

# On the Analysis of Low Quality Flow Boiling

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A generalized analytical treatment of dimensionless equations describing turbulent forced convection and saturated nucleate boiling heat transfer leads to a new flow boiling relationship. This relationship yields a two-phase Nusselt number in terms of dimensionless parameters of the system and includes the effects of voidage and the suppression of boiling caused by the flow. It is applicable to geometries such as flow over tube bundles and flat plates in addition to the more commonly studied case of flow in tubes.

The relationship is shown to correlate well the existing data on water over a wide range of conditions. A single experimentally determined factor is required and this factor is found for water and tentatively for Refrigerant 113 and cyclohexane.

## NOTATION

$A$	total flow cross-sectional area (in eq. (12), a constant)
$A_f$	liquid flow cross-sectional area
$B$	constant (with dimension of length in same units as $D$ )
$Bo$	boiling number, $q_b/Gh_{fg}$
$c$	specific heat of the liquid phase
$C$	constant
$C_f$	constant in liquid-only convection equation
$C_{sf}$	constant in Rohsenow (10) equation
$D$	physical dimension given by: internal diameter for flow in tubes equivalent diameter for flow in non-circular ducts external diameter for flow over tubes and tube banks plate length for flow over plates
$D_b$	bubble departure diameter
$F$	factor used by Chen (7)
$g$	gravitational acceleration
$G$	total mass flow rate per unit total area
$h$	two-phase heat transfer coefficient
$h_f$	heat transfer coefficient when only the liquid is present at the same mass flow rate
$h_{fg}$	specific enthalpy of vaporization (latent heat)
$k$	thermal conductivity of the liquid phase
$m, n$	power indices
$Nu$	two-phase Nusselt number, $hD/k$
$Nu_b$	boiling Nusselt number, $h_b D_b/k$
$Nu_c$	convective Nusselt number $h_c D/k$
$Nu_f$	convective, liquid-only Nusselt number, $h_f D/k$
$Pr$	liquid phase Prandtl number
$q$	heat flux
$Re_b$	boiling Reynolds number, $q_b D_b/h_{fg}\mu$
$Re_c$	convective Reynolds number, $\rho U_f D/\mu$ or $Re_f/(1-\alpha)$
$Re_f$	convective, liquid-only Reynolds number, $GD(1-x)/\mu$
$Re_v$	convective vapour Reynolds number $qD/h_{fg}\mu$
$S$	factor used by Chen (7) for boiling suppression
$u$	liquid velocity
$U$	bulk mean velocity (single phase flow)
$U_f$	bulk mean liquid velocity (two-phase flow)
$Vo$	Voidage number, $1/(1-\alpha)$

$W_f$	mass flow rate of liquid
$x$	dryness fraction or quality
$X_{tt}$	Martinelli parameter (defined following eq. (9))
$\alpha$	voidage
$\Delta T$	temperature difference between the fluid saturation and surface temperatures
$\mu$	dynamic viscosity of liquid phase
$\mu_g$	dynamic viscosity of vapour phase
$\rho$	density of liquid phase
$\rho_g$	density of vapour phase
$\sigma$	surface tension

## Suffixes

c	convective
b	boiling
f	liquid phase alone

## 1 INTRODUCTION

The analysis of saturated nucleate boiling under both pool and flow conditions is often based on the direct summation of convective and boiling components in the form

$$q = q_c + q_b$$

At low and intermediate nucleate boiling heat fluxes this is allowable because the two components depend on separate mechanisms: one involving a uniform boundary layer covering the complete surface and the other involving nucleation at isolated sites on the surface. At high heat fluxes direct summation is inappropriate as the increased site density causes the convection area to be reduced and encourages interference between the sites.

