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FILM BOILING ON SPHERES IN SINGLE- AND TWO-PHASE FLOWS

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FINAL REPORT

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ABSTRACT

Film boiling on spheres in single- and two-phase flows was studied experimentally and theoretically with an emphasis on establishing the film boiling heat transfer closure law, which is useful in the analysis of nuclear reactor core melt accidents.

Systematic experimentation of film boiling on spheres in single-phase water flows was carried out to investigate the effects of liquid subcooling (from 0 to 40 °C), liquid velocity (from 0 to 2 m/s), sphere superheat (from 200 to 900 °C), sphere diameter (from 6 to 19 mm), and sphere material (stainless steel and brass) on film boiling heat transfer. Based on the experimental data a general film boiling heat transfer correlation is developed. Utilizing a two-phase laminar boundary-layer model for the unseparated front film region and a turbulent eddy model for the separated rear region, a theoretical model was developed to predict the film boiling heat transfer in all single-phase regimes.

The film boiling from a sphere in two-phase flows was investigated both in upward two-phase flows (with void fraction from 0.2 to 0.65, water velocity from 0.6 to 3.2 m/s, and steam velocity from 3.0 to 9.0 m/s) and in downward two-phase flows (with void fraction from 0.7 to 0.95, water velocity from 1.9 to 6.5 m/s, and steam velocity from 1.1 to 9.0 m/s). The saturated single-phase heat transfer correlation was found to be applicable to the two-phase film boiling data by making use of the actual water velocity (water phase velocity), and an adjustment factor of $(1-\alpha)^{1/4}$ (with α being the void fraction) for downward flow case only. Slight adjustments of the Reynolds number exponents in the correlation provided an even better interpretation of the two-phase data.

Preliminary experiments were also conducted to address the influences of multisphere structure on the film boiling heat transfer in single- and two-phase flows.

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Nomenclature

Ar	Archimedes number, $g(\rho_l - \rho_v)d^3/(\rho_v \nu_v^2)$
C_p	specific heat, $[J/(kg^\circ C)]$
d, D	diameter of sphere or cylinder, [m]
d'	dimensionless diameter, $d/[\sigma/g/(\rho_l - \rho_v)]^{1/2}$
Fr	Froude number, $U_l^2/(gd)$
g	gravitational constant, $[kg\ m/sec^2]$
Gr	Grashof number, $g\beta_l(T_{sat} - T_l)d^3/\nu_l^2$
h	heat transfer coefficient, $q/\Delta T$
h_{fg}	latent heat of vaporization, $[J/kg]$
h'_{fg}	latent heat plus sensible heat, $h_{fg} + 0.5C_{pv}\Delta T_{sup}$
K	density ratio, ρ_l/ρ_v
k	thermal conductivity, $[W/(m^\circ C)]$
l'	capillary length, $[\sigma/g/(\rho_l - \rho_v)]^{1/2}$
Nu	average Nusselt number, hd/k
Pr	Prandtl number
q	average surface heat flux, $[W/m^2]$
R	radius; or dimensionless ratio, $[(\mu\rho)_v/(\mu\rho)_l]^{1/2}$
Re _l	Reynolds number of liquid phase, $U_l d/\nu_l$
Sc	dimensionless subcooling, $(C_{pl}\Delta T_{sub})/(h_{fg}Pr_l)$
Sc'	dimensionless subcooling, $(C_{pl}\Delta T_{sub})/(h'_{fg}Pr_l)$
Sp	dimensionless superheat, $(C_{pv}\Delta T_{sup})/(h_{fg}Pr_v)$
Sp'	dimensionless superheat, $(C_{pv}\Delta T_{sup})/(h'_{fg}Pr_v)$
T	temperature, $[^\circ C]$
U	velocity, phase velocity
V	superficial velocity

Greek symbols

α	thermal diffusivity, $k/(\rho C_p)$; or void fraction
ΔT_{sub}	liquid subcooling, $T_{\text{sat}} - T_l$
ΔT_{sup}	average superheat of solid or solid wall, $T_w - T_{\text{sat}}$
ϵ	radiation emissivity
λ_c	critical wavelength, $2\pi l'$
μ	viscosity, $[\text{kg}/(\text{m sec})]$
ν	kinematic viscosity, $[\text{m}^2/\text{sec}]$
ρ	density, $[\text{kg}/\text{m}^3]$
σ	surface tension of vapor-liquid interface, $[\text{N}/\text{m}]$
σ_B	Stefan-Boltzman constant, $5.67 \times 10^{-8} [\text{W}/(\text{m}^2 \text{K}^4)]$

Subscripts

c	film boiling without radiation
f,F	forced convection film boiling
l	liquid, or water
nc	natural convection (non film boiling)
p	pool film boiling
r	radiation
s	saturated value, or steam
sat	saturated
sub	subcooled
t	total heat flux
two	two-phase
v	vapor, or steam
w	wall of sphere/cylinder, or water

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Chapter 1

Introduction and Overview

Film boiling is a major heat transfer mechanism that occurs when large temperature differences exist between a liquid and a hot surface. It has applications in cryogenic systems, metallurgic industries, and other areas where heating, cooling, and quenching of high temperature surfaces are encountered.

In recent years, film boiling has become an important concern in the nuclear reactor safety analyses. During postulated core melt accidents (CMAs), the core/fuel may melt, fall into water, and fragment into spheres or sphere-like particles. The subsequent fuel-coolant interaction may result in spontaneous evaporation --- a destructive steam explosion. One of the important concerns in the analysis of fuel-coolant interaction (premixing) is the heat transfer between the fuel particles and the water/steam two-phase flow in all the flow regimes, which cover from natural convection (pool film boiling) to forced convection, from saturated to highly subcooled, and from single-phase flow to two-phase flow.

Recently, Theofanous (1987), Amarasooriya (1991) and Yuen (1994) have developed a three-fluid model for general systems in a wide range of flow regimes. Such models have been found useful for establishing upper limits on the energy of large-scale steam explosions. In the absence of experimental data and appropriate film boiling correlations, the initial computations were based on single-phase film boiling correlation and extrapolations of two-phase formulations. The situations are the same for other fuel-coolant analysis codes. Thus, there is a need for a complete closure law --- the heat transfer correlation for film boiling on spheres in a wide range of single- and two-phase flow regimes.

Regarding previous work on film boiling from spheres, it can be concluded that: For two-phase Flows: No experiment or analysis for sphere film boiling in two-phase flows can be located. For Single-Phase Flows: (1) All the meaningful single-phase film boiling heat transfer data were obtained through short cool-down transients (by passing a preheated sphere through a liquid tank). Although the short cool-down transient technique is good enough for pool or low speed forced convection film boiling experiments, it is not adequate for high velocity forced convection film boiling, because the transient is too short. For example, with sphere speed at 2.0 m/s and pool length 0.5 m, the film boiling cool-down transient is only about 0.25 second! Within such a short time, the entrance thermal response may affect the accuracy of the experimental data. (2) The accuracy of the single-phase data may be very poor as a result of the large heat loss from the support-tube, since the sphere-support-tube diameter to sphere diameter ratios of the previous experimental work are about 0.12 to 0.3. (3) For each single-phase film boiling regime, the basic form of heat transfer correlation has been well developed, but there are some uncertainties in the correlation constants, as shown in Table 2.6.3. (4) There is no general correlation or complete theoretical analysis for film boiling on spheres which covers all the single-phase flow regimes. There is no theoretical model for the heat transfer in the separated rear region of the sphere.

The main purpose of this work is to establish the experimental data base and to derive the heat transfer correlation (closure law) for film boiling on spheres. The emphasis is on very high temperature film boiling (pure film boiling with the sphere temperature higher than the quenching point) in all the flow regimes, which cover from natural convection (pool film boiling) to forced convection, from saturated to highly subcooled, and from single-phase flow to two-phase flow. Besides this, theoretical analysis is carried out to understand the fundamentals of film boiling and to formulate better heat transfer correlations.

A versatile two-phase flow loop/mixer has been built to provide a variety of steady state single- and two- phase flow to obtain reliable, high quality, and complete experiment data. A radio frequency induction heating technique was investigated and

applied to heat metallic spheres directly inside the test section to maintain a steady state film boiling. With these setups, the film boiling heat transfer data can be obtained either by transient mode runs or by steady-state mode runs. An X-ray radiograph technique was applied to measure the void fraction of two-phase flow in the test section. Another unique feature of the present experiment approach is that it provides a convenient way to observe clearly the film boiling process, since the sphere is held still during the run. After the induction power is turned off, the induction coil could be moved away to allow the whole film boiling cool-down transient to be observed. With test spheres directly supported by small sheathed thermocouples (thermocouple O.D. vs. test sphere O.D. being: 0.25 mm / 6.35 mm; 0.5 / 9.35; and 0.81 / 12.7), the heat loss from the support is significantly minimized and is relatively small.

Systematic experimentation of film boiling on sphere in single-phase water flow has been carried out (Chapter 4) to investigate and check the effects of liquid subcooling (from 0 to 40 °C), liquid velocity (from 0 to 2 m/s), sphere superheat (from 200 to 900 °C), sphere diameter (from 6 to 19 mm), and sphere material (stainless steel and brass) on film boiling heat transfer. The single-phase experiment results are presented, discussed, and correlated in saturated pool, saturated forced convection, subcooled pool, and subcooled forced convection film boiling regimes, respectively; and then a general film boiling heat transfer correlation is developed (Chapter 6).

A theoretical analysis of film boiling from a sphere in single-phase flows was conducted (Chapter 5). With the two-phase laminar boundary layer model (an integral method modified from Shigechi, Ito and Nishikawa's (1983) model) for the un-separated front film region and the turbulent eddy model (based on Theofanous, Houze and Brumfield's (1987) model) for the separated rear region, the combined theoretical model can predict the film boiling heat transfer in all regimes.

The film boiling from a sphere in two-phase flows has been investigated (Chapter 7) both in an upward two-phase flow (with void fraction from 0.2 to 0.65, water phase velocity from 0.6 to 3.2 m/s, and steam phase velocity from 3.0 to 9.0 m/s) and

in a downward two-phase flow (with void fraction from 0.7 to 0.95, water phase velocity from 1.9 to 6.5 m/s, and steam phase velocity from 1.1 to 9.0 m/s). The void fractions were measured by an X-ray radiography. The calculated void fraction of downward two-phase flow, which is based on the acceleration due to the gravity and steam drag, agrees well with the X-ray measurements. The saturated single-phase heat transfer correlation is found to be applicable to the two-phase film boiling data by making use of the actual water velocity (water phase velocity), and an adjustment factor of $(1-\alpha)^{1/4}$ for downward flow case only. Slight adjustments of the Reynolds number exponents in the correlation provide an even better interpretation of the two-phase data. The power dependency on the Reynolds number should decrease to 0.2 in the upward two-phase case and increase to 0.75 in the downward two-phase case to correlate better the film boiling heat transfer data respectively.

The film boiling from a multi-sphere group has also been investigated (Chapter 8) through a five-sphere array with four spheres placed in the front of the test sphere. The experiments indicate that: (1) the presence of boiling spheres in the front of the test sphere decreases the film boiling heat transfer in the upward two-phase flow and increases the heat transfer in downward two-phase flow; and (2) in a saturated single-phase upward flow, the presence of boiling spheres in the front of the test sphere increases the heat transfer in pool film boiling regime and decreases the heat transfer in forced convection film boiling regime. These trends are consistent with the expectation using the single-sphere correlation and the theoretical interpretation in conjunction with the flow field changes due to the presence of the four spheres ahead of the test sphere.

Chapter 2

A Review on Film Boiling From Spheres and Horizontal Cylinders in Single-Phase Flows

2.1. Film Boiling and Radiation Effect

Boiling is the evaporation process that occurs at the interface of a liquid and a hot solid or another hot liquid. This process is associated with mass and heat transfer from the hot surface to the liquid through the vapor film. According to the temperature difference across the interface and the configuration of the vapor film around the interface, boiling is usually classified into three sub-groups: nucleate, transition and film boiling.

In film boiling, a complete vapor film is formed at the interface which prevents the liquid directly contacting with the hot solid (or liquid) surface. Film boiling only exists when the temperature difference across the interface is large enough to sustain it (usually over 100 °C ~ 700 °C depending on subcooling, surface condition and liquid flow velocity). In film boiling, the heat transfer mechanism is mainly by heat conduction and radiation across the vapor film. The low conductivity of the vapor film through which the heat must be transported characterizes the low value of heat transfer coefficient. The radiation heat transfer contributes higher and higher percentage in the total heat flux when the temperature difference becomes bigger and bigger.

According to the geometry of the interface, film boiling can be categorized into: film boiling on plane surfaces (vertical, horizontal upward, horizontal downward, inclined and curved), on cylinders, and on spheres. Actually, the film boiling on sphere and on cylinder are quite similar to each other, both in vapor film configurations and in the ways that the heat transfer data are correlated. Here, the review and discussion are limited only on the film boiling from cylinders and spheres.

On the configuration of the vapor film on spheres and cylinders, several papers are relevant, such as Walfort (1969), Stevens and Witte (1971), Aziz and Hewitt (1986), Zvirin, Hewitt, and Kenning (1990), and Dix and Orozco (1990). According to these observations, the conclusions about the vapor film configuration on the front part of the sphere or cylinder (with $\theta < \pi/2$) are consistent: a complete vapor film either smooth or wavy is covered on the front of the sphere/cylinder. But, on the other hand, the descriptions of the film configuration on the rear part are quite diverse. There are two reasons for this diversity. Firstly, the shape of the rear vapor film itself depends on the liquid condition: the velocity of flow and the degree of subcooling. Secondly, the back film configurations are also significantly affected by the sphere supports, which are at the rear side. For most of the previous experimental observations, the support effects were not avoided and even not discussed.

Regarding the influences of the sphere/cylinder itself on the film boiling, some experimental investigations [such as Bromley (1950) and Dhir (1978)] indicate that, if only the convective heat transfer is considered, the materials, the properties and the surface conditions do not affect the film boiling heat transfer significantly. However, the thermal conductivity of the sphere may affect the temperature distribution inside the sphere, and the radiation heat transfer (emissivity of the surface) strongly depends on the material and the surface condition of the spheres/cylinders. Moreover, the materials, properties, and surface conditions of the sphere significantly affect the quenching (or transition) point and the heat transfer during the transition boiling. In conclusion, for film boiling, the sphere itself only affects the radiation heat transfer but not the convective heat transfer.

On the other hand, in regard to the effects from the liquid, many quantities and conditions should be considered. Firstly, the properties of the liquid should be considered. Such properties are densities, heat conductivities and viscosities of the liquid phase and the vapor phase, saturation temperature at certain system pressure, latent heat of evaporation and surface tension at the liquid vapor interface. Secondly, the liquid temperature and the system pressure must be specified, which determines the subcooling of the liquid. According to this, the film boiling is classified into:

saturated and subcooled film boiling. In the case of saturated film boiling, most of the heat that is transferred through the interface becomes the heat source for evaporation, and the produced vapor does not (or hardly) condense. So, the front vapor film is thick and wavy; the film on back of the sphere becomes thicker and thicker and eventually develops into a stable (or unstable) vapor dome; there is a two-phase wake behind the sphere. In contrast, in the case of subcooled film boiling, most of the heat transferred to the interface is convected away into the liquid and only part of the heat becomes the heat source for net evaporation. The bubbles that separated from the vapor film condense very quickly and vanish after a short path in the liquid. Finally, according to the liquid flow velocity, the film boiling can be divided into two regimes: pool (or natural free convection) and forced convection film boiling. Experimental investigations indicate that as long as the square root of the Froude number is smaller than $1.0 \sim 2.0$, the velocity of the liquid does not affect the film boiling heat transfer rate, and this serves as the criterion to distinguish the pool and forced convection film boiling.

There are many aspects of film boiling; a complete review on the film boiling from spheres/cylinders is not intended here. Since our purpose is to get new data by using new techniques and to develop a general correlation for film boiling on spheres in single- and two-phase flows, the review will emphasize existing film boiling analyses, correlations, experimental techniques and experimental results. The review is divided into four sections according to the flow velocity and liquid subcooling, which are saturated pool, subcooled pool, saturated forced convection and subcooled forced convection film boiling.

Before the detailed review on the film boiling which only takes account to the convective heat transfer (with the radiative heat transfer contribution being subtracted away), let us look how the radiation heat transfer is considered first.

Since radiation from the sphere/tube is largely absorbed in a small thickness of the liquid, it serves to produce more vapor that goes into the vapor film. This means that with the existence of radiation the convective contribution of film boiling heat transfer will be affected, and the heat transfer contributions of the convection and

radiation can not be simply added together.

Bromley (1950) did an analysis for the case of pool film boiling and obtained an equation

$$h = h_c (h_c/h)^{1/3} + h_r \quad (2.1.1) .$$

He also indicated that , as long as h_r is smaller than h_c , the simple equation

$$h = h_c + 0.75 h_r \quad (2.1.2) ,$$

gives a good approximation to Eq. (2.1.1) within an accuracy of 5%. If h_r is very large (h_r/h_c ranges up to 10.0) the following equation was suggested

$$h = h_c + h_r \{ 0.75 + 0.25/(2.62h_c/h_r + 1) \} \quad (2.1.3) .$$

Bromley, Leroy and Robbers (1953) did a similar analysis for the case of forced convection film boiling by assuming that all heat transfer above the separation point θ_s is by radiation , and they obtained an expression

$$h = h_c + h_r \{ 1 - \theta_s/(4\pi) \} \quad (2.1.4) .$$

At high velocities, θ_s equals $\pi/2$, so

$$h = h_c + (7/8)h_r = h_c + 0.88h_r \quad (2.1.5) .$$

In all the above cases, h_r is given by

$$h_r = \epsilon \sigma [T_w^4 - T_s^4] / \Delta T_{sup} \quad (2.1.6) .$$

Since then, Eq. (2.1.2) and (2.1.5) are usually used in literature when radiation is concerned. However, recently, Sakurai, Shiotsu and Hata (1990) did a rigorous numerical solution for pool film boiling with subcooling. By comparing the results with and without radiation, they suggested that the radiation effect can be considered in the following way

$$h = h_c + Jh_r \quad (2.1.7) ,$$

$$J = F + (1-F)/(1+1.4h_c/h_r) \quad (2.1.8) \quad ,$$

$$F = [1-0.25\exp(-0.13Sp)] \cdot \exp(-0.64R^{0.6}Pr^{0.65}Sp^{-0.73}Sc^{1.1}) \quad (2.1.9) \quad .$$

The J factors for pool and forced convection film boiling (with the heat transfer coefficient h being estimated by our general correlation, see chapter 6) are plotted in Fig. 2.1 and Fig. 2.2 respectively; they show a big difference between the two ways of calculating the radiation factors, especially when the sphere superheat is low and liquid subcooling is large. The ratios of total heat transfer coefficients obtained by using Eq.(2.1.5) and Eq.(2.1.7) are given in Fig. 2.3 and Fig. 2.4 for pool and forced convection film boiling cases respectively. In the case of pool film boiling, the maximum difference is about 16 % which occurs at the condition of 80 °C liquid subcooling and 1300 °C wall superheat. In the case of forced convection film boiling, the maximum difference is about 11 % at the condition of 80 °C liquid subcooling and 2200 °C wall superheat. These comparisons indicate that the complicated Sakurai's radiation factor dose not differ greatly from the simple Bromley's factor of 7/8, and the Bromley's factor may still be used in a practical sense.

Fig. 2.1 Radiation J factor in the case of pool film boiling.

Fig. 2.2 Radiation J factor in the case of forced convection film boiling.

Fig. 2.3 The ratio of heat transfer coefficients obtained by Eq. (2.1.5) and Eq. (2.1.7) in pool film boiling.

Fig. 2.4 The ratio of heat transfer coefficients Obtained by Eq. (2.1.5) and Eq. (2.1.7) in Forced Film Boiling.

2.2. Saturated Pool Film Boiling

A. 1/4 Power Law

Bromley (1950) is the first work to study the pool film boiling at saturated condition systematically. It showed that heat transfer coefficients are independent of the tube materials except for the radiation contribution. By an analysis that is similar to the Nusselt's analysis for condensation, Bromley obtained:

$$Nu = C[Ar/Sp]^{1/4} \quad (2.2.1)$$

Where the Archimides number Ar is given as: $Ar = g(\rho_l - \rho_v)d^3/(\rho_v \nu_v^2)$. He argued that, in general, the constant C is function of $Sp' = [(\Delta T_{sup} C_{pv})/(h'_{fg} Pr_v)]$. According to the stable pool film boiling experiments that were conducted with electrical heated graphite tubes ($D = 9.8 - 16.2$ mm) and with 7 different fluids (water, soap solution, liquid nitrogen, n-pentane, benzene, carbon tetrachloride and ethyl alcohol) at atmosphere pressure, he concluded that the constant C does not vary significantly and suggested that a constant $C = 0.62$ could be used to correlate all of his data. Thus it gives,

$$Nu = 0.62[Ar/Sp]^{1/4} \quad (2.2.1')$$

Ten years later, Berenson (1961) applied the Taylor-Helmholtz Hydrodynamic Instability concept to his analysis of pool film boiling from a large horizontal surface, and he reached a correlation,

$$Nu_{\lambda_c} = h\lambda_c/k_v = 0.672 [Ar_{\lambda_c}/Sp]^{1/4} \quad (2.2.2)$$

which is similar to that of Bromley's correlation for horizontal cylinder besides the substitution of critical wave length $\lambda_c = 2\pi l'$ for the tube diameter in Nu and Ar number.

In 1963, Frederking and Clark, applied a laminar analysis which is similar to that used by Bromley to the pool film boiling from a sphere and got a similar correlation:

$$Nu = 0.586 [Ar/Sp]^{1/4} \quad (2.2.3) .$$

Dhir and Purohit (1978) did an experiment with 19 and 25.4 mm spheres of steel, copper, and silver in water. They found that the properties of the sphere do not affect the heat transfer coefficient provided a stable film exists, which agrees with Bromley (1950)'s argument. When they correlated their pool film boiling data at saturated condition, they got:

$$Nu = 0.8[Ar/Sp]^{1/4} \quad (2.2.4) .$$

The constant is higher than that in Eq. (2.2.3), however it is not clear which is right because of the following two reasons. First, because their support tube is 3 mm in diameter, the heat loss from the support could contribute large portion in the total heat transfer. Second, their experiment was operated at the "minimum film boiling temperature" state, the boiling may be partially in the transition boiling regime. So their experimental data may over estimated the film boiling heat transfer from sphere, and the correlation constant $C = 0.8$ may be too high for pool film boiling.

In addition to the above literature, there are also some others that obtained or applied the $1/4$ power law for pool film boiling at both saturated and subcooled conditions, such as Farahat and Nasr (1975), Farahat and Halfawy (1975), Farahat (1978), Klimenko (1980), Michiyoshi, Takahashi and Kikuchi (1988), Tso, Low and Ng (1990), Sakurai, Shiotsu, and Hata (1990a,b). Especially, when the diameter effect on the heat transfer is concerned, the base of the correlation is usually the $1/4$ power law, this will be discussed later.

B. $1/3$ Power Law

Through research which concerns the pool film boiling from spheres (usually, $D > 20$ mm) in cryogenic liquids or in some organic liquids which have small capillary length l' , the correlations with $1/3$ power law were developed.

The first work may be attributed to Frederking and Clark (1963). They obtained a $1/4$ power law from their theoretical analysis, however, the $1/3$ power law emerged from the fact that the pool film boiling heat transfer coefficients obtained from their experiments were independent of the diameter of the test sphere over their range of geometry. So they applied the $1/3$ power law to correlate their data in the form of

$$Nu = 0.14[Ar/Sp']^{1/3} \quad (2.2.5) \quad .$$

Merte and Clark (1964) did an experiment by cooling-down a sphere ($D = 25.4$ mm) in liquid nitrogen under different gravity conditions ($0.0 < a/g < 1.0$). Their data were correlated by:

$$Nu = 0.15[Ar/Sp' \cdot (a/g)]^{1/3} \quad (2.2.6) \quad .$$

They argued that the power increasing from Bromley's $1/4$ to $1/3$ is due to the transition from laminar to turbulent flow, and the transition appears to occur at a Rayleigh number of about 5×10^7 .

In the researching of oscillation effect on the pool film boiling heat transfer, Rhea and Nevins (1969) conducted an experiment with 25.4, 19.0 and 12.7 mm diameter spheres in liquid nitrogen, and Schmidt and Witte (1972) did an experiment with 19 mm diameter sphere in Freon-11. Both of them used the same correlation

$$Nu = 0.14[Ar/Sp' \cdot \{a/g + x^2 f^2 / (dg)\}]^{1/3} \quad (2.2.7) \quad ,$$

where x and f are the amplitude and frequency of the oscillation respectively.

There are also some other literature that apply $1/3$ power law to saturated pool film boiling. Such as, Frederking, Chapman, and Wang (1965), Barron and

Gorgolis (1977), Farahat and Nasr (1978), Kliimenko (1980), Grigoriev, Klimenko and Shelepen (1982).

It seems that with big diameter sphere or in cryogenic liquids, 1/3 power law should be used to correlate the pool film boiling heat transfer data if $d/l' > 10$.

C. The Diameter Effect.

Banchero, Barker, and Boll (1955) found that Bromley's equation did not correlate well with their data over a wide range of diameters. They modified Bromley's correlation to the form of

$$Nu = A(1/d+C)d^{1/4}[Ar/Sp]^{1/4} \quad (2.2.8) \quad ,$$

where A and C are constants determined from experiment: A is given as about $0.045 \text{ (in)(ft)}^{-1/4}$, presumably for a variety of liquids, whereas C is 9.5, 36.5 and 84.0 $(\text{in})^{-1}$ for n-pentance, oxygen, and water, respectively, at atmospheric pressure. This correlation does not allow the heat transfer coefficient to approach zero as the diameter approaches infinity. The disadvantage of this correlation is that the constant C depends on the kind of boiling liquid.

In 1962, Bareen and Westwater made another contribution to the understanding of the diameter effect on the pool film boiling heat transfer on cylinders. By introducing the Berenson's (or critical) wave length λ_c , they correlated all available pool film boiling heat transfer data on horizontal tubes by means of an empirical equation:

$$Nu = (0.59+0.069\lambda_c/d)(d/\lambda_c)^{1/4}[Ar/Sp]^{1/4} \quad (2.2.9) \quad .$$

Baumeister and Hamill (1967) developed a theoretical model for pool film boiling from small diameter wires. Following the trend of the experiment data, they suggested a equation

$$Nu = 0.373 \left[\left(\frac{d}{l} \right) \left(1 + \frac{9}{\sqrt{6}} \frac{l}{d} + \frac{8}{3\sqrt{6}} \left(\frac{l}{d} \right)^3 \right) \right]^{1/4} \left(\frac{Ar}{Sp} \right)^{1/4} \quad (2.2.10)$$

Hendick and Baumeister (1969) following the same idea developed a theoretical model for film boiling from spheres. They characterized the size of the vapor dome by the critical wave length λ_c and applied the principle of a maximum rate of entropy production. The correlation is given as

$$Nu = 2 + \frac{l}{4} \left[-\frac{2}{3} \frac{Ar}{Sp} G(B_o) \right]^{1/4} + \left[0.177 \left(\frac{Ar}{Sp} \sqrt{B_o} \right)^{1/4} + \csc(\theta^*(B_o)) \right] (1 + \cos(\theta^*(B_o))) \quad (2.2.11)$$

Where $B_o = (d/l)^2$ and $\theta^*(B_o)$, $G(B_o)$ are functions of B_o given in graphical form. This correlation was verified by experimental data with small spheres ranging from 0.397 to 12.7 mm in diameter by Hendick and Baumeister (1970).

Irving and Westwater (1986) and Westwater, Hwalek and Irving (1986), in their research on the limitation for obtaining boiling curves by the quenching method, indicated that when $d/\lambda_c > 7.8$, the heat transfer coefficient will not depend on the diameter of the sphere.

Grigoriew, Klimenko and Shelepen (1982) combined the 1/4 power law and 1/3 power law to account for the diameter effect and the turbulent effect. They use 1/4 power law for laminar film boiling when diameter is small and 1/3 power law for turbulent film boiling when diameter is large.

$$Nu = 0.7 Ar^{1/4} Pr_v^{1/3} f_1(K), \quad Ar < 3 \times 10^7 \quad (2.2.12a)$$

$$f_1(K) = \begin{cases} 1.0 & K \leq 1.4 \\ 0.92 K^{1/4} & K > 1.4 \end{cases}$$

$$Nu = 0.165 Ar^{1/3} Pr_v^{1/3} f_2(K), \quad Ar \geq 3 \times 10^7 \quad (2.2.12b)$$

$$f_2(K) = \begin{cases} 1.0 & K \leq 1.6 \\ 0.85K^{1/3} & K > 1.6 \end{cases}$$

Where $K = h_{fg}/(C_{pv}\Delta T_{sup})$. This correlation, was claimed, fits well with the data from 11 research groups which include 5 liquids and different size spheres with the diameter ranging from 0.25 to 96 mm.

Sakurai, Shiotsu and et al. (1990a, 1990b, 1992) introduced an empirical diameter-correction factor $K(d')$ to account for the diameter influence in their correlation.

$$Nu/(1+2/Nu) = K(d') (Ar/Sp)^{1/4} M_c^{1/4} \quad (2.2.13) ,$$

$$\begin{aligned} K(d') &= 0.44d'^{-1/4}, & \text{for } d' < 0.14 \\ K(d') &= 0.75/(1+0.28d'), & \text{for } 0.14 < d' < 1.25 \\ K(d') &= 2.1d'/(1+3.0d'), & \text{for } 1.25 < d' < 6.6 \\ K(d') &= 0.415d'^{1/4}, & \text{for } d' > 6.6 \end{aligned}$$

where $M_c(p_{sys}, \Delta T_{sup}, \Delta T_{sub})$ is a function of liquid and steam properties at given $p_{sys}, \Delta T_{sup}, \Delta T_{sub}$, which will be given later when the pool film boiling at subcooled condition is concerned. The correlation was verified by their pool film boiling steady-state experiment with cylinders of diameter ranging from 0.3 to 6.0 mm, in 7 different liquids and at system pressures ranged from 1 bar to 5 bar.

It is interesting to compare the above correlations. The six correlations were plotted in terms of $Nu_l/(Ar_l/Sp)^{1/4}$ and l'/d in Fig. 2.5 at a condition of 1.0 bar system pressure and 600 °C sphere superheat. The nondimensional group for the y coordinate is independent of diameter, and it could be expressed as

$$\frac{Nu_r}{(Ar_r/Sp)^{1/4}} = \frac{Nu_d}{[(Ar_d/Sp)(d/l)]^{1/4}} = h \left\{ \frac{h'_{fg} g k_v \rho_v (\rho_l - \rho_v)}{l' \mu_v \Delta T_{sup}} \right\}^{-1/4}$$

Fig. 2.5 Comparison of pool film boiling correlations for saturated condition. Comparison of experimental data with the correlations.

The comparison shows that in the l'/d range of 0.2 to 1.0, all the correlations are well agreed in a band less than 10% except Hendricks's correlation which is about 20% higher than others. Bromley's correlation is applicable in this range, but out of this range, Bromley's under predicted the pool film boiling heat transfer. For big diameter case, all correlations tend to depart from Bromley's at the l'/d value of about 0.15 and then keep at a constant level in a band of about 13%. On the other hand for small diameter, all correlations start to deviate from Bromley's at the l'/d value of about 2.0 except the Grigoriev's correlation. In the l'/d range of 2.0 to 10.0, the Hendricks's and Breen's correlation are about 30% higher than Baumeister's and Sakurai's. Under other system pressure and superheat conditions, the comparisons is little bit different, but not significantly.

D. Experiments of Pool Film Boiling From Sphere in Water

Besides Dhir (1978), as mentioned above, there are also several other saturated pool film boiling experiment (conducted with cool-down transient of sphere in water) could be found in the literature. Bradfield (1967) obtained film boiling cool-down transient curves by 59 mm diameter pure copper sphere in water at temperatures of 95, 53 and 27 °C and at atmospheric pressure. Marschall and Farrar (1975) obtained the heat transfer coefficient by a partially submerged inconel sphere of 18.75 mm in diameter. Toda and Mori (1982) did an experiment with a 12.7 mm stainless steel sphere. Aziz , Hewitt and Kenning (1986) studied the film boiling on stainless steel spheres of 10 and 20 mm in diameter

Some of the experimental results are given in Table 2.1 in terms of the 1/4 power law $C-1/4$ and $C_f = Nu_f/(Ar_f/Sp)^{1/4}$. The latter are plotted in Figure 2.5. It is clear that the experimental data are about 10-20% higher than the correlations except that Aziz's data are on the line of Hendricks's. There may be two reasons for this: firstly, in the experiments the water was not completely at saturated condition, and secondly, the heat losses from the sphere support have not been concerned or correctly counted.

Table 2.1 Experiments of Saturated Pool Film Boiling

Reference	d(mm)/(d/l')	T _w (°C)	C-1/4	C _f
Bradfield(67)	59.7/23.0	600.0	1.13	0.53
Dhir (1978)	25.4/9.77	210.0	0.86	0.49
Dhir (1978)	19.0/7.31	209.0	0.86	0.51
Toda (1982)	12.7/4.88	320.0	0.88	0.59
Aziz (1986)	20.0/7.69	510.0	0.91	0.45
Aziz (1986)	10.0/3.80	475.0	0.88	0.54

$$C-1/4 = Nu/[Ar/Sp]^{1/4}; \quad C_f = Nu_f/[Ar_f/Sp]^{1/4}$$

2.3 Subcooled Pool Film Boiling

A. The Addition Law

The earliest work on the subcooled pool film boiling may go back to 1960's. Tachibana and Fukui (1961) experimentally and theoretically studied the pool film boiling from fine nickrome wires with diameters of 0.5 and 0.8 mm in water, alcohol and carbon tetrochloride over a subcooling range of 0.0 to 45 °C at atmospheric pressure, but no correlation was generated.

Hamill and Baumeister (1967) did a theoretical analysis of film boiling from a horizontal plate with subcooling and radiation. The analysis is based on the postulation that the rate of entropy production is maximized. The general solution enables the total heat transfer coefficient to be calculated by addition of the saturated pool film boiling coefficient h_{sat} , radiation heat transfer coefficient h_r , and the subcooled turbulent convective heat transfer contribution term, which is given by

$$h_t = h_{sat} + 0.88h_r + 0.12 h_{nc} \Delta T_{sub}/\Delta T_{sup} \quad (2.3.1) .$$

Where $h_{nc} = q_{nc}/\Delta T_{sub}$ is the turbulent free-convection heat transfer coefficient for subcooled liquid, and $h_r = \epsilon \sigma (T_w^4 - T_s^4)/\Delta T_{sup}$ is the radiation heat transfer coefficient.

Bradfield (1967) experimentally studied the effect of subcooling on pool film boiling from a pure copper sphere (59 mm in diameter) and obtained the film boiling cool-down transient heat flux curves in water at temperatures of 95, 53 and 27 °C at atmospheric pressure. The study shows a strong effect of the subcooling.

Siviour and Ede (1970) investigated the subcooled pool film boiling from horizontal tubes of diameter 3.2 and 6.4 mm in water with 0-80 °C liquid subcooling. They correlated their experimental data by the addition law:

$$Nu_t = Nu_{sat} + J Nu_r + Nu_{nc} (Sc/Sp)(\mu_l/\mu_v) \quad (2.3.2) ,$$

$$Nu_{sat} = 0.613[Ar/Sp]^{1/4}, Nu_{nc} = 0.59[GrPr]^{1/4} Pr^{1/4} .$$

Where J is 0.78, $Nu_r = h_r d / k_v$, and $Nu_{nc} = h_{nc} d / k_l$.

Farahat, Eggen and Armstrong (1972,1974,1975) studied the pool film boiling from spheres (with diameters of 25.4, 19.0, and 12.7 mm) in subcooled sodium and water. Following the analysis of Hamill and Baumeister (1967), They correlated their data by

$$Nu_t = Nu_{sat} + 0.88Nu_r + KNu_{nc}(Sc/Sp)(\mu_l/\mu_v) \quad (2.3.3) ,$$

$$Nu_{nc} = 0.75[GrPr^n]^{1/4} ,$$

$$K = 11.9 (\Delta T_{sub})^{-0.7} , \quad \text{For } d = 25.4 \text{ mm}$$

$$K = 18.8 (\Delta T_{sub})^{-0.65} , \quad \text{For } d = 19-13 \text{ mm}$$

Where Nu_{sat} is calculated by Hendricks and Baumeister (1969)'s correlation, $n = 1$ is for nonmetallic fluids and $n = 2$ is for metallic liquids. The K is a function of liquid subcooling and the diameter of the sphere.

Dhir and Purohit (1978) also applied the addition law to correlate their subcooled film boiling data that were obtained by cool-down transient of 19 and 25.4 mm diameter spheres of steel, copper and silver in water at atmospheric pressure. The correlation is given by

$$Nu_t = Nu_{sat} + C_r Nu_r + Nu_{nc}(Sc/Sp)(\mu_l/\mu_v) \quad (2.3.4) ,$$

where

$$Nu_{sat} = 0.8[Ar/Sp]^{1/4} , \quad Nu_{nc} = 0.9[GrPr]^{1/4} .$$

B. Ratio Law

Shih and El-Wakil (1981) carried out an analysis for pool film boiling from a sphere by using an integral method. Based on their theoretical analysis, they obtained:

$$Nu/Nu_{sat} = 1 + 13.91 [ScAr/Gr]^{0.39} \quad (2.3.5) .$$

They emphasized that: for subcooled film boiling it is the ratio, not the difference between the subcooled and the saturated film boiling Nusselt numbers, that is significant. They also claimed that their experiment with stainless steel spheres of 3.2, 4.8 and 6.4 mm in diameter in Freon-11 and Freon-113 with 0-20 °C subcooling supports this correlation.

Michiyoshi, Takahashi and Kikuchi (1988) obtained an universal correlation for subcooled pool film boiling from vertical plate, horizontal plate and sphere by an analysis with integral method,

$$Nu = K (Ar/Sp)^{1/4} M_c^{1/4} \quad (2.3.6) .$$

where

$$\begin{aligned} M_c &= E^3 / [1 + E / (Sp \cdot Pr)] / (R \cdot Pr \cdot Sp)^2 \\ E &= (A + C B^{1/2})^{1/3} + (A - C B^{1/2})^{1/3} + (1/3) Sc^* \\ A &= (1/27) Sc^{*3} + (1/3) R^2 Sp \cdot Pr Sc^* + (1/4) R^2 Sp^2 Pr^2 \\ B &= (-4/27) Sc^{*2} + (2/3) Sp \cdot Pr Sc^* - (32/27) Sp \cdot Pr R^2 \\ &\quad + (1/4) Sp^2 Pr^2 + (2/27) Sc^{*3} / R^2 \\ C &= (1/2) R^2 Sp \cdot Pr, \quad Sc^* = C_{pl} \Delta T_{sub} / h'_{fg} \end{aligned}$$

Where $K = 0.696$ for sphere, $K = 0.610$ for horizontal cylinder, and $K = 0.793$ for vertical plate. Comparing this correlation with their own data in water and potassium and others data from literature, they concluded that: the correlation is suitable for characterizing pool film boiling in various nonmetallic liquids especially for water.

Later in 1990, Tso, Low and Ng presented a detailed derivation of this correlation for sphere case, which is given Appendix A. They also compared their experimental data which were obtained by cool-down transient of copper spheres (20 and 25 mm in diameter) in Freon-12 over a subcooling range of 0 to 70 °C. Their data are clearly (about 50%) higher than the theoretical correlation.

Sakurai, Shiotsu and Hata (1990a) carried out a rigorous solution to a theoretical

pool film boiling model which including the radiation contribution for the case of horizontal cylinder. Their study indicated that the rigorous solution is in good agreement with the simple analytical solution which is the same as that obtained by Michiyoshi (1988).

Sakurai etc. (1990b) conducted a systematic experiment with electrical heated platinum cylinders in various liquids, including water, ethanol, isopropanol, Freon-113, Freon-11, Liquid nitrogen and liquid argon. The experiment was carried out in a wide range of system pressure, liquid subcooling, surface superheat and cylinder diameter. Based on their experiment, they modified their analytical solution by introducing a diameter depending empirical function $K(d')$ on the right hand side of the correlation and a factor $(1+2/Nu)^{-1}$ on the left hand side, which is only effective when Nu is very small. Their correlation is given as

$$Nu/(1+2/Nu) = K(d') (Ar/Sp)^{1/4} M_c^{1/4} \quad (2.3.7)$$

where

$$\begin{aligned} K(d') &= 0.44d'^{-1/4}, & \text{for } d' < 0.14 \\ K(d') &= 0.75/(1+0.28d'), & \text{for } 0.14 < d' < 1.25 \\ K(d') &= 2.1d'/(1+3.0d'), & \text{for } 1.25 < d' < 6.6 \\ K(d') &= 0.415d'^{1/4}, & \text{for } d' > 6.6 \end{aligned}$$

Several other theoretical studies on subcooled pool film boiling from tubes, such as Nishkawa, Ito and Kuroki (1972), Nishkawa and Ito (1966), Srinivasan and Rao (1984), Nakayama and Koyama (1986a,1986b) could also be located.

C. Comparison of the Correlations

The correlations mentioned above are compared in Fig. 2.6 at a condition of 1 bar system pressure, 660 °C sphere superheat and 9.53 mm in diameter. The prediction of Shih's correlation is far above the others. Michiyoshi's, Sakurai's and Dhir's correlations give fairly consistent prediction except in the low subcooling case in which Dhir's is about 15% higher than the other two. Farahat's line is higher than the others. The Siviour's correlation is lower than others when the subcooling is large. It is clear that both correlations in ratio form and addition form show a

common trend: Nu increasing with the liquid subcooling approximately linearly. This suggests us that, with the same Nu_{sat} number at saturated condition and with a proper constant for calculating the Nu_{nc} (the natural convective Nusselt number for liquid side), a correlation in addition form (as the one used by Dhir) should give the same prediction as the one in ratio form (as the one used by Michiyoshi) in a special condition. According to this thought, the following correlation in addition law may be constructed to match Michiyoshi's correlation for 1 bar pressure condition.

$$Nu = Nu_{sat} + Nu_{nc}(Sc/Sp)(\mu_l/\mu_v) \quad (2.3.8) ,$$

$$Nu_{sat} = 0.67[Ar/Sp]^{1/4}, \quad Nu_{nc} = 1.45[GrPr]^{1/4}$$

The two correlations are compared at five sphere temperatures in Figure 2.7. It shows that for the sphere temperature higher than 500 °C, the simple correlation in addition form given by Eq. (2.3.8) agrees well with the correlation in ratio form given by Eq. (2.3.6). However, for the sphere temperature lower than 500 °C, the correlation in addition form tends to give higher predictions than the correlation in ratio form.

Moreover, the two correlations are also compared at system pressures of 1, 10 and 50 bars in Figure 2.8. It is clear that the increase in pressure has much more effect on the correlation in ratio form than in addition form. Since the correlation in ratio form has been verified by experiment at high pressure, as claimed by Sakurai (1990b), the correlation in addition form with a fixed constant is only valid at certain pressure (for this case, it is atmospheric pressure).

From the above comparisons, it can be concluded that the correlation in ratio form given by Eq. (2.3.6) or (2.3.7) are likely valid for various of liquids and a wide range of liquid subcooling, surface superheat and system pressure, while the correlation in addition form is only valid in certain conditions

Fig. 2.6 Comparison of pool film boiling correlations at subcooled condition (with superheat of 660 °C at 1 bar pressure).

Fig. 2.7 Comparison of pool film boiling correlations in addition and ratio laws, at five different sphere superheats and 1 bar pressure.

Fig. 2.8 Pressure effect on pool film boiling correlations.

2.4. Saturated Forced Convection Film Boiling

A. Mode 1 Correlation

Bromley, Leroy and Robbers (1953) conducted a forced convection film boiling experiment with electrically heated graphite tubes ($d = 9.8 - 16.2$ mm) in saturated benzene, ethyl alcohol, n-hexane and carbon tetrachloride at atmospheric pressure. The flow velocity ranged from 0 to 4.0 m/sec. They also carried out an analysis which is based on Bernoulli's theorem and a consideration of the viscous drag to the vapor. From the analysis they obtained two dimensionless groups and correlated them according to their experimental data in a linear form

$$h \left[\frac{d^2 \Delta T_{\text{sup}} \mu_v}{U^2 k_v \rho_v \rho_l h'_{fs}} \right]^{1/4} = 0.88 \left[\frac{gd(\rho_l - \rho_v)}{4U^2 \rho_l} + \frac{3d\mu_v h^2}{U k_v \rho_l} \left(\frac{\pi}{\theta'} \right)^2 \right]^{1/4} \quad (2.4.1)$$

At very low flow regime ($Fr^{1/2} < 1.0$), this equation reduces to Eq. (2.2.1') that is the correlation for pool film boiling. At high flow regime ($Fr^{1/2} > 2.0$) the equation become

$$Nu = 2.70 Re_l^{1/2} \{ (\nu_l/\nu_v)/Sp' \}^{1/2} \quad (2.4.2)$$

Cess and Sparrow (1961) did an analysis on film boiling from a horizontal flat plate. According to their analysis, at conditions of $(\mu\rho)_v/(\mu\rho)_l$ less than 0.1 and large Re , the following equation could be obtained

$$Nu = 1.0 Re_l^{1/2} \{ (\nu_l/\nu_v)/Sp' \}^{1/2} \quad (2.4.3)$$

which is the same correlation as that of Bromley's Eq. (2.4.2) except that the constant is smaller.

Witte (1968) did a simple analysis for film boiling on sphere with a given average vapor velocity $U=3/4U_m \sin\theta$, which is the velocity of the potential flow. The theoretical result is

$$Nu = 0.698 Re_l^{1/2} \{ (\nu_l/\nu_v)/Sp' \}^{1/2} \quad (2.4.4)$$

Based on the ratio of his theoretical constants for sphere and tube and the Bromley's experimental correlation for tube given by Eq. (2.4.2), he suggested that a constant of 2.98 should be used for the saturated film boiling on spheres.

Nakayama and Koyama (1986) carried out an analysis for forced convection film boiling on a vertical plate and reached the same result as that of Cess and Sparrow (1961).

B. Mode 2 Correlation

Kobayasi (1965, 1966), following the analysis used by Bromley (1953), did an analysis for forced convection film boiling on sphere and reached two dimensionless groups

$$\Lambda = \frac{4}{3} \left\{ \frac{\mu_v^2 \rho_l}{\mu_l^2 \rho_v} Sp \right\}^{1/2} \frac{Nu_c^2}{Re_l}$$

$$\Psi = 16 \frac{Nu_c^2 \mu_v}{Re_l \mu_l (1 - \cos \theta')^2} + \frac{4}{9} \frac{1}{Re_l^2} \frac{gd^3}{v_l^2}$$

When $Fr^{1/2} > 1.5$, $\Psi \rightarrow 0$, and $\Lambda \rightarrow 0.2066$, so the Nusselt number may be expressed as

$$Nu = 0.393 Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp]^{1/4} \quad (2.4.5) .$$

Where

$$K = \rho_l/\rho_v, \quad R = [(\mu\rho)_v/(\mu\rho)_l]^{1/2}$$

Wilson (1979) did a theoretical study of film boiling on a sphere in forced convection. For the case of small subcooling, he obtained equation for local Nusselt number

$$Nu(\theta) = C(\theta) Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp]^{1/4} \quad (2.4.6) .$$

Epstein and Hauser (1980) applied similarity boundary theory and the perturbation method to model the forced convection film boiling in the stagnation region of a sphere or cylinder. They obtained an explicit solution for the film thickness. By considering two extremes of slight subcooling and large subcooling, they reached a theoretical Nusselt number for saturated film boiling in forced convection

$$Nu = 0.553 Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp]^{1/4} \quad (2.4.7) .$$

However, by comparing with the experimental data from Bromley (1953), Motte (1957) and Dhir (1978) in both small subcooling and high subcooling cases, they suggest a factor 2.04 should be applied to their theoretical equation to predict the film boiling heat transfer. This is definitely too high for the case of saturated film boiling, as shown later, and it is obvious even from the Fig. 2 in Epstein and Hauser (1980).

Ito, Nishikawa and Shigechi (1981) theoretically studied the saturated forced convection film boiling from a cylinder by means of integral method with two-phase boundary layer. Their numerical solution indicated that for $Fr^{1/2} > 2.0$

$$Nu = 0.46 Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp]^{1/4} \quad (2.4.8)$$

The constant is slightly different for different fluids, with 0.46 for water, 0.48 for ethanol and 0.51 for hexane.

Liu, Shiotsu and Sakurai (1992) carried out an approximate analytical solution for saturated film boiling from a horizontal cylinder. They reached a solution in the form of

$$Nu = C \ln(Fr)[Ar/Sp]^{1/4} \quad (2.4.9)$$

Where C is a constant which depends on the assumption of boundary condition, $\ln(Fr)$ is a function of Fr number. Based on their experiment with water and Freon-113 at system pressures from 1 to 5 bar, they obtained a correlation

$$Nu/(1+2/Nu) = H(Fr, d') K(d') [Ar/SP^* Mc]^{1/4} \quad (2.4.10)$$

$$H(Fr, d') = (1 + 0.68 Fr^{2.5/4})^{1/2.5} + 0.45 \tanh\{0.04(d' - 1.3)Fr\}$$

For the case of forced convection, $H(Fr, d') = 0.75 Fr^{1/4}$ and $H \cdot K(4.0) = 0.554 Fr^{1/4}$. Thus, for forced convection with $d' = 4.0$ ($d = 10$ mm), it gives approximately

$$Nu = 0.554 Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp]^{1/4} \quad (2.4.11)$$

In addition to the studies mentioned above, there are several other papers that are related with the experimental study of saturated forced convection film boiling such as

Dhir and Purohit (1978), Aziz, Hewitt and Kenning (1986), Zvirin, Hewitt and Kenning (1990), and Dix and Orozco (1990). All the experiments were done by pass a preheated sphere through a still water pool. In aspect of analysis, Witte and Orozco (1984) did a theoretical study on the effect of vapor velocity profile shape on saturated and subcooled flow film boiling from sphere and cylinder. Their study indicated: a comparison of the analysis to available experimental data shows that heat transfer results based on a quadratic vapor velocity profile compare much better with experiments than those based on linear profile; the heat transfer results for linear and quadratic vapor velocity profiles become virtually identical as subcooling increases.

C. Comparison and Comments

In order to compare the two kinds of correlations, the mode 1 correlation can be rewritten as

$$\begin{aligned} \text{Nu} &= C_1 \text{Re}_1^{1/2} (\mu_l/\mu_v) [R^4 K / \text{Sp}']^{1/4} [\text{KSp}]^{-1/4} \\ &= C_1 \text{Re}_1^{1/2} (\mu_l/\mu_v) [R^4 / \text{Sp}^2]^{1/4} \end{aligned} \quad (2.4.12)$$

Besides using Sp' instead of Sp , the mode 1 correlation has an extra term $[\text{KSp}]^{-1/4}$ in comparison with the mode 2 correlation (such as given by Eq. (2.4.11)). Because the powers on the Re_1 are the same in both kinds of correlation, the velocity dependences are the same in both cases, however, the dependences on temperature are different. The constant C_1 is about 2.7 while the constant C_2 used in the second mode correlations is about 0.5.

Now the question is: which mode is better in taking account of the surface superheat and other effects, such as system pressure and different liquids? Since the mode 2 correlations come out from more rigorous analyses than mode 1 and the mode 2 correlation used by Liu, et al (1990) was claimed to be verified by various liquids at pressures range from 1 to 5 bar, the mode 2 correlation is likely to be the better one. This is also confirmed by our experimental data, as shown later.

