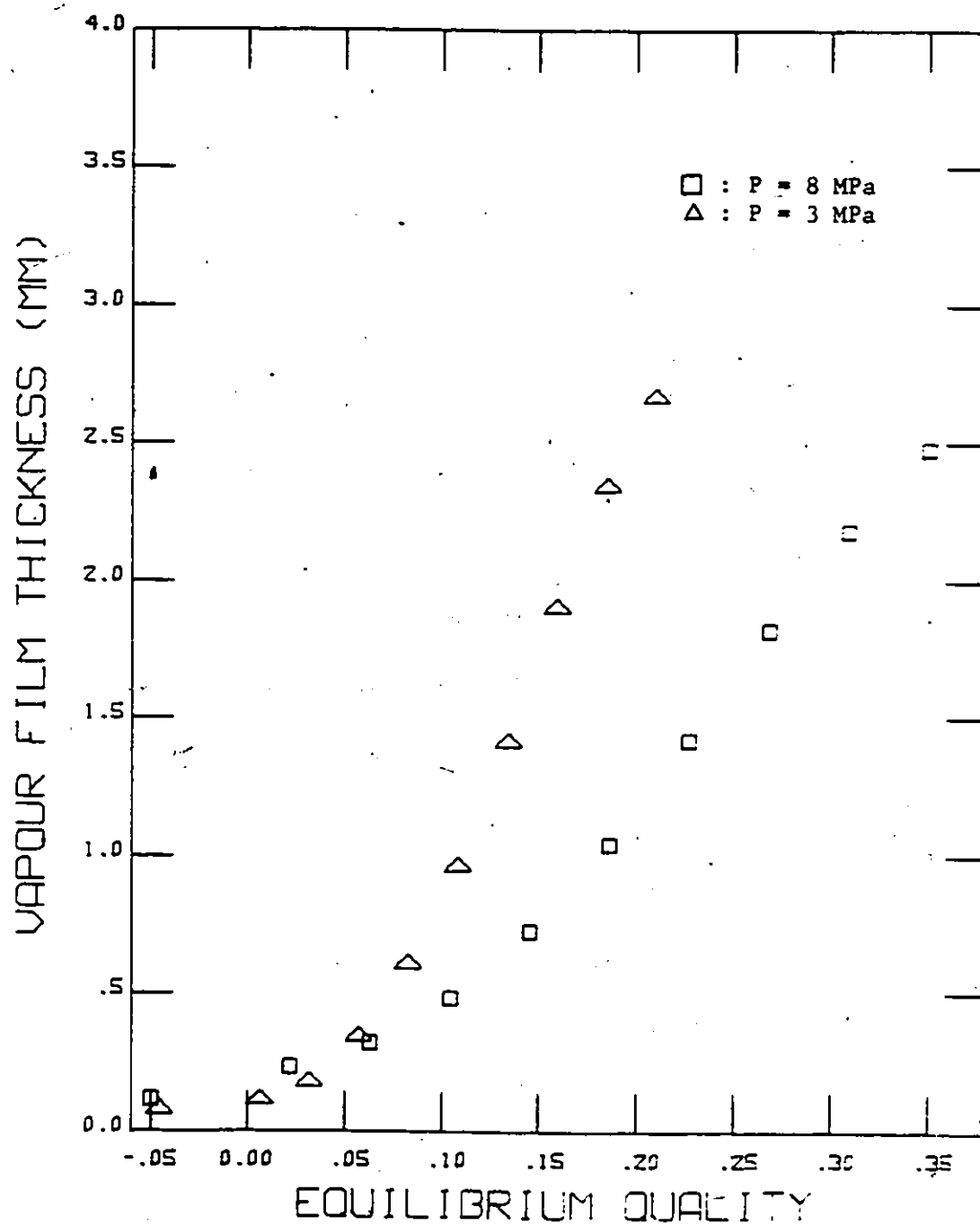


FIGURE IV.6. Predicted vapour film thickness ($P = 8 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 243 \text{ kW.m}^{-2}$ and $P = 3 \text{ MPa}$, $G = 359 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 100 \text{ kJ.kg}^{-1}$, $\phi = 180 \text{ kW.m}^{-2}$)

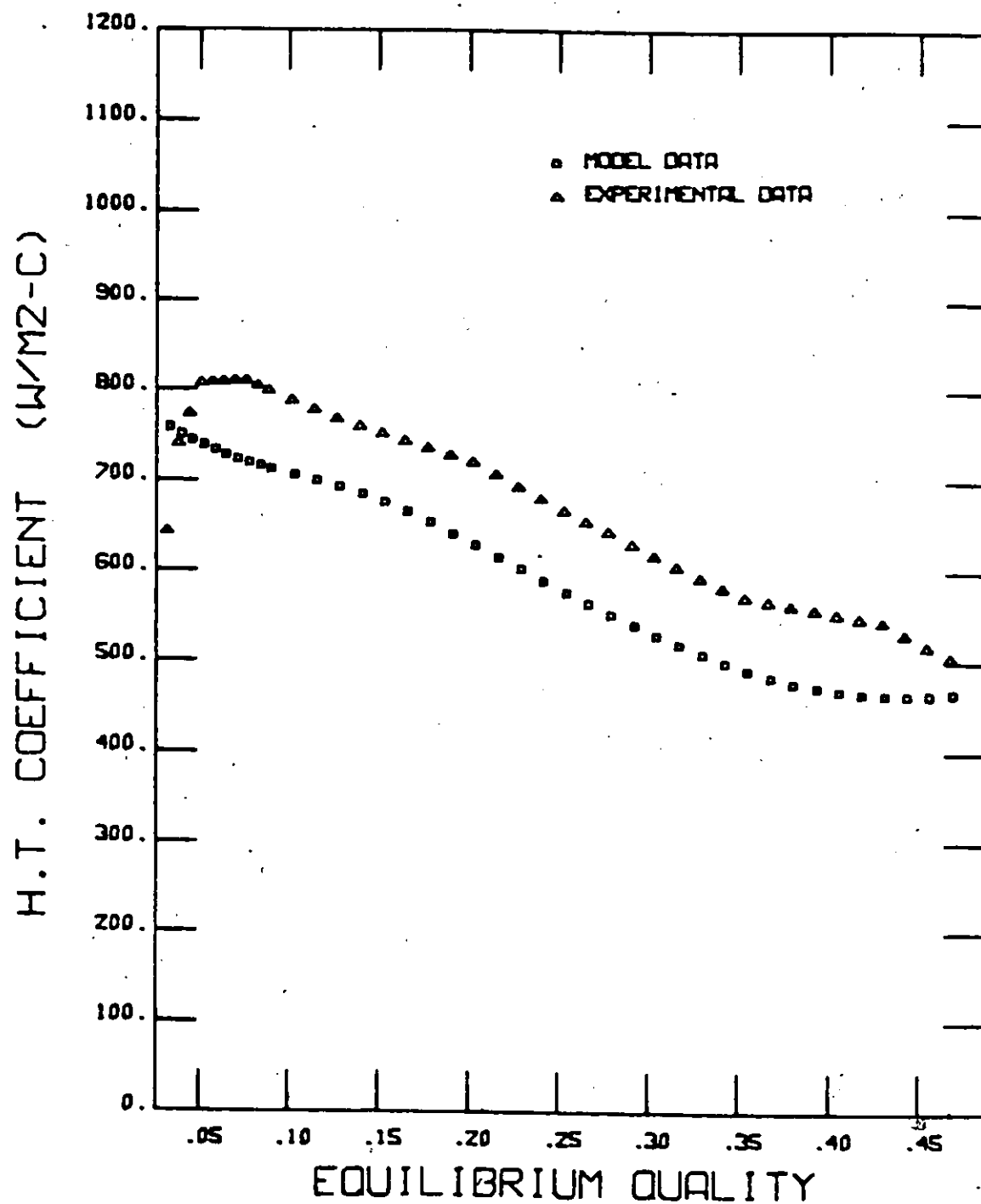


FIGURE IV.7. Predicted and experimental heat transfer coefficient
($P = 8 \text{ MPa}$, $G = 219 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 142 \text{ kJ.kg}^{-1}$,
 $\phi = 177 \text{ kW.m}^{-2}$)

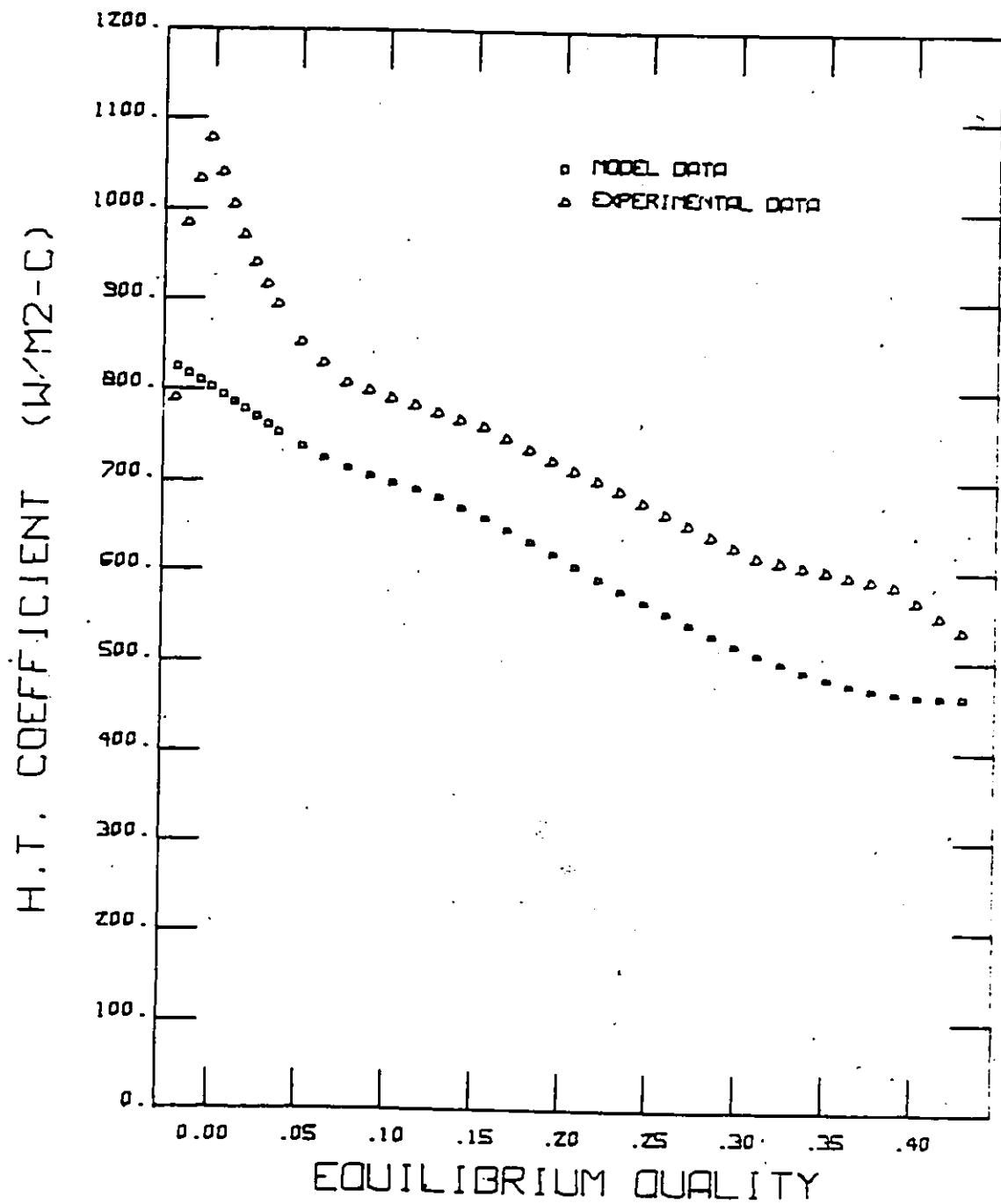


FIGURE IV.8. Predicted and experimental heat transfer coefficient
 ($P = 8 \text{ MPa}$, $G = 219 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{\text{in}} = 228 \text{ kJ.kg}^{-1}$,
 $\phi = 183 \text{ kW.m}^{-2}$)

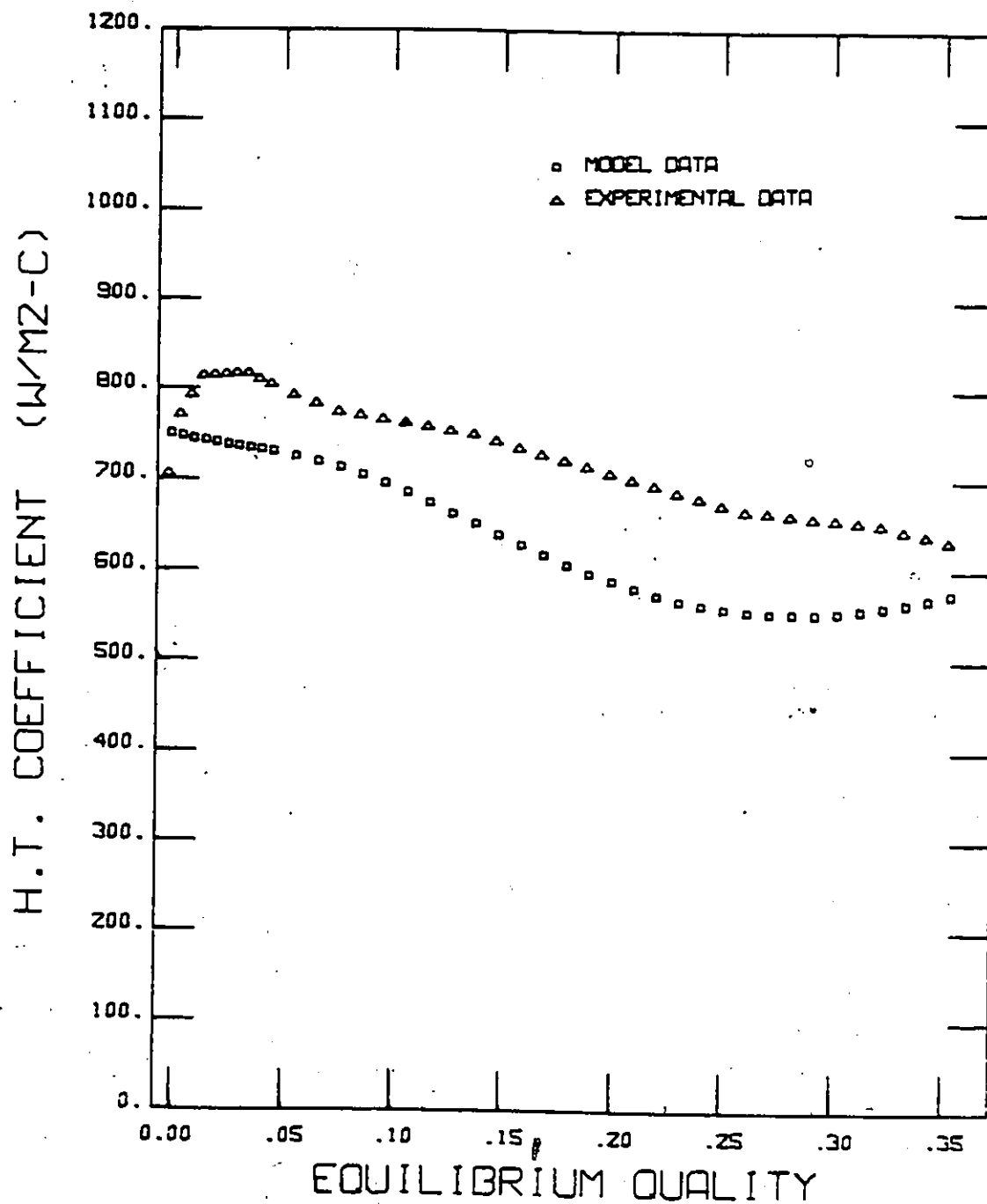


FIGURE IV.9. Predicted and experimental heat transfer coefficient
 ($P = 8 \text{ MPa}$, $G = 369 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{\text{in}} = 133 \text{ kJ.kg}^{-1}$,
 $\phi = 243 \text{ kW.m}^{-2}$)

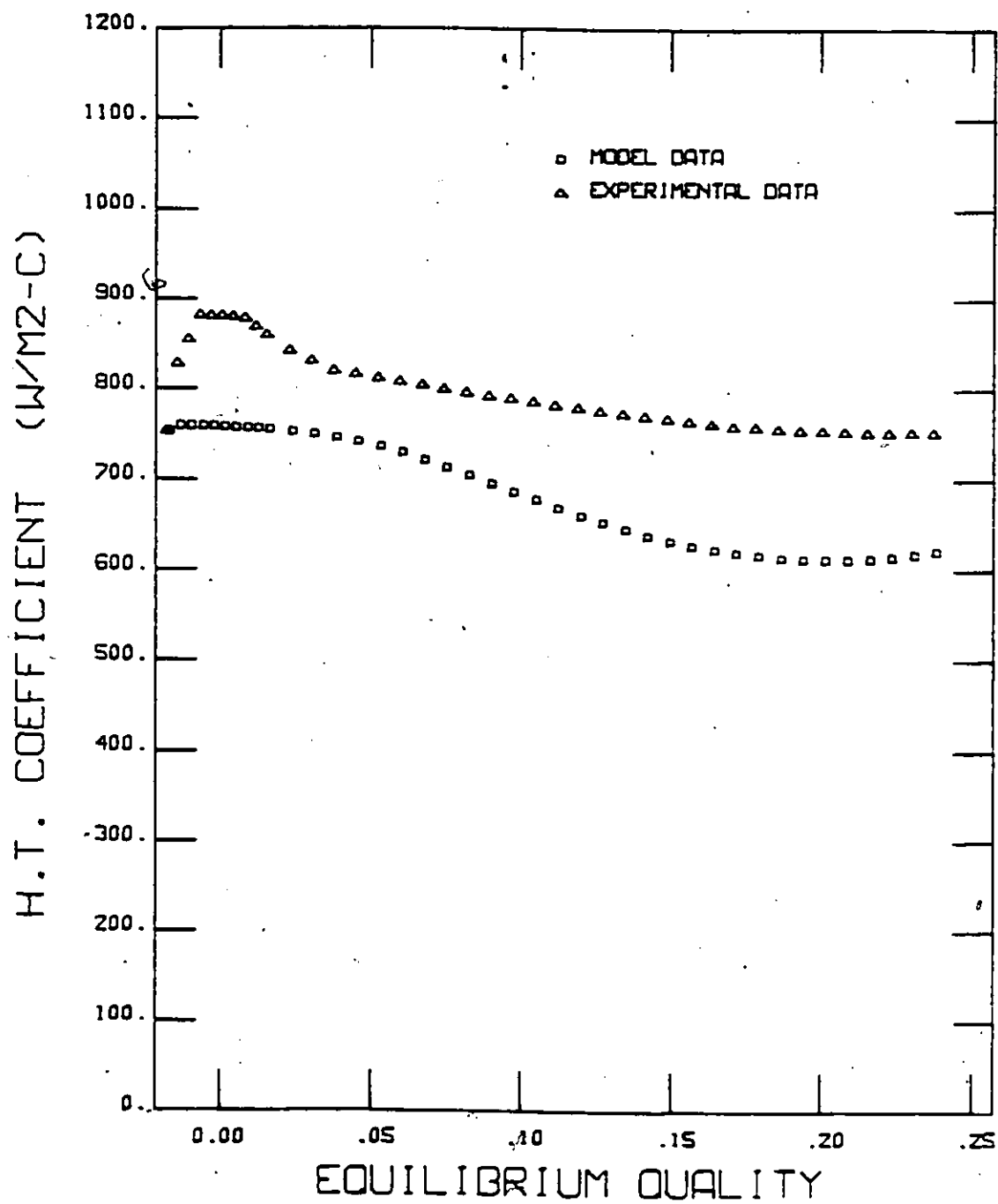


FIGURE IV.10 Predicted and experimental heat transfer coefficient
 ($P = 8 \text{ MPa}$, $G = 523 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 121 \text{ kJ.kg}^{-1}$,
 $\phi = 249 \text{ kW.m}^{-2}$)

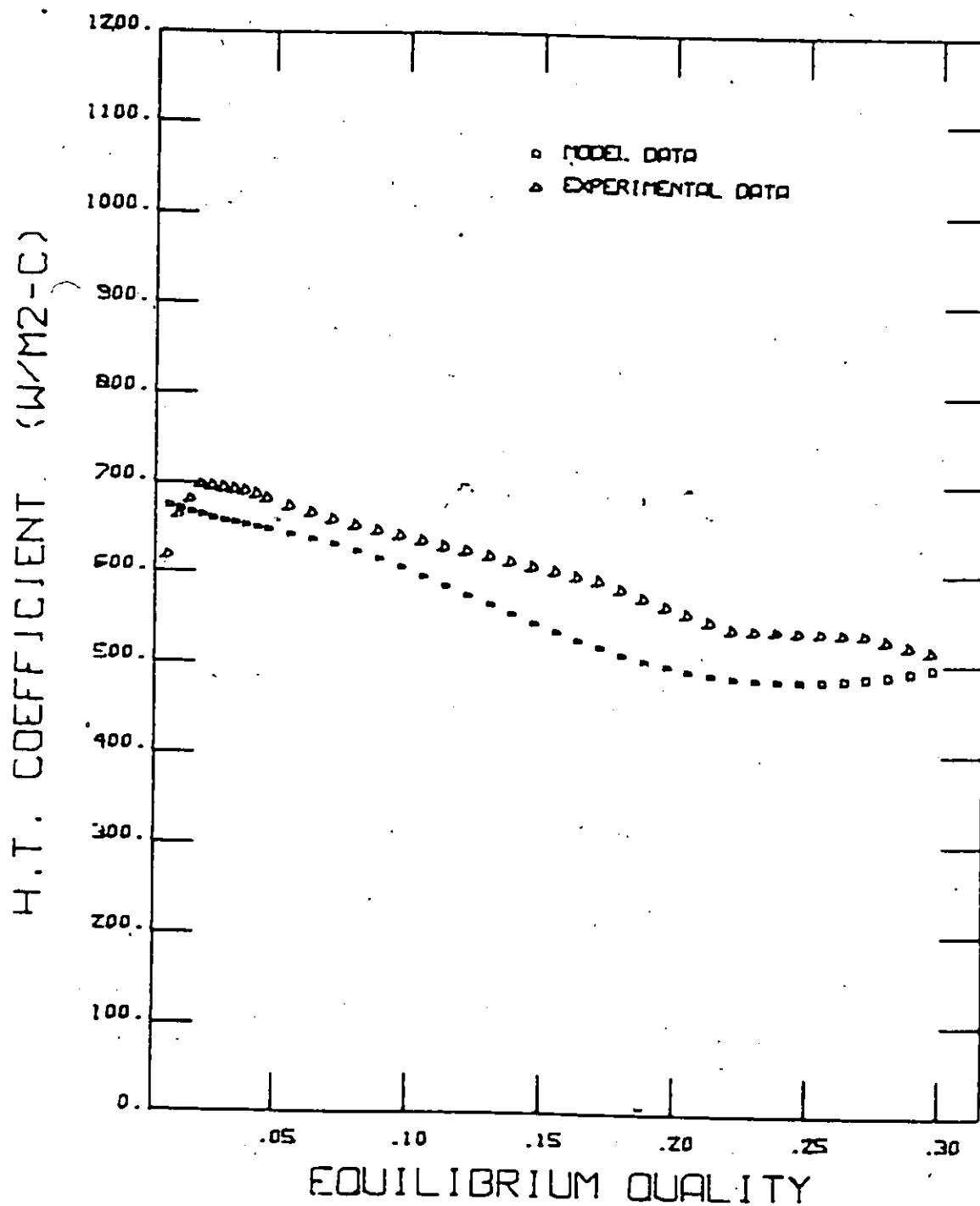


FIGURE IV.11. Predicted and experimental heat transfer coefficient
 ($P = 6 \text{ MPa}$, $G = 370 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 112 \text{ kJ.kg}^{-1}$,
 $\phi = 216 \text{ kW.m}^{-2}$)

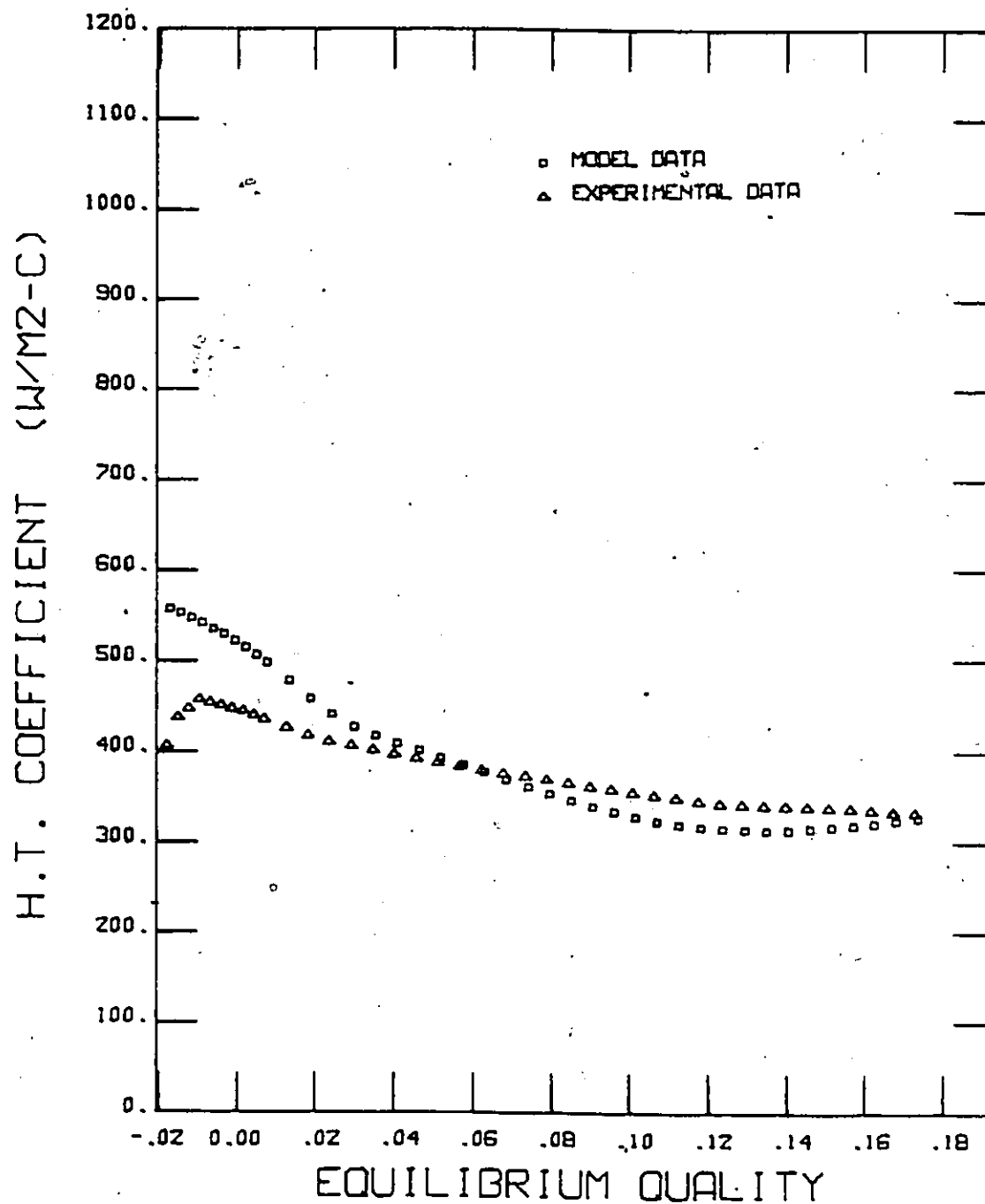


FIGURE IV.12. Predicted and experimental heat transfer coefficient
 ($P = 2 \text{ MPa}$, $G = 357 \text{ kg.m}^{-2}.\text{s}^{-1}$, $\Delta H_{in} = 133 \text{ kJ.kg}^{-1}$,
 $\phi = 165 \text{ kW.m}^{-2}$)

^

V. EXPERIMENTAL DESIGN

V.1. General

✓ The hot patch method[†] was initially used by Groeneveld [1972] with Freon. This method is used to generate stable film boiling. Initially the hot patch is heated above the Leidenfrost temperature with zero flow in the test section. When the flow is introduced, the thermal inertia of the hot patch prevents the rewet front from propagating downstream. Stable film boiling exists underneath the hot patch and the dry patch spreads downstream as the test section power is increased. The tube is directly electrically heated and the heat flux is constant over the length of the tube.

Since no water post-dryout data was available at low flows and low qualities, this technique was also used by Fung [1981] and Stewart [1981] over a moderate range of conditions. Nijhawan [1980] and Gottula et al. [1983] measured the vapour temperature also using this method.

✓ Stable film boiling has only been obtained at low and moderate flow. In this experiment stable film boiling, which is more difficult to achieve at elevated flows and subcooling, was maintained over a wider range of mass flow rates and inlet subcooling. The hot patch was occasionally operated above the melting point of the solder causing a deterioration of the silver solder joint and eventually led to the failure of the test section. Because of the above problem, successful operating conditions are given in Table 2.

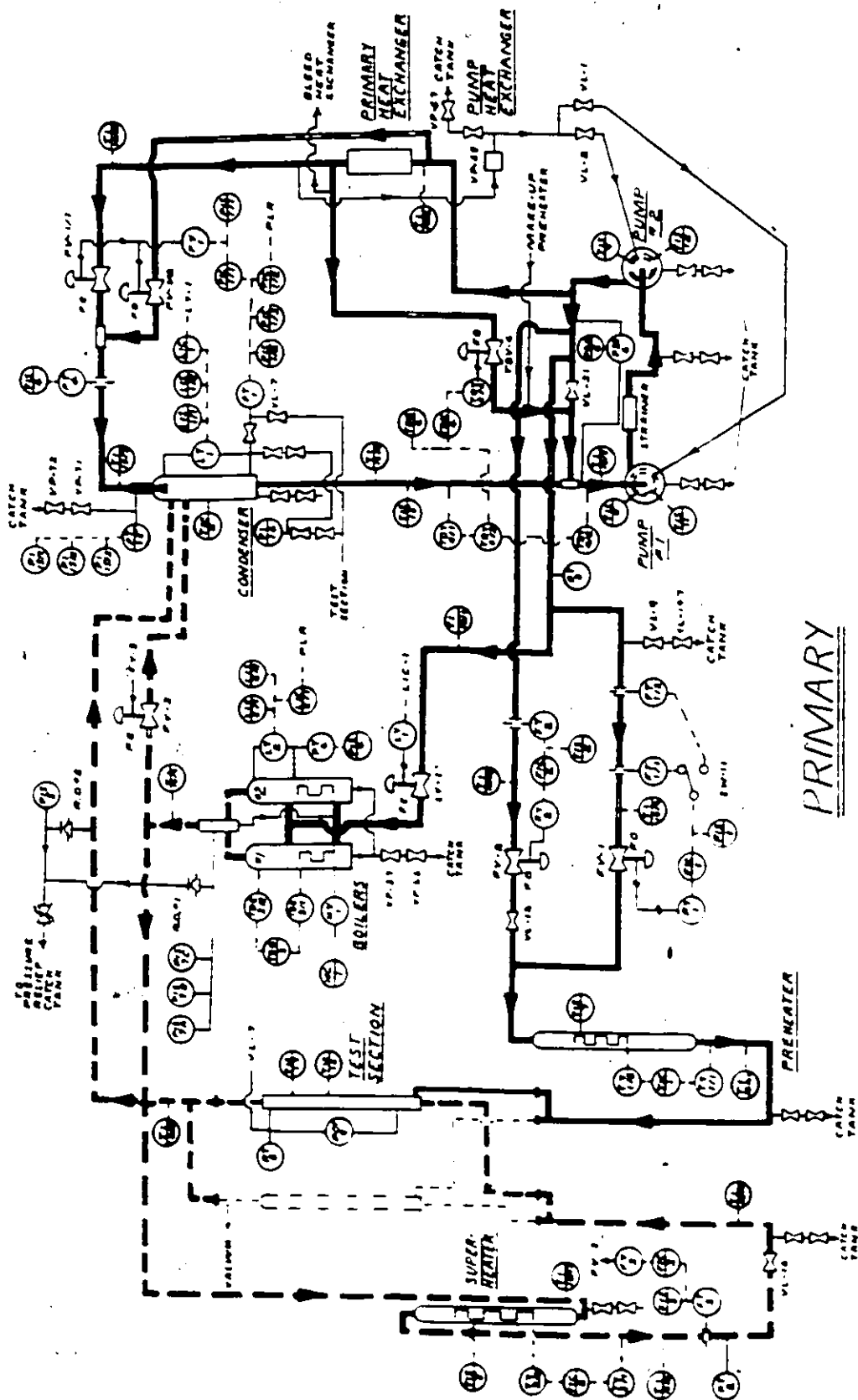
[†] a copper block silver soldered on a tube.

V.2 Experimental Equipment

V.2.1 Experimental loop

The experiment was conducted in the MRI FLARE loop (Fog Loop Advance Reactor Engineering) of the Chalk River Nuclear Laboratories. A schematic drawing of the loop is shown in Figure V.1. The loop is filled with deionized distilled water. The pressure is initially obtained using the make-up system. The water is then circulated and the loop is vented through the top of the condenser to release non-condensables (such as air etc.). Heat is then added to the loop through the electrical boiler. As the temperature of the water reaches saturation, vapour starts to build up in the condenser and the boiler. The level of the condenser and the level of the boiler then decrease. Adjustment of flow is made using control valve FT 1/1 or FT 1/2 depending on the flow range. Pressure adjustment in the loop is obtained using the condenser. More spray water (valve PV-1/1 and PV-1/2) in the condenser will decrease the loop pressure. Test section inlet subcooling can be adjusted using control valve TDV-4. More water flows through the primary heat exchanger as valve TDV-4 is opened. This subcooling is also necessary since it avoids pump cavitation. The inlet temperature to the test section can be fine tuned using a manual adjustment of the preheater. The loop is provided with the following instruments:

- flow control, level controls, temperature controls, pressure controls.
- alarms and trips under adverse conditions to prevent any damage to the loop's equipment, (for example, low pump subcooling, low condenser level, low boiler level, high test section wall temperature). A picture of the control panel is shown in Figure. V.2.



PRIMARY

FIGURE V.1. MRI Flare Loop diagram

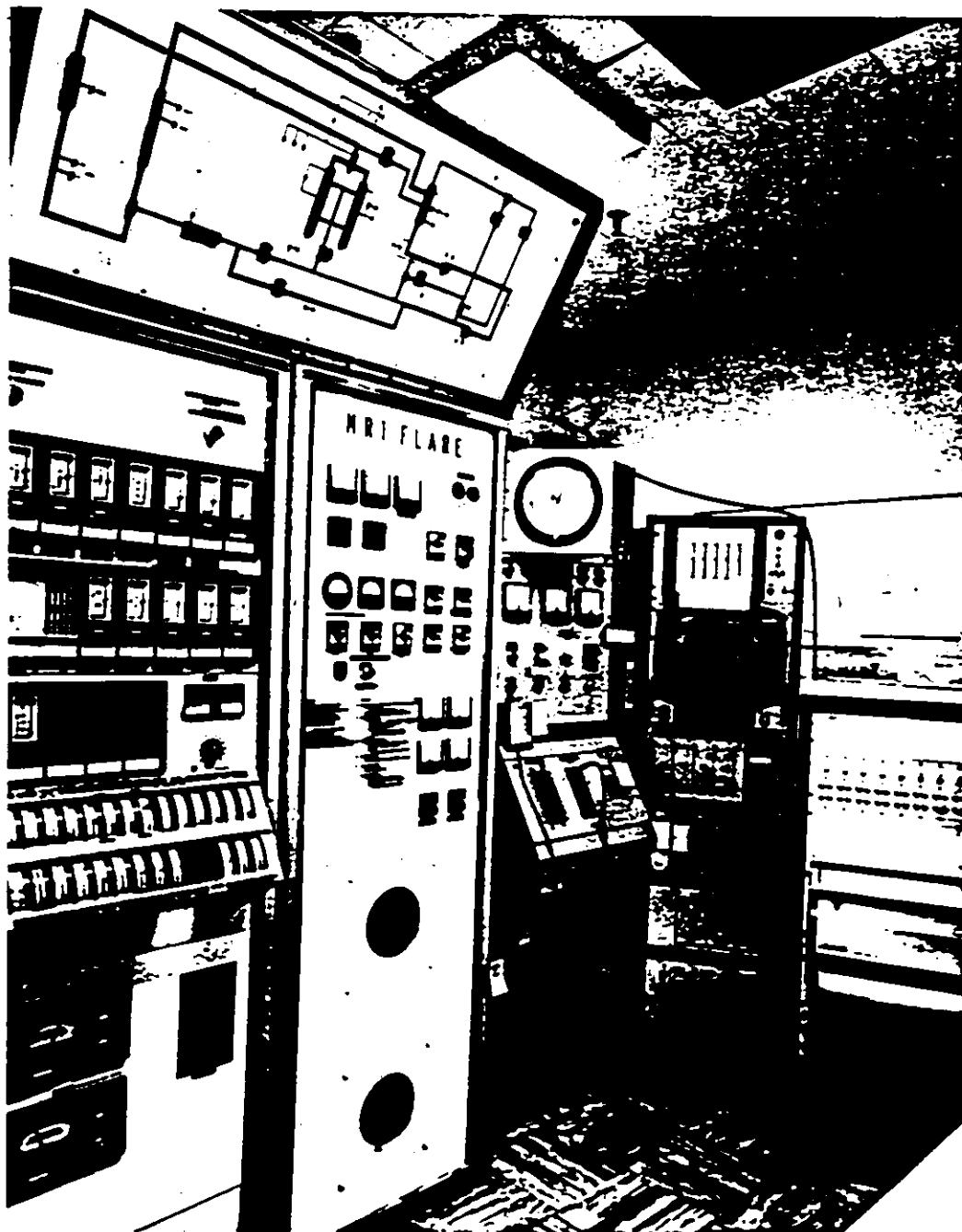


FIGURE V.2. MRI FLARE LOOP control panel

V.2.2: Flow measurement

The flow was measured using calibrated orifice plates. A turbine flowmeter could not be used since the loop temperature is too high. Foxboro $dp^\dagger$ cells (100 inches of water) were used across each orifice plate since the pressure difference is proportional to the flow (the discharge coefficient of the orifice plate is known from calibrations). Correction factors for water temperature through the orifice plate are also included. The maximum flow allowable for flowmeter FT 1/1 is 1400 lbs/hour and 3600 lbs/hour for flowmeter FT 1/2. For better accuracy each flowmeter was never used below one third of the maximum permissible flow and the zero adjustment of the dp cell was done daily.

V.2.3 Test section power supply

The test section was heated by a 1000 amperes and 50 volts DC (direct current) power supply. Maximum power could not be obtained since the resistance of the test section was not perfectly matched to the power supply. Power supply leads are connected to the test section using two semicylindrical grooved silver plated copper plates. A copper compound was also put between the clamps and the test section to avoid oxidation and also provide a better electrical contact.

The heat loss from the test section is expected to be considerable through the power clamps and heat balance runs were obtained as shown in Appendix E. Power was evaluated by measuring the voltage across the two test section power clamps and the voltage across a known

$\dagger dp$ differential pressure

shunt resistor (to calculate the current). Voltage was reduced from 50 V to 50 mV (1:1000) through a divider before being fed to a scanning computer. Multiplication of voltage and current is done by the on-line computer's program.

V.2.4 Test section preheater

The preheater was used for the positive inlet quality runs. A 350 kW DC power supply was directly heating a 0.514 inch (Inconel 718) outside diameter tube. The preheater and the test section were electrically insulated using asbestos phenolic gaskets. Thermocouples were installed at the preheater outlet and connected to a high temperature power supply trip. The preheater power supply provides its internal power value and measurement of voltage and current is not needed. Heat balance runs are shown in Appendix E.

V.2.5 Pressure measurement

Test section outlet pressure was measured by a calibrated pressure cell. The range of the pressure cell was from 0 psia to 1400 psia for a corresponding output of 0 - 50 mV. Since the elevation of the outlet test section pressure taps was above the elevation of the pressure cell, correction for static head was made.

The pressure drop across the test section was measured using a Foxboro dp cell. This dp cell has been calibrated to provide 10-50 mV for 0-8 psi pressure drop. Zero adjustment of this dp cell was also performed daily. A Shaevitz higher range dp cell (0-25 psig) was also used

to measure the pressure drop. Two signals, corresponding to both sides of the dp cell, were fed to a Wheatstone bridge. Excitation voltage and balance of the bridge were adjusted to get 25 mV full scale.

V.3 Test section

V.3.1 Hot Patch

A 10.1 cm long by 8.98 cm outside diameter hot patch was silver soldered on the test section. (Drawing is shown in Figure V.3.) Twelve or eight 500 watt heaters, depending on the test section, were installed in the axial direction of the hot patch. Two auto transformers (220 volts, 20 amperes each) were used as power supplies, each controlling half of the heaters. The heaters were connected in such a way to match the resistance of the power supply. Figure V.4 shows the hot patch installed in the loop. The hot patch power was measured using two Westinghouse VP-840 Watt transducers. The hot patch temperature was measured at three different locations: no appreciable variation in temperature was found.