It must, however, be borne in mind that even at low heat flux the two components are to some extent mutually dependent. For example, in the case of pool boiling on a horizontal flat plate the natural convection component is a strong function of the site spacing. The convection cell initiated by Taylor instability under purely natural convective conditions is generally much larger than the typical cell size imposed by the rising streams of bubbles from the sites. (The common practice of subtracting the non-boiling natural convection heat flux from the total flux in order to obtain the boiling heat flux, as used by Lorenz, Mikic, and Rohsenow (1), Judd and Shoukri (2), Cornwell (3), and others is therefore of doubtful physical significance.)

In the case of forced convective boiling it is well established that the convective part is sensitive to the volume of vapour produced by the boiling process and the boil-

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ing part is sensitive to the flow conditions. The value of  $g_c$  is therefore not necessarily equal to the value which would be obtained under non-boiling conditions at the same temperature difference and similarly the value of  $q_b$  may not be the same as its value under non-flow conditions.

Simple correlations for flow boiling have been developed by Schrock and Grossman (4), Collier and Pulling (5), and others in the form

$$\frac{h}{h_f} = a\{Bo + b(X_{11})^n\} \quad (1)$$

The expressions are successful for particular geometries over limited ranges of flow conditions. Bergles and Rohsenow (6) developed a procedure for plotting the partial boiling curve on the  $\log q - \log \Delta T$  plane. The transition region between the forced convection and the extrapolated pool boiling curve is bridged by a squared difference interpolation formula. Chen (7) proposed a relationship which is essentially of the form

$$h = h_c(F) + h_b(S)$$

with a flow factor  $F = F(X_{11})$  and a boiling suppression factor  $S = S(Re_f, F)$ . Both  $F$  and  $S$  are found from experimentally determined charts. This relationship does include the mutual dependence of the components and successfully correlates much of the available flow boiling data for water and some organics flowing in commercially finished circular tubes. It takes no account, however, of surface effects, which are known to influence the boiling component. It is also completely unsuitable for geometries where the flow component is not described by an equation of the Dittus-Boelter type;

$$Nu = CRe_f^{0.8}Pr^{0.4}$$

It cannot therefore be applied to flow over flat plates or cross-flow over tube banks owing to the different power indices.

This paper describes an attempt to formulate the problem in a general way which will be applicable over a wide range of geometries and to incorporate implicitly the effects of flow, voidage, and suppression without the need for separate charts. The flow regimes over which the analysis is applicable range from the early stages of nucleate boiling through bubbly and slug flow to the intermediate stages of annular flow.

Some areas where the analysis will not be applicable should perhaps be mentioned at this stage. It is assumed that the surface is at all times covered by a convective boundary layer consisting entirely of the liquid phase. At high voidage annular flow therefore, where the boundary layer may be physically reduced owing to insufficient liquid, this model cannot be applied. At the present time only saturated flow has been considered, although the divergence caused by a few degrees sub-cooling as encountered for example in the lower part of a reboiler, will be small. The convective component involves determination of the voidage, generally by estimation. The uncertainties involved in the prediction of this parameter, especially in more complex geometries, will lead to corresponding uncertainties in the value of the convective term.

## 2 THE CONVECTIVE COMPONENT

The convective part of flow boiling is generally considered to be described by a turbulent flow relationship of the form

$$Nu = C_1 Re_c^m Pr^n \quad (2)$$

where  $m$  is typically in the range 0.6 to 0.8 and  $n$  in the range 0.33 to 0.4 depending on the flow geometry and heating arrangement. Complication arises, however, over the definition of Reynolds number and most studies employ a Reynolds number  $Re_f$  based on flow in a tube assuming that the liquid alone is present and occupies the total cross-sectional area of the tube:

$$Re_f = \frac{GD}{\mu} (1 - x) \quad (3)$$

Unfortunately  $Re_f$  does not necessarily reflect the effect of flow on heat transfer. (The liquid velocity usually increases as more vapour is produced and a Reynolds number suitable for heat transfer and therefore based on this liquid velocity would also increase unless compensated by a change in the physical dimension. The Reynolds number defined by eq. (3) shows the opposite tendency.)

