ANL/ET/CP--92074
CONF-9706130--1

A CORRELATION FOR NUCLEATE FLOW BOILING IN SMALL CHANNELS*

T. N. Tran[1,2], M. W. Wambsganss[2], M.-C. Chyu[1], and D. M. France[3]

[1]Department of Mechanical Engineering, Texas Tech University,

Lubbock, TX 79409, U.S.A.

[2]Energy Technology Division, Argonne National Laboratory, Argonne, IL 60439, U.S.A.

[3]Department of Mechanical Engineering, University of Illinois at Chicago,

Chicago, IL 60607, U.S.A.

Submitted to Compact Heat Exchangers for the Process Industries Conference, Snowbird, Utah, June 22-27, 1997.

*Work supported by the U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Office of Industrial Technologies, under Contract W-31-109-Eng-38.

A CORRELATION FOR NUCLEATE FLOW BOILING IN SMALL CHANNELS

T. N. Tran[1,2], M. W. Wambsganss[2], M.-C. Chyu[1], and D. M. France[3]

[1]Department of Mechanical Engineering, Texas Tech University, Lubbock, TX 79409, U.S.A.

[2]Energy Technology Division, Argonne National Laboratory, Argonne, IL 60439, U.S.A.

[3]Department of Mechanical Engineering, University of Illinois at Chicago, Chicago, IL 60607, U.S.A.

ABSTRACT: Compact heat exchangers are becoming more attractive for applications in which energy conservation, space saving, and cost are important considerations. Applications exist in the process industries where phase-change heat transfer realizes more compact designs and improved performance compared to single-phase heat transfer. However, there have been only a few studies in the literature reporting on phase-change heat transfer and two-phase flow in compact heat exchangers, and validated design correlations are lacking. Recent data from experiments on flow boiling of refrigerants in small channels have led researchers to conclude that nucleation is the dominant heat transfer mechanism over a broad range of heat flux and wall superheats. Local heat transfer coefficients and overall two-phase pressure drops were measured for three different refrigerants with circular and non-circular channels in a range of pressures. This data base supports the nucleate boiling mechanism, and it was used to develop a new correlation for heat transfer in nucleate flow boiling. The correlation is based on the Rohsenow [1952] boiling model, introducing a confinement number defined by Kew and Cornwell [1995]. The new correlation predicts the experimental data for nucleate flow boiling of three refrigerants within ±15%.

INTRODUCTION/BACKGROUND

Compact heat exchangers are characterized by small, typically non-circular flow passages. Such heat exchangers have numerous attractive features including high thermal effectiveness, small size, low weight, design flexibility, and low cost when mass-produced. Unlike shell-and-tube heat exchangers they can be designed to operate in a pure counterflow mode and can accommodate multiple fluid streams. Traditionally, compact heat exchangers have found wide application in the transportation industry, where small size, low weight, and low cost are important. However, they have been used only sparingly in the process industries, with one notable exception being in cryogenics. Examples of potential new process-related applications include control of temperature-sensitive processes, integral re-boilers and condensers in diadiabatic distillation [Polley, 1993], and as reactors in which a catalyst coats the inside heat transfer surface [Sobel and Spadaccini, 1995].

A significant barrier to the application of compact evaporators and condensers in the process industries is the lack of validated design correlations and an industrial standard. The design and specification of shell-and-tube heat exchangers are covered by the widely-accepted standards of the Tubular Exchanger Manufacturers Association [TEMA, 1990]. However, no similar standard exists for the design and specification of compact heat exchangers. The closest document to a standard for compact heat exchangers is a guide to plate-fin heat exchangers [HTFS, 1987]. While this guide provides valuable information on design and specification, the information on phase-change heat transfer is based primarily on large-tube data and correlations.

Another barrier to the application of compact heat exchangers in the process industries is the conservative design approach that relies on verified technology. Here, it is safe to conclude that this barrier would be reduced if the lack of validated correlations and industrial standards was changed.

The development of design correlations requires both an accurate data base and an understanding of the physical mechanisms involved. With regard to the physical mechanisms for heat transfer, it is important to know if the dominant heat transfer mechanism is forced convection or nucleation, and for what range of pertinent parameters each is dominant. Flow patterns and flow pattern maps are also important as they provide valuable insights relative to heat transfer mechanisms.

The authors have performed and reported a series of experiments involving flow boiling of two different refrigerants (R-113 and R-12) in three different flow channels, circular and rectangular in cross-section, with hydraulic diameters in the range 2.4 to 2.9 mm [Wambsganss *et al.,* 1993; Tran *et al.,* 1993, 1994, 1996]. Data trends from these studies, interpreted, in part, using flow pattern information from earlier small-channel, two-phase flow investigations [Wambsganss, 1992, 1994], have led to the following conclusions:

- Nucleate boiling dominates over a large range of heat flux ($q'' > 8$ kW/m^2) and wall superheats ($\Delta T_{sat} > 2.7$°C).

- Forced convection dominates at low values of heat flux ($q'' <$ 8 kW/m^2) and wall superheats ($\Delta T_{sat} < 2.7$°C).

- The transition between forced convection and nucleate boiling is very abrupt, occurring at a wall superheat of approximately 2.7°C.
- The transition occurs at a lower value of wall superheat than that predicted by large tube correlations.
- Geometry effects are negligible for the rectangular and circular channels tested.

These results have also been reported by other researchers, including Peng and Wang [1993], Peng *et al.* [1995], and Feldman *et al.* [1996b].