2.5 Subcooled Forced Convection Film Boiling

A. Theoretical analysis

Motte and Bromley (1957) did an analysis for subcooled forced convection film boiling by arguing that

$$q = q_v + q_l \quad (2.5.1)$$

where q is the total film boiling heat flux, q_v the net heat flux that flow into the vapor layer and q_l is the heat flux that flow into the liquid stream. From the Bromley's correlation Eq. (2.4.1), they obtained the q_v . They suggested that q_l may be evaluated in three different ways according to the turbulence intensity of flow system. By a universal expression they give a correlation in the form of

$$h \left(\frac{d\Delta T_{\text{sub}}}{Uk_v \rho_v h'_{fs}} \right)^{1/2} - \frac{7.29}{h} \left(\frac{Uk_v \rho_v h'_{fs}}{d\Delta T_{\text{sub}}} \right)^{1/2} = K\Delta T_{\text{sub}} C_{\rho} \rho_l \left(\frac{(\alpha + \alpha_t)}{\Delta T_{\text{sub}} k_v \rho_v h'_{fs}} \right)^{1/2} \quad (2.5.2).$$

Where α_t is the turbulent eddy diffusivity for heat. This correlation has been used by Chou and Witte (1992) to compare their analysis with experimental data.

Cess and Sparrow (1961b), following their analysis for saturated film boiling in Cess and Sparrow (1961a), carried out a boundary-layer analysis for forced convection film boiling on a flat plate in subcooled liquid. For the case of high subcooling, their explicit expression can be rearranged in the form of average Nusselt number.

$$\text{Nu}_x = (2/\sqrt{\pi}) \text{Re}_{l,x}^{1/2} \text{Pr}_l^{1/2} (\mu_l/\mu_v) (\text{Sc}/\text{Sp}) \quad (2.5.3)$$

Wilson (1979) did a theoretical study of film boiling on a sphere in forced convection. He assumed that the liquid velocity field, including the liquid boundary-layer, could be given by a potential flow distribution. Then the integral method was used to the vapor boundary-layer. By applying Pohlhausen-type integral technique, an ordinary differential equation of the film thickness was obtained. For low subcooling

and high subcooling cases, the differential equation can be integrated analytically. In the case of low subcooling the result is given in the form of Eq. (2.4.6). In the case of large subcooling, the solution can be expressed in the form of local Nusselt number

$$Nu(\theta) = C(\theta) Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp) \quad (2.5.4) .$$

As mentioned early, Epstein and Hauser (1980) did an analysis of forced convection film boiling in the stagnation region of a sphere or cylinder and obtained an explicit solution of the film thickness. By considering the two extremes of slightly subcooling and large subcooling, they obtained two simple expressions. The Nusselt number for the saturated case is given in Eq. (2.4.7), and for the large subcooling case is given as

$$Nu = 0.977 Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp) \quad (2.5.5) .$$

Then they simply combined them for all subcooling cases in the following way

$$Nu = \{Nu_{sat}^4 + Nu_{sub}^4\}^{1/4} \quad (2.5.6) .$$

By comparing with the experimental data from Bromley (1953), Motte (1957) and Dhira (1978) in both small subcooling and high subcooling cases, they suggested that a factor 2.04 should be applied to the theoretical equation Eq. (2.5.6) to predict the film boiling heat transfer.

Fodemski and Hall (1982) in their analysis of forced convection film boiling in subcooled liquid applied the *ordinary* forced convection correlation

$$Nu_{fc}(\theta) = \frac{q_{fc}(\theta)}{\Delta T_{sub}} \frac{d}{k_l} = C(\theta) (Re_l Pr_l)^{1/2} \quad (2.5.7) .$$

to calculate the heat flow into the liquid stream. In the case of large subcooling, the film boiling heat flux q will approximately equal the convective heat flux q_{fc} . Eq. (2.5.7) can be rearranged into the form of Eq. (2.5.4).

Shigechi and Ito (1983) extended Ito et al (1981)'s study for saturated film

boiling to subcooled case. The integral method was applied to both liquid boundary-layer and vapor film layer to get differential equations. The numerical solution given by the Fig. 6 in their report shows that, for $Fr > 0.5$ and large subcooling $Sc > 0.05$, the following expression holds for all the three liquids: water, ethanol and hexane.

$$Nu = 1.15 Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp) \quad (2.5.8)$$

All the above theoretical analyses except Motte et al (1957) reached the same expression except the constant, as given by Eq. (2.5.8), for large subcooled forced convection film boiling, and this may also be interpreted as

$$Nu_{fc} = \frac{q_{fc}}{\Delta T_{sub}} \frac{d}{k_l} = C(Re_l Pr_l)^{1/2} \quad (2.5.9)$$

So, it can be concluded that at large subcooling cases, $q_c \approx q_{fc}$, and Eq. (2.5.8) is the best formula for correlating the subcooled forced convection film boiling heat transfer data, as we know right now.

Since all the studies mentioned above have laminar assumption in the analysis, the $1/2$ power on Reynolds number and Prandtl number may only be correct in laminar flow film boiling. If the flow around the sphere is turbulent the power should be higher than $1/2$ as in the case of conventional forced convection.

Additional theoretical work can be found in Fodemski (1985, 1992), Hsiao, Witte and Cox (1975), Orozco, Stellman and Poulikakos (1987), and Walsh (1979).

B. Experimental Studies

Motte and Bromley (1957) conducted a film boiling experiment with electrically heated graphite tube in subcooled ethyl alcohol, benzene, carbon tetrachloride, and hexane. The diameters of the test tube were 9.8, 12.6 and 16.2 mm; the flow velocity ranged from 1.0 to 4.0 m/s; the liquid subcooling was from 0. to 40 °C, and the system pressure was at atmospheric pressure. They also studied the turbulence effect

by putting screens in the upstream of the test section, and they found that this can cause approximately 30% increase in one of their dimensionless group, which is given by the left hand side of Eq. (2.5.2).

Shigechi and Ito (1983) cited the forced convection film boiling data from Nishikawa et al (1978). The experiment was carried out with 16 mm diameter cylinder in water at 5, 10, and 15 °C subcooling and at atmospheric pressure.

For subcooled forced convection film boiling *on cylinders*, those are the only two experiments could be located. On the other hand, there are lots of experiments for film boiling on sphere in subcooled liquids, some of them are summarized in Table 2.5.1.

Table 2.5.1. Film Boiling Experiments with Spheres

Reference	D (mm)/ L (m)	Material	Liquid	Subcooling (°C)	Velocity (m/sec)
Walford 69	6.35/0.3	nickel	water	5-60	0.50-1.80
Jacobson 70	12.7	steel	water	50-80	0.04-0.27
Stevence 71	19.0/0.9	copper	water	40-76	2.90-6.00
Stevence 73	25.4/0.9	silver	water	23-76	1.52
Dhir 1978	19.0/0.8	s. steel	water	0-50	0.02-0.45
Aziz 1986	20.0/0.2	copper	water	0-20	0.01-1.80
Orozco 86	38.1	copper	Freon-11	9-19	1.60-2.32
Dix 1990	38.4/3.0	copper	Freon-113	0-20	0.50-1.90
Zvirin 90	16, 32/1.5	copper	water	0-40	1.60-2.40

All the experiments were conducted at atmospheric pressure. Most of the experiments were done by a cool-down transient through a still water pool in a very short time interval, except Jacobson and Shair's (1970) and Orozco and Witte's (1986) (will be discussed latter). The maximum sphere traveling length is given in the table by L. The sphere traveling time in the liquid can be calculated from L/U which is much less than 0.5 sec for most of the high speed cases. In such a short time, the thermal response of the sphere and the entrance effect will limit the accuracy of the experimental data. This implies that the accuracy of all the large velocity data obtained from above experiments are questionable.

On the other hand, with low speed (lower than 0.5 m/sec) the obtained data may

be reasonably correct, but they are not in the forced convection regime. Dhir and Purohit (1978)'s experiment is an example of a such case. They suggested a correlation

$$Nu = Nu_{p,s} + 0.8Re_l^{1/2} \{1 + (\mu_l/\mu_v)(Sc/Sp)\} \quad (2.5.10)$$

which is only valid in the regime of $0.046 < Fr^{1/2} < 1.03$ ($1200 < Re_l < 19000$). Where $Nu_{p,s} = 0.8 (Ar/Sp)^{1/4}$ is the Nusselt number at saturated pool film condition.

Jacobson and Shair (1970) conducted an experiment with continually heating the test sphere by induction. The sphere was supported by the liquid flow with a Lavit disk in the front of the sphere. The temperature of the sphere was determined with a Pyro microoptical pyrometer. Although the experimental techniques are quite interesting, the test flow velocity is too low to be interesting.

Orozco and Witte (1986) heated their test sphere by circulating Dowtherm G through a passage in a hemisphere sphere. Although this technique avoids the short transient problem, it may be difficult to account for the heat loss at the back side of the sphere. From their experiment, a correlation was suggested in the form of

$$Nu = 18.73 (\rho_v/\rho_l) [Re_l Sc Pr_l^2]^{5/8} Sp^{-3/2} \quad (2.5.11)$$

Dix and Orozco (1990) carried out a study on local heat transfer distribution on a sphere surface. They concluded that in the stable pool film boiling, the heat flux were relatively uniform as a function of angular position along the surface of the sphere. In forced convection film boiling, significant heat flux were recorded in the vapor wake region of the sphere; and these heat flux varied from 0.3 to 1.5 times of the heat flux at the front stagnation region. The heat flux in rear wake region were most noticeably affected by detachment of the vapor wake because of the subcooling. But the authors did not pay attention to and discuss the effects of the big support tube both on the observation and on local heat transfer result in the rear part of the sphere.

C. Comments and Conclusion

There are no good experiments for film boiling on sphere in subcooled forced

convection regime (with high Fr number) because the cool-down transient technique used by most of the experiments is not suitable for high speed test.

The equations in the form of $Nu = C Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp)$ is likely to be the best way to correlate large subcooled film boiling in forced convection regime. But the $1/2$ power on Re and Pr may not be good for turbulent flow film boiling. How to combine the saturated and large subcooled correlations to express the small subcooled film boiling is a very practical issue and it is addressed in this work.

2.6 Summary of The Previous Studies and The Goals of the Present Work

In conclusion, regarding the previous work on film boiling from spheres, the experimental studies are summarized in Table 2.6.1 and the theoretical studies are summarized in Table 2.6.2. According to the experimental and theoretical studies, the best heat transfer correlations for film boiling on spheres in single-phase flows are listed in Table 2.6.3.

Regarding previous work on film boiling from spheres, it can be concluded that: For two-phase Flows: No experiment or analysis for sphere film boiling in two-phase flows can be located. For Single-Phase Flows: (1) All the meaningful single-phase film boiling heat transfer data were obtained through short cool-down transients (by passing a preheated sphere through a liquid tank). Although the short cool-down transient technique is good enough for pool or low speed forced convection film boiling experiments, it is not adequate for high velocity forced convection film boiling, because the transient is too short. For example, with sphere speed at 2.0 m/s and pool length 0.5 m, the film boiling cool-down transient is only about 0.25 second! Within such a short time, the entrance thermal response may affect the accuracy of the experimental data. (2) The accuracy of the single-phase data may be very poor as a result of the large heat loss from the support-tube, since the sphere-support-tube diameter to sphere diameter ratios of the previous experimental work are

about 0.12 to 0.3. (3) For each single-phase film boiling regime, the basic form of heat transfer correlation has been well developed, but there are some uncertainties in the correlation constants, as shown in Table 2.6.3. (4) There is no general correlation or complete theoretical analysis for film boiling on spheres which covers all the single-phase flow regimes. There is no theoretical model for the heat transfer in the separated rear region of the sphere.

The main purpose of this work is to establish the experimental data base and to derive the heat transfer correlation (closure law) for film boiling on spheres. The emphasis is on very high temperature film boiling (pure film boiling with the sphere temperature higher than the quenching point) in all the flow regimes, which cover from natural convection (pool film boiling) to forced convection, from saturated to highly subcooled, and from single-phase flow to two-phase flow. Besides this, theoretical analysis is carried out to understand the fundamentals of film boiling and to formulate better heat transfer correlations. The key techniques and scopes of the present experimental work are listed in Table 2.6.1 and the basic models used in the present theoretical analysis are shown in Table 2.6.2.

The goals of the present study are:

- Build a robust and flexible experimental system for sphere film boiling experiment in both single- and two- phase flows.
- Obtain high quality single-phase film boiling heat transfer data, clarify the uncertainties and construct a general correlation.
- Develop a theoretical model to predict the film boiling heat transfer in single-phase flows with an emphasis on the modeling of the separated rear region.
- Establish data base for sphere film boiling in two-phase flows and formulate heat transfer correlations.
- Study the influence of multi-sphere array structure on the sphere film boiling heat transfer in single- and two- phase flows.

In the following chapters each individual aspect of these goals will be presented in detail respectively.

Table 2.6.1 Experimental Studies of Film Boiling On Spheres

	Previous Studies		Present Work
Single - Phase	Saturated Pool	Bradfield (1967) Dhir (1978) Toda (1982) Aziz (1986) Farahat (1972) El-Wakil (1981)	<ul style="list-style-type: none"> • Induction heating • Testing in a flow loop • Whole transient cooldown from 900 C to quench • Steady state mode operation • Sphere directly supported by thermocouples (less heat loss) • Experimental Ranges: Velocity: 0 -- 2.3 m/s; $Re < 2.10^5$ $Pr < 80$ Subcooling: 0 -- 40 C Superheat: 900 C to quench Diameter: 6 -- 19 mm Sphere material: stainless steel or brass
	Subcooled Pool	Stick a pre-heated sphere into a liquid Tank	
	Saturated Forced Convection	Pass a pre-heated sphere through a liquid tank	
	Subcooled Forced Convection	Induction Heating Internal Convective Heating	
Two-Phase	Bubbly Two - Phase	None	<ul style="list-style-type: none"> • Two-phase mixer • X-ray radiography for void fraction measurement • Experimental Ranges: Upward two-phase flow: Water Velocity: 0.6 -- 3.2 m/s Steam Velocity: 3.0 -- 9.0 m/s Void Fraction: 0.2 -- 0.65 Downward two-phase flow: Water Velocity: 1.5 -- 6.5 m/s Steam Velocity: 1.1 -- 9.0 m/s Void Fraction: 0.7 -- 0.95
	Mist-Jet Two - Phase	None	

Table 2.6.2 Theoretical Studies Of Film Boiling on Spheres In Single- Phase Flows

	Previous Studies			Present Work
	Reference	Subcooling	Method	
Pool	Frederking (1963)	Saturated	Laminar Boundary Layer, Integral Method	Two-Phase Laminar Boundary Layers Model Integral Method
	Hendick (1969) Shih (1981) Dhir (1978) Michiyoshi (1988)	Subcooled		
	Kobayasi (1965)	Saturated	Bernoulli's Theorem	
	Witte (1968)	Subcooled	Uniform Film Vapor Velocity	
Forced Convection	Wilson (1979)	Subcooled	Similarity	Unseparated Region: Two-Phase Laminar Boundary Layers Model Separated Region: Turbulent Eddy Model Integral Method
	Epstein (1980)	Subcooled	Stagnation Point Perturbation	
	White (1984)	Subcooled	Velocity Distribution Effect	

Table 2.6.3 Heat Transfer Correlations For Film Boiling on Spheres in Single-Phase Flows

Regime	Basic Correlation	Constant	References
Saturated Pool	$Nu = C (Ar/Sp')^{1/4}$	0.62 0.586 0.8 C(d)	T: Bromley (1950) S: Frederking (1963) S: Dhir (1978) T: Bareen (1962)
Subcooled Pool	$Nu = C (Ar/Sp')^{1/4} M_c^{1/4}$	0.696 0.696 K(d)	S: Michiyoshi (1988) S: Tso (1990) T: Sakurai (1990)
Saturated Forced Convection	$Nu = C Re_l^{1/2} \frac{\mu_l}{\mu_s} \left(\frac{KR^4}{Sp'} \right)^{1/4}$	0.393 0.553 0.46	S: Kobayasi (1966) S: Epstein (1980) T: Ito (1981)
Subcooled Forced Convection	$Nu = C Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_s)(Sc/Sp)$	0.977/2.0 1.15 1.13	S: Epstein (1980) T: Shigechi (1983) P: Cess (1961b)
S -- Sphere; T -- Cylinder; P -- Plate; Ar = $g(\rho_l - \rho_v)d^3/(\rho_v \nu^2)$; Mc is given by Eq. (2.3.6)			

Chapter 3

Experimental Approaches and Techniques

In order to get film boiling heat transfer data in well defined flow conditions (either two-phase or single-phase), the flow inside the test section must reach a steady state before the heat transfer data are collected. This requires an adequate and well-controlled heat supply to sustain a high temperature (over ~ 1000 °C) film boiling in a steady-state flow. This is a very significant experimental challenge, since all the conventional heating techniques do not work in this case. For instance, the conventional internal electrical heating cannot work at such a high temperature; the surface heating by electric current is not possible for spherical geometry; and the internal convective heating (circulating hot liquid into a passage inside the test sphere) is impossible in high temperature cases. Moreover, all these techniques require either insulated wires for electric power supply or channels for heating fluid and they unavoidably cause large amounts of heat loss and disturbance on the flow field. After extensive experimentation with various alternatives, the radio frequency induction heating method was selected for our experiment. Eventually, it became possible to heat the sphere and to sustain film boiling at a temperature over 1000 °C in any desired flow conditions. Besides the cool-down transient technique, the heat flux can also be obtained through a steady state operation based on the induction power coupling calibration.

Other major difficulties are the generation and characterization of the two-phase flows in various two-phase flow regimes. A well designed two-phase mixer was used to generate the two-phase flows. An X-ray radiography was applied to measure the void fraction in the test section for two-phase runs.

3.1 Flow Loop

The flow loop is schematically illustrated in Figure 3.1. The water loop is built up with copper tube of 38 mm (1.5 in.) I.D. The steam loop is built up with copper tube of 25.4 mm (1.0 in.) I.D. The arrangement of the mixer/test-section assembly is reversible so that they can be connected to the "top separator" or to the "water tank", to generate upward flow and downward flow respectively. The up-flow geometry is necessary for single-phase flow and low void fraction bubbly two-phase flow, while the down-flow geometry is more suitable for high void fraction two-phase flow. However, for very high liquid velocity two-phase flows, either upward or downward geometry could be used.

Water is recirculated with a pump and thermostatically maintained to a fixed temperature. The steam is supplied from lab lines, filtered, metered and used in a once-through fashion, i.e., vented into the atmosphere or to a condenser after separation. All loop components are externally insulated except the test section.

All flow measurements are made with venturis. The pressure differences given by the venturis are read both by mercury U-tubes and by Validyne pressure transducers which are interfaced with a PC computer through a data acquisition system. Venturis of different sizes can be easily switched into the loop for different ranges of flow rate. The water flow rate measurement is calibrated by volumetric method, providing a measurement accuracy of about $\pm 4\%$. The steam flow rate measurement is calibrated and checked by calorimetric method, which confirms a measurement accuracy of about $\pm 5\%$.

In the present arrangement, the loop system is designed to be operated at atmospheric pressure. However, with modifications of the water tank and test section, the whole loop can be operated at system pressure of 10 bar. The maximum water flow rate is 50 GPM, which provides superficial water flow velocity of 2 m/s in the test section. The peak steam supply rate is 12 gram/sec, which gives an equivalent superficial steam velocity of 20 m/s in the test section.

Fig. 3.1 Schematic of the flow loop.

Fig. 3.2 Schematic illustration of the two-phase mixer. The tubes (69 total) are on a square grid of 3.81 mm center to center.

3.2 Two-Phase Mixer

The two-phase mixer is schematically illustrated in Figure 3.2. The tube needles are made of brass. On each tube needle assembly, there are a total of 69 tubes on a square grid of 3.81 mm center to center. Three tube-needle assemblies are used in the experiment. The sizes of the tubes and the ratios of tube-hole opening area to base area of these assemblies are as follows:

No. 1 tube-needle assembly:	1.37 mm I.D.;	1.83 mm O.D.;	10.2% ,
No. 2 tube-needle assembly:	1.80 mm I.D.;	2.41 mm O.D.;	17.5% ,
No. 3 tube-needle assembly:	2.31 mm I.D.;	2.77 mm O.D.;	28.9% .

With opening and closing the valves, the loop can supply either water or steam into each of two chambers in the mixer. The mixing is achieved by injecting or flowing one phase of the flow into another through the tube needles.

To generate an upward bubbly two-phase flow, the steam is supplied to the needle chamber and introduced into the water flow through the tube needles. On the other hand, to generate high void fraction droplet or jet flow, the water drop or jet is introduced into the steam flow through the tube-needles, and the downward flow arrangement is preferred. To generate a high speed mist/jet flow, either of the above arrangements may be used.

3.3 Test Section and Test Spheres

The test section, as shown in Figure 3.1, consists of two pieces of Pyrex tube and a middle flange which serves as the sphere support. The circular test section is 38 mm in diameter and 330 mm in length. The induction copper-tube coil is tightly wound on the outer side of the test section (which is 47 mm O.D.) and hooked up to a RF induction power supply.

Fig. 3.3 Schematic illustration of the test sphere support flange.

Spheres: Stainless Steel or Brass,
6.35, 9.53, 12.7, or 19.1 mm O.D.
Thermocouples: Stainless Sheathed, K Type,
0.25, 0.50, or 0.81 mm O.D.

Fig. 3.4 Schematic of the instrumented test sphere.

The test sphere is supported from the middle flange, as shown in Figure 3.3, with its distance from the entrance of the test section usually being 170 mm (or 4.3 diameters). Stainless steel and brass balls/spheres of 6.35, 9.53, 12.7, 19.1 mm O.D are used in the experiments. The spheres are drilled, as shown in Figure 3.4, to accept thermocouples (stainless steel sheathed, K type) at three positions: the forward stagnation point, the center, and the rear stagnation point --- these are referred to as positions 1, 2 and 3, respectively. The sphere is directly supported by these sheathed thermocouples (in some tests only one is used to minimize the heat loss from the thermocouples). Approximately 15 mm from the sphere, as shown in Figure 4.3, the thermocouple stems are reinforced by a stainless steel tube of 1.53 mm O.D., which is fixed to the end of a steel duct-beam. At the other end of the duct-beam, four strain gauges are installed on the surface inside the duct to measure the force/drag on the sphere. The duct keeps the strain gauges away from water and steam and also serves as a passage for the thermocouples to pass through. The sphere itself and the thermocouple sheaths provide an adequate shielding from the RF electromagnetic field to allow undisturbed thermocouple signals, even with the induction heating power on.

The reason for supporting the sphere directly by thermocouple is to minimize the heat loss from the support. In our experiments, 0.25 mm O.D. sheathed thermocouples are used for 6.35 mm O.D. spheres; 0.5 mm sheathed thermocouples for 9.53 mm spheres; and 0.81 mm sheathed thermocouples for 12.7 and 19.1 mm spheres. With these selections, the thermocouple to sphere diameter-ratios are less than 0.06 (the corresponding area-ratios are less than 1 %). In contrast, Dhir and Purohit's (1978) diameter-ratios are about 0.15; Aziz, Hewitt and Kenning's (1986) ratios are 0.3 and 0.15; Dix and Orozco's (1990) ratio is 0.25. Thus, the heat loss from the supports is significantly minimized in our experiments. Our experiment also indicates that, in the case 9.53 mm O.D. sphere, using three 0.5 mm O.D. thermocouples instead of only one will cause about a 20% increase in the film boiling heat transfer with all the other conditions being the same. Actually, the mechanism of this increasing is quite complicated because the adding of two more thermocouples not only triples the heat transfer surface but also helps to drain more water to the surface on the back of the sphere. This means the increase of heat loss is more than twice. This will be discussed at some depth later on.

The arrangement of the test-section/support-flange assembly is also reversible such that the sphere support tube can be located either down-stream of the sphere (as shown in Figure 3.1), which is more suitable for heat transfer tests, or up-stream of the sphere, which is more suitable for film configuration observation (because the disturbance of the support to the vapor film on the rear part of the sphere can be avoided). The distance of the test sphere to the entrance of the test section can be adjusted by changing the lengths of the two Pyrex tubes if necessary.

4.4 Induction Heating and Power Calibration

The test section (47 mm O.D.) is wound by 10 turns of 4.5 mm O.D. copper tube coil with the test sphere located at the center of it, as shown in Figure 3.1. The coil is hooked up to an RF induction power supply unit (RFC 3160) which is able to supply a peak of 20 kW induction power at frequency of 250~800 kHz.

The induction power coupling (the heating power obtained by the sphere) not only depends on the relative position of the sphere to the coil but also depends on the physical properties of the sphere and the medium inside the coil. The more ferric the metallic material, the better the coupling. The electromagnetic properties of the sphere may change with temperature and thus the power coupling is also temperature dependent. The medium inside and around the coil influences the intensity of the electromagnetic field, so it also affects the power coupling. If the operating condition of the induction power supply and the medium inside/around the sphere are fixed, the power coupling then only depends on the induction power level (monitored by the plate current) and the sphere temperature.

The main mode of operation, in our experiments, is to follow the transient cool-down of the test sphere from a well established steady-state. The instantaneous heat fluxes can be deduced from the energy balance on the sphere and the quantitative knowledge of induction power coupling is not needed in this case. In this manner, from a single cool-down transient with the sphere temperatures transients and the information of the heat capacity of the sphere, we can obtain a large number of data pairs, heat fluxes at respective sphere temperatures, between the initial and the final

(just before quenching) states.

Besides the transient mode operation, it is also interesting to obtain data at steady-state (without the cool-down transient), which is uniquely possible in the present experimental set-up. In particular, this experimental technique may be used to enhance the robustness of the data base generated by the transient mode runs. However, this method requires the information of the power coupled to the sphere during the steady-state run, which is referred to as induction power calibration, and we discovered that it is not as straightforward as we initially anticipated. Specifically, in the early stages of the work, we didn't know how the presence of water as the medium in the test-section would affect the power coupling, and how we would tackle and quantify the influence of water. After lots of experimentation, eventually, we found out that compared with the case with air as medium, the presence of pure water (distilled or deionized) as medium in the test section decreases the power coupling; and the electric conductivity of the water affects the power coupling significantly. More interesting and useful discovery was that the influence of the electric conductivity of water on the power coupling diminished essentially when the conductivity is over ~ 200 micromho, and beyond this point the power coupling is the same as that with air as the medium. This means that the power calibration obtained in air could be used to determine the power coupling when water or steam is present in the test section during the steady-state mode operation, as long as the water electrical conductivity is higher than 200 micromho. This will be demonstrated latter.

The power calibration in air was carried out as follows. With the test sphere in its usual position in the test section, the power was turned on and maintained at a given level until the sphere reached a temperature of ~ 1000 °C, at which the power was triggered off. This process was repeated at various power levels indicated by the "plate current" of the induction power supply (the plate current is also read by the PC computer). A typical temperature transient is shown in Figure 3.5. From these temperature data, which were recorded on the PC computer at sampling rate of 10 Hz, the power coupled to the sphere could be obtained as a function of the sphere temperature as shown in Figure 3.6 (actually it shows the normalized power with respect to the sphere surface area). This involves getting the temperature gradient

(carried out with a "local" sliding curve-fit to 5 consecutive readings) and using the energy balance equation,

$$q = \frac{m_s C_p(T_w)}{\pi D^2} \frac{dT_w}{dt} + h(T_w - T_a) + \epsilon \sigma (T_w^4 - T_a^4) \quad (3.1)$$

Fig. 3.5 A typical temperature transient during a calibration run in air. The insert shows a magnified portion.

In this equation, the heat capacity of the sphere material (stainless steel or brass) as a function of temperature is from Touloukian and Buyco (1970). The heat transfer coefficient, which accounts for the natural convection heat loss from the sphere to the air, was estimated by the correlation from Yuge (1960). The corrected or the true power obtained from Eq. (3.1) is also shown in Figure 3.6. The emissivity of radiation can be obtained by matching the corrected power to zero for the whole cool-down period; thus obtained emissivities for new and "aged" stainless steel spheres are 0.5 and 0.8 respectively. From these results, a composite of calibration curves could be assembled as shown in Figure 3.7. So, for a given power setting (plate current) and sphere temperature, the coupling power could be determined.

Fig. 3.6 The reduction of data in Figure 3.5, as described in the text.

Fig. 3.7 Calibration (in air) of the power delivered to the sphere with its temperature as the parameter.

3.5 X-Ray Radiography For Void Fraction Measurement

X-ray radiograph is a photographic record produced by the passage of X-rays through an object onto a film. In our application, the X-rays produced by an X-ray tube pass through the test tube and then expose the film behind it, as illustrated in Figure 3.8. Since the water has a much higher attenuation factor than the air and steam, the attenuation of the X-rays and thus the lightness of image on the film depends on the water fraction (or void fraction) inside the test tube. So from the analysis of the lightness of the X-ray radiograph, the average void fraction inside the test tube can be determined.

To obtain a quantitative measurement of void fraction, another glass tube called a calibration tube, which is the same as the test tube was set beside the test tube. A calibration piece, as shown in Figure 3.10, was set inside the calibration tube. The calibration piece, which is made from a stepped aluminum piece with empty holes inside, provides various voids inside the calibration tube, as illustrated in Figure 3.9.

Two typical X-ray radiographs obtained from downward and upward two-phase flows are shown in Figure 3.11 (a) and (b) respectively. By comparing the lightness near the center line of the test tube with the lightness of the holes in the calibration piece on the same film, the average void fraction in the test tube could be obtained. Radiographing the calibration tube and the test tube on the same film by the same exposure not only avoids the error caused by the intensity differences of different X-ray exposures, but also eliminates the errors from the film development processing.

Another nice aspect of this radiograph technique is that with selected film (Kodak Industrix AA film), optimized X-ray generator voltage, and optimized distance from X-ray tube to target, the lightness of the X-ray radiograph is almost linear to the void fraction inside the tube, as indicated by Figure 3.12. This allow us to easily to extrapolate the void fraction that falls between two adjacent samples provided by the calibration piece.

The radiograph can be made either "off line" (without boiling sphere in the test tube) to determine the void fraction of coming two-phase flow at certain flow

conditions, or "on line" (with boiling sphere in the test tube) to have an overall radiograph of the sphere film boiling in a two-phase flow. The results will be presented and discussed in Chapter 7.

Fig. 3.8 Schematic diagram showing the fundamentals of a X-ray radiograph void fraction measurement technique.

Fig. 3.9 Schematic illustration of void made by the calibration piece in the calibration tube.

Fig. 3.10 A photograph of the void fraction calibration piece.

(a)

(b)

Fig. 3.11 X-ray radiographs obtained with (a) downward two-phase flow and (b) upward two phase flow

Fig. 3.12 The correspondent radiograph lightness of the voids generated by the calibration piece.

3.6 Experiment Procedures and Data Reduction

For all the experiments, the film boiling is established by the following procedures: depleting the water from the test section, heating up the sphere to a temperature over the quenching point, turning on the water flow and then adjusting the power level to let the film boiling stabilized at a certain flow condition with a desired sphere temperature.

In a transient mode operation, after the flow reaches steady state and the sphere temperature stabilizes, the induction power is triggered off and the temperature transients are recorded by the PC computer at a sampling rate of 10 Hz. A typical temperature transient is shown in Figure 3.13. The total heat flux is obtained from

$$q_t = \frac{m_s C_p (T_w)}{\pi D^2} \frac{dT_w}{dt} \quad (3.2) ,$$

which is also illustrated in Figure 3.13.

In a steady-state mode operation, the power level is adjusted from one to another (either increase, decrease, or randomly). At each level, the temperature (or film boiling) is allowed to stabilize, as shown in Figure 3.14. The total heat flux (including both convective and radiative contribution) at each steady state, as shown by the plateau in Figure 3.14, can be obtained from the respective plate currents and the sphere average temperature in conjunction with the power calibration that is shown in Figure 3.7.

As usually treated in the literature, the total heat flux is split into convective and radiative contributions. The radiation from the sphere is to be largely absorbed in a thin layer of the liquid and hence will help to produce vapor and thus affect the convective contribution, so a radiation-factor J is usually used when radiation contribution is subtracted out from the total heat flux. Thus the convective heat flux is given by

$$q_c = q_t - J q_r \quad (3.3) ,$$

where $q_r = \epsilon \sigma [T_w^4 - T_s^4]$ and $J = 7/8$ which comes from Bromley, Leroy and Robbers (1953), as discussed in Section 2.1.

Based on the convective heat flux, all the data were reduced to heat transfer coefficients, usually presented in terms of Nusselt numbers. All the properties are evaluated at the film temperatures, which are $(T_w + T_s)/2$ and $(T_s + T_l)/2$ for vapor and liquid respectively.

Fig. 3.13 Measured sphere temperature transients, and implied heat flux transient, in a transient mode run.

Fig. 3.14 Temperatures recorded in a run made in the steady-state mode operation. Each plateau provides a steady-state measurement.

3.7 A Consideration for the Non uniform Temperature Distribution within the Sphere.

As it is known, during the film boiling cool-down transient, neither the heat flux nor the temperature are uniform on the surface of the sphere or within the sphere. For instance, at the front stagnation point, the heat flux is higher and the temperature is lower than any other part of the surface; the center temperature is higher than the temperatures on the surface. However, in most cases the maximum temperature difference within the sphere is much smaller compared to the average sphere superheat, especially when the superheat is high and thermal conductivity of the sphere is large. For example, for a transient cool-down of a stainless steel sphere (which has a very low thermal conductivity) in a saturated single-phase film boiling, the temperature differences between the front stagnation point and the center are only about 50 °C, which is less than 10 % of the superheat. In the case of a brass sphere, the difference is about 10 times smaller than this. So in both the above cases, the temperature distribution inside the sphere and on the surface can be assumed to be uniform and the temperature measured at any location within the sphere could be used as the characteristic sphere temperature.

Based on the discussion above and the concern to avoid too much heat loss from more thermocouples, in most of our single-phase experiments for obtaining the heat transfer data, only one thermocouple was used to support and measure the temperature of the sphere. The tip of the thermocouple is located on the center line and 1/4 diameter from the front stagnation point.

But this isothermal assumption is not good in some extreme cases when the liquid-phase velocity is high and the subcooling is large, especially when the sphere thermal conductivity is low (as in the case of stainless steel). According to our rough analysis for the cool-down transient of stainless steel spheres in a single-phase forced convection with velocity of 2 m/s and 30 °C subcooling, the temperature difference between the surface and the center could be about 20% of the sphere average superheat. This is also confirmed by our experimental measurement with three

thermocouples. However even in these cases, the temperature at $1/4$ diameter from the front stagnation point can also be regarded as the sphere average temperature because the temperature at the front stagnation point is very low and all the back half of the sphere remained at relative high temperatures, which are equal or even higher than the temperature of the center of the sphere.

Chapter 4

Film Boiling in Single-Phase Flows

Experiments on film boiling from spheres in single-phase water flows have been conducted systematically to observe the vapor film configurations and to investigate and check the effects of water flow velocity, water subcooling, sphere superheat, sphere diameter, and sphere material on the film boiling heat transfer. The ranges of the experimental parameters are:

- Water Velocity: 0.0 -- 2.3 m/s; $Re \leq 2 \cdot 10^5$; $Fr \leq 80$
- Water Subcooling: 0.0 -- 40.0 °C
- Sphere Superheat: 900 °C -- Quench
- Sphere Diameter: 6.0 -- 19.0 mm

The flow in the test section was arranged upward. Stainless steel spheres with outer diameter of 6.35, 9.53, 12.7 and 19.1 mm and brass spheres with outer diameter of 9.53 and 12.7 mm were used in the experiments. In order to minimize the heat loss, in most of the tests, only one sheathed thermocouple was installed in the sphere with its tip located at 1/4 diameter from the front stagnation point. The experiments were conducted with distilled water, de ionized water and tap water. Also, tests were made with the sphere at different stages of oxidation: essentially metallic, slightly oxidized but still mostly metallic in appearance, well oxidized (dark, but smooth), and heavily oxidized and corroded. In each case, the emissivity was measured by the power-calibration procedure described in Chapter 3 and was found to gradually progress from 0.5-0.6 to 0.8-0.9 through these four stages of oxidation. Most of the data were obtained from transient cool-down operations; however, steady-state operations were also carried out to obtain the heat transfer data at saturated conditions.

In this chapter, the observations of vapor film configurations and heat transfer

results of film boiling on a 12.7 mm diameter stainless steel sphere are presented first. Then, the heat transfer data are presented in several different ways according to the correlations discussed in Chapter 2 and the fitness of these correlations is discussed; the diameter and material effects are checked and correlated. Finally, the results obtained by steady state mode operation are presented and compared.

4.1 Observations of Film Configurations of Film Boiling

One of the unique features of the present experimental approach is that it provides a convenient way to observe the film boiling process clearly, because the sphere is held still during the test. In the cool-down transient mode operation, after the induction power is triggered off, the induction coil can be moved away so the whole film boiling cool-down transient can be observed.

In order to get rid of the support wire's influence on the rear part of the film, in the runs for observation purpose, the sphere can be held by a small sheathed thermocouple from the front stagnation point. In this way, our experiment provided very unique and realistic observations of the film boiling on the sphere. The observations were videotaped for runs at wide ranges of flow velocity (0 to 2 m/sec), liquid subcooling (0 to 40 °C), sphere superheat (200 to 900 °C) and diameter of the spheres (6.35, 9.53, 12.7, and 19.0 mm). Normal camera pictures were also taken at a shutter speed of 1/2000 second in all typical flow conditions. Typical pictures that were taken from the film boiling on a stainless steel sphere (12.7 mm in diameter) supported with one sheathed thermocouple (0.81 mm O.D.) are shown and discussed as follows.

Pool film boiling in saturated water: As shown in Figure 4.1, the test spheres were supported from the up-stream (front stagnation point) in the first three cases (a,b,c), and were held from the down-stream (rear stagnation point) in the last three cases (d,e,f). The whole vapor surface is quite wavy and the induced free convection is turbulent, especially when the sphere temperature is high, as shown in Figure 4.1 (a) and (d). However, as the sphere temperature decreases, the vapor surface at the front part of the sphere tends to be smoother, especially when the

diameter is small, as shown in Figure 4.1 (c) and (e). On the rear part of the sphere, there is always a periodic generation and detachment of vapor dome, which makes a two-phase wake behind the sphere.

Pool film boiling in subcooled water: As shown in Figure 4.2, for subcooled pool film boiling, the vapor film is smooth on the front part of the sphere. There is a small vapor dome at the back of the sphere. Small bubbles periodically detach from the top of the dome. The frequency of the bubble detachment depends on the sphere temperature and the subcooling of the water. When the sphere temperature is high, the detachment frequency is fast. When the sphere temperature is low, the detachment frequency is low or even zero (with no vapor escaping). Figure 4.2 (f) was taken during the quenching.

Forced convection film boiling in saturated water flow: In forced convection saturated film boiling, as shown in Figure 4.3 (a,b,c), a large and long "perfect" vapor wake can be seen regardless of what the sphere temperature is. The higher the sphere temperature and the faster the flow, the longer is the vapor wake, which may be as long as 10 times the sphere diameter. Big vapor slugs always periodically detach from the tail of the wake. Generally, the film on the front part of the sphere is quite wavy, as in the case of saturated pool film boiling. It is very interesting that a little bit of water subcooling (just about 0.5°C) can destroy the "perfection" of the wake behind the sphere, especially when the sphere temperature is low. In this case, the tail of the wake is torn apart and some water is dripped and bounces on the back surface of the sphere in the wake, as shown in Figure 4.3 (d) and (e). This also happens when the sphere is supported from the up-stream and it is worse when the support is at the rear stagnation point. This may be part of the reason why only one degree subcooling could cause about 30~40 % increase in the overall film boiling heat transfer from a sphere that is supported from the rear stagnation point.

Forced convection film boiling in subcooled water flow: In this regime, the vapor wake at the back of the sphere disappears and a wavy vapor film is formed on the back of the sphere, as shown in Figure 4.4. The film on the front part of the sphere is always smooth, as in the case of subcooled pool film boiling. In the case of moderate water velocity, as shown in Figure 4.4 (a, b, d and e), there is almost no

vapor escaping from the vapor film and the surface of the back film is very wavy in the equator region. When the water velocity becomes large, the occasional escape of vapor bubbles from the vapor film near the upper-equator region can be observed, but the bubbles disappear in the subcooled liquid quickly.

In the subcooled cases, either from up-stream or from down-stream, the support (the thermocouple) makes no difference on the observation. However in the saturated cases, there are some differences between the film configurations that were obtained by up-stream support and by down-stream support, although the differences are not very significant. The best way of supporting is up-stream, if the observation is the only concern.

The observed phenomena for typical film boiling regimes are summarily sketched in Figure 4.5. In general, for saturated film boiling, the front vapor film is wavy and on the back of the sphere there is large vapor wake in forced convection and a periodic detachment of vapor dome in natural convection. For subcooled film boiling, the front film is always smooth, the back film is wavy and very few or no vapor bubbles can be observed in the water.

Most of the above observations are new and could provide good bases for theoretical analysis in helping us to understand the mechanism of film boiling in different film boiling regimes.

Fig. 4.1 Pool film boiling in saturated water

Fig. 4.2 Pool Film Boiling in 10°C subcooled water

Fig. 4.3 Forced convection film boiling in upward saturated water flow

Fig. 4.4 Forced convection film boiling in upward 10°C subcooled water flow

Saturated Pool Film Boiling

Subcooled Pool Film Boiling

completely saturated

~0.5 C subcooling

Saturated Film Boiling in Forced Convection

at moderate velocity

at high velocity

Subcooled Film Boiling in Forced Convection

Fig. 4.5 Sketches of typical film boiling configurations.
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4.2 Film Boiling in Saturated Conditions

The total heat flux of one typical series of transient mode runs (with 316 stainless steel spheres of 12.7 mm in diameter) are shown in Figure 4.6. The influences of temperature and flow velocity on the total heat flux are very clear. The quench temperatures are about 250 °C for pool film boiling and increase gradually with the water flow velocity in the forced convection regime. More data are tabled in Appendix C.

The same data are plotted in terms of Nu vs. Re in Figure 4.7, where $Nu = q_c D / (\Delta T_{sup} / k_v)$, $q_c = q_t - q_r$, $Re = UD / \nu_l$, and the k_v is evaluated at the film temperature $(T_s + 0.5\Delta T_{sup})$. It is obvious that: (1) the Nu is nearly independent of Re , when Re is less than 20000 ($Fr^{1/2} = 1.5$, pool film boiling regime); (2) Nu increases with Re , when Re is larger than 20000 (forced convection film boiling regime). With constant flow rates, the Nu decreases significantly when the sphere temperature increases.

Fig. 4.6 Total heat flux transients from a typical series of transient mode runs in saturated experiment.

Fig. 4.7 Saturated film boiling data plotted in the form of Nu vs. Re.

Pool Film Boiling Interpretation

Basically, there are two ways to correlate the heat transfer data for pool film boiling in saturated liquid, which are regarded (by us) as 1/4-power law and 1/3-power law:

$$\text{1/4-Power Law: } Nu = C_{\text{sat},N} \{Ar/Sp'\}^{1/4} \quad (4.2.1)$$

$$\text{1/3-Power Law: } Nu = C_{\text{sat},N} \{Ar/Sp'\}^{1/3} \quad (4.2.2)$$

The data are plotted in terms of the 1/4-power law in Figure 4.8. For $Fr^{1/2} < 1.5$, the heat transfer characteristic $Nu/(Ar/Sp')^{1/4}$ ranges from 0.59 to 0.70, which approximates the Bromley's (1961) experimental constant of 0.62, Frederking and Clark's (1963) theoretical constant 0.586, and Dhir's (1978) experimental constant 0.8. In Figure 4.9, the data are presented in the form of the 1/3-power law; the heat

transfer characteristic $Nu/(Ar/Sp)^{1/3}$ is about 0.15, which is exactly the same as that of Merte and Clark (1964). The data are also plotted in the form of Michiyoshi's (1988) correlation in Figure 4.10. The $Nu/(M_c Ar/Sp)^{1/4}$ ranges from 0.65 to 0.75, which is in agreement with the theoretical value of 0.696 from Michiyoshi et al. (1988) analysis.

The $Fr^{1/2}$ ranges from 0 to 1.5 for pool film boiling, which is in agreement with that found by Bromley et al. (1961) for cylinders. In general, all the three correlations could be used to express the saturated pool film boiling Nusselt number even without changing the constant, but the $1/3$ -power law correlation is better for taking account of the sphere temperature effect. However, for general purposes, the Michiyoshi's formula (which is in $1/4$ -power law) is preferred since it also covers subcooled conditions, correlates system pressure effects and is better for small diameter spheres.

Fig. 4.8 Saturated film boiling data plotted in the form of $1/4$ power law.

Fig. 4.9 Saturated film boiling data plotted in the form of 1/3 power law.

Fig. 4.10 Saturated film boiling data plotted in the form of Eq. (2.3.6).

Forced Convection Film Boiling In Saturated Water Flow

As mentioned in our literature review, there are two kinds of correlations:

$$\text{Mode 1: } Nu = C_{\text{sat,F1}} Re_1^{1/2} (\mu_l/\mu_v) [R^4/Sp^2]^{1/4} \quad (4.2.3) ,$$

$$\text{Mode 2: } Nu = C_{\text{sat,F2}} Re_1^{1/2} (\mu_l/\mu_v) [R^4 K/Sp]^{1/4} \quad (4.2.4) ,$$

where $Re_1 = Re = UD/\nu_l$. The mode 1 correlation was first obtained by Bromley (1961) and the mode 2 correlation may be derived from Kobayasi's (1965) or Wilson's (1979) analyses, the latter is presented in Appendix B.

Figure 4.11 shows the data plotted in the form of mode 1 correlation; the $C_{\text{sat,F1}} = Nu / \{ Re_1^{1/2} (\mu_l/\mu_v) [R^4/Sp^2]^{1/4} \}$ changes significantly with the sphere temperature and ranges from 2.0 to 3.2. In terms of mode 1 correlation, the value obtained from Bromley's (1961) experiment for cylinder is 2.7. On the other hand, in terms of mode 2 correlation, the data are plotted in Figure 4.12. It is obvious that the $C_{\text{sat,F2}} = Nu / \{ Re_1^{1/2} (\mu_l/\mu_v) [R^4 K/Sp]^{1/4} \}$ depends much less on the sphere temperature and is about 0.5 in the forced convection regimes, which can be compared with the theoretical constants 0.393 of Kobayasi (1965,1966), 0.553 of Epstein and Hauser (1980), 0.46 of Nishikawa et al. (1981), and the experimental constant of 0.554 from Liu, Shiotsu and Sakurai's (1992) cylinder experimental correlation. In addition, Figures 4.11 and 4.12 also indicate that forced convection film boiling starts from $Fr^{1/2}$ of 2.0, which also agrees with that found by Bromley (1953).

It is obvious that the mode 2 correlation is better than mode 1, as it was indicated in our literature review in Chapter 2, and the constant is 0.5 based on our experiment. Therefore, mode 2 correlation is recommended for forced convection film boiling in saturated water flows.

Fig. 4.11 Saturated film boiling data plotted in the form of mode 1 forced convection film boiling correlation.

Fig. 4.12 Saturated film boiling data are plotted in the form of mode 2 forced convection film boiling correlation.

The Sensitive Effect of Little Liquid Subcooling

In saturated film boiling experiments, the liquid is usually regarded as being saturated. But in reality, the liquid may not be 100% saturated because of the heat loss, and this may affect the interpretation of the results. Therefore, experimental assessment of the effect of a little subcooling on the film boiling heat transfer is very useful.

The experimental data obtained at conditions of 0.0, 1.0 and 2.0 °C subcooling are shown in Figure 4.13 in terms of mode 2 correlation. It is obvious that only one or two degrees subcooling can have a significant effect on the film boiling heat transfer, especially in the forced convection regime. It also implies that the so claimed saturated condition by Aziz (1986) may have about one degree of subcooling.

Fig. 4.13 The effect of little liquid subcooling on the film boiling heat transfer.

4.3 Film Boiling In Subcooled Conditions

Figure 4.14 shows the total heat flux of one typical series of transient mode runs of film boiling on a 12.7 mm stainless steel sphere in 80 °C tap water. The influences of temperature and flow velocity on the total heat flux could be easily identified. The quench temperature is about 300 °C in still water and increases slowly with the water velocity when the velocity is small ($0 < U < 0.5$ m/s), and increases quickly when the velocity is high. The quench temperatures are much higher than those obtained at saturated conditions. More data are tabled in Appendix D.

The experimental Nusselt numbers obtained in 10 and 20 °C subcooling conditions are shown in Figure 4.15 and 4.16 respectively. Unlike the case in a saturated condition, the Nu always increases with the Re, first slowly and then quickly when Re is greater than 10000, especially when the subcooling is high. As in the case of the saturated condition, the Nusselt number decreases significantly when the sphere temperature increases.

Fig. 4.14 The total heat flux transients from a typical series of transient mode runs in 20 °C subcooled water.

Fig. 4.15 Subcooled film boiling data plotted in the form of Nu vs. Re .

Fig. 4.16 Subcooled film boiling data plotted in the form of Nu vs. Re .

Pool Film Boiling In Subcooled Water

For pool film boiling in subcooled liquid, correlations in addition law [such as Farahat (1972), Siviour (1970) and Dhir (1978)] and correlations in ratio law [such as Michiyoshi (1988) and Sakurai (1990b)] are usually used to correlate the heat transfer data, as indicated in Chapter 2. Thus, our experimental data are compared with these correlations, as shown in Figure 4.17, at a condition of sphere temperature 760 °C; the experimental data is about 8% higher than Michiyoshi's correlation and about 14% higher than Sakurai's ratio correlation for cylinders. Compared with Dhir's correlation, our data is a little lower for low subcooling and a little higher for large subcooling.

The experimental data, including all the data with sphere temperatures from 900 °C to quench, are plotted in form of Michiyoshi's correlation in Figure 4.18, and it shows agreement within a $\pm 15\%$ band. The Michiyoshi's correlation is given as

$$Nu = 0.696 (Ar/Sp)^{1/4} Mc^{1/4} \quad (4.3.1)$$

where

$$Mc = E^3 / [1 + E / (Sp'Pr)] / (RPrSp)^2$$

$$E = (A + C B^{1/2})^{1/3} + (A - C B^{1/2})^{1/3} + (1/3) Sc^*$$

$$A = (1/27) Sc^{*3} + (1/3) R^2 Sp' Pr Sc^* + (1/4) R^2 Sp'^2 Pr^2$$

$$B = (-4/27) Sc^{*2} + (2/3) Sp' Pr Sc^* - (32/27) Sp' Pr R^2$$

$$+ (1/4) Sp'^2 Pr^2 + (2/27) Sc^{*3} / R^2$$

$$C = (1/2) R^2 Sp' Pr, \quad Sc^* = C_{pl} \Delta T_{sub} / h'_{fg}$$

Fig. 4.17 A comparison of subcooled film boiling data with the correlations, at sphere superheat of 660 °C and 1 bar pressure.

Fig. 4.18 Subcooled film boiling data correlated in the form of Eq. (2.3.6).

Forced Convection Film Boiling In Subcooled Water Flows

To give a sense of how the water subcooling and water velocity affect the film boiling heat transfer, the ratios of experimental Nusselt number over the corresponding saturated pool film boiling Nusselt number are plotted in Figure 4.19. In general, the subcooling has a stronger effect than the flow velocity; the velocity influence is enhanced when the subcooling is large.