V.3.2 Test section

The test section was made of an instrumented 1.11 cm OD, .8928 ID Inconel 600 tube. Each thermocouple is slid in a MgO sleeve and bent. The tip of the Chromel-alumel K type thermocouple is spot welded to the test section. To attach the thermocouple to the test section, two straps are put around the MgO sleeve and spot welded to the test section (Fig. V.5). The thermocouples used were not calibrated since bending and

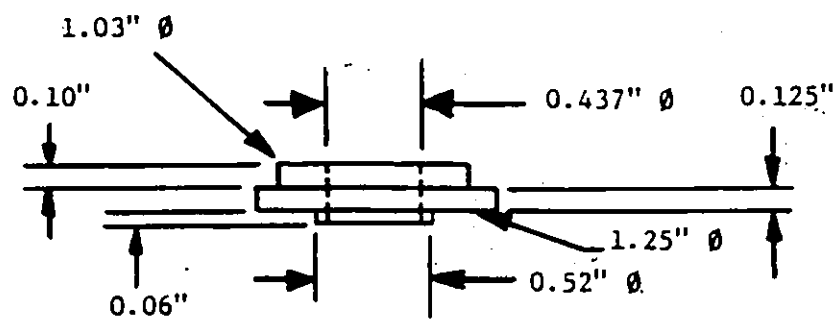
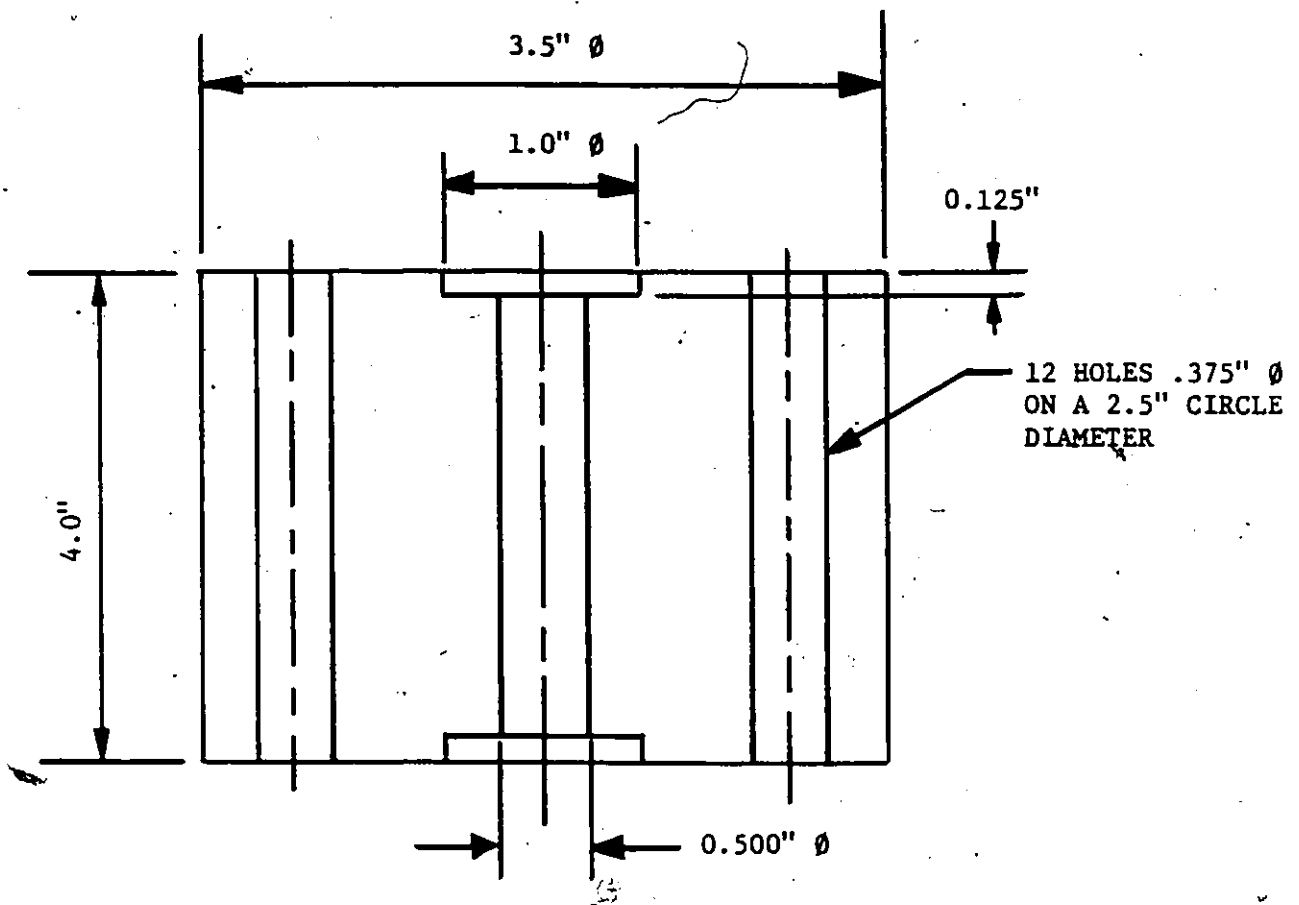


FIGURE V.3. Hot patch

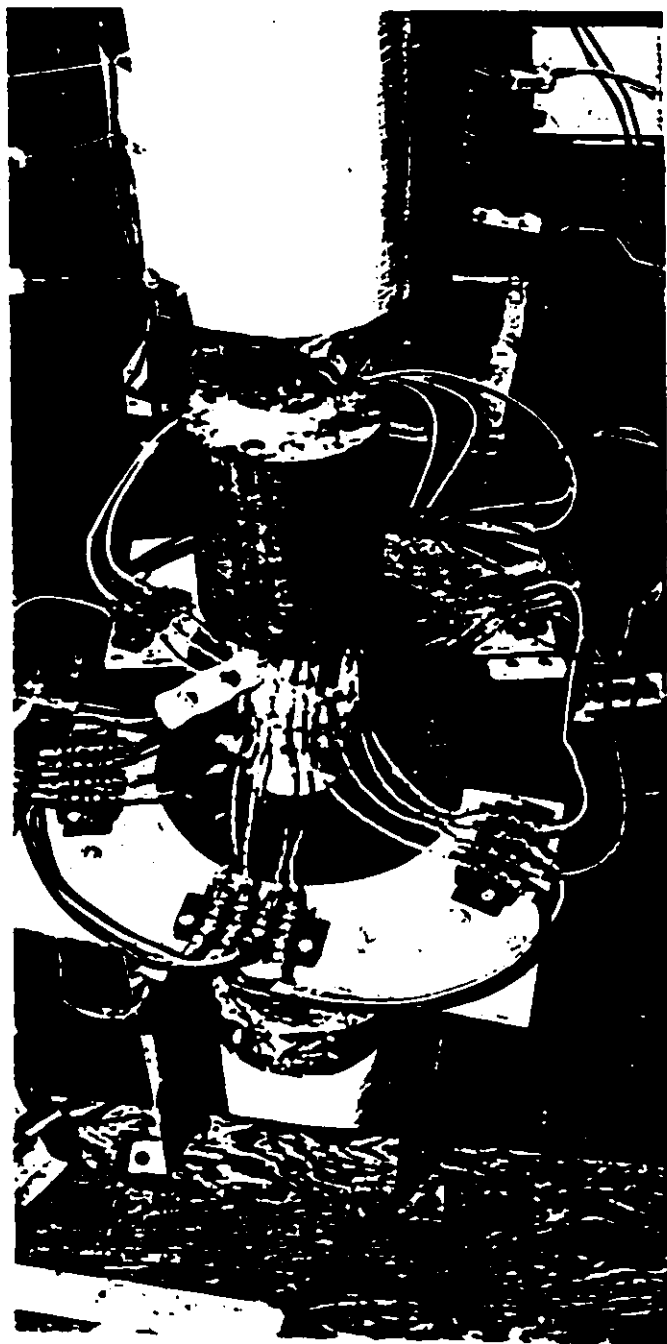


FIGURE V.4. Hot patch installed in the loop

spot welding the thermocouple effects the calibration. The thermocouple resistance to ground was however checked using a 50 Volt ohmmeter. Thermocouples of resistance lower than 20 megohms were not used. The resistance was also checked after the thermocouples were installed on the test section. Thermocouples were connected to a ice reference point using Chromel-alumel extension wires. Copper wires were used from the ice reference point to the analog digital converter (ADC). Thermocouples were also grounded to prevent noisy signals.

The test section was insulated with ceramic fiber in contact with the test section and with fiberglass in contact with ambient air. Ceramic fiber was used since it can sustain higher temperatures. The inside diameter of the fiberglass insulation was 2.5 inches and the outside diameter 4.5 inches. Aluminium foil was wrapped around the insulation close to the hot patch to prevent the insulation from burning (Figure V.4). An overall view of the test section installed in the loop (without hot patch insulation) is shown in Figure V.6.

V.3.3 Test section properties

Since the dimensions are given for ambient temperature, corrections are made for the thermal expansion of the test section. The subroutine DIMENS (see Appendix F) uses the average wall temperature of the test section to calculate the thermal expansion. The method of calculating the inside wall temperature from the measured wall temperature is presented in Appendix G. A correction is also made for the variation of wall conductivity with temperature. A fit has been developed from

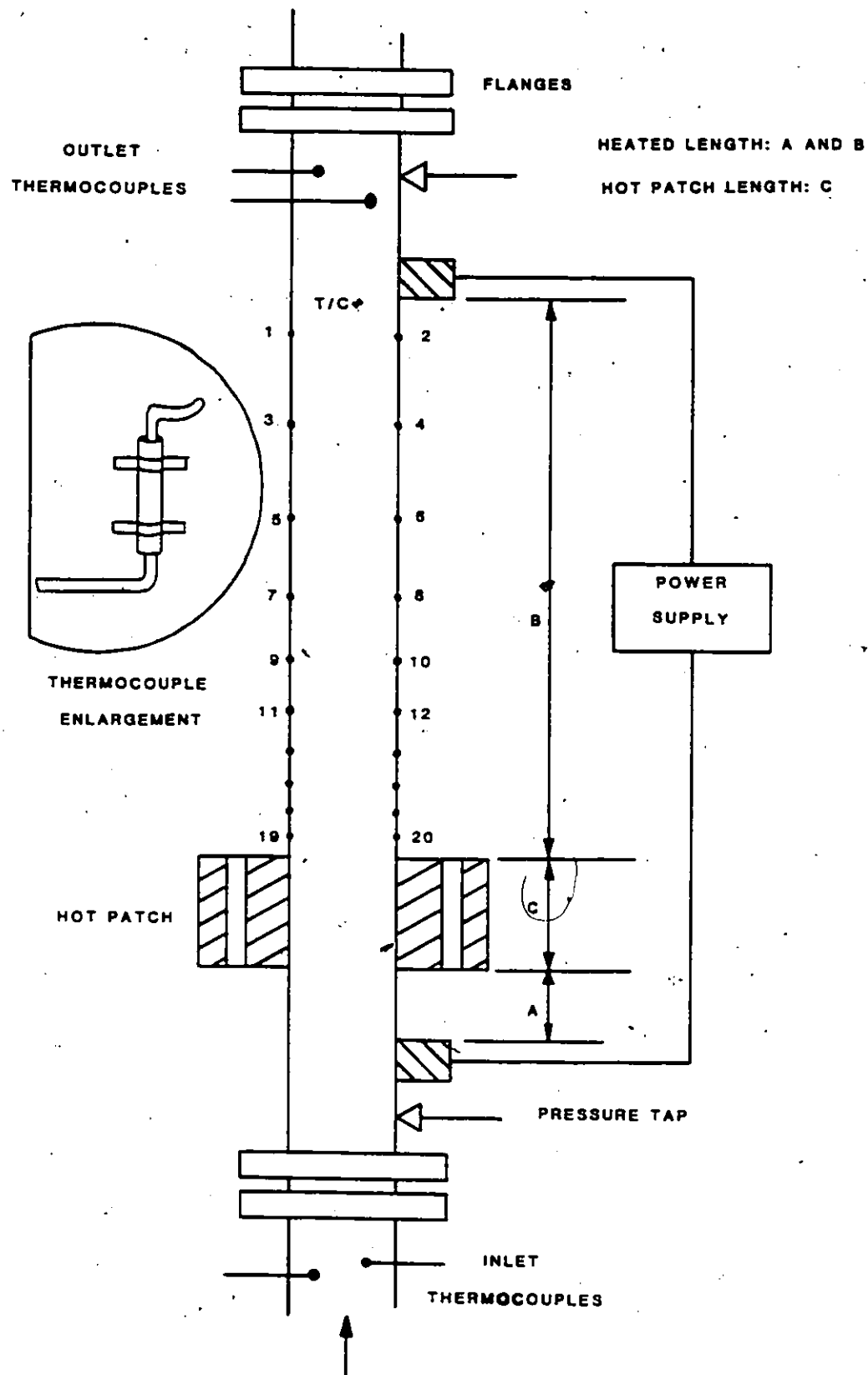


FIGURE V.5. Test section instrumentation

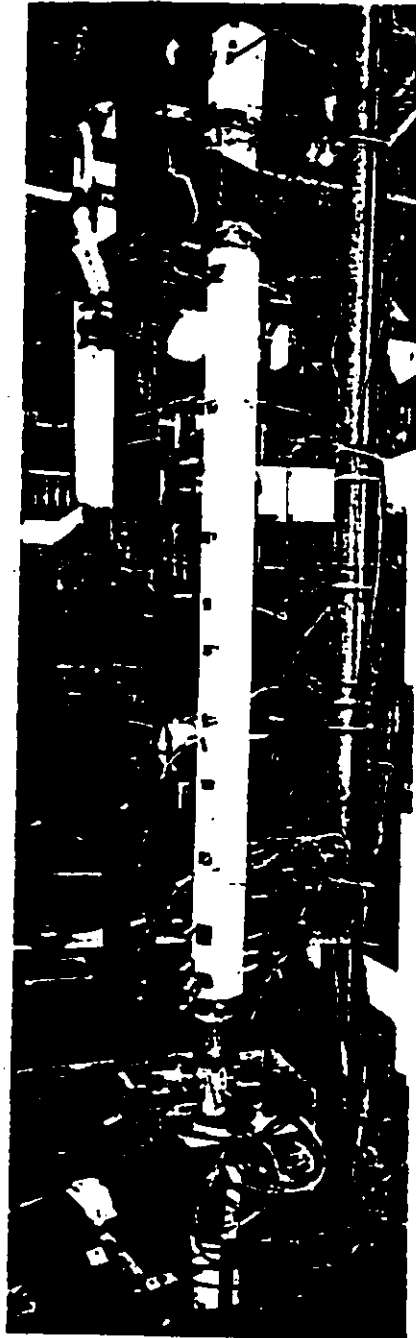


FIGURE V.6. Test section installed in the loop
(without hot patch insulation)

manufacturer's data [Huntington alloy, 1965] and is shown in the subroutine CONDUCT of Appendix F.

A sketch of the test section is shown in Figure V.5. Since the location of the thermocouples varies for the different test sections, Table 3 provides the information for a specific test section.

V.4 Data Acquisition

The signals from the instruments were fed to an analog digital converter (ADC). An on line computer program calculated the required parameters (Temperature, flow, power, pressure) on demand and displayed them on a CRT. The data were stored on magnetic tape for further processing. A temperature pen recorder was also used.

V.5 Experimental Procedure

- 1) The loop pressure was adjusted to the desired value.
- 2) The water was preheated to near saturation. This makes it easier to attain stable film boiling. The hot patch power is also turned on at this time. Preheating of the hot patch is useful since it decreases the time to reach 750°C preventing the water from cooling down too much.
- 3) Flow is valved out using control valve FV-1 and the hot patch is heated up to 750°C at almost full power.

- 4) As the hot patch temperature reaches 750°C , the control valve is gradually reopened. Final adjustment of flow, inlet positive quality or inlet subcooling and pressure is made. For high mass flux, test section power had to be turned on as the flow was reopened to prevent the hot patch from quenching.
- 5) The test section power is then increased and several scans are taken during the dry patch spreading at a constant test section power.
- 6) When all the thermocouples are in dryout, the power is increased until the maximum test section temperature reaches about 760°C .
- 7) Power is then decreased slightly and scans are taken until the minimum film boiling temperature is reached.
- 8) Fast scans are obtained during the wet front propagation. Scans of pressure drop are also taken after the test section and the hot patch have been quenched.

VI. EXPERIMENTAL RESULTS

VI.1. General

The heat transfer coefficient was calculated at each thermocouple location by using the inside wall temperature calculated from the outside wall temperature. This heat transfer coefficient is based on the inside wall temperature minus the saturation temperature evaluated at the outlet pressure. The heat transfer coefficient was plotted versus the equilibrium thermodynamic quality (calculated as shown in the Appendix H). The run number RN described in the appendix I and PHI the heat flux in kW/m^2 are also specified on the plots. All the corrected experimental results are presented in Appendix I.

The following corrections were applied:

- 1) Thermal expansion of the test section using the test section average wall temperature (Appendix F).
- 2) Corrected inside wall temperature using the measured outside wall temperature (Appendix G).
- 3) The inlet quality was corrected due to heat loss through the test section assuming that half of the heat occurs at the bottom clamp. The measured preheater power was corrected as shown in Appendix E. The following equation was used to correct X_{in} :

$$(X_{in})_{\text{corrected}} = X_{in} - \frac{POWLOSST}{(2 H_{fg} W)}$$

X_{in} : Inlet quality at the test section inlet thermocouples.

$(X_{in})_{\text{corrected}}$: Inlet quality downstream of the bottom power clamp.

POWLOSST: Power loss through the test section calculated from equation E.1. (Appendix E).

Equation E.1 is calculated by using the ambient temperature and the subcooled inlet liquid temperature for subcooled water, or the saturation temperature for the positive inlet quality.

- 4) The test section inlet quality for subcooled water was calculated using the corrected inlet temperature. (TINC as shown in Appendix F.)
- 5) The heat flux and the mass flux were calculated using the corrected dimensions of the test section due to thermal expansion.

In this chapter, some experimental results will be discussed. The results were plotted as curves of heat transfer coefficient versus equilibrium quality. However, it should be remembered that the axial conduction of the hot patch affects the reading of the adjacent test section thermocouples which explains the initial increase of heat transfer. The axial conduction in the test section wall between the upper thermocouple and the upper power clamp also affects the upper thermocouple reading. The questionable upper thermocouple reading was removed from the plots.

VI.2. Mass Flux Effects

The effect of mass flux is shown in Figures VI.1, VI.2 and VI.3. The low subcooling runs (9113, 9213, 9313 and 9413) as well as the positive inlet quality runs (Figure IV.2 and Fig. VI.3) show a strong effect of mass flux. The heat transfer coefficient decreases initially followed by an increase. As discussed by Stewart [1981], there are three mechanisms involved:

- 1) vapour superheat: a large degree of vapour superheating will mean a lower heat transfer coefficient. This is most unlikely to occur at low heat flux.
- 2) vapour mixing: The conductive heat transfer dominates initially but high vapour velocity leading to high Reynolds number will improve the convective heat transfer.
- 3) vapour layer thickness: A thinner vapour film will mean a better conductive heat transfer coefficient in the inverted annular flow.

The initial decrease of heat transfer with increase in quality may probably be explained by the increase of the vapour layer thickness (i.e. mechanism 3). A minimum is reached and subsequently the heat transfer coefficient increases due to a much better convective heat transfer coefficient. The rate of increase of heat transfer with quality is even larger at higher mass flux probably due to a higher vapour velocity as well as better vapour mixing and lower vapour superheat. The initial decrease of heat transfer with quality occurs over a longer test section length for the high mass flux run than the low mass flux run (Run

9413 and Run 9113 in Figure VI.1 for example). This is the result of the test section experiencing Inverted Annular Film Boiling over a much longer length at higher mass flux.

A thinner vapour film might result in a higher heat transfer coefficient due to a better conduction. The minimum value of heat transfer could correspond to the transition from inverted annular flow to dispersed flow. This minimum occurs at a high equilibrium quality for some runs which correspond also to a high void fraction as discussed in section II.3.

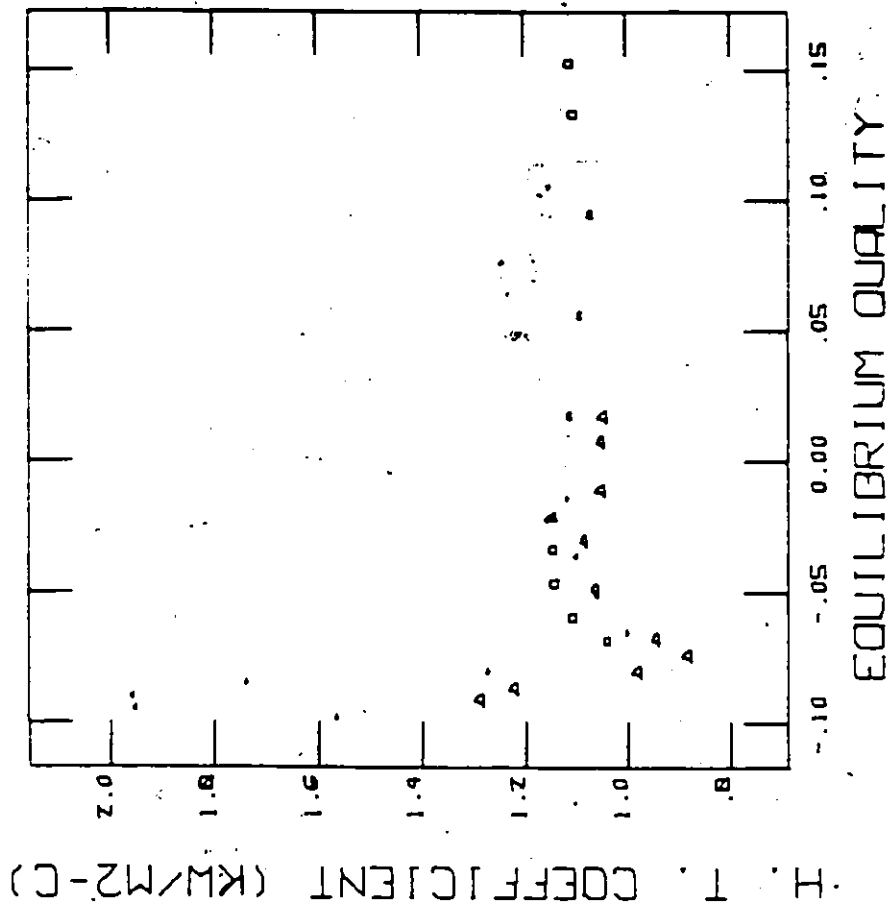
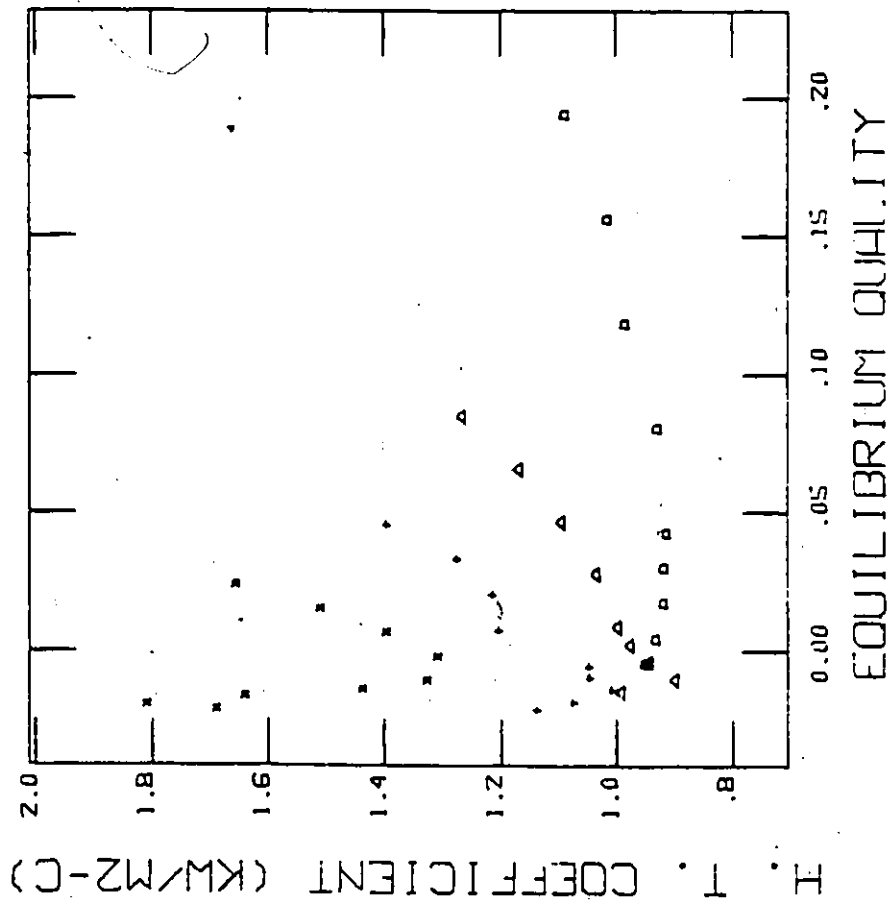
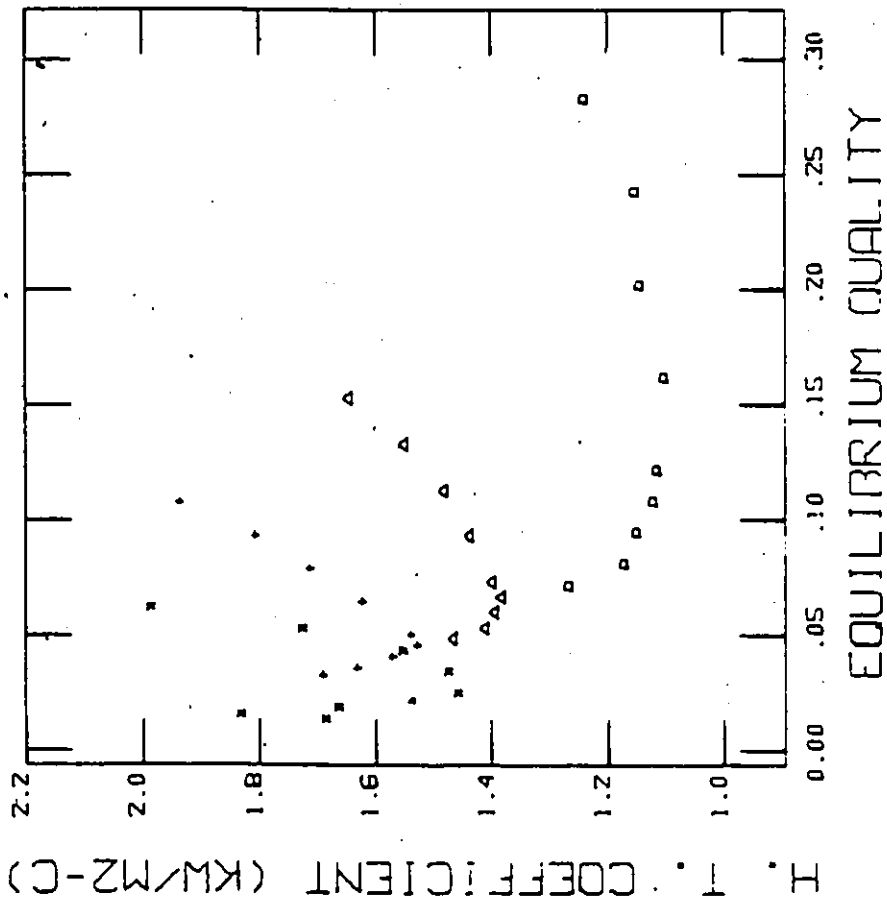


FIGURE VI.1 The effect of mass flux (50 kJ.kg⁻¹ and 150 kJ.kg⁻¹)
 on the heat transfer coefficient.

○ RN:9153 PHI:401.
 △ RN:9253 PHI:386.
 · RN:9353 PHI:387.
 × RN:9443 PHI:413.



○ RN:9153 PHI:401.
 △ RN:9253 PHI:386.
 · RN:9353 PHI:387.

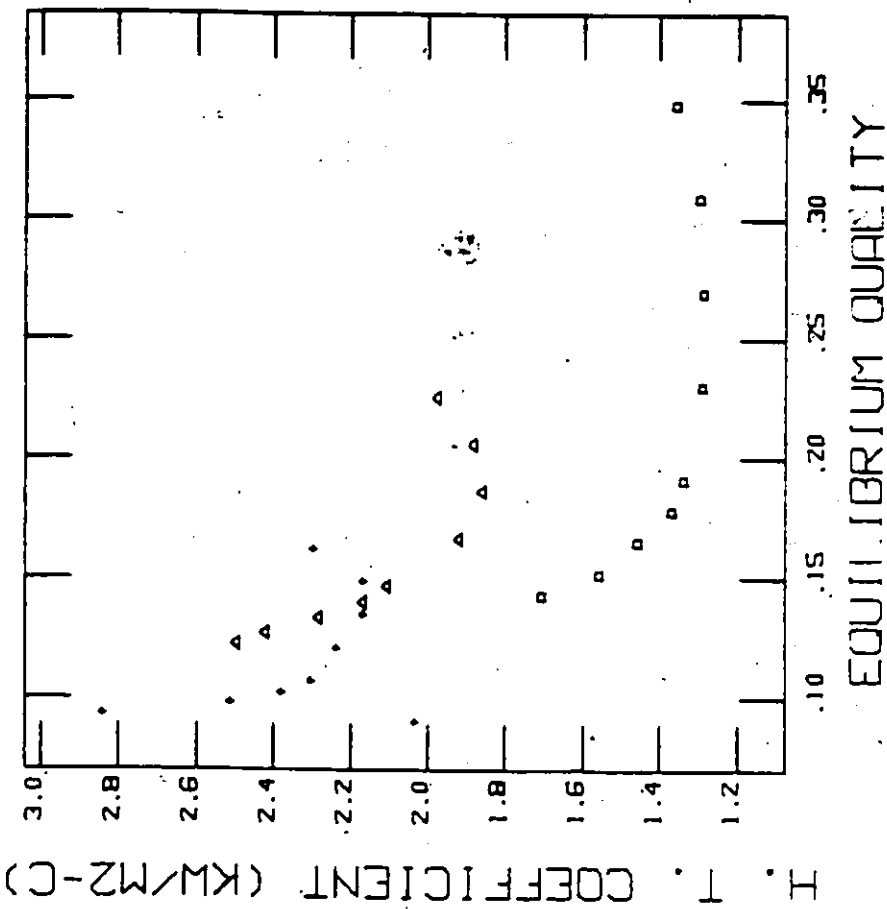


FIGURE VI.2. The effect of mass flux (0.0 Inlet quality and 0.08 inlet quality) on the heat transfer coefficient.

□ RN:9163 PHI:408.
 △ RN:9263 PHI:381.
 • RN:9163 PHI:381.

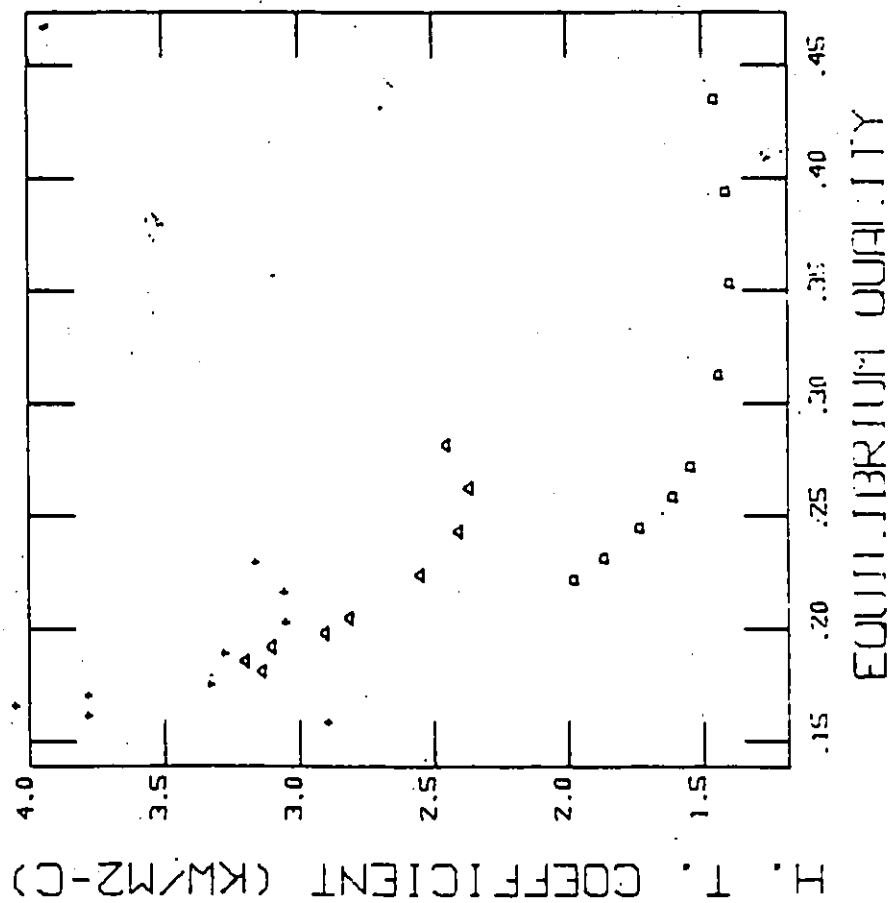


FIGURE VI.3. The effect of mass flux (0.15 inlet quality)
 on the heat transfer coefficient.

VI.3 Inlet Enthalpy Effects

VI.3.1 Subcooled inlet.

The effect of inlet quality is plotted in Figures VI.4 to VI.7 for the positive inlet quality as well as the subcooled inlet. It was found that the low mass flux subcooled inlet runs (9113, 9123, 9133) show a stronger effect than the higher mass flux runs. (9213, 9223, 9313, 9323). For low mass flux and for very low inlet subcooling (Run 9113), the heat transfer coefficient increases with equilibrium quality. The trend of the data is opposite for higher subcooling (Run 9133). This trend can probably be explained by two opposite effects: superheating of the vapour and the change from conductive heat transfer to convective heat transfer. This is summarized in Table 4. The superheating of the vapour is higher at low inlet subcooling (Run 9113) in the upstream part of the test section, which may explain the lower heat transfer coefficient. The total heat flux from the wall is divided into two parts as explained previously (equation B.44): removed of the subcooling and vapour generation at a rate Γ_v . The vapour generation is larger at low subcooling and this will result in change from a conductive heat transfer to a convective heat transfer sooner. However, the removal of subcooling is stronger for a high subcooling inlet (Run 9133) and the vapour film is thinner. Little convective heat transfer occurs due to this very thin vapour film and the low vapour velocity. The increase of vapour superheat with quality is also expected to be higher at high subcooling since the initial superheat of the vapour is almost non-existent. It can also be observed that the curves are converging at an approximate quality of 0.25 for 1000

$\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$. After this value, the effect of inlet conditions on the vapour superheat has disappeared. The heat transfer coefficient for the subcooled inlet for $2000 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ and $2750 \text{ kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ shows the same trend as described in the section VI.2. It is observed that the heat transfer coefficient of the higher inlet subcooling decreases over a longer test section length than the lower subcooling inlet (Figure VI.4). Conductive heat transfer exists over a longer length in the case of the high subcooling inlet.

VI.3.2 Positive Inlet Quality

The effect of quality for the positive inlet quality is plotted in Figures VI.5, VI.6 and VI.7. It was observed that the heat transfer coefficient is much better for a higher inlet quality. This might be explained by a higher vapour velocity for a greater inlet quality. The decrease of heat transfer with equilibrium quality is probably the increase of vapour superheat. The convective heat transfer improves in the downstream part of the test section where the vapour superheat has levelled off. The initial decrease might also be explained by an "extrance effect" which is not well understood. This effect occurs over a longer length for a higher positive inlet quality (Run 9143 and Run 9163). This might suggest that it takes a longer distance to the heat transfer coefficient to establish for a higher inlet void fraction.

A straight line was drawn in Figure VI.6 for 2750 $\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$. The slope is approximately 10 for this mass flux compared to 5 for 2000 $\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$ and 2 for 1000 $\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$. The increase of heat transfer is even greater for 4500 $\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$. This shows that the rate of increase of heat transfer is strongly dependent on the mass flux. The change in slope with mass flux is not linearly proportional to the increase of mass flux due to a much better convective heat transfer and a lower vapour superheat.

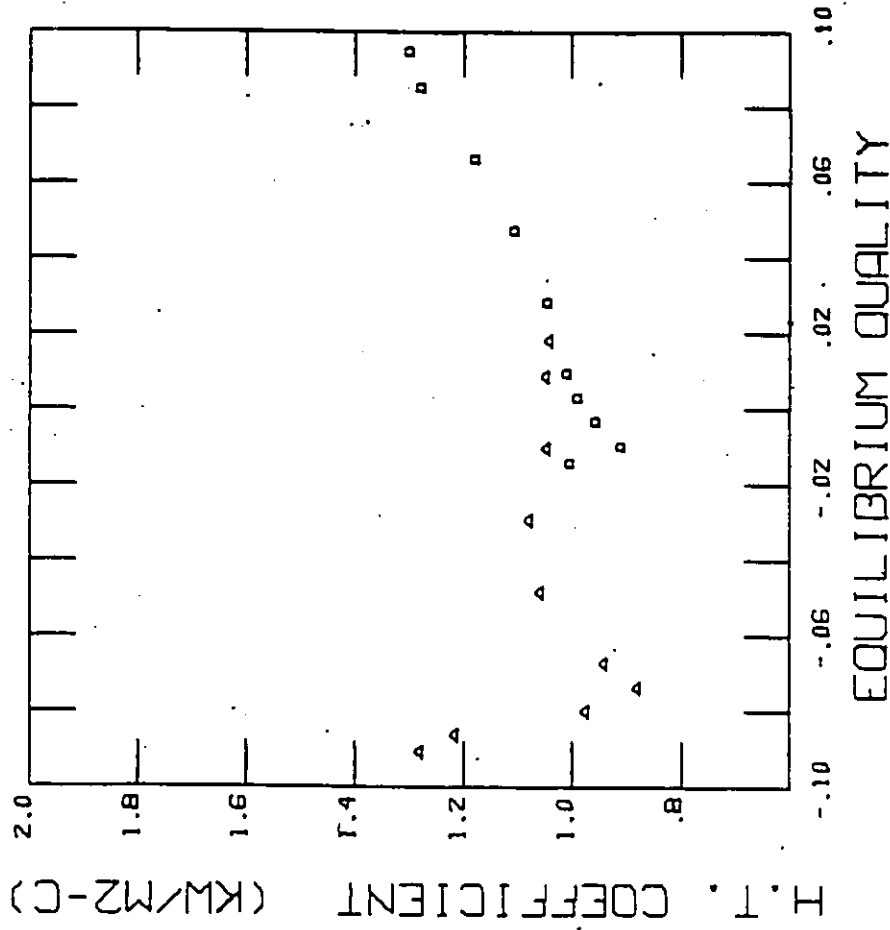
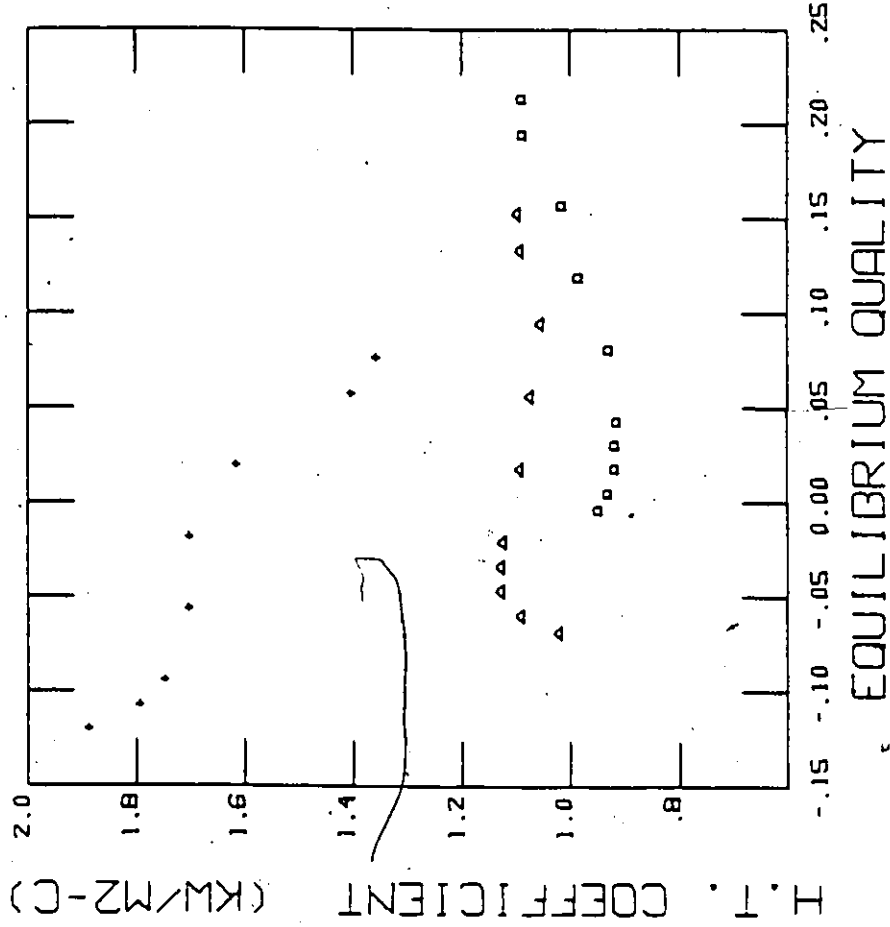
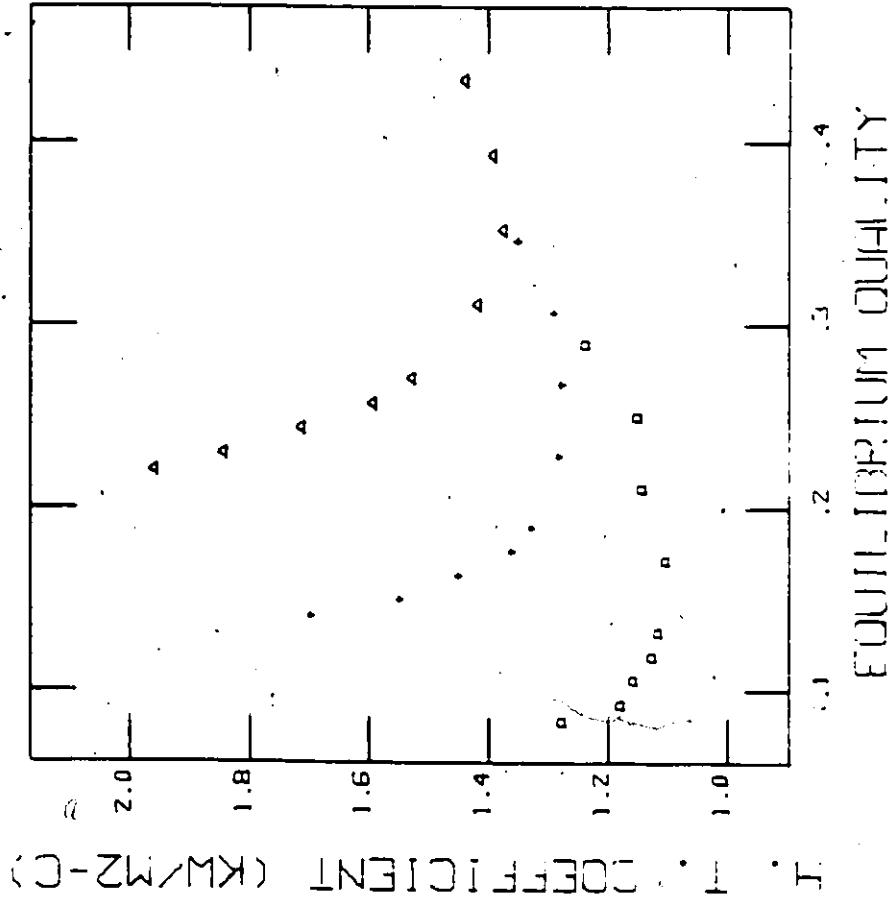


FIGURE VI.4. Effect of inlet subcooling ($G = 1000 \text{ kg.m}^{-2}.\text{s}^{-1}$ and $G = 2000 \text{ kg.m}^{-2}.\text{s}^{-1}$) on the heat transfer coefficient.