The forced-convection/nucleate-boiling transition is important from the standpoint of developing and, in design, applying heat transfer correlations. Tran *et al.* [1996] have shown that large tube correlations predict a transition at wall superheats of approximately 12°C, compared with a measured transition wall superheat of approximately 2.7°C for a 2-3 mm channel. This suggests that the transition is a function of channel size, among other parameters, occurring at a lower wall superheat as channel size decreases. The study of Peng *et al.* [1995] showed that for very small rectangular channels with hydraulic diameters in the range of 0.30 to 0.65 mm, fully developed nucleate boiling took place with no transition from forced convection.

Feldman *et al.* [1996a] have proposed using the product of the boiling number Bo and Lockhart-Martinelli parameter X to define a transition number (BoX). For transition number smaller than the critical value (0.15×10^{-3} for rectangular and corrugated channels), forced convection dominates, while above the critical transition number, nucleate boiling dominates.

Recognizing the abrupt transition as a characteristic of small-channel flow boiling, Feldman *et al.* [1996b] proposed an asymptotic model for the local, small-channel, evaporative heat transfer coefficient in the following form:

$$h = [(Fh_{conv})^n + S(h_{nb})^n]^{1/n} \tag{1}$$

where F is an enhancement factor and S is a suppression factor. Feldman *et al.* suggested calculating the nucleate boiling term with the Cooper pool boiling correlation [Cooper, 1984], and taking the greater value of the nucleate boiling and the convection terms. This effectively uses the asymptotic model with $S = 1$, and $n = \infty$.

It can be shown that the Cooper pool boiling correlation gives results that are very close to the Stephan and Abdelsalam [1980] correlation for natural convection boiling. Tran *et al.* [1996] have shown that the Stephan and Abdelsalam correlation underpredicts the nucleate boiling data at low values of wall superheat and overpredicts the data at high wall superheats. (See Fig. 1.) This suggests that nucleate boiling is enhanced in small channels at low wall superheat heats over what would be predicted by a pool boiling correlation. Consequently, a new correlation is needed that accounts for the enhancement in small channels.

The focus of this paper is on developing an improved correlation for the nucleate flow boiling term in Eq. 1. Data from the experimental studies of Wambsganss *et al.* [1993], and Tran *et al.* [1993, 1994, 1996], together with new (unpublished) data for R-134a at three different saturation pressures, are used as the basis for the development. For completeness, the experimental test set-up and typical experimental results are discussed.

EXPERIMENTS

The development of the correlation for nucleate flow boiling is based on data from experimental studies by the authors. The details of the test apparatus, instrumentation, test procedure, and data reduction method used in these studies have been reported [Wambsganss *et al.,* 1993; Tran *et al.,* 1993, 1994, 1996]. They are briefly described below for completeness. The data base used in the development of the correlation is also summarized.

Test Apparatus and Instrumentation

A schematic of the test apparatus is given in Fig. 2. The test apparatus consisted of a closed-loop system with the system pressure controlled by high-pressure nitrogen via a pressure regulator and a bladder-type expansion tank. Liquid refrigerant was pumped through a filter and a constant displacement flow meter. After passing through the flow meter and a sight glass, the refrigerant entered the test channel in a subcooled condition. The refrigerant was heated to saturation and evaporated to a quality of approximately 0.8 or less by passing a DC electricity through the test channel wall. Heat input to the refrigerant was determined from the electric power input to the channel, accounting for heat loss to the environment.

The test channels tested were approximately 0.9 m in length of either stainless steel or brass. The channels were instrumented to measure inlet pressure, overall pressure drop across the channel length, bulk refrigerant temperatures at three axial locations (inlet, outlet, and one intermediate location), and wall surface temperatures at up to 16 axial locations.

Test Procedure and Data Reduction

Single-phase testing was first performed to validate the overall system performance, and to determine the heat loss to the environment. In performing flow boiling experiments, the establishment of steady-state conditions was verified by monitoring analog records of in-stream and wall temperatures. After steady-state was achieved, all sensor-output voltages were read by the data acquisition system 30 times each and averaged. As an additional check of steady state, the data were averaged in three groups of 10 data scans each and consistency was checked before all 30 scans were averaged together.

To provide the type of data that would facilitate assessment of the heat transfer mechanisms, the flow boiling tests were performed as follows:

- constant exit quality: heat flux and mass flux adjusted to maintain exit quality of approximately 0.8

- constant heat flux: mass flux variable, exit quality varying in the range of 0.3 to 0.8

- constant mass flux: heat flux variable, exit quality varying in the range of 0.3 to 0.8

The data reduction procedure involved determining the saturated outlet pressure p_{out} from the measured outlet temperature T_{out} using the saturation temperature-pressure relationship. The single-phase pressure drop in the subcooled region was calculated to be small (the subcooled length is short and the Reynolds number low). The pressure at the start of boiling p_{SB} is then calculated as the sum of the outlet pressure p_{out} and the measured two-phase pressure drop Δp. The refrigerant temperature at the start of boiling T_{SB} can then be calculated, again, using the saturation temperature-pressure relationship. An energy balance over the subcooled length gives

$$L_{SB} = \frac{\dot{m} c_p (T_{SB} - T_{in})}{P q''} \tag{2}$$

The two-phase pressure drop was small (typically less than 30 kPa) and therefore assumed to be linear over the test channel length. The pressure gradient is calculated as

$$m = \frac{dp}{dz} = \frac{\Delta p}{L - L_{SB}} \tag{3}$$

The temperature of refrigerant in the subcooled region ($z \le L_{SB}$) is calculated as

$$T_f(z) = T_{in} + \frac{z(T_{SB} - T_{in})}{L_{SB}} \tag{4}$$

and the refrigerant temperature in the boiling region ($z \geq L_{SB}$) is determined using the saturation temperature-pressure relationship with

$$p(z) = -mz + [p_{out} + m(L - L_{SB})] \tag{5}$$

The local evaporative heat transfer coefficient at the wall temperature measurement location z, for $z \geq L_{SB}$, is calculated as

$$h(z) = \frac{q''}{T_w(z) - T_{sat}(z)} \tag{6}$$

where $T_w(z)$ is the wall temperature on the inside surface of the channel, calculated from the measured external wall surface temperature $T_w'(z)$ and the conduction temperature drop across the channel wall.