For highly subcooled forced convection film boiling, as mentioned in Chapter 2, several analyses suggested the same correlation in the form of

$$Nu = C_{sub,F} Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc'/Sp') \quad (4.3.2) \quad .$$

This expression is not suitable for saturated or small subcooled cases. So in order to correlate all the forced convection film boiling by one equation, the following expression may be suggested

$$Nu = Nu_s + C'_{sub,F} Re_l^n Pr_l^{1/2} (\mu_l/\mu_v)(Sc'/Sp') \quad (4.3.3) \quad ,$$

where the Nu_s is the corresponding Nusselt number for saturated forced convection film boiling, given by $Nu_s = 0.5 Re_l^{1/2} (\mu_l/\mu_v)[R^4 K/Sp']^{1/4}$. Physically, Eq. (4.3.3) may be interpreted as: the total heat flux equals the heat that is required to sustain the vapor film plus the heat that convected into the liquid stream. The Constant $C'_{sub,F}$ and the power n on Re_l should be determined by experimental data.

In terms of Eq. (4.3.2), the experimental data obtained at 20 °C subcooling are plotted in Figure 4.20. The sphere superheat influence is well correlated and the heat transfer characteristic $C_{sub,F} = Nu / \{Re_l^{1/2} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp)\}$ varies from 1.5 to 2.0, in contrast with the theoretical constant of 0.977 from Epstein and Hauser's (1980) analysis and the suggested (by them) constant of 2.04.

According to our highly subcooled forced convection film boiling experimental data, we found that the best constants for Eq. (4.3.3) are $C'_{sub,F} = 0.072$ and $n =$

0.77 in order to correlate well with the data. Thus, Eq. (4.3.3) becomes

$$Nu = Nu_s + 0.072 Re_l^{0.77} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp') \quad (4.3.4)$$

The reason for our empirical power n being greater than $1/2$ is due to the turbulence effect. The fact that turbulence increases the power on the Reynolds number is also shown in Wang and Peng's (1992) study of film boiling on a horizontal flat plate in forced convection, where the power n is 0.8.

In the form of Eq. (4.3.4), the data are plotted in Figure 4.21; the sphere superheat effect is still well correlated. More data obtained at various subcooling conditions are shown in Figure 4.22. For $Fr_l^{1/2}$ greater than 2.0, the $C'_{sub,F} = Nu/Re_l^{0.77} Pr_l^{1/2} (\mu_l/\mu_v)(Sc/Sp')$ is about 0.072.

Fig. 4.19 The enhancement of liquid subcooling and velocity on the film boiling heat transfer.

Fig. 4.20 Subcooled film boiling data plotted in the form of Eq. (4.3.2).

Fig. 4.21 Subcooled film boiling data plotted in the form of Eq. (4.3.3).

Fig. 4.22 More subcooled data plotted in the form of Eq. (4.3.3).

4.4 Diameter Effect on Film Boiling

Four different sizes of spheres (6.35, 9.53, 12.7 and 19.1 mm O.D.) were used in our experiment to check the effect of sphere diameter on the film boiling heat transfer. The diameter effects on correlation Eq. (4.2.4) for saturated forced convection film boiling and the effects on correlation Eq. (4.3.4) for subcooled forced convection film boiling are shown in Figure (4.23) and (4.24) respectively. The results indicate that in our diameter range from 0.635 to 19.1 mm, the diameter effects on forced convection film boiling correlations are not distinguishable.

The same conclusion was obtained for pool film boiling if Eq. (4.3.1) is used to correlate the data. However, according to several previous studies on pool film boiling, as indicated in the literature review in Chapter 2, this correlation does not work well for very small and very large diameter spheres. Sakurai et al. (1990b),

based on their systematic experimental data obtained from film boiling on cylinders, modified their analytical solution (which is the same as Michiyoshi's correlation Eq. (4.3.1)), by introducing a diameter-effect correction-factor $K(d')$ on the right hand side of the equation and a factor $(1+2/Nu)^{-1}$ on the left hand side to correlate their small wire/cylinder film boiling data well. Since the film boilings on spheres and on cylinders are quite similar, it is reasonable to assume that the Sakurai's diameter-effect correction- factor for cylinders should also apply to the case of spheres. Comparing our data with Sakurai's correlation, we found our experimental data are about 10% to 14% higher than their correlation. In addition, according to Michiyoshi's et al (1988) theoretical analysis, the Nusselt number for spheres is 14% higher than that for horizontal cylinders. Therefore a factor of 1.14 is applied to Sakurai's et al (1990b) correlation to correlate our sphere pool film boiling experimental data, and the agreement turned out to be very good within a $\pm 15\%$ band, as shown in Figure 4.25. The complete correlation for pool film boiling at both saturated and subcooled conditions is given as

$$Nu/(1+2/Nu) = K_c(d') (Ar/Sp)^{1/4} M_c^{1/4} \quad (4.4.1)$$

where M_c is the same as that in Eq. (4.3.1) and $K_c(d')$ is calculated by

$$K_c(d') = 0.5d'^{-1/4}, \quad \text{for } d' < 0.14$$

$$K_c(d') = 0.86/(1+0.28d'), \quad \text{for } 0.14 < d' < 1.25$$

$$K_c(d') = 2.4d'/(1+3.0d'), \quad \text{for } 1.25 < d' < 6.6$$

$$K_c(d') = 0.47d'^{1/4}, \quad \text{for } d' > 6.6$$

Fig. 4.23 Data obtained from different sizes of sphere at saturated condition, plotted in terms of saturated forced convection correlation.

Fig. 4.24 Data obtained from different sizes of sphere at subcooled conditions, plotted in terms of subcooled forced convection correlation.

Fig. 4.25 Data obtained from different sizes of sphere at subcooled pool film boiling conditions (from 0.0 to 40 °C).

4.5 Material and Water Effects on Film Boiling

Our experiments were conducted with distilled water, deionized water, tap water, and deionized water with salt additions. The experiment shows that the kind of water used has no effect on the heat transfer data obtained from the transient mode operation, and it only affects the speed of sphere surface oxidation.

For the cool-down transient experimental technique, one concern is that the nonuniformity of temperature distribution in the stainless steel sphere may affect the accuracy of the film boiling heat transfer data. This was investigated and checked by doing the transient experiment with brass spheres of diameters 9.53 and 12.7 mm, which have a thermal conductivity that is 10 times higher than stainless steel. The total film boiling heat flux obtained with a brass sphere in saturated water is shown in Figure 4.26; it is very close to that obtained with a stainless steel sphere, and the only

difference is that the brass sphere has a little lower quench temperature than the stainless steel sphere, especially when the water velocity is high.

The ratios of experimental Nusselt numbers (obtained with brass) over the predicted Nusselt numbers from Eq. (4.4.1) for pool film boiling are shown in Figure 4.27, which shows excellent agreement when $Fr^{1/2}$ is smaller than 1. For forced convection, the ratios of experimental Nusselt numbers over the predicted Nusselt numbers from Eq. (4.2.4) are shown in Figure 4.28, which also shows good agreement when $Fr^{1/2}$ is greater than 2. This indicates that the film boiling heat transfer does not depend on the sphere material. It also implies that the nonuniformity of temperature distribution in the stainless steel sphere does not affect the accuracy of film boiling heat transfer data significantly.

Fig. 4.26 Total heat flux transients from a typical series of transient mode runs in the saturated condition with a brass sphere.

Fig. 4.27 Data obtained from brass sphere are compared with pool film boiling correlation.

Fig. 4.28 Data obtained from brass sphere are compared with forced convection film boiling correlation.

4.6 Steady-State Mode Operation Results

Steady-state mode operations have been conducted with stainless-steel spheres in waters with different water electrical conductivities. The ratios of experimental data over the predicted values from Eq. (4.2.4) which is obtained from transient mode experiment are shown in Figure 4.29. If we assume that the film boiling heat transfer in the transient state and steady state are the same, these data are seen to confirm the applicability of the power calibration (in air) to an all-water flow system, provided the electrical conductivity is adequately large (over 200 micromho).

Fig. 4.29 The effect of water electrical conductivity (shown in parenthesis in micromho) on the power coupling to the sphere. D.W. is for distilled water; D.I.W. is for deionized water; T-LC.W. and T-HC.W. are for low conductivity and high conductivity tap water respectively.

Chapter 5

A Theoretical Analysis of Film Boiling in Single-Phase Flows

5.1 Introduction to the Theoretical Analysis

As the literature review in Chapter 2 indicates, from theoretical aspect, film boiling on spheres has been studied extensively. However, there is still no theoretical film boiling model that, by itself, covers all the single-phase film boiling regimes for spheres (from natural to forced convection and from completely saturated to highly subcooled conditions). Frederking and Clark's (1963) analysis, which is similar to Nusselt's analysis for condensation on a vertical wall, is only for *saturated natural convection*. Witte's (1968) simple analysis, which uses the potential velocity around the sphere as the vapor velocity, is only for *saturated forced convection*. Kobayasi (1965), following the Bromley's (1953) analysis, is only for *saturated* case. Epstein's (1980) model, which applies similarity boundary layer theory and perturbation method to the front stagnation point of the sphere, is only for *forced convection*, either saturated or highly subcooled. Michiyoshi (1988) and Tso (1990) obtained a close form analytical solution for *natural free convection* film boiling on a sphere.

For film boiling on cylinders, there are also many pieces-wise analyses. For example, Bromley (1953) carried out an analysis for *saturated* cases based on Bernoulli's theorem and a consideration of viscous drag to the vapor. Motte and Bromley (1957) also did an analysis for *subcooled forced convection* film boiling from an energy partition and turbulent point of view. Liu, Shiotu and Sakurai (1992) did an analysis for *saturated forced convection* by means of an integral method with a single-phase boundary layer. Shigechi, Ito and Nishikawa's (1983) analytical model, which is based on an integral method with a two-phase laminar boundary-layer theory, covers all the single-phase film boiling regimes, although it must be solved numerically.

In this chapter, an analysis which is similar to that of Shigechi, Ito and Nishikawa (1983) is carried out for a sphere in all the film boiling regimes. There are two major differences between our model and Shigechi's. First, the boundaries around the sphere are axial symmetric instead of two-dimensional as for cylinders. Secondly, the potential flow over a sphere is different from that over a cylinder. Additionally in our model, the film properties and the effective latent heat of evaporation are used instead of the properties at the saturation point.

Shigechi, Ito and Nishikawa's analysis treats the separated rear region with an arbitrary film thickness function. This is fine for the saturated case since the heat transfer in this region is very small compared with that in the unseparated front region. However, with this arbitrary thickness function for the rear separated region, the overall heat transfer becomes arbitrary for subcooled cases because the rear region also contributes a significant amount in heat transfer. In this work, a turbulent eddy model [Theofanous et al (1976)] is applied to calculate the heat transfer in the separated rear region for the subcooled forced convection cases, which agree with the experimental data very well.

5.2 Physical Model and Fundamental Equations

Consider film boiling from a sphere with a uniform surface temperature T_w , which is placed in a stream of liquid flowing upward with a uniform velocity U_∞ , as shown in Fig. 5.1. The gravitational force is opposite to the coming flow direction. The liquid is subcooled by ΔT_{sub} below the saturation temperature T_s corresponding to the system pressure. The following assumptions are made.

- (1) The vapor film and the liquid (water) flow adjacent to the vapor-liquid interface are regarded as boundary-layers. This is a two-phase boundary-layer.
- (2) The flows in the vapor and liquid boundary-layer are steady laminar flow and the flow outside the liquid boundary-layer is a potential flow.
- (3) Physical properties are constant and are evaluated at the film temperatures. The

density of vapor is negligible compared with that of liquid when the vapor momentum equation is concerned.

(4) The inertia terms in the momentum equation and the convection terms in the energy equation for vapor film layer are neglected for simplicity. The effects of convective terms are compensated by using an effective latent heat of vaporization h'_{fg} at the interface.

$$h'_{fg} = h_{fg} + 0.5C_{pv}\Delta T_{\text{sup}}$$

(5) The vapor-liquid interface is smooth and the pressure difference due to surface tension is small and not considered.

(6) Radiation heat transfer is not considered.

Fig. 5.1 Physical model and coordinate system.

Under the above assumptions, the fundamental equations for the vapor layer are as follows

Continuity equation:

$$\frac{1}{r} \frac{\partial u_v}{\partial \theta} + \frac{\partial v_v}{\partial y} + \frac{u_v}{r} \cot \theta = 0 \quad (1),$$

Momentum equation:

$$0 = -g\rho_v \sin\theta - \frac{1}{r} \frac{\partial p_v}{\partial \theta} + \mu_v \frac{\partial^2 u_v}{\partial y^2} \quad (2),$$

Energy equation:

$$0 = k_v \frac{\partial^2 T_v}{\partial y^2} \quad (3).$$

For the potential flow outside the liquid boundary-layer, the pressure is given by

$$p_l = p_0 + g\rho_l r \cos\theta + \frac{1}{2}\rho_l U_\infty^2 - \frac{1}{2}\rho_l \left(\frac{3}{2} U_\infty \sin\theta \right)^2$$

Thus the pressure gradient is

$$-\frac{1}{r} \frac{\partial p_l}{\partial \theta} = \rho_l \left(g \sin\theta + \frac{9}{8} \frac{U_\infty^2}{r} \sin 2\theta \right)$$

Neglecting the pressure difference due to curvature of the vapor-liquid interface, using the assumption that the pressure in the vapor layer equals the pressure in the liquid layer, which is assumed to be the pressure of potential flow around the sphere, substituting the above equation into the vapor layer momentum Eq. (2) and taking account of the third assumption, the following equation is obtained

$$\frac{\mu_v}{\rho_v} \frac{\partial^2 u_v}{\partial y^2} + g \frac{\rho_l}{\rho_v} \sin\theta + \frac{9}{8} \frac{\rho_l}{\rho_v} \frac{U_\infty^2}{r} \sin 2\theta = 0 \quad (4).$$

For the liquid boundary-layer, the three governing equations are as follows

Continuity equation:

$$\frac{1}{r} \frac{\partial u_l}{\partial \theta} + \frac{\partial v_l}{\partial y} + \frac{u_l}{r} \cot \theta = 0 \quad (5),$$

Momentum equation:

$$\frac{u_l}{r} \frac{\partial u_l}{\partial \theta} + v_l \frac{\partial u_l}{\partial y} = g \beta_l (T_l - T_\infty) \sin \theta + \frac{9}{8} \frac{U_\infty^2}{r} \sin 2\theta + \frac{\mu_l}{\rho_l} \frac{\partial^2 u_l}{\partial y^2} \quad (6),$$

Energy equation:

$$\frac{u_l}{r} \frac{\partial T_l}{\partial \theta} + v_l \frac{\partial T_l}{\partial y} = \frac{k_l}{c_{pl} \rho_l} \frac{\partial^2 T_l}{\partial y^2} \quad (7).$$

The boundary conditions and the compatibility conditions at the vapor-liquid interface are

$$y = 0: \quad u_v = v_v = 0 \quad (8),$$

$$T_v = T_w \quad (9),$$

$$y = \delta: \quad u_v = u_l = u_\delta \quad (10),$$

$$\mu_v \left(\frac{\partial u_v}{\partial y} \right)_\delta = \mu_l \left(\frac{\partial u_l}{\partial y} \right)_\delta \quad (11),$$

$$\rho_v \left(v_v - \frac{u_v}{r} \frac{d\delta}{d\theta} \right)_\delta = \rho_l \left(v_l - \frac{u_l}{r} \frac{d\delta}{d\theta} \right)_\delta \quad (12),$$

$$T_v = T_l = T_\delta \quad (13),$$

$$-k_v \left(\frac{\partial T_v}{\partial y} \right)_\delta = -\dot{m} h'_{fs} - k_l \left(\frac{\partial T_l}{\partial y} \right)_\delta \quad (14),$$

$$y = \delta + \delta_u:$$

$$u_i = u_{i,\delta_s} = \frac{3}{2} U_{\infty} \sin \theta \quad (15),$$

$$\left(\frac{\partial u_i}{\partial y} \right)_{\delta_s} = 0 \quad (16),$$

$$y = \delta + \delta_T:$$

$$T_i = T_{\infty} \quad (17),$$

$$\left(\frac{\partial T_i}{\partial y} \right)_{\delta_T} = 0 \quad (18).$$

The $\dot{m}$ in Eq. (14) is given by

$$-\dot{m} = \rho_v \left(\frac{1}{r} \frac{d}{d\theta} \int_0^{\delta} u_v dy + \frac{\cot \theta}{r} \int_0^{\delta} u_v dy \right) \quad (19).$$

Before transforming these equations into dimensionless forms, integrating Eq. (1) with respect to y from 0 to δ and substituting the result into Eq. (19), we have

$$-\dot{m} = \rho_v \left(\frac{1}{r} \frac{d}{d\theta} \int_0^{\delta} u_v dy + \frac{\cot \theta}{r} \int_0^{\delta} u_v dy \right) = -\rho_v \left(v_v - \frac{u_v}{r} \frac{d\delta}{d\theta} \right)_{\delta} \quad (19)'$$

Integrating the vapor layer energy equation, Eq.(3), with the boundary conditions, Eqs. (9) and (13), we obtain a linear temperature distribution. Substituting it into Eq. (14) and applying Eq. (19), we have

$$h'_{fs} \rho_v \left(\frac{1}{r} \frac{d}{d\theta} \int_0^{\delta} u_v dy + \frac{\cot \theta}{r} \int_0^{\delta} u_v dy \right) = k_v \frac{T_w - T_s}{\delta} + k_l \left(\frac{\partial T_l}{\partial y} \right)_{\delta} \quad (3)'$$

$$h'_{fs} \rho_v \left(v_v - \frac{u_v}{r} \frac{d\delta}{d\theta} \right)_{\delta} = -k_v \frac{T_w - T_s}{\delta} - k_l \left(\frac{\partial T_l}{\partial y} \right)_{\delta} \quad (14)'$$

We define the dimensionless variables and parameters as follows

$$l^* = \left(\frac{\mu_v^2 r}{\rho_v^2 g} \right)^{1/4}; \quad u^* = (gr)^{1/2}; \quad v^* = \left(\frac{\mu_v^2 g}{\rho_v^2 r} \right)^{1/4} \quad (20),$$

$$Y = y/l^*; \quad \Delta = \delta/l^*; \quad \Delta_u = \delta_u/l^*; \quad \Delta_T = \delta_T/l^* \quad (21),$$

$$U_v = u_v/u^*; \quad U_l = u_l/u^*; \quad V_v = v_v/v^*; \quad V_l = v_l/v^* \quad (22),$$

$$F = U_v^2/gr = 2Fr \quad (23),$$

$$K = \rho_l/\rho_v \quad (24),$$

$$R = (\rho_v \mu_v / \rho_l \mu_l)^{1/2} \quad (25),$$

$$\Theta = c_{pl}(T_l - T_\infty) / Pr_l h'_{fs} \quad (26),$$

$$Sc = c_{pl}(T_s - T_\infty) / Pr_l h'_{fs} \quad (27),$$

$$Sp = c_{pv}(T_w - T_s) / Pr_v h'_{fs} \quad (28),$$

$$B_l = \beta_l h'_{fs} / c_{pl} \quad (29).$$

The dimensionless governing equations and boundary conditions now become as follows.

For the vapor film:

$$\frac{\partial U_v}{\partial \theta} + \frac{\partial V_v}{\partial Y} + U_v \cot \theta = 0 \quad (30),$$

$$\frac{\partial^2 U_v}{\partial Y^2} + K \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta = 0 \quad (31),$$

$$\frac{d}{d\theta} \int_0^\Delta U_v dY + \cot \theta \int_0^\Delta U_v dY = \frac{Sp'}{\Delta} + \frac{1}{R^2 K} \left(\frac{\partial \Theta}{\partial Y} \right)_\Delta \quad (32).$$

For the liquid boundary layer:

$$\frac{\partial U_l}{\partial \theta} + \frac{\partial V_l}{\partial Y} + U_l \cot \theta = 0 \quad (33),$$

$$U_i \frac{\partial U_i}{\partial \theta} + V_i \frac{\partial U_i}{\partial Y} = Pr_i B_i \Theta \sin \theta + \left(\frac{1}{RK} \right)^2 \frac{\partial^2 U_i}{\partial Y^2} + \frac{9}{8} F \sin 2\theta \quad (34),$$

$$U_i \frac{\partial \Theta}{\partial \theta} + V_i \frac{\partial \Theta}{\partial Y} = \frac{1}{Pr_i} \left(\frac{1}{RK} \right)^2 \frac{\partial^2 \Theta}{\partial Y^2} \quad (35).$$

The boundary conditions:

$$\begin{aligned} \text{At } Y = 0: \\ U_v = V_v = 0 \end{aligned} \quad (36).$$

$$\begin{aligned} \text{At } Y = \Delta: \\ U_v = U_i = U_\Delta \end{aligned} \quad (37),$$

$$\left(\frac{\partial U_v}{\partial Y} \right)_\Delta = \frac{1}{R^2 K} \left(\frac{\partial U_i}{\partial Y} \right)_\Delta \quad (38),$$

$$\left(V_v - U_v \frac{d\Delta}{d\theta} \right)_\Delta = K \left(V_i - U_i \frac{d\Delta}{d\theta} \right)_\Delta \quad (39),$$

$$\Theta_\Delta = Sc' \quad (40),$$

$$\left(V_v - U_v \frac{d\Delta}{d\theta} \right)_\Delta = -\frac{Sp'}{\Delta} - \frac{1}{R^2 K} \left(\frac{\partial \Theta}{\partial Y} \right)_\Delta \quad (41).$$

$$\begin{aligned} \text{At } Y = \Delta + \Delta_u: \\ U_i = U_{i,\Delta_u} = (3/2) F^{1/2} \sin \theta \end{aligned} \quad (42),$$

$$\left(\frac{\partial U_i}{\partial Y} \right)_{\Delta_u} = 0 \quad (43).$$

$$\begin{aligned} \text{At } Y = \Delta + \Delta_T: \\ \Theta_{\Delta_T} = 0 \end{aligned} \quad (44),$$

$$\left(\frac{\partial \Theta}{\partial Y} \right)_{\Delta_T} = 0 \quad (45).$$

5.3 Method of Solution

Integrating Eq. (31) twice with respect to Y and determining the integral constant by boundary condition Eqs. (36) and (37), we have a vapor layer velocity distribution function

$$U_v = \left\{ \frac{U_\Delta}{\Delta} + \frac{K\Delta}{2} \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta \right\} Y - \left\{ \frac{K}{2} \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta \right\} Y^2 \quad (46).$$

As an approximation, we assume quadratic functions of Y for the radial distributions of tangential velocity U_l and temperature Θ in the liquid boundary layer and determine the constants of these functions by Eqs. (37), (42) and (43) for U_l , and by Eqs. (40), (44) and (45) for Θ . So the velocity and temperature distributions in the liquid layer are

$$U_l = U_\Delta + \left(\frac{3}{2} F^{1/2} \sin \theta - U_\Delta \right) \left[2 \left(\frac{\xi}{\Delta_u} \right) - \left(\frac{\xi}{\Delta_u} \right)^2 \right] \quad (47),$$

$$\Theta = Sc' \left[1 - \left(\frac{\xi}{\Delta_r} \right)^2 \right] \quad (48),$$

where

$$\xi = Y - \Delta = (y - \delta) / u^* \quad (49).$$

Integrating Eqs. (33) and (34) from Δ to $\Delta + \Delta_u$ with respect to Y we have

$$\begin{aligned} \frac{d}{d\theta} \int_{\Delta}^{\Delta + \Delta_u} U_l dY - U_{l,\Delta} \frac{d(\Delta + \Delta_u)}{d\theta} + U_{l,\Delta} \frac{d\Delta}{d\theta} \\ + \cot \theta \int_{\Delta}^{\Delta + \Delta_u} U_l dY + V_{l,\Delta} - V_{l,\Delta} = 0 \end{aligned} \quad (50),$$

$$\begin{aligned} \frac{d}{d\theta} \int_{\Delta}^{\Delta + \Delta_u} U_l^2 dY - U_{l,\Delta}^2 \frac{d(\Delta + \Delta_u)}{d\theta} + U_{l,\Delta}^2 \frac{d\Delta}{d\theta} + (V_l U_l)_{\Delta_u} - (V_l U_l)_{\Delta} = \\ Pr_l B_l \sin \theta \int_{\Delta}^{\Delta + \Delta_u} \Theta dY - \left(\frac{1}{RK} \right)^2 \left(\frac{\partial U_l}{\partial Y} \right)_{\Delta} + \frac{9}{8} F \Delta_u \sin 2\theta \end{aligned} \quad (51).$$

Eliminating $V_{l,\Delta u}$ between the above two equations and transforming the integral variable Y to ξ , we get

$$\begin{aligned} \frac{d}{d\theta} \int_0^{\Delta_u} U_l^2 d\xi - U_{l,\Delta_u} \frac{d}{d\theta} \int_0^{\Delta_u} U_l d\xi + (U_{l,\Delta} - U_{l,\Delta_u}) \left\{ \frac{Sp'}{K\Delta} + \left(\frac{1}{RK} \right)^2 \left(\frac{\partial \Theta}{\partial \xi} \right)_0 \right\} = \\ Pr_l B_l \sin \theta \int_0^{\Delta_u} \Theta d\xi - \left(\frac{1}{RK} \right)^2 \left(\frac{\partial U_l}{\partial \xi} \right)_0 + \frac{9}{8} F \Delta_u \sin 2\theta + U_{l,\Delta_u} \cot \theta \int_0^{\Delta_u} U_l d\xi \end{aligned} \quad (52)$$

Integration of Eq. (35) from Δ to $\Delta + \Delta_T$ with respect to Y and applying Eq. (33) yields

$$\begin{aligned} \frac{d}{d\theta} \int_{\Delta}^{\Delta + \Delta_T} U_l \Theta dY - Sc' \left(V_{l,\Delta} - U_{l,\Delta} \frac{d\Delta}{d\theta} \right) + \cot \theta \int_{\Delta}^{\Delta + \Delta_T} U_l \Theta dY = \\ - \frac{1}{Pr_l} \left(\frac{1}{RK} \right)^2 \left(\frac{\partial \Theta}{\partial Y} \right)_{\Delta} \end{aligned} \quad (53)$$

Applying Eq. (41) and transforming the integration variable Y to ξ , we obtain

$$\frac{d}{d\theta} \int_0^{\Delta_T} U_l \Theta d\xi + \frac{Sc' Sp'}{K\Delta} + \left(Sc' + \frac{1}{Pr_l} \right) \left(\frac{1}{RK} \right)^2 \left(\frac{\partial \Theta}{\partial \xi} \right)_0 + \cot \theta \int_0^{\Delta_T} U_l \Theta d\xi = 0 \quad (54)$$

Substitution of Eqs. (46) and (47) into Eq. (38) yields a relationship between Δ_u and Δ as following:

$$\Delta_u = \frac{4}{R^2 K} \frac{(3/2) F^{1/2} \sin \theta - U_{\Delta}}{2U_{\Delta} - K\Delta^2(1 + (9/4)F \cos \theta) \sin \theta} \Delta \quad (55)$$

Substituting Eq. (46) for U_v , Eq. (47) for U_l , and Eq. (48) for Θ into Eqs. (32), (52) and (54) and eliminating Δ_u by Eq. (55), the final differential equations for

Δ , Δ_T and U_Δ are

$$F_{11} \frac{d\Delta}{d\theta} + F_{12} \frac{dU_\Delta}{d\theta} = F_{1c} \quad (56),$$

$$F_{21} \frac{d\Delta}{d\theta} + F_{22} \frac{dU_\Delta}{d\theta} = F_{2c} \quad (57),$$

$$F_{31} \frac{d\Delta}{d\theta} + F_{32} \frac{dU_\Delta}{d\theta} + F_{33} \frac{d\Delta_T}{d\theta} = F_{3c} \quad (58),$$

where F_{11} , F_{12} , F_{1c} F_{3c} are functions of the variables Δ , U_Δ , Δ_T and θ , and of the parameters B_1 , Fr , K , Pr_1 , R , Sc' and Sp' . The algebraic expressions of these functions for the case of $\Delta_u > \Delta_T$ are as follows:

$$\begin{aligned} F_{11} &= \frac{U_\Delta \Delta}{2} + \frac{K \Delta^3}{4} \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta; \quad F_{12} = \frac{\Delta^2}{2}, \\ F_{1c} &= Sp' - \frac{2Sc' \Delta}{R^2 K \Delta_T} - \frac{K \Delta^4}{12} \left(\cos \theta + \frac{9}{4} F \cos 2\theta \right) \\ &\quad - \left\{ \frac{\Delta^2}{2} U_\Delta + \frac{K \Delta^4}{12} \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta \right\} \cot \theta, \\ F_{21} &= G_u G_{Fa}; \quad F_{22} = \Delta_u \left(\frac{2}{5} U_\Delta - \frac{1}{10} F^{1/2} \sin \theta \right) - 2G_u \left(\frac{2\Delta}{R^2 K} + \Delta_u \right), \\ F_{2c} &= Pr_1 B_1 \frac{Sc' \Delta_T}{3} \sin \theta - \Delta_u \left\{ \frac{2}{5} U_\Delta F^{1/2} \cos \theta - \frac{27}{40} F \sin 2\theta \right\} \\ &\quad + \left(\frac{3}{2} F^{1/2} \sin \theta - U_\Delta \right) \left\{ \frac{Sp'}{K \Delta} - 2 \left(\frac{1}{\Delta_u} + \frac{Sc'}{\Delta_T} \right) \left(\frac{1}{RK} \right)^2 \right\} - G_u G_{Fc} \\ &\quad + \frac{3}{2} F^{1/2} \sin \theta \Delta_u \left(\frac{U_\Delta}{3} + F^{1/2} \sin \theta \right) \cot \theta, \\ F_{31} &= G_T G_{Fa}; \quad F_{32} = \frac{\Delta_T}{3} - \frac{1}{6} \left(\frac{\Delta_T^2}{\Delta_u} - \frac{\Delta_T^3}{5 \Delta_u^2} \right) - 2G_T \left(\frac{2\Delta}{R^2 K} + \Delta_u \right), \\ F_{33} &= \frac{U_\Delta}{3} + \frac{1}{6} \left\{ \frac{3}{2} F^{1/2} \sin \theta - U_\Delta \right\} \left\{ 2 \frac{\Delta_T}{\Delta_u} - \frac{3}{5} \left(\frac{\Delta_T}{\Delta_u} \right)^2 \right\}, \\ F_{3c} &= \frac{2}{\Delta_T} \left(Sc' + \frac{1}{Pr_1} \right) \left(\frac{1}{RK} \right)^2 - \frac{Sp'}{K \Delta} - \frac{1}{4} \left(\frac{\Delta_T^2}{\Delta_u} - \frac{\Delta_T^3}{5 \Delta_u^2} \right) F^{1/2} \cos \theta - G_T G_{Fc} \end{aligned}$$

$$\begin{aligned}
& - \left\{ \frac{U_{\Delta} \Delta_T}{3} + \frac{1}{6} \left(\frac{3}{2} F^{1/2} \sin \theta - U_{\Delta} \right) \left(\frac{\Delta_T^2}{\Delta_u} - \frac{\Delta_T^3}{5 \Delta_u^2} \right) \right\} \cot \theta \\
G_{F\Delta} &= \frac{4}{R^2 K} \left\{ \frac{3}{2} F^{1/2} \sin \theta - U_{\Delta} \right\} + 2 K \Delta \Delta_u \left(1 + \frac{9}{4} F \cos \theta \right) \sin \theta \\
G_{FC} &= \frac{6 \Delta}{R^2 K} F^{1/2} \cos \theta + K \Delta^2 \Delta_u \left(\cos \theta + \frac{9}{4} F \cos 2\theta \right) \\
G_u &= \frac{2 U_{\Delta}^2 - U_{\Delta} F^{1/2} \sin \theta - 3 F \sin^2 \theta}{10 [2 U_{\Delta} - \Delta^2 K (1 + (9/4) F \cos \theta) \sin \theta]} \\
G_T &= \frac{\left\{ (3/2) F^{1/2} \sin \theta - U_{\Delta} \right\} \left\{ (2/5) (\Delta_T / \Delta_u)^3 - (\Delta_T / \Delta_u)^2 \right\}}{6 [2 U_{\Delta} - \Delta^2 K (1 + (9/4) F \cos \theta) \sin \theta]} \quad (59)
\end{aligned}$$

For the case of $\Delta_u < \Delta_T$, which occurs in subcooled natural convection conditions, the functions that are different from those of the case of $\Delta_u > \Delta_T$ are given as follows:

$$\begin{aligned}
F_{2c} &= Pr_i B_i \frac{Sc' \Delta_T}{3} \sin \theta \left[1 - \left(1 - \frac{\Delta_u}{\Delta_T} \right)^3 \right] - \Delta_u \left\{ \frac{2}{5} U_{\Delta} F^{1/2} \cos \theta - \frac{27}{40} F \sin 2\theta \right\} \\
&+ \left(\frac{3}{2} F^{1/2} \sin \theta - U_{\Delta} \right) \left\{ \frac{Sp'}{K \Delta} - 2 \left(\frac{1}{\Delta_u} + \frac{Sc'}{\Delta_T} \right) \left(\frac{1}{RK} \right)^2 \right\} - G_u G_{FC} \\
&+ \frac{3}{2} F^{1/2} \sin \theta \Delta_u \left(\frac{U_{\Delta}}{3} + F^{1/2} \sin \theta \right) \cot \theta \\
F_{31} &= G_T G_{F\Delta}; \quad F_{32} = \frac{\Delta_T}{3} - G'_T - 2 G_T \left(\frac{2 \Delta}{R^2 K} + \Delta_u \right) \\
F_{33} &= \frac{U_{\Delta}}{3} + \left\{ \frac{3}{2} F^{1/2} \sin \theta - U_{\Delta} \right\} \left\{ \frac{1}{3} \left(1 - \frac{\Delta_u}{\Delta_T} \right)^3 + \frac{\Delta_u}{\Delta_T} - \frac{7}{6} \left(\frac{\Delta_u}{\Delta_T} \right)^2 + \frac{2}{5} \left(\frac{\Delta_u}{\Delta_T} \right)^3 \right\} \\
F_{3c} &= \frac{2}{\Delta_T} \left(Sc' + \frac{1}{Pr_i} \right) \left(\frac{1}{RK} \right)^2 - \frac{Sp'}{K \Delta} - \frac{3}{2} F^{1/2} \cos \theta G'_T - G_T G_{FC} \\
&- \left\{ \frac{U_{\Delta} \Delta_T}{3} + \left(\frac{3}{2} F^{1/2} \sin \theta - U_{\Delta} \right) G'_T \right\} \cot \theta
\end{aligned}$$

$$G'_T = \frac{\Delta_T}{3} \left(1 - \frac{\Delta_u}{\Delta_T} \right)^3 + \Delta_u \left[\frac{2}{3} - \frac{5}{6} \frac{\Delta_u}{\Delta_T} + \frac{3}{10} \left(\frac{\Delta_u}{\Delta_T} \right)^2 \right]$$

$$G_T = \frac{\left\{ (3/2) F^{1/2} \sin \theta - U_\Delta \right\} \left\{ (1/3) (\Delta_u / \Delta_T - 1) - 0.1 (\Delta_u / \Delta_T)^2 \right\}}{[2U_\Delta - \Delta^2 K (1 + (9/4) F \cos \theta) \sin \theta]} \quad (60)$$

Since the Prandtl number of water is always larger than 1.0, we may assume that the Δ_u is always larger than or equal to Δ_T at the front stagnation point. Our model shows that this is true. So the functions for the case of $\Delta_u > \Delta_T$ are always used at the front stagnation point. These functions are then continually used as long as $\Delta_u > \Delta_T$, as in the cases of subcooled forced convection. However, for the subcooled natural convection, Δ_T is equal to or almost equal to Δ_u in the front and rear stagnation regions, but larger than Δ_u when away from the stagnation points. In the computation, as long as $\Delta_u < \Delta_T$, the functions given by Eqs. (60) are used.

Because of the singularity at the stagnation point $\theta = 0$, a forward-integration of Eqs (56) to (58) cannot be directly performed. To overcome this difficulty, we assume that $\Delta = A_0 + A_2 \theta^2$, $U_\Delta = B_1 \theta$ and $\Delta_T = C_0 + C_2 \theta^2$ in the region from $\theta = 0$ to 0.05 rad. Substituting them into Eqs. (56), (57) and (58) and solving the resulting equations for A_0 , A_2 , B_1 , C_0 and C_2 by the Newton method, the starting values of Δ , U_Δ and Δ_T are determined. Then starting from $\theta = 0.05$ rad, Eqs. (56), (57) and (58) are forward integrated by means of Fourth-Order-Runge-Kutta method with a step interval of 0.001 rad.

The numerical integration can be performed up to the rear stagnation point without separation as long as $Fr < 2/9$. However, when $Fr > 2/9$, the vapor layer separates at a certain angle between $\theta = \pi/2$ and $\theta = \pi$, depending on the Fr . After the vapor layer separation, the numerical integration based on the model given above can not be performed further. To account for the heat transfer in the region after separation, as a preliminary model, it is assumed that a film with a uniform equivalence thickness Δ_e is covered all over the separated rear part of the sphere. The heat transfer in that region is merely the heat conduction through the film. According to our experimental observations and heat transfer data, it is also assumed that $\Delta_e =$

$2\Delta_s$ for the saturated condition and $\Delta_e = \Delta_s$ for subcooled condition, where Δ_s is the dimensionless vapor film thickness at the separation point. The effect of the choice of these equivalent film thickness on the overall heat transfer will be discussed latter. This is just a preliminary model for the heat transfer in the separated rear region, later a better turbulent heat transfer model based on Theofanous, Houze and Brumfield's (1976) model will be used to calculate the heat transfer in this region.

After solving the Eqs. (56), (57) and (58), the heat transfer characteristics are calculated as follows:

Local heat transfer coefficient in terms of Δ ,

$$h_o = \frac{q_o}{\Delta T_{sup}} = \frac{k_v}{\delta} = \left(\frac{2g}{v_v^2 d} \right)^{1/4} \frac{k_v}{\Delta} \quad (61).$$

Average heat transfer coefficient in terms of Δ_{av} ,

$$h = \frac{q}{\Delta T_{sup}} = \frac{k_v}{\delta_{av}} = \left(\frac{2g}{v_v^2 d} \right)^{1/4} \frac{k_v}{\Delta_{av}} \quad (62).$$

Average Nusselt number in terms of Δ_{av} ,

$$\frac{Nu}{Re_l^{1/2}} \left(\frac{\mu_v}{\mu_l} \right) = \left(\frac{2R^*}{Fr} \right)^{1/4} \frac{1}{\Delta_{av}} \quad (63).$$

where Δ_{av} is defined by

$$\frac{1}{\Delta_{av}} = \frac{1}{\pi d^2} \int_0^\pi \frac{1}{\Delta} \pi d \sin \theta \frac{d}{2} d\theta = 0.5 \int_0^\pi \frac{\sin \theta}{\Delta} d\theta \quad (64).$$

5.4 Results and Discussion

The numerical calculation was carried out for water at atmospheric pressure. The superheat ΔT_{sup} , subcooling ΔT_{sub} and Fr number were given first. Then the density ratio K , $\rho\mu$ ratio R , Prandtl number of liquid Pr_l , buoyant force parameter B_L , dimensionless superheat Sp' , and dimensionless subcooling Sc' were evaluated at film temperatures of $T_{f,v} = (T_w + T_s)/2$ and $T_{f,l} = (T_l + T_s)/2$. The ranges of Fr, Δ_{sup} and Δ_{sub} are:

$$Fr = 0.01 - 500; \quad \Delta_{\text{sup}} = 300 - 2000 \text{ }^\circ\text{C}; \quad \Delta_{\text{sub}} = 0 - 80. \text{ }^\circ\text{C}$$

The corresponding values for K , R , Pr_l and B_L are 2300 – 6200, 0.004 – 0.0055, 1.76 – 3.0 and 0.46 – 0.86 respectively. The corresponding ranges of Sp' and Sc' are 0.25 – 1.2 and 0 – 0.023 respectively.

Separation Point, Vapor Film and Liquid Boundary-Layer Thickness

As mentioned earlier, the vapor film boundary layer separates if Fr is larger than 2/9. This critical Froude number is larger than the 1/8 that was obtained by Shigechi, Ito and Nishikawa (1983) for a cylinder. The separation points of the numerical integration, i.e. the angles at which the integration breaks down, are shown in Fig 5.2. When $Fr > 2/9$, the separation angles can be expressed exactly by

$$\theta_s = \cos^{-1}\left(\frac{2}{9Fr}\right) \quad (5.4.1),$$

for both saturated and subcooled conditions. This is also different from that obtained by Shigechi, Ito and Nishikawa (1983) for a cylinder, which was $\theta_s = \cos^{-1}(1/8Fr)$.

No liquid layer separation (i.e. numerical integration breaking down) was found prior to the separation of vapor layer, although the liquid momentum boundary

layer thickness Δ_u is much larger than the vapor layer thickness Δ for highly subcooled forced convection conditions. In these cases, the liquid-vapor interface velocity is almost equal to the velocity of the potential flow and the liquid momentum boundary layer is almost a layer with uniform velocity. For this case, the thickness Δ_u is not important in the calculation. To verify this argument, another numerical integration scheme based on the assumption that $U_\Delta = 1.5F^{1/2}\sin\theta$ was developed for the cases whenever $\Delta_u/\Delta > 5$. It shows that the vapor film thickness, hence the heat transfer, was not affected significantly by this modification.

Fig. 5.2 The dependence of the separation angle θ_s of the theoretical model on the Froude number Fr .

The angular distributions of the reciprocal of the dimensionless vapor film thickness $1/\Delta$, which is proportional to the local heat transfer coefficient, are shown in Figs. 5.3 and 5.4 for saturated and subcooled cases respectively. In these figures, the slopes of the curve are discontinuous at the separation point for the cases of $Fr = 1.0, 3.0$ and 30.0 . Fig 5.3 shows that, for saturated forced convection cases, there is significant variation in heat transfer in the front part of the sphere and most of the heat transfer is contributed from this regime. In contrast, for subcooled forced

convection, Fig. 5.4 shows that there is less variation in the front part of the sphere (except in the vicinity of the front stagnation point), and the rear part of the sphere is as equally important as the front part in transfer heat.

For saturated film boiling, the angular variation of the ratio Δ_q/Δ is shown in Fig 5.5. It shows that, in the front part of the sphere, this ratio essentially only depends on the superheat ΔT_{sup} and is independent of the angle θ . But, for the rear part of the sphere, this ratio decreases as the angle θ increases in the case of natural free convection film boiling ($Fr < 0.01$).

For subcooled conditions, the ratios of Δ_q/Δ and Δ_T/Δ are shown against the angle in Figs. 5.6 and 5.7 respectively. Fig. 5.6 indicates that, unlike the saturated case, the ratio Δ_q/Δ becomes more Froude number dependent when it is highly subcooled. Fig. 5.7 shows that, except in the vicinity of the front stagnation point, the ratio Δ_T/Δ decreases with the angle for all flow conditions. Comparing Fig. 5.6 with Fig. 5.7, it is obvious that Δ_T/Δ_q is always smaller than or equal to one.

Fig. 5.3 Angular distribution of the dimensionless local heat transfer coefficients for saturated film boiling with $\Delta T_{\text{sup}} = 1000^\circ\text{C}$.

Fig. 5.4 Angular distribution of the dimensionless local heat transfer coefficients at subcooled condition.

Fig. 5.5 Angular variation of Δ_u/Δ at saturated conditions.

Fig. 5.6 Angular variation of Δ_u/Δ at subcooled conditions.

Fig. 5.7 Angular variation of Δ_T/Δ at subcooled conditions.

Heat Transfer Results at Saturated Conditions

The variation of $(Nu/Re_l^{1/2})(\mu_v/\mu_l)$ against the Froude number Fr and superheat ΔT_{sup} is shown in Fig. 5.8. The variation of $C_{sat,F1} = (Nu/Re_l^{1/2})(\mu_v/\mu_l)(Sp'^2/R^4)^{1/4}$, (the heat transfer characteristic of mode 1 forced convection saturated film boiling correlation), is shown in Fig. 5.9 with the superheat as the parameter. It is obvious that both of these two heat transfer characteristics are superheat dependent. As mentioned in the previous chapters, the best way to correlate the saturated film boiling heat transfer is by

$$C_{sat,N} = \frac{Nu}{Re_l^{1/2}} \frac{\mu_v}{\mu_l} \left(\frac{Fr Sp'}{R^4 K} \right)^{1/4} = Nu \left(\frac{Sp'}{Ar} \right)^{1/4} \quad (5.4.2) ,$$

$$C_{sat,F} = \frac{Nu}{Re_l^{1/2}} \frac{\mu_v}{\mu_l} \left(\frac{Sp'}{R^4 K} \right)^{1/4} \quad (5.4.3) ,$$

for natural free convection and forced convection respectively. The dependences of these two heat transfer characteristics on $Fr^{1/2}$ are shown in Fig. 5.10 at two significantly different superheat conditions. It shows that they are much less dependent on the superheat than are the $(Nu/Re_l^{1/2})(\mu_v/\mu_l)$ and $C_{sat,F1} = (Nu/Re_l^{1/2})(\mu_v/\mu_l)(Sp'^2/R^4)^{1/4}$.

In the pool film boiling regime, $C_{sat,N}$ is about 0.6, which can be compared with the value of 0.58 from Ito, Nishikawa and Shigechi's (1981) analysis for a cylinder, 0.62 from Bromley's (1950) experimental data for cylinders, 0.586 from Frederking's (1963) analysis for a sphere and ~ 0.60 from Michiyosh's (1988) analysis for a sphere. On the other hand, for forced convection, $C_{sat,F}$ is about 0.5, which is very close to the values of 0.46 obtained by Ito, Nishikawa and Shigechi (1981), 0.454 from Kobayasi's (1965) theory for spheres and 0.553 from Epstein and Hauser's (1980) analysis for a sphere and cylinder.

Fig. 5.8 The relationship between $(Nu/Re_l^{1/2})(\mu_v/\mu_l)$ and Froude number Fr at saturated film boiling conditions.

Fig. 5.9 The relationship between $C_{sat,Fl}$ and Froude number Fr at saturated film boiling conditions.

Fig. 5.10 The dependence of $C_{sat,N}$ and $C_{sat,F}$ on Froude number Fr at saturated film boiling conditions.

Figs. 5.11 and 5.12 compared two sets of our experimental data with the theory in terms of $C_{sat,N}$ and $C_{sat,F}$ respectively. They indicate that the agreement of the theory with the experiment is perfect in the forced convection regime. However, in the pool film boiling regime, the experimental data are about 10-20% higher than the theory.

In the theoretical model, the equivalent film thickness beyond the separation point is arbitrarily assumed to be two times that of the film thickness at the separation point, i.e., $\Delta_e = 2\Delta_s$. It is interesting to see what would be the result if the Δ_e is assumed to be infinity, i.e. no heat transfer contribution from the region beyond the separation. The comparison is shown in Fig. 5.13. It shows that with the assumption of $\Delta_e = 2\Delta_s$, the rear part of the sphere contributes about 16% in the overall heat transfer in forced convection regime for this particular case.

In conclusion, for saturated film boiling, our theoretical analysis agrees well with the existing analyses either for spheres or for cylinders and also agrees well with our experiment.

Fig. 5.11 A comparison of saturated experimental data with the theory in terms of $C_{\text{sat},N}$.

Fig. 5.12 A comparison of saturated experimental data with the theory in terms of $C_{\text{sat},F}$.

Fig. 5.13 The influence of the equivalent film thickness Δ_e on the overall heat transfer results at saturated film boiling conditions.

Heat Transfer Results at Subcooled Conditions.

Figs. 5.14 and 5.15 show the dependence of $(Nu/Re_l^{1/2})(\mu_v/\mu_l)$ on the Froude number with subcooling ΔT_{sub} as the parameter at superheat ΔT_{sup} of 600 °C and 1200 °C respectively. These two figures indicate that the effect of subcooling is more remarkable for the larger Froude number Fr and the smaller superheat ΔT_{sup} .

Based on the discussion in the previous chapters, the following two heat transfer characteristics are the best choices to correlate the subcooled film boiling heat transfer

$$C_{sat,N} = Nu \left(\frac{Sp'}{ArMc} \right)^{1/4} \quad \text{for pool film boiling} \quad (5.4.4) ,$$

$$C_{sub,F} = \frac{Nu}{Re_l^{1/2}} \frac{\mu_v}{\mu_l} \frac{Sp'}{Sc' Pr_l^{1/2}} \quad \text{for forced convection} \quad (5.4.5) ,$$

Fig. 5.14 The relationship between $(Nu/Re_l^{1/2})(\mu_w/\mu_l)$ and Froude number Fr at subcooled film boiling conditions with $\Delta T_{sup} = 600 \text{ } ^\circ\text{C}$.

Fig. 5.15 The relationship between $(Nu/Re_l^{1/2})(\mu_w/\mu_l)$ and Froude number Fr at subcooled film boiling conditions with $\Delta T_{sup} = 1200 \text{ } ^\circ\text{C}$.

Fig. 5.16 The dependence of $C_{sub,N}$ and $C_{sub,F}$ on Froude number Fr at subcooled film boiling conditions.

Fig. 5.17 The dependence of $C_{sub,N}$ on subcooling ΔT_{sub} in natural free convection film boiling.

Fig. 5.18 The dependence of $C_{sat,F}$ and $C_{sub,F}$ on subcooling ΔT_{sub} in forced convection film boiling.

where Mc in Eq. (5.4.4) is a complicated function of subcooling ΔT_{sub} , superheat ΔT_{sup} and sphere diameter d , which is given by Eq. (4.3.1). The variation of these two dimensionless groups are shown in Fig. 5.16 at three conditions. It is obvious that $C_{sub,N}$ and $C_{sat,F}$ are essentially independent of subcooling ΔT_{sub} , superheat ΔT_{sup} and the Froude number Fr in free convection and forced convection film boiling regimes respectively.

For pool film boiling, the variation of $C_{sub,N}$ against subcooling ΔT_{sub} is plotted in Fig. 5.17 with superheat ΔT_{sup} as the parameter. When ΔT_{sub} equals 0, i.e. at a saturated condition, $C_{sub,N}$ equals 0.696, which is exactly the same as that obtained by Michiyosh (1988). However, it decreases slightly as the subcooling increases and reaches ~ 0.63 at 80 °C subcooling. In Michiyoshi's analysis, linear velocity and temperature profiles were assumed for the liquid layer, but in the present model, quadratic profiles were used. This could be the reason for the small difference in heat transfer between the present model and their analysis when subcooling exists.

In the case of forced convection, the variations of $C_{sub,F}$ and $C_{sat,F}$ against subcooling ΔT_{sub} are shown in Fig. 5.18 with the superheat ΔT_{sup} as the parameter. It indicates that the higher the superheat, the less the $C_{sat,F}$ depends on subcooling. This means that when the superheat is very high, the saturated forced convection correlation may be approximately used for the subcooled condition. It also shows that the lower the superheat, the earlier the $C_{sub,F}$ levels out (i.e. $C_{sub,F}$ tends to be a constant at smaller subcooling). The constant that the $C_{sub,F}$ tends to reach is about 1.45. In comparison, the $C_{sub,F}$ obtained by Shigechi, Ito and Nishikawa (1983) for a cylinder is 1.16, and the one obtained from Epstein and Hauser's (1980) analysis for a sphere and cylinder is 0.977.

The comparisons of the experimental data of this work with the predictions of the theory in terms of $C_{sub,F}$ are shown in Figs. 5.19 (a) and 5.19 (b) for two subcooling conditions respectively. The comparisons indicate that the theory agrees well with the experiment in the natural convection and the transition regimes. However, the experimental data are higher than the theoretical prediction in the forced convection regime. The experimental data also clearly exhibit a higher power dependency on the Reynolds number than the 0.5 that is predicted by the theory as given by Eq. (5.4.5).