○ RN,9143 PHI,391.
 △ RN,9163 PHI,408.
 • RN,9153 PHI,401.



○ RN,9323 PHI,416.
 △ RN,9313 PHI,413.

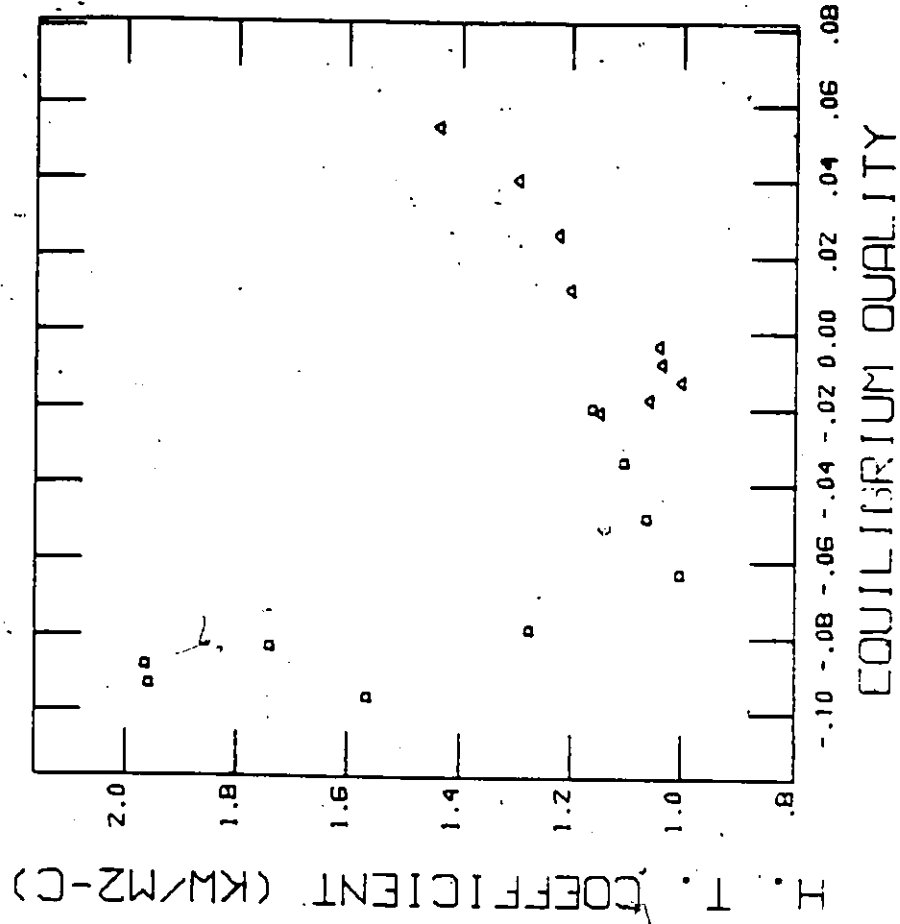
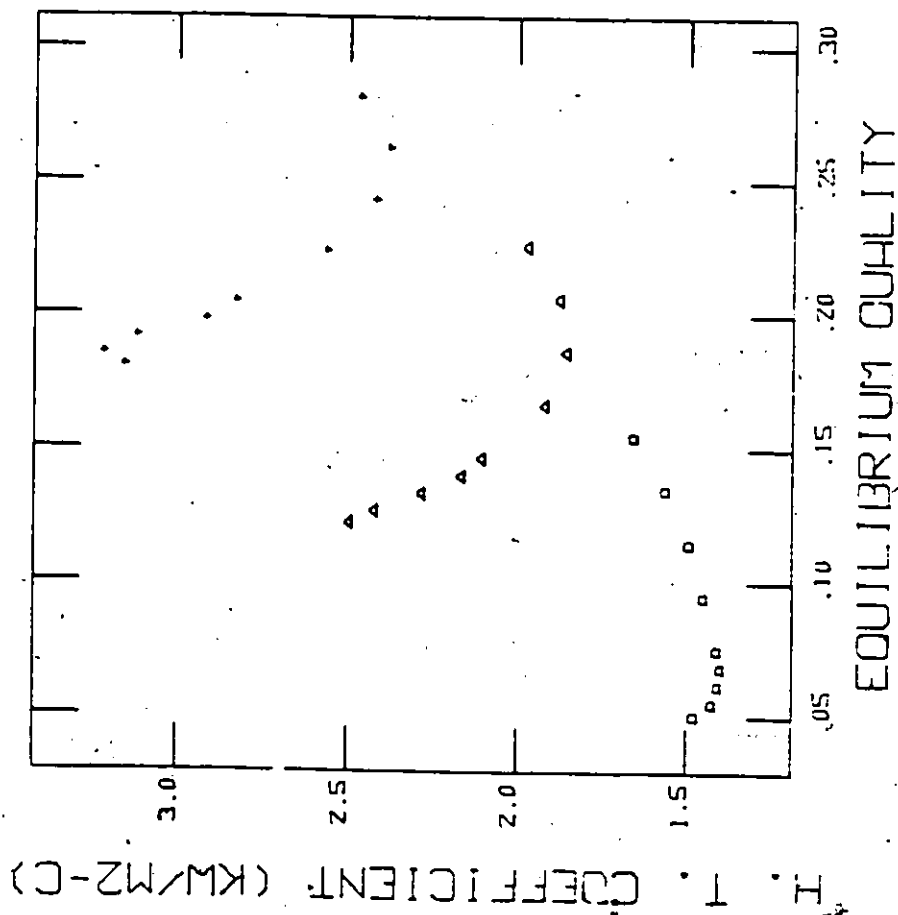


FIGURE VI.5. Effect of positive inlet quality ($G = 1000 \text{ kg.m}^{-2}.\text{s}^{-1}$)
 and effect of inlet subcooling ($G = 2750 \text{ kg.m}^{-2}.\text{s}^{-1}$)
 on the heat transfer coefficient.

◻ RN:9243 PHI:392.
 ◻ RN:9253 PHI:386.
 • RN:9263 PHI:381.



◻ RN:9343 PHI:387.
 ◻ RN:9353 PHI:387.
 • RN:9363 PHI:381.

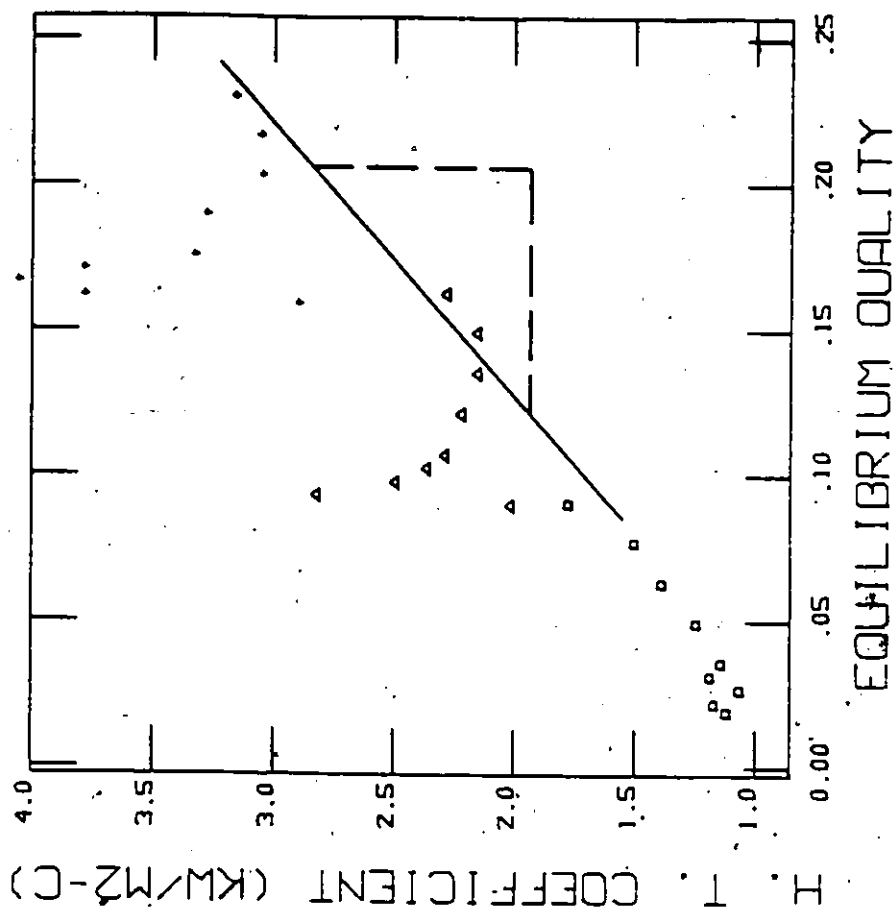


FIGURE VI.6. Effect of positive inlet quality ($G = 2000 \text{ kg.m}^{-2}.\text{s}^{-1}$ and $G = 2750 \text{ kg.m}^{-2}.\text{s}^{-1}$) on the heat transfer coefficient.

◻ RN:9443 PHI:413.
 ◻ RN:9413 PHI:422.

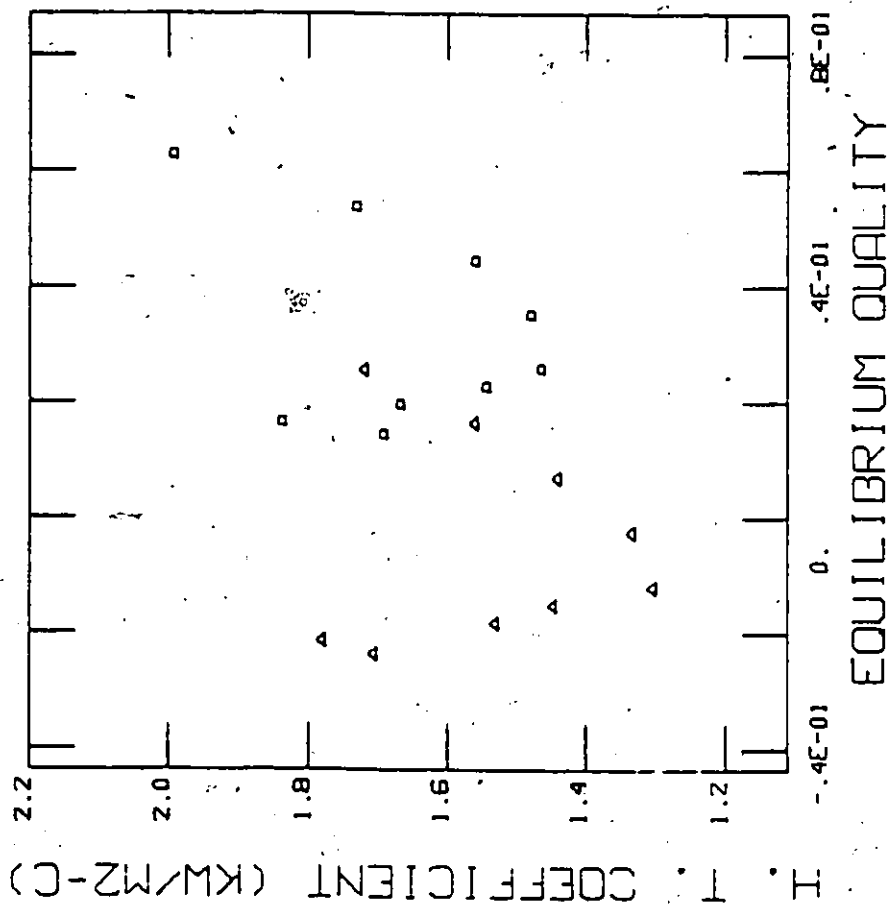


FIGURE VI.7. Effect of inlet quality ($C = 4500 \text{ kg.m}^{-2}.\text{s}^{-1}$)
 on the heat transfer coefficient.

VI.4 Heat Flux Effects

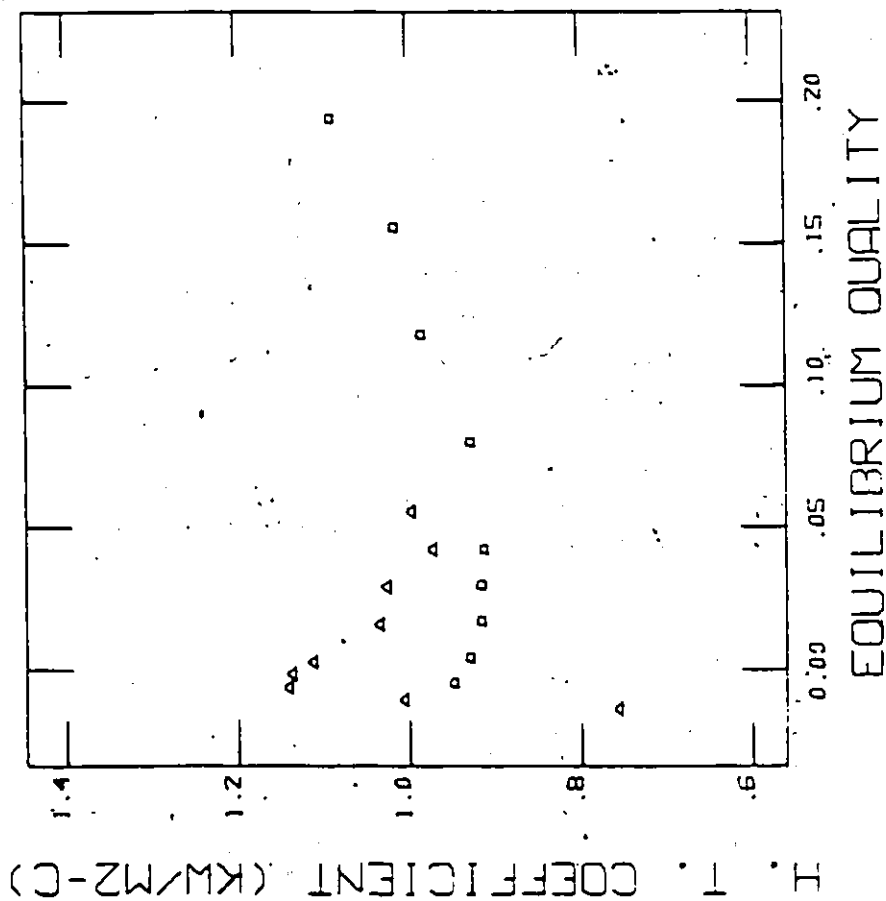
VI.4.1 Subcooled inlet

The effect of heat flux on the heat transfer coefficient is plotted in Figure VI.8 to Figure VI.10. It was generally found that the heat transfer coefficient is higher for the lower heat flux. Initially, the flow regime for the subcooled inlet is thought to be inverted annular. The vapour film is thinner for lower heat flux which can explain the higher conductive heat transfer coefficient. The decrease of the heat transfer coefficient with quality in Figure VI.8 (Run 9123) might be explained by the increase of vapour superheat as mentioned previously. This increase might be more important at lower heat flux and it tends to decrease the heat transfer coefficient.

The trend of the data is different for the Run 9213 (Figure VI.9). The heat transfer coefficient is increasing with quality which was not observed for the Run 9123. The rate of increase of heat transfer is greater initially for $185 \text{ kW} \cdot \text{m}^{-2}$ until the quality reaches approximately zero. This is probably the subcooled water present in the liquid core. However, it was observed as previously that the lower the heat flux, the higher the heat transfer coefficient.

Figure IV.10 shows the experimental results obtained at the higher mass flux (Run 9413). It is observed that the heat flux effect is very strong in the upstream part of the test section especially at the lower heat flux. The effect of heat flux however decreases in the downstream part. This might be explained by a lot better mixing and the reduction of the vapour superheat due to the very high mass flux.

○ RN:9113 PHI:374.
 △ RN:9113 PHI:133.



○ RN:9123 PHI:425.
 △ RN:9123 PHI:375.
 • RN:9123 PHI:300.
 ■ RN:9123 PHI:260.

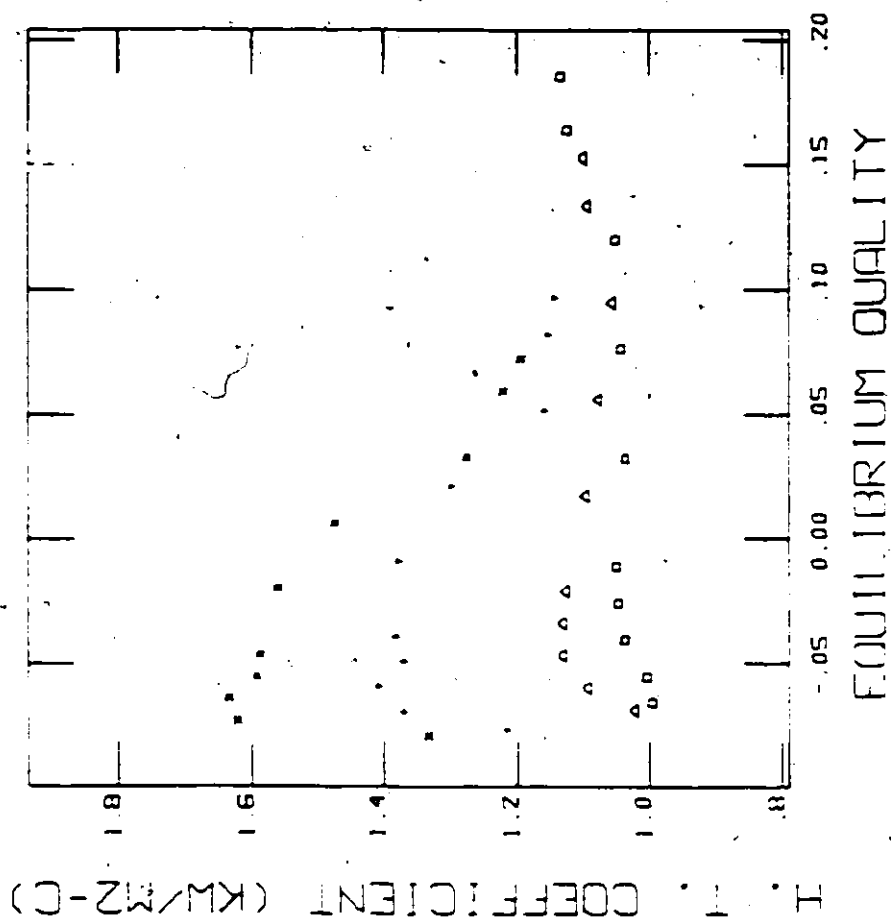
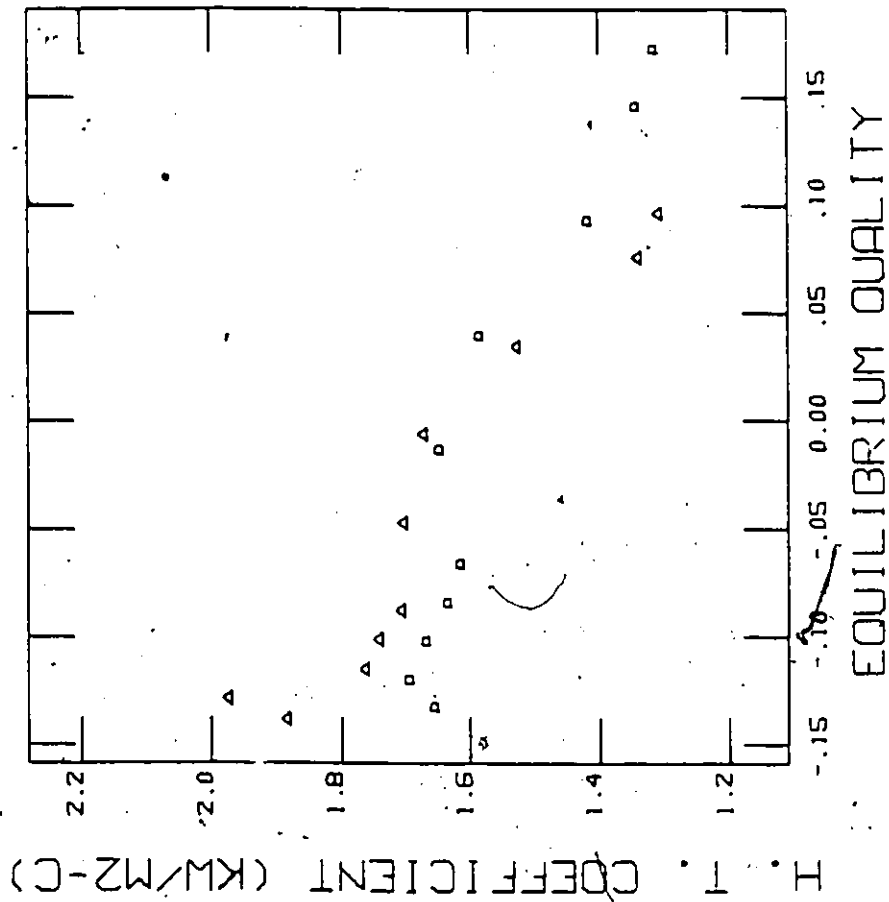


FIGURE VI.8. Effect of heat flux on the heat transfer coefficient

(Run 9113 and Run 9123)

○ RN:9133 PHI:527.
 △ RN:9133 PHI:402.



○ RN:9213 PHI:418.
 △ RN:9213 PHI:296.
 ; RN:9213 PHI:185.

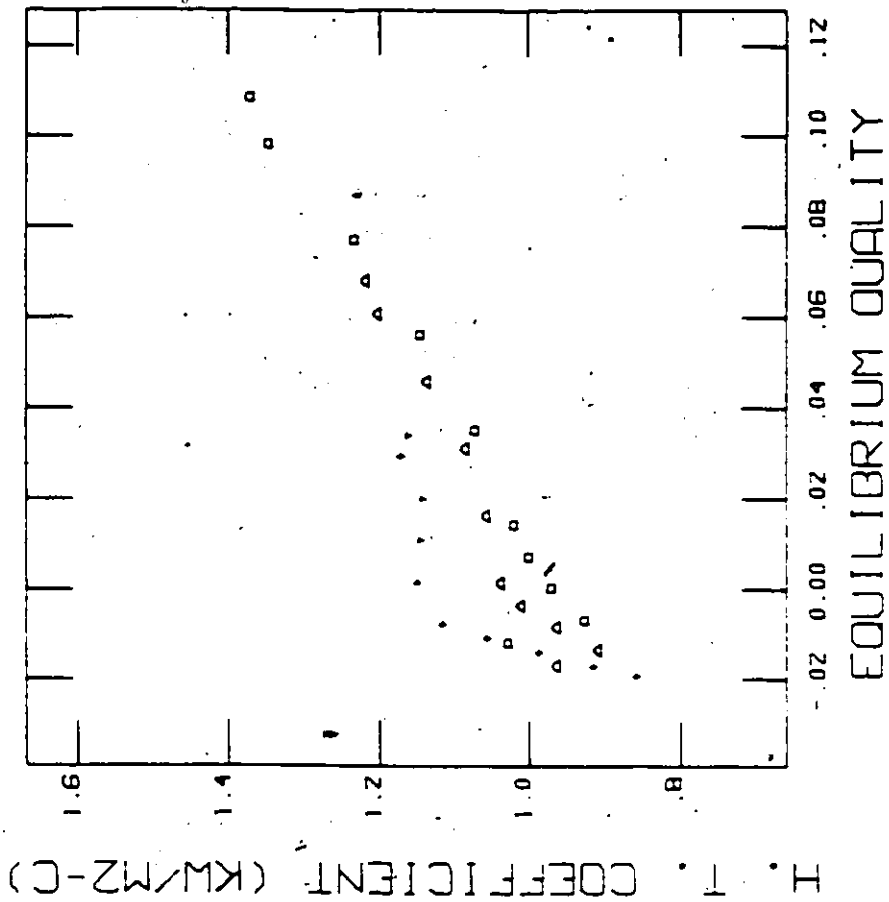
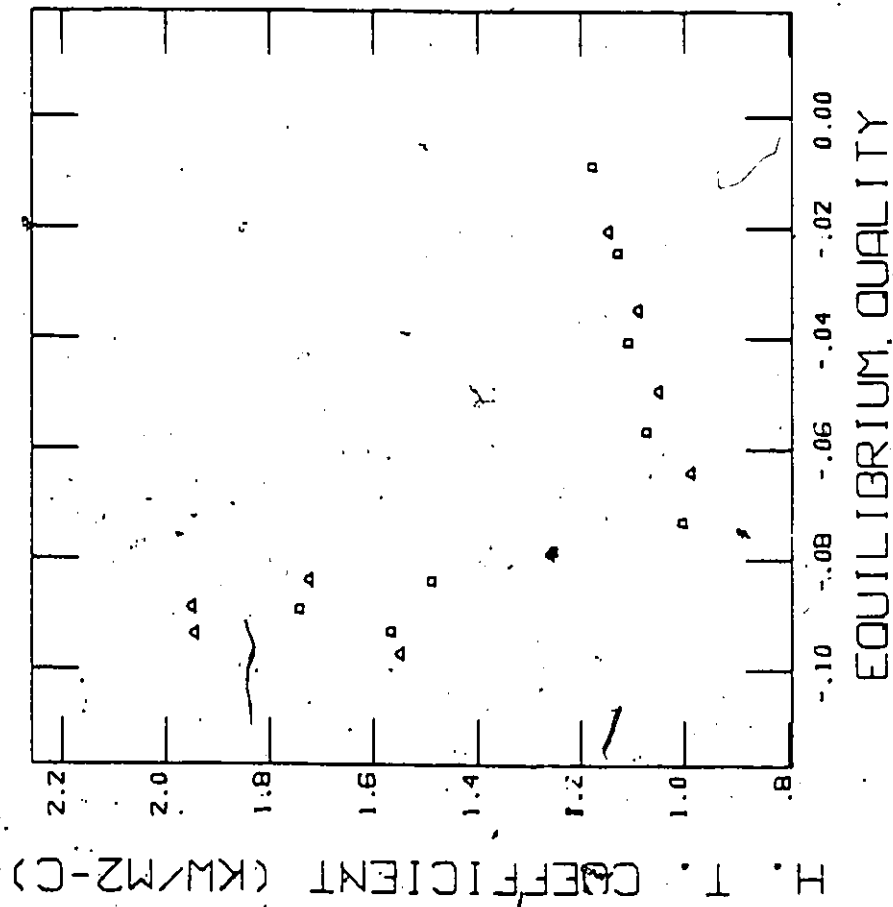


FIGURE VI.9. Effect of heat flux on the heat transfer coefficient.

(Run 9133 and Run 9213)

○ RN:9323 PHI:457.
 △ RN:9323 PHI:416.



○ RN:9413 PHI:509.
 △ RN:9413 PHI:306.
 · RN:9413 PHI:196.

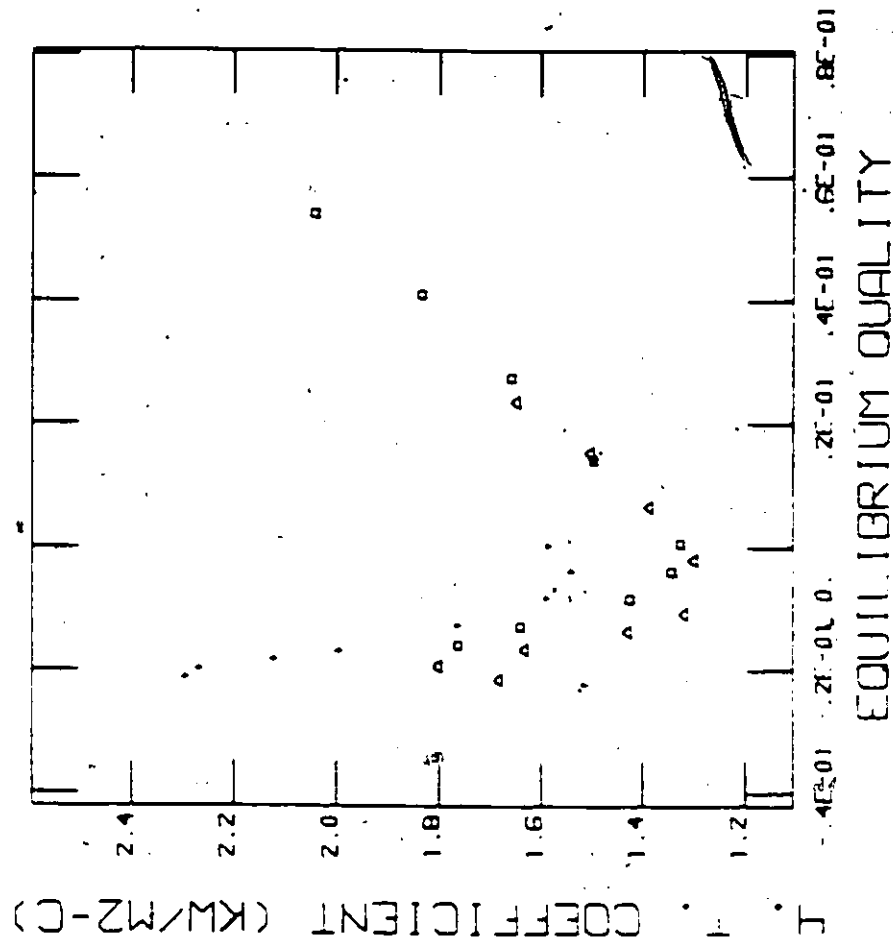


FIGURE VI.10. Effect of heat flux on the heat transfer coefficient

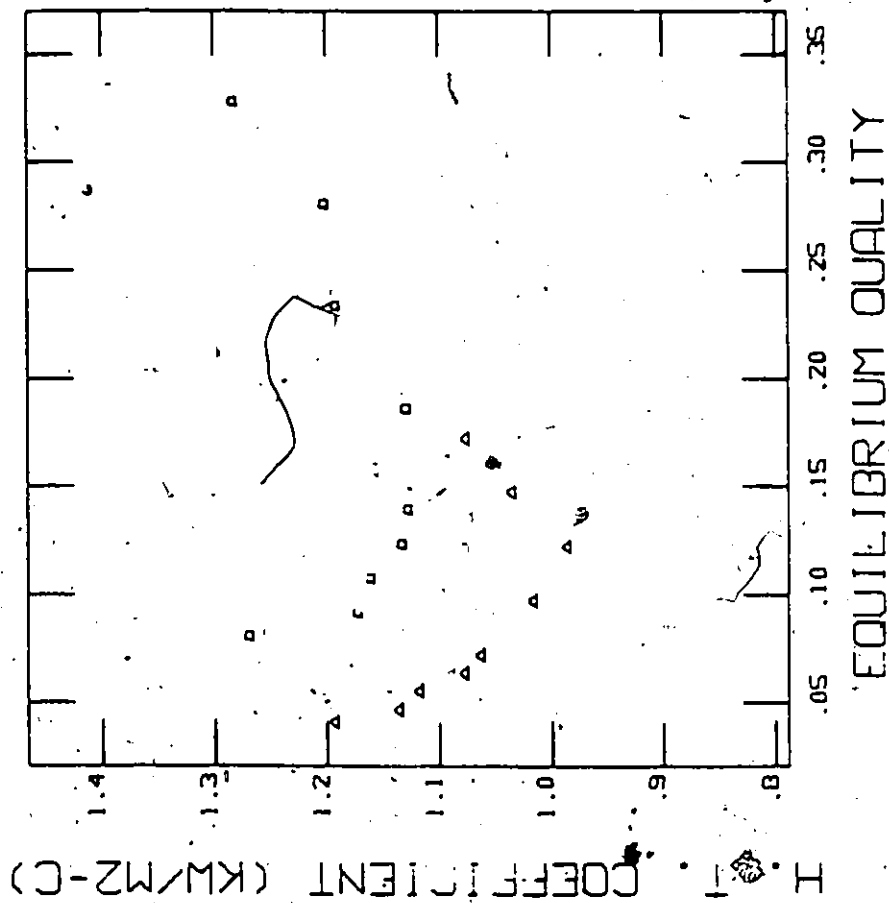
(Run 9323 and Run 9413)

VI.4.2 Positive Inlet Quality

The effect of heat flux is different for the positive inlet quality specially for the 0.08 Inlet and the 0.15 Inlet (Figure VI.11 - Figure VI.13). An increase in heat flux generally increases the heat transfer coefficient. The flow regime is thought to be dispersed for a high quality inlet. This could mean a higher heat flux from the vapour to the droplets and a larger vapour velocity which will improve the heat transfer coefficient.

The trend was found to be different for zero inlet quality as shown in Figure VI.12 (Run 9243). The heat transfer coefficient is higher for $122 \text{ kW}\cdot\text{m}^{-2}$ compared to $326 \text{ kW}\cdot\text{m}^{-2}$. This is not well understood due to the flow regime involved. However, this could be caused by the fluctuation in the inlet quality since it was found that the heat transfer coefficient is very sensitive to the inlet quality between 0.0 and 0.08 as discussed in the section VI.3. A low heat flux could also imply subcooled water over a longer distance downstream of the hot patch will tend to increase the heat transfer coefficient.

○ RN:9143 PHI:463.
 △ RN:9143 PHI:255.



○ RN:9153 PHI:545.
 △ RN:9153 PHI:297.
 • RN:9153 PHI:131.

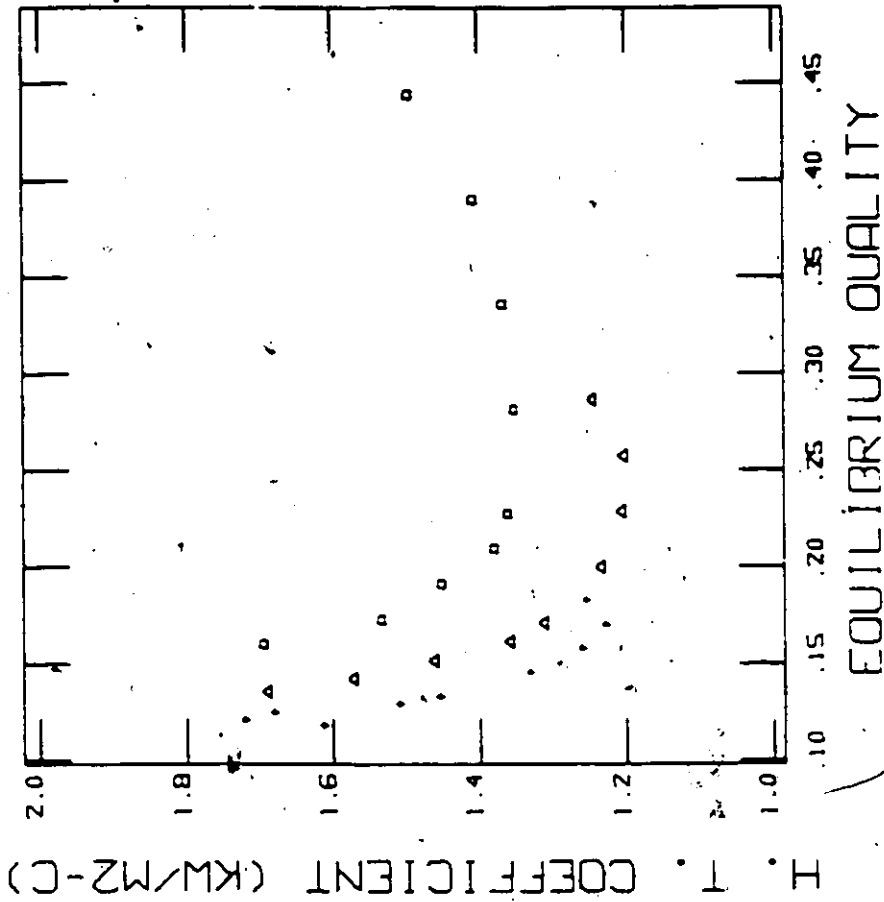


FIGURE VI.11. Effect of heat flux on the heat transfer coefficient

(Run 9143 and Run 9153)

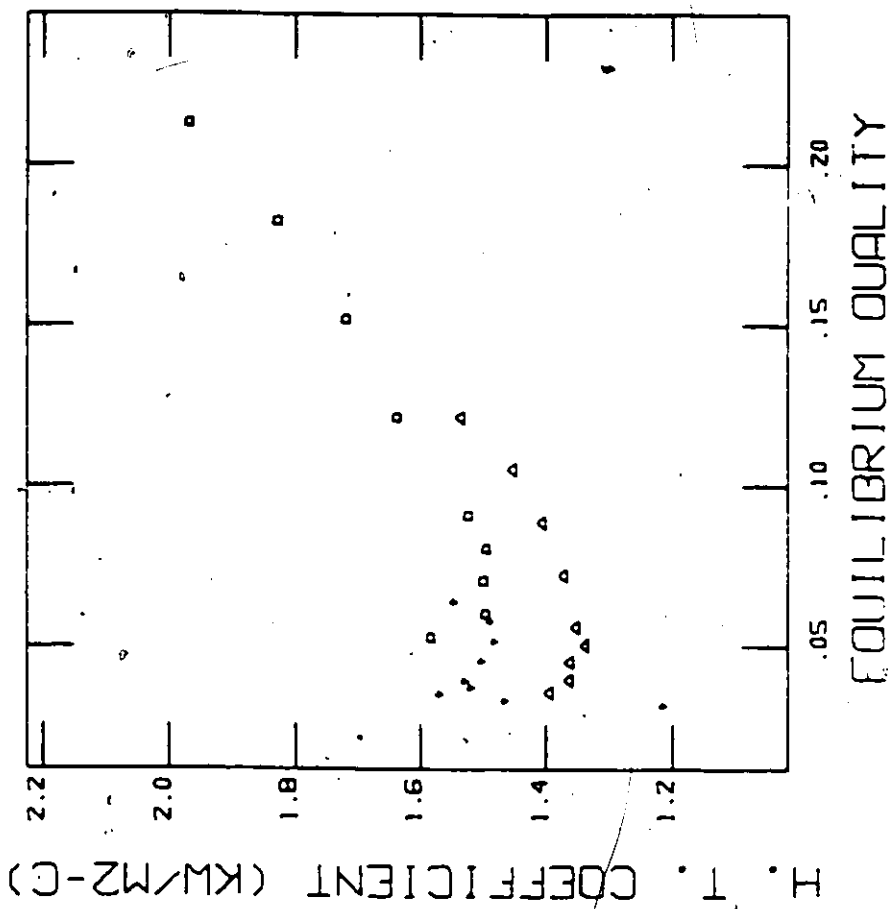
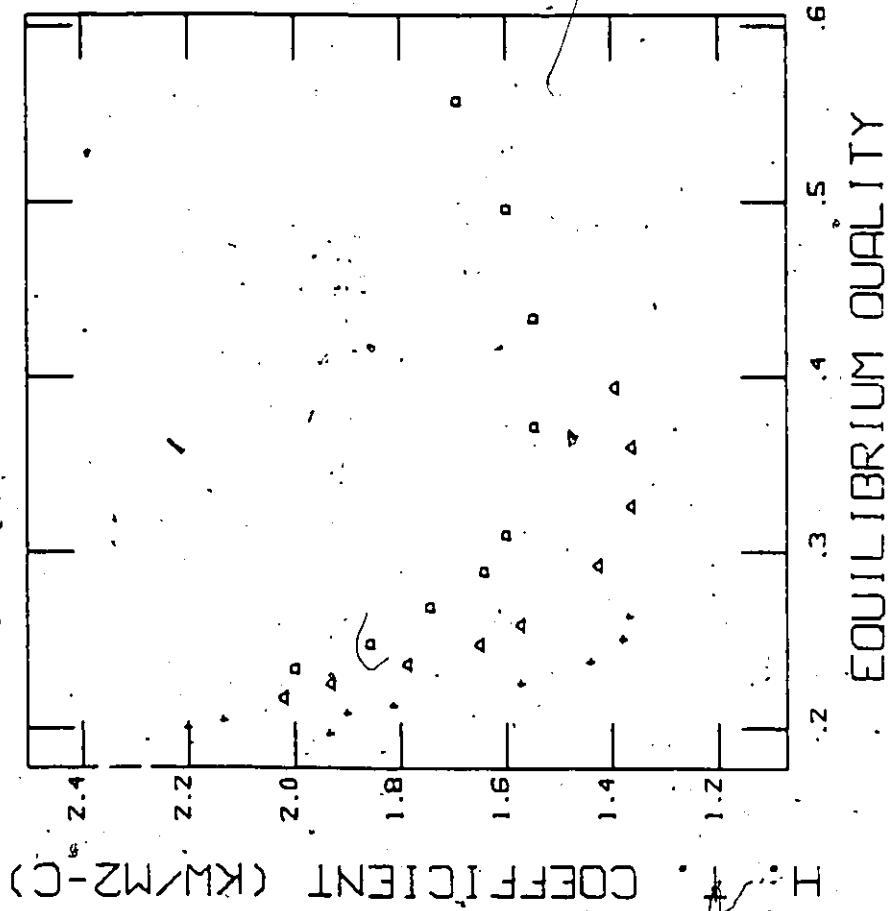
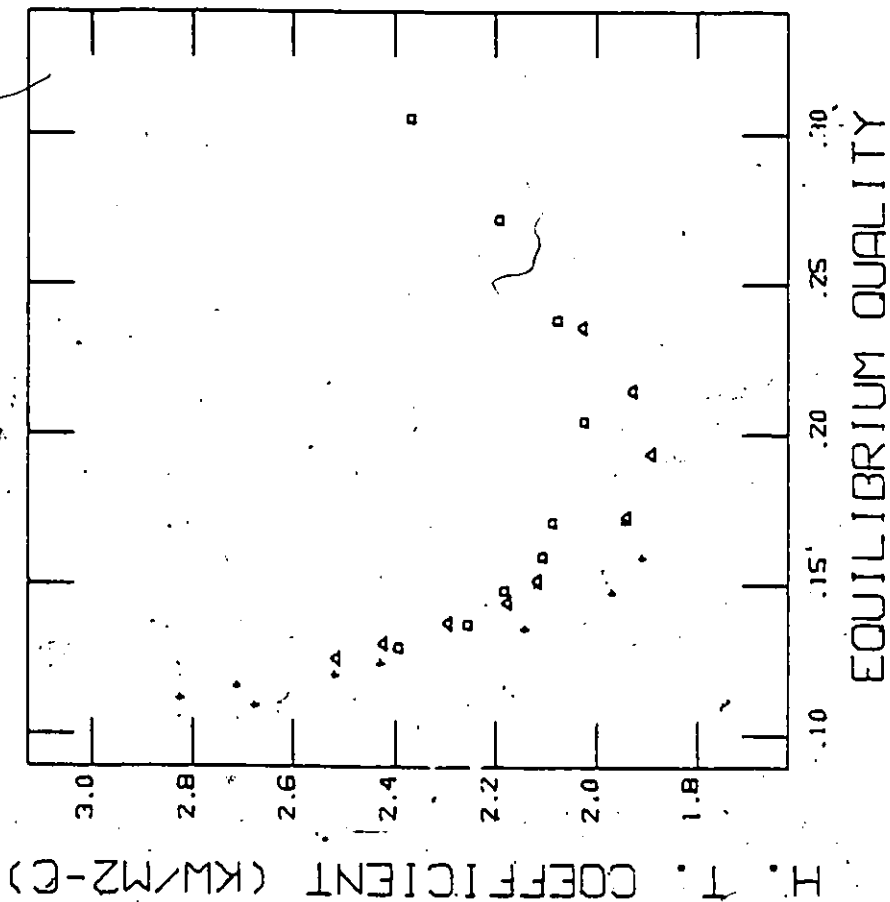


FIGURE VI.12. Effect of heat flux on the heat transfer coefficient
 (Run 9163 and Run 9243)

□ RN:9253 PHI:656.
 △ RN:9253 PHI:411.
 · RN:9253 PHI:236.