The quality x at measurement location z is calculated as

$$x(z) = \frac{q'' S(z - L_{SB})}{\dot{m} i_{fg}} \tag{7}$$

The exit quality x_e is obtained by evaluating Eq. 7 with $z = L$.

A typical result showing measured and calculated temperature distributions is given in Fig. 3.

Uncertainty Analysis

The estimated uncertainties of key parameters are as following: temperatures, ±0.2°C, pressure drop ±0.7 kPa, channel dimension,

±0.075 mm, heat loss factor, ±2%, heat input, ±1%. The accuracy of the data base was assessed by performing an uncertainty analysis using the method of sequential perturbations [Moffat, 1988]. Uncertainty in the local heat transfer coefficient was found to be very sensitive to the wall superheat as shown in Table 1, for a representative test series.

TABLE 1

Uncertainty in Heat Transfer Coefficient

ΔT_{sat}(C)	Uncertainty
>3	5–10%
2–3	10–15%
1.5–2	12–20%
1–1.5	18–30%
<1	>30%

From Table 1, it is clear that the uncertainty of the heat transfer coefficient is large for low values of wall superheat. The data base that will be used in correlation development is for wall superheats greater than approximately 2.75°C, which is the range in which nucleate boiling is dominant. Therefore, the expected uncertainty in the two-phase heat transfer coefficient will be in the range of 5 to 10 percent.

Experimental Results

Experimental results were obtained from flow boiling tests with three different refrigerants (R-113, R-12, and R-134a) at up to three saturation pressures, in circular, rectangular, and square channels. As reported previously [Tran *et al.,* 1994, 1995], in the nucleate boiling dominant region the local heat transfer coefficients are effectively independent of quality for precritical-heat-flux qualities in the range of 0.2 to 0.8. Therefore, an average heat transfer coefficient over the quality range tested was calculated for each test run.

The data base used in developing the correlation for nucleate flow boiling heat transfer coefficient consists of a total 431 test runs as summarized in Table 2.

TABLE 2

Summary of Nucleate Flow Boiling Test Data Base

($\Delta T_{sat} > 2.75°C$ and $x > 0.2$)

	Test Series			
	(1)*	(2)*	(3)*	(4)*
Refrigerant	R-113	R-12	R-12	R-134a
Channel Material	Stainless steel	Brass	Brass	Brass
Channel Geometry	Circular	Circular	Rectangular	Circular
d_h (mm)	2.92	2.46	2.40 (1.70×4.06)	2.46
P_R	0.045	0.12, 0.20	0.20	0.10, 0.15, 0.20
G (kg/m^{2}s)	50-400	63-832	44-505	92-476
q'' (kW/m^2)	8.8-90.8	7.5-59.5	7.7-129	7.9-49.8
Bo	0.00075-0.0023	0.00020-0.0017	0.00028-0.0016	0.00039-0.00081
ΔT_{sat} (°C)	7.2-18.2	2.8-6.6	2.8-8.2	2.8-7.1
No. of Tests	27	104	118	182

*Test Series (1): Wambsganss et al. [1993]
Test Series (2): Tran et al. [1993]
Test Series (3): Tran et al. [1996]
Test Series (4): New (unpublished) data

DESIGN CORRELATION

Since nucleate-boiling-dominant heat transfer is a function of heat flux, and is independent of mass flux, the following form of correlation is suggested:

$$h = c_1 q''^{c_2} \tag{8}$$

This is the form of pool boiling correlations of Stephan and Abdelsalam [1980] and Cooper [1984]. However, a heat-flux-only correlation cannot be expected to adequately represent a flow boiling situation.

Two correlations [Lazarek and Black, 1982; and Tran *et al.,* 1996] have been proposed for the nucleate flow boiling of refrigerants in small channels. Lazarek and Black based their correlation on their data of refrigerant R-113 flow boiling in a small diameter (3.17 mm) tube. Their correlation included the product of the boiling number and Reynolds number, and is given as

$$Nu = 30\, Re_\ell^{0.857}\, Bo^{0.714} \tag{9}$$

It is noteworthy that the exponents of these two dimensionless parameters are such that the mass flux effect is very small, a feature that allowed the correlation to follow the trends of the small-channel nucleation-dominant heat transfer data. And, indeed, the correlation did show some success when compared to the data of Tran *et al* [1996].

The recent correlation reported by Tran *et al.* [1996] which includes boiling number and Weber number—the Weber number eliminates viscous effects in favor of surface tension—is given as

$$h = 8.4 \times 10^5 \left(Bo^2 We_{\ell o}\right)^{0.3} \left(\frac{\rho_g}{\rho_\ell}\right)^{0.4} \tag{10}$$

where the units of the heat transfer coefficient h are W/m^{2}C. In the above correlation the fluid property variations due to operating conditions are represented by the vapor-to-liquid density ratio. Tran *et al.* [1996] have shown that this correlation works well with their flow boiling data base for R-12 and R-113. However, the correlation showed less accuracy when evaluated against new data for R-134a, obtained at three different system pressures.