When subcooling in forced convection conditions is high, the overall heat transfer result is very sensitive to the choice of the equivalent film thickness of the rear part because the film that covers the rear part of the sphere is thin. As indicated earlier, in our preliminary model, the assumption $\Delta_e = \Delta_s$ is completely arbitrary, so it certainly cannot predict the experiment well.

In conclusion, when there is no separation (i.e. for the pool film boiling), the two-phase laminar boundary-layer theory agrees well with the experiment, the existing pool film boiling analyses and correlations. This preliminary model also works well for saturated forced convection film boiling, even when there is separation. However, this preliminary model does not work well in the subcooled forced convection regime because it does not model the heat transfer in the rear part correctly. A model that considers the heat transfer in the rear separated region for subcooled forced convection film boiling, therefore, developed next.

Fig. 5.19(a) A comparison of 20 °C subcooled experimental data with the preliminary theory in terms of $C_{sub,F}$.

Fig. 5.19(b) A comparison of 30 °C subcooled experimental data with the preliminary theory in terms of $C_{sub,F}$.

5.5 A Turbulent Heat Transfer Model for the Separated Rear Region

As mentioned in Chapter 4 Section 1, the experimental observation shows that a thin vapor film covers all of the sphere in the case of subcooled forced convection. The film on the front part is always smooth, but wavy on the rear side. So, we may still use the laminar model to calculate the heat transfer for the front unseparated region, and use a turbulent eddy model to calculate the heat transfer in the rear separated region.

Theofanous, Houze and Brumfield (1976) studied the turbulent mass transfer at the free gas-liquid interface. By a dimensional analysis or an *unsteady mass transfer eddy model*, they obtained two mass transfer coefficient correlations, with the constants being determined by experiments. This model can be readily converted to a heat transfer situation, and Liang and Peter-Griffith (1994) did so by applying this model to calculate the condensation of steam in subcooled water. In the following, the same model will be extended to calculate the turbulent heat transfer across the vapor-water interface in the rear separated region when the Froude number Fr is larger than 6.

It is reasonable to consider the flow behind the sphere as large turbulent eddies, with length scale $l = r$ and velocity $u = cU$, where U is the main stream flow velocity and c is a constant. Because of the viscous dissipation, it may also be assumed that the large eddies break down to Kolmogorov microscale eddies with their length and velocity being

$$l_k = \left(\frac{v_l^3}{\epsilon} \right)^{1/4} \quad \text{and} \quad u_k = (v_l \epsilon)^{1/4} \quad (5.5.1).$$

Because of the existence of film on the sphere surface, the eddies behind the sphere behave like eddies in a free stream. Therefore, the dissipation rate may be related with the large turbulent eddies by the well known empirical formula [Tennekes and Lumley (1972)],

$$\varepsilon = u^3 / l \quad (5.5.2).$$

The small eddies are carried by the large eddies to the free vapor-water interface on the back of the sphere to participate in heat transfer from time $t = 0$ to $t_{\text{exp}} = r/u$. Following the analysis of Theofanous et al (1976), unsteady heat transfer into the small eddy is calculated by taking the velocity field around the small eddy to be that of the Fortescue-Pearson (1970) roll cell. The mathematical governing equation, then, is

$$\frac{\partial T}{\partial t} + \left(u_m \sin \frac{\pi x}{l_k} \right) \frac{\partial T}{\partial x} + \left(-\frac{\pi u_m y}{l_k} \cos \frac{\pi x}{l_k} \right) \frac{\partial T}{\partial y} = \alpha_l \frac{\partial^2 T}{\partial y^2} \quad (5.5.3),$$

where $u_m = 2^{1/2} u_k$. The initial and boundary conditions are

$$\begin{aligned} T &= T_i - T_\infty = 0 & \text{at } t = 0 &, \\ T &= 0 & \text{at } y = \infty &, \\ T &= \Delta T_{\text{sub}} & \text{at } y = 0, t > 0 &. \end{aligned}$$

Defining $\eta = y/\delta(x, t)$, the problem becomes a search for $T = T(\eta)$ with the following two ordinary differential equations

$$-T' \eta \xi = \alpha_l T'' \quad (5.5.4),$$

$$\frac{1}{2} \frac{\partial(\delta^2)}{\partial t} + \frac{u_m}{2} \sin \frac{\pi x}{l_k} \frac{\partial(\delta^2)}{\partial x} + \frac{\pi u_m}{l_k} \cos \frac{\pi x}{l_k} \delta^2 = \xi \quad (5.5.5),$$

where ξ is an arbitrary constant. The solution is

$$T = \Delta T_{\text{sub}} \operatorname{erfc} \left\{ \frac{y}{2} \left[\frac{\pi u_m}{\alpha_l l_k} \left(\coth \left(\frac{\pi u_m t}{l_k} \right) + \cos \left(\frac{\pi x}{l_k} \right) \right) \right]^{1/2} \right\} \quad (5.5.6).$$

Hence, the average heat transfer coefficient h is

$$h = \frac{1}{l_k t_{exp} \Delta T_{sub}} \int_{t=0}^{t_{exp}} \int_{x=0}^{l_k} k_f \frac{\partial T}{\partial y} \Big|_{y=0} dt dx$$

$$= 1.07 \rho_f C_{p,f} F(\tau) \left(\frac{\alpha_f u_k}{l_k} \right)^{1/2} \quad (5.5.7),$$

where

$$F(\tau) = \frac{\pi}{2\sqrt{2}\tau} \int_0^{\tau} \int_0^1 (\coth \pi x' + \cos \pi x')^{1/2} dx' dt' \quad (5.5.8),$$

and

$$\tau = \sqrt{2} t_{exp} u_k / l_k \quad (5.5.9).$$

In the case when $Fr > 6$, ($Fr^{1/2} > 2.5$), $Re > 10000$ and

$$\tau = \sqrt{2} \frac{r/u}{l_k/u_k} \gg 1 \quad (5.5.10).$$

So, $F(\tau)$ is constant and is equal to one.

As done by Theofanous et al (1976), the experimental constant of 0.25 is used instead of the analytical constant of 1.07. Finally, the heat transfer coefficient for the rear separated region is

$$h_{sep} = 0.25 \rho_f C_{p,f} \left(\frac{\alpha_f u_k}{l_k} \right)^{1/2} = 0.25 \rho_f C_{p,f} Pr_f^{-1/2} \left(\frac{v_f c^3 U^3}{r} \right)^{1/4}$$

$$= 0.25 \rho_f C_{p,f} Pr_f^{-1/2} (4 v_f^2 c^6 g^3 d)^{1/8} Fr^{3/8} \quad (5.5.11).$$

In this formula, the only unknown is the constant c , which turned out to be 0.32 ± 0.1 by matching the computation with the experimental data.

Eq. (5.5.11) is only good for forced convection film boiling when $Fr > 6.0$. Applying this model to the transition region (from $Fr = 2/9$ to 6) will result in a 10 ~ 20 % higher heat transfer in comparison to the experimental data. For a better prediction in the transition regime, the following equation is used

$$h_{\text{top}} = h_{Fr=6} \left(\frac{Fr}{6} \right)^n \quad (5.5.12),$$

where $h_{Fr=6}$ is calculated from Eq. (5.5.11) with $Fr = 6$, and n is $3/4$, which is determined by matching the calculated Nusselt number with experimental data in the transition regime.

The results of the new model are compared with the experiment data in Figs. 5.20 (a) and 5.20 (b) for the cases of 20 °C and 30 °C subcooling respectively. The new model agrees with the experiment much better than the preliminary model of the previous section. However the experimental data still show a little higher power dependency on the velocity (or the Froude number) than the theory. There may be two reasons for this discrepancy. First, the heat transfer in the unseparated front part may also be affected by the turbulence in the coming flow. Secondly, this rear turbulent model is not good enough to account for the heat transfer perfectly, because the flow and vapor film in this separated region is so complicated.

Since it is not clear to us which one of the two is the main reason, we tried to modify the rear turbulent model to match the experimental data better just by increasing the power on the Froude number in the Eq. (5.5.11). The modified heat transfer coefficient formula is

$$h_t = 0.25 \rho_l C_{p,l} Pr_l^{-1/2} (4v_l^2 c^6 g^3 d)^{1/8} Fr^n \quad (5.5.11)',$$

where $n = 0.45$ and $c = 0.27$. The results of the modified turbulent model are compared with the experiment data in Figs. 5.21 (a) and 5.21 (b) for the cases of 20

°C and 30 °C subcooling respectively. They show that the slopes of the modified turbulent model agrees better with the experimental data in the forced convection regime.

Coupled with the turbulent model, the variation of the two predicted heat transfer characteristics, $C_{sub,N}$ and $C_{sub,F}$, are shown in Fig. 5.22. It indicates that $C_{sub,N}$ and $C_{sub,F}$ are still essentially independent of subcooling ΔT_{sub} and superheat ΔT_{sup} in the free convection and forced convection regimes respectively.

In conclusion, with the laminar model for the front unseparated film and the turbulent eddy model for the rear separated region, the combined theoretical model can predict well all the cases of single-phase film boiling.

Fig. 5.20(a) A comparison of 20 °C subcooled experimental data with the turbulent model theory in terms of $C_{sub,F}$.

Fig. 5.20(b) A comparison of 30 °C subcooled experimental data with the turbulent model theory in terms of $C_{\text{sub},F}$.

Fig. 5.21(a) A comparison of 20 °C subcooled experimental data with the modified turbulent model theory in terms of $C_{\text{sub},F}$.

Fig. 5.21(b) A comparison of 30 °C subcooled experimental data with the modified turbulent model theory in terms of $C_{sub,F}$.

Fig. 5.22 The dependence of $C_{sub,N}$ and $C_{sub,F}$ of modified turbulent model on Froude number Fr at subcooled film boiling conditions.

Chapter 6

A General Correlation for Film Boiling in Single-Phase Flows

6.1. Constructing the General Correlation

The experimental data presented in Chapter 4 and theoretical analysis in Chapter 5 indicate that the following "partial " correlations could be well used to correlate and predict the film boiling heat transfer in particular regimes.

Pool Film Boiling Correlation

$$\text{Nu}_p/(1+2/\text{Nu}_p) = K_c(d') (Ar/Sp')^{1/4} M_c^{1/4} \quad (6.1.1)$$

$$M_c = E^3/[1+E/(Sp'Pr)] / (RPrSp')^2$$

$$E = (A + C B^{1/2})^{1/3} + (A - C B^{1/2})^{1/3} + (1/3) Sc^*$$

$$A = (1/27)Sc^{*3} + (1/3)R^2Sp'PrSc^* + (1/4)R^2Sp'^2Pr^2$$

$$B = (-4/27)Sc^{*2} + (2/3)Sp'PrSc^* - (32/27)Sp'PrR^2 \\ + (1/4)Sp'^2Pr^2 + (2/27)Sc^{*3}/R^2$$

$$C = (1/2)R^2Sp'Pr, \quad Sc^* = 0.93 Pr^{0.22} C_{pl}\Delta T_{sub}/h_{fg}'$$

$$K_c(d') = 0.5d'^{-1/4}, \quad \text{for } d' < 0.14$$

$$K_c(d') = 0.86/(1+0.28d'), \quad \text{for } 0.14 < d' < 1.25$$

$$K_c(d') = 2.4d'/(1+3.0d'), \quad \text{for } 1.25 < d' < 6.6$$

$$K_c(d') = 0.47d'^{1/4}, \quad \text{for } d' > 6.6$$

This is applicable for both saturated and subcooled conditions as long as $Fr^{1/2}$ is smaller than 0.2. The analytical derivation of the base formula of this correlation is given in Appendix A. The $K_c(d')$ ranges from 0.70 to 0.78 as sphere diameter changes from 6.0 to 19.0 mm.

At the saturated condition, $M_c^{1/4}$ is about 0.86, so $[K_c(d')M_c^{1/4}]$ varies from

0.6 to 0.67, and thus Eq. (6.1.1) becomes

$$Nu_p = 0.64 (Ar/Sp)^{1/4} \quad (6.1.1)'$$

Forced Convection Film Boiling At Saturated Conditions

$$Nu_s = 0.5 Re_i^{1/2} \frac{\mu_l}{\mu_v} \left(\frac{KR^d}{Sp'} \right)^{1/4} \quad (6.1.2).$$

Forced Convection Film Boiling At Subcooled Conditions

$$Nu_f = Nu_s + 0.072 Re_i^{0.77} Pr_i^{0.5} \frac{\mu_l}{\mu_v} \frac{Sc'}{Sp'} \quad (6.1.3).$$

Eq. (6.1.3) coupled with Eq. (6.1.2) could essentially predicate all the forced convection film boiling ($Fr^{1/2} > 2.5$), including all the saturated, small subcooled and large subcooled cases. The comparisons of the forced convection film boiling experimental data with the predication of Eq. (6.1.3) are given in Fig. 6.1, 6.2 and 6.3 for saturated, small subcooling ($0 < \Delta T_{sub} < 20^\circ C$), and large subcooling respectively. The agreements are good in all the three cases.

Now we are ready to construct a general correlation. After some trial and error, we found that Eq. (6.1.1) and Eq. (6.1.3) could be combined in the following way for film boiling on spheres in all regimes:

Combined Correlation

$$Nu = [Nu_p^5 + \{F(Fr)Nu_f\}^5]^{1/5} \quad (6.1.4)$$

where the empirical function

$$F(Fr) = 1 - \frac{0.2}{1 + |Fr^{0.5} - 1|}$$

Fig. 6.1 Forced convection data obtained in saturated water flow are compared with Eq. (6.1.2).

Fig. 6.2 Forced convection data obtained in little subcooled ($0 \sim 20^\circ\text{C}$) water flow are compared with Eq. (6.1.3).

Fig. 6.3 Forced convection data obtained in large subcooled ($> 20^\circ\text{C}$) water flow are compared with Eq. (6.1.3).

Fig. 6.4 Data obtained at all conditions are compared with correlation (6.1.4).

is introduced to correlate the transition data. Figure 6.4 gives the comparison of the experimental data obtained at all conditions with the correlation (6.1.4), and it shows that most of the data points are in a band within $\pm 15\%$.

6.2. Comparisons of the General Correlation With the Theory

The Nusselt number calculated from the general correlation Eq. (6.1.4), at saturated condition, are compared with the theory and experiment in terms of $C_{sat,N}$ and $C_{sat,F}$, in Figs. 6.5 and 6.6 respectively. For subcooled cases, with the modified turbulent model Eq. (5.5. 11)', the comparisons are given in Figs. 6.7 (a) and 6.7 (b) in terms of $C_{sub,F}$. They all show good agreement among the experiment, theory and correlation. For subcooled cases, with the turbulent model Eq. (5.5.11), comparisons are given in Figs 6.8 (a) and 6.8 (b).

Fig. 6.5 A comparison of the general correlation with the theory and experiment in terms of $C_{sat,N}$ at saturated conditions.

Fig. 6.6

A comparison of the general correlation with the theory and the experiment in terms of $C_{sat,F}$ at saturated conditions.

Fig. 6.7 (a)

A comparison of general correlation with the modified turbulent model theory and the experiment at 20 °C subcooling.

Fig. 6.7 (b) A comparison of general correlation with the modified turbulent model theory and the experiment at 30 °C subcooling.

Fig. 6.8 (a) A comparison of general correlation with the turbulent model theory and the experiment at 20 °C subcooling.

Fig. 6.8 (b) A comparison of general correlation with the turbulent model theory and the experiment at 30 °C subcooling.

6.3. Comparisons of the General Correlation with Others

In the case of saturated pool film boiling, the comparison is given in Fig. 6.9; it shows that our correlation is a little lower than Hendrik's correlation (which is for sphere) and a little higher than others. For the subcooled pool film boiling, the comparison at the sphere temperature of 760 °C is shown in Fig. 4.17; it shows our correlation is lower than Farahat's and higher than all others.

For saturated forced convection, the comparison is given in Fig. 6.10. Our correlation is quite close to Bromley's correlation but is only about half of Epstein's et al (1980) correlation. Dhir's et al (1978) correlation, which is not valid for this high velocity case, gives much higher Nusselt number than the others.

Fig. 6.11 shows the comparison for the case of subcooled forced convection.

It indicates that our correlation agrees very well with Epstein's correlation at high sphere superheat, and about 10 ~ 20 % lower than it at low superheat (400~600 °C). Compared to our correlation, Dhir's correlation is lower when the sphere temperature is low and higher when the sphere temperature is high. Orozco's correlation is extremely low compared with the others.

Figs. 6.12 - 6.14 give more comparisons of our correlation with Epstein's and Dhir's correlations. The general conclusions are: (1) for complete saturated film boiling, both Epstein's and Dhir's correlations give much higher (more than a factor of two) predication than ours; (2) for large subcooling and high velocity, our correlation agrees with Epstein's correlation and is higher than Dhir's correlation. (3) At the transition regime from natural to forced convection with large subcooling, our correlation gives the lowest value and Epstein's gives the highest value.

Fig. 6.9 The present general correlation is compared with other saturated pool film boiling correlations.

Fig. 6.10 Comparison of our general correlation with other correlations at saturated forced convection condition ($U = 1.5$ m/s).

Fig. 6.11 Comparison of our general correlation with other correlations at highly subcooled forced convection condition ($\Delta T_{\text{sub}} = 30$ °C, $U = 1.5$ m/s).

Fig. 6.12 Comparison of our general correlation with other correlation at forced convection regime ($U = 1.5$ m/s).

Fig. 6.13 Comparison of our general correlation with other correlation at saturated condition.

Fig. 6.14 Comparison of our general correlation with other correlation at subcooled condition.

Chapter 7

Film Boiling in Two-Phase Flows

The film boiling from a sphere in two-phase flows has been studied in two parts, film boiling in upward two-phase flow and film boiling in downward two-phase flow. The experiments were carried out with 316 stainless steel spheres of 12.7 mm in diameter, saturated water and saturated steam in atmospheric pressure.

In both cases, prior to heat transfer tests, the void fractions (or water fraction) in the test section were investigated and measured "off line" by the X-ray radiography. The X-ray radiographs were also taken during the "on line" heat transfer runs. The two-phase flow that we characterize and communicate here is the two-phase flow that comes towards the test sphere, other than the disturbed flow behind the boiling spheres.

All two-phase film boiling heat transfer experiments were conducted by transient mode runs. The experimental data are interpreted and correlated in different ways, and the final experimental correlations are suggested.

7.1 Upward Two-Phase Flow Generation and Void Fraction Measurement

The upward two-phase flows were generated within the two-phase mixer by injecting saturated steam into a saturated upward water flow through the tube-needle assembly, as shown in Fig. 3.2. Only No.1 tube-needle assembly, see Section 3.2, was used in the upward two-phase experiment to generate the two-phase flows.

By varying the water and steam flow rates, different two-phase flows were generated in the test tube. In the our present experiment, the superficial water velocity

covers from 0.3 to 1.6 m/s, and the superficial steam velocity varies from 1.4 to 4.5 m/s. The void fractions in the test tube, which can not be determined theoretically, were detected and measured by the X-ray radiographic method, as presented in Section 3.5. For each combination of water and steam flow rates, the radiograph of the generated two-phase flow was taken, and the void fraction was determined by analyzing the darkness of the obtained X-ray pictures. Typical X-ray radiographs of the center part of the test tube obtained at different steam flow rates are shown in Figures 7.1 - 7.5 respectively. The void fractions in the test section were determined in an "off line" fashion, without the test sphere inside the test section.

The direct observation of two-phase flow inside the test tube and the pictures in Figure 7.1 - 7.5 indicate that there are three flow regimes: (1) with low water flow rate, especially when the steam flow rate is also low, the two-phase flow is "bubbly" flow with void fraction from 0.65 to 0.4; (2) with high water flow rate but low steam flow rate, the flow looks like a single-phase flow with lots of small bubbles; (3) with both high water flow rate and high steam flow rate, the two-phase flow appears as an upward mist-jet flow with void fraction about 0.5.

Because of the variant nature of the two-phase flows (both in time and in space), especially for the bubbly flow case, several X-ray pictures were taken for each water/steam flow combination to get the average void fraction for that flow condition. The obtained water fractions at superficial steam velocities of 1.4, 2.1, 2.7, 3.2 and 4.5 m/s are shown in Figures 7.6 - 7.10, respectively. The water fraction covers range from 35% to 80%.

After knowing the superficial water and steam velocities (V_w and V_s) and the void fraction of the two-phase flow (α) the water and the steam velocities can be calculated by

$$U_w = V_w/\alpha_w = V_w/(1-\alpha) \quad (7.1),$$

$$U_s = V_s/\alpha = V_s/(1-\alpha_w) \quad (7.2).$$

In our present experiment, the water velocity varies from 0.6 to 3.2 m/s and the steam

velocity varies from 3.0 to 9.0 m/s.

Typical radiographs of film boiling on sphere in two-phase flow are shown in Fig 7.11. For most two-phase cases, it is impossible to distinguish the vapor film from the steam void of the coming two-phase flow, except the cases when the void fraction is very low. For comparison, the radiographs of film boiling on a sphere in saturated single-phase flow are also shown in Fig. 7.11. It is very clear that the film configuration in single-phase flow observed from X-ray pictures are completely consistent with those from photo pictures.

7.2 Film Boiling in Upward Two-Phase Flows

Setting the steam flow rate at certain level, the film boiling heat transfer experiment was carried out by adjusting the water flow rate from one value to another. The water and steam velocities were calculated by Eqs. (7.1) and (7.2). Typical experiment data are tabled in Appendix E.

Choosing the superficial water velocity V_w as the characteristic flow velocity and defining the Reynolds number as

$$Re = \frac{V_w D}{\nu_l} \quad (7.3),$$

the experimental Nusselt numbers

$$Nu = \frac{hD}{k_l} \quad (7.4),$$

were plotted in Figures 7.12 and 7.13 for two steam flow rates respectively. As in the case of single-phase film boiling, the Nu number depends both on the superficial water velocity and the sphere superheat. The comparison of Fig. 7.12 with Fig. 7.13 also indicates that the higher the steam flow rate, the higher the Nu number. In terms of the saturated single-phase film boiling heat transfer characteristic

$$C'_{sat,F} = \frac{Nu}{Re^{1/2}} \frac{\mu_v}{\mu_l} \left(\frac{Sp'}{R^4 K} \right)^{1/4} \quad (7.5),$$

the data that are plotted in Figures 7.12 and 7.13 were re-plotted in Figures 7.14 and 7.15, respectively. In the same way, more data obtained with other steam flow rates are shown in Figure 7.16. They show that the sphere superheat effect is well correlated in all the upward two-phase cases as in the case of single-phase flow. However, the $C'_{sat,F}$ is no longer a constant of 0.5 as in the case of single-phase flow, and it varies from 0.45 to 0.95. They also show that the heat transfer characteristic $C'_{sat,F}$ decreases with Re and increases with the superficial steam velocity. It is obvious that the superficial water velocity is not a good characteristic velocity for the upward two-phase film boiling.

Instead of using the superficial water velocity, now, the water velocity (or called water phase velocity) U_w calculated from Eq. (7.1) is used as the characteristic velocity to define the Reynolds number

$$Re_l = \frac{U_w D}{\nu_l} \quad (7.3a).$$

Based on this definition, all the experiment data were re-plotted in Figure 7.17 in terms of

$$C_{sat,F} = \frac{Nu}{Re_l^{1/2}} \frac{\mu_v}{\mu_l} \left(\frac{Sp'}{R^4 K} \right)^{1/4} \quad (7.5a).$$

It shows that the effects of water flow and steam flow are correlated much better in this way, and the $C_{sat,F}$ ranges from 0.4 to 0.5, which is very close to the single-phase flow value of 0.5.

Figure 7.17 also clearly shows that there are two two-phase flow regimes. One is the upward mist-jet flow regime (include the nearly single-phase flow regime) with Reynolds number bigger than 7×10^4 . In this regime, the data are well correlated by

$$Nu = 0.42 Re_i^{1/2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \quad (7.6),$$

within a $\pm 10\%$ band. The other regime is the upward bubbly two-phase flow regime with the Reynolds number smaller than 7×10^4 .

In order to correlated the data in the bubbly flow regime better, we tried to adjust the exponent on the Reynolds number in Eq. (7.5a) to fit the experiment data. With an exponent of 0.2 on Reynolds number, all the experiment data are re-plotted in Figure 7.18. The thus resulted correlation is

$$Nu = 12.5 Re_i^{0.2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \quad (7.7),$$

which correlates the data within a $\pm 10\%$ band, for Reynolds number less than $1.2 \cdot 10^5$.

$V_w = 0.38$ m/s

0.55

0.79

0.97

1.26

1.61

Fig. 7.1 X-Ray radiographs obtained with superficial steam velocity at 1.4 m/s.

$V_w = 0.38$ m/s

0.55

0.79

0.97

1.26

1.61

Fig. 7.2 X-Ray radiographs obtained with superficial steam velocity at 2.1 m/s.

$V_w = 0.38 \text{ m/s}$

0.55

0.79

0.97

1.26

1.61

Fig. 7.3 X-Ray radiographs obtained with superficial steam velocity at 2.7 m/s.

$V_w = 0.38 \text{ m/s}$

0.55

0.79

0.97

1.26

1.61

Fig. 7.4 X-Ray radiographs obtained with superficial steam velocity at 3.2 m/s.

$V_w = 0.38 \text{ m/s}$

0.55

0.79

0.97

1.26

1.61

**Fig. 7.5 X-Ray radiographs obtained with
superficial steam velocity at 4.5 m/s.**

Fig. 7.6 Measured water fraction. The symbols represent measurements obtained at different time.

Fig. 7.7 Measured Water Fraction. The symbols represent measurements obtained at different time.

Fig. 7.8 Measured water fraction. The symbols represent measurements obtained at different time.

Fig. 7.9 Measured water fraction. The symbols represent measurements obtained at different time.

Fig. 7.10 Measured water fraction. The symbols represent measurements obtained at different time.

3.2/0.3/0.65

3.2/0.8/0.55

3.2/1.2/0.50

1.4/0.38/0.45

1.4/1.00/0.35

1.4/1.61/0.20

0.0/0.38/0.00

0.0/1.60/0.00

Fig. 7.11 X-Ray radiographs of film boiling from sphere in upward two- and single- phase flow. The numbers stand for: superficial Steam velocity/superficial water velocity/void fraction.

Fig. 7.12 Film boiling Nusselt number obtained in upward two-phase flow with superficial steam velocity at 1.4 m/s.

Fig. 7.13 Film boiling Nusselt number obtained in upward two-phase flow with superficial steam velocity at 4.5 m/s.

Fig. 7.14 Film boiling heat transfer data obtained in upward two-phase flow with superficial steam velocity at 1.4 m/s.

Fig. 7.15 Film boiling heat transfer data obtained in upward two-phase flow with superficial steam velocity at 4.5 m/s.

Fig. 7.16 All the film boiling heat transfer data obtained in upward two-phase flow, plotted in terms of superficial water velocity V_w . Repeated symbols represent data of different sphere superheats.

Fig. 7.17 All the film boiling heat transfer data obtained in upward two-phase flow, plotted in terms of water velocity U_w . Repeated symbols represent data of different sphere superheats.

Fig. 7.18 All the film boiling heat transfer data obtained in upward two-phase flow, plotted in terms of Eq. (7.7). Repeated symbols represent data of different sphere superheats.

7.3 Downward Two-Phase Flow Generation, Void Fraction Measurement and Prediction

The downward two-phase flows were generated by injecting water jets into a downward steam flow. All the three tube-needle assemblies (see Section 3.2) were used as the water injector. Both water and steam flow rates can be adjusted independently to produce the downward two-phase flows. The superficial water velocity varies from 0.07 to 1.6 m/s, and the superficial steam velocity changes from 1.0 to 4.5 m/s.

The water jet velocity and the void fraction at the test sphere location can be predicated based on the geometry of the tube-needle assembly and the acceleration of the water due to the gravity and the drag force of the steam flow. The equations that govern the acceleration of the water drop are

$$\frac{dU_w}{dt} = g + \frac{3 C_D \rho_v}{4 D \rho_w} |U_s - U_w| (U_s - U_w) \quad (7.8),$$

$$\frac{dh}{dt} = U_w \quad (7.9).$$

where D is the sphere diameter and C_D is the drag coefficient.

In the computation, the inner diameter of the tube-needle was used to approximate the diameter of the water drops in the test section, and a value of 0.5 was chosen for the drag coefficient C_D . The drag force of the water-steam interaction is much smaller than the gravity in our case. So the calculation is not sensitive to the selections of the drop diameter and the drag coefficient. Once the water jet velocity at the tube outlet $U_{w,0}$ (calculated based on the water flow rate and tube-needle inner diameter) and the water drop-jet velocity at the test sphere location U_w are known, the water fraction and the void fraction can be calculated as follows

$$\alpha_w = \frac{U_{w,0} A_{hole}}{U_w A_{base}} \quad (7.10),$$

$$\alpha = 1 - \alpha_w \quad (7.11).$$

The void fractions of the downward two-phase flow are also measured by the X-ray radiography as in the case of upward two-phase flow. Figures 7.19, 7.20 and 7.21 show the X-ray pictures of the downward two-phase flows at the center part of the test section (where the test sphere is located) with No.1, No.2 and No.3 tube-needle assemblies respectively. These radiographs are obtained with the superficial steam velocity at 4.5 m/s, and the radiographs with other superficial steam velocities 1.0, 1.4, 2.1 and 3.2 m/s are quite the same. These radiographs clearly show that when the water jet velocity is low, the two-phase flow looks more like a droplet flow; and when the water jet velocity is high, the two-phase flow shows more like a jet flow.

Figure 7.22 shows typical X-ray pictures of the sphere film boiling in downward two-phase flow. We cannot get detailed information of the vapor film on the sphere, although somewhat higher void fraction regions can be seen behind the spheres.

The measured and the predicted void fractions are compared in Figures 7.23, 7.24 and 7.25 for the two-phase flows generated by No.1, No.2 and No.3 tube-needle assemblies, respectively. They show that the predicted void fractions reasonably agree with the X-ray measurements. Therefore, Eqs. (7.8) - (7.11) can be used to calculate the water velocity and void fraction of the two-phase flow at the test sphere location.

With all combinations of the water flow rate and steam flow rate, the downward two-phase flows cover void fraction range from 70% to 95%, with the water velocity ranges from 1.9 to 6.5 m/s and the steam velocity ranges from 1.1 to 9.0 m/s.

7.4 Film Boiling of Downward Two-Phase Flows

All the three tube-needle assemblies are used in the downward two-phase flow film boiling heat transfer experiment. With each tube-needle assembly, the superficial steam flow rate were first set at one of following values 1.0, 1.4, 2.1, 3.2 and 4.5 m/s, then transient mode runs were carried out by varying the water flow rate. Typical downward two-phase heat transfer experimental data are tabled in Appendix F.

Defining the Reynolds number with the superficial water velocity, as in Eq. (7.3), the experiment data are plotted in terms of the saturated single-phase film boiling heat transfer characteristic $C'_{sat,F}$ in Figures 7.26, 7.27 and 7.28 for No.1, No.2 and No.3 tube-needle assemblies respectively. In these figures the superficial steam velocity are shown as the parameters, and the repeated symbols represent data of different sphere superheats (from 400 to 800 °C). These Figures indicate that, as in the case of upward single- and two-phase flows, the effect of sphere superheat is well correlated in all the cases of downward two-phase flows. They also show that the heat transfer characteristic $C'_{sat,F}$ is no longer a constant of 0.5, and it varies from 0.7 to 1.3. When the Reynolds number is high, the steam velocity has very little or no effect on the $C'_{sat,F}$; but when the Reynolds number is small, the steam velocity has a significant effect on the $C'_{sat,F}$. Comparing the three figures that obtained with different tube-needle assemblies, it is obvious that the smaller the tube inner diameter, the higher the heat transfer characteristic $C'_{sat,F}$.

As in the case of upward two-phase flow, we now define the Reynolds number Re_j based on the water velocity at the test sphere location, which is calculated by Eqs. (7.8) to (7.11). In terms of the heat transfer characteristic $C_{sat,F}$, the data obtained with superficial steam velocities at 1.0 and 4.5 m/s are plotted in Figure 7.29 and 7.30 respectively, with the number of tube-needle assembly as the parameter. It is obvious that: the $C_{sat,F}$ is still not a constant; it increases with the Reynolds number; the smaller the diameter of the tube diameter, the higher is the $C_{sat,F}$; the steam velocity almost has no influence on the $C_{sat,F}$. In general, based on the water-phase

characteristic to correlate the downward two-phase flow heat transfer data.

There are reasons for this. First, in the upward two-phase flows the water fractions are higher (from 35% to 80%) than those in the downward two-phase flows (from 5% to 30%). Secondly, in the case of upward two-phase flows, the gravity is opposite to the flow direction, so it helps to accumulate more water at the front stagnation point than in the case of downward two-phase flows. So for the film boiling in upward two-phase flows, from the sphere point of view, it is like in a saturated single-phase water flow. This is why the saturated single-phase film boiling heat transfer characteristic $C_{sat,F}$ still works well for the film boiling in upward two-phase flows. On the other hand, in the downward two-phase flows, there is no much water accumulated at the front stagnation point, and the film boiling may be thought as in a low density single-phase flow with an equivalent density

$$\rho_e = \alpha_w \rho_{water} \quad (7.12).$$

Substituting this equivalent density as the liquid phase density into the heat transfer characteristic $C_{sat,F}$, Eq. (7.5a), we have new heat transfer characteristic

$$C_{D,two} = \frac{Nu}{Re_l^{1/2}} \frac{\mu_r}{\mu_l} \left(\frac{Sp'}{R^* K} \right)^{1/4} \frac{1}{\alpha_w^{1/4}} \quad (7.13),$$

which is just the $C_{sat,F}$ divided by a factor of $\alpha_w^{1/4}$.

With the new heat transfer characteristic $C_{D,two}$, the experiment data that plotted in Figures 7.29 and 7.30 are re-plotted in Figures 7.31 and 7.32 respectively. In the same way, Figure 7.33 shows all the data of downward two-phase flow, which represents all the data obtained with different sphere superheats, different tube-needle assemblies, and different steam and water velocities. It is obvious that this heat transfer characteristic correlates well all the effects, except that it increases slightly with the Reynolds number Re_l (or water-phase velocity). Moreover, the $C_{D,two}$ only varies only from 0.4 to 0.5, and it is very close to the saturated single-phase value

$C_{\text{sat},F} = 0.5$. So, in an approximation sense, the heat transfer characteristic Eq. (7.13) can be used to correlate the downward two-phase film boiling heat transfer in the following formula

$$Nu = 0.45 Re_i^{1/2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \alpha_w^{1/4} \quad (7.14),$$

within a $\pm 20\%$ band.

The systematically increasing of the $C_{D,\text{two}}$ with Re_i suggests us that the exponent on the Reynolds number in Es. (7.13) or (7.14) should be higher. By fitting the correlation with the experimental data, it was found that a power of 0.75 could be used instead of 0.5. In terms of the further modified new heat transfer characteristic

$$C_{D,\text{two}}^* = \frac{Nu}{Re_i^{0.75}} \frac{\mu_v}{\mu_l} \left(\frac{Sp'}{R^* K} \right)^{1/4} \frac{1}{\alpha_w^{1/4}} \quad (7.15),$$

all the downward two-phase data are shown in Figure 7.34. It correlates the data much better than $C_{D,\text{two}}$, and the corresponding correlation is

$$Nu = 0.0235 Re_i^{0.75} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \alpha_w^{1/4} \quad (7.16),$$

which correlates the data within a $\pm 10\%$ band.

7.5 Conclusion

The void fractions of upward and downward two-phase flow were measured by the X-ray radiography. The calculated void fraction of downward two-phase flow, which is based on the acceleration due to the gravity and steam drag, agrees well with the radiographic measurements.

The upward two-phase flows cover void fraction range from 0.2 to 0.65 with the water velocity from 0.6 to 3.2 m/s and the steam velocity from 3.0 to 9.0 m/s. When the water velocity is low, the flow appears like a bubbly flow; when the water velocity is high, the flow shows like a mist-jet flow. Based on the water velocity, the film boiling heat transfer data can be essentially correlated by the saturated single-phase heat transfer characteristic $C_{sat,F}$, with an equation of

$$Nu = 0.42 Re_i^{1/2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \quad (7.6).$$

The modified correlation

$$Nu = 12.5 Re_i^{0.2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp'} \right)^{1/4} \quad (7.7),$$

fits the experimental data better (within a $\pm 10\%$ band), for $3 \times 10^4 < Re_i < 1.2 \times 10^5$.

The downward two-phase flows are droplet-jet flows and cover void fraction from 0.7 to 0.95 with water velocity from 1.9 to 6.5 m/s and steam velocity from 1.1 to 9.0 m/s. Based on the water phase velocity and with an adjustment factor $\alpha_w^{1/4}$, the saturated single-phase film boiling heat transfer characteristic $C_{sat,F}$ can also correlate all the downward two-phase film boiling heat transfer data within a $\pm 20\%$ band with the equation

$$Nu = 0.45 Re_i^{1/2} \frac{\mu_l}{\mu_w} \left(\frac{R^4 K}{Sp'} \right)^{1/4} \alpha_w^{1/4} \quad (7.14).$$

With an increase of the exponent on the Reynolds number (from 0.5 to 0.75), the following Equation fits the experimental data even better (within a $\pm 10\%$ band)

$$Nu = 0.0235 Re_i^{0.75} \frac{\mu_l}{\mu_w} \left(\frac{R^4 K}{Sp'} \right)^{1/4} \alpha_w^{1/4} \quad (7.16).$$

$U_{w,0} = 0.67 \text{ m/s}$

1.36

2.19

2.96

4.05

5.78

Fig. 7.19 X-Ray radiographs of downwrd two-phase flow generated by No. 1 tube-needle assembly.

$U_{w,0} = 0.56 \text{ m/s}$

1.64

2.02

2.93

3.40

5.69

Fig. 7.20 X-Ray radiographs of downwrd two-phase flow generated by No. 2 tube-needle assembly.

$U_{w,0} = 0.59 \text{ m/s}$

1.22

2.04

2.51

3.58

4.65

Fig. 7.21 X-Ray radiographs of downwrd two-phase flow generated by No. 3 tube-needle assembly.

$U_{w,o} = 0.40 \text{ m/s}$

$U_{w,o} = 0.80 \text{ m/s}$

$U_{w,o} = 4.18 \text{ m/s}$

$U_{w,o} = 5.14 \text{ m/s}$

Fig. 7.22 X-Ray radiographs of film boiling on sphere in downward two-phase flow generated by No. 1 tube-needle assembly.

Fig. 7.23 Comparison of measured void fraction with the predicted

Fig. 7.24 Comparison of measured void fraction with the predicted

Fig. 7.25 Comparison of measured void fraction with the predicted

Fig. 7.26 Film boiling heat transfer data obtained in downward two-phase flow generated by No. 1 tube-needle assembly, plotted in terms of superficial water velocity V_w .

Fig. 7.27 Film boiling heat transfer data obtained in downward two-phase flow generated by No. 2 tube-needle assembly, plotted in terms of superficial water velocity V_w .

Fig. 7.28 Film boiling heat transfer data obtained in downward two-phase flow generated by No. 3 tube-needle assembly, plotted in terms of superficial water velocity V_w .

Fig. 7.29 Film boiling heat transfer data obtained in downward two-phase flow with superficial steam velocity at 1.0 m/s, plotted in terms of $C_{sat,F}$.

Fig. 7.30 Film boiling heat transfer data obtained in downward two-phase flow with superficial steam velocity at 4.5 m/s, plotted in terms of $C_{sat,F}$.

Fig. 7.31 Film boiling heat transfer data obtained in downward two-phase flow with superficial steam velocity at 1.0 m/s, plotted in terms of $C_{D,two}$.

Fig. 7.32 Film boiling heat transfer data obtained in downward two-phase flow with superficial steam velocity at 4.5 m/s, plotted in terms of $C_{D,two}$.

Fig. 7.33 All the film boiling heat transfer data obtained in downward two-phase flow, plotted in terms of $C_{D,two}$. Repeated symbols represent data from different tube-needle assemblies and with different sphere superheats.

Fig. 7.34 All the film boiling heat transfer data obtained in downward two-phase flow, plotted in terms of $C_{D,two}^*$. Repeated symbols represent data from different tube-needle assemblies and with different sphere superheats.

Chapter 8

Film Boiling From A Multi-Sphere Array

In the analysis of fuel-coolant interaction (or premixing), the concern about film boiling is from a group of spheres instead of one sphere. Obviously, the presence of solid spheres and the addition of vapor from the film boiling will change the local water flow velocity and the void fraction. It may be that the single sphere film boiling correlations are not applicable to the film boiling from a multi-sphere group. However, because the local velocity and void fraction around the sphere in question already represent all the effects, as long as they are known, the film boiling correlation for a single sphere in two-phase flow could be used to calculate the film boiling heat transfer from a sphere that is among a multi-sphere group.

In order to verify this, the "on-line" local velocity and void fraction should be measured with multi-sphere array present in the test section and film boiling on. But, it is impossible to get these measurements with the present X-ray radiographic method. However, in this work, as a preliminary step towards the problem of film boiling from multi-sphere arrays, the influence on heat transfer with the presence of four spheres in the front of the test sphere are investigated experimentally.

The experiments have been conducted in upward single-phase flow, upward two-phase flow and downward two-phase flow. The multi-sphere array is constructed as illustrated in Figure 8.1, with the back-center sphere as the test sphere. All these experiments are carried out with transient mode runs. All the four fore-spheres and the test sphere were heated up and experience cool down transients (in film boiling condition) from 900 °C to quench. In all the multi-sphere experiments, the runs were conducted and the data from the test sphere were reduced in the same way as in the corresponding single sphere cases, and the reduced heat transfer characteristics are compared with those obtained with a single sphere.

8.1 Film Boiling From A Multi-Sphere Array in Saturated Single-Phase Flows

For film boiling from a multi-sphere array in saturated single-phase flows, the superficial water velocity is used to characterize the flow. The experiment data is shown in Figure 8.2 in terms of the pool film correlation ($1/4$ -power law). It shows that for $Fr^{1/2} < 1.0$, the $Nu/(Ar/Sp)^{1/4}$ values are about 0.87, which is higher than the 0.64 that is obtained for a single sphere. The same data are re-plotted in Figure 8.3 in terms of forced convection film boiling correlation. For $Fr^{1/2} > 2.0$, the heat transfer characteristic is about 0.4, which is lower than the 0.5 that is obtain for a single sphere.

More clearly, the comparison is given in Figure 8.4 in terms of the ratios of the multi-sphere experimental data over the general correlation for a single sphere, Eq. (6.1.4). It is obvious that, in the pool film boiling regime, the multi-sphere data are about 20% higher than that of the single sphere. This is because the boilings on the fore-spheres make the convection in the water more turbulent and thus enhance the film boiling heat transfer on the test sphere. However, in the forced convection regime, the boiling on the fore-spheres creates some void fraction and, possibly, some superheated vapor in the water flow, so the film boiling heat transfer decreases, which is also about 20% .

8.2 Film Boiling From A Multi-Sphere Array in Upward Two-Phase Flows

As in the case of single-sphere film boiling in upward two-phase flows, the upward two-phase flow was created by injecting saturated steam into a saturated water flow through the No.1 tube-needle assembly. The runs were carried out in three subsets with the superficial steam velocity at 1.4, 2.7 and 4.5 m/s, respectively. For each subset, the superficial water velocity was changed from 0.3 to 1.4 m/s to carry out each run.

Again, as in the case of single-sphere upward two-phase flow, the data were reduced with water and steam properties at film temperatures and with the coming water-phase velocity as the characteristic velocity. In terms of Eq. (6.7), which is more applicable for high Reynolds number, the data are shown in Figure 8.5. In terms of Eq. (6.8), which is more proper for low Reynolds number, the same data are shown in Figure 8.6. In these figures, the repeated symbols represent the data of different sphere superheats. They show clearly that, as in the forced convection single-phase flow, the film boiling heat transfer decreases about 15% when four boiling spheres are placed in the front of the test sphere. The reason for this is that the boiling on the fore-spheres increases the void fraction and may produce superheated vapor in the front of the test sphere.

8.3 Film Boiling From A Multi-Sphere Array in Downward Two-Phase Flows

Basically, here we just repeat the film boiling heat transfer experiment in downward two-phase flow with four fore-spheres placed in the front of the test sphere, as shown in Figure 8.1. The experiment was carried out in three subsets with No.1, No.2 and No.3 tube-needle assemblies respectively. For each subset, the superficial steam velocity was set at certain value (1.0, 2.1 or 4.5 m/s) first, and then the water velocity was changed to make each run.

The experiment data are reduced in the same way as in the case of a single-sphere. The data obtained with No.1, No.2 and No.3 tube-needle assemblies are shown in Figures 8.7 to 8.9 respectively. For each case, the data are plotted in terms of Eq. (7.14) and Eq. (7.16) respectively in (a) and (b), and the repeated symbols represent the data with different sphere superheats. These data indicate that, in all cases, the film boiling heat transfer increases significantly with four boiling spheres placed in the front of the test sphere. They also show that the bigger the tube-needle inner diameter, the higher is the increase. The reason of this enhancement is quite obvious: the presence of the fore-spheres concentrates more water towards the front of the test sphere, which yields high water fraction and high water flow velocity.

8.4 Conclusion

The film boiling from a multi-sphere group has also been investigated through a five-sphere array with four spheres placed in the front of the test sphere. The experiments indicate that: (1) the presence of boiling spheres in the front of the test sphere decreases the film boiling heat transfer in the upward two-phase flow and increases the heat transfer in downward two-phase flow; and (2) in a saturated single-phase upward flow, the presence of boiling spheres in the front of the test sphere increases the heat transfer in pool film boiling regime and decreases the heat transfer in forced convection film boiling regime. These trends are consistent with the expectation using the single-sphere correlation and the theoretical interpretation in conjunction with the flow field changes due to the presence of the four spheres ahead of the test sphere.

Fig. 8.1 Schematic of the multi-sphere array.

Fig. 8.2 Heat transfer data of film boiling from multi-sphere array in saturated single-phase flow, plotted in terms of pool film boiling correlation.

Fig. 8.3 Heat transfer data of film boiling from multi-sphere array in saturated single-phase flow, plotted in terms of forced convection film boiling correlation.

Fig. 8.4 Comparison of the data from multi-sphere array in single-phase flow with the general correlation Eq. (6.1.4).

Fig. 8.5 Heat transfer data of film boiling from multi-sphere array in upward two-phase flow, plotted in terms of Eq. (7.6).

Fig. 8.6 Heat transfer data of film boiling from multi-sphere array in upward two-phase flow, plotted in terms of Eq. (7.7).

Fig. 8.7(a) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 1 tube-needle assembly, plotted in terms of Eq. (7.14).

Fig. 8.7(b) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 1 tube-needle assembly, plotted in terms of Eq. (7.16).

Fig. 8.8(a) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 2 tube-needle assembly, plotted in terms of Eq. (7.14).

Fig. 8.8(b) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 2 tube-needle assembly, plotted in terms of Eq. (7.16).

Fig. 8.9(a) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 3 tube-needle assembly, plotted in terms of Eq. (7.14).

Fig. 8.9(b) Heat transfer data of film boiling from multi-sphere array in downward two-phase flow generated by No. 3 tube-needle assembly, plotted in terms of Eq. (7.16).

Chapter 9

Conclusions

1. A flexible experimental system has been built and robust experimental techniques have been developed to test film boiling from very high temperature spheres in a wide variety of two-phase flows.
2. Single-phase film boiling experiments have been conducted systematically (with liquid subcooling from 0 to 40 °C, liquid velocity from 0 to 2 m/s, sphere superheat from 200 to 900 °C, sphere diameter from 6 to 19 mm) to obtain high quality heat transfer data. A general correlation has been developed.
3. Utilizing an two-phase laminar boundary-layer model for the unseparated front film region and a turbulent eddy model for the separated rear region, a theoretical model has been developed to predict the film boiling heat transfer in all single-phase regimes.
4. Two-phase film boiling data have been obtained in a wide variety of flow regimes. The upward two-phase flows cover void fraction from 0.2 to 0.65, water velocity from 0.6 to 3.2 m/s, and steam velocity from 3.0 to 9.0 m/s. The downward two-phase flows cover void fraction from 0.7 to 0.95, water velocity from 1.9 to 6.5 m/s, and steam velocity from 1.1 to 9.0 m/s. The saturated single-phase heat transfer correlation is found to be applicable to the two-phase film boiling data by making use of the actual water velocity (water phase velocity), and an adjustment factor of $(1-\alpha)^{1/4}$ (with α being the void fraction) for downward flow case only.
5. Preliminary experiments on film boiling from a multi-sphere array indicate that: (1) the presence of boiling spheres in the front of the test sphere decrease the heat transfer in upward two-phase flows and increase the heat transfer in downward two-phase flows; and (2) in a saturated single-phase upward flow, the presence of boiling spheres in the front of the test sphere increases the heat transfer in pool film boiling and decreases the heat transfer in forced convection film boiling.

References

1. Amarasooriya, W.H. Amarasooriya and T.G. Theofanous, "Premixing of Steam Explosions: A Three-Fluid Model," Nuclear Engineering and Design 126, pp 23-39 (1991).
2. Aziz, A. Aziz, G.F. Hewitt, and D.B.R. Kenning, "Heat Transfer Regimes in Forced-Convection Film Boiling on Spheres", Proc. 8th Int. Heat Transfer Conference, pp 2149--2154, (1986).
3. Banchemo, J.T. Banchemo, G.E. Barker, and R.H. Boll, "Stable Film Boiling of Liquid Oxygen Outside Single Horizontal Tubes and Wires", Heat Transfer, Chemical Engineering Progress, Symposium Series, American Institute of Chemical Engineers, New York, Vol. 51, No. 17, p. 21, (1955).
4. Barron, R.F. Barron, and A.K. Gorgolis, "Film Boiling of Liquid Nitrogen on A Sphere in An Enclosure", Adv. Cryog. Eng. Vol. 23, pp 305-312, (1977).
5. Baumeister, K.J. Baumeister and T.D. Hamill, "Laminar Flow Analysis of Film Boiling From A Horizontal Wire", NASA TN D-4035, (1967).
6. Bradfield, W.S. Bradfield, "Periodic Contact Behavior in Film Boiling", I&EC Fundamentals, pp 200--204, (1966).
7. Bradfield, W.S. Bradfield, "On the Effect of Subcooling on Wall Superheat in Pool Boiling", J. of Heat Transfer, Transactions of the ASME, August, pp 269-270, (1967).
8. Breen, B.P. Breen and J.W. Westwater, "Effect of Diameter of Horizontal Tubes on Film Boiling Heat Transfer", Chemical Engineering Progress, Vol. 58, No. 7, July (1962).
9. Bromley, L. A. Bromley, "Heat Transfer in Stable Film Boiling", Chemical Engineering Progress, Vol. 46, No. 5, pp 221-227, May (1950).
10. Bromley, L.A. Bromley, N.R. Leroy, and J.A. Robbers, "Heat Transfer in Forced Convection Film Boiling", Industrial and Engineering Chemistry, Vol. 45, No. 12, pp 2639--2646, (1953).
11. Bui, T. D. Bui and V.K. Dhir, "Film Boiling Heat Transfer on An Isothermal Vertical Surface", Transactions of the ASME, J. of Heat Transfer, Vol. 107, pp 764-771, (1985).
12. Cess, R.D. Cess and E.M. Sparrow, "Subcooled Forced-Convection Film Boiling on A Flat Plate", J. of Heat Transfer, Trans. of The ASME, pp 377-379, August, (1961).