□ RN:9263 PHI:683.
 △ RN:9263 PHI:431.
 · RN:9263 PHI:243.

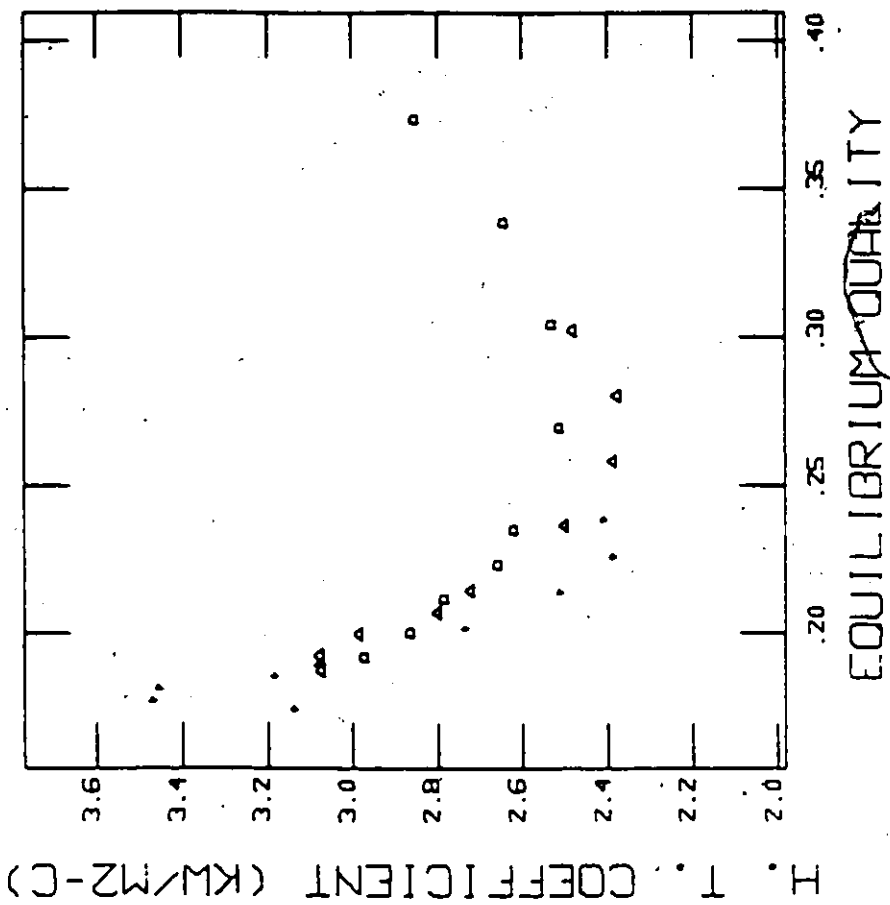


FIGURE VI.13. Effect of heat flux on the heat transfer coefficient

(Run 9253 and Run 9263)

VI.5 Comparison with Correlations

The experimental results were compared with empirical correlation for a given heat flux. Three correlations were used to predict the heat transfer coefficient (based on $(T_w - T_{sat})$) knowing the heat flux.

The correlations used are:

- 1) Berenson [1961]
- 2) Dougall-Rohsenow [1963]
- 3) Groeneveld-Delorme [1976].

The first correlation is a pool type of correlation and the properties were calculated at the film temperature which requires an iterative procedure. The second correlation is basically a Dittus-Boelter correlation for the vapour and ignores the non-equilibrium. Finally, the third correlation is a non-equilibrium correlation.

Non-equilibrium means that the vapour and liquid temperatures are not the same. This means that the actual quality defined as the vapour mass flow fraction of the total mass flow is lower than the thermodynamic equilibrium quality. The Dougall-Rohsenow correlation evaluates the steam properties at saturation.

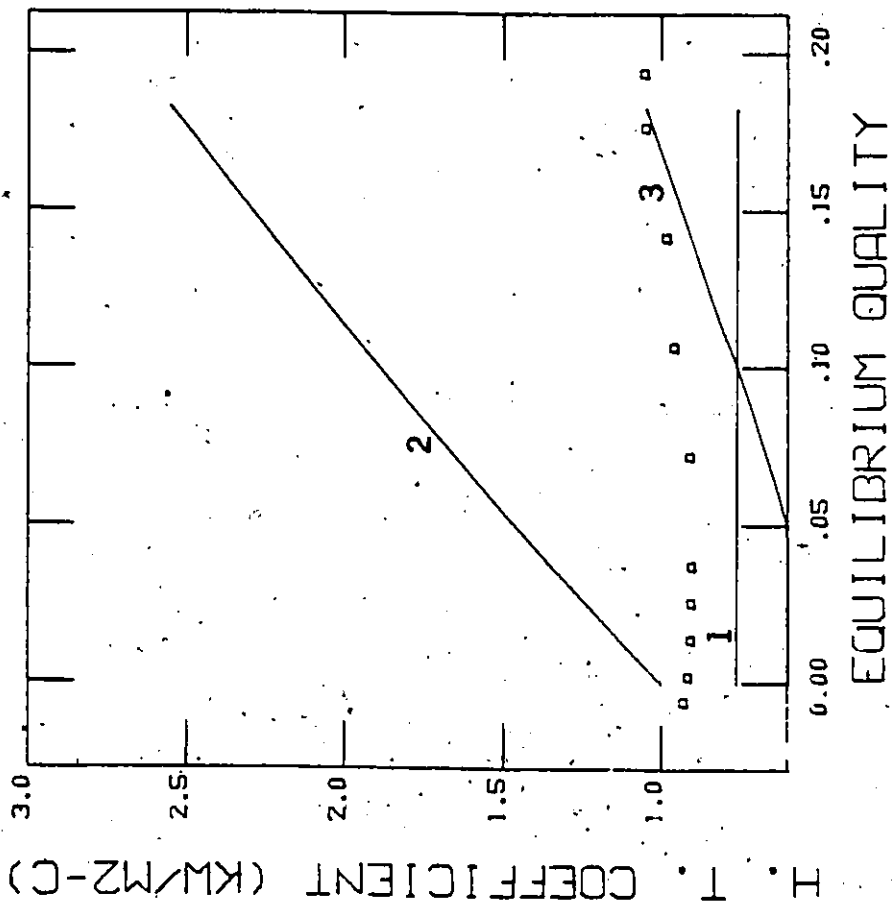
It was observed that the Dougall-Rohsenow correlation always overpredicts the heat transfer coefficient. This was also reported by Morris et al. [1981] from a water upflow dispersed flow film boiling experiment through a rod bundle. Koizumi [1982] generated a steam-water two-phase flow in an annular flow path with a uniformly heated rod and mentioned also that the heated rod surface temperature calculated by the Dougall-Rohsenow equation is much lower than the one measured.

The Berenson correlation underpredicts the heat transfer coefficient. This correlation is derived for saturated pool boiling liquid and a better agreement was generally found when the inlet temperature is close to the saturation temperature. This was observed for the subcooled inlet (Run 9113, Run 9123 and Run 9133) as well as the positive inlet (Run 9143, Run 9153 and Run 9163). This also applies at higher mass flux (Run 9213 and Run 9223; Run 9243, Run 9253 and Run 9263; Run 9343, Run 9353 and Run 9363). This correlation is a pool boiling correlation as mentioned previously and the agreement is better at lower mass flux. (Run 9143 and Run 9443).

The Groeneveld-Delorme correlation predicts the heat transfer coefficient more accurately for a high quality inlet. (Run 9153, Run 9163, Run 9253, Run 9263, Run 9353 and Run 9363). This is expected since this correlation is derived for the dispersed flow regime region. The trend of this correlation is identical to the trend of the Dougall-Rohsenow's correlation. It predicts an increase in the heat transfer coefficient for an higher quality. The Groeneveld-Delorme correlation usually also overpredicts the heat transfer coefficient in the downstream part of the test section. The result is different from Morris [1981] and Koizumi [1982]. They both found that the Groeneveld-Delorme correlation underpredicts the heat transfer coefficient. However, the inlet quality of their experiments was higher (0.5 to 1.0 for Koizumi)

□ RN:9113 PHI:342.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme



□ RN:9123 PHI:375.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme

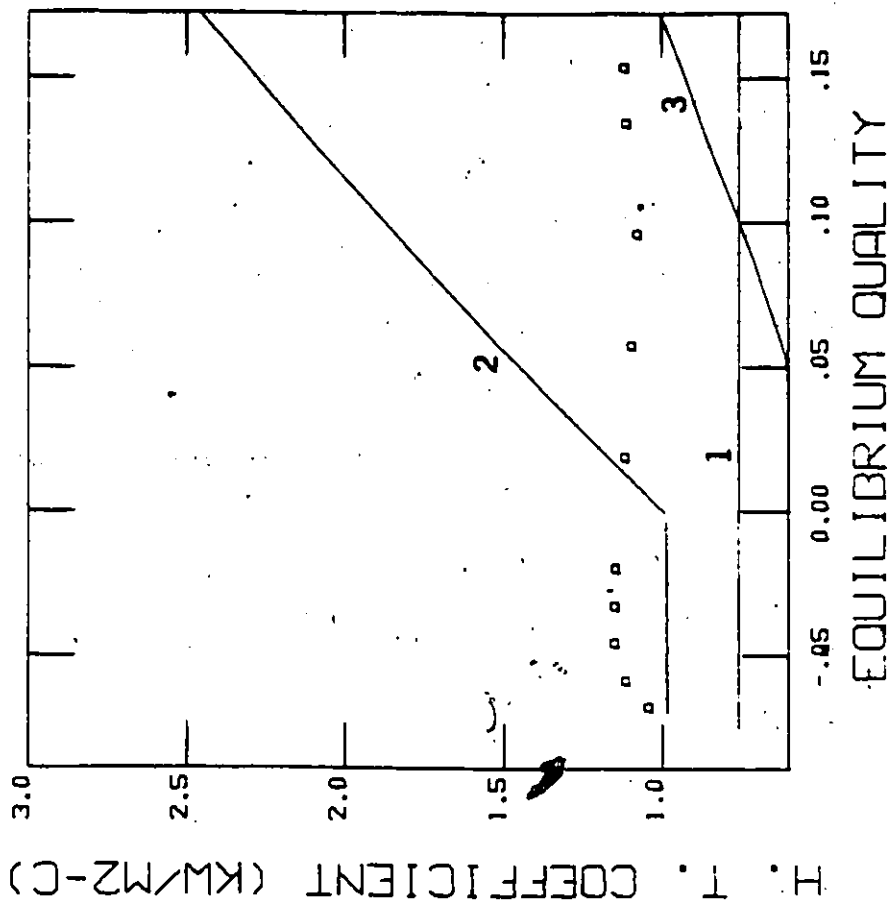
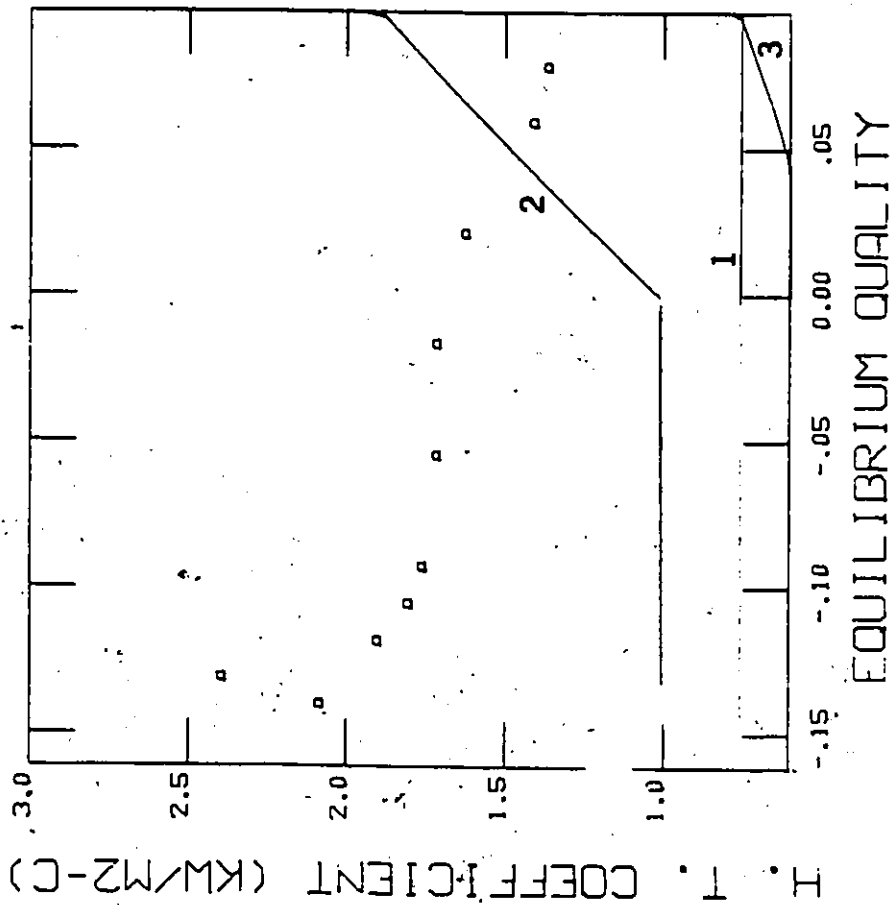


FIGURE VI.14. Comparison of experimental heat transfer coefficient with three correlations (Run 9113 and Run 9123)

□ RN 9133 PHI .379.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme



□ RN 9213 PHI .375.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme

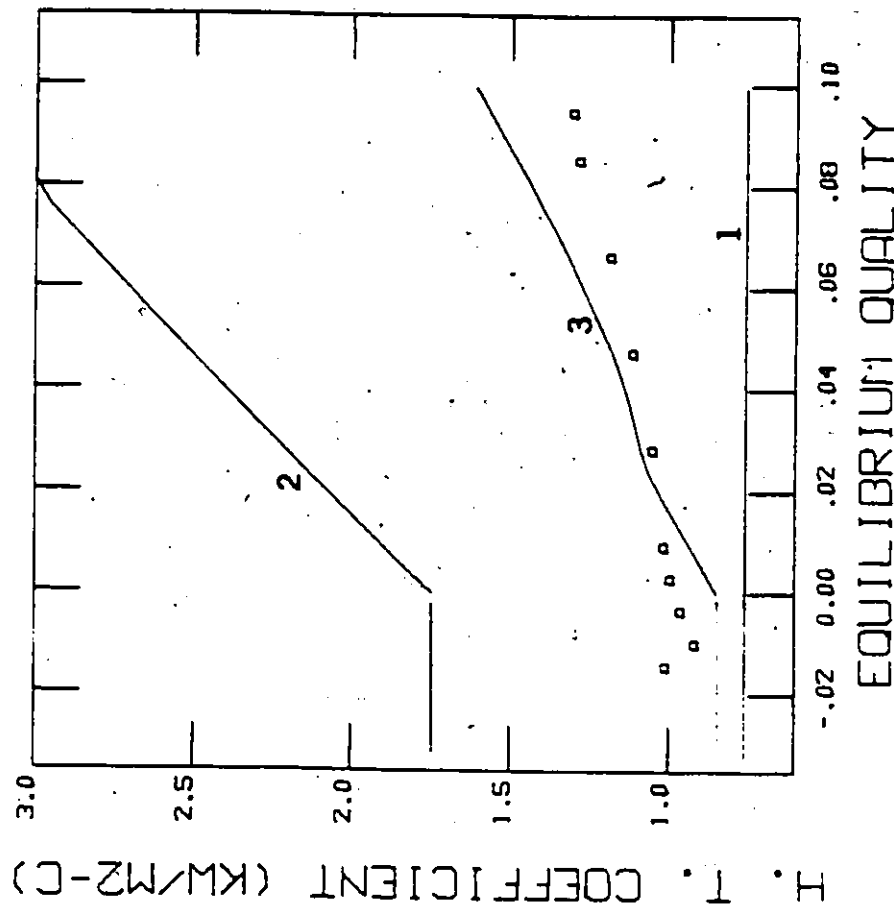
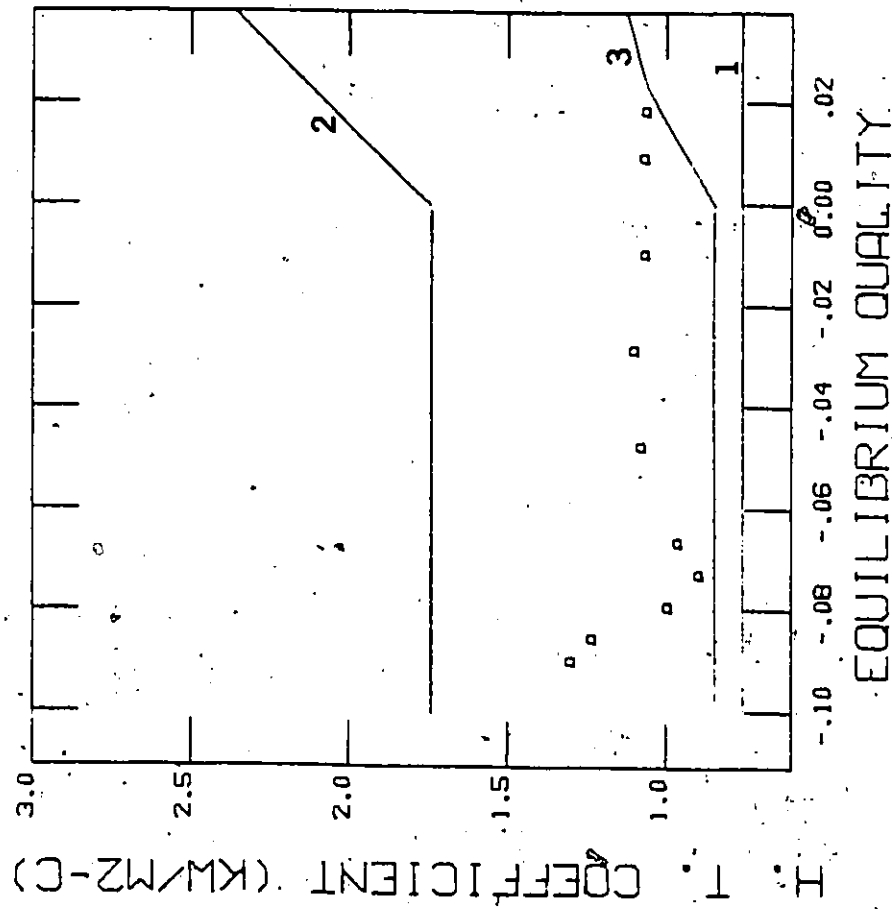


FIGURE VI.15. Comparison of experimental heat transfer coefficient with three correlations (Run 9133 and Run 9213)

□ RN.9223 PHI.371.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme



□ RN.9313 PHI.413.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme

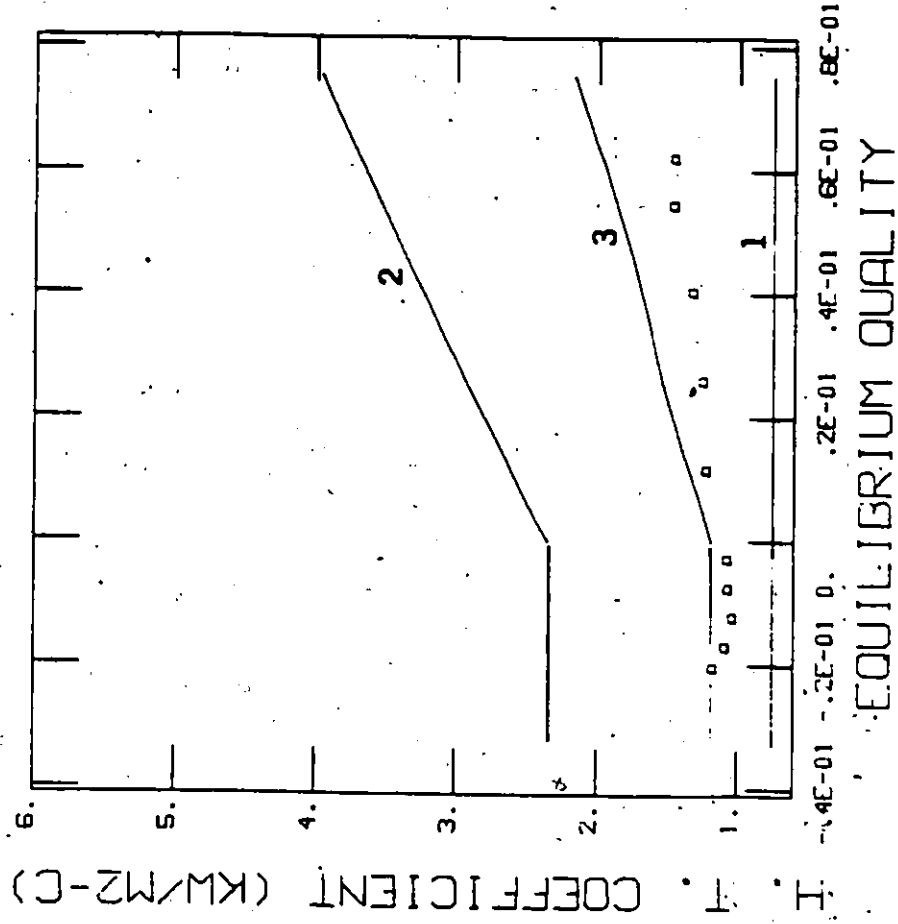
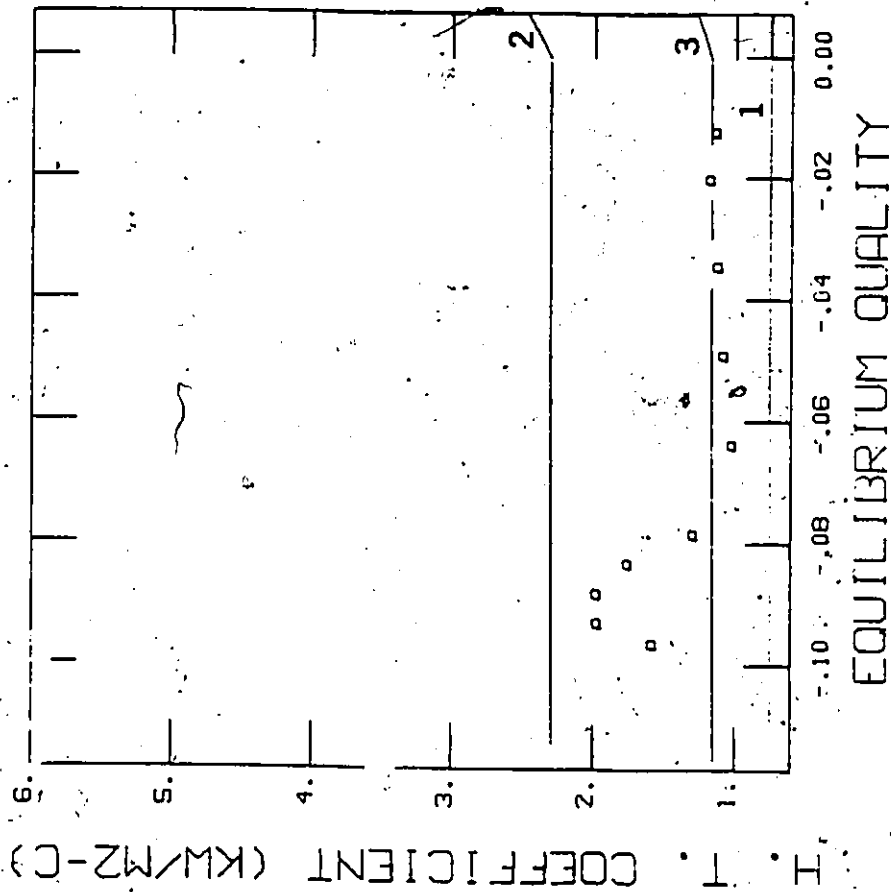


FIGURE VI.16. Comparison of experimental heat transfer coefficient with three correlations (Run 9223 and Run 9313)

○ RN:9323 PHI:416.
 1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme



○ RN:9413 PHI:422.
 1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

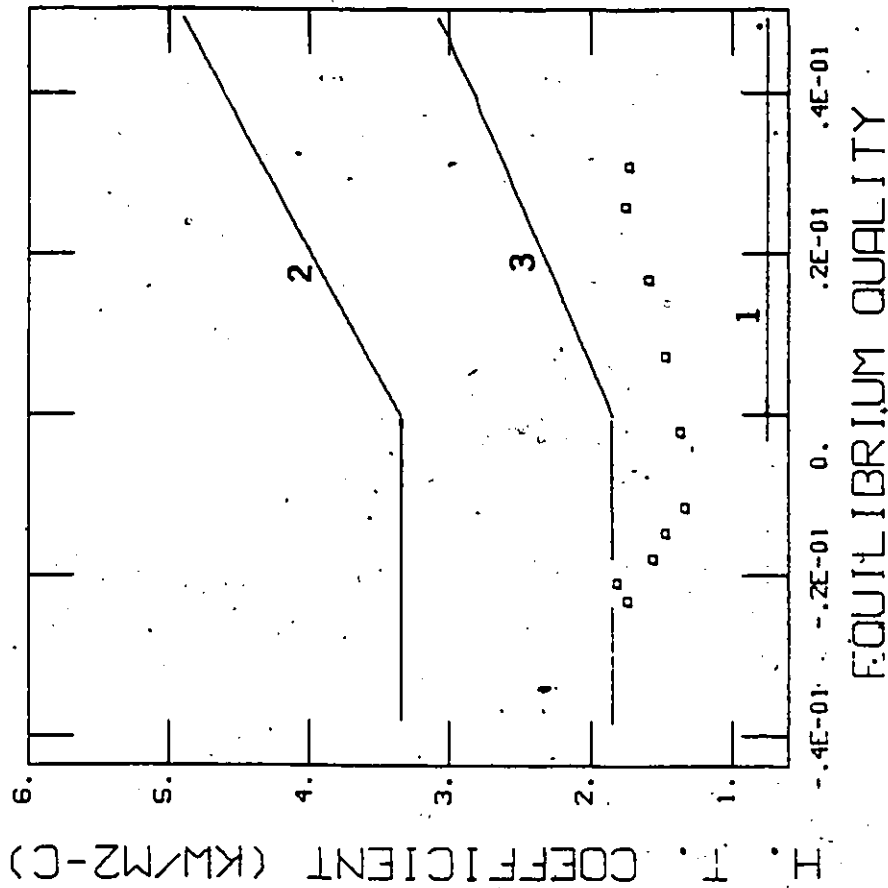
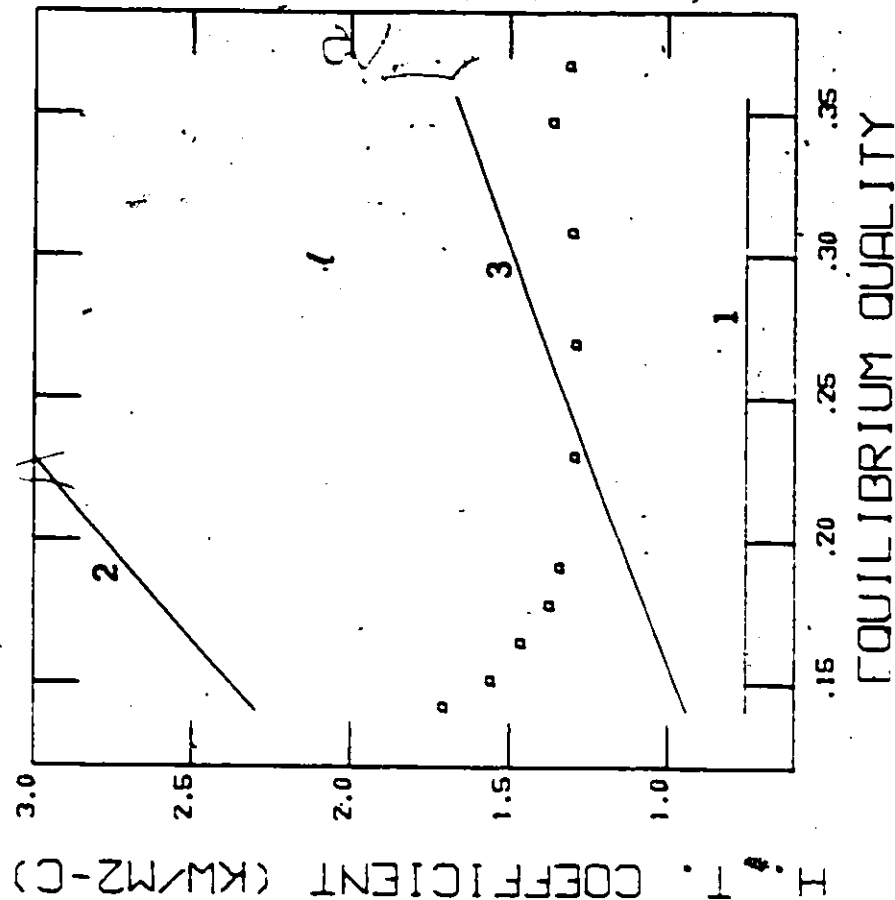


FIGURE VI.17. Comparison of experimental heat transfer coefficient
 with three correlations (Run 9323 and Run 9413)

1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

Run 9153 PH1, 401.



1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

Run 9143 PH1, 391.

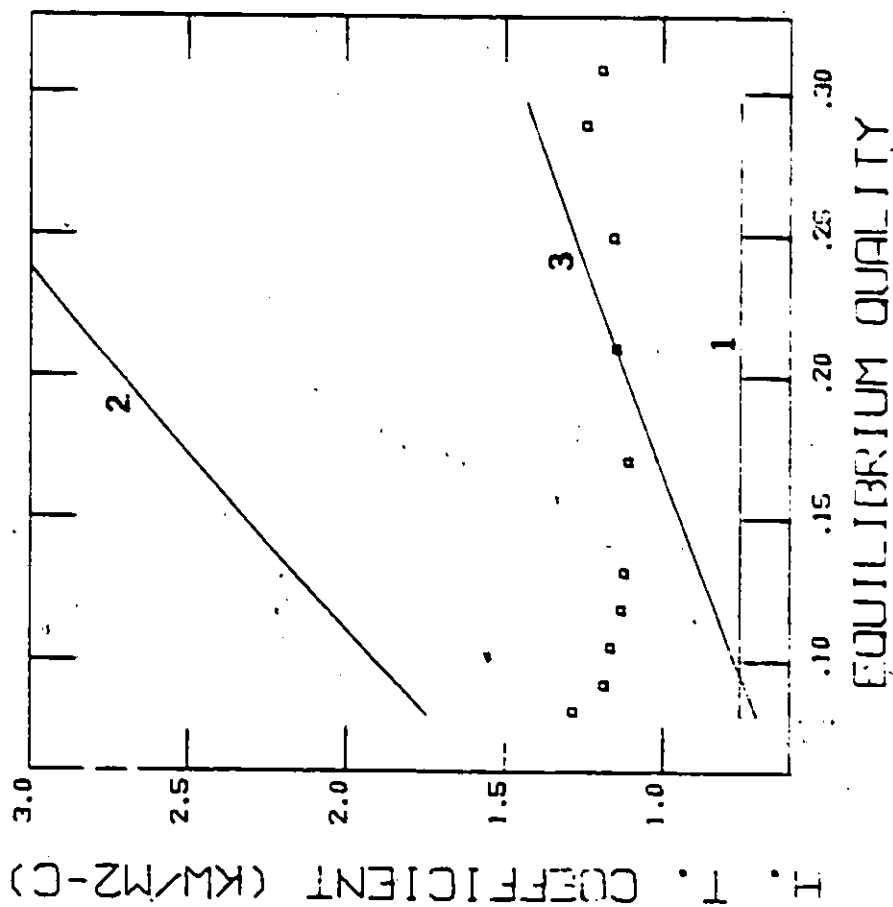
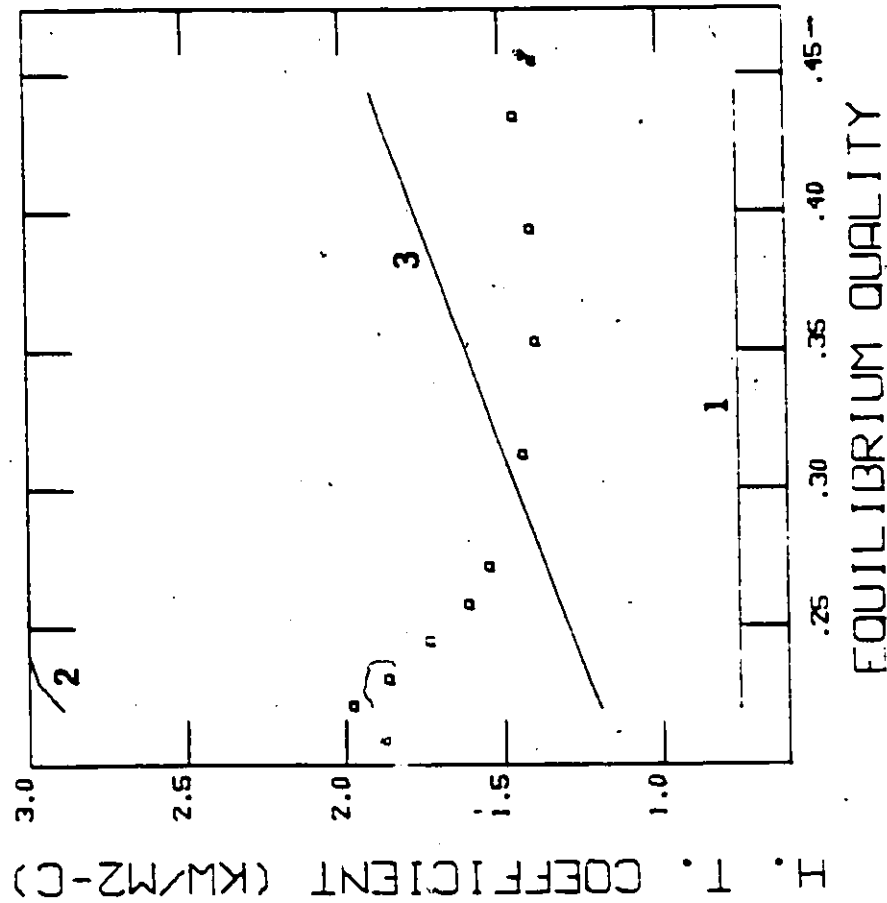


FIGURE VI.18. Comparison of experimental heat transfer coefficient with three correlations (Run 9143 and Run 9153)