In an attempt to correlate flow boiling data for all three refrigerants (R-113, R-12, and R-134a), at different pressures, we used the well-known nucleate boiling model from Rohsenow [1952] was used as the starting point,

$$Nu_b = \frac{hL_b}{k_\ell} = A\,Re_b^m\,Pr_\ell^n \tag{11}$$

where the Reynolds number is based on bubble diameter, L_b is the characteristic length based on bubble diameter, and A, m, and n are constants. This approach is reasonable, considering a nucleation mechanism, which implicitly involves bubble dynamics.

Rohsenow's nucleate boiling equation can be derived using the concept of bubble departure diameter suggested by Fritz [1937]. The bubble departure diameter is given as

$$D_b = c_b \theta \left[\frac{2\sigma}{g(\rho_\ell - \rho_g)} \right]^{0.5} \tag{12}$$

where c_b is a constant and θ is a contact angle (or wetted angle) dependent on the fluid; for a refrigerant $c_b = 0.0146$ and $\theta = 35°$. The final form of Rohsenow's correlation is

$$\left\{ \frac{q"}{\mu_\ell i_{fg}} \left[\frac{\sigma}{g(\rho_\ell - \rho_g)} \right]^{0.5} \right\}^{r_1} = \left(\frac{1}{c_{sg}} \right) (Pr_\ell)^{-r_2} \left[\frac{c_{p,\ell}(T_w - T_{sat})}{i_{fg}} \right] \tag{13}$$

where r_1 and r_2 are constants. The left hand side of Eq. 12 defines parameter groupings which are a function of heat flux and fluid properties—the important components of nucleate boiling.

For small-channel, nucleate-flow-boiling, one would expect the confinement provided by a small channel to affect the boiling process, including bubble growth and coalescence. This has been shown to be the case [Kasza and Wambsganss, 1995; Kasza et al. 1997]. To account for this confinement effect, Kew and Cornwell (1995) suggested a new form for the evaporative heat transfer in small channel:

$$Nu = C\, Re_b^{a_1}\, Re_\ell^{a_2}\, N_{conf}^{a_3}\, Pr_\ell^{a_4} \tag{14}$$

where N_{conf} is a new dimensionless group, termed the "confinement number," defined as

$$N_{conf} = \frac{\left[\dfrac{\sigma}{g\left(\rho_\ell - \rho_g\right)}\right]^{0.5}}{D} \tag{15}$$

and C, a_1, a_2, a_3, and a_4 are constants. The term in the numerator of Eq. 15 is found in the definition of the bubble departure diameter (see Eq. 12); D in the denominator is the channel hydraulic diameter. The confinement number therefore relates the bubble diameter to the size of the channel.

Using these concepts, the heat transfer coefficient for nucleate flow boiling in small channels is expressed in terms of heat flux, fluid properties, and the confinement number. Taking the product of the boiling number, Reynolds number, and confinement number, gives the left-hand-side of Rohsenow's correlation (Eq. 13):

$$Bo\,Re\,N_{conf} = \frac{q''}{i_{fg}\mu}\left[\frac{\sigma}{g\left(\rho_\ell - \rho_g\right)}\right]^{0.5} \tag{16}$$

This suggests expressing the heat transfer coefficient in terms of this parameter grouping. The form of the proposed new correlation is

$$Nu = \frac{hD}{k_\ell} = c_1\left(Bo\,Re_\ell\,N_{conf}\right)^{c_2}\left(\frac{\rho_g}{\rho_\ell}\right)^{c_3} \tag{17}$$

Surface tension is expected to be important in small-channel flow boiling, where the contribution from bubble growth and shape to the heat transfer process is significant. Following Kew and Cornwell [1995], and introducing the confinement number into the heat transfer coefficient includes this fluid property.

Using the experimental data from the tests summarized in Table 2, to evaluate the coefficients c_1, c_2, and c_3, in Eq. 16, obtains

$$Nu = 770\left(Bo\, Re_\ell\, N_{conf}\right)^{0.62}\left(\frac{\rho_g}{\rho_\ell}\right)^{0.297} \tag{18}$$

In Figs. 4-7, the predicted evaporative heat transfer coefficients for nucleate flow boiling, using Eq. 18, are compared with experimental data for the following test series: R-113 in a 2.92 mm, circular, stainless-steel tube (Fig. 4); R-12 in a 2.46 mm, circular, brass tube (Fig. 5); R-12 in a 1.70 × 4.06 mm (2.40 mm hydraulic diameter), rectangular, brass channel (Fig. 6); and R-134a in a 2.46 mm, circular, brass tube (Fig. 7). In Fig. 8, predicted coefficients are compared with experimental data (431 data points) from all four test series. The comparison given in Figs. 4-8 shows that the present correlation (Eq. 18) predicts the vast majority of the data within ±15%.

DISCUSSION

Previously reported data obtained with refrigerants R-113 and R-12 were used together with new (unpublished) data on R-134a to provide the basis for the development of a nucleate flow boiling heat transfer correlation over the quality range $0.2 < x < 0.8$.

The correlation is an improvement for the nucleate boiling term in the asymptotic model (Eq. 1) representing the heat transfer coefficient for flow boiling in small channels. The correlation was developed using experimental data from tests with three different refrigerants boiling at different pressures in circular and rectangular channels, with channel sizes in the range 2.4 to 2.9 mm. The new correlation predicted the heat transfer within ±15%.