In the case of liquid alone, flowing under conditions of forced convection, the heat transfer coefficient is primarily dependent upon the velocity of flow within the surface boundary layer. This velocity is conventionally expressed in terms of a distribution law together with a bulk mean velocity of liquid flow, defined as

$$U = \frac{\int u \, dA}{A}$$

where  $A$  is the total cross-sectional area and  $u$  is the velocity at any particular distance from the surface. In the case of two-phase flow the heat transfer coefficient is also primarily dependent upon the velocity of flow in the boundary layer which is assumed to consist entirely of liquid. (Under conditions of high voidage annular flow the boundary layer may be physically affected by the thickness of the liquid film on the wall. It is for this reason that the present analysis is restricted to moderate voidages as found in bubbly flow and the initial stages of annular flow.) The velocity of liquid flow is similarly expressed in terms of a distribution law together with a bulk mean velocity of liquid flow, defined as

$$U_f = \frac{\int u \, dA_f}{A_f}$$

Now if relationships between the Nusselt and Reynolds numbers based on the flow of liquid alone are to be applied to the two-phase flow, two conditions must be fulfilled. The same velocity distribution law must apply in each case and the Reynolds number  $Re_c$  for two-phase flow, must be based on  $U_f$ . The first condition is generally assumed without comment (although the eddy diffusivities could differ). The second condition leads to the following expression for  $Re_c$ .

The mass flow rate  $W_f$  of liquid in the two-phase flow is given by

$$W_f = \int_{A_f} \rho u \, dA_f = G(1 - x)A$$

where  $G$  is the total mass flow rate per unit total area. Rearrangement of the above formulae gives

$$U_f = \frac{G(1 - x)}{\rho} \frac{A}{A_f}$$

The convective Reynolds number for the two phase flow then becomes (noting that the voidage,  $\alpha = 1 - A_f/A$ ),

$$\begin{aligned} Re_c &= \frac{\rho U_f D}{\mu} \\ &= \frac{GD}{\mu} \left( \frac{1 - x}{1 - \alpha} \right) \\ &= Re_f \left( \frac{1}{1 - \alpha} \right) \end{aligned} \quad (4)$$

Some confirmation that the product of  $Re_f$  and  $1/(1 - \alpha)$  should replace the Reynolds number in the case of two-phase flow may be obtained by an examination of the factor  $F$  used by Chen (7). This factor is defined by

$$Nu_c = C Re_f^{0.8} Pr^{0.4} F$$

where  $Nu_c$  is the convective Nusselt number. From the above reasoning,  $Nu_c$  for the two-phase pipe flow is given by

$$Nu_c = C \left[ Re_f \left( \frac{1}{1 - \alpha} \right) \right]^{0.8} Pr^{0.4}$$

and for similar Nusselt numbers and  $C$  values this indicates that

$$F = \left( \frac{1}{1 - \alpha} \right)^{0.8}$$

The right-hand term was estimated using the Lockhart and Martinelli (8) curves and its variation as a function of  $X_{tt}$  is compared with  $F$  in Fig. 1. The equality holds

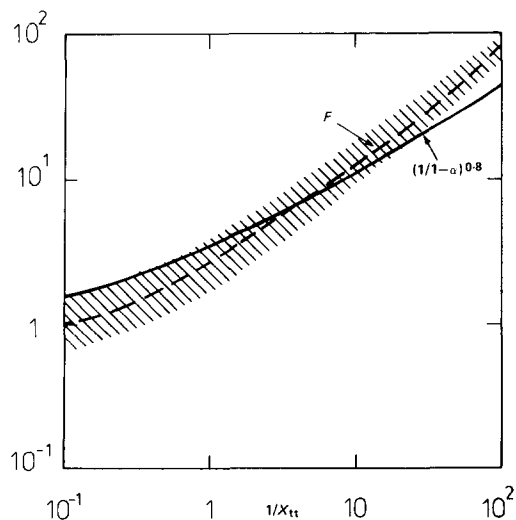


Fig. 1. Comparison with experimental factor  $F$  from Chen (7)

within the range of Chen's data up to voidages of about 0.97. At very high voidages this theory is in any case not applicable as mentioned earlier.