• RM.9163 PHI, 408.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme



• RM.9243 PHI, 382.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme

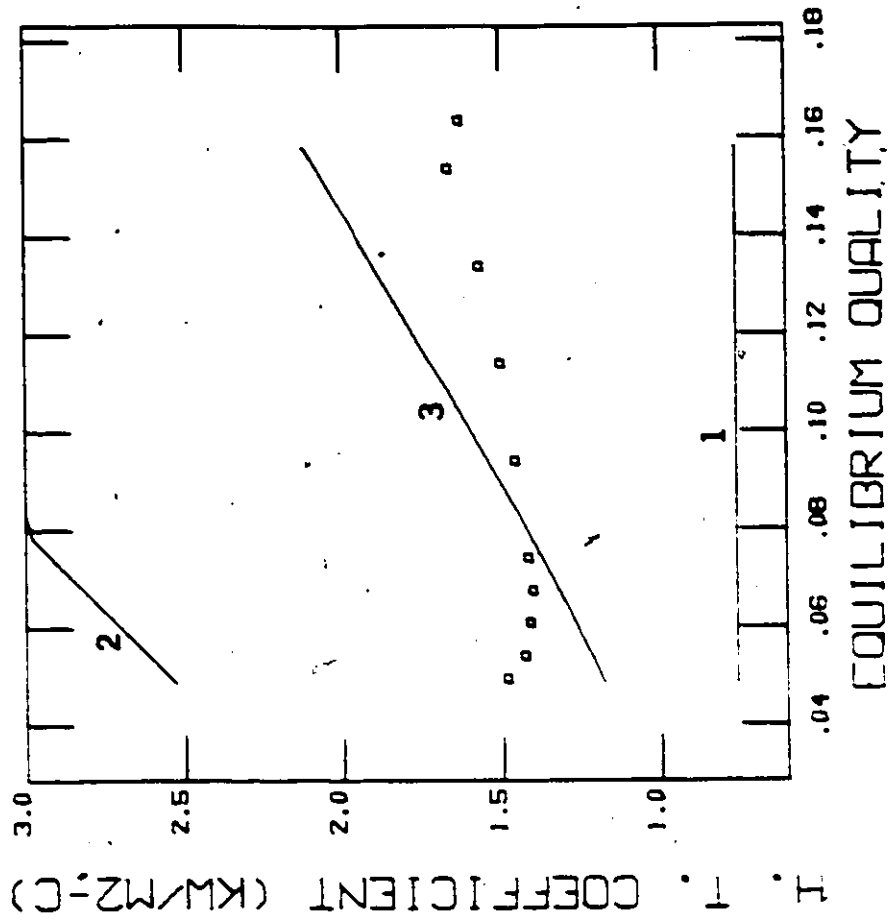
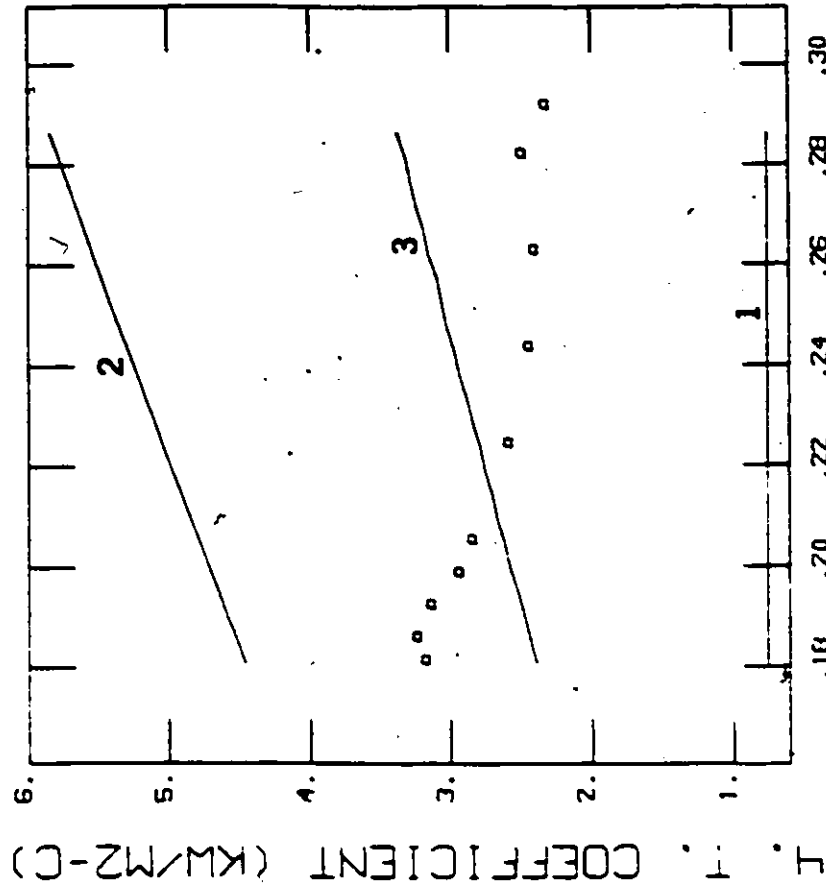


FIGURE VI.19. Comparison of experimental heat transfer coefficient with three correlations (Run 9163 and Run 9243)

1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

Run 9263 (PHI .381)



1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

Run 9253 (PHI .386)

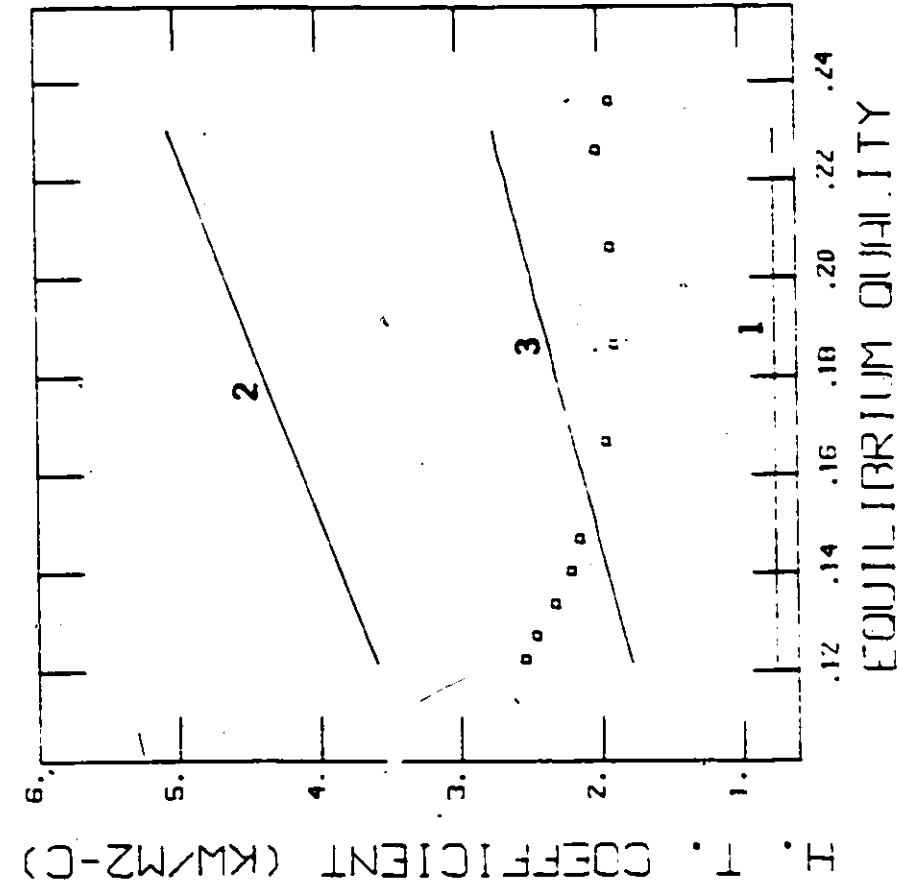
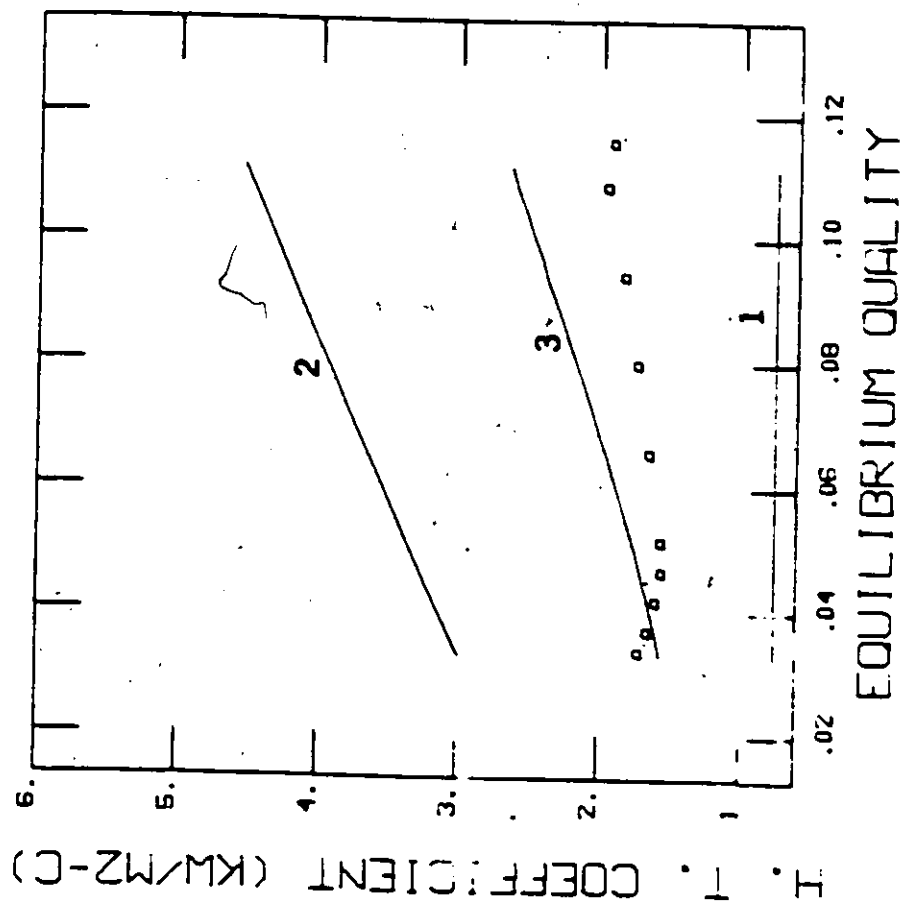


FIGURE VI.20. Comparison of experimental heat transfer coefficient with three correlations (Run 9253 and Run 9263)

◦ RM 9343 PH1.394.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme



◦ RM 9353 PH1.387.

- 1 : Berenson
- 2 : Dougall-Rohsenow
- 3 : Groeneveld-Delorme

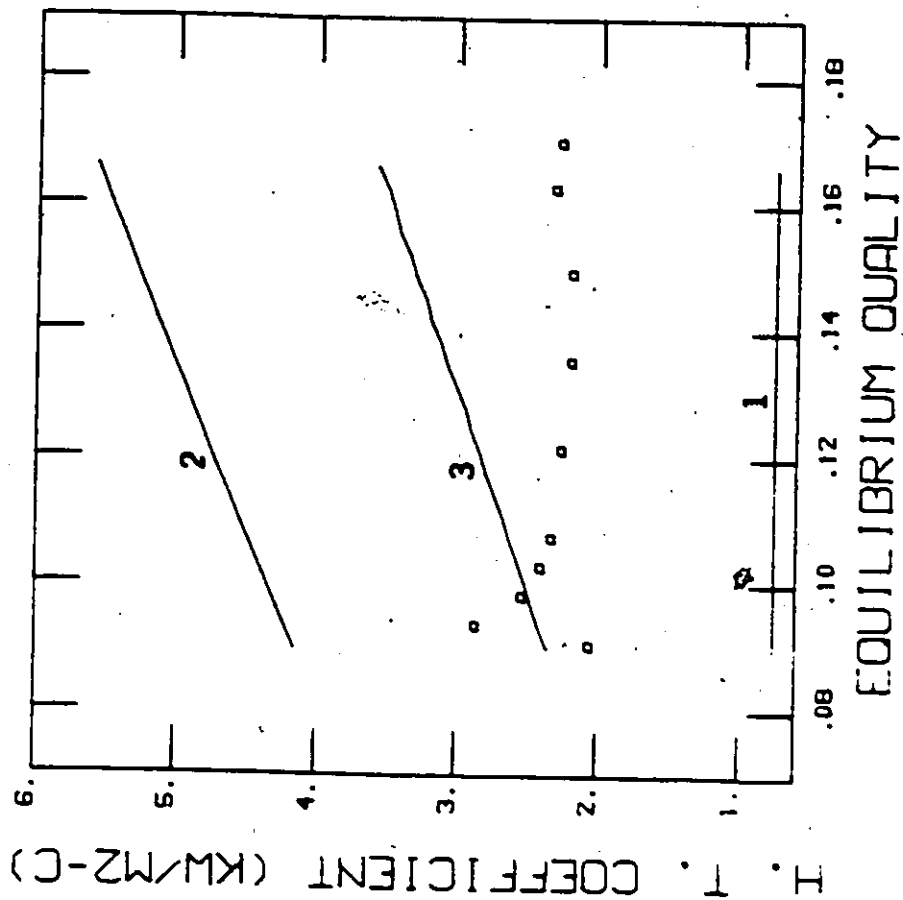
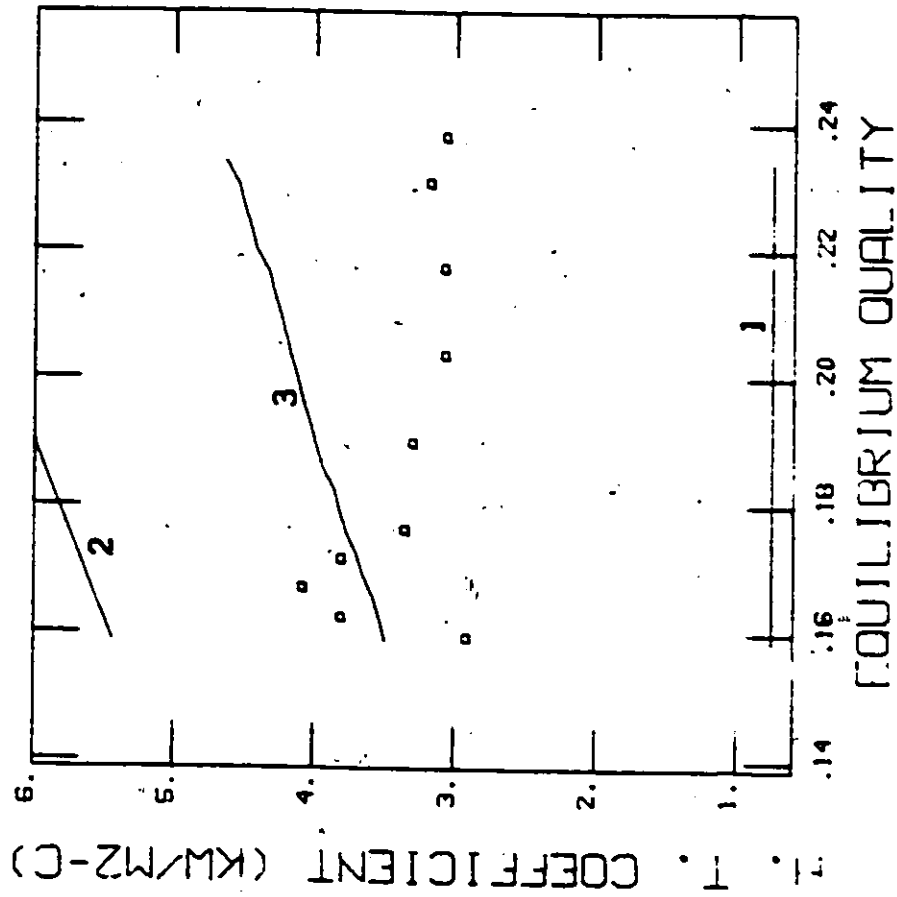


FIGURE VI.21. Comparison of experimental heat transfer coefficient with three correlations (Run 9343 and Run 9353)

• RN, 9363 PHI, 381.
 1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme



• RN, 9443 PHI, 413.
 1 : Berenson
 2 : Dougall-Rohsenow
 3 : Groeneveld-Delorme

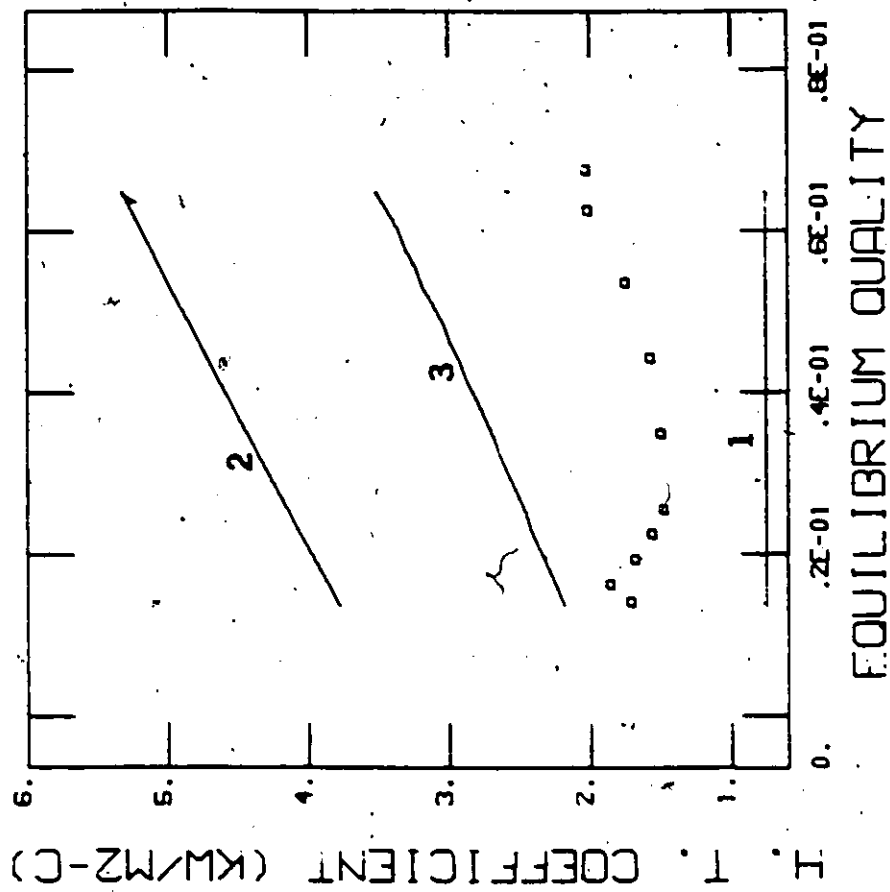


FIGURE VI.22. Comparison of experimental heat transfer coefficient with three correlations (Run 9363 and Run 9443)

VII. CONCLUSIONS AND RECOMMENDATIONS

1. The film boiling region was investigated experimentally/as well as analytically. The proposed model shows reasonable agreement with the experimental data obtained by Stewart [1981]. The initial conditions which must be included are the mass flux, the heat flux, the inlet temperature, as well as the tube geometry.
2. Several simplifications were done and some improvements should be done. This includes a better velocity profile in the turbulent region. A uniform velocity profile was assumed in the liquid core which could also be improved. The model was limited to low mass flux (less than $500 \text{ kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$) but it could be useful to expand this prediction method to a wider range. Several parameters as slip ratio and void fraction were also investigated and it was found that even with a lack of experimental data of these given parameters, it still predicts some "reasonable" values.
3. The comparison of the prediction method with experimental data was limited to a set of data of one experimenter and more extensive comparison should be carried out with different sets of data. This could include the comparison of data of rewetting film boiling. The prediction method is using data of stable film boiling but the initial velocity at the quench front can be subtracted from the quench velocity which will bring the phenomenon to a stable film boiling case. This, however, neglects the axial conduction effect of the rewetting front.
4. The entrainment of the vapour in the liquid core was neglected but it was mentioned in the section III.5 that some vapour formed appears in

the core which becomes a mixture of liquid and vapour. This phenomenon could in fact increase the rate of removal of subcooling in the liquid core due to the condensation of the vapour.

5. Stable film boiling was obtained over a very wide range of mass flux and quality at a nominal outlet pressure of 9.6 MPa. The range of the data was summarized in Table 2. The wide range of data has shown that film boiling is very complex. The trends of the data vary depending on the system parameters such as mass flux, inlet quality and heat flux. The data were mainly analysed qualitatively but this however provides a useful initial step to develop a mechanistic model covering a wide range of conditions. Possible causes of the observed data trends were also given by separating some effects such as mass flux, inlet subcooling and heat flux. The minimum film boiling temperature measurements show a net increase of the minimum film boiling temperature in the subcooled region but was found fairly constant in the positive quality region.
6. Experimental heat transfer coefficients were also compared with three empirical correlations (Berenson [1961], Groeneveld-Delorme [1976], Dougall-Rohsenow [1963]). Agreement was found to be poor. Berenson's correlation was found to be more accurate at low qualities as well as inlet temperature close to saturation. Dougall-Rohsenow's correlation overpredicts approximately by a factor of two the heat transfer coefficient specially in the high quality region. Groeneveld-Delorme's correlation better predicts the heat transfer coefficient in the high quality region.

TABLE 1. LOW QUALITY FILM BOILING EXPERIMENTS

(from Groeneveld & Gardiner (1977))

AUTHORS	EQUATION	RANGE OF APPLICABILITY					COMMENTS
		TEST FLUID	GEOMETRY	PRESSURE MPa	FLOW RATE kg m ⁻² s ⁻¹	SUBCOOLING °C OR QUALITY	
Bromley [75,79]	$h = h_{FB} + 3/4 h_{rad}$ $h_{FB} = 0.62 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right]^{1/4}$ $h_{rad} = \frac{5.67 \times 10^{-8} (T_b^4 - T_s^4)}{\left(\frac{1}{\epsilon_b} + \frac{1}{\epsilon_s} - 1 \right) \Delta T_s}$ $h_{FS} = h_{FB} \left[1 + \frac{0.4 C_p \Delta T_s}{h_{FB}} \right]$	Nitrogen water n-pentane benzene carbon tetrachloride ethyl alcohol	Horizontal carbon cylinders OD - 6.4 mm - 9.5 mm - 12.7 mm	Not given	Pool boiling	Saturated	Pool film boiling from external surfaces. Laminar theory.
Bernome [14]	$h_{FB} = 0.423 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s \sqrt{\frac{\rho_b}{\Delta T_s (T_b - T_s)}}} \right]^{1/4}$		Horizontal surface		Pool boiling	Saturated	Theory only.
Lawrence and Ede [80]	$C_{total} = C_s + C_r$ $C_s = 0.613 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right]^{1/4} + C_{rad}$ $C_r = 0.31 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right]^{1/4} \left[\frac{C_p \Delta T_s}{h_{FB}} \right]^{1/4}$ $C_r = 0.31 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right]^{1/4}$	Water water plus detergent	Horizontal cylinders OD - 1.2 mm - 6.4 mm	0.101	Pool boiling	0 - 90°C	Correlated their data well over the entire range of sub-coolings.
Anderson et al. [45]	$h_s = C_1 \left[\frac{k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right]^{1/4}$ $0.3321 \leq C_1 \leq 0.3400$				Low flow rates	Saturated	Close slightly higher heat transfer coefficient than Bernome's equation for $\Delta T_s = 100^\circ\text{C}$. Theory only.
Leffler [12]	$h = \left[\frac{0.613 k_b^3 \rho_b (T_b - T_s) h_{fg}}{12 \Delta T_s \mu_b} \right]^{1/4} + h_{rad}$	water	Vertical annulus OD - 63.5 mm ID - 6.4 mm length - 76.2 mm	0.11 - 0.61	(0.136 - 1.33) $\times 10^3$	15°C - 36°C	
Leffler et al. [46]	a) laminar annular flow i) outermost region $Nu = 0.0012 \left(\frac{200}{Pr} \right)^{0.4} \left[1 + 1.22 \exp(-0.630 \frac{2}{Pr}) \right]$ ii) non-outermost region $Nu = 0.0078 \left(\frac{0.613 k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right)^{0.75} Pr^{0.7}$ $Pr = \left[1 + (3 \times 10^{-6}) Pr + 1.1 \exp(-0.10^{-6} Pr) \right]$ $\nu = \frac{C_p \Delta T_s}{h_{FB}} \left(\frac{0.613 k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \right)^{0.75}$ b) Slug flow $Nu_s = 30 Nu_{FB}^{-0.25} \left[1 + 0.25 \exp(-0.85 \frac{2}{Pr}) \right]$ $Nu_s = \frac{h_s D}{k_b} \frac{\rho_b \mu_b}{C_p (T_b - T_s)}$ $Nu_{FB} = \frac{h_{FB} D}{k_b} \frac{\rho_b \mu_b}{C_p (T_b - T_s)}$	Nitrogen	Horizontal flow in a vertical tube ID - 4 mm to 20 mm length - 100 mm	0.61 - 2.13	(6.12 - 16.9) $\times 10^3$	5 - 35°C	Model required to evaluate the film thickness
Duggall and Roseman [4]	$Nu = \frac{0.613 k_b^3 \rho_b (T_b - T_s) h_{fg}}{\mu_b \Delta T_s} \sqrt{\frac{\rho_b}{\Delta T_s (T_b - T_s)}}$ $\int_0^L \frac{dx}{1 + Pr \frac{1}{\nu}}$	Freon-113	Vertical tube ID - 10.2 mm and 6.6 mm length - 301.0 mm	0.101	(0.45 - 1.11) $\times 10^3$	Exit quality up to 100	

TABLE 2
RUN MATRIX

Mass flux ($\text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$)	1000	2000	2750	4500
$\Delta H_{\text{in}} = 50 \text{ kJ/Kg}$	9.6*	9.6	9.6	9.6
150	9.6, 4.0	9.6	9.6	
250	9.6			
$x_{\text{in}} = 0.0$	9.6	9.6	9.6	9.6
0.08	9.6	9.6	9.6	
0.15	9.6	9.6	9.6	

* This number indicates the nominal outlet pressure (MPa).

TABLE 3
TEST SECTIONS SPECIFICATIONS

Inside diameter	0.8928 cm		
Outside diameter	1.114 cm		
Date used	(Oct-Nov)	(Apr-Dec)	(May)
Heated length between power clamps (cm)	189.5	189.3	188.3
Heated length before hot patch (cm)	12.3	12.3	11.0
Heated length beyond last T/S thermocouple	3.0	2.0	2.3
Hot patch length (cm)	10.1	10.1	10.1
Cartridge heaters	12×500W	8×500W	12×500W
Thermocouple location			
Elevation (cm) from top of hot patch			
TC(1) and TC(2)	174.2	175.0	175.0
TC(3) and TC(4)	159.2	159.0	159.0
TC(5) and TC(6)	129.2	130.0	130.0
TC(7) and TC(8)	99.2	100.0	100.0
TC(9) and TC(10)	69.2	70.0	70.0
TC(11) and TC(12)	39.3	39.9	39.9
TC(13) and TC(14)	29.2	29.9	30.0
TC(15) and TC(16)	19.2	20.0	20.0
TC(17) and TC(18)	9.1	10.0	10.0
TC(19) and TC(20)	2.0	3.0	3.0
Lower pressure tap from top of hot patch (cm)	64.6	50.7	64.6
Higher pressure tap from top of hot patch (cm)	208.6	209.8	209.8

TABLE 4

PHYSICAL PHENOMENA DOWNSTREAM OF THE HOT PATCH

Inlet conditions	Physical phenomena	Results
Low inlet subcooling	High vapour superheat	Low heat transfer coefficient
	High vapour generation rate	Thick vapour film (poor conductive heat transfer coefficient)
	High vapour Reynolds number	Good convective heat transfer
High inlet subcooling	Low vapour superheat	High heat transfer coefficient
	Low vapour generation rate	Thin vapour film (High conductive heat transfer coefficient)
	Low vapour Reynolds number	Poor convective heat transfer

APPENDIX A KALININ'S MODEL EQUATIONS

$$\pi \rho_v U_o H_{fg} \frac{d}{dz} \left[\frac{(D-\delta)\delta}{\frac{\rho_v}{\rho_l} + (1 - \frac{\rho_v}{\rho_l})(1 - \frac{2\delta}{D})^2} \right]$$

$$= \frac{\pi k_v (T_w - T_s) A (U_o D / v_v)^m}{(\frac{2\delta}{D})^{1-m} [\frac{\rho_v}{\rho_l} - (1 - \frac{\rho_v}{\rho_l})(1 - \frac{2\delta}{D})^2]^m}$$

$$- \frac{\pi k_l (T_s - T_l) B (U_o D / v_l)^n (1 - 2\delta/D)^n}{[\frac{\rho_v}{\rho_l} + (1 - \frac{\rho_v}{\rho_l})(1 - \frac{2\delta}{D})^2]^n} \quad \dots A.1'$$

For the left part of the equation, saying $b = 2\delta/D$, we can reduce:

$$\pi \rho_v U_o H_{fg} \frac{d}{dz} \left[\frac{\frac{2}{D}(D-\delta) \frac{2}{D} \delta \frac{D^2}{4}}{[\frac{\rho_v}{\rho_l} (2b-b^2) + (1-b)^2]} \right] \quad \dots A.2$$

$$\pi \frac{D^2}{4} \rho_v U_o H_{fg} \frac{d}{dz} \left[\frac{(2-b)b}{[\frac{\rho_v}{\rho_l} b(2-b) + (1-b)^2]} \right] \quad \dots A.3$$

$$\pi \frac{D^2}{4} \rho_v U_o H_{fg} \left[\frac{(\frac{\rho_v}{\rho_l} b(2-b) + (1-b)^2)(2-2b) \frac{db}{dz}}{[\frac{\rho_v}{\rho_l} b(2-b) + (1-b)^2]^2} \right]$$

$$- \frac{(2-b)b [\frac{\rho_v}{\rho_l} (2-2b) - 2(1-b)] \frac{db}{dz}}{[\frac{\rho_v}{\rho_l} b(2-b) + (1-b)^2]^2} \quad \dots A.4$$

Expanding and simplifying, the left side becomes:

$$\frac{\pi \rho_v U_o H_{fg} \frac{D^2}{4} 2(1-b) \frac{db}{dz}}{\left[\frac{\rho_v}{\rho_l} b(2-b) + (1-b)^2 \right]^2}$$

...A.5

Finally:

$$\pi \rho_v U_o H_{fg} \frac{D^2}{4} \frac{db}{dz} =$$

$$\frac{\pi k_v (T_w - T_s) A (U_o D / v_v)^m \left[(1-b)^2 + \frac{\rho_v}{\rho_l} (2-b)b \right]^{2-m}}{2 b^{1-m} (1-b)}$$

$$- \frac{\pi k_l (T_s - T_l) B (U_o D / v_l)^n \left[(1-b)^2 + \frac{\rho_v}{\rho_l} (2-b)b \right]^{2-n}}{2(1-b)^{1-n}}$$

...A.6

APPENDIX B

THE PROPOSED MODEL

B.1 General Comments

The heat flux from the wall is assumed to be transferred by conduction, convection and radiation. For a laminar film, heat is transferred by conduction. As the Reynolds number increases and reaches a value corresponding to the turbulent zone, heat is transferred by convection. However, in the model, both modes will be included in the film boiling region. Initially, the vapour film velocity is very small and the vapour layer is thin, hence conduction heat transfer term is larger compared to the convection heat transfer term. As the distance from the quench front increases downstream, the vapour layer gets thicker and the vapour velocity increases. Hence the convection term increases and the conduction term decreases. Strictly speaking, these two modes can not be included together. However, this permits a smooth transition from the laminar zone to the turbulent zone and acts as a buffer zone.

A more rigorous method was used by Fung [1981]. His approach was similar to Hsu [1959]. For the turbulent vapour film, Fung assumes that the vapour film consists of a laminar sublayer and a turbulent layer. In the turbulent region, the power law velocity profile was assumed to hold each half of this turbulent region symmetrical about the line of maximum velocity. However a thermal sublayer δ_3 which is different from the laminar sublayer was used to get better agreement with experimental results.

In our case, only a laminar velocity profile was introduced. The wall friction factor is calculated from the velocity profile equation explained later. (The velocity profile equation is differentiated at the wall to obtain the shear stress.) Of course, the wall friction factor can take different values depending on the velocity profile but this method is thought to be more accurate than the one chosen by Kaufman [1976], who assumed a constant wall friction factor (0.005). The interfacial shear stress could be calculated at the interface by an identical procedure but an empirical equation which accounts for the waviness of the interface is used to predict the interface friction factor.

The total heat flux transferred from the wall can be divided into three components using the assumptions previously mentioned (see figure IV.1).

1) Calculation of $Q''_{\text{conduction}}$

The heat transfer by conduction through the film is calculated by:

$$Q''_{\text{conduction}} = \frac{k_v}{\delta} (T_w - T_{\text{sat}}) \quad \dots B.1$$

The vapour conductivity is obtained at the wall-saturation mean temperature. The vapour layer thickness δ is calculated by integrating the differential equation $\frac{d\delta}{dz}$ given in Appendix D. At the quench front, the vapour layer thickness is calculated from the Berenson's pool boiling correlation. This method was also used by Hsu and Westwater [1959].

2) Calculation of $Q''_{\text{radiation}}$

The radiation is evaluated by assuming an exchange between two infinite gray plates:

$$Q''_{\text{radiation}} = \frac{\sigma(T_w^4 - T_{\text{sat}}^4)}{\left(\frac{1}{\epsilon_w} + \frac{1}{\epsilon_l} - 1\right)} \quad \dots B.2$$

The emissivity of the liquid core (ϵ_l) was assumed to be equal to 1.0.

The wall emissivity is obtained from manufacturer's data. If the vapour film is not assumed to be a completely transparent medium, a more rigorous radiation heat transfer can be calculated using the Hottel [Holman, 1976] tabulations of gas emittance. The equations and discussion are given in Appendix C.

3) Calculation of $Q''_{\text{convection}}$

The convection heat transfer takes place from the wall to the vapour and then from the vapour to the interface.

(1) wall to vapour heat flux

The turbulent convective heat transfer from Groeneveld [1982] was used:

$$h_{w-v} = 0.0083 \frac{C k_{vw}}{(2\delta)} \left[\frac{\rho_v U_v 2\delta}{\mu_{vw}} \right]^{.877} \text{Pr}_{vw}^{.611} \quad \dots B.3$$

C: correction for increase of interfacial Area (1.5).

(ii) vapour to interface heat flux

The equation of Groeneveld [1982] can be used to predict the vapour to interface heat transfer coefficient.

$$h_{v-i} = 0.0083 \frac{k_{vs} C}{(2\delta)} \left[\frac{\rho_v (\bar{U}_v - U_1) 2\delta}{\mu_{vs}} \right]^{.877} Pr_{vs}^{.611} \quad \dots B.4$$

where U_1 is the interface velocity defined by $(\bar{U}_v + U_1)/2.0$ and U_1 is the liquid core velocity.

The total heat flux leaving the wall is then given by

$$Q''_{total} = \frac{k_v}{\delta} (T_w - T_{sat}) + h_{w-v} (T_w - T_v) + Q''_{radiation} \quad \dots B.5$$

The vapour temperature is assumed to be above saturation. A large degree of superheat was observed in the superheated steam temperature measurements obtained by Laperriere [1983]. It was shown that the superheated steam temperature follows the same trend as the wall temperature for a variation of mass flux, heat flux and inlet subcooling. The vapour temperature could be more accurately evaluated by an energy balance of the vapour film as shown in Figure B.1.

$$h_{w-v} (T_w - T_v) - h_{v-i} (T_v - T_i) = \text{heat absorbed by the vapour} \quad \dots B.6$$

$$P_v h_{w-v} (T_w - T_v) \Delta z - P_i h_{v-i} (T_v - T_i) \Delta z = (W_{v1} + \Delta W_v) H_{v2} - W_{v1} H_{v1} \quad \dots B.7$$

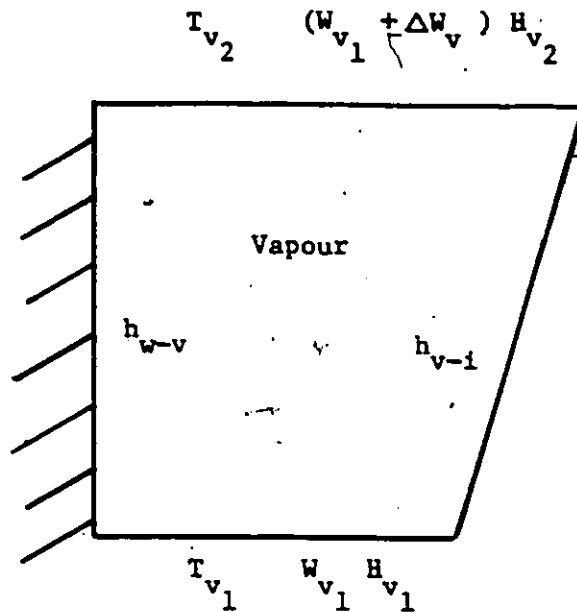


FIGURE B.1. Vapour Film Energy Balance

To solve this equation, the interfacial velocity U_i must be known to calculate h_{v-i} . It is difficult at this point to predict the validity of the assumption that $\bar{U}_i = (\bar{U}_v + U_i)/2$ so this method was not used to calculate the vapour temperature. Further work will have to be done in the interface velocity measurement to better understand the basic mechanisms. The vapour is assumed to be at the average of the wall temperature and the saturation temperature.

B.2 Description of Differential Equations

Six differential equations are prescribed to describe the inverted annular flow:

- 1) mass conservation equation for the vapour film
- 2) mass conservation equation of the liquid in the core
- 3) momentum conservation equation for the vapour film
- 4) momentum conservation equation for the core
- 5) energy equation of the liquid in the core
- 6) energy conservation equation in the vapour film.

The appropriate equations for the annular geometry are:

$$\text{liquid core area } (A_l) = \pi (r-\delta)^2 \quad \dots B.8$$

$$\text{vapour film area } (A_v) = \pi (r^2 - (r-\delta)^2) \quad \dots B.9$$

$$\text{liquid core perimeter } (P_l) = 2\pi (r-\delta) \quad \dots B.10$$

$$\text{vapour film perimeter } (P_v) = 2\pi r \quad \dots B.11$$

Each differential equation is now described:

- 1) The mass conservation equation for the vapour film is equal to:

$$\Gamma_v = \frac{d(\bar{U}_v A_v \rho_v)}{dz} \quad \dots B.12$$

Neglecting the change of density ρ_v with z , the equation can be rewritten

$$\Gamma_v = \rho_v \frac{d(\bar{U}_v A_v)}{dz} \quad \dots B.13$$

This equation means that the evaporation rate, Γ_v , generated at the interface must equal the increase in the vapour film flow rate.

Expanding, one obtains:

$$\Gamma_v = \rho_v \left[\bar{U}_v \frac{dA_v}{dz} + A_v \frac{d\bar{U}_v}{dz} \right] \quad \dots B.14$$

From the geometry:

$$A_v = \pi[r^2 - (r-\delta)^2] \quad \dots B.15$$

$$\frac{dA_v}{dz} = 2(r-\delta) \pi \frac{d\delta}{dz} \quad \dots B.16$$

Finally,

$$\frac{d\delta}{dz} = (\Gamma_v / \rho_v - A_v \frac{d\bar{U}_v}{dz}) / (2\pi(r-\delta)\bar{U}_v) \quad \dots B.17$$

2) Momentum conservation equation for the vapour film.

A friction factor, f_1 , is assumed at the interface and f_w on the wall. The momentum conservation equation for the film may be written as:

$$-\frac{dp}{dz} = \rho_v \bar{U}_v \frac{d\bar{U}_v}{dz} + \rho_v g + \frac{f_1 \rho_v P_v (\bar{U}_v - U_1)^2}{2A_v} + \frac{f_w \rho_v P_v \bar{U}_v^2}{2A_v} + \frac{\Gamma_v}{A_v} (\bar{U}_v - U_1) \quad \dots B.18$$

The terms on the right-hand side of the equation represent the acceleration, the gravitational force, the interface friction, the wall friction and the evaporation. For a vertical surface, various boundary conditions may be imposed [Kakac, 1977]

- 1) zero-interfacial shear stress ($\tau_i = 0$)
- 2) zero-interfacial velocity ($U_i = 0$)
- 3) zero wall shear stress ($\tau_w = 0$)
- 4) τ_i , U_i and $\tau_w \neq 0$.

The equation of the velocity profile is [Costigan, 1983]:

$$U_v(x) = U_v \left[1 - \left(\frac{x}{n\delta} - 1 \right)^2 \right] \quad \dots B.19$$

where $0.333 \leq n \leq 1.0$ depending on the velocity profile.

In this equation, it can be shown that U_v represents the maximum velocity which has not been clearly defined by Costigan [1983]. For zero interfacial shear (see Figure B.2), n is equal to 1.0 and the equation is reduced to:

$$U_v(x) = \frac{U_v}{\delta^2} (2x\delta - x^2) \quad \dots B.20$$

In a general form

$$U_v(x) = \frac{U_v}{\delta^2} \left(\frac{2x\delta}{n} - \frac{x^2}{n^2} \right) \quad \dots B.21$$

The previous equation can be differentiated and one obtains:

$$\frac{dU_v}{dx} = \frac{U_v}{\delta^2} \left(\frac{2\delta}{n} - \frac{2x}{n^2} \right) \quad \dots B.22$$

To calculate the value of x corresponding to the maximum value of U_v , the equation is set equal to zero

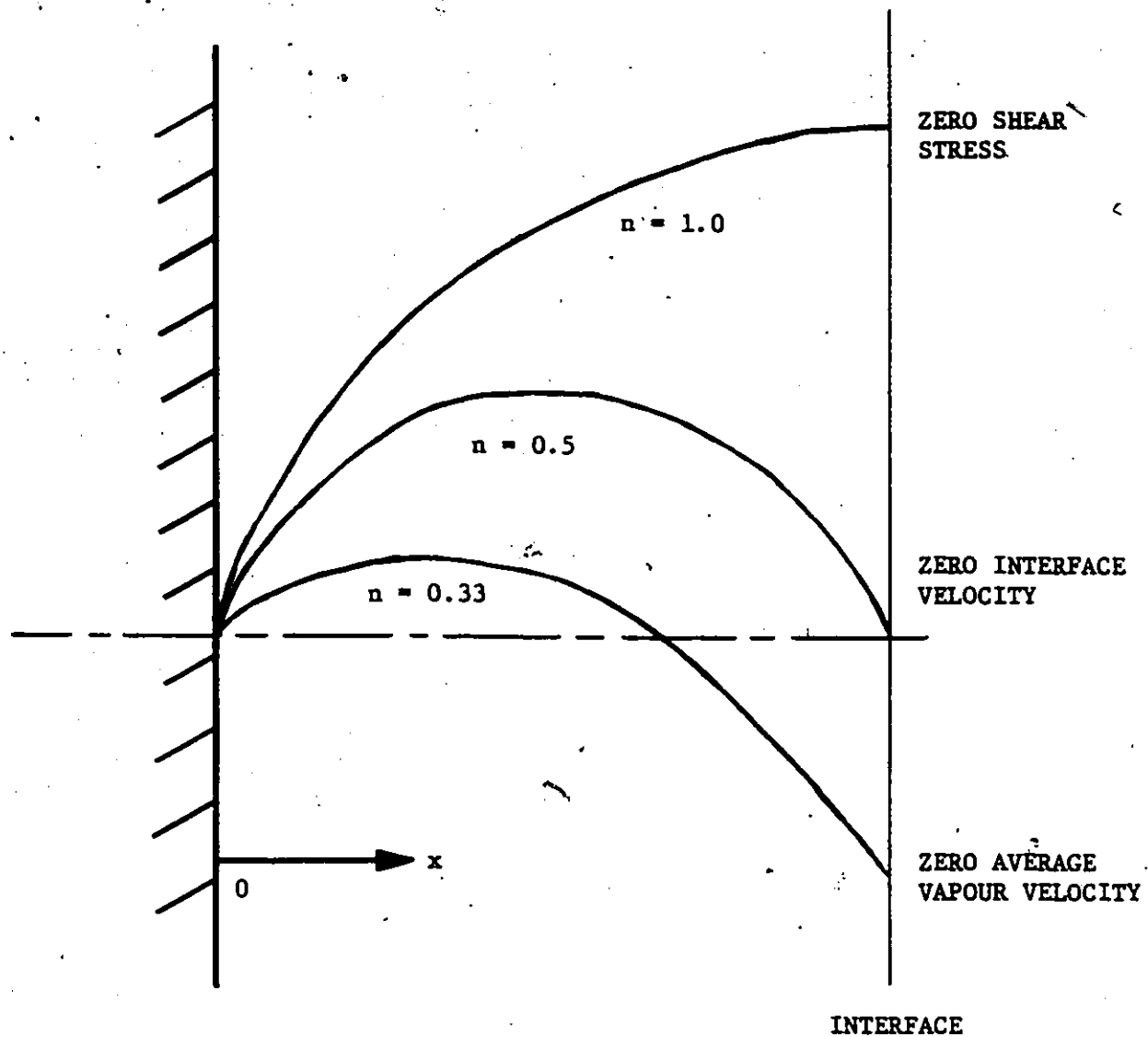


FIGURE B.2. Velocity profile in the vapour film

$$\frac{2\delta}{n} = \frac{2x}{n^2} \quad \dots B.23$$

$$x = n\delta \quad \dots B.24$$

For $n = 1$ for example at $x = \delta$:

$$U_v(\delta) = U_v \quad \dots B.25$$

This proves that U_v in the equation B.19 represents the maximum value of the velocity and not the average velocity corresponding to a given velocity profile.