The developed heat transfer correlation is an improvement over the use of either the Cooper or Stephan and Abdelsalam pool boiling correlations to represent nucleate flow boiling in small channels, as both pool boiling correlations underpredict the data over a broad range of wall superheats. The proposed new correlation also includes surface tension, an important fluid property in small-channel flow boiling which is not found in the correlation of Lazarek and Black [1982] (Eq. 9), but which was found in the earlier correlation of Tran *et al.* [1995] to help predict the R-113 and R-12 data well (Eq. 10). The new correlation is expected to be more representative, than either of these two earlier correlations, of flow boiling in small channels, as attested to by the good agreement with the experimental data of three refrigerants over a range of pressures, as shown in Figs. 4-8.

In common with many correlations for phase-change heat transfer, the correlation was developed from a limited data set. The particular data set in this case included only refrigerants. There is a need to evaluate the correlation against other fluids with vastly different properties, including surface tension, which is expected to be important in small channel flow boiling. Also, the data base included a very small range of hydraulic diameters (2.4 to 2.9 mm) and only two channel geometries (circular and rectangular). Channel size is important as it affects the nucleate flow boiling heat transfer coefficient, and also as it affects the forced-convection/nucleate-boiling transition. Channel cross-sectional geometry may also be important as it affects the flow patterns and liquid distribution via capillary action. Because of the importance of the nucleation mechanism, the effect of channel surface conditions, as it determines the size, number, and distribution of nucleation sites, is another factor which could be considered and studied.

To apply the correlation, the designer must be able to predict the conditions under which nucleation is dominant. Alternatively, he must have confidence in his prediction of forced convection heat transfer, such that he can follow the suggestion of Feldman *et al.* [1996b] in applying Eq. 1, by taking the larger of the forced convection and nucleate flow boiling terms.

Feldman *et al.* [1996a] proposed an approach to predict the forced-convection/nucleate-flow-boiling transition. However, additional experimental data are needed to evaluate and improve this predictive method. In particular, test data are needed at different hydraulic diameters to provide the basis for characterizing the effect of channel size on the transition.

This paper and much of the recently published data on small-channel flow boiling focus on the nucleate boiling region. There is a need to study the forced convection region which dominates at low mass flux, high qualities, and low wall superheats. There are only limited data in this region – more data are needed. The generation of such data requires careful experimentation because of the small temperature differences that are involved.

ACKNOWLEDGMENTS

This work was supported by the U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Office of Industrial Technologies, under Contract W-31-109-Eng-38.

The authors thank Roger Smith for his contributions in fabricating the test apparatus, instrumenting the test channel, and performing some of the tests, and Ms. Joyce Stephens for her contributions in processing much of the data, and preparing the figures and overall manuscript for publication.

NOMENCLATURE

c_p	specific heat
g	gravitational acceleration
h	two-phase heat transfer coefficient
h_{conv}	forced-convective heat transfer coefficient
h_{nb}	nucleate boiling heat transfer coefficient
i_{fg}	latent heat of evaporation
k_{ℓ}	thermal conductivity of liquid
m	pressure gradient (Eq. 3)
$\dot{m}$	refrigerant mass flow rate
n	coefficient (Eq. 1)

p	refrigerant pressure
p_{CR}	critical pressure
p_{in}	inlet pressure
p_{out}	outlet pressure
q''	heat flux
x	quality
x_e	exit quality (outlet of test channel)
z	axial distance along tube measured from start of heating
Bo	boiling number ($= q''/i_{fg}G$)
D	hydraulic diameter
D_b	bubble departure diameter
F	enhancement factor
G	mass flux
L	test channel length (heated)
L_b	characteristic length based on bubble diameter
L_{SB}	subcooled length (length from test channel inlet to start of boiling)
N_{conf}	confinement number
Nu	Nusselt number ($= hD/k_\ell$)
Nu_b	Nusselt number ($= hL_b/k_\ell$)
P	heat transfer perimeter
Pr	Prandtl number
P_R	reduced pressure ($= p/p_{CR}$)
Re	Reynolds number based on tube diameter
Re_b	Reynolds number based on bubble diameter
S	suppression factor
T_{in}	refrigerant inlet temperature
T_f	refrigerant temperature
T_{out}	refrigerant outlet temperature
T_{sat}	refrigerant saturation temperature

T_{SB}	refrigerant temperature at start of boiling ($z = L_{SB}$)
T_w	inside wall temperature
T'_w	measured outside wall temperature
We	Weber number ($= G^2D/\rho_\ell\sigma$)
X	Martinelli parameter
Δp	pressure drop
ΔT_{sat}	wall superheat ($= T_w - T_f$)
m	viscosity
ρ_g	vapor density
ρ_ℓ	liquid density
σ	surface tension

Subscripts

g	gas (vapor)
ℓ	liquid

REFERENCES

Cooper, M. G., Saturation nucleation pool boiling – a simple correlation, *IChemE Symposium Series*, No. 86, 785-793, 1984.

Feldman, A., Margat, L., Marvillet, C., and Thonon, B., Transition from nucleate boiling to convective boiling in compact heat exchangers, *Proc. Eurotherm Conf.* No. 48 Pool Boiling, Paderborn, Germany, 1996a.

Feldman, A., Marvillet, C., and Lebouche, M., An experimental study of boiling in plate fin heat exchangers, in *2nd European Thermal-Sciences and 14th UIT National Heat Transfer Conference*, edited by G. P. Celata, P. Di Marco, and A. Mariani, pp. 445-450, Edizioni ETS, 1996b.