The value of  $C_1$  as defined in eq. (2) is not necessarily the same as the constant used in the case of single phase convection. This is because the turbulence level may be greater, as mentioned earlier, and  $C_1$  may therefore be larger than the single phase constant. Thus if the single phase Nusselt number for the flow, as if liquid only is present, is defined as

$$Nu_f = C_f Re_f^m Pr^n$$

and a Voidage number is defined as

$$Vo = \left( \frac{1}{1 - \alpha} \right)$$

then by comparison with eq. (2)

$$Nu_c = \left( \frac{C_1}{C_f} \right) Vo^m Nu_f \quad (5)$$

### 3 THE BOILING COMPONENT

The boiling component may also be expressed in the general form

$$Nu_b = C Re_b^a Pr^b \quad (6)$$

where the physical dimension is the bubble diameter  $D_b$  at departure or some group of properties with a unit of length (as discussed by Whitaker (9)).  $Re_b$  is based on a mass velocity of vapour of  $q_b/h_{fg}$  and the equation may therefore be written as

$$\frac{h_b D_b}{k} = C \left( \frac{q_b D_b}{h_{fg} \mu} \right)^a Pr^b \quad (7)$$

The commonly used Rohsenow (10) correlation is generally expressed in the form

$$\frac{c \Delta T}{h_{fg}} = C_{sf} \left[ \frac{q_b}{h_{fg} \mu} \left( \frac{\sigma}{g(\rho - \rho_g)} \right)^{1/2} \right]^{1/3} Pr^{1.7} \quad (8)$$

Substitution of  $\Delta T = q_b/h_b$  and the conventional expression for pool-boiling gravity-controlled bubble release

$$D_b = \text{constant} \times \left( \frac{\sigma}{g(\rho - \rho_g)} \right)^{1/2}$$

then yields eq. (7) with  $a = 0.667$ ,  $b = -0.7$ , and  $c = 1/C_{sf}$ . Kuloor and Radhakrishnan (11) list four other correlations of this form in which  $a$  varies from 0.625 to 0.72 and  $b$  is about  $\frac{1}{3}$ . Some Soviet workers ((12), (13) for example), reduce the expression to

$$h = C(p) q^{0.7}$$

where  $C(p)$  depends on the fluid, surface, and pressure.

The Rohsenow relationship has been criticized in the literature (14), (15) on the grounds that  $C_{sf}$ ,  $a$ , and  $b$  vary in practice. This variation is to be expected as the nucleate boiling curve on a  $\log q - \log \Delta T$  plane forms an 'S' shape and the relationship yields a straight line. However, the relationship gives a good overall mean to the complete nucleate boiling curve from incipience to burnout (see for example (16)) although the local variations in slope may be considerable. Criticisms regarding

the pressure and heat flux dependence of  $C_{sf}$  are no doubt valid, but the ranges of pressure and heat flux over which  $C_{sf}$  is reasonably constant are considerable (for water about 1 to 150 bar and 50 to 1000 kW/m<sup>2</sup>). This is generally as large as the range of application of empirical equations in other areas of convection. The general form of the Rohsenow relationship is used to describe the boiling part of flow boiling in the following section.