Equation B.21 must then be integrated over the length of the vapour film to evaluate the mean vapour velocity. Mathematically:

$$\bar{U}_v = \frac{\int_0^\delta U_v(x) dx}{\delta} \quad \dots B.26$$

$$\bar{U}_v = U_v \left(\frac{3n-1}{3n^2} \right) \quad \dots B.27$$

where $\bar{U}_v$ is the average velocity in the vapour film and U_v is the maximum vapour velocity of the velocity profile. The local velocity based on the average velocity is then given by:

$$U_v(x) = \frac{3n^2}{(3n-1)} \cdot \frac{\bar{U}_v}{\delta^2} \left[\frac{2x\delta}{n} - \frac{x^2}{n^2} \right] \quad \dots B.28$$

The wall shear stress can be expressed using Newton's law of viscosity [Schlichting, 1960]:

$$\tau_w = \mu_v \left(\frac{dU_v}{dx} \right)_{x=0} \quad \dots B.29$$

$$= \frac{1}{2} f_w \rho_v \bar{U}_v^2 \quad \dots B.30$$

This means:

$$f_w = \frac{2}{\rho_v \bar{U}_v^2} \mu_v \left(\frac{dU_v}{dx} \right)_{x=0} \quad \dots B.31$$

$$f_w = \frac{12 \mu_v n}{\rho_v \bar{U}_v \delta (3n-1)} \quad \dots B.32$$

3) Momentum conservation equation for the liquid core:

$$-\frac{dp}{dz} = \rho_l U_l \frac{dU_l}{dz} + \rho_l g - \frac{f_1 \rho_v P_1 (\bar{U}_v - U_l)^2}{2 A_1}$$

As in the momentum equation for the vapour film (see Figure B.3), the right-hand-side of the equation represents the acceleration, the gravity effect and the interfacial shear friction.

4) Energy equation of the liquid in the core.

This equation is used to describe the removal of the subcooling of the water in the liquid core:

$$\rho_l A_l U_l (1-\alpha) c_{p_l} \frac{dT_l}{dz} = h_l (T_s - T_l) P_1 \quad \dots B.34$$

The void fraction in the core is α . However, α is equal to zero since all the vapour is moving in the film. The equation can be rewritten as:

$$\frac{dT_l}{(T_s - T_l)} = \frac{h_l P_1 dz}{\rho_l A_l U_l c_{p_l}} \quad \dots B.35$$

$$(T_s - T_1) = D e^{-Az} \quad \dots B.36$$

where $A = h_1 P_1 / (\rho_1 A_1 U_1 c_{p1})$.

The value of D can be evaluated from the initial conditions:

$$\text{at } z = 0, \quad D = (T_s - T_{10})$$

where T_{10} is the liquid temperature at $z = 0$.

5) Mass conservation equation of the liquid in the core.

The evaporation rate per unit length may be expressed by:

$$-\Gamma_v = \frac{d}{dz} [\rho_1 A_1 U_1] \quad \dots B.37$$

Neglecting the change of ρ_1 , the equation is rewritten:

$$-\Gamma_v = \rho_1 \frac{d}{dz} [A_1 U_1] \quad \dots B.38$$

6) Energy conservation equation in the vapour film.

A heat balance is performed on the vapour film as shown in Figure B.4.

$$Q''_v P_v \Delta z - Q''_1 P_1 \Delta z = (W_{v1} + \Delta W_v) H_{v2} - W_{v1} H_{v1} - \Delta W_v H_f \quad \dots B.39$$

The LHS of the equation can be rewritten as:

$$P_v Q''_v \Delta z \quad \dots B.40$$

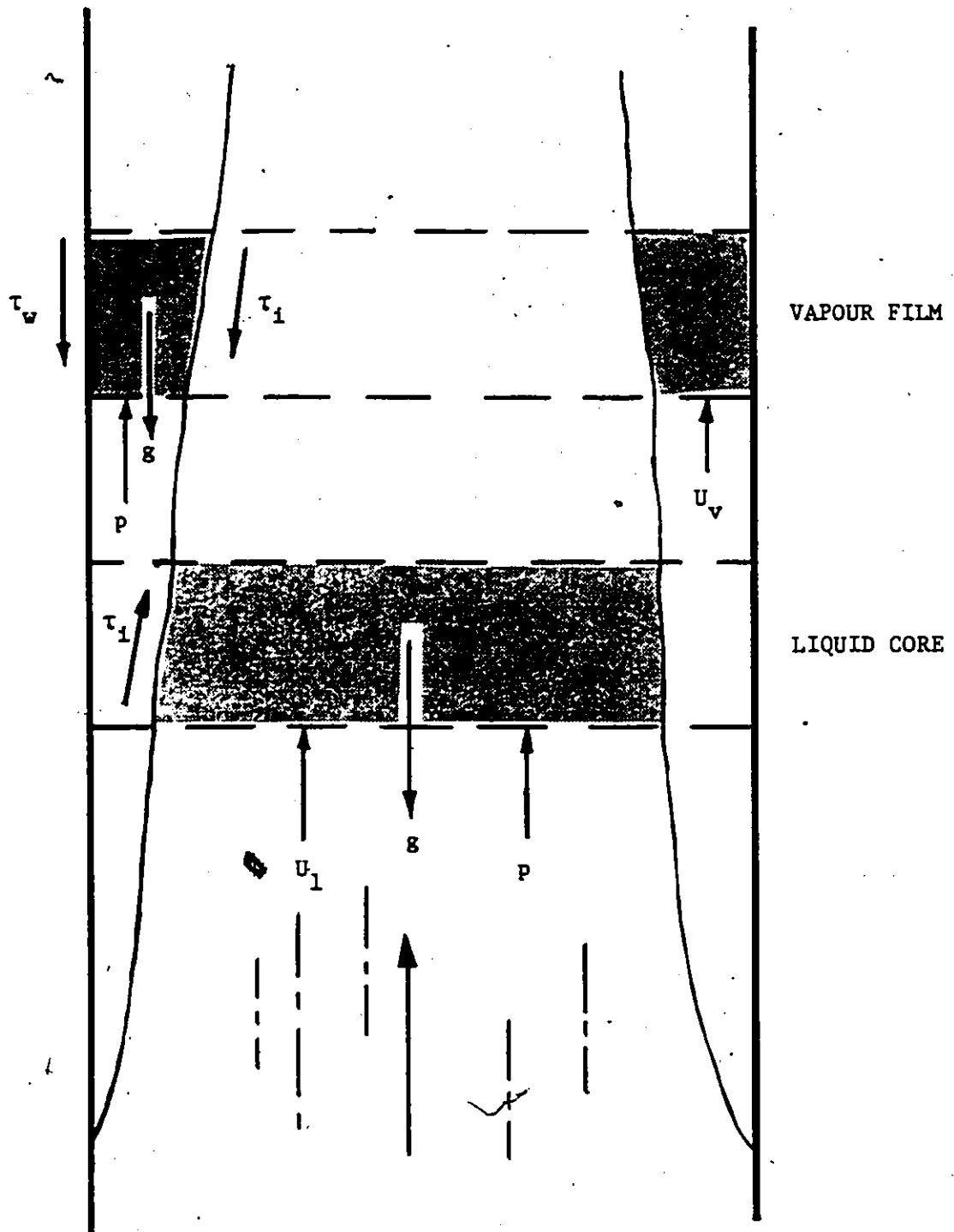


FIGURE B.3. Control volume of the inverted annular film boiling

Q_v'' is evaluated knowing the total heat flux and also the heat flux used to remove the subcooling (Q_1''). The energy equation is then simplified to:

$$\Delta W_v = \frac{P_v Q_v'' \Delta z - W_v (H_{v2} - H_{v1})}{(H_{v2} - H_f)} \quad \dots B.41$$

The vapour enthalpy change from 1 to 2 is small compared to the other term $P_1 Q_v''$ and the vapour temperature was assumed to be at $(T_w + T_{sat})/2$

$$\frac{W_v}{\Delta z} = \Gamma_v = \frac{P_v Q_v''}{H_{fg} + c_{p_v} (T_v - T_{sat})} \quad \dots B.42$$

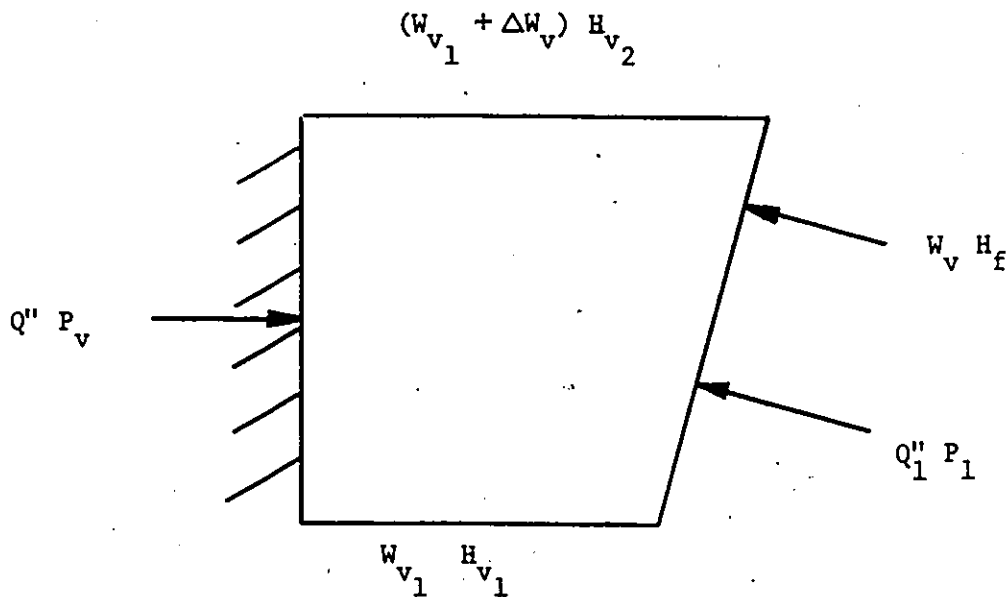


FIGURE B.4: Vapour film heat balance

B.3 Constitutive Equations

B.3.1 Interfacial vapour generation rate

The total vapour generation, Γ_v , is expressed by:

$$\Gamma_v = \frac{Q_v'' P_v}{(H_{fg} + c_{p_v}(T_v - T_{sat}))} \quad \dots B.43$$

A modified heat of vaporization takes into account the superheating of the vapour in equation B.43 as developed in section B.2. Q_v'' is evaluated knowing the total heat flux, Q'' , and also the heat flux used to remove the subcooling (Q_1''). At the interface

$$P_v Q'' = P_v Q_v'' + P_l Q_1'' \quad \dots B.44$$

Then

$$Q_v'' = (P_v Q'' - P_l Q_1'') / P_v \quad \dots B.45$$

The removal of the core water subcooling (Q_1'') can be evaluated using the Dittus-Boelter correlation

$$h_l = \frac{0.023 k_l C}{2(r-\delta)} \left(\frac{(U_1 - U_l) 2(r-\delta)}{\nu_l} \right)^{0.8} \text{Pr}_l^{0.4} \quad \dots B.46$$

C: correction for increase of interfacial Area (1.5)

$$Q_1'' = h_l (T_{sat} - T_l) \quad \dots B.47$$

where U_1 is the liquid velocity in the core and U_i is the interface velocity. The interface velocity, U_i , is the average of the liquid velocity and the vapour velocity.

$$U_i = (\bar{U}_v + U_1)/2.$$

B.3.2 Velocity Profile

As shown in the velocity profile equation, several values of n can be chosen. However, the value of n can be calculated assuming that the velocity at the interface is the average of the vapour velocity and the liquid core velocity at $x = \delta$:

$$U_v(\delta) = \frac{3n^2}{(3n-1)} \frac{\bar{U}_v}{\delta^2} \left[\frac{2\delta^2}{n} - \frac{\delta^2}{n^2} \right] \quad \dots B.48$$

$$U_v(\delta) = \frac{3n^2}{(3n-1)} \frac{2n-1}{n^2} \bar{U}_v \quad \dots B.49$$

$$\left[\frac{2n-1}{3n-1} \right] 3 \bar{U}_v = \left(\frac{\bar{U}_v + U_1}{2} \right) \quad \dots B.50$$

$$6 \left[\frac{2n-1}{3n-1} \right] = \frac{\bar{U}_v + U_1}{\bar{U}_v} \quad \dots B.51$$

Saying

$$\frac{\bar{U}_v + U_1}{\bar{U}_v} = C \quad \dots B.52$$

$$(12n - 6) = (3n-1) C \quad \dots B.53$$

$$n = \frac{6-C}{12-3C} \quad \dots B.54$$

B.3.3 Interfacial Friction Factor

The interfacial friction factor or effective interface roughness, f_1 , initially derived by Wallis [1970] for annular flow was used.

$$\frac{f_1}{f_v} = (1 + 300 \delta/D) \quad \dots B.55$$

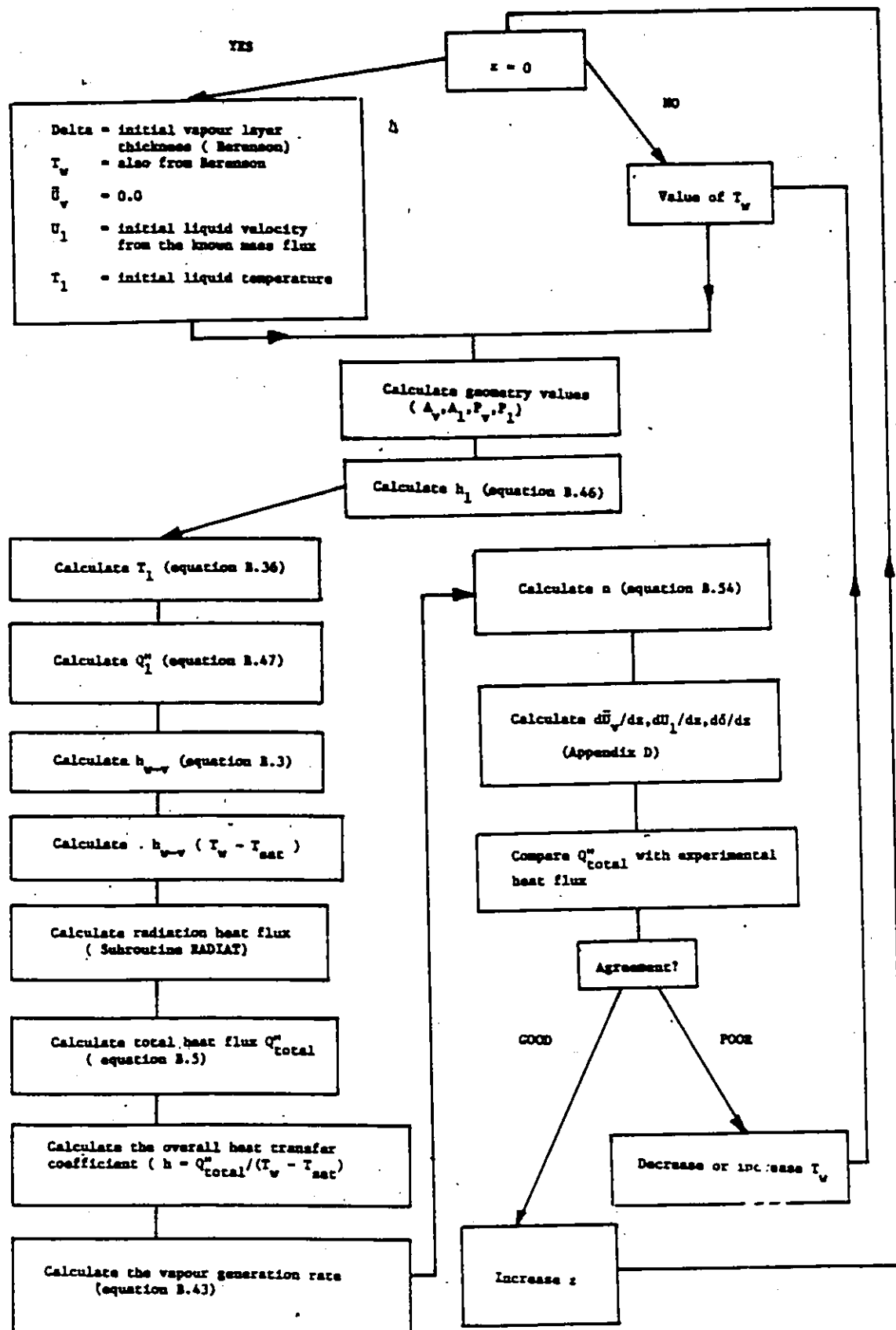
where f_1 is the interfacial friction factor and f_v is the smooth tube value. The equation shows that the interfacial friction factor increases with δ . The interface becomes disturbed by ripples with an increase in the vapour layer thickness. A value of 0.005 was chosen for f_v as proposed by Kaufman [1976].

B.3.4 Calculation of wall temperature and vapour properties

In this model, an iterative procedure was used to evaluate the wall temperature. An initial value of the wall temperature was estimated. The experimental heat flux was then compared with the heat flux predicted by the model. If the heat flux of the model was lower than the experimental heat flux, the wall temperature was increased. In the opposite case, the wall temperature was decreased. This iterative method was used since the properties of the vapour had to be evaluated at the mean of the wall temperature and saturation temperature. Then the heat transfer coefficient could be easily calculated knowing the heat flux (based on the wall temperature minus the saturation temperature).

B.4 Numerical Solution

An AECL package, Forsim VI [Carver, 1979] was used to solve the system of differential equations. A computer listing of the model is given at the end of the appendix. The main program is divided into four principal parts: storage of information; initial conditions at the quench front, dynamic section to solve the differential equations and an output section.



SUBROUTINE UPDATE

CALCULATION OF FILM BOILING HEAT
TRANSFER ABOVE THE QUENCH FRONT DURING REFLOODING
THIS PROGRAM WAS WRITTEN BY A. LAPERRIERE

NOMENCLATURE OF VARIABLES

DELTA	VAPOR LAYER THICKNESS	M
DELTA Z	VAPOR LAYER THICKNESS VARIATION	
EM	LIQUID CORE EMISSIVITY	
EV	VAPOR EMISSIVITY	
EW	WALL EMISSIVITY	
FLAG	1: PRINT OF ALL RESULTS2: FEW RESULTS	
G	GRAVITATION ACCELERATION	M/SEC**2
GA	VAPOR IN FILM AREA	M**2
GGP	VAPOR GENERATION RATE IN THE FILM	KG/(M**2*SEC)
GP	VAPOR PERIMETER	M
GV	VAPOR VELOCITY	M/SEC
GVZ	VARIATION OF VAPOR VELOCITY	
HFT	HEAT FLUX TEST SECTION	M/M**2
HFMF	HEAT FLUX HOT PATCH	M/M**2
HL	SUBCOOLED LIQUID HEAT TRANSFER	M/(M**2*SEC C)
HPL	HOT PATCH LENGTH	M
HT	HEAT TRANSFER (MODEL)	M/(M**2*SEC C)
HTDL	HEATED LENGTH EXCLUDING HOT PATCH	M
HTE	HEAT TRANSFER (EXPERIMENTAL)	M/(M**2*SEC C)
ICHECK	DIAGNOSTIC PRINT CONTROL	
ISUB	INLET SUBCOOLING	KJ/KG
KTS	TEST SECTION CONDUCTIVITY	M/(M*SEC C)
MA	LIQUID CORE AREA	M**2
METHOD	INTEGRATION METHOD	
MP	LIQUID CORE PERIMETER	M
PARA	QSL = PARA HL (TS - TLC)	
P1	OUTLET PRESSURE	KPA
QDC	TEST SECTION POWER	M
QMP	HOT PATCH POWER	M
QS	HEAT FLUX (RADIAT., CONV. AND COND.)	M/M**2
QSL	HEAT FLUX TO THE LIQUID CORE	M/M**2
QSP	HEAT FLUX BY RADIATION	M/M**2
QSV	HEAT FLUX TO THE VAPOR	M/M**2
QUA	QUALITY IN THE TEST SECTION	
RAD	RADIUS(INSIDE)	M
RADO	RADIUS(OUTSIDE)	M
TFIN	FINAL VALUE OF INTEGRATION	M
TL	LIQUID INLET TEMPERATURE	DEG C
TLC	CORRECTED INLET TEMPERATURE	DEG C
TS	SATURATION TEMPERATURE	DEG C
TSLBHP	TEST SECTION LENGTH BEFORE HOT PATCH	M
TV	VAPOUR TEMPERATURE	C
TW	INSIDE WALL TEMPERATURE	DEG C
TWB	BEPENSON WALL TEMPERATURE	DEG C
TWE	INSIDE WALL TEMPERATURE(EXP)	DEG C
TWA	OUTSIDE WALL TEMPERATURE(EXP)	DEG C
TWIC	INSIDE WALL TEMPERATURE	DEG C
TWEXP	EXPERIMENTAL INSIDE WALL TEMP.	DEG C
TWMOO	MODEL INSIDE WALL TEMPERATURE	DEG C
TW1	VAPOUR TEMPERATURE USED IN PROPER	DEG C
TW2	MEAN WALL VAPOUR TEMPERATURE	DEG C
TW3	MEAN VAPOUR SATURATION TEMPERATURE	DEG C
TW22	OUTSIDE WALL TEMPERATURE(EXP)	DEG C
VM	LIQUID CORE VELOCITY	M/SEC
VHZ	LIQUID CORE VELOCITY VARIATION	
M	MASS FLOW	KG/M
M1	MASS FLOW	KG/SEC
M2	MASS FLUX	KG/(M**2*SEC)
XI, XI'	INLET QUALITY OF THE TEST SECTION	
X1	QUALITY (INLET OF HOT PATCH)	
X2	QUALITY (OUTLET OF HOT PATCH)	

```

80      C      THIS PROGRAM WAS DERIVED USING THE FOLLOWING TEST SECTION SPECIFICATION:
      C
      C      INCONEL 600 TUBING
      C      I.O. = 0.8928 CM
      C      O.D. = 1.114 CM
      C      HEATED LENGTH EXCLUDING HOT PATCH: 197.0 CM
85      C      HOT PATCH LENGTH: 28.1 CM
      C      TEST SECTION LENGTH BEFORE HOT PATCH: 16.5 CM
      C      TEST SECTION LENGTH AFTER HOT PATCH: 184.5 CM
      C
      C      THE TEST SECTION CONDUCTIVITY USED IS FROM REFERENCE (1)
90      C      THE TEST SECTION EMISSIVITY USED IS FROM REFERENCE (2)
      C      REFERENCE: 1. GROENEVELD, D.C., THE THERMAL BEHAVIOUR OF A HEATED SURFACE
      C                      AT AND BEYOND DRYOUT, AECL-4309
      C                      2. STEWART, J.C., LOW QUALITY FILM BOILING AT INTERMEDIATE AND
95      C                      ELEVATED PRESSURE, M. S. J. OTTAWA UNIV., 1981
      C      FLAG.....1 PRINTOUT OF ALL RESULTS AND PARAMETERS
      C      .....2 PRINTOUT OF FEW RESULTS
      C
      C      .....
100     C
      C      .....SECTION 1.....STORAGE
      C      .....
105     C      COMMON/RESERV/Z,DZ,DTOUT,EMAX,TFIN,METHOD,CTMAX,CTMIN,ICMEK
      C      COMMON/PLOT/X2,XOUT
      C      COMMON/CNTRPOL/INOUT
      C      COMMON/INTEGR/GV,VH,DELTA
      C      COMMON/DERIVT/GVZ,VHZ,DELTAZ
110     C      COMMON/FUNTEL/TWA(14,2)
      C      COMMON/TEMP/TS,TM1,TL,P1
      C      COMMON/PROP/CG,CL,DL,GCON,GD,GDYVIS,HF,HFG,HIN,KL,KVIS,PRL,JOH
      C      COMMON/PROP1/GDC,DLC,HV
      C      COMMON/DATA/QUA,HT,HTE,TW,TWE
      C      REAL 14,HP,KL,KVIS,KTS,XSUB

115     35      CONTINUE
      C      INITIALIZE PARAMETERS OF FORSIM
      C      TFIN=1.849
      C      METHOD = 8
      C      ICMEK = 1
120     C      FLAG = 2.0
      C
      C      THE GEOMETRY OF THIS GIVEN TEST SECTION INCLUDING THE HOT
      C      PATCH IS GIVEN IN THE FOLLOWING LINES:
125     C      RAD = 0.884464
      C      RADO = 0.88557
      C      HTDL = 1.97
      C      HPL = 0.101
      C      TSLBNP = 0.165
      C
      C      THE INLET TEMPERATURE, SATURATION TEMPERATURE, TEST SECTION POWER,
      C      HOT PATCH POWER, PRESSURE, MASS FLOW, GRAVITATION ACCELERATION,
      C      PAPA AND THE AVERAGE WALL TEMPERATURE ARE DEFINED:
130     C      TL = 210.2
      C      QDC = 11880.0
      C      QHP = 1730.0
135     C      P1 = 4072.0
      C      H = 83.3
      C      G = 9.8
      C      PAPA = 1.50
140     C      CALL TSATUR(P1,TS)
      C
      C      EVALUATE HEAT FLUX
      C      TWE2 = 273.0 + FUNC(Z,TWA,14)
      C      HFHP = QHP/(2.*3.1416*PAD*HPL)
145     C      HFT = QDC/(2.*3.1416*PAD*HTDL)
      C      CALCULATE TEST SECTION CONDUCTIVITY FOR DIFFERENT WALL TEMPERATURE
      C      KTS = (0.14375 + 0.00019*(TWE2 - 273.0))*103.0
      C      CALCULATE EXPERIMENTAL INSIDE WALL TEMPERATURE FROM
      C      OUTSIDE EXPERIMENTAL WALL TEMPERATURE.
150     C      IF (Z.LT. HPL) TWE = TWE2 - (HFHP*PAC*2*LOG(RADO/PAD))/KTS

```

```

C1 = (RADO**2. / (RACO**2. - RADO**2.)) * ALOG(RADO/RAC) - 0.51
C2 = 100 / (2.0 * 3.1416 * KTS * HTD)
IF (Z .GE. MPL) TNE = TNE2 - C2 * C1
155 C INTERPOLATION OF THE INSIDE WALL TEMPERATURE
C AN INITIAL GUESS OF THE WALL TEMPERATURE
C IS MADE USING BERENSON'S CORRELATION
C THIS ALSO PROVIDES THE INITIAL
C VAPOR FILM THICKNESS
IF (Z.GT.0.0) GO TO 70
160 X = 0.0
N2 = W / (3600.0 * 3.1416 * RAO**2.0)
CALL BERENSON(N2,X,P1,TWB,HFT)
CALL TSATIN(P1,TS)
HTR = HFT / (TWB - TS)
165 CALL PROPER
DELTA = GCON / HTR
TW = TWB + 273.0
70 CONTINUE

170 C CALCULATE THERMODYNAMIC PROPERTIES USING CENL POLS-W ROUTINE

C *** NOTE: THIS POLS-W ROUTINE IS DIFFERENT FROM
C THE ADVANCE ENGINEERING BRANCH POLS-W
C ROUTINE
175 CALL PROPER

C
C
C
C SECTION 2...INITIAL CONDITIONS
C
C
C
185 IF (Z.GT. 0.) GO TO 20
N = 0
GV = 0.01
C CALCULATION OF THE CORRECTED INLET TEMPERATURE AND THE CORRECTED
C INLET ENTHALPY DUE TO HEAT LOSS.
C THE AMBIENT TEMPERATURE OF 30.0 DEG C IS ASSUMED.
190 XIN = ((HIN-HF) / HFG) - 4.66 * (TL-30.0) / (2.4 * W2 * 3.1416 * RAD**2. * HFG)
CALL TEMP(XIN,TL)
TL = TLC
CALL PROPER
VM = W / (3600.0 * 3.1416 * RAD**2. * OIN)
195 ISUB = XIN * HFG / 1000.0
W2 = W / (3600 * 3.1416 * RAD**2.)

C
C
C
205 C SECTION 3...DYNAMIC SECTION
C
C
C
205 C GEOMETRY VALUES ARE DEFINED
20 GA = 3.1416 * (RAD**2. - (RAD-DELTA)**2.)
HA = 3.1416 * (RAD - DELTA) ** 2.
MP = 2 * 3.1416 * (RAD-DELTA)
GP = 2 * 3.1416 * RAD
215 C
C
C
215 C CALCULATION OF THE QUALITY
C CORRECTIONS ARE MADE FOR HEAT LOSS AT THE POWER CLAMPS.
W1 = W / 3600.
XI = XIN
X1 = XI * HFT * 3.1416 * 2. * RAD * TSLBHP / (W1 * HFG)
220 X2 = X1 * OMP / (W1 * HFG)
F1 = HFT * 3.1416 * 2. * RAD / (W1 * HFG)
IF (Z.LT.MPL) QUA = (X2 - X1) * 2 / MPL * X1
IF (Z.GT.MPL) QUA = X2 + F1 * (Z - MPL)
IF (Z.GT.1.73) XOUT = QUA + 0.01
C HEAT FROM REMOVAL OF SURF COOLING
225 DO 15 N1 = 1.5
VI = (VM + GV) / 2.0
PEL = ABS((VM - VI) * 2. * (RAD - DELTA) / KVIS)
ML = 0.023 * (KL / (Z * (RAD - DELTA))) * PEL**0.6 * PRL**0.6

```

```

235 C = HL*HP*2/(DL*MA*VM*CL)
      B = (TS-TLC)/EXP(C)
      TL = (TS-B)
      QSL = HL * B * PARA
      CALL PROPER
235 15 CONTINUE
      C WALL-VAPOUR CONVECTIVE HEAT TRANSFER COEFFICIENT
      TW2 = ((TM-273.0+TS)/2.0 + TM-273.6)/2.0
      C CALCULATE THE VAPOR TEMPERATURE
      TV = (TM-273.0 + TS)/2.0
      TW1 = TV
      CALL PROPER
245 PEV1 = 2.0 * DELTA * GO
      C THE THERMODYNAMIC PROPERTIES ARE CALCULATED AT MEAN WALL-VAPOUR TEMPERATURE.
      TM1 = TW2
      IF(TM1.GT.799.0) TM1 = 799.0
      CALL PROPER
245 REV = REV1 * GV/GOYVIS
      HVK1 = 0.0083 * ABS(REV)**0.877 * (CG*GOYVIS/GCON)**0.611
      * GCON/(2.0 * DELTA)
      C TM1 HAS TO BE RETURNED TO THE MEAN WALL-INTERFACE TEMPERATURE.
      TM1 = TV
250 CALL PROPER
      C CALCULATES HEAT TRANSFER FOR — EXPERIMENTAL
      C AND — MODEL
      TS = AMAX1(1.E-6,DELTA)
      A = (GCON/TS)*(TM-TS-273.0)
255 C CALCULATION OF THE RADIATION HEAT
      CALL RADIA(TM,RAD,DELTA,TS,P1,QSR)
      C THE HEAT FLUX IS CALCULATED USING THE CONVECTION, CONDUCTION
      C AND RADIATION
      QS=A+QSR+PARA*HVK1*(TM-273.0-TV)
260 HTE=HFT/(TME-TS-273.0)
      HT = TS / (TM - TS - 273.0)
      C VAPOR GENERATION RATE CALCULATED
      QSV=(GP-QS-HP*QSL)/GP
      GGR = (QSV-GP)/(HFG+CG*(TM1-TS))
265 C SELECTION OF N(RN) DEPENDING ON THE VELOCITY PROFILE
      C = (GV + VM)/GV
      RN = (6.0 - C)/(12.0 - 3.8°C)
270 IF(RN.GT.1.0) RN = 0.999
      IF(RN.LT.0.333) RN = 0.333
      C VOID FRACTION CALCULATION
      VOID = GA/(MA + GA)
275 C
      C
      C VARIATION IS CALCULATED FOR THE FOLLOWING:
      C 1. DELTA
      C 2. VM
      C 3. GV
280 C
      C CALCULATION OF INTERFACIAL FRICTION FACTOR
      FI = 0.005*(1.0 + 300.*DELTA/(2.0 * RAC))
      Y1 = 2.*3.1416*GV*(RAD-DELTA)+(GA*VM**2.*DL*2.*3.1416
285
      A*(PAD-DELTA)/(GO*GV*MA)
      Y2 = GGR*(1+GA*VM/(MA*GV))/(GO-GA*(DL-GO)*G/(GO*GV))+
      A*GP*GOYVIS*RN/((3*RN-1)*DELTA-GO)+GGR*(1-VI/GV)/GO
      A + FI*HP*(GV-VI)**2.0*GA/(2.0*MA*GV)
      A + FI * GP*(GV-VI)**2.0*(2.0*GV)
      DELTAZ = Y2/Y1
290
      VMZ = (-GGP-VM*DL*2.*
      A*3.1416*(DELTA-PAD)*DELTAZ)/(MA-DL)
295
      GVZ = (G*(DL-GO)+DL*VM*VMZ-6*GP*GOYVIS*GV*RN/(DELTA*GA
      A*(3*RN-1)) - GGR*(GV-VI)/GA
      A-FI*GP*HP*(GV-VI)**2.0/(2.0*MA)
      A-FI*GO*GP*(GV-VI)**2.0/(2.0*GA))/(GO*GV)
300 PV=QSR/QS
      PARTC=A/QS
      PARTCV=(QS-QSP-A)/QS
      C
      C ITERATION OF THE INSIDE WALL TEMPERATURE
305 IF(Z.EQ.0.0) GO TO 28
      QDIF = QS - HFT
      IF(Z.LT.HPL) QERROR = ABS(QDIF/HFT)
      IF(Z.GE.HPL) QERROR = ABS(QDIF/HFT)
      IF(QERROR.GT.0.01) AUG = 1.0
310 IF(QERROR.GT.0.02) AUG = 3.0

```


[illegible]

```

1      SURROUTINE MYPLOT(N,X,HTCHOD,HTCEXP,TWMOO,TWEXP)
      .....
5      THIS PROGRAM WILL PLOT
      1. HEAT TRANSFER COEFFICIENT .VS. QUALITY
      2. WALL TEMPERATURE .VS. QUALITY
10     THESE DATA POINTS WERE OBTAINED ON A DIFFERENT PROGRAM
      - USING THE FORSIM PACKAGE AT CRNL
      .....
15     REAL X(N),HTCMOD(N),HTCEXP(N),TWMOD(N),TWEXP(N)
      REAL X1(50),Y1(50),Y2(50),Y3(50),Y4(50)

      COMMON/SCALES/XMIN,XMAX,TICX,FXSCAL,YMIN,YMAX,TICY,FYSCAL
      COMMON/PLOT/X2,XOUT
      COMMON/FORNTAT/XL,YL,FTITLE(2),FSUBTIT(2),FXAXTIT(3),FYAXTIT(3),
      * MSTVARS,STVARS(10)
      COMMON/SYMEAN/SYMCOD(10),FSYMB1(2),FSYMB2(2),FSYMB3(2),FSYMB4(2),
      * FSYMB5(2),FSYMB6(2),FSYMB7(2),FSYMB8(2),FSYMB9(2),
      * FSYMB10(2),SYMLOCX,SYMLOCY

25     C      INITIALIZE VARIABLES
      SYMB = SYMCOD(1) = 1.0
      XL = 4.0
      YL = 4.0
      XMIN = X2
      XMAX = XOUT + 0.01
      YMIN = 0.0
      YMAX = 1200.0
      FXAXTIT(1) = 10H(=EQUILIBR
      FXAXTIT(2) = 10H1UM QUALIT
      FXAXTIT(3) = 10H(Y=)
      FYAXTIT(1) = 10H(= 4.T. CO
      FYAXTIT(2) = 10HEFFICIENT
      FYAXTIT(3) = 10H (W/H2-G)
      FYAXTIT(4) = 10H(=)

35     C      COLLECT THE DATA POINTS EXCLUDING
      C      THE FIRST CALCULATED POINTS
      IC = 0
      DO 20 I = 6,N
      IC = IC + 1

```

```

45      X1(IC) = X(2)
        Y1(IC) = M*CHOO(I)
        Y2(IC) = M*CEXP(I)
        Y3(IC) = T*MOO(I)
        Y4(IC) = T*EXP(I)
50      20 CONTINUE
        C PRINT OF THE DATA OF THE PLOTS
45      FOPMAT (5X,F9.3,4X,I3.3X,F6.1,2X,F6.1,3X,F6.1,5X,F6.1)
        PRINT*, 'QUALITY N M*CHOO M*CEXP T*MOO(C) T*EXP(C)'
        DO 10 I=1,IC
55      PRINT 45,X1(I),I,Y1(I),Y2(I),Y3(I),Y4(I)
        10 CONTINUE
        ENCODE(40,260,FSYMB1)

        CALL SIMPLT(1,X1,Y1,IC,0.1)
        SYMCOO(1)=3.0
60      ENCODE(40,250,FSYMB1)
        CALL SIMPLT(3,X1,Y2,IC,2.3)
        FYAXTIT(1)=10H(*WALL TEM
        FYAXTIT(2)=10H(TEMPERATURE
        FYAXTIT(3)=10H(DEG C)*)
65      YMIN=200.0
        YMAX=1800.0
        SYMCOO(1)=1.0
        ENCODE(40,260,FSYMB1)
        CALL SIMPLT(1,X1,Y3,IC,8.1)
70      SYMCOO(1)=3.0
        ENCODE(40,250,FSYMB1)
        CALL SIMPLT(3,X1,Y4,IC,8.3)

250     FOPMAT(21H(*EXPERIMENTAL DATA*))
75     260     FOPMAT(14H(*MODEL DATA*))

        RETURN
        END

```

```

1      SUBROUTINE BERENSH(G1,X,PKPA,TWB,PHI)
        C THIS SUBROUTINE IS USED TO CALCULATE THE WALL C
        C TEMPERATURE USING THE BERENSON CORRELATION C
        C J.OF HEAT TRANSFER,83,PP553-558(1982) C
5      CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
        COMMON/TEMP/TS,TW1,TL,P1
        COMMON/PROP/CG,CL,OL,GCON,GD,GDYVIS,HF,HFG,HIN,KL,KVIS,PR,CON
        COMMON/PROPL/GOC,DLC,HV
        C SINCE THE PROPERTIES MUST BE EVALUATED AT THE FILM C
        C TEMPERATURE ((TW+TSAT)/2.0), AN ITERATIVE PROCEEDURE IS USED C
        C TO CALCULATE THE WALL TEMPERATURE KNOWING THE HEAT FLUX C
        C AN INITIAL GUESS OF TW(DEG C) IS DONE 1700
        TW = 780
        P1 = PKPA
        CALL TSATUP(PKPA,TSAT)
15      SIG = SIGMA(TSAT)
        DO 40 I = 1,200
        TW1 = (TW + TSAT)/2.0
        CALL PROPER
        HT = 0.425*((GCON**3.0*9.81*GD*(DLC-GD)*(HV-HF))/
        3(GDYVIS*(TW - TSAT) * (SIG/(9.81*(DLC-GD))**0.5))**0.25
        TW2 = (PHI/HT) + TSAT
        DIF = TWB - TW
        IF(DIF.LT.0.0) TW = TW + 0.99
25      IF(DIF.GT.0.0) TW = TW + 1.01
        DIFA = ABS(DIF)
        IF(DIFA.LT.5.0) GO TO 20
        40 CONTINUE
        20 CONTINUE
        RETURN
        END