Fritz, W., Berechnung des maximalvolumens von dampfblasen, *Physikalische Zeitschrift,* 36, 379, 1937.

Heat Transfer and Fluid Flow Services, *Plate-fin Heat Exchangers: Guide to their Specification and Use*, edited by M. A. Taylor, 1st ED., Harwell Laboratory, Oxon, U.K., 1987.

Kasza, K. E., and Wambsganss, M. W., Flow visualization of microscale thermal mechanics of boiling in small channels, in *Flow Visualization VII - Proceedings of 7th International Symposium on Flow Visualization,* edited by J. P. Crowder, pp. 262-267, Begell House, Inc., New York, 1995.

Kasza, K. E., Didascalou, T., and Wambsganss, M. W., Microscale flow visualization of nucleate boiling in small channels: mechanisms influencing heat transfer, in *Proc. Conference on Compact Heat Exchangers for the Process Industries,* edited by R. K. Shah, Begell House, Inc., New York (to be published, 1997).

Kew, P. A., and Cornwell, K., Confined bubble flow and boiling in narrow channels, *10th Int. Heat Transfer Conf.,* Brighton, U.K., 1995.

Lazarek, G. M., and Black, S. H., Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113, *Int. J. Heat Mass Transfer,* 25(7), 945-960, 1982.

Moffat, R. J., Describing the uncertainties in experimental results, *Experimental Thermal and Fluid Science*, 1, 3-17, 1988.

Peng, X. F., and Wang, B.-X., Forced convection and flow boiling heat transfer for liquid flowing through microchannels, *Int. J. Heat Mass Transfer,* 36(14), 3421-3427, 1993.

Peng, X. F., Wang, B.-X., Peterson, G. P., and Ma, H. B., Experimental investigation of heat transfer in flat plates with rectangular microchannels, *Int. J. Heat Mass Transfer,* 38(1), 127-137, 1995.

Polley, G. T., Heat exchangers for the future, *The Chemical Engineer,* 10, 14-16, 1993.

Rohsenow, W. M., A method of correlating heat-transfer data for surface boiling of liquids, *Trans. ASME,* 74, 969-975, 1952.

Sobel, D. R., and Spadaccini, L. J., Hydrocarbon fuel cooling technologies for advanced propulsion, ASME Paper No. 95-GT-226, presented at the *Int. Gas Turbine and Aerospace Congress & Exposition*, Houston, TX, 1995.

Stephan, K., and Abdelsalam, M., Heat transfer correlations for natural convection boiling, *Int. J. Heat Mass Transfer,* 23, 73-87, 1980.

Tran, T. N., Wambsganss, M. W., France, D. M., and Jendrzejczyk, J. A., Boiling heat transfer in a small, horizontal, rectangular channel, *Heat Transfer–Atlanta,* AIChE Symp. Series 89(295), pp. 253-261, 1993.

Tran, T. N., Wambsganss, M. W., and France, D. M., Boiling heat transfer in compact heat exchangers, presented at *AIChE Annual Meeting*, San Francisco, CA, 1994.

Tran, T. N., Wambsganss, M. W., and France, D. M., Small circular and rectangular channel boiling with two refrigerants, *Int. J. Multiphase Flow,* 22(3), 485-498, 1996.

Tubular Exchanger Manufacturers Association, *Standards of the Tubular Exchanger Manufacturers Association*, edited by V. J. Stachura, 7th ED., Tubular Exchanger Manufacturers Association, Inc., Tarrytown, NY, 1988.

Wambsganss, M. W., Jendrzejczyk, J. A., and France D. M., Two-phase flow patterns and transitions in a small, horizontal, rectangular channel, *Int. J. Multiphase Flow,* 17(3), 327-342, 1991.

Wambsganss, M. W., France, D. M., Jendrzejczyk, J. A., and Tran T. N., Boiling heat transfer in a horizontal small-diameter tube, *Trans. ASME, J. Heat Transfer,* 115, 963-972, 1993.

Wambsganss, M. W., Jendrzejczyk, J. A., and France, D. M., Two-phase flow and pressure drop in flow passages of compact heat exchangers, in *Vehicle Thermal Management*, SAE PT-46, edited by A. C. Alkidas, pp. 257-266, Society of Automotive Engineers, Inc., Warrendale, PA, 1994.

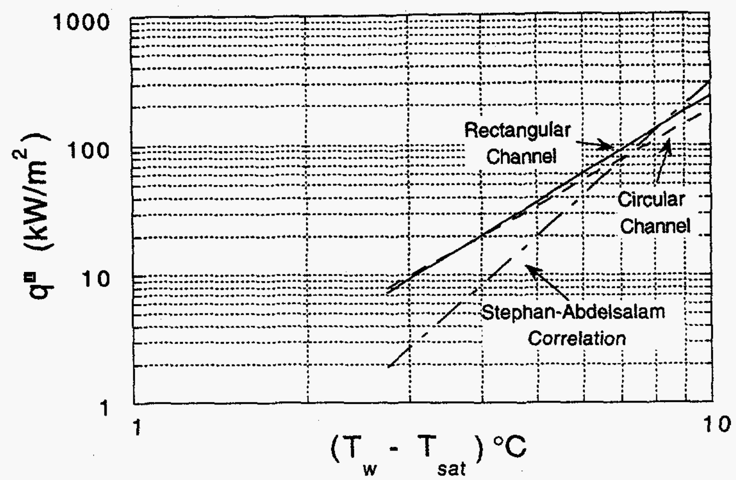

Figure 1. Heat transfer behavior of small rectangular and circular channels and pool boiling prediction of Stephan and Abdelsalam [1980] for R-12 at $\Delta T_{sat} > 2.75°C$; $p_{sat} \approx 825$ kPa