Under flow boiling conditions the constant  $C$  of eq. (7) must be modified as it is no longer equal to the pool boiling value ( $1/C_{sf}$ ), but is reduced by a factor,  $C_2$ , which is dependent upon the flow. The need for this factor is due to the onset of nucleate boiling occurring at higher surface superheat under flow conditions. (This effect is sometimes called boiling suppression and the reduction factor is effectively the same as the suppression factor  $S$  used by Chen (7).) Under flow boiling conditions therefore

$$C = \frac{C_2}{C_{sf}} \quad (9)$$

#### 4 FLOW BOILING ANALYSIS

The total heat transfer is taken as the sum of the two mutually dependent components as discussed earlier:

$$q = q_c + q_b$$

or

$$h = h_c + h_b \quad (10)$$

where each heat transfer coefficient is based on  $\Delta T_{sat}$ . Substitution for  $h_c$  and  $h_b$  from eqs. (5), (6), and (9) yields for flow boiling

$$h = \left(\frac{C_1}{C_f}\right) Vo^m Nu_f \frac{k}{D} + \left(\frac{C_2}{C_{sf}}\right) Re_b^a Pr^b \frac{k}{D_b} \quad (11)$$

The equation is difficult to use in this form because a value for  $D_b$  is required and determination of  $Re_b$  requires a knowledge of  $q_b$ . The following rearrangement leads to a more practical equation for engineering use although it does unfortunately make the simple concepts on which it is based less obvious. The boiling heat flux can be arranged as

$$\begin{aligned} q_b &= q - q_c \\ &= q \left(1 - \frac{q_c}{q}\right) \\ &= q \left(1 - \frac{h_c}{h}\right) \end{aligned}$$

The boiling Reynolds number may then be written as

$$Re_b = \frac{qD}{h_f \mu} \left(1 - \frac{h_c}{h}\right) \frac{D_b}{D} = Re_v \left(1 - \frac{h_c}{h}\right) \frac{D_b}{D}$$

where  $Re_v$  is a convenient dimensionless group and is termed the two-phase vapour flow Reynolds number. (It is effectively the product of the liquid flow Reynolds number and the vapour/liquid mass flow ratio.) Substi-

tution in eq. (11) then yields

$$\begin{aligned} h &= \left(\frac{C_1}{C_f}\right) Vo^m Nu_f \frac{k}{D} \\ &+ \left(\frac{C_2}{C_{sf}}\right) Re_v^a \left(1 - \frac{h_c}{h}\right)^a Pr^b \left(\frac{D_b}{D}\right)^a \frac{k}{D_b} \end{aligned}$$

or in dimensionless form

$$\begin{aligned} Nu &= AVo^m Nu_f \\ &+ Re_v^a \left(1 - \frac{AVo^m Nu_f}{Nu}\right)^a Pr^b \left(\frac{D}{B}\right)^{1-a} \quad (12) \end{aligned}$$

where

$$A = \left(\frac{C_1}{C_f}\right) \quad \text{and} \quad B = \left(\frac{C_{sf}}{C_2}\right)^{1/(1-a)} D_b$$

Equation (12) allows the possibility of direct calculations between the two-phase heat transfer coefficient,  $h$ , and the total heat flux. The solution is of an iterative form as  $Nu$  occurs on both sides of the equation.  $A$  and  $B$  are parameters which may vary with fluid, flow conditions, surface, and geometry and are found experimentally in the following section. The power index  $m$  is the same as that to which the Reynolds number is raised in the forced convection relationship for  $Nu_f$ . Thus  $m$  is 0.8 for flow through tubes and ducts and 0.63 for flow over in-line tube banks under most conditions (17). Following the earlier comments, the power index  $a$  is taken as 0.7 and, for the present,  $b$  is taken as -0.7 although this latter value is discussed later.