```

```

1      FUNCTION SIGMA(TEMP)
        CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
        C THIS FUNCTION IS USED TO CALCULATE THE SURFACE TENSION C
        C AT A GIVEN SATURATION TEMPERATURE (N/M) C
5      CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
        DIMENSION A(5)
        DATA A/1.160936807E-31,1.121404686E-03,-5.752805150E-06,
        1.276274650E-08,-1.149719290E-11/
        B = 0.83
        TC = 647.3
        T = TEMP + 273.0
        SIGMA = A(1) + (TC-T)**2/(1.0 + B * (TC - T))
        DO 10 I = 2,5
10      SIGMA = SIGMA + A(I) * (TC - T) ** I
        SIGMA = SIGMA/1000.0
        RETURN
        END

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1      SUBROUTINE PROPER
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F O P S I M V I

- FORTRAN ORIENTED DISTRIBUTED SYSTEM SIMULATION PACKAGE •
- FOR PARTIAL AND ORDINARY DIFFERENTIAL EQUATIONS •
- USERS MANUAL - ATOMIC ENERGY OF CANADA REPORT AECL 5821 •
- AUTHORS M B CARVER AND D G STEWART, C.R.N.L./A.E.C.L. •
- OPTIMIZATION OPTION INSERTED BY C F AUSTIN AND M B •
- CARVER, C.R.N.L./A.E.C.L. USER MANUAL AECL E6242 •

FILM BOILING HEAT TRANSFER ABOVE THE QUENCH FRONT

TABLE TWA IS 13 BY 2 DEPENDANT VARIABLE IS LISTED ON THE RIGHT

0.	.752000E+03
.101000E+00	.752000E+03
.129000E+00	.480000E+03
.138000E+00	.466000E+03
.298000E+00	.521000E+03
.398000E+00	.563000E+03
.498000E+00	.549000E+03
.798000E+00	.626000E+03
.189900E+01	.658000E+03
.139800E+01	.685000E+03
.169800E+01	.697000E+03
.184900E+01	.713000E+03
.190600E+01	.713000E+03

PRINTOUT TIMES ARE LISTED BELOW

0.	.100E+02	.250E+01	.500E+01	.750E+01	.102E+00	.125E+00	.150E+00
.178E+00	.200E+00	.275E+00	.250E+00	.275E+00	.300E+00	.325E+00	.350E+00
.400E+00	.450E+00	.500E+00	.550E+00	.600E+00	.650E+00	.700E+00	.750E+00
.800E+00	.850E+00	.900E+00	.950E+00	.100E+01	.105E+01	.110E+01	.115E+01
.120E+01	.125E+01	.130E+01	.135E+01	.140E+01	.145E+01	.150E+01	.155E+01
.168E+01	.175E+01	.170E+01	.175E+01	.180E+01	.185E+01		

CASE NUMBER	1 USING	ADAMS	(VARIABLE STEP)	INTEGRATION ALGORITHM	WITH	SPARSE	JACOBIAN ANALYSIS
				INTEGRATING	3 EQUATIONS		
X = -.181E+00	HTCMOD=	.631E+03	HTCEXP=	.448E+03	TWMOD=	630.54	TWEXP= 720.40 N= 1
X = -.180E+00	HTCMOD=	.579E+03	HTCEXP=	.448E+03	TWMOD=	618.54	TWEXP= 726.40 N= 1
X = -.897E-01	HTCMOD=	.365E+03	HTCEXP=	.448E+03	TWMOD=	533.54	TWEXP= 725.40 N= 2
X = -.788E-01	HTCMOD=	.325E+03	HTCEXP=	.448E+03	TWMOD=	504.54	TWEXP= 726.40 N= 3
X = -.585E-01	HTCMOD=	.679E+03	HTCEXP=	.448E+03	TWMOD=	568.54	TWEXP= 737.05 N= 5
X = -.530E-01	HTCMOD=	.677E+03	HTCEXP=	.816E+03	TWMOD=	568.54	TWEXP= 513.75 N= 6
X = -.492E-01	HTCMOD=	.673E+03	HTCEXP=	.977E+03	TWMOD=	568.54	TWEXP= 470.47 N= 7
X = -.454E-01	HTCMOD=	.670E+03	HTCEXP=	.100E+04	TWMOD=	568.54	TWEXP= 465.30 N= 8
X = -.416E-01	HTCMOD=	.666E+03	HTCEXP=	.102E+04	TWMOD=	570.54	TWEXP= 461.50 N= 9
X = -.378E-01	HTCMOD=	.662E+03	HTCEXP=	.955E+03	TWMOD=	572.54	TWEXP= 475.60 N= 10
X = -.340E-01	HTCMOD=	.657E+03	HTCEXP=	.699E+03	TWMOD=	574.54	TWEXP= 469.40 N= 11

X= .186E+00 HTCMOD= .413E+03 HTCEXP= .435E+03 TMMOD= 765.54 TWEXP= 692.65 N= 42
X= .194E+00 HTCMOD= .415E+03 HTCEXP= .479E+03 TMMOD= 765.54 TWEXP= 697.94 N= 43
X= .201E+00 HTCMOD= .418E+03 HTCEXP= .473E+03 TMMOD= 765.54 TWEXP= 703.24 N= 44

/// FUNCTION TABLE TMA REFERENCED OUT OF BOUNDS
THE NEAREST END VALUE OF Y WILL BE USED AND COMPUTATION CONTINUES
VALUE REQUESTED FOR X WAS 1.3025

/// FUNCTION TABLE TMA REFERENCED OUT OF BOUNDS
THE NEAREST END VALUE OF Y WILL BE USED AND COMPUTATION CONTINUES
VALUE REQUESTED FOR X WAS 1.3025

/// FUNCTION TABLE TMA REFERENCED OUT OF BOUNDS
THE NEAREST END VALUE OF Y WILL BE USED AND COMPUTATION CONTINUES
VALUE REQUESTED FOR X WAS 1.3025

/// FUNCTION TABLE TMA REFERENCED OUT OF BOUNDS
THE NEAREST END VALUE OF Y WILL BE USED AND COMPUTATION CONTINUES
VALUE REQUESTED FOR X WAS 1.3025

X= .209E+00 HTCMOD= .421E+03 HTCEXP= .462E+03 TMMOD= 758.54 TWEXP= 706.44 N= 45

QUALITY	N	HTCMOD	HTCEXP	TMMOD(C)	TWEXP(C)
-.053	1	676.8	815.8	568.5	513.7
-.049	2	673.3	977.3	568.5	470.5
-.045	3	669.6	1000.6	568.5	465.4
-.042	4	666.1	1017.7	570.5	461.6
-.038	5	661.5	954.9	572.5	475.6
-.034	6	657.1	899.4	574.5	489.4
-.030	7	652.2	850.8	576.5	503.2
-.026	8	647.2	806.5	579.5	516.7
-.023	9	642.0	775.7	582.5	527.3
-.019	10	636.3	747.1	585.5	537.8
-.011	11	623.1	696.5	591.5	558.6
-.004	12	608.8	668.1	599.5	571.6
.004	13	593.3	642.4	608.5	584.4
.012	14	577.8	630.7	613.5	590.6
.019	15	563.5	619.4	627.5	596.7
.027	16	552.1	608.5	635.5	602.9
.034	17	543.1	597.9	642.5	609.1
.042	18	534.2	587.2	648.5	615.3
.049	19	526.5	578.0	655.5	621.4
.057	20	517.4	569.8	662.5	626.8
.065	21	507.7	561.8	670.5	632.1
.072	22	497.5	554.0	678.5	637.4
.080	23	487.3	546.4	687.5	642.8
.087	24	477.2	539.1	696.5	648.1
.095	25	467.7	531.9	707.5	653.4
.103	26	458.4	526.0	716.5	657.9
.110	27	449.4	520.2	723.5	662.5
.118	28	442.2	514.5	736.5	667.8
.125	29	434.3	509.0	748.5	671.5
.133	30	427.9	503.5	747.5	676.1
.141	31	422.7	498.3	754.5	680.5
.148	32	418.3	496.0	758.5	682.5
.156	33	415.2	493.7	762.5	684.5
.163	34	413.1	491.4	764.5	686.5
.171	35	412.1	489.2	765.5	688.5
.179	36	412.0	487.0	765.5	690.5
.186	37	413.1	484.6	765.5	692.7
.194	38	415.1	478.8	765.5	698.0
.201	39	418.3	473.2	765.5	703.3
.209	40	420.6	467.8	758.5	708.5

MASS FLUX, HEAT FLUX, PRESSURE AND SUBCOOLING ARE PRINTED
M2= 369.6 HFT=21355.5 P1= 4072.4 ISUB= -214.1

/// THIS RUN TERMINATED BECAUSE THE CURRENT VALUE OF TIME IS 1.8500

/// TERMINAL COMMENTS ///

// NO CONVERGENCE PROBLEMS DETECTED DURING THIS RUN
// BOUNDS OF FUNCTION TABLE TMA WERE EXCEEDED 5 TIMES DURING THIS RUN

EXECUTION OPTIME FOR THIS RUN = 1146.926

FIELD LENGTH UTILISED 53824 (1511008)

.....

F O R S I M V I

• FORTRAN ORIENTED DISTRIBUTED SYSTEM SIMULATION PACKAGE •

• FOR PARTIAL AND OR ORDINARY DIFFERENTIAL EQUATIONS •

• USERS MANUAL - ATOMIC ENERGY OF CANADA REPORT AECL 5821 •

• AUTHORS M B CARVER AND D G STEWART, C.R.N.L./A.E.C.L. •

• OPTIMIZATION OPTION INSEPTED BY C F AUSTIN AND M B •

• CARVER, C.R.N.L./A.E.C.L. USER MANUAL AECL 56842 •

.....

JOB TERMINATED - NO MORE DATA
 EXECUTION CP TIME - 1146.9560

MFA NOS/BE 1.3 RELEASE 99 P.D 18.17

17.57.18. LAPFORN FROM NOS/HP NON-PRIME

17.57.18. GP 00003968 WORDS - FILE INPUT , DC 04

17.57.19. LAPFORI.80331-02236.T1488.1048.

17.57.19. ROUTE.OUTPUT.DC=PM.TID=05.0EF.

17.57.19. FTN(P=0)

17.57.32. 1.348 CP SECONDS COMPILATION TIME

17.57.32. ATTACH.FORSIM.

17.57.32. PFM IS

17.57.32. FORSIM

17.57.32. MF CYCLE NO. = 818

17.57.32. COPYN.0.EXEC.FORSIM.LGO.

17.57.35. RETURN.FORSIM.LGO.

17.57.35. ATTACH.FORSIM.CY=1.

17.57.35. PFM IS

17.57.35. FORSIM

17.57.35. ATTACH(LIBPOLSAW)

17.57.35. LFM IS

17.57.35. LIBPOL

17.57.35. MF CYCLE NO. = 881

17.57.35. LOSET(LIB=FORSIM/LIBPOL)

17.57.35. EXEC.

17.57.58. NON-FATAL LOADER ERRORS -

17.57.58. COMMON BLOCK REDEFINITION - FUNTBL

17.57.58. LAST PROGRAM READ - TABLE

17.57.58. LAST FILE ACCESSED- EXEC

17.57.53. RELEASE FORSIM

18.56.38. STOP

18.56.38. 151188 MAXIMUM EXECUTION FL.

18.56.38. 1147.172 CP SECONDS EXECUTION TIME.

18.56.38. GP 00007680 WORDS - FILE OUTPUT , DC 42

18.56.38. GP 00008256 WORDS - FILE PLOT , DC 38

18.56.38. MS 21584 WORDS (107520 MAX USED)

18.56.38. CPA 1150.415 SEC. 694.562 ADJ.

18.56.38. IO 27.711 SEC. 2.909 ADJ.

18.56.38. CM 155395.774 KWS. 48.949 ADJ.

18.56.38. SS 746.422

18.56.38. PP 44.258 SEC. DATE 83-06-14

18.56.38. EJ END OF JOB. MP

APPENDIX C RADIATION HEAT TRANSFER

The two surfaces 1 and 2 (shown in Figure C.1) are separated by water vapour at the temperature T_v , average of the wall temperature and saturation temperature.

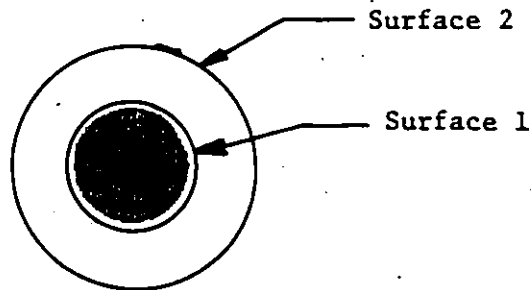


FIGURE C.1. Radiation Heat Transfer Surfaces

Assuming that the two surfaces are uniform temperature and also that their emissivity and absorptivity is constant over the surfaces, the net heat flux over a surface 1 is given by:

$$\frac{Q_1}{A_1} = J_1 - G_1 \quad \dots C.1$$

G_1 : incident radiation upon the surface 1.

J_1 : leaving radiation from the surface 1.

Q_1/A_1 : net heat flux over the surface 1.

The equation A.1 can also be written as:

$$\frac{Q_1}{A_1} = J_1 - \sum_j \tau_{g1j} F_{1j} J_j - \epsilon_g E_{bg} \quad \dots C.2$$

where

- j : surface number ($j = 1, 2$)
- τ_{g1j} : vapour transmissivity
- F_{1j} : view factor
- ϵ_g : vapour emissivity
- E_{bg} : vapour emissive power assumed as a black body over all wavelengths.

The second right hand term is the incident radiation emitted from the surrounding surfaces and reaching the surface 1 through the vapour of transmissivity τ_g . The third right-hand term is the incident radiation to the surface 1 from the vapour. Since two surfaces are exchanging heat by radiation, equation C.2 is expanded:

$$\frac{Q_1}{A_1} = J_1 - \tau_{g11} F_{11} J_1 - \tau_{g12} F_{12} J_2 - \epsilon_g E_{bg} \quad \dots C.3$$

$$\frac{Q_2}{A_2} = J_2 - \tau_{g21} F_{21} J_1 - \tau_{g22} F_{22} J_2 - \epsilon_g E_{bg} \quad \dots C.4$$

The equation C.1 can also be written as [Holman, 1976]:

$$\frac{Q_1}{A_1} = \frac{\epsilon_1}{(1 - \epsilon_1)} (E_{b1} - J_1) \quad \dots C.5$$

Expanding for the two surfaces:

$$\frac{Q_1}{A_1} = \frac{\epsilon_1}{(1 - \epsilon_1)} (E_{b1} - J_1) \quad \dots C.6$$

$$\frac{Q_2}{A_2} = \frac{\epsilon_2}{(1 - \epsilon_2)} (E_{b2} - J_2) \quad \dots C.7$$

The view factors are calculated using the enclosure's property and also the reciprocity relation:

$$F_{11} + F_{12} = 1 \quad \dots C.8$$

$$F_{21} + F_{22} = 1 \quad \dots C.9$$

$$A_1 F_{12} = A_2 F_{21} \quad \dots C.10$$

Since surface 1 can not see itself, the view factor F_{11} is zero and F_{12} is 1.

$$F_{21} = \frac{A_1}{A_2} \quad \dots C.11$$

$$F_{22} = 1 - \frac{A_1}{A_2} \quad \dots C.12$$

Using equations C.3, C.4, C.5 and C.6, a system of two unknowns, two equations are obtained:

$$[A] [J] = [B]$$

$$A = \begin{bmatrix} F_{11} \tau_{g_{11}} - \frac{\epsilon_1}{(1 - \epsilon_1)} - 1 & F_{12} \tau_{g_{12}} \\ F_{21} \tau_{g_{21}} & F_{22} \tau_{g_{22}} - \frac{\epsilon_2}{(1 - \epsilon_2)} - 1 \end{bmatrix}$$

$$B = \begin{bmatrix} -\epsilon_g E_{bg} - \frac{\epsilon_1}{(1 - \epsilon_1)} E_{b_1} \\ -\epsilon_g E_{bg} - \frac{\epsilon_2}{(1 - \epsilon_2)} E_{b_2} \end{bmatrix}$$

$$J = \begin{bmatrix} J_1 \\ J_2 \end{bmatrix}$$

A subroutine has been written in the model's computer program to solve this system of equations and is shown in appendix B (Subroutine RADIAT). Comparison between the rigorous calculation of radiative heat transfer and the simplified method assuming two gray infinite planes was done and is shown in Figure C.2. The radiation heat flux is constant using the simplified method. However, the radiation heat flux is lower following an increase of the vapour film thickness for a constant wall temperature using the rigorous method. This might be explained by a part of the radiation leaving the wall which is incident to the same wall.

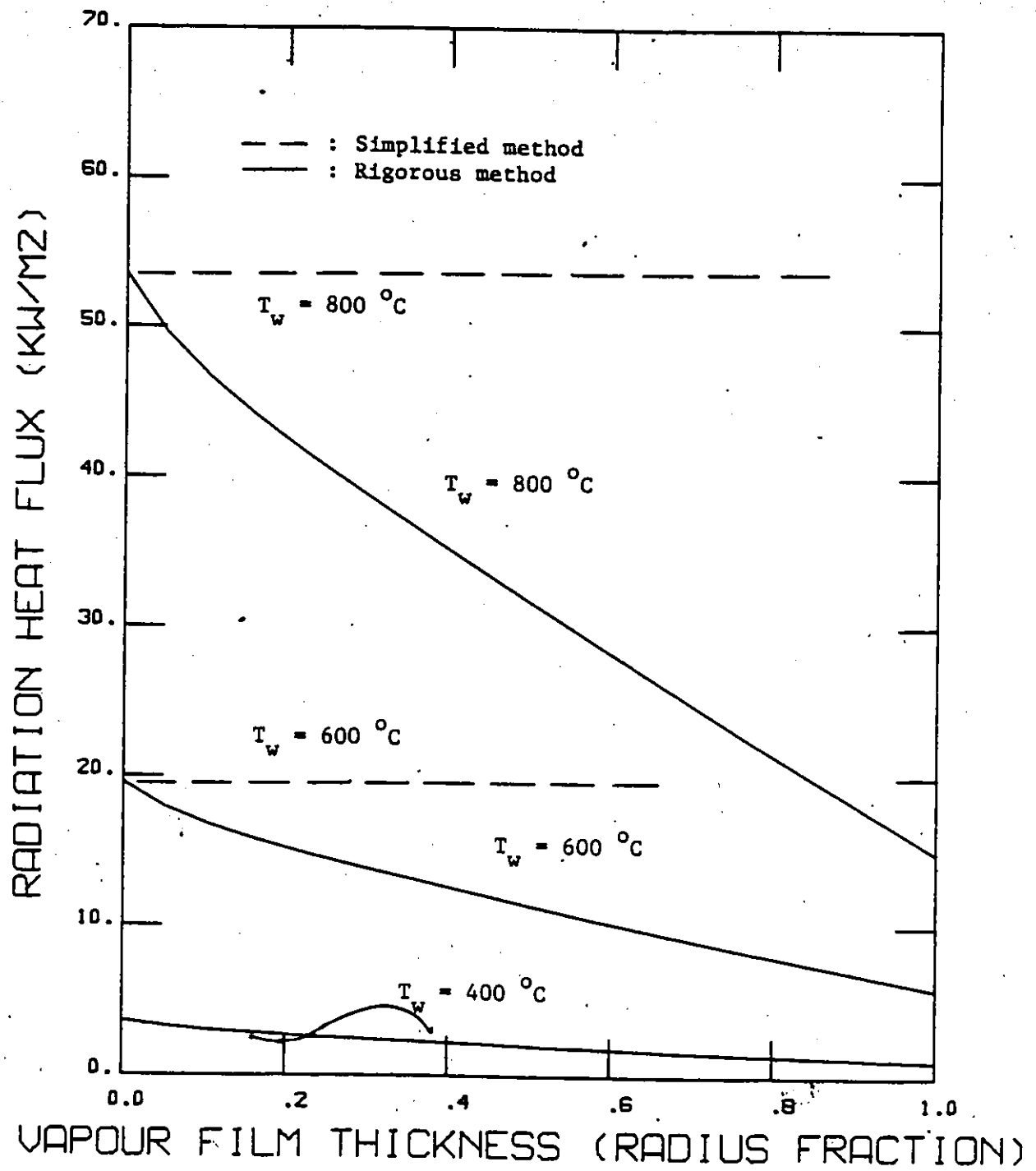


FIGURE C.2. Radiation heat flux at the wall for several vapour film thickness ($P = 9.6 \text{ MPa}$)

APPENDIX D

DERIVATION OF THE DIFFERENTIAL EQUATIONS

1) Derivation of dU_v/dz

Using the liquid core momentum equation and the vapour film momentum equation:

$$\begin{aligned} \rho_l g + \rho_l U_l \frac{dU_l}{dz} - \frac{f_l \rho_v P_l (\bar{U}_v - U_l)^2}{2A_l} \\ = \rho_v \bar{U}_v \frac{d\bar{U}_v}{dz} + \rho_v g + \frac{2P_v \mu_v \bar{U}_v 3n}{\delta A_v (3n-1)} + \frac{\Gamma_v}{A_v} (\bar{U}_v - U_l) + \frac{f_l \rho_v P_v (\bar{U}_v - U_l)^2}{2A_v} \quad \dots D.1 \end{aligned}$$

$$\begin{aligned} \frac{d\bar{U}_v}{dz} = \left[\rho_l \left(g + U_l \frac{dU_l}{dz} \right) - \rho_v g - \frac{6P_v \mu_v \bar{U}_v n}{\delta A_v (3n-1)} - \frac{\Gamma_v}{A_v} (\bar{U}_v - U_l) \right. \\ \left. - \frac{f_l \rho_v P_l (\bar{U}_v - U_l)^2}{2A_l} - \frac{f_l \rho_v P_v (\bar{U}_v - U_l)^2}{2A_v} \right] / (\rho_v \bar{U}_v) \quad \dots D.2 \end{aligned}$$

2) Derivation of $\frac{dU_l}{dz}$

Using the mass conservation equation of the liquid:

$$-\Gamma_v = \frac{d}{dz} (U_l A_l \rho_l) \quad \dots D.3$$

$$-\Gamma_v = \rho_l \left[A_l \frac{dU_l}{dz} + U_l \frac{dA_l}{dz} \right] \quad \dots D.4$$

$$A_l \rho_l \frac{dU_l}{dz} = -\Gamma_v - U_l \rho_l \frac{dA_l}{dz} \quad \dots D.5$$

From

$$A_1 = \pi(r - \delta)^2 \quad \dots D.6$$

$$\frac{dA_1}{dz} = 2\pi(\delta - r) \frac{d\delta}{dz} \quad \dots D.7$$

$$\frac{dU_1}{dz} = \frac{[-\Gamma_v - U_1 \rho_1 2\pi(\delta - r) \frac{d\delta}{dz}]}{[A_1 \rho_1]} \quad \dots D.8$$

3) Derivation of $\frac{d\delta}{dz}$

Using the equation of mass conservation for the vapour film:

$$\frac{d\delta}{dz} = (\Gamma_v / \rho_v - A_v \frac{d\bar{U}_v}{dz}) / [2\pi(r - \delta)\bar{U}_v] \quad \dots D.9$$

$$\begin{aligned} \frac{d\delta}{dz} = & \left[\Gamma_v / \rho_v - \frac{A_v}{(\rho_v \bar{U}_v)} \left[g(\rho_1 - \rho_v) + \rho_1 U_1 \frac{dU_1}{dz} - 6 \frac{P_v \mu_v \bar{U}_v n}{\delta A_v (3n-1)} \right. \right. \\ & \left. \left. - \frac{\Gamma_v}{A_v} (\bar{U}_v - U_1) - \frac{f_1 \rho_v P_1 (\bar{U}_v - U_1)^2}{2A_1} - \frac{f_1 \rho_v P_v (\bar{U}_v - U_1)^2}{2A_v} \right] \right] / (2\pi \bar{U}_v (r - \delta)) \quad \dots D.10 \end{aligned}$$

$$\begin{aligned} \frac{d\delta}{dz} = & \left[\Gamma_v / \rho_v - \frac{A_v}{(\rho_v \bar{U}_v)} \left[g(\rho_1 - \rho_v) + \frac{\rho_1 U_1}{A_1 \rho_1} [-\Gamma_v - U_1 \rho_1 2\pi(\delta - r) \frac{d\delta}{dz}] \right. \right. \\ & \left. \left. - \frac{6 P_v \mu_v \bar{U}_v n}{\delta A_v (3n-1)} - \frac{\Gamma_v}{A_v} (\bar{U}_v - U_1) - \frac{f_1 \rho_v P_1 (\bar{U}_v - U_1)^2}{2A_1} - \frac{f_1 \rho_v P_v (\bar{U}_v - U_1)^2}{2A_v} \right] \right] / (2\pi \bar{U}_v (r - \delta)) \quad \dots D.11 \end{aligned}$$

$$\begin{aligned}
 2\pi \bar{U}_v(r-\delta) \frac{d\delta}{dz} - \frac{A_v U_1^2 \rho_1 2\pi(\delta-r)}{\rho_v \bar{U}_v A_1} \frac{d\delta}{dz} &= \frac{\Gamma_v}{\rho_v} - \frac{A_v g}{\rho_v \bar{U}_v} (\rho_1 - \rho_v) \\
 + \frac{\Gamma_v U_1 A_v}{\rho_v \bar{U}_v A_1} + \frac{6 P_v \mu_v n}{\delta \rho_v (3n-1)} + \frac{\Gamma_v}{\rho_v} \left[1 - \frac{U_1}{\bar{U}_v}\right] \\
 + \frac{f_1 P_1 (\bar{U}_v - U_1)^2 A_v}{2 A_1 \bar{U}_v} + \frac{f_1 P_v (\bar{U}_v - U_1)^2}{2 \bar{U}_v}
 \end{aligned} \tag{...D.12}$$

To simplify

$$Y1 = 2\pi \bar{U}_v(r-\delta) + \frac{A_v U_1^2 \rho_1 2\pi(r-\delta)}{\rho_v \bar{U}_v A_1} \tag{...D.13}$$

$$\begin{aligned}
 Y2 = \frac{\Gamma_v}{\rho_v} \left[1 + \frac{A_v U_1}{A_1 \bar{U}_v}\right] - \frac{A_v g}{\rho_v \bar{U}_v} (\rho_1 - \rho_v) + \frac{6 P_v \mu_v n}{\delta \rho_v (3n-1)} + \frac{\Gamma_v}{\rho_v} \left[1 - \frac{U_1}{\bar{U}_v}\right] \\
 + \frac{f_1 P_1 (\bar{U}_v - U_1)^2 A_v}{2 A_1 \bar{U}_v} + \frac{f_1 P_v (\bar{U}_v - U_1)^2}{2 \bar{U}_v}
 \end{aligned} \tag{...D.14}$$

$$\frac{d\delta}{dz} = Y2/Y1 \tag{...D.15}$$

APPENDIX E

HEAT BALANCE

E.1 Test Section Heat Balance

Zero power and power heat balance runs were obtained to verify power measurements, flow measurement and temperature measurement. Results of the test section zero power heat loss are shown in figure E.1. These data were obtained during the warm up of the loop. Heat losses are given as a function of the average wall temperature minus the ambient temperature. An eye fit line was drawn through the dots and the following equation was derived:

$$\text{POWLOSST} = 3.063 \times 10^{-3} (T_{wa} - T_a)$$

POWLOSST: Zero power heat loss (kW)

T_{wa} : Average wall temperature ($^{\circ}\text{C}$)

T_a : Ambient temperature ($^{\circ}\text{C}$)

The lowest flow of the run matrix ($G = 1000 \text{ Kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$) was used to evaluate the heat loss. The reading of the inlet thermocouple and the reading of the outlet temperature was also checked at ambient temperature. Differences between the two values were found to be negligible.

Test section heat balance runs using the power loss equation previously mentioned are shown in figure E.2. Results show reasonable agreement even if the flow was measured with an orifice plate which is less accurate than a turbine flowmeter. The heat balance runs obtained are compatible with some previous heat balance runs of the Advance Engineering Branch.

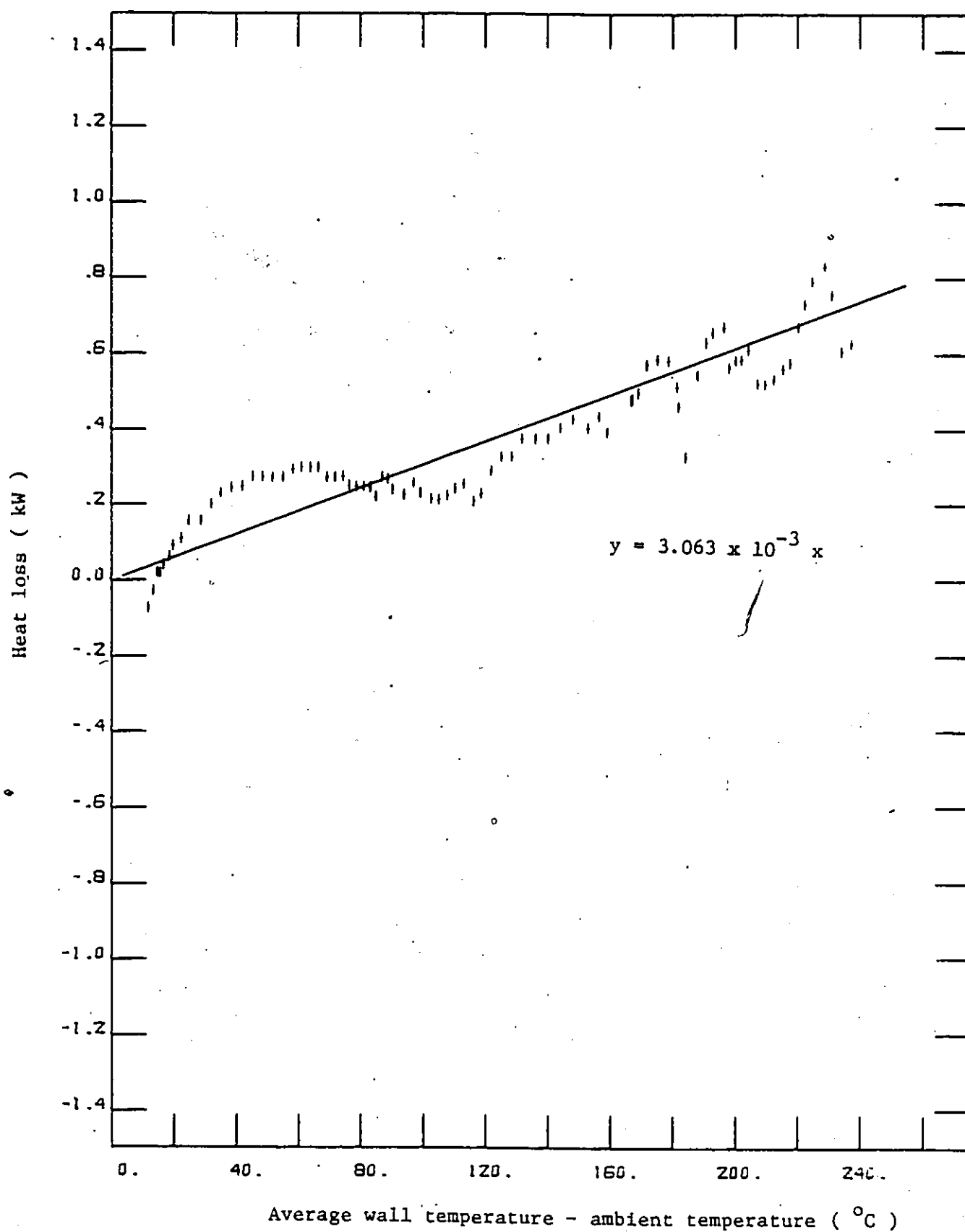


FIGURE E.1. Test section zero power heat loss

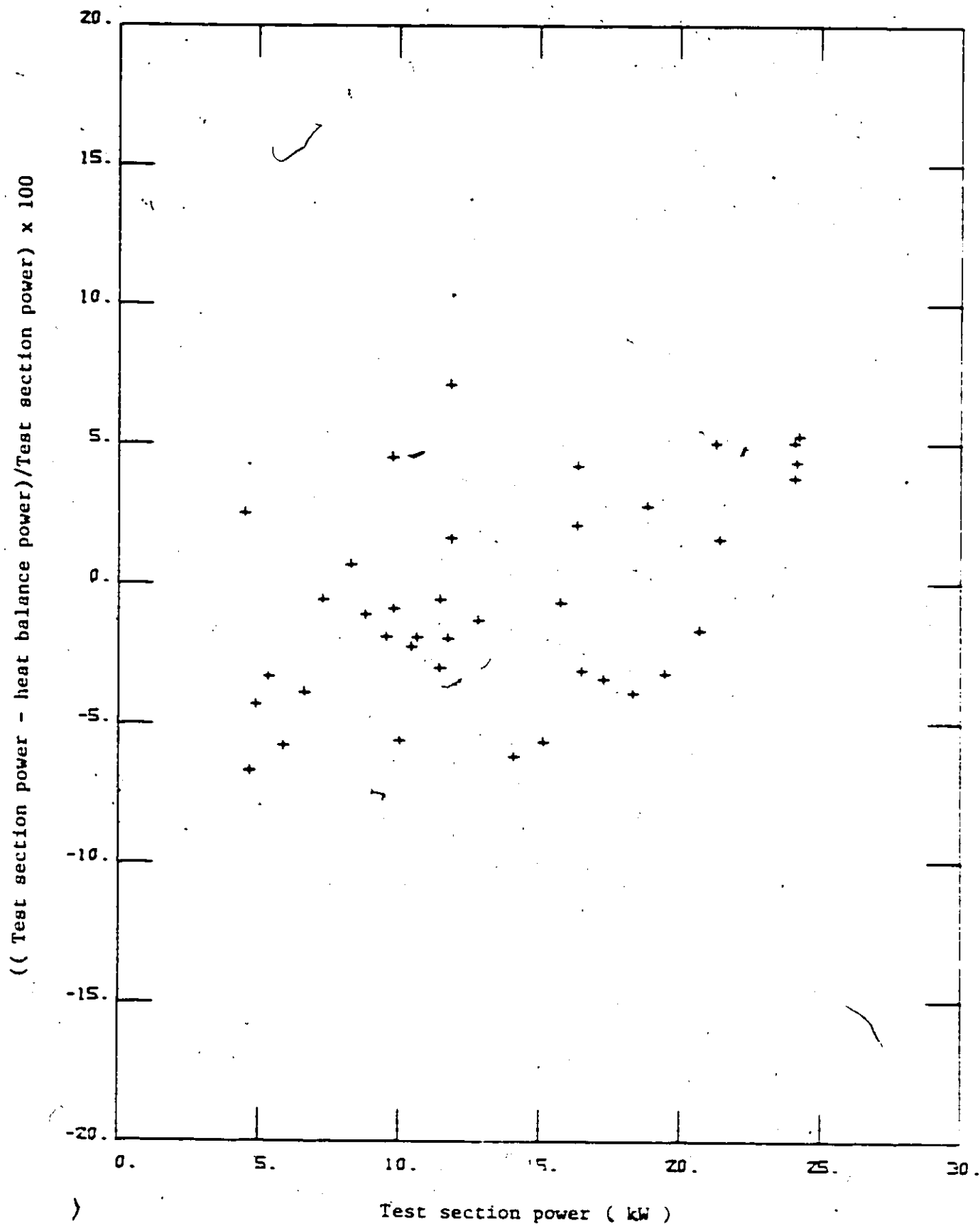


FIGURE E.2. Test section heat balance runs with heat loss correction

The heat loss was assumed to occur only at the power clamps. This means that half of the heat loss occurs at the lower clamps and half at the higher clamps.

E.2. Preheater heat balance

Power heat balance runs were also obtained for the preheater (350 kW power supply). It was found necessary to correct significant discrepancies in the electrical power measurements. The change of enthalpy of the water during the single-phase liquid heat balance were less than the electrical energy supplied as shown in Figure E.3. The following equation derived from the heat balance data was used to correct the measured power:

$$PA = -2.7650 + 1.0364 PM - .001203 PM^2$$

PA: accurate preheater power (kW)

PM: measured preheater power (kW)

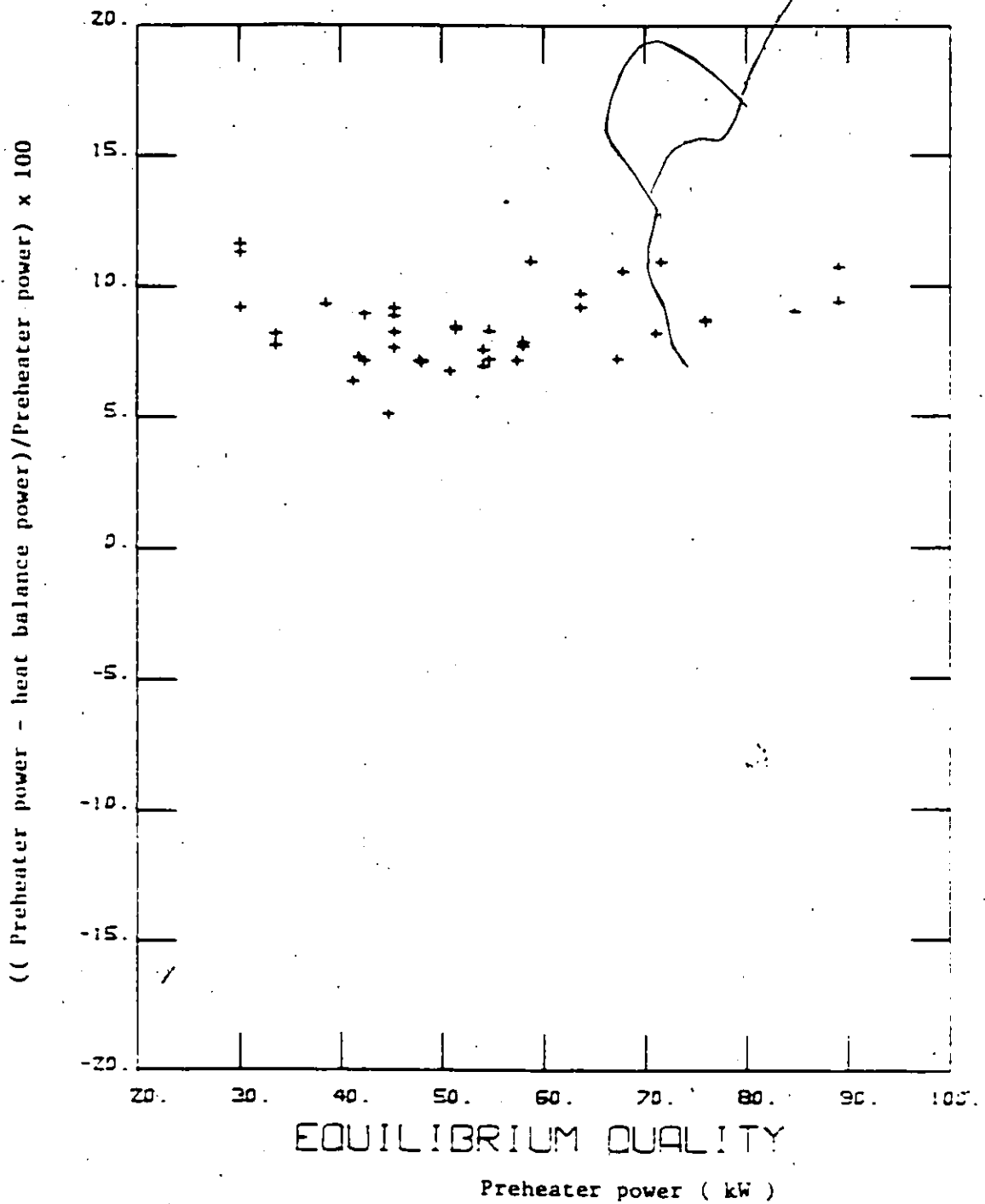


FIGURE E.3. Preheater heat balance runs

Several subroutines used in the data processing are listed.