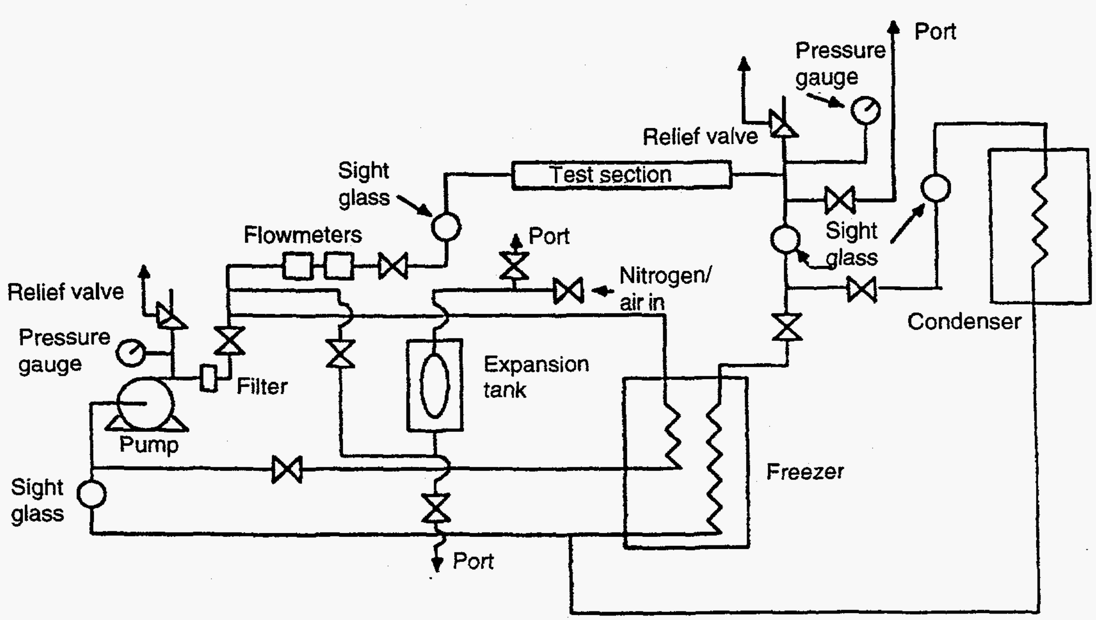

Figure 2. Schematic diagram of test apparatus

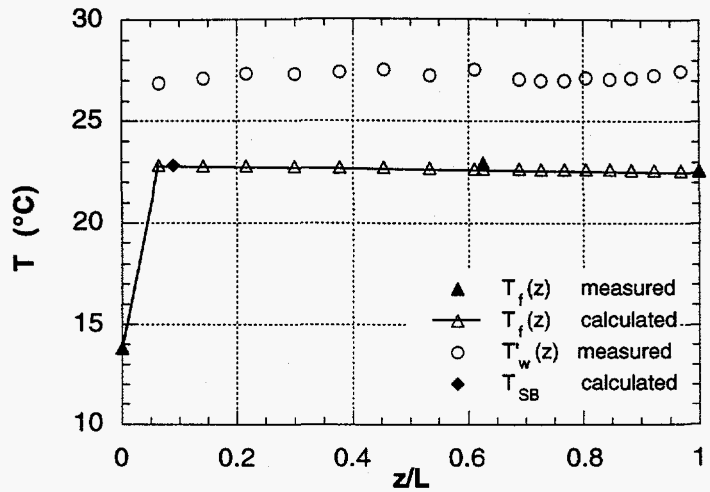

Figure 3. Measured and calculated temperature distributions for a typical test run with R-134a

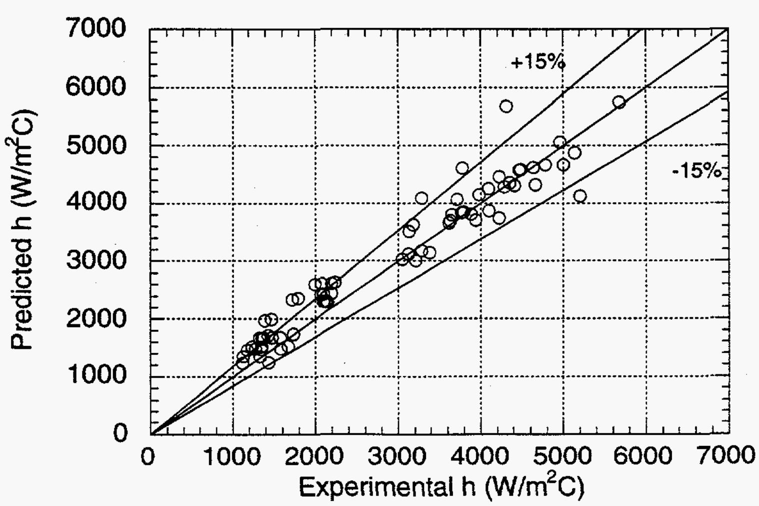

Figure 4. Comparison of predicted heat transfer coefficients (Eq. 18) with experimental data from R-113 in a 2.92 mm, circular, stainless-steel tube