#### 5 COMPARISON WITH EXPERIMENTAL RESULTS

Figure 2 shows the correlation of experimental values for three fluids with eq. (12) and is arranged so that constants  $A$  and  $B$  may be found from the ordinate intersection and the slopes. The Voidage number may be determined from homogeneous flow theory or, for separated flow, from the appropriate Lockhart-Martinelli parameter,  $X_{tt}$ , by using charts or a voidage correlation such as Chisholm (18). From Butterworth (19), the Lockhart-Martinelli curve is closely approximated by the following expression when  $X_{tt}$  values are in the range 0.01 to 100:

$$\frac{1 - \alpha}{\alpha} = 0.28 X_{tt}^{0.71}$$

Algebraic rearrangement yields

$$Vo = \frac{1}{1 - \alpha} = 1 + \left(\frac{6}{X_{tt}}\right)^{0.71} \quad (13)$$

where

$$X_{tt} = \left(\frac{1 - x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho}\right)^{0.5} \left(\frac{\mu}{\mu_g}\right)^{0.1}$$

Table 1 gives relevant details of the fluids, flow geometries, and references. When a reference provides a large number of possible data points for a particular fluid and geometry, only a few representative points (generally about six), are shown in the figure to show the general area of the results from that experiment. This is con-

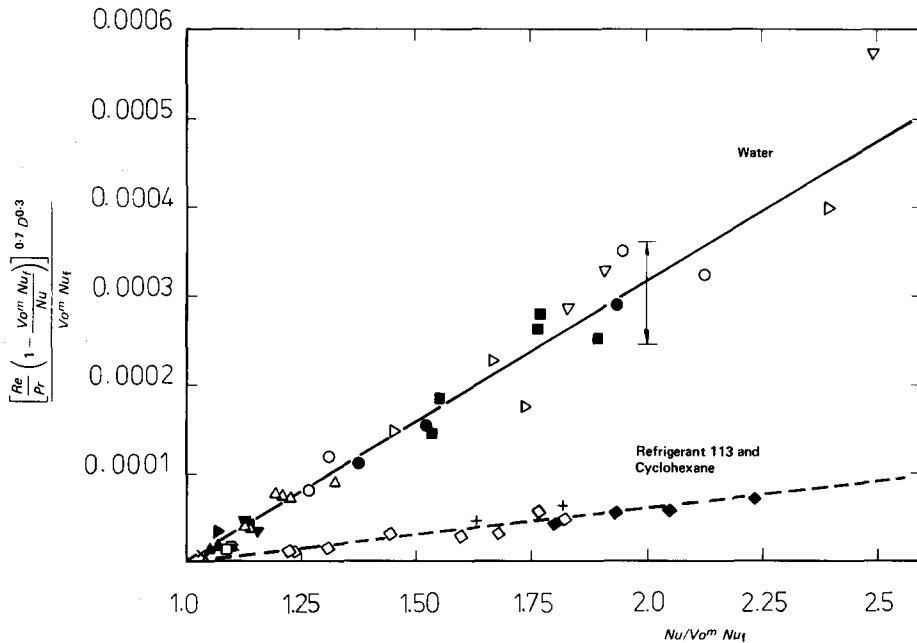


Fig. 2. Determination of the constants in eq. (12)

$m = 0.8$ , internal flow     $m = 0.63$ , tube banks     $a = 0.7$ ,  $b = -0.7$     Data point code from Table 1

sidered to provide a more even weighting to the points than if all the values from all the runs for each test are included. Reference (23) for example gives results for 67 runs, while reference (21) gives results for only 8 runs from a rig yielding data of similar accuracy. With almost pure convection, experimental scatter may yield a small negative value for the boiling term of eq. (12), in which case pure convection is assumed to exist.

The first conclusion which may be drawn from Fig. 2 is that the function passes through the origin, indicating that  $A = 1$ . This suggests that earlier concern regarding the reduction in thickness of the 'convection' layer is unfounded and that the boundary layer thickness between the areas of influence of the boiling sites remains constant over the range of data available. It should be mentioned, however, that the graphical arrangement necessitated to isolate constants  $A$  and  $B$  has led to a reduced sensitivity to  $Nu$ . Some indication of this sensitivity is given in Fig. 2 by showing the variation in the ordinate axis (at an abscissa value of 2.0) for a change in  $Nu$  of 25 per cent with other parameters constant.