13. Cess, R.D. Cess, and E.M. Sparrow, "Film Boiling in A Forced-Convection Boundary-Layer Flow", J. of Heat Transfer, Trans. of ASME, pp 370-376, August, (1961).
14. Cess, R.D. Cess, "Forced-Convection Film Boiling on A Flat Plate With Uniform Surface Heat Flux", J. of Heat Transfer, Trans. of the ASME, p 395, (1962).
15. Chou, X.S. Chou, and L.C. Witte, "A Theoretical Model for Flow Film Boiling Across Horizontal Cylinders", AIAA 92-4051, National Heat Transfer Conference, San Diego, (1992).
16. Dhir, V. K. Dhir, G.P. Purohit, "Subcooled Film-Boiling Heat Transfer From Spheres", Nuclear Engineering and Design, Vol. 47, pp 49-66, (1978).
17. Dix, D. Dix and J. Orozco, "Film Boiling Heat Transfer From A Sphere in Natural and Forced Convection of Freon-113", Experimental Heat Transfer, Vol. 3, pp 129-148, (1990).
18. Epstein, M. Epstein and G.M. Hauser, "Subcooled Forced-Convection Film Boiling in the Forward Stagnation Region of A Sphere or Cylinder", Int. J. of Heat and Mass Transfer, Vol. 23, pp 197-189, (1980).
19. Farahat, M.M. Farahat, and T.N. Nasr, "Film Boiling Heat Transfer From Small Spherical Particles to Saturated Sodium and Water", Atomkernenergie (ATKE) Bd. 26, Lfg. 2, pp 99-102, (1975).
20. Farahat, M.M. Farahat, and T.N. Nasr, "Natural Convection Film Boiling From Spheres to Saturated Liquid, An Integral Approach", Int. J. Heat Mass Transfer, Vol. 21, pp 256--258, (1978).
21. Farahat, M.M.K. Farahat and D.T. Eggen, "Pool Boiling in Subcooled Sodium at Atmospheric Pressure", Nuclear Science and Engineering, Vol. 53, pp 240-253, (1974).
22. Farahat, M.M.K. Farahat, "Transient-Boiling Heat Transfer From Spheres To Sodium", ANL-7909, (1972).
23. Fodemski, T.R. Fodemski, "The Influence of Liquid Viscosity and System Pressure on Stagnation Point Vapor Thickness During Forced-Convection Film Boiling", Int. J of Heat and Mass Transfer, Vol. 28, No. 1, pp 69-80, (1985).
24. Fodemski, T.R. Fodemski, W.B. Hall, "Forced Convection Film Boiling on A Sphere in (a) Sub-Cooled or (b) Superheated Liquid", Proc. 7th Heat Transfer Conf. Munchen 4, pp 375-379, (1982).
25. Fodemski, T.R. Fodemski, "Forced Convection Film Boiling in the Stagnation Region of A Molten Drop and Its Application to Vapor Explosions", Int. J. Heat & Mass Transfer, Vol. 35, No. 8, pp 2005-2016, (1992).

26. Fortescue, G.E. Fortescue and J.R.A. Pearson, "On Gas Absorption Into A Turbulent Liquid", Chem. Engng Sci. Vol. 22, pp 1163-1176, (1967).
27. Frederking, T.H.K. Frederking, and J.A. Clark, "Natural Convection Film Boiling on A Sphere", Adv. Cryog. Eng. Vol.8, pp 501--506, (1963).
28. Frederking, T.H.K. Frederking, R.C. Chapman, and S. Wang, "Heat Transport and Fluid Motion During Cooldown of Single Bodies to Low Temperatures", Adv. Cryog. Eng., Vol 10, Part 2, pp 353-360, (1965).
29. Grigoriev, V.A. Grigoriev, V.V. Klimenko and A.G. Shelepen, "Pool Film Boiling From Submerged Spheres", Proc. of 7th Int. Heat Transfer Conf., Munich, Germany, Vol. 4, pp 387-392, (1982).
30. Hamill, T.D. Hamill and K.J. Baumeister, "Effect of Subcooling and Radiation on Film-Boiling Heat Transfer From A Flat Plate", NASA TN D-3925, (1967).
31. Hendricks, R.C. Hendricks and K. J. Baumeister, "Film Boiling From Submerged Spheres", NASA TN-5124, (1969).
32. Hendricks, R.C. Hendricks and K. J. Baumeister, "Similarity and Curvature Effects in Pool Film Boiling", Proc. of the 4th Int. Heat Transfer Conf., Paris-Versailles, Vol.5, B3.7, (1970).
33. Hesson, J.C. Hesson, and L.C. Witte, "Comment on 'Film Boiling Heat Transfer Around A Sphere in Forced Convection' by K. Kobayasi", J. Nuclear Science and Technology, Vol. 3 , No. 10, pp 448--449, (1966).
34. Hishikawa, K. Hishikawa, and T. Ito, "Two-phase Boundary-Layer Treatment of Free-convection Film Boiling", Int. J. of Heat Mass Transfer, Vol. 9, pp 103-115. (1966).
35. Hsiao, K.H. Hsiao, L.C. Witte, J.E. Cox, "Transient Film Boiling From A Moving Sphere", Int. J. of Heat and Mass Transfer, Vol. 18, pp 1343-1350, (1975).
36. Irving, M.E. Irving and J.W. Westwater, "Limitations for Obtaining Boiling Curves by the Quenching Method with Spheres", Proc. of 8th Int. Heat Transfer Conf., Vol. 4, pp 2061-2066, (1986).
37. Ito, T. Ito, K. Nishikawa, and T. Shigechi, "Forced Convection Film Boiling Heat Transfer From A Horizontal Cylinder to Liquid Cross-flow Upward (1st Report, Saturated Liquid)", Bulletin of the JSME, Vol. 24, No. 198, pp 2107-2114, Dec., (1981).
38. Ito, Takehiro Ito, and Kaneyasu Hishikawa, " Two-Phase Boundary-layer Treatment of Forced-Convection Film Boiling", Int. J. of Heat and Mass Transfer, Vol. 9, pp 117-130, (1966).

39. Jacobs, H.R. Jacobs and R.F. Boehm, "An Analysis of Effects of Body Force and Forced Convection on Film Boiling", Proc. of Int. Heat Transfer, Vol.5, B3.9, pp 1-11, (1970).
40. Jacobson, R.N. Jacobson, and F.H. Shair, "Film Boiling From A Sphere During Forced Convection of Subcooled Water", I&EC Fundamentals Vol. 9, No. 1, (1970).
41. Klimenko, V.V. Klimenko, "Film Boiling on A Horizontal Plate-- New Correlation", Int. J. of Heat and Mass Transfer, Vol. 24, pp 69-79, (1981).
42. Kobayasi, K. Kobayasi, "Counter-Comment to Hesson & Witte's Comment on 'Film Boiling Heat Transfer Around A Sphere in Forced Convection'", J. of Nuclear Science and Technology, Vol. 3, No. 10, pp 449-450, (1966).
43. Kobayasi, K. Kobayasi, "Film Boiling Heat Transfer Around A Sphere in Forced Convection", Journal of Nuclear Science and Technology, Vol. 2, No.2, pp 62-67, (1965).
44. Liang, Kuo-Shing Liang and Peter-Griffith, "Experimental and Analytical Study of Direct Contact Condensation of Steam in Water", Nuclear Engineering and Design, Vol. 147, pp 425-435, (1994).
45. Liu, C. Liu, T.G. Theofanous and W.W. Yuen "Film Boiling From Spheres in Single- and Two-Phase Flow", ANS Proceedings, National Heat Transfer Conference, HTC-Vol. 6, San Diego, August, (1992).
46. Liu, Q. S. Liu, M. Shiotsu, and A. Sakurai, "A Correlation for Forced Convection Film Boiling Heat Transfer From A Horizontal Cylinder", HTD-Vol. 197, Two-phase Flow and Heat Transfer, Proceeding of 28th National Heat Transfer Conference, ASME (1992).
47. Marschall, E. Marschall and L.C. Farrar, "Film Boiling From A Partly Submerged Sphere", Int. J. of Heat and Mass Transfer, Vol. 18, pp 875-878. (1975).
48. McAdams, W.H. McAdams, Heat Transmission, 3rd ed., McGraw-Hill Book Co., New York, p. 172, (1954).
49. Merte, H. Merte Jr, J.A. Clark, "Boiling Heat Transfer With Cryogenic Fluids At Standard, Fractional, and Near-Zero Gravity", J. of Heat Transfer, Trans of ASME, pp 351-359, August, (1964).
50. Michiyoshi, I. Michiyoshi, O. Takahashi, and Y. Kikuchi, "Heat Transfer and the Low Limit of Film Boiling", Proc. of the First World Conf. on Experimental Heat Transfer, Fluid Mechanics and Thermodynamics, Dubrovnik, Yugoslavia, pp 1404-1415, (1988).

51. Motte, E.I. Motte, L.A. Bromley, "Film Boiling of Flowing Subcooled Liquids", *Industrial and Engineering Chemistry*, Vol. 49, No. 11, (1957).
52. Nakayama, A. Nakayama and H. Koyama, "An Integral Method in Laminar Film Pool Boiling From Curved Surfaces", *Transactions of the ASME, J. of Heat Transfer*, Vol. 108, pp 490-493, (1986).
53. Nakayama, A. Nakayama, and H. Koyama, "Analysis of Combined Free and Forced Convection Film Boiling Part I: Forced and Free Convection Regions", *A.I.Ch.E. JI*, Vol. 32, No. 1, pp 142-145, (1986).
54. Nakayama, A. Nakayama, and H. Koyama, "Analysis of Combined Free and Forced Convection Film Boiling Part II: Combined Free and Forced Convection Region", *A.I.Ch.E. JI*, Vol. 32, No. 1, pp 146-148, (1986).
55. Nakayama, A. Nakayama, "Subcooled Forced-Convection Film Boiling in the Presence of A Pressure Gradient", *AIAA Journal*, Vol. 24, No. 2 , pp 230-236, (1986).
56. Nishikawa, K. Nishikawa, T. Ito, and T. Kuroki, "Pool Film Boiling Heat Transfer From A Horizontal Cylinder to Saturated Liquids", *Int. J. of Heat and Mass Transfer*, Vol. 15, pp 853-862, (1972).
57. Nishikawa, Kaneyasu Nishikawa, T. Ito, and K. Matsumoto, "Investigation of Variable Thermophysical Property Problem Concerning Pool Film Boiling From Vertical Plate with Prescribed Uniform Temperature", *Int. J. of Heat and Mass Transfer*, Vol. 19, pp 1173-1182, (1976).
58. Orozco, J. Orozco and H. Francisco, "Free Convection Film Boiling Heat Transfer From A Rotating Surface", *J. of Heat Transfer, Trans. of the ASME*, Vol. 114, August, pp 695-702, (1992).
59. Orozco, J. Orozco, R. Stellman, and D. Poulikakos, "Dynamics Response of A Liquid-Vapor Interface During Flow Film Boiling From A Sphere", *J. of Heat Transfer, Transactions of the ASME*, Vol. 109, pp 1051-1055, (1987).
60. Orozco, J.A. Orozco, L.C. Witte, "Flow Film Boiling From A Sphere to Subcooled Freon-11", *J. of Heat Transfer. Transactions of ASME*, Vol. 108, Nov. (1986).
61. Rhea, L.G. Rhea, R.G. Nevins, "Film Boiling Heat Transfer From An Oscillating Sphere", *J. of Heat Transfer, Transactions of the ASME*, pp 267-272, May, (1969).
62. Sakurai , A. Sakurai, M. Shiotsu, and K. Hata, "A General Correlation for Pool Film Boiling Heat Transfer From A Horizontal Cylinder to Subcooled Liquid: Part 2 -- Experimental Data for Various Liquids and Its Correlation", *J. of Heat Transfer, Transactions of the ASME*, Vol. 112, pp 441--450, May, (1990b).

63. Sakurai, A. Sakurai and M. Shiotsu, "Pool Film Boiling Heat Transfer and Minimum Film Boiling Temperature", Pool and External Flow Boiling, ASME (1992).
64. Sakurai, A. Sakurai, M. Shiotsu, and K. Hata, "A General Correlation for Pool Film Boiling Heat Transfer From A Horizontal Cylinder to Subcooled Liquid: Part 1 -- A Theoretical Pool Film Boiling Heat Transfer Model Including Radiation Contributions and Its Analytical Solution", J. of Heat Transfer, Transactions of the ASME, Vol. 112, pp 430-440, May, (1990a).
65. Sakurai, A. Sakurai, M. Shiotsu, and K. Hata, "Effect of System Pressure on Film-Boiling Heat Transfer, Minimum Heat Flux, and Minimum Temperature", Nuclear Science and Engineering, Vol.88, pp 321-330, (1984).
66. Schmidt, W.E. Schmidt, and L.C. Witte, "Oscillation Effects Upon Film Boiling From A Sphere", J. of Heat Transfer, Transactions of the ASME, pp 491-493, Nov., (1972).
67. Sheppard, J.J. Sheppard, W.S. Bradfield, "Stagnation Free-Convection Film Boiling on A Hemisphere", (1971).
68. Shigechi, T. Shigechi, T. Ito, and K. Nishikawa, "Forced Convection Film Boiling Heat Transfer From a Horizontal Cylinder to Liquid Cross-flow Upward (2nd Report, Subcooled Liquid)", Bulletin of the JSME, Vol. 26, No. 214, pp 554-561, April, (1983).
69. Shih, C. Shih, and M.M. El-Wakil, "Film Boiling and Vapor Explosions From Small Spheres", Nuclear Science and Engineering, Vol. 77, pp 470-479, (1981).
70. Siviour, J.B. Siviour and A.J. Ede, "Heat Transfer in Subcooled Pool Film Boiling", Proc. of 4th Int. Heat Transfer Conf., Vol.5, B3.12, Paris-Versailles, (1970).
71. Sparrow, E.M. Sparrow, "The Effect of Radiation on film-boiling heat transfer", Int. J. of Heat Mass Transfer, Vol. 7, pp 229-238, (1964).
72. Srinivasan, J. Srinivasan and N.S. Rao, "Numerical Study of Heat Transfer In Laminar Film Boiling By the Finite-Difference Method", Int. J. of Heat and Mass Transfer, Vol. 27, pp 77-84, (1984).
73. Stevens, J.W. Stevens, and L.C. Witte, "Destablization of Vapor Film Boiling Around Spheres", Int. J. Heat & Mass Transfer, Vol.16, pp. 669-678. (1973).
74. Stevens, J.W. Stevens, L.C. Witte, "Transient Film and Transition Boiling From A Sphere", J. Of Heat Transfer, Vol. 14, pp 443-450., (1971).
75. Tachibana, F. Tachibana, and S. Fukui, "Heat Transfer in Film Boiling to

- Subcooled Liquids", Int. Developments in Heat Transfer, ASME, pp 219-223, 1961.
76. Tennekes, H. Tennekes and J.L. Lumley, A First Course In Turbulence, the MIT Press, (1972).
 77. Theofanous, T.G. Theofanous, B. Najafi and E. Rumble, "An Assessment of steam-Explosion-Induced Containment Failure. Part I: Probabilistic Aspects," Nuclear Science and Engineering 97, pp 259-281 (1987).
 78. Theofanous, T.G. Theofanous, R.N. Houze and L.K. Brumfield, "Turbulent Mass Transfer at Free, Gas-Liquid Interface, With Applications To Open-Channel, Bubble and Jet Flows", Int. J. of Heat Mass Transfer, Vol. 19, pp 613-624, (1976).
 79. Theofanous, W.H. Amarasekera and T.G. Theofanous, "Premixing of Steam Explosions: A Three-Fluid Model", Nuclear Engineering and Design, Vol. 126, pp 23-39, (1991).
 80. Toda, Saburo Toda, and Michitsugu Mori, "Subcooled Film Boiling and The Behavior of Vapor Film on A Horizontal Wire and A Sphere", Proc. of the 7th Int. Heat Transfer Conf., Munich, Germany, Vol. 4, pp 173-178, (1982).
 81. Touloukian, Y.S. Touloukian and E.H. Buyco, "Specific Heat, Metallic Elements and Alloys", Thermophysical Properties of Matter 4, IFI/Plenum, New York (1970).
 82. Tso, C.P. Tso, H.G. Low, and S. M. Ng, "Pool Film Boiling From Spheres to Saturated and Subcooled Liquids of Freon-12 and Freon-22", Int. J. Heat and Fluid Flow, Vol. 11, No. 2, pp 154-159, (1990).
 83. Walford, F.J. Walford, "Transient Heat Transfer From A Hot Nickel Sphere Moving Through Water", Int. J. Heat Mass Transfer, pp 1621--1625, (1970).
 84. Walsh, S. Walsh and S.D.R. Wilson, "Boundary-Layer Flow In Forced-Convection Film-Boiling on A Wedge", Int. J. of Heat and Mass Transfer, Vol. 22, pp 569-574, (1979).
 85. Wang, B.X. Wang and X.F. Peng, "Film Boiling Heat Transfer For Liquid Flowing With High Velocity", Int. J. of Heat and Mass Transfer, Vol. 35, No. 3, pp 675-682, (1992).
 86. Westwater, J.W. Westwater, J.J. Hwalek, and Mark E. Irving, "Suggested Standard Method for Obtaining Boiling Curves by Quenching", Ind. Eng. Chem. Fundamentals, Vol. 25, pp 685-692, (1986).
 87. Wilson, S.D.R. Wilson, "Steady and Transient Film Boiling on A Sphere in Forced Convection", Int. J. Heat Mass Transfer, Vol. 22, pp 207--218, (1979).

88. Witte, L.C. Witte, and J. Orozco, "The Effect of Vapor Velocity Profile Shape on Flow Film Boiling From Submerged Bodies", J. of Heat Transfer, Trans. of the ASME, Vol. 106, pp 191--197, (1984).
89. Witte, L.C. Witte, "Film Boiling From A Sphere", I&EC Fundamentals, pp 517--518, (1968).
90. Yuen, W.W. Yuen, X. Chen and T.G. Theofanous, "On the Fundamental Microinteractions That Support the Propagation of Steam Explosions", Nuclear Engineering and Design, Vol. 146, pp 133-146, (1994).
91. Yuge, T. Yuge, "Experiments on Heat Transfer from Spheres Including Combined Natural and Forced Convection", J. Heat Transfer ser. C 82, p 214 (1960).
92. Zvirin, Y. Zvirin, G.F. Hewitt, D.B.R. Kenning, "Boiling on Free-Falling Spheres: Drag and Heat Transfer Coefficients", Experimental Heat Transfer, Vol.3, pp 185-214, (1990).
93. Zvirin, Y. Zvirin, S. Aziz, G.F. Hewitt, and D.B.R. Kenning, "Experimental Investigation of the Transition From Boiling on Free Falling Spheres", A.N.S. Proc. 23rd U.S. National Heat Transfer Conf., Houston, Texas, pp 209-216, (1988).

APPENDIX A

A Derivation of the Correlation for Pool Film Boiling on Sphere

Figure A1 shows the physical model and coordinates, with the sphere radius, vapor layer thickness and liquid layer thickness being R , Δ , and δ , respectively. Based on the assumptions of thin layer in an incompressible fluid and no radiation effects, the one dimensional laminar momentum and energy boundary layer governing equations for the buoyancy-induced vapor motion are

$$\rho_v u_v \frac{\partial u_v}{\partial x} = g(\rho_l - \rho_v) \sin \phi + \mu_v \frac{\partial^2 u_v}{\partial y^2} \quad (\text{A1})$$

$$\rho_v c_{pv} u_v \frac{\partial T_v}{\partial x} = k_v \frac{\partial^2 T_v}{\partial y^2} \quad (\text{A2})$$

The corresponding equations of the free-convection in the liquid boundary layer are

$$\rho_l u_l \frac{\partial u_l}{\partial x} = g\beta_l \rho_l (T_l - T_\infty) \sin \phi + \mu_l \frac{\partial^2 u_l}{\partial z^2} \quad (\text{A3})$$

$$\rho_l c_{pl} u_l \frac{\partial T_l}{\partial x} = k_l \frac{\partial^2 T_l}{\partial z^2} \quad (\text{A4})$$

Figure A1 Model and coordinate system for subcooled pool film boiling

In order to carry out an analytical solution to the above boundary layer equations, further assumptions are made:

- (1) Convective terms are neglected, and the effects are compensated by using an effective latent heat of vaporization h'_{fg} in the later on energy integral equation.
- (2) The buoyancy term in the liquid momentum equation is neglected.
- (3) The wall temperature is uniform at T_w , and the vapor-liquid interface is smooth and is at T_{sat} ($T_w > T_{sat} > T_\infty$)
- (4) The thickness of the momentum boundary layer is assumed to be the same as that of the thermal boundary layer in liquid phase.

The necessary velocity boundary conditions are assumed to be non-slip condition at the wall and at the vapor-liquid interface, continuous interfacial shear, and stationary bulk liquid beyond the boundary of the liquid layer. The full set of conditions are expressed by

At $y=0$:

$$\begin{aligned} u_v &= 0 \\ T_v &= T_w \end{aligned} \quad (A5)$$

At $y=\Delta$ and $z=\delta$:

$$\begin{aligned} u_v &= u_l \\ \mu_v \frac{\partial u_v}{\partial y} &= \mu_l \frac{\partial u_l}{\partial z} \\ T_v &= T_l = T_{sat} \end{aligned} \quad (A6)$$

At $z=\delta$:

$$\begin{aligned} u_l &= 0 \\ T_l &= T_\infty \end{aligned} \quad (A7)$$

The governing equation may now be solved completely in terms of the unknowns Δ and δ . The results are

$$u_v = \frac{g(\rho_l - \rho_v) \sin \phi \Delta^2}{2\mu_v} \left[\left(\frac{f + 2\xi}{f + \xi} \right) \left(\frac{y}{\Delta} \right) - \left(\frac{y}{\Delta} \right)^2 \right] \quad (A8)$$

$$T_v = T_w - \Delta T_{\text{sup}} \frac{y}{\Delta} \quad (\text{A9})$$

$$u_i = \frac{g(\rho_l - \rho_v) \sin \phi \Delta^2}{2\mu_l} \left(\frac{f\xi}{f + \xi} \right) \left(1 - \frac{z}{\delta} \right) \quad (\text{A10})$$

$$T_l = T_{\text{sat}} - \Delta T_{\text{sub}} \frac{z}{\Delta} \quad (\text{A11})$$

where $f = \mu_l/\mu_v$ and $\xi = \delta/\Delta$. The integral heat balance are performed on the vapor and liquid layers, giving

$$h_{fs} \frac{d}{dx} \int_0^\Delta u_v \rho_v 2\pi R \sin \phi dy = 2\pi R \sin \phi \left[-k_v \frac{\partial T_v}{\partial y} \Big|_{y=0} + k_l \frac{\partial T_l}{\partial z} \Big|_{z=0} \right] \quad (\text{A12})$$

$$C_{pl} \frac{d}{dx} \int_0^\delta u_l \rho_l (T_l - T_w) 2\pi R \sin \phi dz = 2\pi R \sin \phi \left[-k_l \frac{\partial T_l}{\partial z} \Big|_{z=0} \right] \quad (\text{A13})$$

Substituting Equations (A8 - A11) into the two integral equations (A12) and (A13), and assuming ξ to be independent of x , yields

$$\frac{d}{dx} (\Delta^3 \sin^2 \phi) = \left(\frac{f + \xi}{f + 4\xi} \right) \left[\frac{12\mu_v k_v \Delta T_{\text{sup}}}{g(\rho_l - \rho_v) \rho_v h_{fs}} - \frac{12\mu_v k_l \Delta T_{\text{sub}}}{g(\rho_l - \rho_v) \rho_v h_{fs} \xi} \right] \frac{\sin \phi}{\Delta} \quad (\text{A14})$$

$$\frac{d}{dx} (\Delta^3 \sin^2 \phi) = \left(\frac{f + \xi}{f \xi^2} \right) \left[\frac{6\mu_l k_l}{g(\rho_v - \rho_l) \rho_l C_{pl} \xi} \right] \frac{\sin \phi}{\Delta} \quad (\text{A15})$$

The solution to equations (A14) and (A15), respectively, are

$$\Delta = I(\phi) \left\{ \left(\frac{f + \xi}{f + 4\xi} \right) \left[\frac{16R\mu_v k_v \Delta T_{\text{sup}}}{g(\rho_l - \rho_v) \rho_v h_{fs}} - \frac{16R\mu_v k_l \Delta T_{\text{sub}}}{g(\rho_l - \rho_v) \rho_v h_{fs} \xi} \right] \right\}^{1/4} \quad (\text{A16})$$

$$\Delta = I(\phi) \left\{ \left(\frac{f + \xi}{f \xi^2} \right) \left[\frac{8R\mu_l k_l}{g(\rho_v - \rho_l) \rho_l C_{pl} \xi} \right] \right\}^{1/4} \quad (\text{A17})$$

where

$$I(\phi) = \frac{\left[\int \sin^{5/3} \phi d\phi \right]^{1/4}}{\sin^{2/3} \phi} \quad (\text{A18})$$

Combining Equations (A16) and (A17) produces a cubic equation for ξ , which has at least one real root:

$$\xi = (W_1 + W_2^{1/2})^{1/3} + (W_1 - W_2^{1/2}) + \frac{k_i \Delta T_{sub}}{3k_v \Delta T_{sup}} \quad (\text{A19})$$

where

$$W_1 = \left(\frac{C_2}{3C_1} \right)^3 + \frac{2C_2 C_3}{2C_1^2 f} + \frac{C_3}{2C_1} \quad (\text{A20})$$

$$W_2 = \left(\frac{C_3}{2C_1} \right)^2 + \left(\frac{C_3}{C_1} \right) \left(\frac{C_2}{3C_1} \right) + \left(\frac{C_3^2}{fC_1^3} \right) \left[\frac{2C_2}{3} - \frac{4C_2^2}{27fC_1} - \frac{64C_3}{27f^2} \right] \quad (\text{A21})$$

$$C_1 = \frac{12\mu_v k_v \Delta T_{sup}}{g(\rho_l - \rho_v) \rho_v h_{fg}} \quad C_2 = \frac{12\mu_v k_i \Delta T_{sub}}{g(\rho_l - \rho_v) \rho_v h_{fg}} \quad C_3 = \frac{6\mu_l k_l}{g(\rho_l - \rho_v) \rho_l C_{pl}} \quad (\text{A22})$$

Now we are ready to evaluate the heat transfer coefficient. The average heat flux from the sphere surface is

$$\bar{q}_x = \frac{1}{4\pi R^2} \int_0^\pi (2\pi R \sin \phi) R d\phi \left(-k_v \frac{\partial T_v}{\partial y} \Big|_{y=0} \right) = 0.5 k_v \Delta T_{sup} \int_0^\pi \frac{\sin \phi}{\Delta} d\phi \quad (\text{A23})$$

Hence the average Nusselt number is

$$Nu_c = \frac{\bar{h} d_s}{k_v} = \left(\frac{\bar{q}_x}{\Delta T_{sup}} \right) \frac{d_s}{k_v} = \frac{1}{2} d_s \int_0^\pi \frac{\sin \phi}{\Delta} d\phi \quad (\text{A24})$$

Substituting Equation (A17) into (A24) yields

$$Nu_c = 2^{-3/2} (Ar Pr_v)^{1/4} \left[\left(\frac{f + \xi}{\xi^3} \right) \left(\frac{k_l C_{pv} \rho_v}{k_v C_{pl} \rho_l} \right) \right]^{1/4} \int_0^\pi \frac{\sin \phi}{I(\phi)} d\phi \quad (\text{A25})$$

By using Equation (A19) and rearranging Equation (A25), we have

$$Nu_c = C_0 (Ar/Sp')^{1/4} M_c^{1/4} \quad (A26)$$

where

$$\begin{aligned} M_c &= E^3/[1+E/(Sp'Pr)]/(RPrSp')^2 \\ E &= (A + C B^{1/2})^{1/3} + (A - C B^{1/2})^{1/3} + (1/3) Sc^* \\ A &= (1/27)Sc^{*3} + (1/3)R^2Sp'PrSc^* + (1/4)R^2Sp'^2Pr^2 \\ B &= (-4/27)Sc^{*2} + (2/3)Sp'PrSc^* - (32/27)Sp'PrR^2 \\ &\quad + (1/4)Sp'^2Pr^2 + (2/27)Sc^{*3}/R^2 \\ C &= (1/2)R^2Sp'Pr, \\ h_{fg}' &= h_{fg} + 0.5C_{pv}\Delta T_{sup}, \quad Sp' = C_{pv}\Delta T_{sup}/(h_{fg}'Pr_v) \\ Sc^* &= C_{pl}\Delta T_{sub}/h_{fg}', \quad R=[(\mu\rho)_w/(\mu\rho)_l]^{1/2} \end{aligned}$$

$$C_0 = 2^{-3/2} \int_0^\pi \frac{\sin \phi}{I(\phi)} d\phi = 0.696 \quad (A27)$$

The similar analysis can be carried out for pool film boiling on horizontal cylinder and on vertical plate. The same formulation can be obtained except that the constant C_0 is different; C_0 for cylinder case is 0.61 and for plate case is 0.793.

APPENDIX B

A Derivation of the Correlation for Forced Convection Film Boiling on Sphere

A sphere of radius R and temperature T_w is held in a uniform stream of liquid which has undisturbed velocity U and temperature T_l . We suppose that T_w is constant but may depend on θ . The temperature at the vapor-liquid interface is the saturation temperature T_s . We suppose that the thickness of the vapor film and of the thermal boundary layer in the liquid phase is small compared with radius R . In these regions thin film theory or boundary-layer theory is applied.

Liquid layer

It is supposed that the velocity field is given by a potential flow. According to K.H. Hsiao, L.C. Witte and J.E. Cox (1975), in spherical polar coordinates (r, θ, ϕ) centered on the sphere and with θ measured from the direction of the incoming steam, the velocity components are

$$U_l \left[-\left(1 - \frac{R^3}{r^3}\right) \cos \theta, \left(1 + \frac{R^3}{2r^3}\right) \sin \theta, 0 \right]$$

We also confine the analysis in the region where the flow remains attached (unseparated).

To find the temperature field, a dimensionless boundary-layer coordinate $y = [(r/R) - 1]Pe^{1/2}$ is introduced, and then the dimensionless energy equation takes the form

$$\frac{3}{2} \sin \theta \frac{\partial T^*}{\partial \theta} - 3y \cos \theta \frac{\partial T^*}{\partial y} = \frac{\partial^2 T^*}{\partial y^2} \quad (B1)$$

where

$$T^* = (T - T_l) / (T_s - T_l) = (T - T_l) / \Delta T_{sub}$$

and the boundary condition are

$$T^* = 1 \text{ on } y = 0; \quad T^* = 0 \text{ at } y = \infty$$

The solution of this problem is

$$T^* = \operatorname{erfc} [y f(\theta)] \quad (\text{B2})$$

where

$$f(\theta) = \frac{3}{4} \sin^2 \theta \left(1 - \frac{3}{2} \cos \theta + \frac{1}{2} \cos^3 \theta \right) \quad (\text{B3})$$

A constant of integration has been determined by the requirement that $f(0) \neq 0$; in fact $f(0) = (3/2)^{0.5}$.

Thus, the the temperature gradient at the liquid-vapor interface ($y = 0$) can be found as

$$\left(\frac{\partial T}{\partial r} \right)_{y=0} = -\frac{2\Delta T_{\text{sub}}}{\pi^{1/2} R} P e^{1/2} f(\theta) = -\frac{\Delta T_{\text{sub}}}{R} \frac{1}{\delta_T} \quad (\text{B4})$$

where

$$\delta_T = \frac{\pi^{1/2}}{2 P e^{1/2}} \frac{1}{f(\theta)} \quad (\text{B5})$$

Vapor Layer

Here we choose coordinates (s, n) where s is length measured around the sphere from $\theta = 0$ and n is length measured normal to the surfaces; thus $s = \theta R$ and $n = r - R$. The velocity components are (u, v). Making the usual boundary-layer approximations, the equations of continuity, momentum and energy take the forms of

$$\frac{1}{\sin \theta} \frac{\partial}{\partial s} (u \sin \theta) + \frac{\partial v}{\partial n} = 0 \quad (\text{B6})$$

$$u \frac{\partial u}{\partial s} + v \frac{\partial u}{\partial n} = -\frac{1}{\rho_v} \frac{\partial p}{\partial s} + \nu_v \frac{\partial^2 u}{\partial n^2} \quad (\text{B7})$$

$$0 = -\frac{1}{\rho_v} \frac{\partial p}{\partial n} \quad (\text{B8})$$

$$u \frac{\partial T}{\partial s} + v \frac{\partial T}{\partial n} = k_v \frac{\partial^2 T}{\partial n^2} \quad (\text{B9})$$

The vapor-liquid interface is located at $n=n_i(s)$. The boundary conditions are

$$\begin{array}{ll} u = 0, \quad T = T_w & \text{on } n = 0 \\ u = U_s = 3/2 U \sin \theta, \quad T = T_s & \text{on } n = n_i \end{array} \quad (\text{B10})$$

Equation (B8) indicates that the pressure in the vapor layer is independent of n as usual in boundary layer theory, and therefore equals $p(n_i)$, the pressure at the interface. By continuity of normal stress at the interface, this is equal to the pressure in the liquid at the interface, and this in turn may be found from the potential flow of the liquid by means of Bernoulli's equation, which gives

$$p = \text{constant} - \frac{1}{2} \rho_l \left(\frac{3}{2} U \sin \theta \right)^2 \quad (\text{B11})$$

To effect an analytical solution, further assumption is made: convective terms (on the left side) in the momentum equation (B7) and energy equation (B9) are neglected. Thus the solutions can be obtained as

$$u = A_1 \frac{n}{n_i} + A_2 \frac{n^2}{n_i^2} \quad (\text{B12})$$

$$T - T_s = \Delta T_{\text{sup}} \left(1 - \frac{n}{n_i} \right) \quad (\text{B13})$$

where A_1 and A_2 are determined from the following two equations, one is from momentum equation (B7) and the other one is from the boundary condition (B10).

$$0 = -\frac{1}{\rho_v} n_i \frac{\partial p}{\partial s} + \frac{2A_2 v_v}{n_i} \quad (\text{B14})$$

$$A_1 + A_2 = U_s \quad (\text{B15})$$

Integrating continuity Equation (B6) from $n = 0$ to $n = n_i$ gives

$$\frac{1}{\sin \theta} \frac{d}{ds} \left[\sin \theta \int_0^{n_i} u dn \right] = \left(u \frac{dn_i}{ds} - v \right)_{n_i} = G_v \quad (\text{B16})$$

where G_v is the volume rate of evaporation of liquid at the location θ per unit interface area. G_v is coupled to the temperature fields by means of heat balance and evaporation at the interface, thus it gives

$$\frac{1}{\sin \theta} \frac{d}{ds} \left[\sin \theta \int_0^{n_i} u dn \right] = G_v = \frac{1}{\rho_v h_{fg}} \left[\left(k_l \frac{\partial T}{\partial r} \right)_{y=0} - \left(k_v \frac{\partial T}{\partial n} \right)_{n_i} \right] \quad (\text{B17})$$

Substituting Equations (B4), (B12) and (B13) into Equation (B17) yields

$$\frac{1}{\sin \theta} \frac{d}{d\theta} \left[\sin \theta \left(\frac{A_1 n_i}{2} + \frac{A_2 n_i}{3} \right) \right] = \frac{1}{\rho_v h_{fg}} \left(\frac{k_v \Delta T_{\text{sup}}}{n_i} - \frac{k_l \Delta T_{\text{sub}}}{R \delta_T} \right) \quad (\text{B18})$$

A_1 and A_2 can be eliminated by means of Equations (B14) and (B15). Then, in terms of the dimensionless vapor film thickness $n^* = n_i/R$ Equation (B18) becomes

$$\frac{1}{\sin \theta} \frac{d}{d\theta} \left[\sin \theta \left(n^{*3} \sin \theta \cos \theta + \alpha n^* \sin \theta \right) \right] = \frac{\beta}{n^*} - \gamma(\theta) \quad (\text{B19})$$

where

$$\begin{aligned} \alpha &= \frac{4\mu_v}{UR\rho_l} \\ \beta &= \frac{16}{3} \frac{k_v \mu_v \Delta T_{\text{sup}}}{\rho_v \rho_l U^2 R^2 h_{fg}} \\ \gamma &= \frac{16}{3} \frac{k_l \mu_v \Delta T_{\text{sub}}}{\rho_v \rho_l U^2 R^2 h_{fg}} \frac{2}{\pi^{1/2}} Pe^{1/2} \end{aligned}$$

Equation (B19) cannot be solved in closed form, but can be integrated numerically in any particular case. However, this is not convenient for practical uses and not helpful for developing general correlations. So, instead we will show how the equation may be simplified in two special cases and then integrate it analytically. Therefore, we can gain more insight into which terms are likely to be important. Furthermore, the simplified solutions have the great advantage to provide dimensionless groups for developing correlations with only the constant to be determined by experiment.

Large Subcooling case

Here we begin by supposing that the left hand side of (B19) is negligible, and the main balance is between the two terms on the right. This express mathematically the situation in which almost all the heat arriving at the interface is convected away in the liquid, and very little is used to produce vapor. This can be justified by following. First we suggest that the order of magnitude of n^* is β/γ .

$$n^* = \frac{\beta}{\gamma} H \quad (B20)$$

Thus, after some rearrangement of (B19) we have

$$\varepsilon \frac{1}{\sin \theta} \frac{d}{d\theta} [\sin \theta (H^3 \sin \theta \cos \theta + AH \sin \theta)] = \frac{1}{H} - f(\theta) \quad (B21)$$

where

$$\varepsilon = \frac{\beta^3}{\gamma^4} = \frac{3\pi^2 k_v^3 \rho_v \rho_l \alpha_l^2 h_{fg} (\Delta T_{sup})^3}{256 k_l^4 \mu_v (\Delta T_{sub})^4} \approx 5 \times 10^{-5} \frac{\Delta T_{sup}^3}{\Delta T_{sub}^4}$$

$$A = \frac{\alpha \gamma^2}{\beta^2} = \frac{16 \mu_v k_l^2 (\Delta T_{sub})^2}{\pi \rho_l k_v^2 \alpha_l (\Delta T_{sup})^2} \approx 1.6 \times 10^2 \frac{\Delta T_{sub}^2}{\Delta T_{sup}^2}$$

If ε is small and A is not large, the left hand side of (B21) could be reasonably be neglected. It is easy to see that these requirements can be met in the case of large subcooling. For water/steam system, if $\Delta T_{sup} = 800^\circ\text{C}$, ΔT_{sub} need only be 20°C to give $\varepsilon = 0.16$ and $A = 0.1$. The higher is the subcooling and the lower is the sphere superheat the better is the approximation.

So the solution is simply

$$H = \frac{1}{f(\theta)} \quad (B22)$$

to the first approximation.

It is easily to show that the Nusselt number is essentially the reciprocal of the nondimensional film thickness

$$Nu_0 = \frac{1}{n^*} \quad (B23)$$

Therefore, for the case of large subcooling

$$Nu_0 = \frac{1}{n^*} = \frac{1}{H(\theta)} \frac{\gamma}{\beta} = \frac{2}{\pi^{1/2} H(\theta)} \text{Re}_l^{1/2} \text{Pr}_l^{1/2} \frac{\mu_l}{\mu_v} \frac{Sc}{Sp} \quad (B24)$$

This the local Nusselt number. The average Nusselt number is defined as

$$Nu = \frac{\int_0^{\theta_s} Nu_\theta dA}{4\pi R^2} = \frac{1}{2} \int_0^{\theta_s} \sin \theta Nu_\theta d\theta \quad (B25)$$

Where θ_s is the angle of separation, which is π for the case of large subcooling. Thus, the average Nusselt number is in the form of

$$Nu = C Re_i^{1/2} \frac{\mu_l}{\mu_v} \frac{Sc}{Sp} \quad (B26)$$

where

$$C = \frac{1}{\pi^{1/2}} \int_0^\pi \frac{\sin \theta}{H(\theta)} d\theta = 0.5642 \quad (B27)$$

This constant is much smaller than the experimental constant of 1.5 to 2.0.

Saturated case

In this case we begin by neglecting the subcooling altogether, so that we set $\gamma = 0$. in Eq. (B21). After some trial and error it is found that the main balance is between the first term in the bracket on the left, which comes from the pressure gradient, and the term in β/n^* on the right. This suggests that the order of magnitude of n^* is $\beta^{1/4}$, and it gives

$$n^* = \beta^{1/4} \Psi \quad (B28)$$

Substituting it into (B19) yields

$$\frac{1}{\sin \theta} \frac{d}{d\theta} [\sin \theta (\Psi^3 \sin \theta \cos \theta + \delta \Psi \sin \theta)] = \frac{1}{\Psi} \quad (B29)$$

where the δ is given by

$$\delta = \frac{\alpha}{\beta^{1/2}} = \left(\frac{3\mu_v \rho_l h_{fg}}{k_v \rho_v} \right)^{1/2} \frac{1}{\Delta T_{sup}^{1/2}} \approx 1.1 \times \frac{1}{\Delta T_{sup}^{1/2}} \quad (B30)$$

where the numerical values are appropriate for water/steam system. δ is small as long as ΔT_{sup} is large, so the term with δ in (B29) is clearly negligible in most cases. Thus the analytical solution for Ψ is

$$\Psi^4 = \frac{4}{3} (\sin \theta)^{-3/3} (\cos \theta)^{-4/3} \int_0^\theta (\sin \theta')^{5/3} (\cos \theta')^{1/3} d\theta' \quad (\text{B31})$$

It should be noted that this solution is valid only in the region $\theta < \pi/2$. Actually, for saturated forced convection film boiling, the vapor film gradually develops into a big vapor wake/dome, starting from the region near $\theta \approx \pi/2$.

Therefore for small subcooling (or saturated) cases, the Nusselt number can be expressed as

$$Nu_\theta = \frac{1}{n^*} = \frac{1}{\Psi(\theta)} \frac{1}{\beta^{1/4}} = \left(\frac{3}{64}\right)^{1/4} \frac{1}{\Psi(\theta)} Re_l^{1/2} \left(\frac{1}{Sp} \frac{\mu_l^2 \rho_v}{\mu_v^2 \rho_l}\right)^{1/4} \quad (\text{B32})$$

By the definition of (B25) and integration from $\theta = 0$ to $\pi/2$, the average Nusselt number may be correlated in the form of

$$Nu = C Re_l^{1/2} \left(\frac{1}{Sp} \frac{\mu_l^2 \rho_v}{\mu_v^2 \rho_l}\right)^{1/4} = C Re_l^{1/2} \frac{\mu_l}{\mu_v} \left(\frac{R^* K}{Sp}\right)^{1/4} \quad (\text{B33})$$

where

$$R = \left(\frac{\mu_v \rho_v}{\mu_l \rho_l}\right)^{1/2}, \quad K = \frac{\rho_l}{\rho_v}$$

and

$$C = \frac{1}{2} \left(\frac{3}{64}\right)^{1/4} \int_0^{\pi/2} \frac{\sin \theta}{\Psi} d\theta = 0.198 \quad (\text{B34})$$

As in the case of large subcooling, this theoretical constant is considerably low, comparing with the constant 0.5 which is obtained from the experiment.

Appendix C

Typical Saturated Single-Phase Experiment Data

Test Sphere Diameter : 6.35 mm
 Support Thermocouple Wire Diameter: 0.51 mm
 Test Sphere Material : 316 Stainless Steel

RUN 1 $T_i=100.1(C)$ $V_i=0.010(m/s)$ $Fr= 0.002$ $Re= 225.1$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc
672.	162.33	138.92	284.21	243.22
790.	190.06	152.30	275.99	221.16
575.	133.30	118.33	281.21	249.64
493.	109.65	99.87	279.45	254.51
426.	87.36	80.78	269.00	248.73
369.	72.60	68.08	271.12	254.25
320.	61.57	58.44	281.12	266.80
278.	52.26	50.06	294.91	282.54

RUN 2 $T_i=100.0(C)$ $V_i=0.112(m/s)$ $Fr= 0.197$ $Re= 2500.1$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc
698.	182.60	156.47	305.94	262.16
600.	146.54	129.68	293.79	259.98
518.	121.16	109.97	290.49	263.66
448.	100.64	93.07	289.56	267.79
389.	83.35	78.15	289.17	271.14
338.	70.33	66.72	296.30	281.06
294.	60.35	57.82	312.20	299.11

RUN 3 $T_i=100.1(C)$ $V_i=0.216(m/s)$ $Fr= 0.730$ $Re= 4815.2$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc
814.	226.60	185.18	317.74	259.66
693.	175.58	149.95	296.43	253.15
595.	140.68	124.21	284.85	251.49
513.	115.77	104.88	281.08	254.66
442.	99.30	92.03	291.21	269.87
380.	83.27	78.37	298.09	280.57
328.	70.18	66.85	309.61	294.91
284.	57.01	54.70	311.88	299.26

RUN 4 $T_i=100.3(C)$ $V_i=0.232(m/s)$ $Fr= 0.839$ $Re= 5167.1$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc
750.	202.52	170.10	311.79	261.89
645.	167.30	146.56	307.75	269.61
556.	134.94	121.34	296.78	266.89
481.	110.53	101.40	290.97	266.94

417.	94.14	87.90	297.83	278.10	46.22	46.22
362.	79.26	74.95	303.47	286.97	51.31	51.31
315.	67.33	64.31	314.19	300.11	57.34	57.34
275.	55.10	52.96	315.67	303.41	61.36	61.36

RUN 5 $Tl=100.5(C)$ $VI=0.333(m/s)$ $Fr= 1.733$ $Re= 7426.9$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
758.	207.37	173.98	315.64	264.82	29.70	29.70
644.	164.20	143.50	302.23	264.12	33.45	33.45
550.	136.22	122.98	303.09	273.63	38.59	38.59
472.	111.28	102.60	299.95	276.56	42.90	42.90
406.	91.36	85.53	299.12	280.06	47.20	47.20
351.	73.58	69.61	294.64	278.76	50.63	50.63
304.	62.92	60.18	310.35	296.82	57.69	57.69
265.	51.50	49.57	314.32	302.52	62.12	62.12

RUN 6 $Tl=100.5(C)$ $VI=0.446(m/s)$ $Fr= 3.110$ $Re= 9951.5$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
857.	236.82	188.44	313.40	249.38	25.34	25.34
709.	189.31	161.93	311.34	266.31	31.45	31.45
593.	143.73	127.37	291.90	258.67	34.72	34.72
499.	117.28	107.20	294.79	269.43	40.47	40.47
421.	93.78	87.38	292.90	272.92	45.11	45.11
357.	75.46	71.30	294.49	278.25	50.09	50.09
305.	59.64	56.87	292.49	278.91	54.12	54.12
262.	49.25	47.37	306.22	294.55	60.75	60.75
226.	40.36	39.07	321.94	311.68	67.75	67.75

RUN 7 $Tl=100.6(C)$ $VI=0.538(m/s)$ $Fr= 4.515$ $Re= 11998.0$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
739.	207.11	176.19	324.74	276.25	31.61	31.61
620.	157.80	139.21	303.73	267.96	34.86	34.86
525.	127.03	115.45	299.87	272.52	39.65	39.65
446.	100.79	93.35	292.36	270.77	43.38	43.38
379.	80.17	75.31	287.87	270.40	47.22	47.22
325.	71.28	68.01	317.94	303.37	57.14	57.14
280.	55.18	52.96	308.35	295.91	59.47	59.47
242.	46.00	44.46	325.60	314.72	66.79	66.79
211.	37.31	36.24	340.01	330.32	73.55	73.55

RUN 8 $Tl=100.7(C)$ $VI=0.772(m/s)$ $Fr= 9.319$ $Re= 17244.8$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
769.	245.55	210.68	367.54	315.33	34.97	34.97
674.	212.47	188.83	370.55	329.32	40.36	40.36
590.	177.25	161.13	362.20	329.26	44.34	44.34
519.	148.61	137.37	355.62	328.72	48.16	48.16
458.	124.43	116.43	348.60	326.19	51.47	51.47
406.	105.96	100.15	347.34	328.30	55.35	55.35
360.	91.65	87.40	353.55	337.15	60.44	60.44

320. 77.26 74.12 352.08 337.75 64.05 64.05

RUN 9 $Tl=100.7(C)$ $VI=0.955(m/s)$ $Fr=14.240$ $Re=21316.6$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
788.	253.93	216.40	369.57	314.95
34.26				34.26
692.	237.11	211.58	400.92	357.75
43.01				43.01
608.	204.65	187.16	403.94	369.40
48.78				48.78
535.	168.64	156.38	388.14	359.92
51.68				51.68
473.	142.79	134.06	383.88	360.42
55.84				55.84
419.	123.78	117.48	389.51	369.67
61.30				61.30
371.	101.88	97.28	376.78	359.77
63.52				63.52
330.	92.09	88.70	402.30	387.49
72.48				72.48

RUN 10 $Tl=100.8(C)$ $VI=1.310(m/s)$ $Fr=26.835$ $Re=29274.4$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
786.	247.84	210.54	361.58	307.17
33.47				33.47
700.	281.92	255.52	470.46	426.41
50.84				50.84
619.	239.14	220.66	461.40	425.75
55.47				55.47
549.	202.32	189.15	451.24	421.88
59.58				59.58
488.	178.65	169.17	462.08	437.56
66.60				66.60
432.	150.99	144.13	455.67	434.97
70.88				70.88
385.	135.16	130.10	475.53	457.73
79.32				79.32
343.	122.96	119.23	508.76	493.30
90.61				90.61

RUN 11 $Tl=100.8(C)$ $VI=1.645(m/s)$ $Fr=42.304$ $Re=36756.3$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
768.	198.48	163.74	297.53	245.46
27.25				27.25
696.	289.39	263.44	486.17	442.57
52.99				52.99
624.	250.98	232.10	479.87	443.77
57.50				57.50
560.	211.77	197.90	461.73	431.49
60.20				60.20
503.	187.73	177.38	466.42	440.71
65.82				65.82
454.	162.16	154.37	459.90	437.78
69.45				69.45
409.	143.75	137.82	466.65	447.42
75.14				75.14
369.	130.60	126.08	487.86	471.00
83.46				83.46

RUN 12 $Tl=100.8(C)$ $VI=1.838(m/s)$ $Fr=52.767$ $Re=41045.8$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
726.	286.09	256.71	457.60	410.59
47.60				47.60
650.	267.38	246.11	486.73	448.01
56.36				56.36
582.	230.48	215.01	479.40	447.23
60.82				60.82
520.	203.93	192.62	486.63	459.64
67.25				67.25
465.	176.21	167.85	483.34	460.40
71.97				71.97
418.	154.54	148.26	487.13	467.33
77.55				77.55
376.	134.28	129.54	488.40	471.13
82.67				82.67

Test Sphere Diameter : 9.53 mm
 Support Thermocouple Wire Diameter: 0.51 mm
 Test Sphere Material : 316 Stainless Steel

RUN 1 $T_i=100.5(C)$ $V_i=0.010(m/s)$ $Fr= 0.001$ $Re= 340.5$
 $T_b(C)$ $q_t,q_c(kW/m^2)$ $h_t,h_c(W/m^2/C)$ Nut Nuc
 731. 154.77 124.74 245.45 197.82 33.26 33.26
 636. 128.44 108.48 240.00 202.70 37.79 37.79
 554. 106.69 93.18 235.34 205.55 42.09 42.09
 484. 87.99 78.70 229.74 205.49 45.82 45.82
 424. 75.85 69.35 235.00 214.85 51.63 51.63
 372. 63.03 58.41 232.64 215.60 55.48 55.48
 327. 53.55 50.23 236.60 221.92 60.78 60.78
 289. 46.84 44.42 249.18 236.33 68.42 68.42
 256. 43.47 41.70 281.01 269.60 81.83 81.83