```

SUBROUTINE TMCOR(TM)
      THIS SUBROUTINE IS USED TO EVALUATE THE INSIDE WALL
      TEMPERATURE FROM THE OUTSIDE WALL TEMPERATURE
      THE OUTSIDE WALL TEMPERATURE
      THE INSIDE WALL TEMPERATURE
      COMMON/TEMP/HFT,C,XIN,SUB(18),HFWP,MSUB,XOUT,X1,TINC,X2
      COMMON/CONST/PI(4)
      COMMON/OUT/RAD,RADO
      REAL TM(2)
      DO 10 I = 1,2
      CALL COMQUAT77(M(I))
      IF(I.EQ.1) GO TO 20
      IF(I.EQ.2) GO TO 20
      CORRECTION FOR HOT PATCH
      TM(I) = TM(I) - (HFWP * RAD * ALOG(RADO /RAD)/RXTS)
      GO TO 30
20  CONTINUE
      CORRECTION FOR TEST SECTION
      C1 = (RADO**2./(RAD**2.-RAD**2.)*ALOG(RADO/RAD) - 8.5)
      TM(I) = TM(I) - HFT * RAD * C1/RXTS
30  CONTINUE
40  CONTINUE
      RETURN
      END

```



```

80      REAL K,R(13),TMC
      COMMON/L128/K(128)
      COMMON/PROP/VT,VC,V,NHFC,NF,NG,NL,UF,UG,UL,
      : AKF,AKG,AKL,CPL,CPC,CPL,SP,SL,
85      : RHOF,RHOG,RHOL,PRF,PRG,PAL,PSAT,TSAT,
      : VFC,UFC
      P = PRESS/6.894757
      T = TEMP/1.8 + 32.
90
      *****
      IF THE PRESSURE (PRESS) PASSED INTO PROPER
      * EQUALS 0.0 THEN SATURATION PROPERTIES AS A
      * FUNCTION OF TEMPERATURE HAVE BEEN REQUESTED.
      * IF THE PRESSURE IS NOT EQUAL TO 0.0 THEN
95      * SATURATION PROPERTIES AS A FUNCTION OF
      * PRESSURE ARE CALCULATED.
      *****
100     IF(PRESS .EQ. 0.0) CALL POLSAN(0.0,T,K,R,1)
      IF(PRESS .NE. 0.0) CALL POLSAN(P,0.0,K,R,1)
      VT = R(1) = 0.002428
      VC = R(2) = 0.002428
105     NFG = R(3) = 2.3855
      NF = R(4) = 0.002428
      NG = R(5) = 0.002428
      NL = R(6) = 0.002428
      UF = R(7) = 0.002428
110     UG = R(8) = 0.002428
      UL = R(9) = 0.002428
      AKF = R(10) = 0.00173735
      AKG = R(11) = 0.00173735
      AKL = R(12) = 0.00173735
115     CPL = R(13) = 0.00173735
      CPC = R(14) = 0.00173735
      CPL = R(15) = 0.00173735
      SP = R(16) = 0.00173735
      SL = R(17) = 0.00173735
      RHOF = 1.0/R(18) = 15.911
      RHOG = 1.0/R(19) = 12.911
120     PRF = R(20) = R(3) * 3682./R(4)
      PRG = R(21) = R(3) * 3682./R(4)
      PAL = R(22) = R(3) * 3682./R(4)
      IF(PRESS .EQ. 0.0) PSAT = R(13) = 6.894757
      IF(PRESS .NE. 0.0) TSAT = (R(13) - 32.) / 1.8
125
      *****
      IF THE TEMPERATURE (TEMP) OR THE PRESSURE (PRESS)
      * PASSED INTO PROPER EQUALS 0.0 THEN ONLY THE
      * SATURATED VALUES HAVE BEEN REQUESTED.
      *****
130
      IF(TEMP .EQ. 0.0 .OR. PRESS .EQ. 0.0) RETURN
135
      *****
      CALCULATE THE SUBCOOLED WATER OR SUPERHEATED STEAM
      * PROPERTIES
      *****
      CALL POLSAN(P,T,K,R,1)
      VT = R(1) = 0.002428
      VC = R(2) = 0.002428
140     NFG = R(3) = 2.3855
      NF = R(4) = 0.002428
      NG = R(5) = 0.002428
      NL = R(6) = 0.002428
      UF = R(7) = 0.002428
145     UG = R(8) = 0.002428
      UL = R(9) = 0.002428
      AKF = R(10) = 0.00173735
      AKG = R(11) = 0.00173735
      AKL = R(12) = 0.00173735
      CPL = R(13) = 0.00173735
      CPC = R(14) = 0.00173735
      CPL = R(15) = 0.00173735
      SP = R(16) = 0.00173735
      SL = R(17) = 0.00173735
      RHOF = 1.0/R(18) = 15.911
      RHOG = 1.0/R(19) = 12.911
      PRF = R(20) = R(3) * 3682./R(4)
      PRG = R(21) = R(3) * 3682./R(4)
      PAL = R(22) = R(3) * 3682./R(4)
      RETURN
      END

```

```

1      SUBROUTINE TESTPA(FLOW,ITN,POUT,TMA,TA,NL,IOATE)
      *****
      THIS SUBROUTINE IS USED TO CALCULATE THE LOOP
      PARAMETERS
      *****
      MFT      TEST SECTION HEAT FLUX      M/HR**2
      G        MASS FLUX                    KG/HR**2 SEC)
      MPL      HOT PATCH LENGTH             M
      HTL      HEATED LENGTH                M
20     HTLANP   HEATED LENGTH AFTER H-PATCH M
      HTLBNP   HEATED LENGTH BEF. HOT PATCH M
      XIN      INLET QUALITY
      XOUT     OUTLET QUALITY
15     QUA      QUALITY
      X1       INLET QUALITY HOT PATCH
      X2       OUTLET QUALITY HOT PATCH
      TSPON    TEST SECTION POWER          KW
      PRPON    PREHEATER POWER              KW
20     PRPONR   CORRECTED REAL PREH. POWER  KW
      M1       INLET MASS FLOW INLET FLAC
      TMA      AVERAGE MET TEMPERATURE
      TA       AMBIENT TEMPERATURE
      ITN      INLET TEMPERATURE
25     TINC     CORRECTED INLET TEMP.(HEAT LOSS)
      POUT     OUTLET PRESSURE              KPA
      *****
      COMMON/TESTP/HFT,G,XIN,QUA(10),MFT,MPL,HTL,HTLANP,HTLBNP,
      : XIN,XOUT,X1,X2,TMA,TA,ITN,TINC,X2
      COMMON/HEAT/PRPON
      COMMON/POWER/TSPON,PRPON,PRPONR
      COMMON/PROP/VT,VC,V,NHFC,NF,NG,NL,UF,UG,UL,AKF,AKG,AKL,
      : RHOF,RHOG,RHOL,PRF,PRG,PAL,PSAT,TSAT,VFC,UFC
30     COMMON/DIM/RAD,RAD
      DIMENSION Z(10),Z1(10),Z2(10)
      DATA Z1/0.02,0.05,0.1,0.15,0.2,0.25,0.3,0.35,0.4,0.45,0.5/
      DATA Z2/0.02,0.05,0.1,0.15,0.2,0.25,0.3,0.35,0.4,0.45,0.5/
      DATA Z/0.02,0.05,0.1,0.15,0.2,0.25,0.3,0.35,0.4,0.45,0.5/
      SELECTION OF TEST SECTION DEPENDING ON THE MONTH

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IDATES = IDATE/100 - IDATE/10000 * 100
DO 100 J = 1,10
  Z(J) = 0.0
150 CONTINUE
IF (IDATES.EQ.12.OR.IDATES.EQ.4) GO TO 12
IF (IDATES.EQ.8) GO TO 14
C TEST SECTION FOR OCTOBER AND NOVEMBER
DO 27 I = 1,10
  Z(I) = Z(I)
27 CONTINUE
NPL = 1.095
NPL = 0.101
NPLAMP = 1.772
NPLBWP = 0.123
50 IF (IDATES.NE.11.AND.IDATES.NE.10) PRINT*,#CHECK NOV-DEC IN TEST#
GO TO 10
C 17 MONTH OF MAY
DO 57 I = 1,10
  Z(I) = Z(I)
57 CONTINUE
NPL = 1.093
NPL = 0.101
NPLAMP = 1.772
NPLBWP = 0.123
60 IF (IDATES.NE.5) PRINT*,#CHECK THE MONTH OF MAY FOR T/C LOC.#
GO TO 10
C 12 MONTH OF DECEMBER AND APRIL
DO 70 I = 1,10
  Z(I) = Z(I)
70 CONTINUE
NPL = 1.093
NPL = 0.101
NPLAMP = 1.77
NPLBWP = 0.123
75 IF (IDATES.NE.12.AND.IDATES.NE.4) PRINT*,#CHECK DEC-IN TEST#
GO TO 10
14 CONTINUE
80 CORRECTION OF WALL DIMENSION
CALL DIMENS(TMA)
FOR THE MONTH OF APRIL, TWO HOT PATCH HEATERS WERE CONNECTED
FULL POWER AND WERE NOT MEASURED BY THE MATTMETER BUT BY
AN AMPMETER. THE POWER IS ADDED TO THE BLOCK(312) AND BLOCK(313)
FOR THE MONTH OF MAY, FOUR HOT PATCH HEATERS WERE CONNECTED
FULL POWER AND BYPASS THE ON LINE COMPUTER MATTMETER.
IF (IDATES.EQ.5) POMP = POMP + 2.0
CALCULATION OF TEST SECTION HEAT FLUX
MFT = TSPON * 1000.0/(NPL*3.141593*2.0 * RAD)
95 CALCULATION OF HOT PATCH HEAT FLUX
MHP = POMP * 1000.0/(NPL * 3.141593 * 2.0 * RAD)
CALCULATION OF FLOW(KG/S)
M1 = FLOW/3600.0
CALCULATION OF MASS FLUX
G = M1 / (3.141593 * RAD ** 2.0)
IF (M1.EQ.0) CONTINUE
IF (M1.EQ.0) GO TO 170
110 CALCULATION OF INLET QUALITY(SINGLE PHASE INLET)
CALL PROPER(POUT,TIN)
MIN = M1
MUS = (MIN - MFT)/MFC
XIN = (MIN - MFT)/MFC
115 IF (XIN.LT.0.0) PRINT*,#CHECK XIN IN SUB TESTPA#
MODIFICATION OF XIN DUE TO HEAT LOSS(SINGLE PHASE INLET)
N = 1
TIN1 = TIN
CALL HTLOS(HTLOSS,TIN,TA,POUT,N,XIN)
XIN = XIN - HTLOSS/(2.0 * MFC * M10)
CALL TEMP(POUT,XIN,TINC)
GO TO 100
120 CONTINUE
170 CALCULATION OF INLET QUALITY(TWO PHASE INLET)
CALL PROPER(POUT,TIN)
MIN = M1
XINPH = (MIN - MFT)/MFC
XIN = XINPH * PRPON / (M1 * MFC)
N = 2
CALL HTLOS(HTLOSS,TIN,TA,POUT,N,XIN)
XIN = XINPH * PRPON / (M1 * MFC)
130 IF (XIN.LT.0.0) PRINT*,#CHECK XIN IN SUB TESTPA# . XIN
MODIFICATION OF XIN DUE TO HEAT LOSS
CORRECTIONS MUST BE MADE FOR:
21 TEST SECTION
TIN1 = TSAT
N = 1
CALL HTLOS(HTLOSS,TIN,TA,POUT,N,XIN)
HTLOSS1 = HTLOSS
140 XIN = XIN - HTLOSS1/(2.00*MFC*M1)
IF (XIN.LT.0.0) CALL TEMP(POUT,XIN,TINC)
IF (XIN.LT.0.0) TINC = TSAT
150 CONTINUE
180 X1 = XIN*MFT*3.141593*2.0*RAD*HTLBNP/(M1*MFC*1000.0)
X2 = X1 + POMP * 1000.0/(M1 * MFC * 1000.0)
C CALCULATION OF QUALITY

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185

C

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DO 168 J = 1,18
  P1 = HFT1 * 3.141593 * 2. = RAD/(W2 * WFG * 1800.0)
  QUA(J) = X2 * P1 * Z(J)

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186

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186 CONTINUE
  XOUT = X2 * P1 * HTLAMP
  RETURN
END

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1

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SUBROUTINE PLOTZ (IRUNI, G1, HFT1, P, TWE1, W7, QUA)
  ***** THIS SUBROUTINE IS USED TO PLOT INSIDE HEAT TRANSFER
  COEFFICIENT VERSUS EQUILIBRIUM THERMODYNAMIC QUALITY
  COMPARISON WITH COUGALL-KONSEMAN, GROENEVELD-DELOME, BERENSON
  CAN ALSO BE PLOTTED

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  FCOMP = 1.....COMPARISON PLOTTED
  FCOMP = 2.....COMPARISON NOT PLOTTED
  COMMON/FORMAT/IL, VL, P, TITLE(8), PSUBTIT(8), FXACT1(8), FXACT2(8),
  SMSTVAR5, STVAR5(18)
  COMMON/PROP/VP, V, VL, WFG, W, W2, W3, W4, W5, W6, W7, W8, W9, W10, W11, W12, W13, W14, W15, W16, W17, W18, W19, W20, W21, W22, W23, W24, W25, W26, W27, W28, W29, W30, W31, W32, W33, W34, W35, W36, W37, W38, W39, W40, W41, W42, W43, W44, W45, W46, W47, W48, W49, W50, W51, W52, W53, W54, W55, W56, W57, W58, W59, W60, W61, W62, W63, W64, W65, W66, W67, W68, W69, W70, W71, W72, W73, W74, W75, W76, W77, W78, W79, W80, W81, W82, W83, W84, W85, W86, W87, W88, W89, W90, W91, W92, W93, W94, W95, W96, W97, W98, W99, W100, W101, W102, W103, W104, W105, W106, W107, W108, W109, W110, W111, W112, W113, W114, W115, W116, W117, W118, W119, W120, W121, W122, W123, W124, W125, W126, W127, W128, W129, W130, W131, W132, W133, W134, W135, W136, W137, W138, W139, W140, W141, W142, W143, W144, W145, W146, W147, W148, W149, W150, W151, W152, W153, W154, W155, W156, W157, W158, W159, W160, W161, W162, W163, W164, W165, W166, W167, W168, W169, W170, W171, W172, W173, W174, W175, W176, W177, W178, W179, W180, W181, W182, W183, W184, W185, W186, W187, W188, W189, W190, W191, W192, W193, W194, W195, W196, W197, W198, W199, W200, W201, W202, W203, W204, W205, W206, W207, W208, W209, W210, W211, W212, W213, W214, W215, W216, W217, W218, W219, W220, W221, W222, W223, W224, W225, W226, W227, W228, W229, W230, W231, W232, W233, W234, W235, W236, W237, W238, W239, W240, W241, W242, W243, W244, W245, W246, W247, W248, W249, W250, W251, W252, W253, W254, W255, W256, W257, W258, W259, W260, W261, W262, W263, W264, W265, W266, W267, W268, W269, W270, W271, W272, W273, W274, W275, W276, W277, W278, W279, W280, W281, W282, W283, W284, W285, W286, W287, W288, W289, W290, W291, W292, W293, W294, W295, W296, W297, W298, W299, W300, W301, W302, W303, W304, W305, W306, W307, W308, W309, W310, W311, W312, W313, W314, W315, W316, W317, W318, W319, W320, W321, W322, W323, W324, W325, W326, W327, W328, W329, W330, W331, W332, W333, W334, W335, W336, W337, W338, W339, W340, W341, W342, W343, W344, W345, W346, W347, W348, W349, W350, W351, W352, W353, W354, W355, W356, W357, W358, W359, W360, W361, W362, W363, W364, W365, W366, W367, W368, W369, W370, W371, W372, W373, W374, W375, W376, W377, W378, W379, W380, W381, W382, W383, W384, W385, W386, W387, W388, W389, W390, W391, W392, W393, W394, W395, W396, W397, W398, W399, W400, W401, W402, W403, W404, W405, W406, W407, W408, W409, W410, W411, W412, W413, W414, W415, W416, W417, W418, W419, W420, W421, W422, W423, W424, W425, W426, W427, W428, W429, W430, W431, W432, W433, W434, W435, W436, W437, W438, W439, W440, W441, W442, W443, W444, W445, W446, W447, W448, W449, W450, W451, W452, W453, W454, W455, W456, W457, W458, W459, W460, W461, W462, W463, W464, W465, W466, W467, W468, W469, W470, W471, W472, W473, W474, W475, W476, W477, W478, W479, W480, W481, W482, W483, W484, W485, W486, W487, W488, W489, W490, W491, W492, W493, W494, W495, W496, W497, W498, W499, W500, W501, W502, W503, W504, W505, W506, W507, W508, W509, W510, W511, W512, 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W1725, W1726, W1727, W1728, W1729, W1730, W1731, W1732, W1733, W1734, W1735, W1736, W1737, W1738, W1739, W1740, W1741, W1742, W1743, W1744, W1745, W1746, W1747, W1748, W1749, W1750, W1751, W1752, W1753, W1754, W1755, W1756, W1757, W1758, W1759, W1760, W1761, W1762, W1763, W1764, W1765, W1766, W1767, W1768, W1769, W1770, W1771, W1772, W1773, W1774, W1775, W1776, W1777, W1778, W1779, W1780, W1781, W1782, W1783, W1784, W1785, W1786, W1787, W1788, W1789, W1790, W1791, W1792, W1793, W1794, W1795, W1796, W1797, W1798, W1799, W1800, W1801, W1802, W1803, W1804, W1805, W1806, W1807, W1808, W1809, W1810, W1811, W1812, W1813, W1814, W1815, W1816, W1817, W1818, W1819, W1820, W1821, W1822, W1823, W1824, W1825, W1826, W1827, W1828, W1829, W1830, W1831, W1832, W1833, W1834, W1835, W1836, W1837, W1838, W1839, W1840, W1841, W1842, W1843, W1844, W1845, W1846, W1847, W1848, W1849, W1850, W1851, W1852, W1853, W1854, W1855, W1856, W1857, W1858, W1859, W1860, W1861, W1862, W1863, W1864, W1865, W1866, W1867, 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W2011, W2012, W2013, W2014, W2015, W2016, W2017, W2018, W2019, W2020, W2021, W2022, W2023, W2024, W2025, W2026, W2027, W2028, W2029, W2030, W2031, W2032, W2033, W2034, W2035, W2036, W2037, W2038, W2039, W2040, W2041, W2042, W2043, W2044, W2045, W2046, W2047, W2048, W2049, W2050, W2051, W2052, W2053, W2054, W2055, W2056, W2057, W2058, W2059, W2060, W2061, W2062, W2063, W2064, W2065, W2066, W2067, W2068, W2069, W2070, W2071, W2072, W2073, W2074, W2075, W2076, W2077, W2078, W2079, W2080, W2081, W2082, W2083, W2084, W2085, W2086, W2087, W2088, W2089, W2090, W2091, W2092, W2093, W2094, W2095, W2096, W2097, W2098, W2099, W2100, W2101, W2102, W2103, W2104, W2105, W2106, W2107, W2108, W2109, W2110, W2111, W2112, W2113, W2114, W2115, W2116, W2117, W2118, W2119, W2120, W2121, W2122, W2123, W2124, W2125, W2126, W2127, W2128, W2129, W2130, W2131, W2132, W2133, W2134, W2135, W2136, W2137, W2138, W2139, W2140, W2141, W2142, W2143, W2144, W2145, W2146, W2147, W2148, W2149, W2150, W215
```

```

C THE LOWER VALUE AND THE HIGHER VALUE OF EXPERIMENTAL QUALITY ARE USED
C FOR THE BOUNDARY FOR THE COMPARISON
JOBOPT = 3
X1 = QUA(1,I)
95 X2 = QUA(18,I)
DIF = (X2 - X1)/23.6
N = 8
DO 58 J = 1,23
N = N + 1
120 IF(N.EQ.1) X(N) = X1
IF(N.GT.1) X(N) = X(N-1) + DIF
IF(X1.LT.0) X(2) = 0.0000000000000001
IF(X(N).LT.0) X(N) = 0.0000000000000002
XSM(N) = X(N)
105 CALL GROENV0(G1,X(N),P,TM,MFT1(I))
TMC(N) = TM
CALL HERENSW(G1,X(N),P,TM,MFT1(I))
TMS(N) = TM
110 CALL DOUGALL(G1,X(N),P,TM,MFT1(I))
TMD(N) = TM
58 CONTINUE
WRITE(2,18)
18 FORMAT(//,28X,2MTBZ,18X,2HTGZ,18X,2HTDZ,18X,2 X Z,
3 8X,2TMZ,6X,2TMSZ,6X,2TMSZ)
C HEAT TRANSFER IS CALCULATED FROM WALL TEMPERATURE
DO 78 J3 = 1,23
MTB(J3) = MFT1(I)/(TMS(J3) - TSAT)
120 HTG(J3) = MFT1(I)/(TMC(J3) - TSAT)
HTD(J3) = MFT1(I)/(TMD(J3) - TSAT)
IF(XSM(J3).GT.0) XMAX(J3) = XMAX
IF(HTG(J3).GT.0) YMAX(J3) = YMAX
IF(HTD(J3).GT.0) YMIN(J3) = YMIN
IF(MTB(J3).GT.0) YMAX(J3) = YMAX
IF(MTD(J3).GT.0) YMIN(J3) = YMIN
125 WRITE(2,27)MTB(J3),HTG(J3),HTD(J3),XSM(J3),
3 TMC(J3),TMS(J3),TMD(J3)
27 FORMAT(1X,E18.3,1X,E18.3,1X,E18.3,1X,E18.3,
3 3X,E18.3,3X,E18.3,3X,E18.3)
78 CONTINUE
CALL SIMPLT(JOBOPT,XSM,NTG,N.1.8)
CALL SIMPLT(JOBOPT,XSM,NTS,N.1.8)
CALL SIMPLT(JOBOPT,XSM,MTD,N.1.8)
45 CONTINUE
48 CONTINUE
135 PRINT*,*
PRINT*,* THE VALUE OF NBS OF RUN FOR THIS PLOT IS: 2-N7
RETURN THE PLOT IS HEAT TRANSFER RATE FOR QUALITY 2
END

```

```

1 FUNCTION SIGMA(TEMP)
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
C THIS FUNCTION IS USED TO CALCULATE THE SURFACE TENSION
C AT A GIVEN SATURATION TEMPERATURE
CCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCCC
5 DIMENSION A(8)
DATA A/1.1649264E-01,E-121684688E+03,-5.752895188E-06,
3 -1.36274458E-08,-1.149719298E-11/
18 B = 8.8
TC = 647.3
T = TEMP - 273
SIGMA = A(1) * (TC-T)**2/(1.0 + B * (TC - T))
DO 15 I = 1,8
15 SIGMA = SIGMA + A(I) * (TC - T) ** I
SIGMA = SIGMA/1000.0
RETURN
END

```

APPENDIX G
CALCULATION OF THE INSIDE WALL TEMPERATURE

The temperature difference between the inside wall and the outside wall can be found using the equation for a uniformly distributed generation [Chapman, 1974]

$$\frac{d^2T}{dr^2} + \frac{1}{r} \frac{dT}{dr} + \frac{Q^*}{k} = 0 \quad \dots G.1$$

The equation of T by integration is

$$T = -\frac{r^2 Q^*}{4k} + B \ln r + C \quad \dots G.2$$

The boundary conditions are:

$$\text{at } r = r_1 \quad T = T_1$$

$$\text{at } r = r_0 \quad T = T_0$$

By the two boundary limit conditions, the values of B and C can be calculated. The heat flux per unit length may be found at the outlet surface by using:

$$\frac{Q}{L} = -k \, 2\pi \, r \, \frac{dT}{dr} \quad \dots G.3$$

Assuming a perfect insulation on the outside surface of the heated tube, it follows that $\frac{Q}{L}$ at the outside radius (r_0) is zero and the inside wall temperature is:

$$T_1 = T_o - \frac{Q^*}{2k} \left[r_o^2 \ln \left(\frac{r_o}{r_1} \right) - \frac{r_o^2 - r_1^2}{2} \right] \quad \dots G.4$$

Q^* : heat generation rate (W/m^3)

r_o : outside radius (m)

r_1 : inside radius (m)

This equation may be rewritten as

$$T_1 = T_o - \frac{Q}{2\pi k L} \left[\frac{r_o^2 \ln(r_o/r_1)}{(r_o^2 - r_1^2)} - \frac{1}{2} \right] \quad \dots G.5$$

Q : power generated in the test section excluding the hot patch (Watts)

L : heated length excluding the hot patch (m)

r_o : outside radius of the tube (m)

r_1 : inside radius of the tube (m)

k : thermal conductivity of the Inconel wall evaluated at the outside wall temperature ($W \cdot m^{-1} \cdot ^\circ C^{-1}$)

In the hot patch section, no heat is generated by the voltage drop of the tube. The electrical resistance of the hot patch is low compared with the electrical resistance of the tube. The general heat conduction equation in cylindrical coordinate is reduced to the following expression:

$$\frac{d^2 T}{dr^2} + \frac{1}{r} \frac{dT}{dr} = 0 \quad \dots G.6$$

With two temperature boundary limits, it can be proved that:

$$\frac{Q}{L} = \frac{2\pi k(T_1 - T_o)}{\ln(r_o/r_1)} \quad \dots G.7$$

$$\frac{Q}{2\pi r_1 L} = \frac{k(T_1 - T_o)}{r_1 \ln(r_o/r_1)} \quad \dots G.8$$

$$Q'' = \frac{k(T_1 - T_o)}{r_1 \ln(r_o/r_1)} \quad \dots G.9.$$

$$T_1 - T_o = \frac{Q'' r_1}{k} \ln \left(\frac{r_o}{r_1} \right) \quad \dots G.10$$

Q'' : heat flux at the hot patch (W/m^2)

APPENDIX H

CALCULATION OF THE EQUILIBRIUM QUALITY

The quality was calculated by the following equation:

$$X = \frac{H_{in} - H_f}{H_{fg}} + \frac{\phi \pi D L}{W H_{fg}} \quad \dots H.1$$

At the upstream location of the hot patch ($z=0$), the quality is expressed by:

$$X_1 = \frac{H_{in} - H_f}{H_{fg}} + \frac{\phi_{TS} \pi D TSLBHP}{W H_{fg}} \quad \dots H.2$$

where

TSLBHP: Heated length before the hot patch

ϕ_{TS} : Test section heat flux.

At the outlet of the hot patch, the quality is given by:

$$X_2 = X_1 + \left[\frac{\text{Power}_{HP}}{W H_{fg}} \right] \quad \dots H.3$$

Power_{HP}: Hot Patch Power (Watts)

In the hot patch region, the heat is provided only by the cartridge heater. The electrical resistance of the test section is low and the voltage drop across the test section in the hot patch region is negligible. The quality downstream of the hot patch is evaluated by:

$$X = X_2 + \frac{\phi_{TS} \pi D (z - 0.101)}{W H_{fg}} \quad \dots H.4$$

APPENDIX I

DATA REDUCTION

Corrected data are presented as following:

RUN: A B C D

A: Pressure (MPa)

9: 9.6 MPa

4: 4 MPa

B: Mass Flux

1: 1000 Kg/(m²·s)

2: 2000 Kg/(m²·s)

3: 2750 Kg/(m²·s)

4: 4500 Kg/(m²·s)

C: Inlet subcooling or positive quality

1: 50 KJ/Kg

2: 150 KJ/Kg

3: 250 KJ/Kg

4: X = 0.0

5: X = 0.08

6: X = 0.15

D: Type of measurement

1. Zero power heat balance

2. Power heat balance

3. Post-dryout (PDO) data

4. Spreading velocity

5. Quenching velocity

6. Minimum film boiling runs

7. Pre-dryout pressure drop at the same local condition
and heat flux

PRESSURE: Outlet pressure (kPa)
G: Mass flux ($\text{kg}/(\text{m}^2\text{-s})$)
TINC: Corrected inlet temperature due to heat loss ($^{\circ}\text{C}$)
PHI: Test section heat flux excluding the hot patch (kW/m^2)
HPPW: Hot patch power (kW)
XIN: Test section inlet equilibrium quality using corrected inlet temperature
XOUT: Test section outlet equilibrium quality
X1: Hot patch upstream quality
X2: Hot patch downstream quality
THPC: Corrected inside wall hot patch temperature ($^{\circ}\text{C}$)
TC: Corrected inside wall temperature ($^{\circ}\text{C}$)
(1-20)
PDO: Test section pressure drop (kPa)

Note: Measurement was not obtained when a zero value is encountered.

DATE	TIME	PRESSURE	C	TIME	PA	WIND	DIR	HEAT	K1	K2	THC
(DD)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)	(MM)
01/11/02	21:20:04	9999.7	1961.4	201.1	201.9	2.9	-114	025	-108	-080	754.3
01/11/02	21:21:06	9999.4	1964.1	201.1	201.3	2.9	-114	026	-108	-080	754.3
01/11/02	21:22:08	9999.3	1962.8	201.1	200.4	2.9	-113	025	-107	-079	754.8
01/11/02	21:23:10	9999.3	1965.5	201.1	200.4	2.9	-113	026	-104	-087	757.4
01/11/02	21:24:11	9997.6	1963.3	200.9	196.8	2.9	-114	023	-104	-080	754.3
01/11/02	21:25:13	9991.8	1964.9	200.9	194.9	2.9	-113	022	-105	-080	754.3
01/11/02	21:26:16	9994.7	1969.9	201.8	192.9	2.9	-115	021	-107	-080	754.3
01/11/02	21:27:18	9994.7	1969.9	201.8	192.9	2.9	-115	021	-107	-080	754.3
01/11/02	21:28:20	9997.4	1971.8	201.1	194.3	2.9	-116	020	-109	-081	777.0
01/11/02	21:29:22	9993.3	1963.2	201.1	194.4	2.9	-116	027	-109	-082	777.2
01/11/02	21:30:24	9998.0	1969.9	201.1	192.6	2.9	-117	024	-110	-093	776.8
01/11/02	21:31:26	9998.8	1961.2	201.1	194.9	2.9	-117	024	-110	-093	777.0
01/11/02	21:32:28	9993.3	1967.7	201.1	196.7	2.9	-116	026	-109	-092	777.2
01/11/02	21:33:30	9994.9	1968.8	201.1	194.4	2.9	-116	024	-109	-092	777.0
01/11/02	21:34:32	9997.8	1968.8	201.8	194.1	2.9	-114	020	-107	-080	777.2
01/11/02	21:35:34	9993.8	1968.6	201.1	194.9	2.9	-115	027	-109	-091	777.2
01/11/02	21:36:36	9997.2	1966.3	201.1	197.2	2.9	-115	023	-107	-080	777.3
01/11/02	21:37:38	9998.3	1961.3	201.8	196.5	2.9	-112	021	-105	-080	776.0
01/11/02	21:38:40	9992.1	1968.9	201.1	191.9	2.9	-114	020	-107	-080	776.1
01/11/02	21:39:42	9977.0	1967.2	201.1	192.6	2.9	-113	019	-104	-080	776.3
01/11/02	21:40:44	9990.5	1972.8	201.1	194.1	2.9	-113	020	-104	-080	776.8
01/11/02	21:41:46	9990.3	1966.9	201.1	192.1	2.9	-113	019	-104	-080	776.8
01/11/02	21:42:48	9990.3	1978.6	201.1	192.7	2.9	-113	020	-104	-080	779.3
01/11/02	21:43:50	9992.0	1962.4	201.1	192.8	2.9	-113	011	-104	-080	779.3
01/11/02	21:44:52	9990.2	1968.7	200.9	194.2	2.9	-111	014	-104	-087	779.3
01/11/02	21:45:54	9992.0	1961.9	201.8	192.6	2.9	-113	020	-104	-080	779.7
01/11/02	21:46:56	9993.0	1961.3	201.8	192.9	2.9	-113	014	-104	-080	780.0
01/11/02	21:47:58	9998.3	1969.9	201.8	197.7	2.9	-113	013	-104	-080	780.0
01/11/02	21:48:00	9994.3	1968.4	200.8	192.5	2.9	-112	007	-104	-080	780.3
01/11/02	21:49:02	9992.1	1963.3	200.9	197.2	2.9	-115	008	-104	-080	780.3
01/11/02	21:50:04	9994.9	1969.9	200.9	192.9	2.9	-113	008	-107	-080	780.8
01/11/02	21:51:06	9997.2	1964.6	200.9	194.6	2.9	-113	013	-104	-091	780.1
01/11/02	21:52:08	9992.8	1968.9	200.9	194.6	2.9	-114	011	-109	-092	780.2
01/11/02	21:53:10	9994.9	1964.9	201.0	194.7	2.9	-117	014	-109	-092	780.3
01/11/02	21:54:12	9994.9	1964.9	201.0	194.7	2.9	-117	014	-109	-092	780.3
01/11/02	21:55:14	9992.3	1961.2	200.7	194.7	2.6	-103	007	-176	-145	780.8
01/11/02	21:56:16	9994.7	1968.8	200.6	194.8	2.6	-103	006	-176	-145	780.8
01/11/02	21:57:18	9994.9	1961.3	200.6	194.8	2.6	-104	006	-177	-146	776.4
01/11/02	21:58:20	9994.9	1971.1	200.6	191.4	2.6	-104	103	-177	-145	771.1
01/11/02	21:59:22	9990.0	1964.9	200.6	194.6	2.6	-104	103	-177	-145	772.0
01/11/02	22:00:24	9990.2	1968.2	200.6	194.9	2.6	-104	007	-176	-147	772.3
01/11/02	22:01:26	9997.4	1969.6	200.6	191.3	2.6	-104	009	-177	-146	772.0
01/11/02	22:02:28	9998.8	1970.0	200.7	199.3	2.6	-104	103	-177	-146	772.0
01/11/02	22:03:30	9997.7	1969.7	201.4	194.3	2.6	-103	102	-173	-144	772.1
01/11/02	22:04:32	9999.7	1961.6	200.3	194.2	2.6	-103	000	-176	-145	770.9
01/11/02	22:05:34	9994.6	1968.3	200.6	194.7	2.6	-102	007	-175	-144	765.7
01/11/02	22:06:36	9992.1	1969.0	200.6	194.9	2.6	-102	007	-175	-145	764.2
01/11/02	22:07:38	9994.9	1963.3	200.6	191.8	2.6	-100	100	-174	-143	763.7
01/11/02	22:08:40	9992.1	1963.4	200.6	194.4	2.6	-102	009	-175	-140	762.0
01/11/02	22:09:42	9999.7	1969.9	200.6	191.2	2.6	-102	004	-175	-144	757.9

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	PCU
(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)	(C)
111.5	112.1	113.4	111.1	114.9	110.6	117.0	104.2	107.3	105.6	118.0	101.9	100.9	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.1
111.6	111.6	114.5	110.7	114.0	109.7	103.9	106.3	106.7	107.0	111.0	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.9	111.9	111.3	110.6	112.3	107.4	106.7	106.6	106.6	106.6	112.1	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
111.9	110.9	111.3	110.6	112.3	107.4	106.7	106.6	106.6	106.6	112.1	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
111.6	110.7	112.1	109.7	109.7	107.3	105.4	106.1	107.2	106.1	110.1	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
111.3	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
106.4	106.6	104.4	104.0	104.3	104.0	104.0	103.7	103.7	103.6	108.7	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
106.3	112.3	104.4	104.3	104.3	104.1	104.1	104.1	104.1	104.1	107.8	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
106.3	112.3	104.4	104.3	104.3	104.1	104.1	104.1	104.1	104.1	107.8	100.4	103.0	105.0	104.0	103.0	102.0	101.0	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3	106.3	104.3	102.3	100.3	101.3	100.7	112.4	100.7	103.4	105.3	104.2	103.7	102.5	101.1	0.0	0.0	0.0
111.6	110.8	109.3	108.3																	

7



RUN	DATE	TIME	T/C	P	PHI	G	XIN	HPPW	XMFB	TMFB
9126	25/10/82	21:38:56	12	9584.6	245.7	984.1	-.116	2.2	-.0463	455
9226	01/11/82	21:51:08	14	9637.4	334.5	1971.8	-.116	2.9	-.0751	494
9136	02/11/82	17:03:19	18	9607.2	363.5	971.5	-.192	2.8	-.1315	438
9146	16/04/83	16:26:09	14	9471.5	68.4	1008.6	.025	2.3	.0608	370
9166	16/04/83	21:20:42	14	9629.8	116.3	1011.2	.149	3.0	.1994	366
9156	16/04/83	22:31:16	16	9592.1	109.8	1058.9	.072	2.6	.1117	375
9246	16/04/83	23:36:20	16	9501.6	80.0	1958.9	.020	2.7	.0406	357
9256	17/04/83	00:41:18	16	9614.7	222.5	1960.0	.082	3.5	.1146	392
9266	17/04/83	01:44:28	16	9599.7	224.4	1981.5	.149	3.8	.1839	370
9346	17/04/83	03:01:59	16	9584.6	125.5	2558.6	.024	3.5	.0454	362
9346	24/05/83	19:44:12	16	9592.1	55.0	2869.7	-.005	3.7	.0119	348
9353	24/05/83	20:39:42	14	9622.3	185.3	2774.3	.057	4.8	.0865	372
9366	24/05/83	21:57:28	18	9509.2	275.5	2795.3	.130	5.5	.1598	362
9323	24/05/83	23:12:42	16	9562.0	278.9	2801.5	-.122	4.3	-.0888	520
9316	25/05/83	01:04:24	14	9524.3	80.2	2831.9	-.038	3.8	-.0189	363
9446	25/05/83	03:56:19	16	9592.1	97.4	4480.6	-.003	5.3	.0132	340
9446	25/05/83	02:47:37	16	9592.1	86.3	4494.9	-.006	5.2	.0100	339
9416	25/05/83	03:26:42	16	9516.7	154.6	4554.7	-.041	5.4	-.0238	340

T/C : Thermocouple number at the minimum film boiling location.

P : Pressure (kPa)

XMFB : Local thermodynamic equilibrium quality at the minimum film boiling location.

TMFB : Minimum film boiling temperature ($^{\circ}\text{C}$)

APPENDIX J

MINIMUM FILM BOILING RESULTS

High flow and high pressure minimum film boiling temperature data were obtained in this study. The minimum film boiling temperature separates the transition boiling region characterized by medium to high heat transfer coefficient from the film boiling region which is characterized by a low heat transfer coefficient. The minimum film boiling temperature is thus of great importance in reactor safety analysis. Minimum film boiling temperature has also been obtained by other experimenters. However, the temperature from previous experimenters is the apparent quench temperature which may be affected by axial conduction due to a resulting front. The true quench temperature resulting from a spontaneous collapsing was measured by reducing the power using the hot patch technique. The true quench temperature was also obtained by Fung [1981] and Stewart [1981] at lower mass flux and lower pressure using the hot patch technique. (The true quench temperature is lower than the apparent quench temperature.)

The data from the appendix I are plotted in Figure J.1. The resulting mechanisms are expected to depend on quality. Rewetting in the low quality region occurs probably due to a spontaneous collapse of the vapour film since the flow regime is expected to be inverted annular. In the high quality region (dispersed flow regime), the rewetting is caused by intermittent contact of the liquid droplets with the wall. It was found that the minimum film boiling temperature is approximately constant in the liquid dispersed region at 9.6 MPa.

The following two main mechanisms appear to govern the minimum film boiling temperature (Groeneveld [1982])

(i) Hydrodynamic mechanism. Separation of the liquid-vapour interface from the wall can only be maintained as long as the vapour thrust force (due to evaporation from and acting normal to the heated surface) exceeds the liquid force.

(ii) Thermodynamic mechanism. Here it is assumed that liquid can never exist beyond a "maximum liquid temperature" which depends only on the liquid properties and hence is a unique function of pressure".

Both mechanisms seem present in the data obtained at 9.6 MPa.

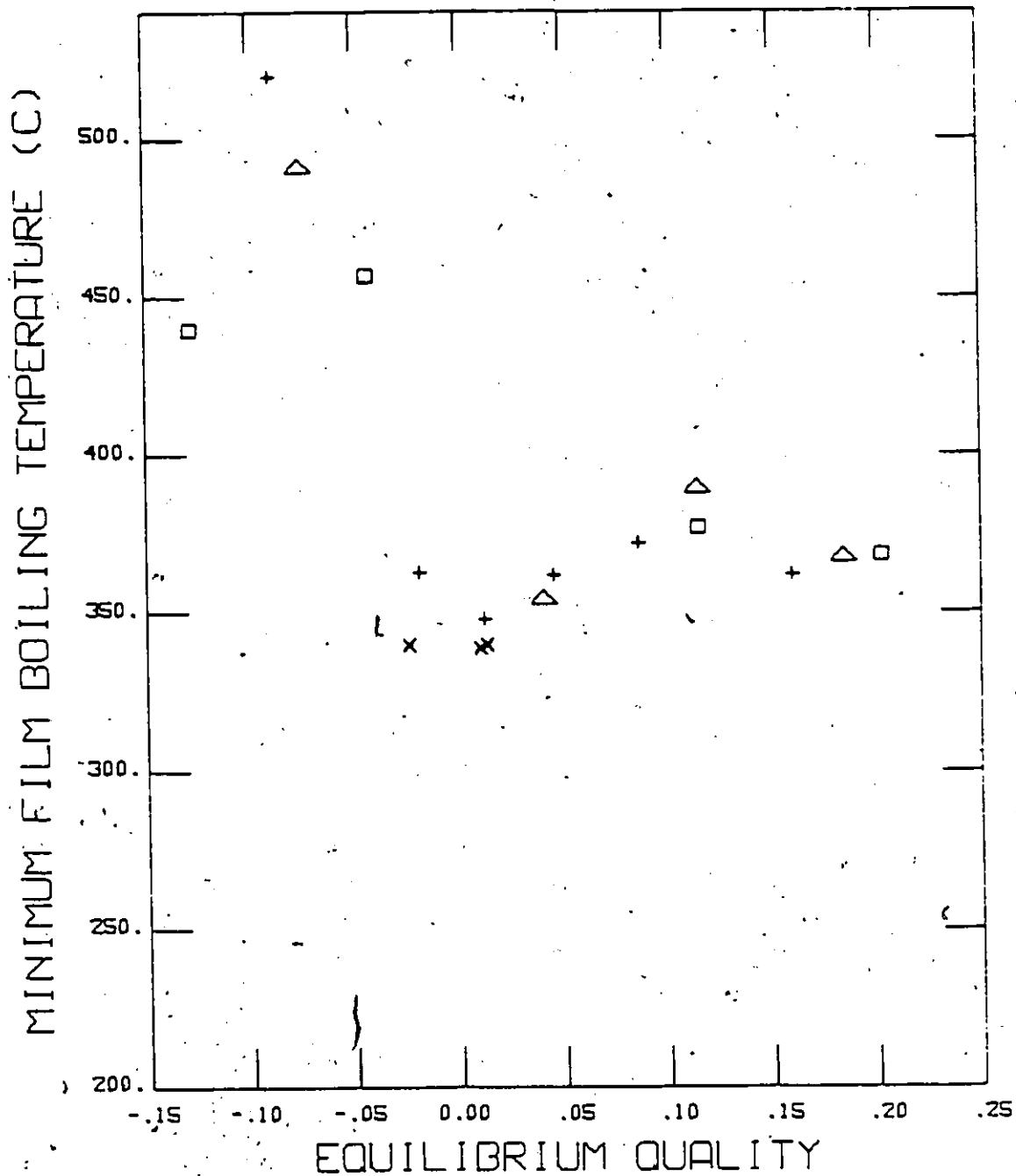


FIGURE J.1. The effect of the equilibrium quality on the minimum film boiling temperature data.

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