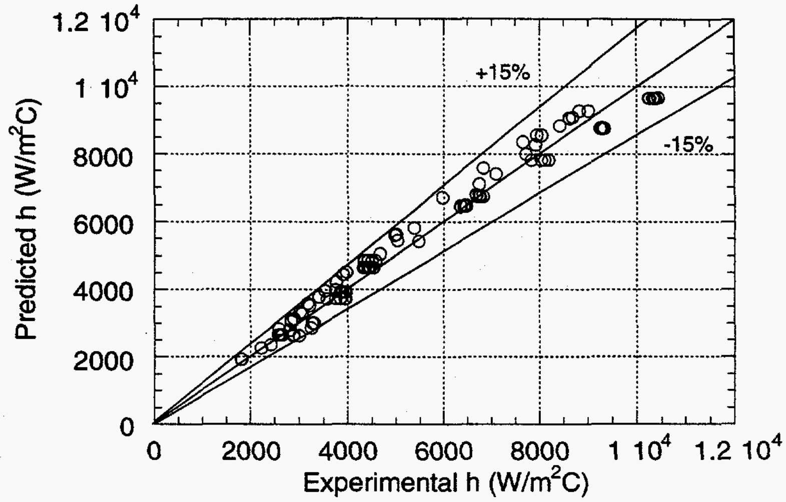

Figure 5. Comparison of predicted heat transfer coefficients (Eq. 18) with experimental data from R-12 in a 2.46 mm, circular, brass tube

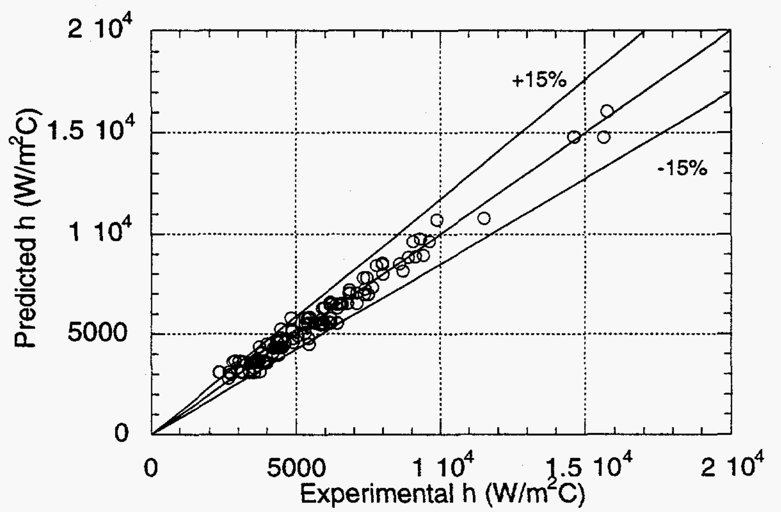

Figure 6. Comparison of predicted heat transfer coefficients (Eq. 18) with experimental data from R-12 in a 1.70×4.06 mm (2.40 mm hydraulic diameter), rectangular, brass channel

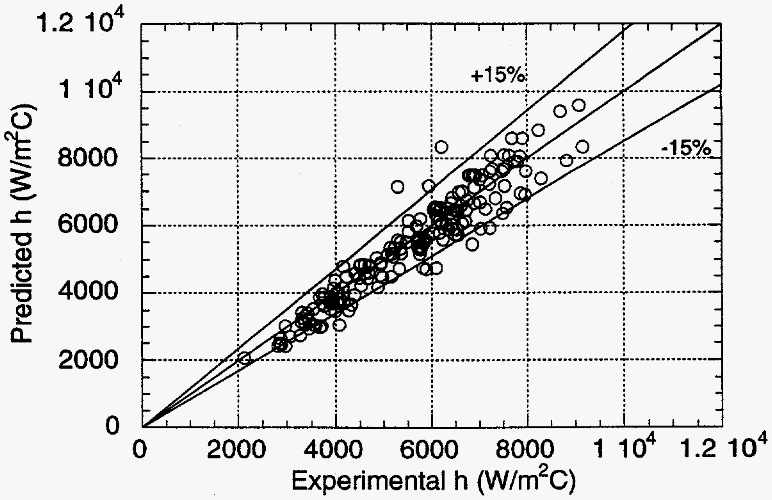

Figure 7. Comparison of predicted heat transfer coefficients (Eq. 18) with experimental data from R-134a in a 2.46 mm, circular, brass tube

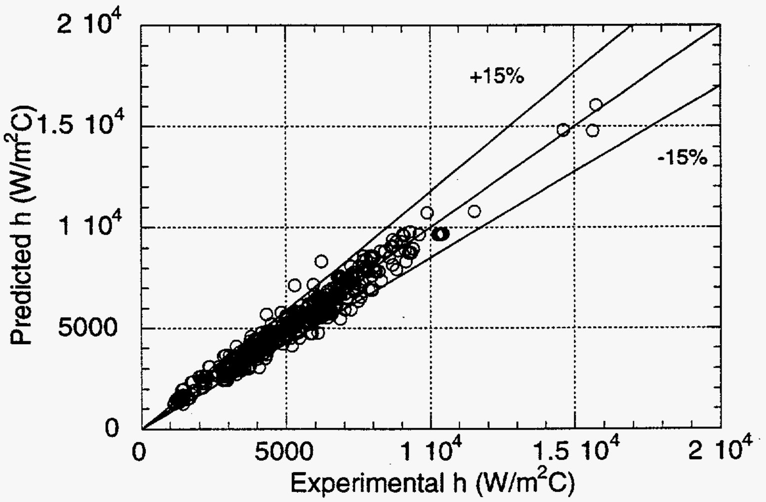

Figure 8. Comparison of predicted heat transfer coefficients (Eq. 18) with experimental data from the four test series summarized in Table 2 (431 data points)