RUN 2 $T_i=100.5(C)$ $V_i=0.116(m/s)$ $Fr= 0.145$ $Re= 3782.8$
 $T_b(C)$ $q_t,q_c(kW/m^2)$ $h_t,h_c(W/m^2/C)$ Nut Nuc
 743. 158.87 127.44 247.55 198.59 33.00 33.00
 643. 129.33 108.71 238.46 200.45 37.07 37.07
 559. 108.78 94.96 237.54 207.37 42.24 42.24
 487. 90.21 80.77 233.80 209.34 46.52 46.52
 425. 76.82 70.26 236.87 216.62 51.95 51.95
 373. 64.31 59.66 236.57 219.47 56.41 56.41
 328. 53.56 50.22 235.82 221.11 60.48 60.48
 289. 45.51 43.08 241.31 228.44 66.07 66.07
 256. 40.60 38.83 262.07 250.64 76.05 76.05

RUN 3 $T_i=100.6(C)$ $V_i=0.167(m/s)$ $Fr= 0.299$ $Re= 5435.6$
 $T_b(C)$ $q_t,q_c(kW/m^2)$ $h_t,h_c(W/m^2/C)$ Nut Nuc
 731. 154.88 124.89 245.77 198.18 33.34 33.34
 640. 128.07 107.79 237.75 200.11 37.16 37.16
 560. 109.44 95.51 238.14 207.82 42.25 42.25
 492. 92.87 83.13 237.18 212.29 46.86 46.86
 434. 77.61 70.69 233.22 212.42 50.40 50.40
 383. 67.05 62.06 237.71 220.03 55.78 55.78
 339. 57.72 54.08 242.32 227.04 61.14 61.14
 301. 49.04 46.36 245.39 231.99 66.06 66.06
 267. 42.73 40.76 257.30 245.41 73.27 73.27

RUN 4 $T_i=100.6(C)$ $V_i=0.240(m/s)$ $Fr= 0.617$ $Re= 7812.0$
 $T_b(C)$ $q_t,q_c(kW/m^2)$ $h_t,h_c(W/m^2/C)$ Nut Nuc
 725. 151.27 122.03 242.44 195.58 33.12 33.12
 636. 128.93 108.99 241.00 203.72 37.99 37.99
 559. 107.21 93.36 233.88 203.66 41.46 41.46
 493. 91.33 81.58 233.14 208.24 45.96 45.96
 435. 79.16 72.20 237.11 216.24 51.23 51.23

384.	67.88	62.86	239.66	221.92	56.17	56.17
340.	59.09	55.43	247.16	231.83	62.35	62.35
302.	49.94	47.25	248.96	235.53	66.99	66.99
268.	44.07	42.08	264.41	252.49	75.32	75.32

RUN 5 $Tl=100.6(C)$ $VI=0.345(m/s)$ $Fr= 1.273$ $Re= 11221.3$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
745.	164.63	132.90	255.56	206.29	34.20	34.20
649.	134.18	112.99	244.66	206.03	37.85	37.85
567.	110.52	96.09	236.89	205.97	41.53	41.53
497.	92.88	82.90	234.55	209.34	45.97	45.97
436.	79.19	72.17	236.27	215.32	50.93	50.93
384.	65.69	60.67	232.27	214.54	54.34	54.34
338.	57.70	54.08	242.88	227.63	61.35	61.35
299.	49.79	47.15	251.08	237.76	67.85	67.85
265.	43.15	41.21	263.25	251.45	75.31	75.31

RUN 6 $Tl=100.6(C)$ $VI=0.426(m/s)$ $Fr= 1.945$ $Re= 13870.7$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
760.	159.92	126.24	242.58	191.48	31.25	31.25
656.	136.67	114.89	246.39	207.13	37.79	37.79
567.	107.35	92.93	230.16	199.25	40.19	40.19
493.	95.51	85.73	243.55	218.62	48.22	48.22
429.	73.89	67.15	225.00	204.49	48.79	48.79
375.	63.89	59.18	233.23	216.01	55.36	55.36
328.	52.66	49.31	231.64	216.92	59.32	59.32
288.	44.82	42.42	239.21	226.39	65.59	65.59
254.	38.93	37.19	253.96	242.60	73.79	73.79

RUN 7 $Tl=100.6(C)$ $VI=0.557(m/s)$ $Fr= 3.318$ $Re= 18118.4$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
766.	168.47	133.99	253.26	201.43	32.68	32.68
668.	142.07	119.04	250.39	209.80	37.75	37.75
585.	113.92	98.23	235.53	203.10	40.16	40.16
513.	99.95	89.07	242.64	216.21	46.55	46.55
451.	81.68	74.00	233.35	211.41	49.08	49.08
398.	70.95	65.45	239.07	220.53	54.83	54.83
351.	59.20	55.21	236.48	220.56	58.39	58.39
311.	54.59	51.68	259.99	246.12	69.03	69.03
276.	46.97	44.82	268.18	255.90	75.43	75.43

RUN 8 $Tl=100.6(C)$ $VI=0.800(m/s)$ $Fr= 6.850$ $Re= 26031.1$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
750.	211.72	179.34	326.07	276.19	45.54	45.54
663.	165.94	143.42	295.14	255.09	46.15	46.15
587.	135.96	120.06	279.49	246.81	48.65	48.65
521.	119.34	107.94	283.75	256.65	54.67	54.67
465.	98.21	89.88	269.97	247.09	56.40	56.40
414.	86.31	80.19	275.49	255.93	62.26	62.26

370.	76.04	71.49	282.80	265.87	68.61	68.61
330.	67.28	63.89	293.69	278.87	76.07	76.07
295.	59.34	56.79	305.43	292.29	83.88	83.88

RUN 9 $Tl=100.6(C)$ $VI=0.988(m/s)$ $Fr=10.466$ $Re=32181.8$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
786.	205.97	168.70	300.57	246.19	39.14	39.14
696.	178.68	152.77	300.38	256.83	44.87	44.87
617.	154.66	136.37	299.67	264.23	50.34	50.34
548.	136.53	123.46	305.50	276.25	57.00	57.00
488.	113.30	103.82	293.00	268.47	59.61	59.61
434.	101.81	94.89	305.82	285.02	67.61	67.61
386.	85.78	80.67	300.50	282.62	71.33	71.33
343.	79.13	75.39	327.20	311.73	83.51	83.51
304.	72.95	70.21	359.90	346.37	98.20	98.20

RUN 10 $Tl=100.7(C)$ $VI=1.291(m/s)$ $Fr=17.852$ $Re=42046.9$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
813.	241.06	199.84	338.64	280.74	43.45	43.45
722.	206.01	177.14	331.82	285.33	48.48	48.48
644.	184.71	164.07	340.41	302.37	55.90	55.90
574.	162.86	147.99	344.47	313.01	62.67	62.67
511.	137.52	126.71	334.98	308.66	66.57	66.57
457.	127.86	119.92	359.54	337.22	77.75	77.75
406.	111.91	106.11	367.30	348.28	85.69	85.69
359.	100.55	96.33	389.28	372.93	97.64	97.64
317.	90.41	87.33	417.46	403.26	112.02	112.02

RUN 11 $Tl=100.7(C)$ $VI=1.539(m/s)$ $Fr=25.381$ $Re=50130.4$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
766.	219.15	184.68	329.49	277.66	45.05	45.05
686.	217.96	193.06	372.27	329.74	58.18	58.18
615.	186.47	168.33	362.51	327.24	62.46	62.46
551.	162.63	149.32	360.98	331.43	68.10	68.10
494.	146.50	136.70	373.05	348.08	76.73	76.73
441.	133.70	126.44	392.63	371.33	87.24	87.24
393.	117.93	112.59	403.54	385.26	96.36	96.36
349.	104.87	100.96	423.20	407.42	108.23	108.23
309.	97.10	94.24	467.11	453.34	127.55	127.55

RUN 12 $Tl=100.6(C)$ $VI=2.040(m/s)$ $Fr=44.599$ $Re=66428.8$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
784.	244.07	207.07	357.14	302.99	48.26	48.26
700.	236.96	210.62	395.73	351.73	61.20	61.20
554.	178.63	165.17	394.62	364.89	74.78	74.78
493.	159.60	149.80	406.62	381.66	84.15	84.15
438.	143.75	136.63	426.13	405.04	95.54	95.54
388.	125.55	120.40	437.76	419.80	105.76	105.76
345.	105.76	101.96	433.81	418.24	111.73	111.73

Test Sphere Diameter : 12.7 mm
 Support Thermocouple Wire Diameter: 0.81 mm
 Test Sphere Material : 316 Stainless Steel

RUN 1 Tl=100.1(C) Vl=0.011(m/s) Fr= 0.001 Re= 478.2
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 748. 145.70 118.92 225.06 183.69 40.46 40.46
 655. 124.72 106.62 225.11 192.44 46.84 46.84
 574. 102.06 89.67 215.91 189.69 50.64 50.64
 503. 88.15 79.56 219.34 197.95 57.54 57.54
 441. 74.13 68.09 217.83 200.09 62.69 62.69
 387. 63.96 59.68 223.39 208.45 70.07 70.07
 340. 54.75 51.69 228.69 215.91 77.38 77.38
 300. 47.00 44.79 236.65 225.54 85.78 85.78
 265. 38.87 37.26 236.86 227.02 90.63 90.63

RUN 2 Tl=100.0(C) Vl=0.123(m/s) Fr= 0.121 Re= 5309.8
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 832. 172.69 135.80 236.19 185.73 37.60 37.60
 713. 136.38 113.15 222.73 184.81 42.25 42.25
 613. 113.77 98.78 221.96 192.71 49.15 49.15
 528. 95.47 85.64 223.51 200.48 56.48 56.48
 455. 79.94 73.39 225.91 207.41 63.90 63.90
 393. 65.07 60.63 223.02 207.82 69.35 69.35
 341. 53.18 50.10 221.22 208.39 74.58 74.58
 298. 44.97 42.79 228.10 217.04 82.72 82.72

RUN 3 Tl=100.4(C) Vl=0.176(m/s) Fr= 0.249 Re= 7640.1
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 824. 166.24 130.43 229.92 180.40 36.82 36.82
 712. 133.78 110.68 218.97 181.17 41.48 41.48
 617. 111.93 96.69 216.90 187.37 47.60 47.60
 535. 93.50 83.29 215.29 191.78 53.57 53.57
 465. 78.69 71.76 216.37 197.31 60.05 60.05
 404. 65.93 61.14 217.34 201.56 66.24 66.24
 353. 54.91 51.54 217.61 204.26 71.90 71.90
 309. 46.71 44.31 224.03 212.53 79.66 79.66

RUN 4 Tl=100.6(C) Vl=0.253(m/s) Fr= 0.514 Re= 10982.2
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 724. 134.36 110.10 215.70 176.76 39.95 39.95
 637. 114.53 97.82 213.58 182.41 45.28 45.28
 561. 99.08 87.43 215.21 189.89 51.42 51.42
 494. 85.07 76.86 216.13 195.27 57.33 57.33
 436. 73.62 67.77 219.69 202.24 63.79 63.79
 385. 63.52 59.32 223.78 208.96 70.46 70.46
 340. 54.14 51.09 226.42 213.65 76.60 76.60

301. 46.80 44.57 233.92 222.75 84.54 84.54
 267. 40.44 38.80 244.02 234.12 93.25 93.25

RUN 5 $Tl=100.7(C)$ $VI=0.364(m/s)$ $Fr= 1.062$ $Re= 15784.2$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
749.	140.09	113.19	216.03	174.55	38.41 38.41
655.	117.71	99.61	212.44	179.77	43.76 43.76
574.	100.27	87.88	212.16	185.95	49.65 49.65
502.	84.87	76.31	211.60	190.27	55.36 55.36
440.	72.39	66.38	213.24	195.55	61.34 61.34
387.	61.49	57.23	215.09	200.17	67.32 67.32
340.	53.16	50.10	222.14	209.35	75.04 75.04
300.	45.44	43.22	228.05	216.90	82.42 82.42
265.	38.80	37.18	236.47	226.63	90.48 90.48

RUN 6 $Tl=100.7(C)$ $VI=0.449(m/s)$ $Fr= 1.622$ $Re= 19512.3$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
780.	147.28	116.97	216.93	172.29	36.75 36.75
678.	120.84	100.83	209.43	174.76	41.49 41.49
590.	102.18	88.74	208.75	181.28	47.48 47.48
515.	86.36	77.17	208.49	186.32	53.33 53.33
450.	72.84	66.48	208.73	190.51	59.05 59.05
394.	62.94	58.47	214.86	199.60	66.50 66.50
345.	53.01	49.85	217.35	204.37	72.78 72.78
303.	44.89	42.62	222.26	211.02	79.86 79.86
266.	38.74	37.10	234.03	224.14	89.30 89.30

RUN 7 $Tl=100.7(C)$ $VI=0.587(m/s)$ $Fr= 2.768$ $Re= 25491.9$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
750.	140.26	113.35	216.25	174.75	38.45 38.45
661.	118.57	100.01	211.83	178.67	43.22 43.22
583.	101.05	88.08	209.61	182.70	48.25 48.25
515.	86.04	76.85	207.70	185.53	53.10 53.10
456.	76.12	69.53	214.48	195.93	60.28 60.28
402.	65.66	60.93	217.76	202.07	66.57 66.57
356.	57.16	53.72	224.07	210.60	73.84 73.84
316.	48.46	45.93	225.69	213.93	79.45 79.45
280.	42.62	40.76	237.72	227.34	88.82 88.82

RUN 8 $Tl=100.8(C)$ $VI=0.843(m/s)$ $Fr= 5.713$ $Re= 36629.8$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
774.	166.57	136.90	247.35	203.28	43.60 43.60
684.	140.91	120.37	241.64	206.42	48.68 48.68
605.	123.67	109.26	245.31	216.72	55.81 55.81
534.	105.63	95.46	243.62	220.16	61.55 61.55
473.	91.05	83.77	244.71	225.16	67.84 67.84
419.	77.49	72.22	243.37	226.81	73.08 73.08
372.	68.91	65.07	254.48	240.29	82.46 82.46
330.	59.70	56.88	260.84	248.50	90.41 90.41

293. 52.03 49.95 271.39 260.54 100.04 100.04

RUN 9 $Tl=100.8(C)$ $VI=1.042(m/s)$ $Fr= 8.729$ $Re= 45285.8$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
781.	183.35	152.87	269.46	224.65	47.85	47.85
695.	158.66	137.18	267.26	231.07	53.90	53.90
617.	139.24	123.99	269.74	240.20	61.00	61.00
548.	118.50	107.57	264.78	240.36	66.08	66.08
488.	105.26	97.35	272.05	251.60	74.46	74.46
434.	90.55	84.78	272.00	254.66	80.55	80.55
386.	78.67	74.42	275.83	260.95	87.84	87.84

RUN 10 $Tl=100.8(C)$ $VI=1.361(m/s)$ $Fr= 14.890$ $Re= 59158.8$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
784.	196.34	165.48	287.24	242.10	51.40	51.40
701.	195.98	173.94	326.74	289.99	67.20	67.20
630.	155.35	139.21	293.90	263.36	65.95	65.95
567.	144.56	132.59	310.43	284.73	76.63	76.63
504.	137.12	128.45	339.77	318.29	92.32	92.32
451.	110.81	104.39	316.14	297.84	92.14	92.14
404.	100.75	95.96	331.98	316.19	103.90	103.90
361.	91.12	87.57	350.71	337.02	117.40	117.40
320.	87.82	85.21	400.73	388.79	143.45	143.45

RUN 11 $Tl=100.7(C)$ $VI=1.741(m/s)$ $Fr= 24.348$ $Re= 75589.7$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
807.	215.58	181.92	305.16	257.51	53.44	53.44
723.	206.65	182.52	332.43	293.61	66.45	66.45
646.	182.90	165.49	335.41	303.49	74.59	74.59
577.	160.02	147.41	336.00	309.53	82.30	82.30
514.	146.44	137.30	354.36	332.24	95.22	95.22
456.	127.59	121.01	359.54	340.99	104.92	104.92
404.	116.49	111.70	384.19	368.42	121.10	121.10
355.	106.47	103.05	418.42	404.97	142.12	142.12
312.	90.86	88.42	431.15	419.55	156.72	156.72

RUN 12 $Tl=100.3(C)$ $VI=2.244(m/s)$ $Fr= 40.476$ $Re= 97295.1$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
892.	145.39	99.76	183.86	126.15	24.12	24.12
796.	275.51	243.22	396.15	349.72	73.38	73.38
707.	242.23	219.64	399.93	362.63	83.49	83.49
625.	213.88	198.06	407.95	377.76	95.06	95.06
551.	188.78	177.69	419.08	394.46	108.07	108.07
484.	170.23	162.47	444.01	423.77	125.94	125.94
423.	152.98	147.59	475.28	458.54	147.09	147.09
367.	133.86	130.15	503.70	489.73	169.22	169.22
317.	115.96	113.40	535.70	523.88	194.05	194.05

Test Sphere Diameter : 19.1 mm
 Support Thermocouple Wire Diameter: 0.81 mm
 Test Sphere Material : 316 Stainless Steel

RUN 1 $T_i=100.8(C)$ $V_i=0.004(m/s)$ $Fr= 0.000$ $Re= 255.7$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
782.	178.04	129.21	261.55	189.81	60.63	60.63
676.	138.22	106.49	240.41	185.22	66.11	66.11
589.	112.58	91.27	230.88	187.17	73.68	73.68
515.	90.67	75.98	218.95	183.48	78.79	78.79
453.	77.49	67.15	220.37	190.97	88.49	88.49
399.	64.51	57.09	216.16	191.30	94.92	94.92
354.	54.00	48.59	213.66	192.26	101.45	101.45
314.	47.34	43.36	222.23	203.54	113.66	113.66

RUN 4 $T_i=100.8(C)$ $V_i=0.300(m/s)$ $Fr= 0.482$ $Re= 19538.6$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
782.	164.04	115.11	240.79	168.97	53.94	53.94
679.	127.12	95.01	220.04	164.46	58.52	58.52
593.	106.47	84.70	216.39	172.14	67.43	67.43
520.	87.23	72.13	208.01	171.99	73.38	73.38
458.	74.10	63.40	207.22	177.29	81.55	81.55
405.	62.32	54.61	204.74	179.41	88.32	88.32
359.	53.75	48.13	208.33	186.56	97.73	97.73
318.	45.95	41.82	211.21	192.22	106.63	106.63
283.	41.06	38.01	225.77	209.00	122.04	122.04

RUN 5 $T_i=100.8(C)$ $V_i=0.431(m/s)$ $Fr= 0.994$ $Re= 28074.0$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
797.	167.48	115.65	240.53	166.09	52.23	52.23
696.	122.60	87.98	205.93	147.78	51.61	51.61
612.	108.45	84.63	212.21	165.61	63.47	63.47
540.	88.71	71.99	202.15	164.05	68.36	68.36
478.	78.35	66.36	207.68	175.89	78.99	78.99
424.	67.01	58.32	207.36	180.46	86.69	86.69
377.	58.20	51.81	210.67	187.54	95.83	95.83
337.	48.96	44.20	207.77	187.57	101.38	101.38
301.	42.76	39.20	214.15	196.30	111.81	111.81

RUN 6 $T_i=100.8(C)$ $V_i=0.577(m/s)$ $Fr= 1.785$ $Re= 37621.8$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
791.	162.22	111.69	235.22	161.94	51.26	51.26
695.	138.62	104.23	233.45	175.53	61.40	61.40
613.	109.24	85.27	213.19	166.41	63.68	63.68
542.	93.84	76.94	212.80	174.46	72.51	72.51
427.	70.44	61.61	216.31	189.20	90.60	90.60
380.	62.05	55.53	222.38	199.05	101.34	101.34
339.	54.77	49.91	229.91	209.53	112.83	112.83
303.	46.72	43.08	231.30	213.31	121.08	121.08

RUN 7 $Tl=100.8(C)$ $VI=0.696(m/s)$ $Fr= 2.591$ $Re= 45337.7$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
776.	175.41	127.69	259.96	189.24	60.80 60.80
684.	135.28	102.40	231.95	175.57	62.11 62.11
607.	112.36	89.08	221.98	175.99	67.83 67.83
541.	95.26	78.45	216.56	178.35	74.22 74.22
484.	81.47	69.06	212.55	180.19	80.33 80.33
431.	72.53	63.45	219.62	192.12	91.46 91.46
386.	61.01	54.23	214.05	190.24	96.08 96.08
344.	57.81	52.77	237.69	216.97	116.02 116.02
307.	48.69	44.92	236.04	217.77	122.83 122.83

RUN 8 $Tl=100.9(C)$ $VI=0.999(m/s)$ $Fr= 5.349$ $Re= 65142.0$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
806.	201.25	147.64	285.36	209.34	65.24 65.24
711.	159.27	122.45	261.07	200.72	69.00 69.00
631.	138.08	112.04	260.33	211.23	79.19 79.19
563.	115.71	96.93	250.58	209.90	85.12 85.12
503.	97.60	83.82	242.61	208.35	90.79 90.79
452.	80.41	70.13	229.28	199.98	92.78 92.78
406.	75.31	67.56	246.85	221.45	108.92 108.92
366.	63.82	57.92	241.09	218.82	113.58 113.58
328.	55.25	50.81	243.64	224.04	122.64 122.64

RUN 9 $Tl=100.9(C)$ $VI=1.235(m/s)$ $Fr= 8.173$ $Re= 80534.8$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
823.	222.48	165.38	308.07	229.00	70.16 70.16
730.	182.86	143.06	290.69	227.42	76.61 76.61
652.	150.14	121.61	272.62	220.81	80.92 80.92
584.	128.22	107.38	265.48	222.33	87.99 87.99
525.	110.96	95.46	261.57	225.04	95.45 95.45
474.	96.00	84.31	257.47	226.10	102.08 102.08
428.	84.49	75.56	257.95	230.68	110.20 110.20
388.	73.60	66.74	256.79	232.87	117.36 117.36
350.	66.58	61.29	266.74	245.57	130.15 130.15

RUN 10 $Tl=100.9(C)$ $VI=1.613(m/s)$ $Fr= 13.942$ $Re=105200.9$

	Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
832.	251.01	191.93	343.18	262.40	79.66 79.66
744.	206.10	163.92	320.30	254.74	84.52 84.52
669.	177.53	146.72	312.44	258.20	92.83 92.83
602.	156.04	133.28	311.24	265.84	103.04 103.04
542.	139.19	122.29	315.65	277.31	115.26 115.26
487.	119.36	106.72	308.75	276.06	122.60 122.60
438.	110.84	101.38	329.01	300.93	142.05 142.05
393.	98.76	91.66	338.47	314.15	157.25 157.25

Appendix D

Typical Subcooled Single-Phase Experiment Data

Test Sphere Diameter : 12.7 mm
Support Thermocouple Wire Diameter: 0.81 mm
Test Sphere Material : 316 Stainless Steel

RUN 1 $Tl=95.4(C)$ $VI=0.011(m/s)$ $Fr=0.001$ $Re=468.4$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
768.	167.33	138.33	250.66	207.21	44.71	44.71
669.	138.72	119.44	244.03	210.11	50.34	50.34
584.	117.89	104.87	244.17	217.21	57.32	57.32
510.	97.53	88.61	238.60	216.78	62.47	62.47
444.	84.76	78.60	246.68	228.76	71.38	71.38
388.	72.34	68.04	252.00	237.02	79.59	79.59
339.	60.77	57.74	255.42	242.70	87.17	87.17
297.	50.54	48.38	257.41	246.38	94.00	94.00
262.	43.33	41.76	269.12	259.39	104.02	104.02

RUN 2 $Tl=95.1(C)$ $VI=0.123(m/s)$ $Fr=0.122$ $Re=5195.7$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
741.	169.27	143.27	264.50	223.87	49.70	49.70
653.	146.58	128.63	265.46	232.96	56.82	56.82
576.	123.79	111.27	260.75	234.39	62.43	62.43
508.	106.10	97.25	260.50	238.78	68.94	68.94
448.	93.32	87.02	268.51	250.37	77.74	77.74
395.	82.66	78.16	281.24	265.93	88.49	88.49
347.	74.11	70.90	301.70	288.64	102.55	102.55
303.	64.38	62.11	318.59	307.34	116.29	116.29
264.	58.83	57.24	361.52	351.73	140.69	140.69

RUN 3 $Tl=95.3(C)$ $VI=0.177(m/s)$ $Fr=0.251$ $Re=7472.2$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
743.	169.55	143.31	263.99	223.14	49.42	49.42
654.	145.79	127.75	263.48	230.87	56.24	56.24
576.	124.95	112.40	262.89	236.48	62.94	62.94
507.	107.48	98.70	264.77	243.13	70.31	70.31
445.	94.16	87.98	273.67	255.72	79.75	79.75
390.	83.61	79.25	289.31	274.24	91.85	91.85
341.	73.36	70.29	306.15	293.35	105.11	105.11
296.	65.28	63.14	334.61	323.63	123.69	123.69
255.	60.89	59.43	394.55	385.05	155.91	155.91

RUN 4 $Tl=95.5(C)$ $VI=0.254(m/s)$ $Fr=0.519$ $Re=10741.7$

Tb(C)	qt,qc(kW/m ²)	ht,hc(W/m ² /C)	Nut	Nuc		
741.	175.94	149.96	274.98	234.37	52.04	52.04
656.	150.70	132.53	271.55	238.80	58.07	58.07

581.	131.49	118.65	273.95	247.20	65.44	65.44
514.	114.15	105.01	276.20	254.09	72.82	72.82
455.	99.15	92.61	280.27	261.78	80.65	80.65
402.	87.01	82.30	289.35	273.70	90.27	90.27
354.	77.17	73.79	305.34	291.97	102.71	102.71
310.	72.01	69.61	345.18	333.68	125.04	125.04
268.	68.23	66.56	407.71	397.75	158.06	158.06

RUN 5 $Tl=95.5(C)$ $VI=0.410(m/s)$ $Fr=1.352$ $Re=17345.6$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
766.	191.84	163.17	288.63	245.48	53.13 53.13
683.	159.64	139.16	274.07	238.90	56.39 56.39
609.	146.27	131.61	288.07	259.20	66.48 66.48
541.	125.73	115.22	285.76	261.87	72.65 72.65
480.	112.60	105.02	296.88	276.89	82.70 82.70
425.	99.78	94.32	307.86	291.00	93.08 93.08
374.	94.64	90.74	346.90	332.60	113.83 113.83
326.	85.82	83.08	381.52	369.35	135.14 135.14
282.	76.10	74.21	420.45	410.00	159.81 159.81

RUN 8 $Tl=95.3(C)$ $VI=0.525(m/s)$ $Fr=2.210$ $Re=22162.5$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
847.	230.89	192.00	309.65	257.50	51.39 51.39
750.	197.83	170.91	304.95	263.44	57.95 57.95
663.	177.59	158.85	316.02	282.67	68.21 68.21
585.	155.68	142.56	321.37	294.29	77.51 77.51
515.	137.19	128.03	331.62	309.48	88.64 88.64
451.	122.54	116.14	350.08	331.80	102.70 102.70
394.	110.51	106.02	376.53	361.24	120.27 120.27
340.	99.21	96.15	414.81	402.03	144.14 144.14
290.	90.30	88.26	476.43	465.67	179.35 179.35

RUN 6 $Tl=95.4(C)$ $VI=0.589(m/s)$ $Fr=2.791$ $Re=24914.3$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
753.	211.06	183.80	323.81	281.99	61.84 61.84
674.	188.72	169.06	329.39	295.07	70.36 70.36
602.	168.64	154.42	336.46	308.10	79.62 79.62
537.	151.35	141.05	347.13	323.51	90.18 90.18
477.	136.82	129.38	363.83	344.04	103.15 103.15
422.	121.74	116.37	378.89	362.17	116.26 116.26
371.	115.96	112.12	428.78	414.60	142.35 142.35
323.	105.87	103.20	477.48	465.44	171.09 171.09
278.	95.99	94.17	542.95	532.66	208.84 208.84

RUN 7 $Tl=95.4(C)$ $VI=0.847(m/s)$ $Fr=5.761$ $Re=35790.6$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
814.	282.70	248.25	396.68	348.33	71.83 71.83
726.	250.83	226.37	401.41	362.27	81.71 81.71
647.	220.71	203.26	404.39	372.42	91.48 91.48
574.	206.91	194.50	437.34	411.10	109.69 109.69
506.	184.93	176.19	456.62	435.04	125.96 125.96

444. 162.76 156.63 474.75 456.87 142.70 142.70
 387. 148.31 144.05 519.00 504.09 169.56 169.56
 333. 136.13 133.24 586.51 574.04 207.88 207.88
 284. 127.20 125.28 696.07 685.56 266.55 266.55

RUN 9 $Tl=95.3(C)$ $VI=1.047(m/s)$ $Fr=8.805$ $Re=44226.1$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 884. 338.29 293.83 431.89 375.14 72.22 72.22
 791. 314.38 282.79 455.79 409.98 86.51 86.51
 705. 274.78 252.35 455.03 417.90 96.41 96.41
 627. 248.56 232.63 472.75 442.44 111.15 111.15
 553. 227.47 216.27 502.90 478.15 130.72 130.72
 485. 205.25 197.48 534.70 514.43 152.80 152.80
 422. 186.18 180.81 579.60 562.90 180.72 180.72
 363. 170.22 166.59 648.27 634.45 220.18 220.18
 308. 160.42 158.04 773.80 762.34 286.22 286.22

RUN 10 $Tl=95.3(C)$ $VI=1.367(m/s)$ $Fr=15.021$ $Re=57761.3$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 908. 382.17 333.88 473.47 413.65 77.86 77.86
 813. 404.41 370.04 567.97 519.70 107.24 107.24
 723. 367.28 343.06 590.00 551.09 124.62 124.62
 640. 329.95 313.00 611.77 580.35 143.58 143.58
 564. 298.77 286.96 645.45 619.96 167.40 167.40
 492. 270.98 262.87 692.43 671.71 197.75 197.75
 426. 247.07 241.56 759.19 742.26 237.02 237.02
 365. 222.60 218.93 842.56 828.66 286.94 286.94
 310. 196.57 194.17 941.67 930.16 348.48 348.48

RUN 12 $Tl=95.3(C)$ $VI=1.748(m/s)$ $Fr=24.555$ $Re=73844.5$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 766. 466.33 437.58 700.93 657.72 142.25 142.25
 683. 426.55 406.11 732.87 697.75 164.77 164.77
 605. 395.36 380.98 785.04 756.49 194.95 194.95
 532. 352.44 342.39 817.37 794.08 222.64 222.64
 466. 316.15 309.16 866.11 846.97 257.37 257.37
 405. 281.56 276.74 925.05 909.22 298.41 298.41
 350. 254.30 251.01 1020.27 1007.05 355.98 355.98

RUN 12 $Tl=95.3(C)$ $VI=2.208(m/s)$ $Fr=39.158$ $Re=93246.9$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 816. 558.24 523.50 780.83 732.23 150.65 150.65
 723. 506.50 482.31 814.17 775.30 175.39 175.39
 636. 464.40 447.80 868.51 837.47 208.28 208.28
 556. 411.58 400.23 904.51 879.57 239.69 239.69
 483. 367.21 359.52 961.35 941.19 280.19 280.19
 416. 324.26 319.08 1027.49 1011.07 326.99 326.99
 357. 289.35 285.90 1131.40 1117.89 391.61 391.61

RUN 1 $Tl=90.1(C)$ $VI=0.011(m/s)$ $Fr=0.001$ $Re=457.3$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc

772.	193.08	163.63	287.54	243.69	52.37	52.37
677.	164.53	144.62	285.72	251.14	59.70	59.70
593.	142.29	128.69	289.25	261.60	68.32	68.32
518.	121.99	112.65	292.33	269.96	76.98	76.98
453.	103.48	97.02	294.22	275.85	85.21	85.21
396.	87.78	83.24	297.34	281.96	93.67	93.67
350.	71.46	68.17	286.77	273.56	96.71	96.71
310.	62.88	60.48	301.29	289.78	108.57	108.57
273.	55.03	53.28	319.17	309.04	121.92	121.92

RUN 2 $Tl=90.0(C)$ $VI=0.124(m/s)$ $Fr=0.123$ $Re=5075.8$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
875.	240.36	197.30	310.49	254.87
755.	194.35	166.86	297.19	255.16
653.	161.43	143.46	292.26	259.74
565.	139.29	127.42	300.13	274.54
487.	122.19	114.32	316.52	296.12
417.	107.89	102.70	341.68	325.25
353.	95.84	92.47	379.87	366.51
295.	84.19	82.06	432.89	421.94
242.	79.36	78.08	562.19	553.13

RUN 3 $Tl=90.1(C)$ $VI=0.177(m/s)$ $Fr=0.253$ $Re=7295.0$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
853.	232.51	192.77	309.33	256.46
749.	193.35	166.46	298.21	256.74
660.	168.23	149.77	301.16	268.10
578.	147.53	134.84	308.97	282.41
505.	131.64	122.95	325.86	304.35
438.	118.30	112.38	350.92	333.35
377.	106.97	102.99	388.11	373.69
319.	95.69	93.09	437.85	425.94
265.	91.17	89.55	555.23	545.39

RUN 4 $Tl=90.1(C)$ $VI=0.255(m/s)$ $Fr=0.522$ $Re=10483.5$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
794.	215.09	183.14	310.54	264.42
714.	188.13	164.78	306.63	268.58
643.	170.58	153.44	314.87	283.24
577.	156.73	144.16	329.49	303.06
515.	142.96	133.79	345.31	323.15
457.	132.36	125.72	371.58	352.94
402.	123.43	118.71	409.76	394.09
350.	118.19	114.90	474.72	461.52
300.	112.06	109.85	564.21	553.10

RUN 5 $Tl=91.5(C)$ $VI=0.366(m/s)$ $Fr=1.077$ $Re=15155.4$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc
741.	199.58	173.55	311.72	271.07
672.	179.41	159.89	314.00	279.82
608.	164.34	149.71	323.95	295.11

549. 150.58 139.64 336.38 311.96 85.76 85.76
 492. 144.92 136.82 370.59 349.89 103.04 103.04
 437. 134.94 129.07 401.86 384.37 121.13 121.13
 386. 119.27 115.03 418.24 403.36 135.78 135.78
 338. 113.21 110.20 476.65 463.94 166.73 166.73
 293. 105.86 103.77 550.99 540.12 207.28 207.28

RUN 6 $Tl=91.3(C)$ $VI=0.526(m/s)$ $Fr=2.223$ $Re=21762.8$
 $Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 876. 279.57 236.32 360.56 304.79 59.12 59.12
 787. 247.09 215.98 360.34 314.96 66.73 66.73
 706. 222.57 199.99 367.57 330.28 76.06 76.06
 634. 198.01 181.54 371.50 340.60 84.87 84.87
 567. 186.80 174.82 400.96 375.24 100.97 100.97
 503. 170.73 162.14 424.82 403.44 117.27 117.27
 440. 165.54 159.54 487.96 470.28 147.55 147.55
 382. 144.56 140.43 514.41 499.72 169.15 169.15
 328. 136.80 134.02 602.49 590.23 215.30 215.30

RUN 7 $Tl=91.7(C)$ $VI=0.730(m/s)$ $Fr=4.287$ $Re=30280.4$
 $Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 765. 297.58 269.03 448.43 405.40 87.83 87.83
 689. 257.33 236.32 437.37 401.67 94.22 94.22
 619. 238.86 223.50 461.32 431.66 109.44 109.44
 552. 226.12 214.97 500.80 476.11 130.28 130.28
 490. 206.73 198.70 530.68 510.06 150.48 150.48
 432. 186.75 181.05 564.38 547.16 173.50 173.50
 375. 187.38 183.44 683.20 668.84 228.46 228.46
 321. 165.04 162.40 749.43 737.45 271.68 271.68

RUN 8 $Tl=91.9(C)$ $VI=0.954(m/s)$ $Fr=7.312$ $Re=39592.3$
 $Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 747. 345.85 319.17 535.03 493.76 108.88 108.88
 674. 309.99 290.34 541.23 506.94 120.91 120.91
 605. 281.19 266.79 557.90 529.33 136.35 136.35
 540. 263.36 252.87 599.27 575.41 159.73 159.73
 479. 244.43 236.89 646.05 626.12 187.21 187.21
 422. 222.52 217.15 692.64 675.94 217.01 217.01

RUN 9 $Tl=90.9(C)$ $VI=1.138(m/s)$ $Fr=10.407$ $Re=46993.2$
 $Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 773. 422.80 393.32 629.38 585.49 125.79 125.79
 694. 382.59 361.15 644.94 608.79 142.06 142.06
 620. 349.47 334.01 673.02 643.24 162.80 162.80
 551. 325.98 314.89 723.64 699.02 191.52 191.52
 486. 297.75 289.89 772.17 751.80 222.84 222.84
 426. 277.53 272.03 853.31 836.39 267.14 267.14

RUN 10 $Tl=91.0(C)$ $VI=1.441(m/s)$ $Fr=16.695$ $Re=59524.2$
 $Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 838. 500.41 462.74 679.12 628.01 126.43 126.43

750. 497.97 471.01 767.29 725.75 159.61 159.61
 667. 443.65 424.58 783.88 750.18 180.24 180.24
 590. 399.79 386.39 817.96 790.55 207.21 207.21
 518. 367.60 358.25 880.37 857.98 244.58 244.58
 452. 338.13 331.69 962.87 944.53 291.95 291.95
 389. 318.00 313.69 1105.37 1090.37 365.82 365.82

RUN 11 $Tl=91.1(C)$ $VI=1.753(m/s)$ $Fr=24.698$ $Re=72455.5$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 791. 566.03 534.35 819.85 773.98 163.22 163.22
 713. 556.75 533.50 908.87 870.91 199.04 199.04
 639. 512.58 495.69 951.88 920.52 227.97 227.97
 570. 463.07 450.90 987.20 961.25 257.69 257.69
 506. 417.72 408.96 1030.76 1009.16 292.10 292.10
 446. 389.01 382.78 1125.84 1107.82 344.85 344.85
 390. 366.36 362.01 1269.12 1254.07 420.20 420.20

RUN 1 $Tl=79.6(C)$ $VI=0.011(m/s)$ $Fr=0.001$ $Re=436.6$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 856. 269.52 205.12 356.90 271.62 53.71 53.71
 752. 224.61 181.04 344.72 277.85 60.94 60.94
 664. 183.99 153.89 326.91 273.43 65.92 65.92
 587. 161.99 140.80 333.04 289.47 76.08 76.08
 521. 132.03 116.89 314.49 278.43 79.15 79.15
 463. 117.67 106.66 324.69 294.31 89.72 89.72
 410. 107.83 99.86 348.42 322.67 105.19 105.19
 361. 98.98 93.27 380.28 358.34 124.75 124.75

RUN 1 $Tl=79.6(C)$ $VI=0.054(m/s)$ $Fr=0.023$ $Re=2090.4$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 800. 251.83 199.52 360.42 285.56 59.73 59.73
 714. 214.07 176.80 349.21 288.42 65.88 65.88
 640. 181.90 154.84 337.55 287.33 71.13 71.13
 574. 161.50 141.62 341.20 299.19 79.81 79.81
 516. 145.40 130.66 350.53 314.99 90.10 90.10
 463. 132.91 121.93 367.28 336.95 102.78 102.78
 414. 113.52 105.37 362.87 336.84 109.33 109.33
 368. 105.30 99.31 394.22 371.78 128.23 128.23

RUN 2 $Tl=79.2(C)$ $VI=0.124(m/s)$ $Fr=0.124$ $Re=4841.0$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 825. 259.10 201.64 357.95 278.57 56.81 56.81
 738. 225.74 184.61 354.32 289.77 64.52 64.52
 659. 207.96 178.49 372.66 319.84 77.51 77.51
 585. 188.35 167.37 388.86 345.54 91.02 91.02
 515. 178.32 163.61 430.32 394.82 112.99 112.99
 449. 162.89 152.75 467.56 438.46 136.00 136.00
 388. 148.66 141.80 518.63 494.70 166.21 166.21
 330. 137.79 133.27 602.03 582.29 211.85 211.85

RUN 3 $Tl=79.2(C)$ $VI=0.179(m/s)$ $Fr=0.256$ $Re=6955.4$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
918.	305.28	225.48	373.80	276.09	51.49	51.49
827.	275.45	217.43	379.15	299.28	60.87	60.87
745.	244.68	202.42	379.98	314.34	69.49	69.49
669.	226.96	196.21	399.79	345.62	82.88	82.88
596.	215.34	193.27	435.14	390.54	101.66	101.66
524.	198.77	183.33	469.33	432.88	122.50	122.50
457.	190.44	179.83	534.96	505.16	155.22	155.22
392.	178.08	171.02	612.21	587.95	196.43	196.43

RUN 4 $Tl=79.5(C)$ $VI=0.279(m/s)$ $Fr=0.626$ $Re=10887.0$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
953.	336.69	246.83	395.07	289.62	52.24	52.24
865.	317.87	251.42	416.08	329.10	64.53	64.53
782.	289.88	241.00	425.69	353.91	75.34	75.34
702.	275.57	240.10	458.41	399.41	92.42	92.42
627.	249.88	224.40	475.38	426.90	107.26	107.26
552.	238.35	220.59	528.91	489.50	134.08	134.08
482.	227.86	215.61	597.99	565.84	168.64	168.64
414.	215.57	207.43	689.22	663.20	215.28	215.28
348.	200.32	195.12	809.75	788.73	279.54	279.54

RUN 5 $Tl=79.5(C)$ $VI=0.369(m/s)$ $Fr=1.093$ $Re=14378.3$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
820.	313.84	257.45	436.63	358.18	73.41	73.41
746.	291.73	249.27	452.23	386.41	85.33	85.33
676.	273.44	241.68	475.39	420.16	99.95	99.95
608.	263.66	240.32	520.33	474.27	121.78	121.78
542.	244.95	228.02	555.17	516.81	143.16	143.16
480.	235.76	223.67	622.57	590.65	176.53	176.53
419.	222.89	214.49	701.43	674.99	217.68	217.68

RUN 6 $Tl=79.6(C)$ $VI=0.494(m/s)$ $Fr=1.961$ $Re=19273.1$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
798.	344.97	292.96	494.79	420.20	88.02	88.02
733.	326.00	285.69	515.67	451.91	101.15	101.15
670.	313.35	282.41	550.57	496.21	118.81	118.81
610.	291.31	267.74	572.55	526.23	134.80	134.80
551.	277.45	259.73	616.17	576.80	158.06	158.06
496.	262.69	249.46	665.25	631.74	185.20	185.20
441.	251.36	241.70	738.73	710.35	222.59	222.59

RUN 7 $Tl=81.7(C)$ $VI=0.595(m/s)$ $Fr=2.840$ $Re=23431.6$

Tb(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc		
835.	387.68	328.01	528.07	446.78	90.18	90.18
762.	355.96	310.64	538.06	469.55	101.94	101.94
692.	336.47	302.47	569.22	511.70	119.67	119.67
624.	314.00	288.84	600.45	552.32	139.20	139.20
560.	292.88	274.40	638.62	598.31	162.36	162.36
498.	269.07	255.68	677.74	644.02	188.33	188.33

RUN 8 $Tl=82.0(C)$ $VI=0.854(m/s)$ $Fr=5.862$ $Re=33701.0$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 845. 486.09 424.19 653.17 570.00 113.92 113.92
 775. 483.05 435.46 716.61 646.00 138.47 138.47
 707. 449.74 413.59 742.51 682.82 157.21 157.21
 642. 427.68 400.36 790.52 740.02 182.75 182.75
 578. 415.09 394.88 870.85 828.45 220.16 220.16
 515. 388.18 373.45 936.36 900.84 257.75 257.75

RUN 9 $Tl=81.9(C)$ $VI=1.074(m/s)$ $Fr=9.270$ $Re=42367.6$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 940. 647.30 561.39 771.86 669.42 122.30 122.30
 852. 599.80 536.24 798.12 713.55 141.61 141.61
 770. 546.75 500.05 817.04 747.25 160.97 160.97
 692. 515.83 481.76 871.95 814.36 190.36 190.36
 616. 493.97 469.67 958.44 911.28 231.67 231.67
 544. 459.13 442.08 1037.18 998.66 276.15 276.15

RUN 10 $Tl=79.9(C)$ $VI=1.423(m/s)$ $Fr=16.280$ $Re=55610.3$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 787. 731.38 681.61 1066.67 994.08 210.62 210.62
 709. 704.09 667.51 1157.05 1096.94 251.79 251.79
 635. 664.54 638.06 1244.33 1194.75 297.36 297.36
 563. 628.42 609.59 1359.42 1318.69 356.32 356.32

RUN 11 $Tl=79.8(C)$ $VI=1.646(m/s)$ $Fr=21.779$ $Re=64285.3$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 1014. 964.34 855.19 1056.45 936.88 160.05 160.05
 861. 865.34 799.91 1139.12 1052.98 207.33 207.33
 719. 769.78 731.70 1245.14 1183.54 268.85 268.85
 587. 694.15 672.99 1428.03 1384.50 363.99 363.99

RUN 12 $Tl=79.8(C)$ $VI=2.035(m/s)$ $Fr=33.282$ $Re=79443.9$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 737. 912.38 871.45 1434.67 1370.31 305.49 305.49
 620. 849.42 824.73 1637.44 1589.85 402.62 402.62

RUN 1 $Tl=71.4(C)$ $VI=0.011(m/s)$ $Fr=0.001$ $Re=418.3$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 890. 341.66 269.15 433.15 341.23 65.36 65.36
 768. 265.49 219.08 397.69 328.17 70.81 70.81
 671. 218.21 187.19 383.02 328.58 78.63 78.63
 586. 184.61 163.60 380.92 337.56 88.89 88.89
 511. 161.76 147.41 394.69 359.67 103.51 103.51
 444. 144.14 134.35 420.66 392.08 122.49 122.49
 383. 127.73 121.09 453.32 429.77 145.33 145.33
 328. 114.35 109.90 503.59 483.97 176.53 176.53

RUN 1 $Tl=70.4(C)$ $VI=0.011(m/s)$ $Fr=0.001$ $Re=416.3$
 Tb(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc
 875. 332.81 263.91 429.87 340.87 66.19 66.19

757. 261.68 217.37 399.02 331.46 72.39 72.39
 658. 219.12 189.71 393.00 340.24 82.50 82.50
 573. 187.81 168.04 397.77 355.90 95.07 95.07
 497. 162.16 148.84 409.34 375.70 109.98 109.98
 429. 147.99 139.03 451.27 423.97 134.95 134.95
 368. 126.04 120.06 472.72 450.32 155.42 155.42

RUN 2 $Tl=70.4(C)$ $VI=0.125(m/s)$ $Fr=0.126$ $Re=4623.5$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 913. 328.46 249.81 404.31 307.51 57.59 57.59
 836. 331.32 271.47 450.80 369.36 74.50 74.50
 763. 303.84 258.40 458.84 390.23 84.66 84.66
 695. 284.26 249.85 478.60 420.66 98.09 98.09
 629. 271.88 246.16 515.26 466.52 116.95 116.95
 565. 253.07 234.10 545.75 504.86 136.19 136.19
 503. 233.21 219.41 579.42 545.14 158.34 158.34
 447. 220.97 210.99 639.13 610.26 189.92 189.92

RUN 3 $Tl=70.5(C)$ $VI=0.180(m/s)$ $Fr=0.259$ $Re=6645.2$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 830. 345.66 287.17 474.38 394.12 79.99 79.99
 760. 318.39 273.46 482.91 414.77 90.25 90.25
 694. 300.40 266.09 506.38 448.55 104.67 104.67
 629. 290.97 265.18 550.80 501.98 125.75 125.75
 567. 272.70 253.56 585.67 544.55 146.57 146.57
 507. 252.11 238.07 621.35 586.75 169.73 169.73

RUN 4 $Tl=70.6(C)$ $VI=0.258(m/s)$ $Fr=0.535$ $Re=9552.4$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 868. 384.32 317.14 501.03 413.44 80.83 80.83
 759. 361.81 317.09 549.72 481.76 104.95 104.95
 654. 350.97 322.10 634.30 582.13 141.81 141.81
 555. 310.45 292.39 683.83 644.05 175.72 175.72

RUN 5 $Tl=70.6(C)$ $VI=0.371(m/s)$ $Fr=1.104$ $Re=13730.1$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 841. 435.18 374.18 587.95 505.54 101.44 101.44
 777. 408.80 360.83 604.64 533.69 114.17 114.17
 716. 386.63 349.04 628.60 567.49 129.35 129.35
 656. 363.95 334.92 656.19 603.84 146.88 146.88
 598. 351.66 329.31 706.77 661.84 171.77 171.77
 542. 333.46 316.55 756.06 717.71 198.85 198.85

RUN 6 $Tl=70.7(C)$ $VI=0.497(m/s)$ $Fr=1.983$ $Re=18402.8$

$Tb(C)$ $qt, qc(kW/m^2)$ $ht, hc(W/m^2/C)$ Nut Nuc
 832. 472.39 413.36 646.06 565.32 114.44 114.44
 775. 452.05 404.55 671.09 600.56 128.79 128.79
 720. 419.05 380.89 677.27 615.59 139.76 139.76
 666. 407.77 377.32 721.08 667.22 160.39 160.39
 615. 394.47 370.34 767.62 720.65 183.52 183.52
 562. 394.03 375.28 853.89 813.26 219.96 219.96

511. 406.24 391.88 991.12 956.09 275.15 275.15

RUN 7 $Tl=70.7(C)$ $VI=0.599(m/s)$ $Fr=2.879$ $Re=22177.6$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
884.	537.90	466.79	686.80	596.00	114.75
818.	515.57	459.53	718.98	640.84	131.56
755.	495.07	451.08	757.02	689.76	150.92
693.	475.81	441.61	803.13	745.41	174.09
634.	448.42	422.07	841.23	791.78	197.28
577.	412.91	392.75	867.14	824.79	219.31

RUN 8 $Tl=70.9(C)$ $VI=0.860(m/s)$ $Fr=5.941$ $Re=31898.3$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
898.	729.73	655.21	915.99	822.46	156.38
822.	699.67	642.71	969.78	890.84	182.09

RUN 9 $Tl=71.0(C)$ $VI=1.063(m/s)$ $Fr=9.077$ $Re=39444.2$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
848.	829.97	767.49	1111.37	1027.70	204.89
777.	808.00	759.96	1194.40	1123.38	240.22

RUN 10 $Tl=70.9(C)$ $VI=1.388(m/s)$ $Fr=15.488$ $Re=51484.0$

	$Tb(C)$	$qt, qc(kW/m^2)$	$ht, hc(W/m^2/C)$	Nut	Nuc
992.	1050.83	948.90	1179.19	1064.81	185.42
888.	1012.78	940.81	1287.48	1195.99	229.53
786.	947.76	898.13	1383.68	1311.21	278.01
687.	886.76	853.42	1511.95	1455.10	341.97

Appendix E

Typical Upward Two-Phase Experiment Data

Test Sphere Diameter : 12.7 mm
 Test Sphere Material : 316 Stainless Steel
 No. 1 Tube needle Assembly

***** TEST: TU1-1 *****

RUN 1 Water V =0.690(m/s) Steam V = 3.218(m/s)
 Re = 29794.9 Void Fraction =0.436