The second conclusion is that  $B$  appears to be strongly dependent on the fluid but independent or only weakly dependent on the geometry, surface material, and pressure. This latter point is particularly the case for water where there is sufficient data to give a wide range to these parameters. The value of  $B$  is calculated as  $4.5 \times 10^{-9}$  metre for water and tentatively  $1.7 \times 10^{-11}$  metre for the two non-aqueous liquids studied within the limited data available. The general magnitude of  $B$  under conditions approximating to pool-boiling may be checked by noting that with  $a = 0.7$ ,

$$C_{sf} = C_2 \left( \frac{B}{D_b} \right)^{0.3} \quad (14)$$

For the case of water, substitution of approximate values of 0.003 m for the pool-boiling bubble diameter  $D_b$ , and 0.5 for the suppression constant  $C_2$ , yields a value of

0.009 for  $C_{sf}$ . This compares favourably with the quoted values of between 0.006 and 0.013 for pool-boiling of water. The dimensionless group  $(D/B)$  may be considered as the ratio of the convective and boiling based physical dimensions.

Equation (8) may now be expressed in the following simpler form:

$$Nu = Vo^m Nu_t + Re_v^{0.7} \left( 1 - \frac{Vo^m Nu_t}{Nu} \right)^{0.7} Pr^b \left( \frac{D}{B} \right)^{0.3} \quad (15)$$

Figure 3 is a plot of the calculated two-phase Nusselt numbers as a function of the experimentally determined Nusselt numbers for all the values included in Fig. 2. It yields some confirmation of the concepts used and in particular of the way in which the components have been rationalized and combined.

There remain two areas which require amplification and would reward further experimental study. The power index  $b$  of the Prandtl number has been taken as  $-0.7$  as this value results from rearrangement of the Rohsenow boiling correlation. The value has since been modified for water (14) and the present authors can find no physical reason why the Prandtl number should be raised to a negative power for any liquid in this non-dimensional representation. It is suspected that any error in the power index for the data used in this study is included in the value of  $B$  which is dependent on the fluid.

Equation (10) implies that for any particular fluid/surface combination, where  $C_{sf}$  and  $B$  are constant, that the suppression factor  $C_2$  is proportional to  $D_b^{0.3}$ . The suppression and the bubble diameter at departure both decrease with increasing flow rate and this proportionality is not unfeasible. However, the elongation of the bubble in the direction of flow, as described by Kenning and Cooper (29), makes the physical interpretation of  $D_b$  under flow conditions rather difficult.

Table 1  
Experimental Details

Symbol Figs. 2 and 3	Fluid and surface	Geometry	Equivalent dia. $D$ (mm)	Pressure (bar)	Quality $x$ %	Ref.
■	Water, Stainless steel	Horizontal flow in tube	11.8	3.10	1-15	20
○	Water, Stainless steel	Horizontal flow in tube	11.8	6.21	1-17.9	20
●	Water, Stainless steel	Horizontal flow in tube	11.8	8.27	2-9.3	20
▽	Water, Stainless steel	Horizontal flow in tube	11.8	10.34	12-16	20
▼	Water, Copper	Upward flow in vertical tubes	25.4	1-2.5	1.5-7.7	21
△	Water, Stainless steel	Upward flow in annulus	2.90	Atmos.	20.1-23.3	22
▲	Water, Stainless steel	Upward flow in annulus	3.43	Atmos.	17.5-27	22
□	Water, Stainless steel	Upward flow in annulus	4.52	Atmos.	26.6-27.8	22
×	Water, Stainless steel	Downflow in vertical tube	18.3	1.09-2.13	1.5-7.9	23
▶	Water, Brass	Horizontal flow in tube	8.97	1 to 2	0.15-0.68	24
▷	Water, Inconel	Upward flow in helically coiled tube	12.5	Atmos.	5-20	25
+	Freon 113, Stainless steel	Upward flow over horizontal tube arrays	25.4	Atmos.	1.6-7.2	26
◆	Freon 113, Stainless steel	Upward flow over horizontal tube arrays	25.4	Atmos.	0	27
◇	Cyclohexane, Brass	Upward flow in vertical tube	25.4	Atmos.	0-5	28