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
742.	182.8	151.5	284.9	236.1	63.3	52.4 0.568
666.	153.7	130.9	271.5	231.2	65.3	55.6 0.560
634.	141.2	121.4	264.5	227.4	66.0	56.7 0.552
575.	120.6	105.6	253.9	222.3	67.7	59.3 0.540
549.	112.8	99.6	251.4	222.1	69.2	61.1 0.539
500.	107.2	97.1	268.4	243.1	78.4	71.0 0.589
455.	89.5	81.6	252.1	230.0	77.7	70.9 0.553
435.	85.2	78.2	254.5	233.7	80.5	73.9 0.560
415.	78.6	72.5	249.5	230.0	80.9	74.6 0.549
397.	79.1	73.6	266.6	248.1	88.6	82.4 0.590
379.	73.3	68.4	262.9	245.5	89.4	83.5 0.580
362.	67.6	63.3	257.9	241.4	89.8	84.0 0.567
345.	73.7	69.9	300.8	285.2	107.2	101.6 0.666

RUN 2 Water V =0.801(m/s) Steam V = 3.319(m/s)
 Re = 34573.5 Void Fraction =0.422

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
738.	193.2	162.3	302.8	254.4	67.5	56.7 0.568
698.	168.5	142.3	281.6	237.8	65.5	55.3 0.534
632.	140.1	120.5	263.5	226.7	65.9	56.7 0.511
604.	134.8	117.6	267.5	233.4	69.1	60.2 0.527
576.	128.6	113.6	270.3	238.7	72.1	63.6 0.538
524.	107.2	95.6	252.5	225.3	71.5	63.8 0.507
437.	88.4	81.3	262.1	241.1	82.6	76.0 0.537
418.	81.8	75.5	257.2	237.5	83.1	76.7 0.527
400.	77.6	72.0	258.7	240.1	85.6	79.4 0.530
383.	77.8	72.8	275.1	257.5	93.1	87.1 0.566
366.	75.1	70.7	282.8	266.1	98.0	92.2 0.581

RUN 3 Water V =0.934(m/s) Steam V = 3.457(m/s)
 Re = 40316.1 Void Fraction =0.406

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
736.	204.1	173.5	321.0	272.9	71.7	60.9 0.565
696.	186.9	160.9	313.6	270.1	73.1	63.0 0.561
662.	158.2	135.8	281.6	241.7	68.1	58.5 0.503
631.	147.2	127.7	277.2	240.4	69.3	60.2 0.501

603.	140.5	123.4	279.5	245.5	72.3	63.5	0.513
575.	122.3	107.3	257.3	225.7	68.6	60.2	0.471
525.	104.9	93.3	246.9	219.6	69.9	62.2	0.458
481.	92.4	83.3	242.9	218.9	72.6	65.4	0.455
460.	95.7	87.6	265.9	243.4	81.5	74.6	0.504
421.	84.7	78.3	263.6	243.6	84.8	78.4	0.501
386.	77.9	72.8	272.1	254.2	91.6	85.6	0.518

RUN 4 Water V = 1.106(m/s) Steam V = 3.666(m/s)
 Re = 47737.7 Void Fraction = 0.382

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
726.	251.1	221.7	401.0	354.1	90.5	79.9	0.674
688.	208.9	183.9	355.3	312.7	83.5	73.5	0.597
656.	177.9	156.0	319.9	280.6	77.8	68.3	0.537
604.	139.6	122.3	276.9	242.7	71.5	62.6	0.466
556.	126.2	112.6	276.9	247.0	75.5	67.4	0.474
534.	118.2	106.0	272.4	244.3	76.3	68.4	0.469
513.	109.1	98.2	264.2	237.8	75.9	68.3	0.456
493.	109.4	99.6	278.4	253.5	82.0	74.6	0.485
456.	100.3	92.4	281.6	259.4	86.7	79.8	0.493
422.	96.0	89.5	298.4	278.4	96.0	89.5	0.526
405.	84.2	78.4	275.9	257.0	90.7	84.4	0.484

RUN 5 Water V = 1.242(m/s) Steam V = 3.864(m/s)
 Re = 53596.5 Void Fraction = 0.363

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
739.	242.9	211.9	379.9	331.4	84.5	73.7	0.594
700.	214.4	188.0	357.2	313.2	82.9	72.7	0.564
667.	175.2	152.4	309.3	269.0	74.4	64.7	0.486
638.	155.2	135.2	288.9	251.5	71.7	62.5	0.455
611.	147.4	129.6	288.5	253.6	73.8	64.9	0.459
562.	120.2	106.1	260.1	229.7	70.4	62.2	0.416
518.	118.6	107.4	283.9	257.1	81.1	73.4	0.465
497.	114.3	104.3	287.9	262.7	84.4	77.0	0.475
459.	101.6	93.6	283.5	261.1	87.0	80.1	0.469
441.	98.1	90.9	288.1	266.9	90.4	83.8	0.478
407.	92.4	86.6	301.6	282.6	98.9	92.7	0.502

RUN 6 Water V = 1.453(m/s) Steam V = 4.251(m/s)
 Re = 62688.1 Void Fraction = 0.330

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
721.	278.2	249.4	447.9	401.5	101.6	91.1	0.667
685.	217.4	192.7	372.0	329.8	87.8	77.8	0.550
629.	155.5	136.2	294.3	257.8	73.8	64.7	0.431
605.	149.1	131.8	295.2	260.9	76.1	67.2	0.437
582.	142.7	127.2	296.0	263.9	78.3	69.8	0.442
541.	142.2	129.6	322.6	293.9	89.6	81.6	0.492
501.	114.9	104.7	286.5	261.0	83.5	76.1	0.436
484.	106.9	97.6	278.3	254.0	82.8	75.6	0.424
467.	106.6	98.2	290.4	267.4	88.2	81.2	0.445
451.	93.6	85.9	267.0	245.1	82.8	76.0	0.406

RUN 7 Water V =1.615(m/s) Steam V = 4.645(m/s)
 Re = 69709.9 Void Fraction =0.302

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
788.	310.6	273.1	451.4	396.9	95.6	84.0 0.621
743.	257.8	226.4	401.2	352.4	89.0	78.2 0.554
706.	211.2	184.2	348.6	303.9	80.4	70.1 0.480
648.	166.5	145.5	304.1	265.7	74.7	65.2 0.421
597.	147.7	131.1	297.5	264.1	77.4	68.7 0.420
573.	142.0	127.1	300.1	268.7	80.2	71.8 0.427
529.	121.6	109.7	283.1	255.5	79.7	71.9 0.405
510.	114.1	103.4	278.6	252.4	80.3	72.8 0.400
491.	108.8	99.2	278.7	254.0	82.3	75.0 0.402
472.	107.6	98.9	289.2	265.8	87.3	80.3 0.420

RUN 8 Water V =1.912(m/s) Steam V = 5.739(m/s)
 Re = 82536.3 Void Fraction =0.244

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
721.	299.9	271.1	482.9	436.5	109.6	99.0 0.632
681.	238.5	214.2	410.8	368.9	97.3	87.4 0.536
649.	198.7	177.6	362.4	323.9	88.9	79.5 0.472
595.	160.1	143.6	323.4	290.0	84.3	75.6 0.424
571.	150.2	135.5	318.6	287.4	85.3	77.0 0.420
550.	137.4	124.2	305.8	276.5	84.0	76.0 0.403
509.	125.4	114.7	306.6	280.5	88.5	81.0 0.409
489.	122.4	112.8	314.4	289.8	93.0	85.7 0.421
453.	110.8	103.1	313.9	291.9	97.0	90.2 0.422
435.	118.6	111.6	354.1	333.3	111.9	105.4 0.480

RUN 9 Water V =2.047(m/s) Steam V = 6.522(m/s)
 Re = 88347.9 Void Fraction =0.215

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
883.	355.1	301.9	453.4	385.4	87.4	74.3 0.529
833.	284.3	239.9	388.0	327.3	78.5	66.3 0.452
790.	255.0	217.3	369.9	315.1	78.2	66.6 0.438
717.	196.3	168.1	318.5	272.6	72.6	62.2 0.382
652.	181.0	159.6	327.9	289.1	80.1	70.7 0.407
595.	159.2	142.8	322.0	288.7	84.0	75.3 0.408
543.	141.5	128.8	319.6	290.9	88.6	80.6 0.410
519.	129.3	118.1	308.6	281.8	88.0	80.3 0.397
497.	120.6	110.6	304.0	278.8	89.1	81.7 0.392
475.	120.6	111.7	321.6	298.0	96.7	89.6 0.418

***** TEST: TU1-2 *****

RUN 1 Water V =0.854(m/s) Steam V = 3.887(m/s)
 Re = 36869.8 Void Fraction =0.544

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
710.	199.8	172.3	327.5	282.4	75.2	64.8 0.613
673.	169.6	146.0	295.8	254.8	70.7	60.8 0.554
639.	154.3	134.1	286.3	248.7	71.0	61.7 0.542
608.	134.5	116.9	264.7	230.2	68.0	59.1 0.503

551.	121.2	107.9	269.0	239.6	73.8	65.8	0.523
525.	115.2	103.6	271.2	243.9	76.8	69.1	0.532
477.	105.2	96.3	279.5	255.8	83.9	76.8	0.556
454.	95.6	87.7	269.7	247.6	83.2	76.4	0.536
434.	89.0	82.1	266.8	246.1	84.5	77.9	0.530
415.	78.5	72.3	249.5	229.9	81.0	74.6	0.494
362.	72.0	67.7	274.6	258.1	95.5	89.8	0.545
346.	67.4	63.5	274.0	258.4	97.5	92.0	0.542

RUN 2 Water V = 0.993(m/s) Steam V = 3.961(m/s)
 Re = 42843.3 Void Fraction = 0.534

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
700.	194.5	168.1	324.2	280.2	75.2 65.0 0.565
663.	168.6	146.1	299.5	259.5	72.3 62.7 0.524
630.	151.3	131.9	285.6	248.9	71.5 62.4 0.504
599.	138.8	122.1	278.5	244.9	72.3 63.6 0.496
493.	107.8	98.0	274.7	249.8	80.9 73.6 0.505
469.	102.4	93.9	277.8	254.6	84.2 77.2 0.512
447.	86.1	78.6	248.1	226.5	77.2 70.5 0.454
427.	90.1	83.4	275.7	255.4	88.1 81.6 0.510
388.	78.3	73.1	271.6	253.7	91.3 85.3 0.502
370.	77.2	72.6	285.7	268.8	98.3 92.5 0.528
353.	74.5	70.5	294.6	278.6	103.8 98.2 0.544

RUN 3 Water V = 1.376(m/s) Steam V = 4.199(m/s)
 Re = 59386.5 Void Fraction = 0.504

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
690.	201.8	176.5	341.9	299.0	80.2 70.1 0.512
655.	172.8	151.1	311.3	272.1	75.8 66.3 0.467
624.	159.3	140.4	303.9	267.8	76.6 67.5 0.460
595.	147.8	131.3	298.5	265.2	77.8 69.1 0.457
519.	117.0	105.8	279.5	252.7	79.7 72.1 0.434
453.	99.7	92.0	282.5	260.4	87.3 80.5 0.444
434.	95.2	88.3	285.5	264.7	90.4 83.8 0.449
380.	79.6	74.7	283.9	266.4	96.4 90.4 0.446
364.	78.0	73.6	295.1	278.6	102.4 96.7 0.464

RUN 4 Water V = 1.376(m/s) Steam V = 4.199(m/s)
 Re = 59386.5 Void Fraction = 0.504

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
706.	244.0	216.9	402.7	358.0	92.9 82.6 0.612
668.	196.3	173.4	345.9	305.5	83.1 73.4 0.524
635.	173.5	153.6	324.2	287.0	80.7 71.5 0.493
579.	138.6	123.4	289.7	257.8	76.9 68.5 0.444
553.	138.1	124.6	304.9	275.3	83.5 75.3 0.473
528.	127.6	115.8	298.0	270.4	84.0 76.2 0.465
505.	120.4	110.0	297.5	271.7	86.3 78.9 0.467
483.	110.3	101.1	288.2	264.1	85.9 78.7 0.452
442.	101.1	93.8	296.0	274.7	92.8 86.1 0.467
423.	94.6	88.1	293.2	273.1	94.1 87.7 0.463
371.	83.0	78.4	306.8	289.9	105.6 99.8 0.484

RUN 5 Water V = 1.548(m/s) Steam V = 4.327(m/s)
 Re = 66790.6 Void Fraction = 0.489

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
734.	263.9	233.6	416.5	368.7	93.2	82.5 0.593
690.	241.2	215.8	408.5	365.6	95.8	85.7 0.590
654.	195.6	173.9	352.9	313.9	86.0	76.5 0.508
595.	154.2	137.7	311.5	278.2	81.2	72.5 0.452
570.	139.7	125.1	297.6	266.5	79.9	71.6 0.432
522.	128.4	117.0	304.7	277.6	86.6	78.9 0.450
499.	123.2	113.1	308.6	283.3	90.2	82.8 0.459
458.	111.2	103.2	310.6	288.3	95.4	88.5 0.464
439.	104.7	97.6	309.4	288.4	97.4	90.8 0.462
402.	92.2	86.5	304.8	286.0	100.5	94.3 0.455
386.	89.5	84.4	313.4	295.6	105.7	99.7 0.468

RUN 6 Water V = 1.815(m/s) Steam V = 4.560(m/s)
 Re = 78331.1 Void Fraction = 0.464

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
702.	270.3	243.8	449.3	405.1	104.1	93.8 0.604
666.	212.9	190.1	376.4	336.1	90.6	80.9 0.502
609.	169.5	151.8	333.0	298.3	85.4	76.6 0.447
584.	163.1	147.5	337.2	304.9	89.1	80.5 0.457
539.	137.6	125.2	313.9	285.5	87.4	79.5 0.427
497.	127.4	117.4	320.9	295.8	94.0	86.6 0.442
459.	115.4	107.4	321.5	299.1	98.6	91.7 0.444
425.	101.8	95.2	313.5	293.4	100.4	94.0 0.433
408.	98.6	92.7	320.1	300.9	104.7	98.5 0.443

RUN 7 Water V = 2.023(m/s) Steam V = 4.779(m/s)
 Re = 87287.0 Void Fraction = 0.442

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
726.	288.1	258.8	460.4	413.5	103.9	93.3 0.582
687.	257.0	232.1	438.0	395.4	103.1	93.1 0.559
626.	185.4	166.3	352.6	316.3	88.7	79.6 0.449
578.	153.9	138.7	322.2	290.5	85.7	77.3 0.412
535.	140.2	128.0	322.4	294.2	90.1	82.3 0.417
497.	126.6	116.7	319.2	294.1	93.6	86.2 0.416
479.	121.5	112.5	321.0	297.2	96.1	89.0 0.420
444.	109.6	102.2	318.3	296.9	99.4	92.7 0.417
412.	98.8	92.7	316.8	297.4	103.2	96.9 0.415

RUN 8 Water V = 2.339(m/s) Steam V = 5.198(m/s)
 Re = 100922.4 Void Fraction = 0.407

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
716.	306.4	278.2	497.4	451.7	113.5	103.0 0.592
680.	247.8	223.6	427.0	385.2	101.2	91.3 0.506
625.	188.0	169.0	358.2	322.0	90.2	81.1 0.425
579.	162.1	146.8	338.4	306.5	89.8	81.4 0.405
538.	142.9	130.5	326.6	298.2	91.0	83.1 0.393
500.	129.2	119.0	322.7	297.3	94.2	86.8 0.392

466.	119.3	110.9	326.1	303.2	99.2	92.2	0.397
449.	118.7	111.1	339.8	318.1	105.5	98.7	0.416
418.	111.6	105.4	351.6	331.9	113.7	107.3	0.431

RUN 9 Water V = 2.578(m/s) Steam V = 5.616(m/s)
Re = 111241.4 Void Fraction = 0.377

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
729.	298.1	268.4	473.8	426.4	106.5
693.	273.8	248.2	461.7	418.5	107.9
636.	203.6	183.7	380.2	343.0	94.6
590.	172.4	156.3	351.7	318.8	92.2
570.	166.6	152.0	354.9	323.9	95.3
531.	144.9	132.9	336.1	308.2	94.4
514.	136.7	125.8	330.7	304.2	94.9
480.	131.2	122.1	345.5	321.6	103.4
432.	127.3	120.5	383.6	362.9	121.7

***** TEST: TU1-3 *****

RUN 1 Water V = 1.214(m/s) Steam V = 3.965(m/s)
Re = 52390.1 Void Fraction = 0.679

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
727.	207.7	178.2	331.4	284.3	74.7
686.	185.1	160.3	316.1	273.8	74.5
617.	153.4	135.2	297.0	261.6	75.5
558.	127.3	113.5	278.1	248.1	75.7
504.	114.4	104.0	282.9	257.1	82.1
457.	104.4	96.4	292.8	270.5	90.1
414.	86.5	80.3	275.2	255.6	89.3
376.	79.0	74.3	286.0	268.7	97.6
343.	69.0	65.2	284.1	268.7	101.6

RUN 2 Water V = 1.359(m/s) Steam V = 4.083(m/s)
Re = 58630.9 Void Fraction = 0.659

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
729.	213.2	183.5	339.1	291.9	76.3
688.	196.8	171.8	335.0	292.4	78.8
620.	152.1	133.6	292.7	257.0	74.2
562.	134.5	120.5	291.5	261.1	79.0
535.	121.3	109.1	279.0	250.9	78.0
487.	111.1	101.6	287.3	262.9	85.2
423.	92.9	86.4	287.9	267.8	92.4
385.	82.5	77.4	289.2	271.5	97.6
368.	73.9	69.4	275.8	259.0	95.2

***** TEST: TU1-3 *****

RUN 3 Water V = 1.519(m/s) Steam V = 4.243(m/s)
Re = 65567.5 Void Fraction = 0.635

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
717.	238.2	209.9	386.0	340.0	87.9
678.	203.9	179.9	352.9	311.3	83.8

615.	156.6	138.5	304.4	269.2	77.6	68.6	0.441
560.	138.2	124.3	300.6	270.4	81.6	73.4	0.443
535.	127.7	115.5	293.8	265.6	82.2	74.3	0.435
489.	114.9	105.4	295.6	271.0	87.4	80.2	0.442
448.	105.5	98.0	303.4	281.8	94.4	87.7	0.457
410.	90.5	84.6	292.2	273.0	95.4	89.2	0.439
376.	85.9	81.1	310.8	293.5	106.1	100.2	0.467
345.	74.8	70.9	305.1	289.6	108.7	103.2	0.456

RUN 4 Water V = 1.708(m/s) Steam V = 4.486(m/s)
 Re = 73732.5 Void Fraction = 0.600

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
691.	230.6	205.2	390.2	347.2	91.4 81.4 0.534
652.	189.6	168.2	343.7	304.9	84.0 74.6 0.470
587.	153.3	137.5	315.0	282.4	82.9 74.3 0.436
531.	136.9	124.9	317.6	289.7	89.2 81.4 0.447
481.	117.0	107.9	306.9	282.9	91.6 84.4 0.435
437.	105.0	97.9	311.3	290.3	98.1 91.5 0.443
398.	90.2	84.7	302.9	284.4	100.5 94.4 0.430
362.	82.6	78.3	315.2	298.8	109.7 104.0 0.446
345.	82.5	78.7	336.7	321.2	120.0 114.5 0.477

RUN 5 Water V = 1.845(m/s) Steam V = 4.715(m/s)
 Re = 79620.2 Void Fraction = 0.571

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
721.	270.7	242.0	436.3	390.0	99.0 88.5 0.575
681.	232.3	208.0	400.0	358.2	94.8 84.9 0.530
617.	172.4	154.1	333.3	297.9	84.7 75.7 0.442
566.	150.1	135.8	322.2	291.5	86.9 78.6 0.433
519.	132.3	121.1	315.9	289.1	90.1 82.4 0.429
477.	118.1	109.2	313.3	289.6	94.0 86.9 0.428
439.	108.6	101.5	320.4	299.3	100.7 94.1 0.439
404.	95.6	89.8	314.3	295.4	103.4 97.2 0.430
372.	84.0	79.3	308.4	291.3	105.8 100.0 0.420
357.	86.4	82.3	335.8	319.6	117.6 112.0 0.459

RUN 6 Water V = 2.037(m/s) Steam V = 5.157(m/s)
 Re = 87915.9 Void Fraction = 0.522

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
733.	276.6	246.4	437.1	389.4	97.9 87.2 0.546
692.	255.7	230.3	432.4	389.4	101.3 91.2 0.548
627.	190.4	171.3	361.6	325.3	90.9 81.8 0.460
574.	157.0	142.0	331.0	299.5	88.4 80.0 0.424
528.	139.9	128.1	326.8	299.2	92.1 84.4 0.423
487.	123.4	113.9	318.9	294.4	94.5 87.3 0.415
450.	112.1	104.5	320.9	299.1	99.6 92.8 0.419
415.	102.6	96.4	325.9	306.4	105.8 99.4 0.426
383.	94.0	89.0	332.6	315.0	112.6 106.6 0.434
367.	91.7	87.3	343.1	326.3	118.5 112.7 0.447

RUN 7 Water V =2.172(m/s) Steam V = 5.602(m/s)
 Re = 93714.1 Void Fraction =0.481
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C
 700. 286.5 260.2 477.7 433.8 110.9 100.7 0.591
 661. 229.3 207.0 408.6 368.7 98.8 89.2 0.504
 602. 170.6 153.5 340.1 306.2 88.0 79.3 0.420
 552. 149.3 135.9 330.1 300.6 90.4 82.3 0.412
 508. 132.2 121.6 324.1 298.1 93.7 86.2 0.408
 467. 121.7 113.2 331.4 308.3 100.6 93.6 0.419
 431. 112.9 106.1 340.9 320.3 108.3 101.7 0.433
 398. 101.8 96.3 342.3 323.8 113.6 107.5 0.434
 381. 98.6 93.6 350.4 332.8 118.8 112.9 0.444

RUN 8 Water V =2.380(m/s) Steam V = 6.732(m/s)
 Re = 102711.6 Void Fraction =0.400
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C
 670. 259.3 236.1 454.8 414.1 109.0 99.2 0.540
 639. 215.8 195.5 400.1 362.6 99.2 89.9 0.474
 613. 188.6 170.6 367.5 332.5 93.9 84.9 0.435
 567. 159.7 145.3 341.9 311.0 92.0 83.7 0.407
 546. 152.6 139.6 342.0 313.0 94.4 86.3 0.409
 506. 143.5 133.0 353.3 327.5 102.4 94.9 0.428
 488. 132.4 122.9 341.6 317.1 101.2 93.9 0.413
 453. 123.2 115.4 349.4 327.4 108.0 101.2 0.424
 436. 116.9 109.9 348.3 327.4 110.0 103.4 0.423
 420. 114.5 108.1 358.2 338.4 115.5 109.1 0.436

RUN 9 Water V =2.479(m/s) Steam V = 7.653(m/s)
 Re = 107005.4 Void Fraction =0.352
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C
 636. 206.6 186.7 385.7 348.5 96.0 86.7 0.446
 612. 188.2 170.3 367.7 332.8 94.0 85.1 0.427
 589. 179.7 163.6 367.2 334.4 96.4 87.8 0.429
 568. 167.8 153.4 358.7 327.8 96.5 88.2 0.420
 548. 158.9 145.9 355.1 325.9 97.8 89.8 0.418
 529. 145.2 133.4 338.8 311.2 95.5 87.7 0.399
 511. 135.8 125.0 330.6 304.3 95.2 87.7 0.390
 494. 134.4 124.6 341.5 316.6 100.5 93.1 0.405
 477. 129.3 120.4 343.0 319.3 102.9 95.8 0.407
 461. 124.3 116.1 344.4 321.8 105.4 98.5 0.409
 445. 120.4 113.0 348.7 327.2 108.8 102.1 0.415
 430. 119.5 112.7 362.3 341.8 115.3 108.8 0.432

***** TEST: TU1-4 *****

RUN 1 Water V =1.103(m/s) Steam V = 5.040(m/s)
 Re = 47610.6 Void Fraction =0.647
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C
 686. 199.2 174.4 340.0 297.5 80.1 70.1 0.569
 647. 176.9 156.0 323.5 285.2 79.5 70.1 0.547
 612. 152.4 134.5 297.6 262.7 76.1 67.2 0.505

581.	143.6	128.2	298.9	266.9	79.2	70.7	0.513
522.	122.9	111.4	291.1	264.0	82.7	75.0	0.507
470.	106.9	98.3	288.6	265.4	87.3	80.3	0.507
447.	97.8	90.3	281.9	260.3	87.8	81.0	0.495
404.	88.2	82.5	290.4	271.5	95.6	89.4	0.512
366.	76.9	72.5	289.7	273.1	100.4	94.6	0.508
332.	70.5	67.1	304.4	289.6	110.6	105.2	0.532
316.	66.7	63.7	309.1	295.1	114.9	109.7	0.538

RUN 2 Water V =1.264(m/s) Steam V = 5.141(m/s)
 Re = 54567.0 Void Fraction =0.634

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
701.	225.1	198.6	374.8	330.7	86.9	76.7	0.591
660.	187.7	165.5	335.4	295.8	81.3	71.7	0.529
593.	149.2	132.9	302.8	269.6	79.1	70.5	0.484
564.	136.9	122.7	295.3	264.7	79.8	71.6	0.475
510.	119.8	109.1	292.4	266.2	84.3	76.8	0.477
462.	105.7	97.5	291.9	269.3	89.2	82.3	0.479
419.	91.9	85.5	287.9	268.1	92.9	86.5	0.473
400.	87.3	81.7	291.3	272.7	96.4	90.3	0.479
363.	77.0	72.7	292.5	275.9	101.6	95.9	0.479
316.	63.1	60.1	292.2	278.1	108.5	103.3	0.473

RUN 3 Water V =1.453(m/s) Steam V = 5.276(m/s)
 Re = 62703.6 Void Fraction =0.618

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
725.	220.9	191.7	353.8	307.0	80.0	69.4	0.510
680.	219.1	194.9	377.9	336.1	89.6	79.7	0.561
609.	163.2	145.6	320.7	286.1	82.3	73.4	0.479
551.	135.4	122.1	300.4	271.0	82.5	74.4	0.454
525.	129.9	118.3	305.9	278.6	86.6	78.9	0.466
476.	111.7	102.8	296.8	273.1	89.1	82.0	0.455
413.	95.3	89.2	304.1	284.6	98.8	92.5	0.468
376.	84.0	79.3	304.8	287.6	104.1	98.2	0.468
343.	75.8	72.0	311.8	296.4	111.5	105.9	0.476
328.	71.3	68.0	313.5	298.9	114.5	109.2	0.477

RUN 4 Water V =1.689(m/s) Steam V = 5.474(m/s)
 Re = 72874.6 Void Fraction =0.595

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C		
734.	277.1	246.7	436.8	388.9	97.7	87.0	0.599
691.	235.1	209.8	398.1	355.2	93.3	83.2	0.549
623.	172.7	153.9	330.1	294.1	83.3	74.2	0.456
568.	152.8	138.3	326.3	295.4	87.7	79.4	0.459
519.	133.0	121.7	317.3	290.5	90.5	82.8	0.451
475.	117.6	108.8	314.0	290.5	94.5	87.4	0.449
454.	110.9	103.1	313.3	291.2	96.7	89.9	0.448
416.	98.4	92.2	311.5	291.9	100.9	94.5	0.446
398.	94.5	89.0	317.0	298.4	105.1	99.0	0.454
365.	83.3	78.9	314.1	297.4	108.8	103.1	0.447
336.	77.3	73.7	328.0	313.0	118.5	113.1	0.465

RUN 5 Water V = 1.869(m/s) Steam V = 5.654(m/s)
 Re = 80642.3 Void Fraction = 0.577

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
684.	255.6	231.0	437.9	395.7	103.4	93.4 0.582
647.	206.5	185.5	377.2	338.8	92.7	83.2 0.499
589.	161.1	145.1	329.6	296.9	86.6	78.0 0.439
563.	151.6	137.5	327.3	296.8	88.5	80.3 0.438
516.	132.7	121.6	318.8	292.2	91.2	83.6 0.431
474.	120.5	111.8	322.4	298.9	97.1	90.0 0.439
435.	108.2	101.2	322.9	302.0	102.0	95.5 0.440
400.	93.8	88.2	312.5	293.8	103.3	97.2 0.425
369.	86.1	81.6	320.7	303.9	110.6	104.8 0.435
339.	82.0	78.4	343.0	327.8	123.3	117.8 0.464

RUN 6 Water V = 2.138(m/s) Steam V = 5.985(m/s)
 Re = 92263.3 Void Fraction = 0.545

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
704.	289.2	262.4	479.1	434.7	110.7	100.5 0.597
663.	232.9	210.4	414.0	374.1	100.0	90.4 0.515
599.	171.7	154.9	344.4	310.7	89.4	80.7 0.429
546.	148.7	135.8	333.7	304.7	92.1	84.1 0.420
499.	128.3	118.2	321.7	296.3	94.0	86.6 0.408
457.	119.3	111.3	334.2	311.9	102.8	95.9 0.427
419.	106.1	99.8	333.0	313.2	107.5	101.1 0.425
384.	93.7	88.7	330.3	312.6	111.6	105.6 0.421
353.	86.7	82.7	343.6	327.7	121.2	115.6 0.436

RUN 7 Water V = 2.338(m/s) Steam V = 6.297(m/s)
 Re = 100897.4 Void Fraction = 0.518

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
732.	304.8	274.7	482.7	435.1	108.3	97.6 0.569
687.	269.2	244.2	458.3	415.7	107.8	97.8 0.546
620.	191.2	172.6	367.5	331.7	93.1	84.0 0.438
567.	162.9	148.5	349.0	318.2	94.0	85.7 0.420
519.	142.3	131.0	339.3	312.5	96.7	89.1 0.412
477.	128.8	119.9	342.2	318.5	102.7	95.6 0.418
457.	121.5	113.5	340.6	318.3	104.8	97.9 0.416
420.	108.4	102.1	339.3	319.4	109.4	103.0 0.415
386.	100.4	95.3	351.2	333.4	118.4	112.4 0.429
370.	93.8	89.2	347.4	330.5	119.6	113.8 0.423

RUN 8 Water V = 2.629(m/s) Steam V = 6.903(m/s)
 Re = 113442.2 Void Fraction = 0.472

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
706.	342.1	315.0	564.6	519.9	130.2	119.9 0.644
666.	276.5	253.7	488.4	448.1	117.5	107.8 0.556
606.	198.6	181.3	392.7	358.3	101.1	92.3 0.446
557.	167.3	153.6	365.9	335.9	99.6	91.5 0.418
515.	146.7	135.7	353.4	326.9	101.3	93.7 0.406
496.	139.6	129.7	352.9	327.9	103.6	96.2 0.407

459.	127.6	119.6	355.9	333.5	109.2	102.3	0.412
425.	117.3	110.7	361.0	340.8	115.6	109.1	0.418
394.	106.5	101.1	362.7	344.5	121.0	114.9	0.419

RUN 9 Water V =2.838(m/s) Steam V = 7.515(m/s)

Re = 122485.3 Void Fraction =0.434

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
667.	279.7	256.9	493.8	453.4	118.8	109.1 0.541
636.	234.8	214.8	437.9	400.7	109.0	99.7 0.479
609.	208.5	190.9	409.5	374.8	105.0	96.1 0.449
563.	173.6	159.5	375.0	344.5	101.4	93.2 0.413
542.	164.5	151.8	372.2	343.5	103.2	95.2 0.411
522.	153.6	142.1	363.8	336.7	103.3	95.6 0.403
485.	135.7	126.3	352.1	327.8	104.6	97.4 0.391
468.	133.2	124.7	361.7	338.6	109.7	102.7 0.403
436.	122.7	115.7	365.4	344.5	115.4	108.8 0.407
405.	113.6	107.9	372.3	353.3	122.3	116.1 0.415

***** TEST: TU1-5 *****

RUN 1 Water V =1.182(m/s) Steam V = 6.813(m/s)

Re = 51012.9 Void Fraction =0.670

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
885.	302.8	249.4	385.9	317.8	74.3	61.2 0.574
828.	267.6	224.1	367.9	308.0	74.9	62.7 0.560
734.	209.3	179.0	330.2	282.4	73.9	63.2 0.520
654.	177.4	155.8	320.5	281.5	78.2	68.7 0.521
584.	149.9	134.3	309.8	277.4	81.8	73.2 0.515
523.	131.1	119.6	309.9	282.8	87.9	80.2 0.524
495.	120.4	110.5	304.5	279.5	89.4	82.0 0.518
469.	114.2	105.7	309.5	286.4	93.8	86.8 0.528
422.	101.1	94.7	314.3	294.4	101.1	94.7 0.538
378.	92.5	87.7	333.0	315.6	113.4	107.5 0.570

RUN 2 Water V =1.357(m/s) Steam V = 6.931(m/s)

Re = 58555.1 Void Fraction =0.659

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
915.	317.7	258.5	390.0	317.3	73.0	59.4 0.532
801.	247.9	208.4	353.7	297.4	73.9	62.2 0.507
753.	221.2	188.5	338.7	288.5	74.3	63.3 0.494
669.	186.4	163.3	327.4	286.7	78.5	68.8 0.495
596.	161.3	144.7	325.0	291.6	84.6	75.9 0.506
563.	148.0	133.9	319.6	289.0	86.4	78.2 0.501
503.	125.5	115.1	311.0	285.4	90.4	83.0 0.494
451.	108.6	100.9	309.1	287.2	95.7	88.9 0.493
428.	103.9	97.2	317.2	296.9	101.2	94.7 0.507
383.	95.0	90.0	335.1	317.5	113.3	107.4 0.536
362.	88.1	83.8	335.9	319.4	116.9	111.1 0.535

RUN 3 Water V =1.562(m/s) Steam V = 7.087(m/s)
Re = 67406.0 Void Fraction =0.644

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
911.	356.3	297.7	439.3	367.1	82.5	68.9 0.574
850.	290.4	243.2	387.2	324.2	77.1	64.5 0.512
799.	249.5	210.3	356.9	300.9	74.7	63.0 0.478
711.	204.5	176.8	334.6	289.4	76.7	66.4 0.464
672.	188.7	165.3	329.9	288.9	78.9	69.1 0.465
602.	164.2	147.2	327.4	293.5	84.8	76.0 0.474
570.	149.8	135.2	319.0	288.0	85.6	77.3 0.465
512.	129.3	118.5	314.1	287.8	90.4	82.8 0.464
485.	124.0	114.6	321.8	297.4	95.6	88.4 0.478
437.	107.2	100.1	318.1	297.1	100.3	93.7 0.474
415.	102.2	96.1	325.2	305.6	105.5	99.2 0.485
371.	98.8	94.2	365.3	348.4	125.7	119.9 0.546

RUN 4 Water V =1.819(m/s) Steam V = 7.313(m/s)
Re = 78514.3 Void Fraction =0.625

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
917.	378.8	319.0	463.4	390.3	86.5	72.9 0.565
859.	317.6	268.8	418.7	354.4	82.6	70.0 0.517
808.	272.2	231.7	384.3	327.0	79.7	67.8 0.481
724.	221.9	192.8	355.9	309.3	80.5	70.0 0.459
651.	189.4	168.1	344.0	305.3	84.2	74.7 0.456
618.	175.7	157.4	339.6	304.1	86.3	77.3 0.455
557.	154.7	141.0	338.8	308.9	92.4	84.2 0.462
503.	132.8	122.4	329.3	303.6	95.8	88.3 0.454
456.	117.5	109.6	330.3	308.1	101.7	94.9 0.457
412.	111.7	105.7	358.0	338.7	116.6	110.3 0.498
391.	109.6	104.3	377.2	359.1	126.3	120.3 0.525

RUN 5 Water V =2.017(m/s) Steam V = 7.517(m/s)
Re = 87031.1 Void Fraction =0.608

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
899.	375.3	319.0	469.5	399.1	89.2	75.8 0.551
843.	321.3	275.2	432.3	370.3	86.6	74.2 0.515
753.	242.6	209.9	371.5	321.4	81.5	70.5 0.452
715.	222.1	194.1	361.4	315.8	82.6	72.1 0.446
645.	191.3	170.4	350.8	312.6	86.4	77.0 0.443
584.	168.3	152.7	348.1	315.8	91.9	83.4 0.449
555.	155.9	142.3	342.5	312.7	93.5	85.4 0.444
529.	147.0	135.2	343.0	315.4	96.7	88.9 0.448
480.	127.4	118.3	335.3	311.3	100.2	93.1 0.440
458.	121.5	113.5	339.8	317.5	104.4	97.6 0.447
415.	114.0	107.8	362.2	342.7	117.5	111.2 0.479
394.	115.8	110.5	394.6	376.4	131.7	125.6 0.523

RUN 6 Water V =2.313(m/s) Steam V = 7.886(m/s)
 Re = 99827.3 Void Fraction =0.579

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
921.	401.4	340.9	489.2	415.5	91.1	77.3 0.533
866.	361.7	311.7	472.5	407.2	92.6	79.8 0.527
818.	305.8	263.7	425.7	367.1	87.4	75.4 0.478
778.	266.9	230.9	394.0	340.8	84.3	72.9 0.446
706.	227.6	200.6	375.8	331.2	86.7	76.4 0.437
642.	198.2	177.7	365.6	327.7	90.3	81.0 0.434
613.	189.0	171.1	368.6	333.6	94.1	85.2 0.443
559.	164.3	150.5	358.0	327.9	97.3	89.1 0.435
535.	153.5	141.3	353.3	325.2	98.9	91.0 0.431
489.	138.2	128.6	355.3	330.7	105.1	97.8 0.437
468.	131.6	123.1	358.1	335.0	108.7	101.7 0.442
427.	127.9	121.3	391.7	371.4	125.2	118.7 0.486

RUN 7 Water V =2.534(m/s) Steam V = 8.229(m/s)
 Re = 109376.9 Void Fraction =0.555

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
817.	329.7	287.9	460.2	401.8	94.7	82.7 0.500
776.	282.6	246.8	418.3	365.4	89.7	78.3 0.457
739.	256.5	225.5	401.4	353.0	89.4	78.6 0.443
705.	237.7	210.7	392.7	348.2	90.6	80.3 0.439
644.	203.3	182.6	373.9	335.9	92.2	82.8 0.425
616.	192.4	174.2	372.9	337.6	94.9	85.9 0.428
564.	171.7	157.5	370.0	339.4	100.0	91.7 0.430
540.	161.3	148.8	366.7	338.1	101.9	94.0 0.428
496.	145.6	135.7	368.1	343.0	108.0	100.7 0.434
455.	134.1	126.2	377.9	355.8	116.5	109.7 0.447

RUN 8 Water V =2.857(m/s) Steam V = 8.878(m/s)
 Re = 123316.8 Void Fraction =0.514

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C
816.	350.9	309.2	490.0	431.7	100.8	88.8 0.506
773.	296.5	261.1	440.9	388.3	94.8	83.5 0.458
735.	266.4	236.0	419.9	372.0	93.9	83.2 0.440
667.	229.3	206.4	404.5	364.1	97.3	87.5 0.433
607.	199.4	181.9	393.4	358.9	101.2	92.3 0.429
579.	187.4	172.2	391.2	359.3	103.8	95.4 0.429
553.	175.6	162.2	387.7	358.1	106.1	98.0 0.427
505.	158.4	148.0	391.1	365.3	113.5	106.0 0.436
482.	153.0	143.8	400.4	376.3	119.4	112.2 0.447
459.	157.1	149.0	437.4	414.9	134.1	127.2 0.491

Appendix F

Typical Downward Two-Phase Experiment Data

Test Sphere Diameter : 12.7 mm
Test Sphere Material : 316 Stainless Steel

TD1 -- No. 1 Tube needle Assembly
TD2 -- No. 2 Tube needle Assembly
TD3 -- No. 3 Tube needle Assembly

***** TEST: TD1-2A *****

RUN 1 Water V = 2.003(m/s) Steam V = 1.129(m/s)
 Re = 86437.4 Void Fraction = 0.964

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
679.	91.8	71.7	158.6	123.9	37.7	29.4	0.405
642.	87.5	70.4	161.4	129.8	39.9	32.1	0.425
576.	75.2	62.6	157.9	131.5	42.0	35.0	0.431
547.	70.0	59.2	156.8	132.6	43.3	36.6	0.435
493.	59.8	51.7	152.4	131.7	44.9	38.8	0.431
468.	55.9	48.8	151.9	132.6	46.1	40.2	0.432
424.	48.3	42.9	149.3	132.5	47.9	42.5	0.428
404.	45.2	40.5	149.0	133.3	49.1	43.9	0.429
367.	40.0	36.3	149.9	136.0	51.8	47.0	0.432
350.	36.9	33.6	147.6	134.5	52.2	47.6	0.425
319.	32.5	29.9	148.2	136.4	54.8	50.4	0.425
292.	28.9	26.9	151.0	140.2	58.1	54.0	0.430

RUN 2 Water V = 2.128(m/s) Steam V = 1.144(m/s)
 Re = 91852.5 Void Fraction = 0.952

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
671.	102.4	83.0	179.5	145.6	43.0	34.9	0.428
638.	96.2	79.4	178.9	147.7	44.4	36.7	0.436
577.	83.8	71.2	175.7	149.3	46.8	39.7	0.441
524.	73.1	63.5	172.6	149.9	48.9	42.5	0.442
476.	64.2	56.8	170.9	151.2	51.3	45.4	0.444
434.	56.6	50.8	169.5	152.2	53.7	48.2	0.444
396.	49.4	44.9	167.1	151.7	55.6	50.5	0.439
362.	44.3	40.7	169.2	155.5	58.9	54.1	0.444
347.	41.1	37.9	166.7	153.7	59.3	54.7	0.436
318.	37.4	34.9	171.8	160.0	63.7	59.3	0.449
304.	36.1	33.8	176.4	165.2	66.7	62.4	0.460

RUN 3 Water V = 2.362(m/s) Steam V = 1.161(m/s)
 Re = 101942.0 Void Fraction = 0.938

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
720.	121.9	98.0	196.7	158.1	44.7	35.9	0.413

682.	116.3	96.0	199.9	165.0	47.3	39.0	0.432
646.	108.2	90.8	198.1	166.3	48.7	40.9	0.437
581.	92.9	80.0	193.2	166.5	51.2	44.1	0.438
524.	80.6	71.0	190.3	167.6	54.0	47.5	0.441
473.	70.2	62.9	188.1	168.5	56.7	50.8	0.441
429.	61.1	55.5	186.0	169.0	59.3	53.9	0.438
389.	53.5	49.1	185.1	170.1	62.1	57.1	0.437
354.	47.6	44.2	187.6	174.3	66.1	61.4	0.442
322.	42.6	39.9	191.8	179.9	70.7	66.3	0.450
307.	40.7	38.4	196.5	185.1	74.0	69.7	0.460

RUN 4 Water V =2.774(m/s) Steam V = 1.178(m/s)
Re = 119733.2 Void Fraction =0.924

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
668.	127.2	108.1	224.2	190.5	53.9 45.8 0.439
635.	119.7	103.2	223.7	192.8	55.7 48.0 0.445
576.	105.1	92.5	220.7	194.3	58.8 51.8 0.449
524.	91.6	82.0	216.4	193.7	61.4 54.9 0.447
477.	80.5	73.0	213.7	194.0	64.2 58.2 0.446
455.	76.2	69.6	214.6	196.2	66.2 60.5 0.449
415.	67.2	62.1	213.2	196.8	69.1 63.8 0.447
380.	59.3	55.2	211.9	197.4	72.0 67.1 0.444
348.	52.9	49.7	213.8	200.7	75.9 71.3 0.446
318.	49.3	46.7	225.7	213.9	83.6 79.2 0.469

RUN 5 Water V =3.135(m/s) Steam V = 1.186(m/s)
Re = 135302.9 Void Fraction =0.918

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
676.	136.2	116.4	236.5	202.0	56.3 48.1 0.428
644.	129.0	111.8	237.1	205.4	58.5 50.6 0.436
585.	113.7	100.6	234.3	207.3	61.8 54.6 0.441
533.	100.4	90.3	232.0	208.7	65.0 58.5 0.444
486.	87.8	80.0	227.7	207.4	67.6 61.6 0.439
444.	77.0	70.9	224.0	206.2	70.0 64.5 0.434
406.	69.3	64.5	226.6	210.8	74.4 69.2 0.440
372.	61.4	57.5	225.8	211.7	77.6 72.7 0.437
341.	55.5	52.4	230.2	217.4	82.5 77.9 0.444
327.	54.2	51.4	239.1	227.0	87.5 83.0 0.460

RUN 6 Water V =3.460(m/s) Steam V = 1.192(m/s)
Re = 149322.5 Void Fraction =0.913

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
687.	148.7	128.0	253.5	218.1	59.7 51.3 0.434
655.	141.4	123.3	254.9	222.2	62.1 54.2 0.443
596.	124.2	110.4	250.5	222.7	65.3 58.0 0.446
543.	110.0	99.4	248.4	224.4	68.8 62.2 0.449
496.	96.8	88.5	244.7	223.8	71.8 65.7 0.447
453.	85.7	79.2	242.6	224.2	74.9 69.3 0.444
415.	76.9	71.8	244.1	227.8	79.2 73.9 0.448
381.	67.7	63.6	241.1	226.5	81.8 76.9 0.441
350.	60.6	57.4	243.1	230.0	86.1 81.4 0.443

RUN 7 Water V = 4.199(m/s) Steam V = 1.200(m/s)
Re = 181222.4 Void Fraction = 0.907

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
697.	171.8	150.1	287.7	251.3	67.0	58.5 0.446
668.	169.3	150.1	298.0	264.2	71.6	63.4 0.470
613.	150.8	135.8	293.8	264.6	75.0	67.5 0.472
564.	135.3	123.5	291.8	266.3	78.9	72.0 0.475
519.	119.9	110.5	286.4	264.1	81.7	75.3 0.471
478.	108.2	100.8	286.5	266.7	85.9	80.0 0.474
441.	97.0	91.0	284.9	267.2	89.4	83.9 0.471
407.	88.4	83.6	288.3	272.4	94.5	89.3 0.477
376.	80.2	76.3	290.9	276.5	99.4	94.5 0.480
361.	76.2	72.6	291.7	278.1	101.7	96.9 0.480

RUN 8 Water V = 5.321(m/s) Steam V = 1.206(m/s)
Re = 229636.8 Void Fraction = 0.903

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
702.	205.3	183.1	340.9	304.0	78.9	70.4 0.474
676.	204.8	185.0	355.9	321.4	84.8	76.6 0.502
650.	195.5	177.8	355.8	323.6	87.2	79.3 0.506
600.	176.4	162.3	352.5	324.3	91.4	84.0 0.508
556.	160.1	148.8	351.5	326.7	95.9	89.1 0.512
514.	143.5	134.3	346.2	324.1	99.3	93.0 0.507
477.	130.9	123.4	347.4	327.6	104.3	98.3 0.511
442.	118.9	112.8	347.3	329.5	108.8	103.2 0.510
410.	108.9	103.9	350.9	334.8	114.5	109.3 0.515

RUN 9 Water V = 6.249(m/s) Steam V = 1.209(m/s)
Re = 269669.7 Void Fraction = 0.901

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
674.	228.3	208.6	397.7	363.4	94.9	86.7 0.521
652.	221.7	203.9	402.0	369.6	98.3	90.4 0.530
609.	203.2	188.6	399.6	370.8	102.6	95.2 0.533
569.	186.0	173.9	396.8	371.0	106.6	99.7 0.533
532.	169.4	159.4	392.2	369.0	110.1	103.5 0.530
498.	156.0	147.6	392.0	370.9	114.7	108.6 0.532
466.	143.5	136.5	391.7	372.5	119.1	113.2 0.532
437.	144.0	138.1	427.6	410.1	134.8	129.3 0.582

***** TEST: TD1-2C *****

RUN 1 Water V = 2.034(m/s) Steam V = 2.407(m/s)
Re = 87777.0 Void Fraction = 0.965

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
798.	130.8	98.2	187.3	140.7	39.2	29.5 0.452
748.	122.1	95.4	188.5	147.3	41.6	32.5 0.476
659.	99.1	80.7	177.4	144.5	43.1	35.1 0.471
583.	83.3	70.4	172.5	145.6	45.6	38.5 0.476
519.	69.0	59.6	164.8	142.4	47.0	40.6 0.465
490.	63.9	55.9	164.0	143.5	48.5	42.4 0.467
438.	54.9	49.0	162.6	145.1	51.2	45.7 0.468

393.	47.2	42.8	161.4	146.2	53.9	48.8	0.467
353.	41.0	37.7	162.2	148.9	57.2	52.5	0.469
318.	36.1	33.5	165.3	153.5	61.2	56.8	0.476
303.	33.2	30.9	164.0	152.8	62.2	57.9	0.470

RUN 2 Water V =2.154(m/s) Steam V = 2.438(m/s)
 Re = 92956.1 Void Fraction =0.952

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
732.	119.7	94.6	189.3	149.6	42.5	33.5	0.437
691.	114.7	93.5	193.9	158.1	45.4	37.0	0.463
618.	96.7	81.4	186.9	157.4	47.5	40.0	0.463
554.	83.2	72.0	183.5	158.8	50.2	43.4	0.468
498.	71.0	62.7	178.6	157.6	52.3	46.1	0.463
449.	61.2	54.9	175.6	157.4	54.5	48.9	0.459
406.	53.7	48.9	176.0	160.1	57.8	52.6	0.463
367.	47.0	43.3	175.9	161.9	60.8	56.0	0.462
334.	41.5	38.6	177.9	165.4	64.4	59.9	0.466
303.	37.7	35.4	185.6	174.4	70.3	66.1	0.484

RUN 3 Water V =2.387(m/s) Steam V = 2.474(m/s)
 Re = 103004.4 Void Fraction =0.939

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
716.	129.9	106.4	211.0	172.9	48.2	39.5	0.451
677.	119.4	99.4	206.8	172.3	49.2	41.0	0.450
608.	103.1	88.6	203.2	174.5	52.2	44.8	0.458
576.	95.3	82.8	200.2	173.9	53.3	46.3	0.456
519.	82.2	72.8	196.2	173.8	55.9	49.6	0.456
493.	78.2	70.0	198.9	178.1	58.5	52.4	0.466
446.	66.8	60.5	193.0	175.0	60.1	54.5	0.454
404.	59.3	54.5	194.8	179.0	64.1	58.9	0.461
367.	51.1	47.4	191.3	177.3	66.1	61.3	0.451
350.	47.4	44.1	189.2	176.0	66.9	62.3	0.445
319.	43.6	41.0	198.9	187.1	73.6	69.2	0.466

RUN 4 Water V =2.813(m/s) Steam V = 2.509(m/s)
 Re = 121387.1 Void Fraction =0.926

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
698.	135.6	113.8	226.7	190.2	52.7	44.2	0.436
663.	127.7	109.0	226.9	193.6	54.8	46.8	0.445
598.	110.0	96.1	221.0	192.9	57.4	50.1	0.445
541.	95.8	85.3	217.3	193.4	60.3	53.7	0.445
515.	88.8	79.6	214.1	192.0	61.4	55.0	0.442
467.	78.7	71.6	214.3	195.2	65.1	59.3	0.446
425.	68.9	63.5	212.3	195.5	68.0	62.6	0.444
387.	61.7	57.4	215.1	200.2	72.4	67.4	0.450
370.	57.5	53.8	213.5	199.5	73.6	68.7	0.446
337.	51.2	48.2	215.8	203.2	77.7	73.2	0.448
322.	49.5	46.8	222.5	210.6	81.9	77.5	0.461

RUN 5 Water V =3.187(m/s) Steam V = 2.527(m/s)
 Re = 137551.9 Void Fraction =0.919