## 6 SUMMARY AND CONCLUSION

The correlation for flow boiling heat transfer in saturated liquids has been developed in non-dimensional form as

$$Nu = Vo^m Nu_f + Re_v^{0.7} \left( 1 - \frac{Vo^m Nu_f}{Nu} \right)^{0.7} Pr^{-0.7} \left( \frac{D}{B} \right)^{0.3}$$

The dimensionless numbers are given as

$$Nu = \frac{hD}{k} \quad (\text{Two-phase Nusselt number})$$

$$Nu_f = C_f Re_f^m Pr^n$$

(Normal liquid phase Nusselt number for the geometrical arrangement with  $Re_f = GD(1-x)/\mu$ )

$$Vo = \frac{1}{1-\alpha} \quad (\text{Voidage number})$$

$$= \frac{x(\rho - \rho_g) + \rho_g}{\rho_g(1-x)} \quad (\text{for homogeneous flow})$$

$$= 1 + \left( \frac{6}{X_{tt}} \right)^{0.77} \quad (\text{for separated flow})$$

$$Re_v = \frac{gD}{h_{fg}\mu}$$

$$Pr = c\mu/k$$

The constant  $m$  is defined by the conventional turbulent flow convection expression for  $Nu_f$  and is 0.8 for flow in tubes and ducts and about 0.63 for flow over in-line tube banks. The value of  $B$  is found to be  $4.5 \times 10^{-9}$  metre for water and tentatively  $17.0 \times 10^{-12}$  metre for cyclohexane and Refrigerant 113. Its value for other liquids may differ considerably from these figures as it depends upon the fluid/surface combination and probably the liquid Prandtl number. Determination of  $Nu$  requires an iterative solution owing to its presence in the boiling term.

The relationship is restricted to voidages less than about 0.95 and to geometrical situations for which suitable liquid-phase convective heat transfer expressions are established. In addition it is not applicable near the

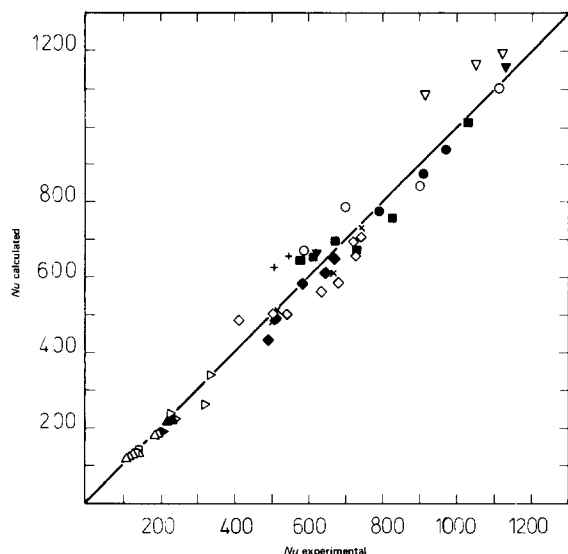


Fig. 3. Correlation of experimental data with eq. (15)

$m = 0.8$ , internal flow

$m = 0.63$ , tube banks

$B = 4.5 \times 10^{-9}$  metre for water

$B = 17.0 \times 10^{-12}$  metre for R113 and cyclohexane

Data point code from Table 1

critical heat flux and has not been verified for pressures of more than 10 bar. It is unfortunate that so much of the experimental work in the literature is reported in insufficient detail to be correlated with a relationship of this type. Further experimental work using a variety of fluids and geometrical flow arrangements has been initiated and the correlation has recently been applied (30) to boiling in a reboiler tube bundle.

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