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
727.	154.3	129.7	246.1	206.9	55.5	46.6	0.435
690.	145.9	124.8	247.3	211.5	58.0	49.6	0.446
655.	135.1	117.0	243.4	210.8	59.3	51.4	0.445
591.	118.5	105.0	241.2	213.8	63.2	56.0	0.453
562.	109.6	97.9	237.4	212.1	64.3	57.4	0.449
509.	95.6	86.7	233.9	212.2	67.5	61.3	0.449
485.	89.2	81.4	232.0	211.8	69.0	63.0	0.447
440.	78.0	72.0	229.4	211.7	72.0	66.5	0.443
400.	69.3	64.7	230.9	215.3	76.3	71.2	0.447
365.	59.9	56.2	225.7	211.9	78.2	73.4	0.435
334.	54.2	51.3	231.8	219.3	83.9	79.4	0.444

RUN 6 Water V =3.522(m/s) Steam V = 2.538(m/s)
Re = 151984.6 Void Fraction =0.915

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
718.	162.0	138.3	262.1	223.8	59.7	50.9	0.442
683.	150.7	130.3	258.8	223.7	61.2	52.9	0.444
649.	141.1	123.5	257.1	225.0	63.1	55.2	0.447
588.	122.9	109.6	252.0	224.8	66.3	59.1	0.448
533.	107.0	96.8	246.9	223.6	69.2	62.7	0.445
485.	93.5	85.7	242.9	222.7	72.2	66.2	0.442
442.	82.6	76.5	241.6	223.9	75.7	70.2	0.441
403.	73.7	69.0	243.3	227.6	80.2	75.0	0.444
368.	65.3	61.5	243.5	229.5	84.0	79.2	0.443
352.	62.2	58.9	246.9	233.6	87.1	82.5	0.448

RUN 7 Water V =4.276(m/s) Steam V = 2.554(m/s)
Re = 184551.6 Void Fraction =0.909

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
714.	185.2	161.9	301.5	263.5	68.9	60.2	0.465
684.	178.6	158.1	306.1	271.0	72.3	64.0	0.480
626.	160.2	144.3	304.6	274.4	76.7	69.1	0.487
574.	142.3	129.9	300.5	274.4	80.3	73.3	0.488
526.	126.4	116.7	296.5	273.6	83.8	77.3	0.486
484.	112.7	105.0	293.7	273.6	87.4	81.4	0.484
445.	101.4	95.2	293.8	275.9	91.7	86.1	0.485
410.	91.0	86.0	293.6	277.5	95.8	90.6	0.484
378.	82.9	78.9	298.4	284.0	101.7	96.8	0.491

RUN 8 Water V =5.412(m/s) Steam V = 2.568(m/s)
Re = 233578.1 Void Fraction =0.904

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
702.	216.4	194.2	359.7	322.9	83.3	74.8	0.501
676.	212.0	192.2	368.2	333.7	87.7	79.5	0.519
651.	204.7	186.9	371.6	339.3	90.9	83.0	0.528
604.	185.2	170.9	367.8	339.4	95.0	87.6	0.530
582.	176.7	163.8	366.9	340.2	97.1	90.1	0.531
540.	160.5	150.0	364.9	341.1	101.4	94.8	0.532
502.	145.7	137.2	362.8	341.6	105.7	99.5	0.532
484.	138.7	131.0	361.6	341.4	107.6	101.6	0.530

467.	132.2	125.2	360.7	341.6	109.6	103.8	0.529
434.	122.5	116.7	366.7	349.4	116.1	110.6	0.538
419.	115.2	110.0	361.8	345.3	116.8	111.5	0.530

RUN 9 Water V = 6.347(m/s) Steam V = 2.574(m/s)
Re = 273932.2 Void Fraction = 0.902

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
683.	236.4	216.0	405.7	370.6	95.9 87.6 0.529
660.	229.4	210.9	409.9	376.9	99.3 91.3 0.538
615.	210.2	195.1	408.0	378.7	103.9 96.5 0.542
574.	193.3	180.9	407.8	381.6	108.9 101.9 0.547
555.	184.8	173.5	406.4	381.6	111.0 104.2 0.546
518.	168.9	159.5	404.0	381.7	115.4 109.0 0.546
484.	155.3	147.6	404.6	384.4	120.4 114.4 0.548
452.	154.7	148.3	439.3	421.0	135.9 130.2 0.597

***** TEST: TD1-2E *****

RUN 1 Water V = 2.326(m/s) Steam V = 5.264(m/s)
Re = 100370.5 Void Fraction = 0.969

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
694.	111.7	90.3	188.2	152.2	44.0 35.6 0.479
657.	108.2	89.9	194.3	161.5	47.2 39.3 0.509
622.	100.8	85.2	193.2	163.3	48.8 41.3 0.515
559.	86.3	74.7	187.8	162.7	51.0 44.2 0.514
531.	81.1	71.1	188.2	165.0	52.9 46.4 0.521
480.	69.4	61.8	182.7	162.8	54.7 48.7 0.512
456.	63.8	57.2	179.3	160.7	55.2 49.5 0.503
414.	55.7	50.6	177.8	161.5	57.8 52.5 0.502
394.	53.1	48.6	180.6	165.3	60.2 55.1 0.511
376.	49.7	45.8	180.4	166.1	61.6 56.7 0.510
326.	40.7	37.9	179.8	167.6	65.8 61.3 0.505
298.	36.7	34.5	185.5	174.5	70.8 66.6 0.518

RUN 2 Water V = 2.415(m/s) Steam V = 5.328(m/s)
Re = 104202.7 Void Fraction = 0.958

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
674.	119.3	99.6	207.7	173.4	49.5 41.3 0.494
639.	109.6	92.8	203.4	172.1	50.4 42.7 0.492
606.	101.9	87.5	201.6	173.0	51.9 44.6 0.495
575.	94.5	82.0	199.1	172.8	53.1 46.1 0.495
545.	87.2	76.5	195.9	171.7	54.1 47.4 0.491
518.	81.1	71.7	194.0	171.7	55.4 49.0 0.491
446.	66.9	60.7	193.7	175.8	60.4 54.8 0.498
404.	58.3	53.6	192.0	176.3	63.2 58.0 0.495
367.	51.6	47.8	193.3	179.3	66.8 62.0 0.497
350.	48.3	45.0	193.1	179.9	68.3 63.7 0.496
305.	40.4	38.1	197.3	186.1	74.6 70.3 0.502

RUN 3 Water V = 2.595(m/s) Steam V = 5.407(m/s)
Re = 111973.3 Void Fraction = 0.944

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
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702.	126.7	104.5	210.3	173.4	48.7	40.1	0.443
668.	122.4	103.2	215.6	181.9	51.8	43.7	0.466
634.	114.6	98.1	214.5	183.6	53.5	45.8	0.471
603.	106.8	92.5	212.3	183.9	54.8	47.5	0.473
546.	93.2	82.4	209.1	184.9	57.7	51.1	0.475
495.	81.1	72.9	205.4	184.6	60.3	54.2	0.473
450.	71.2	64.9	203.6	185.4	63.2	57.5	0.472
410.	63.1	58.1	203.8	187.8	66.6	61.3	0.474
374.	55.6	51.7	203.3	189.1	69.7	64.8	0.472
342.	49.3	46.2	204.2	191.4	73.1	68.6	0.472
313.	44.2	41.8	208.1	196.5	77.7	73.4	0.478

RUN 4 Water V = 2.946(m/s) Steam V = 5.493(m/s)
Re = 127133.2 Void Fraction = 0.929

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
697.	138.4	116.7	232.0	195.7	54.0	45.6	0.443
663.	132.4	113.6	235.2	202.0	56.8	48.8	0.458
630.	124.5	108.3	234.8	204.2	58.8	51.1	0.464
571.	109.0	96.7	231.3	205.4	62.0	55.0	0.468
518.	94.7	85.3	226.3	203.9	64.6	58.2	0.464
472.	83.5	76.3	224.7	205.3	67.9	62.0	0.465
430.	73.7	68.1	223.5	206.4	71.1	65.7	0.463
393.	65.0	60.6	222.3	207.1	74.3	69.2	0.461
359.	58.7	55.2	226.8	213.3	79.3	74.6	0.469
328.	53.3	50.5	233.7	221.5	85.3	80.9	0.481

RUN 5 Water V = 3.278(m/s) Steam V = 5.538(m/s)
Re = 141449.1 Void Fraction = 0.921

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
694.	147.1	125.6	247.5	211.4	57.8	49.4	0.443
660.	138.4	119.9	247.2	214.2	59.9	51.9	0.449
597.	120.5	106.6	242.2	214.3	63.0	55.7	0.451
542.	105.0	94.4	237.7	213.8	65.9	59.3	0.450
492.	91.8	83.7	234.0	213.3	68.9	62.9	0.448
448.	81.8	75.5	235.0	216.9	73.1	67.4	0.452
409.	73.1	68.1	236.8	220.8	77.4	72.2	0.456
373.	63.6	59.7	232.6	218.4	79.7	74.9	0.446
341.	57.5	54.4	238.2	225.4	85.3	80.8	0.455

RUN 6 Water V = 3.590(m/s) Steam V = 5.567(m/s)
Re = 154921.6 Void Fraction = 0.917

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
700.	152.4	130.4	254.1	217.5	59.0	50.5	0.429
667.	147.2	128.1	259.7	226.1	62.5	54.4	0.447
606.	128.8	114.4	254.7	226.1	65.6	58.2	0.448
551.	112.5	101.4	249.4	224.8	68.4	61.7	0.445
526.	105.9	96.2	248.5	225.7	70.2	63.8	0.447
480.	94.0	86.4	247.2	227.3	73.9	67.9	0.448
439.	83.3	77.3	245.8	228.3	77.3	71.8	0.447
402.	74.0	69.3	245.3	229.7	81.0	75.8	0.446
338.	58.6	55.6	246.8	234.1	88.9	84.3	0.444

RUN 7 Water V =4.332(m/s) Steam V = 5.605(m/s)
Re = 186943.5 Void Fraction =0.910

	Tw(C)	qt, qc(kW/m2)	ht, hc(W/m2/C)	Nut	Nuc	C'
691.	175.3	154.2	296.6	260.7	69.5	61.1 0.460
661.	166.6	148.1	297.2	264.1	72.0	63.9 0.467
604.	147.7	133.4	293.3	264.9	75.7	68.4 0.469
553.	129.5	118.3	286.2	261.5	78.4	71.6 0.463
507.	114.9	106.2	282.7	261.1	81.9	75.6 0.462
465.	103.7	96.8	284.2	265.2	86.5	80.7 0.466
427.	92.6	87.0	282.8	265.9	90.3	84.9 0.464
393.	83.1	78.7	283.7	268.5	94.7	89.6 0.465
377.	79.8	75.8	288.1	273.7	98.3	93.3 0.472

RUN 8 Water V =5.506(m/s) Steam V = 5.631(m/s)
Re = 237601.6 Void Fraction =0.906

	Tw(C)	qt, qc(kW/m2)	ht, hc(W/m2/C)	Nut	Nuc	C'
679.	199.2	179.0	344.0	309.2	81.6	73.4 0.478
654.	191.6	173.6	346.0	313.5	84.4	76.5 0.486
606.	176.0	161.6	348.2	319.7	89.7	82.4 0.497
561.	158.4	146.8	343.6	318.4	93.2	86.3 0.495
520.	144.2	134.8	343.2	320.7	97.7	91.3 0.498
483.	130.5	122.8	340.8	320.6	101.5	95.5 0.496
449.	118.8	112.5	340.9	322.8	105.9	100.3 0.496
417.	109.0	103.8	343.9	327.5	111.3	106.0 0.500

RUN 9 Water V =6.475(m/s) Steam V = 5.643(m/s)
Re = 279443.8 Void Fraction =0.904

	Tw(C)	qt, qc(kW/m2)	ht, hc(W/m2/C)	Nut	Nuc	C'
684.	222.0	201.4	379.8	344.6	89.6	81.3 0.489
662.	218.4	199.7	388.3	355.1	93.8	85.8 0.504
641.	210.6	193.6	389.3	357.9	96.3	88.6 0.509
600.	194.7	180.6	389.5	361.3	101.0	93.7 0.515
562.	178.2	166.5	386.0	360.7	104.6	97.7 0.514
526.	164.3	154.6	385.7	362.9	109.0	102.6 0.517
493.	151.3	143.2	385.0	364.3	113.3	107.2 0.518
462.	138.3	131.4	381.8	362.9	116.6	110.9 0.513

***** TEST: TD2-1A *****

RUN 1 Water V =1.966(m/s) Steam V = 1.218(m/s)
Re = 84842.0 Void Fraction =0.947

	Tw(C)	qt, qc(kW/m2)	ht, hc(W/m2/C)	Nut	Nuc	C'
732.	111.5	86.4	176.4	136.6	39.5	30.6 0.406
691.	105.1	83.9	177.7	141.9	41.6	33.2 0.424
553.	76.0	64.8	167.7	143.0	45.9	39.2 0.429
497.	65.5	57.1	164.8	143.8	48.3	42.1 0.431
449.	56.8	50.5	163.0	144.9	50.7	45.0 0.430
406.	49.1	44.3	160.7	144.9	52.8	47.6 0.426
368.	42.1	38.4	157.3	143.3	54.3	49.5 0.417
334.	37.9	35.0	161.9	149.4	58.6	54.1 0.429
304.	33.2	30.9	162.6	151.4	61.5	57.2 0.428

RUN 2 Water V = 2.055(m/s) Steam V = 1.244(m/s)
 Re = 88685.9 Void Fraction = 0.927

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
718.	108.6	85.0	175.9	137.5	40.0	31.3	0.370
679.	107.9	87.7	186.2	151.5	44.2	35.9	0.409
609.	92.5	77.8	181.8	152.9	46.6	39.2	0.415
548.	79.4	68.5	177.3	153.0	48.8	42.1	0.415
520.	73.3	63.9	174.5	152.1	49.7	43.3	0.412
470.	62.9	55.7	170.0	150.7	51.5	45.6	0.406
447.	58.8	52.6	169.5	151.5	52.8	47.2	0.407
406.	50.4	45.5	164.9	149.0	54.1	49.0	0.397
369.	45.5	41.7	169.5	155.5	58.5	53.7	0.409
336.	40.4	37.4	171.4	158.9	61.9	57.4	0.413
320.	37.6	35.0	170.7	158.8	63.0	58.6	0.409

RUN 3 Water V = 2.226(m/s) Steam V = 1.276(m/s)
 Re = 96056.8 Void Fraction = 0.904

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
700.	120.0	98.0	200.0	163.3	46.4	37.9	0.395
629.	103.8	87.8	196.5	166.1	49.3	41.7	0.403
566.	89.8	77.9	193.0	167.4	52.1	45.1	0.407
510.	77.0	68.1	187.8	166.0	54.1	47.8	0.403
485.	72.3	64.5	187.8	167.5	55.8	49.8	0.406
440.	62.6	56.6	184.3	166.7	57.9	52.4	0.401
399.	54.9	50.2	183.5	168.0	60.8	55.6	0.400
363.	48.0	44.4	182.5	168.8	63.4	58.7	0.397
331.	42.7	39.9	184.9	172.5	67.2	62.7	0.401
316.	40.4	37.9	186.9	175.1	69.4	65.1	0.404

RUN 4 Water V = 2.386(m/s) Steam V = 1.296(m/s)
 Re = 102965.2 Void Fraction = 0.890

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
689.	128.2	107.2	217.6	181.9	51.1	42.7	0.411
655.	118.8	100.7	214.1	181.5	52.2	44.2	0.411
592.	103.1	89.6	209.6	182.1	54.8	47.6	0.413
537.	90.5	80.2	207.3	183.8	57.9	51.3	0.417
511.	83.4	74.4	202.9	181.0	58.4	52.1	0.410
465.	73.9	67.0	202.5	183.5	61.7	55.9	0.413
424.	64.6	59.2	199.5	182.8	64.0	58.6	0.408
387.	57.4	53.1	200.2	185.3	67.4	62.4	0.410
354.	50.7	47.4	200.1	186.7	70.5	65.8	0.408
324.	45.9	43.2	205.2	193.2	75.4	71.0	0.417

RUN 5 Water V = 2.537(m/s) Steam V = 1.311(m/s)
 Re = 109472.0 Void Fraction = 0.880

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
704.	136.0	113.7	225.3	188.3	52.1	43.5	0.403
669.	128.2	109.0	225.6	191.8	54.1	46.0	0.411
635.	118.3	101.7	221.0	190.0	55.0	47.3	0.408
575.	102.8	90.3	216.6	190.3	57.8	50.8	0.410

521.	90.4	81.0	214.9	192.4	61.2	54.7	0.414
473.	79.1	71.9	212.2	192.6	64.0	58.1	0.412
431.	69.2	63.6	209.4	192.3	66.6	61.1	0.408
393.	60.9	56.4	208.1	192.9	69.5	64.4	0.406
359.	53.9	50.3	208.2	194.6	72.8	68.0	0.405
343.	51.8	48.6	213.0	200.2	76.1	71.5	0.414

RUN 6 Water V =2.898(m/s) Steam V = 1.336(m/s)
Re = 125045.7 Void Fraction =0.863

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
695.	153.0	131.4	257.0	220.8	60.0 51.5 0.429
664.	146.9	128.1	260.6	227.2	62.9 54.8 0.442
634.	138.2	121.7	258.9	228.1	64.6 56.9 0.444
578.	121.7	109.0	254.5	228.0	67.6 60.6 0.445
529.	107.9	98.1	251.8	228.8	71.0 64.5 0.446
484.	95.3	87.6	248.4	228.2	73.9 67.9 0.443
444.	84.6	78.5	246.2	228.4	77.0 71.4 0.440
408.	75.7	70.8	246.0	230.1	80.6 75.4 0.440
375.	68.2	64.2	248.1	233.8	84.9 80.0 0.443
360.	64.9	61.4	250.3	236.7	87.4 82.7 0.446

RUN 7 Water V =3.477(m/s) Steam V = 1.359(m/s)
Re = 150067.0 Void Fraction =0.849

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
702.	179.9	157.8	299.1	262.4	69.3 60.8 0.453
670.	173.7	154.4	304.9	271.0	73.1 64.9 0.468
611.	153.3	138.5	300.1	271.1	76.9 69.4 0.470
584.	143.4	130.4	296.6	269.7	78.4 71.3 0.468
533.	127.8	117.7	295.2	271.9	82.7 76.2 0.471
510.	119.0	110.1	290.5	268.8	83.8 77.5 0.466
467.	107.2	100.2	292.4	273.3	88.8 83.0 0.471
447.	100.8	94.5	290.7	272.7	90.5 84.9 0.468
410.	90.2	85.2	291.2	275.2	95.1 89.9 0.469
376.	81.2	77.2	293.7	279.4	100.3 95.4 0.471

RUN 8 Water V =3.975(m/s) Steam V = 1.371(m/s)
Re = 171548.3 Void Fraction =0.841

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
676.	195.1	175.3	338.8	304.3	80.7 72.5 0.486
647.	186.2	168.7	340.2	308.2	83.6 75.7 0.493
594.	166.4	152.7	337.0	309.4	88.0 80.8 0.496
545.	149.3	138.6	335.5	311.3	92.7 86.0 0.499
523.	139.9	130.3	331.0	308.4	94.0 87.5 0.494
481.	125.7	118.1	330.0	310.0	98.5 92.6 0.495
443.	112.4	106.3	327.6	309.9	102.5 97.0 0.491
408.	102.7	97.7	333.0	317.0	109.0 103.7 0.499
392.	97.5	93.1	333.9	318.8	111.6 106.6 0.499

RUN 9 Water V =4.691(m/s) Steam V = 1.383(m/s)
Re = 202455.2 Void Fraction =0.834

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
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700.	228.1	206.1	380.1	343.4	88.2	79.7	0.499
673.	222.6	203.0	388.3	354.1	92.7	84.6	0.515
622.	203.1	187.6	389.3	359.4	98.4	90.9	0.525
598.	191.5	177.5	384.6	356.6	100.0	92.7	0.521
553.	173.2	162.0	382.4	357.7	104.6	97.9	0.522
512.	156.1	147.1	378.8	356.8	108.9	102.6	0.520
475.	142.1	134.8	379.5	359.9	114.2	108.3	0.523
440.	128.9	122.9	379.1	361.4	119.0	113.5	0.522
424.	123.0	117.6	379.7	363.0	121.8	116.4	0.522

RUN 10 Water V = 5.314(m/s) Steam V = 1.389(m/s)
Re = 229326.1 Void Fraction = 0.830

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
710.	247.0	224.1	404.9	367.4	93.0 84.3 0.498
683.	242.7	222.2	416.1	381.0	98.3 90.0 0.518
632.	223.9	207.6	421.0	390.3	105.2 97.6 0.532
585.	200.8	187.7	414.4	387.5	109.4 102.2 0.529
541.	182.6	172.1	414.0	390.1	114.9 108.3 0.532
502.	163.3	154.8	406.9	385.6	118.6 112.4 0.525
465.	150.0	143.1	411.0	391.9	125.1 119.3 0.531
448.	140.2	133.9	402.9	384.8	125.3 119.7 0.520

***** TEST: TD2-1E *****

RUN 1 Water V = 2.276(m/s) Steam V = 5.662(m/s)
Re = 98237.7 Void Fraction = 0.954

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
664.	120.4	101.6	213.5	180.1	51.5 43.4 0.519
629.	111.9	95.8	211.5	181.0	53.0 45.4 0.523
597.	102.8	88.9	207.0	179.1	53.9 46.6 0.518
566.	96.2	84.2	206.3	180.7	55.6 48.7 0.523
537.	89.4	79.1	204.5	180.9	57.0 50.4 0.523
485.	77.2	69.4	200.6	180.3	59.6 53.6 0.519
461.	71.8	65.0	198.8	180.0	60.8 55.1 0.516
418.	63.5	58.3	200.1	183.7	64.7 59.4 0.522
398.	59.6	55.0	200.3	184.9	66.5 61.4 0.523
345.	48.6	45.5	198.8	185.8	70.9 66.3 0.516
300.	41.8	39.6	209.2	198.1	79.6 75.4 0.539

RUN 1 Water V = 2.276(m/s) Steam V = 5.662(m/s)
Re = 98237.7 Void Fraction = 0.954

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
713.	132.0	108.8	215.4	177.5	49.3 40.6 0.510
675.	127.3	107.5	221.3	186.9	52.7 44.5 0.539
640.	116.8	99.8	216.4	185.0	53.6 45.8 0.534
607.	108.0	93.5	213.3	184.6	54.9 47.5 0.534
576.	102.1	89.6	214.8	188.5	57.3 50.3 0.545
519.	88.3	78.9	210.8	188.4	60.1 53.7 0.544
469.	76.5	69.4	207.4	188.1	62.8 57.0 0.540
425.	67.4	61.9	207.7	190.9	66.5 61.1 0.544
385.	58.9	54.7	206.5	191.7	69.7 64.7 0.541

350.	51.0	47.7	203.6	190.4	72.0	67.4	0.530
319.	45.7	43.1	208.4	196.5	77.1	72.7	0.540
305.	43.1	40.8	210.2	198.9	79.4	75.1	0.542

RUN 2 Water V =2.340(m/s) Steam V = 5.770(m/s)
Re = 100975.6 Void Fraction =0.936

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
682.	124.7	104.4	214.5	179.5	50.8 42.5 0.469
645.	116.1	98.8	213.0	181.1	52.4 44.6 0.475
611.	107.9	93.0	211.0	181.9	54.0 46.6 0.478
550.	93.0	82.0	206.8	182.3	56.8 50.1 0.479
496.	80.0	71.7	202.3	181.4	59.4 53.2 0.475
448.	69.9	63.6	201.1	183.0	62.6 56.9 0.476
406.	61.4	56.6	201.1	185.3	66.0 60.9 0.477
368.	53.6	49.8	199.8	185.8	69.0 64.1 0.473
335.	47.4	44.5	201.9	189.4	73.0 68.5 0.476
305.	43.5	41.2	212.4	201.1	80.2 76.0 0.498

RUN 2 Water V =2.340(m/s) Steam V = 5.770(m/s)
Re = 100975.6 Void Fraction =0.936

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
723.	142.4	118.2	228.7	190.0	51.8 43.0 0.495
681.	133.8	113.4	230.1	195.1	54.5 46.2 0.510
643.	123.5	106.3	227.5	195.9	56.2 48.4 0.513
607.	113.6	99.1	224.1	195.4	57.6 50.3 0.513
542.	97.3	86.7	219.9	196.0	61.0 54.3 0.514
486.	82.6	74.8	214.0	193.7	63.5 57.5 0.507
437.	71.2	65.4	211.5	194.0	66.7 61.2 0.503
394.	62.7	58.3	213.7	198.5	71.3 66.2 0.510
355.	54.4	51.0	213.1	199.7	74.9 70.2 0.506
322.	47.8	45.2	215.9	203.9	79.6 75.2 0.509
306.	45.2	42.8	219.3	208.0	82.7 78.4 0.515

RUN 3 Water V =2.469(m/s) Steam V = 5.913(m/s)
Re = 106546.9 Void Fraction =0.914

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
676.	131.4	111.5	227.9	193.4	54.2 46.0 0.457
642.	122.3	105.2	225.7	194.1	55.8 48.0 0.459
579.	106.5	93.7	222.3	195.7	59.0 52.0 0.464
523.	93.7	84.1	221.3	198.7	62.8 56.3 0.470
474.	82.3	75.0	220.2	200.7	66.4 60.5 0.473
430.	71.8	66.2	218.0	200.9	69.4 63.9 0.469
390.	63.6	59.2	219.1	204.1	73.4 68.4 0.472
372.	59.3	55.4	217.8	203.6	74.7 69.9 0.468
339.	52.3	49.3	218.8	206.1	78.7 74.1 0.468
309.	47.2	44.8	225.5	214.1	84.6 80.3 0.479

RUN 4 Water V =2.597(m/s) Steam V = 6.011(m/s)
Re = 112096.2 Void Fraction =0.899

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
683.	139.7	119.3	239.8	204.7	56.7 48.4 0.453

649.	131.2	113.6	239.0	206.9	58.6	50.7	0.458
617.	121.4	106.1	234.8	205.3	59.7	52.2	0.456
587.	114.1	100.8	234.3	207.1	61.6	54.5	0.460
532.	100.5	90.5	232.8	209.6	65.4	58.8	0.465
482.	88.2	80.5	230.7	210.6	68.8	62.8	0.465
438.	78.0	72.0	230.4	212.8	72.5	67.0	0.467
399.	69.3	64.7	231.9	216.4	76.8	71.7	0.471
364.	61.1	57.4	231.6	217.8	80.4	75.6	0.468
332.	53.4	50.5	229.8	217.4	83.4	78.9	0.461
318.	50.6	48.0	232.2	220.4	86.1	81.7	0.465

RUN 5 Water V = 2.724(m/s) Steam V = 6.084(m/s)
 Re = 117557.3 Void Fraction = 0.888

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
705.	148.5	126.1	245.7	208.6	56.7 48.2 0.439
669.	140.6	121.4	247.4	213.6	59.4 51.3 0.450
602.	122.3	108.1	243.5	215.2	63.0 55.6 0.455
544.	105.7	95.0	238.2	214.2	65.9 59.3 0.453
492.	92.3	84.2	235.4	214.7	69.4 63.3 0.453
446.	80.7	74.4	233.0	215.0	72.6 67.0 0.450
405.	71.3	66.5	233.5	217.7	76.7 71.5 0.452
369.	62.6	58.8	232.7	218.6	80.2 75.4 0.448
336.	56.0	53.1	237.2	224.6	85.6 81.0 0.455
321.	53.4	50.7	241.2	229.3	89.0 84.6 0.461

RUN 6 Water V = 3.325(m/s) Steam V = 6.283(m/s)
 Re = 143476.7 Void Fraction = 0.860

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
721.	181.0	157.0	291.7	253.1	66.2 57.5 0.454
687.	178.8	158.0	304.9	269.5	71.8 63.4 0.485
622.	158.8	143.2	304.3	274.4	76.9 69.4 0.496
565.	138.9	127.0	298.9	273.4	80.7 73.8 0.495
514.	121.1	112.0	293.0	270.9	84.1 77.8 0.490
468.	106.6	99.6	290.0	270.8	88.0 82.2 0.487
427.	95.7	90.1	292.7	275.7	93.5 88.1 0.492
390.	84.3	79.9	290.6	275.6	97.4 92.4 0.487
357.	75.7	72.2	294.5	281.0	103.2 98.5 0.491

RUN 7 Water V = 3.590(m/s) Steam V = 6.332(m/s)
 Re = 154925.8 Void Fraction = 0.853

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
714.	192.6	169.3	313.8	275.8	71.7 63.0 0.471
682.	188.4	168.1	323.9	288.9	76.6 68.4 0.495
621.	167.5	152.0	321.5	291.7	81.4 73.8 0.502
566.	148.0	136.0	317.4	291.7	85.5 78.6 0.502
518.	129.9	120.6	311.1	288.9	88.9 82.5 0.497
474.	116.0	108.7	310.5	291.0	93.6 87.7 0.498
434.	103.0	97.3	308.6	291.3	97.7 92.2 0.495
398.	92.6	88.0	310.9	295.4	103.1 98.0 0.498
366.	83.8	80.1	315.5	301.6	109.3 104.5 0.503

RUN 8 Water V =4.080(m/s) Steam V = 6.393(m/s)
 Re = 176094.1 Void Fraction =0.845
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 696. 210.4 188.8 353.2 316.9 82.3 73.9 0.502
 665. 201.0 182.1 355.8 322.3 85.7 77.7 0.512
 636. 190.4 173.8 355.5 324.5 88.5 80.8 0.516
 608. 179.6 165.0 353.8 325.1 90.9 83.6 0.518
 556. 158.4 147.0 347.5 322.6 94.8 88.0 0.514
 509. 141.3 132.4 345.4 323.6 99.7 93.4 0.515
 467. 124.7 117.6 339.7 320.5 103.2 97.3 0.507
 429. 113.0 107.3 343.2 326.2 109.3 103.9 0.512
 395. 101.1 96.6 343.3 328.0 114.4 109.3 0.511

RUN 9 Water V =4.807(m/s) Steam V = 6.445(m/s)
 Re = 207463.6 Void Fraction =0.838
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 702. 227.2 205.0 377.5 340.6 87.4 78.9 0.492
 675. 224.8 205.1 391.2 356.9 93.3 85.1 0.516
 623. 205.8 190.2 393.7 363.7 99.4 91.9 0.528
 599. 195.7 181.8 392.7 364.7 102.0 94.7 0.530
 553. 175.9 164.8 388.7 364.0 106.4 99.7 0.528
 511. 158.9 149.9 386.6 364.8 111.3 105.0 0.529
 492. 151.4 143.3 386.8 366.2 114.1 108.0 0.530
 455. 136.9 130.3 385.7 367.2 118.9 113.2 0.528
 422. 123.1 117.7 382.8 366.2 123.1 117.8 0.523

RUN 10 Water V =5.452(m/s) Steam V = 6.472(m/s)
 Re = 235303.7 Void Fraction =0.835
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 691. 253.3 232.2 428.7 392.9 100.4 92.1 0.530
 666. 244.0 225.0 430.9 397.3 103.7 95.6 0.537
 619. 223.2 207.8 430.0 400.3 109.0 101.5 0.542
 576. 203.5 191.0 427.8 401.5 114.0 107.0 0.544
 536. 186.3 176.1 427.8 404.4 119.6 113.0 0.548
 498. 169.2 160.8 424.7 403.6 124.2 118.1 0.546
 464. 153.4 146.5 421.3 402.3 128.4 122.6 0.541
 448. 145.8 139.5 418.8 400.7 130.2 124.6 0.538

***** TEST: TD3-3A *****

RUN 1 Water V =1.948(m/s) Steam V = 1.439(m/s)
 Re = 84055.1 Void Fraction =0.923
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 640. 95.7 78.8 177.3 145.9 43.9 36.1 0.400
 608. 89.7 75.1 176.6 147.8 45.4 38.0 0.406
 550. 78.3 67.3 174.2 149.7 47.9 41.1 0.411
 498. 68.3 60.0 171.8 150.7 50.3 44.1 0.413
 452. 59.5 53.1 169.4 151.1 52.4 46.8 0.411
 411. 52.3 47.3 168.3 152.2 54.9 49.7 0.411
 375. 46.4 42.4 168.9 154.6 57.8 52.9 0.413
 342. 40.7 37.6 168.3 155.4 60.2 55.6 0.410
 313. 36.0 33.5 169.1 157.5 63.1 58.8 0.410

300. 34.7 32.5 173.7 162.6 66.1 61.9 0.420

RUN 1 Water V =1.948(m/s) Steam V = 1.439(m/s)
Re = 84055.1 Void Fraction =0.923

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
658.	93.6	75.2	167.8	134.9	40.7 32.8 0.370
625.	87.2	71.4	166.2	136.1	41.9 34.3 0.374
594.	81.3	67.7	164.7	137.0	43.0 35.8 0.377
538.	70.2	59.8	160.4	136.7	44.7 38.1 0.375
488.	62.4	54.5	160.9	140.5	47.6 41.6 0.385
444.	53.6	47.5	155.9	138.0	48.7 43.1 0.375
405.	47.6	42.8	156.2	140.4	51.3 46.2 0.378
370.	42.0	38.2	155.9	141.9	53.7 48.9 0.378
338.	37.1	34.1	155.8	143.1	56.1 51.5 0.377
310.	33.0	30.6	157.4	145.9	59.0 54.7 0.379

RUN 2 Water V =2.017(m/s) Steam V = 1.487(m/s)
Re = 87026.0 Void Fraction =0.894

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
674.	105.2	85.5	183.4	149.1	43.8 35.6 0.370
639.	100.4	83.5	186.1	154.8	46.1 38.4 0.385
576.	87.2	74.7	183.1	156.8	48.8 41.8 0.391
520.	75.6	66.1	179.7	157.2	51.2 44.8 0.391
471.	65.9	58.7	177.5	158.0	53.6 47.8 0.391
428.	59.1	53.5	180.3	163.4	57.5 52.1 0.401
408.	54.7	49.8	177.8	161.9	58.2 53.0 0.396
371.	47.1	43.2	173.6	159.4	59.7 54.8 0.385
339.	42.6	39.6	178.8	166.2	64.3 59.8 0.397
309.	38.2	35.8	182.7	171.2	68.6 64.3 0.403

RUN 3 Water V =2.151(m/s) Steam V = 1.549(m/s)
Re = 92826.0 Void Fraction =0.858

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
681.	120.5	100.2	207.5	172.7	49.2 40.9 0.385
646.	112.5	95.1	206.1	174.2	50.7 42.9 0.390
613.	104.0	89.0	202.7	173.5	51.8 44.3 0.389
582.	96.8	83.9	200.8	174.0	53.1 46.0 0.390
526.	85.1	75.4	199.9	177.0	56.5 50.0 0.396
476.	75.1	67.7	199.7	180.0	60.0 54.0 0.401
432.	64.8	59.0	195.0	177.8	61.9 56.4 0.393
393.	57.1	52.6	194.9	179.7	65.1 60.0 0.394
358.	50.0	46.6	194.3	180.8	68.1 63.3 0.391
326.	44.7	42.0	198.0	185.9	72.5 68.1 0.397
311.	42.4	40.0	200.8	189.2	75.1 70.8 0.401

RUN 4 Water V =2.417(m/s) Steam V = 1.629(m/s)
Re = 104327.7 Void Fraction =0.816

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
672.	137.7	118.2	240.8	206.6	57.6 49.4 0.408
639.	129.6	112.8	240.7	209.5	59.7 52.0 0.415

607.	121.2	106.6	239.1	210.4	61.5	54.1	0.417
577.	112.3	99.7	235.4	209.0	62.7	55.6	0.414
522.	100.0	90.5	236.9	214.3	67.3	60.8	0.424
497.	93.0	84.7	234.2	213.2	68.6	62.5	0.422
451.	80.3	73.9	228.6	210.3	70.8	65.1	0.413
410.	72.0	67.0	232.1	216.0	75.7	70.5	0.420
374.	63.4	59.5	231.7	217.5	79.4	74.5	0.419
357.	60.2	56.7	234.4	220.9	82.2	77.5	0.423
326.	54.8	52.0	242.9	230.8	89.0	84.6	0.436

RUN 5 Water V = 2.605(m/s) Steam V = 1.667(m/s)
Re = 112415.7 Void Fraction = 0.797

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
684.	154.0	133.4	263.7	228.5	62.2 53.9 0.424
653.	145.3	127.4	262.9	230.5	64.2 56.3 0.428
623.	137.0	121.3	262.0	232.0	66.2 58.6 0.432
568.	120.8	108.8	258.4	232.7	69.5 62.6 0.434
542.	114.3	103.7	258.5	234.6	71.7 65.0 0.437
495.	100.9	92.7	255.3	234.4	75.0 68.8 0.436
453.	89.6	83.1	253.8	235.4	78.4 72.7 0.435
434.	83.7	78.0	251.1	233.8	79.5 74.0 0.430
397.	75.4	70.8	253.6	238.2	84.2 79.1 0.434
365.	67.0	63.4	253.2	239.4	87.8 83.0 0.432
335.	61.3	58.3	260.8	248.2	94.3 89.7 0.443

RUN 6 Water V = 2.864(m/s) Steam V = 1.707(m/s)
Re = 123585.9 Void Fraction = 0.779

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
668.	162.0	142.8	285.1	251.4	68.5 60.4 0.436
640.	153.3	136.5	284.3	253.0	70.5 62.7 0.439
612.	144.7	129.8	282.4	253.3	72.2 64.8 0.440
562.	130.8	119.1	283.5	258.2	76.8 69.9 0.449
516.	116.2	107.0	279.8	257.7	80.1 73.8 0.447
474.	104.3	97.0	279.1	259.5	84.1 78.2 0.449
436.	93.5	87.6	278.1	260.7	87.8 82.3 0.448
402.	83.7	79.0	277.4	261.8	91.6 86.4 0.446
370.	76.0	72.2	281.1	267.0	96.7 91.9 0.450
342.	69.4	66.3	287.3	274.5	102.9 98.3 0.458

RUN 7 Water V = 3.251(m/s) Steam V = 1.748(m/s)
Re = 140287.5 Void Fraction = 0.760

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
674.	181.3	161.6	316.0	281.8	75.4 67.3 0.449
645.	172.5	155.2	316.4	284.6	77.9 70.1 0.454
592.	155.5	142.0	315.9	288.4	82.6 75.4 0.462
544.	137.5	126.8	309.6	285.5	85.6 79.0 0.456
501.	125.0	116.5	312.2	291.0	91.1 84.9 0.465
461.	111.6	104.8	309.3	290.5	94.7 88.9 0.461
425.	100.9	95.4	310.7	293.8	99.5 94.1 0.463
392.	91.5	87.1	313.8	298.7	105.0 99.9 0.467
362.	82.9	79.3	317.1	303.4	110.5 105.7 0.469

RUN 8 Water V = 3.597(m/s) Steam V = 1.775(m/s)
 Re = 155245.6 Void Fraction = 0.749

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
674.	196.9	177.2	343.0	308.7	81.8	73.7 0.462
645.	188.3	170.9	345.4	313.6	85.1	77.2 0.470
618.	176.2	160.9	340.4	310.9	86.5	79.0 0.467
591.	167.1	153.6	340.1	312.6	89.0	81.8 0.470
543.	149.9	139.3	338.7	314.7	93.9	87.2 0.473
498.	134.0	125.6	336.6	315.6	98.5	92.4 0.473
458.	120.1	113.4	335.7	317.1	103.1	97.4 0.472
421.	108.3	103.0	337.3	320.6	108.5	103.2 0.474
388.	96.9	92.6	336.6	321.7	113.1	108.1 0.472
372.	92.3	88.5	339.0	324.8	116.3	111.5 0.474

RUN 9 Water V = 4.208(m/s) Steam V = 1.806(m/s)
 Re = 181588.2 Void Fraction = 0.736

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
685.	217.0	196.3	370.8	335.5	87.4	79.1 0.458
657.	213.4	195.1	383.0	350.2	93.1	85.1 0.479
604.	192.5	178.1	382.0	353.6	98.6	91.3 0.485
579.	180.8	168.0	377.3	350.7	100.2	93.1 0.481
533.	162.6	152.5	375.5	352.2	105.3	98.7 0.483
491.	145.9	137.8	373.1	352.5	110.1	104.0 0.482
453.	131.8	125.3	373.5	355.2	115.5	109.8 0.483
418.	118.6	113.4	372.9	356.4	120.5	115.1 0.481
402.	113.5	108.8	376.2	360.6	124.2	119.0 0.485

RUN 10 Water V = 4.742(m/s) Steam V = 1.824(m/s)
 Re = 204649.2 Void Fraction = 0.728

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
681.	240.5	220.2	413.7	378.8	97.9	89.7 0.484
653.	229.1	211.1	414.2	381.7	101.1	93.2 0.489
600.	206.0	191.9	411.8	383.7	106.8	99.5 0.493
552.	187.3	176.2	414.8	390.2	113.7	107.0 0.501
508.	166.7	157.8	408.8	387.2	118.2	111.9 0.496
468.	150.3	143.2	408.9	389.7	124.1	118.3 0.497
431.	136.1	130.5	411.5	394.4	130.8	125.3 0.499
414.	128.6	123.5	410.1	393.9	133.3	128.0 0.497

RUN 11 Water V = 5.665(m/s) Steam V = 1.845(m/s)
 Re = 244490.4 Void Fraction = 0.720

	Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'
690.	278.6	257.6	472.6	437.0	110.9	102.6 0.507
667.	275.6	256.5	485.9	452.2	116.8	108.7 0.526
646.	266.9	249.6	489.3	457.5	120.4	112.6 0.532
585.	235.3	222.2	485.6	458.6	128.1	121.0 0.535
548.	216.7	205.9	484.3	460.0	133.4	126.7 0.536
513.	199.2	190.1	482.2	460.2	138.5	132.2 0.536
481.	185.1	177.5	486.3	466.3	145.3	139.3 0.541
465.	175.8	168.8	481.1	462.0	146.4	140.6 0.535

***** TEST: TD3-3E *****

RUN 1 Water V =2.289(m/s) Steam V = 6.550(m/s)
Re = 98801.1 Void Fraction =0.935

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
655.	116.4	98.4	210.0	177.4	51.2	43.2	0.467
621.	109.7	94.2	210.4	180.6	53.2	45.7	0.476
591.	101.6	88.2	207.2	179.8	54.3	47.1	0.475
534.	87.4	77.3	201.4	178.0	56.4	49.8	0.469
509.	82.2	73.4	201.3	179.6	58.2	51.9	0.473
461.	72.1	65.3	199.7	180.8	61.1	55.3	0.473
419.	62.6	57.4	196.3	179.8	63.3	58.0	0.467
400.	60.3	55.6	201.0	185.5	66.5	61.4	0.480
364.	53.0	49.4	200.9	187.1	69.7	64.9	0.478
332.	47.1	44.2	202.9	190.5	73.6	69.1	0.480
304.	41.7	39.4	205.0	193.8	77.6	73.3	0.482

RUN 2 Water V =2.338(m/s) Steam V = 6.739(m/s)
Re = 100887.8 Void Fraction =0.909

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
675.	121.6	101.9	211.6	177.3	50.4	42.3	0.424
641.	117.1	100.1	216.6	185.1	53.6	45.8	0.444
609.	109.4	94.7	215.1	186.3	55.2	47.8	0.447
550.	95.4	84.4	212.1	187.6	58.3	51.5	0.450
498.	83.6	75.2	210.0	188.9	61.4	55.3	0.453
474.	78.1	70.7	208.7	189.1	62.8	56.9	0.451
411.	63.1	58.1	203.1	187.0	66.2	61.0	0.441
374.	56.7	52.8	207.0	192.7	70.9	66.0	0.449
326.	47.4	44.7	209.8	197.7	76.8	72.4	0.452
298.	43.7	41.5	220.9	209.9	84.3	80.1	0.473

RUN 3 Water V =2.436(m/s) Steam V = 7.001(m/s)
Re = 105145.1 Void Fraction =0.874

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
662.	130.3	111.7	232.0	198.7	56.1	48.0	0.430
629.	121.5	105.4	229.7	199.2	57.6	50.0	0.432
598.	114.6	100.7	230.2	202.2	59.8	52.5	0.439
569.	105.8	93.7	225.8	199.9	60.7	53.7	0.434
515.	92.3	83.1	222.3	200.2	63.7	57.3	0.434
468.	80.9	73.8	219.9	200.7	66.7	60.9	0.433
426.	72.1	66.6	221.4	204.6	70.9	65.5	0.438
354.	55.9	52.5	219.9	206.6	77.4	72.7	0.433
339.	52.6	49.6	220.5	207.8	79.3	74.7	0.433
310.	48.1	45.7	229.2	217.7	86.0	81.6	0.447

RUN 4 Water V =2.646(m/s) Steam V = 7.360(m/s)
Re = 114189.5 Void Fraction =0.832

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
667.	153.2	134.1	270.3	236.6	65.0	56.9	0.457
636.	144.2	127.6	269.4	238.4	67.0	59.3	0.461

606.	136.0	121.6	269.0	240.4	69.3	61.9	0.466
501.	104.9	96.4	261.5	240.3	76.2	70.0	0.465
479.	99.1	91.6	261.9	242.1	78.5	72.5	0.467
417.	83.2	78.0	262.0	245.6	84.7	79.4	0.468
382.	73.4	69.2	260.4	245.8	88.3	83.3	0.464
350.	65.5	62.2	262.3	249.2	92.9	88.2	0.465
321.	58.7	56.1	266.2	254.3	98.2	93.9	0.468

RUN 5 Water V = 2.802(m/s) Steam V = 7.545(m/s)
 Re = 120943.1 Void Fraction = 0.811

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
679.	166.7	146.5	287.8	253.0	68.3	60.0	0.461
647.	158.0	140.5	289.0	257.1	71.1	63.2	0.469
586.	138.6	125.5	285.1	258.0	75.1	67.9	0.472
533.	120.7	110.6	279.1	255.8	78.3	71.7	0.468
485.	108.3	100.5	281.6	261.4	83.7	77.7	0.476
442.	94.6	88.5	276.7	259.0	86.7	81.2	0.468
403.	84.5	79.7	278.7	263.0	91.8	86.6	0.472
368.	75.9	72.2	283.0	269.0	97.7	92.8	0.477
337.	68.6	65.6	289.5	276.9	104.4	99.8	0.485

RUN 6 Water V = 3.028(m/s) Steam V = 7.744(m/s)
 Re = 130677.9 Void Fraction = 0.791

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
655.	172.3	154.2	310.7	278.2	75.7	67.8	0.476
626.	164.3	148.4	312.5	282.4	78.7	71.1	0.484
572.	146.6	134.3	310.7	284.7	83.2	76.2	0.488
524.	129.5	119.9	305.9	283.2	86.7	80.3	0.485
480.	114.8	107.2	302.2	282.2	90.4	84.4	0.482
441.	102.8	96.8	301.9	284.2	94.8	89.2	0.482
405.	92.2	87.4	302.8	287.1	99.6	94.4	0.483
372.	83.3	79.5	306.4	292.2	105.2	100.4	0.486
342.	75.7	72.6	312.6	299.8	111.8	107.2	0.493

RUN 7 Water V = 3.381(m/s) Steam V = 7.959(m/s)
 Re = 145899.0 Void Fraction = 0.769

Tw(C)	qt,qc(kW/m2)	ht,hc(W/m2/C)	Nut	Nuc	C'		
686.	194.5	173.8	332.2	296.9	78.3	70.0	0.468
655.	183.8	165.7	331.3	298.6	80.7	72.8	0.472
625.	181.1	165.2	344.7	314.6	86.8	79.2	0.498
570.	160.2	148.0	340.8	314.9	91.4	84.5	0.499
520.	140.5	131.1	334.6	312.2	95.3	88.9	0.494
497.	133.5	125.1	336.1	315.1	98.5	92.3	0.498
475.	126.4	119.0	336.9	317.3	101.3	95.4	0.500
454.	119.0	112.5	335.9	317.4	103.6	97.9	0.498
435.	112.2	106.4	335.4	318.0	106.1	100.6	0.497
416.	105.2	100.1	333.2	316.8	107.9	102.6	0.493
398.	100.2	95.6	336.1	320.7	111.5	106.4	0.497
381.	94.6	90.5	336.5	321.9	114.1	109.2	0.497
365.	89.8	86.2	339.2	325.3	117.6	112.8	0.499

RUN 8 Water V = 3.708(m/s) Steam V = 8.095(m/s)
 Re = 160028.7 Void Fraction = 0.756
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 681. 213.7 193.4 367.7 332.8 87.1 78.8 0.494
 648. 201.0 183.5 366.8 334.8 90.0 82.2 0.498
 617. 188.6 173.4 365.2 335.7 92.9 85.4 0.501
 587. 177.4 164.2 364.6 337.4 95.9 88.8 0.504
 559. 166.7 155.2 363.6 338.5 98.9 92.1 0.505
 507. 146.9 138.1 361.2 339.6 104.6 98.3 0.506
 460. 128.8 122.0 357.6 338.9 109.5 103.8 0.501
 419. 115.2 110.0 361.8 345.3 116.8 111.5 0.507
 381. 101.9 97.8 362.9 348.3 123.1 118.2 0.506

RUN 9 Water V = 4.303(m/s) Steam V = 8.255(m/s)
 Re = 185700.7 Void Fraction = 0.742
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 657. 220.5 202.3 395.9 363.0 96.2 88.2 0.494
 631. 211.0 194.7 397.2 366.6 99.4 91.7 0.500
 606. 201.1 186.6 397.4 368.8 102.3 94.9 0.503
 582. 190.7 177.8 395.6 368.8 104.7 97.6 0.504
 537. 173.1 162.7 395.9 372.3 110.4 103.8 0.508
 516. 158.9 149.6 381.6 359.4 109.2 102.8 0.490
 478. 149.2 141.7 395.2 375.5 118.5 112.6 0.510
 442. 134.4 128.4 393.7 376.0 123.4 117.9 0.507
 409. 122.5 117.6 397.3 381.3 130.0 124.7 0.511
 393. 116.0 111.5 395.8 380.6 132.2 127.1 0.508

RUN 10 Water V = 4.837(m/s) Steam V = 8.343(m/s)
 Re = 208741.3 Void Fraction = 0.734
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 674. 242.0 222.3 421.6 387.3 100.6 92.4 0.493
 646. 233.4 216.0 427.9 396.0 105.3 97.5 0.505
 592. 206.8 193.2 420.3 392.7 110.0 102.8 0.502
 567. 197.6 185.6 423.0 397.3 113.9 107.0 0.508
 521. 177.8 168.3 422.8 400.3 120.3 113.9 0.511
 499. 169.5 161.1 425.2 404.1 124.3 118.2 0.515
 458. 150.8 144.1 421.1 402.5 129.3 123.6 0.510
 421. 136.5 131.2 425.0 408.4 136.7 131.4 0.514

RUN 11 Water V = 5.778(m/s) Steam V = 8.435(m/s)
 Re = 249336.8 Void Fraction = 0.726
 Tw(C) qt,qc(kW/m2) ht,hc(W/m2/C) Nut Nuc C'
 677. 266.5 246.6 461.8 427.2 109.8 101.6 0.494
 651. 262.2 244.4 476.2 443.9 116.6 108.7 0.514
 600. 237.5 223.5 475.1 447.0 123.2 115.9 0.519
 554. 215.0 203.8 474.1 449.3 129.6 122.9 0.521
 532. 204.7 194.7 474.0 450.8 133.1 126.5 0.523
 511. 194.4 185.4 473.0 451.1 136.2 129.9 0.523
 491. 185.3 177.2 473.8 453.1 139.8 133.7 0.524
 472. 175.8 168.6 472.7 453.2 142.7 136.8 0.523
 454. 168.6 162.1 476.5 458.1 147.1 141.4 0.527