H. Müller-Steinhagen

N. Epstein

A. P. Watkinson

Department of Chemical Engineering,
The University of British Columbia,
Vancouver, B.C.,
V6T 1W5, Canada

Subcooled-Boiling and Convective Heat Transfer for Heptane Flowing Inside an Annulus and Past a Coiled Wire: Part II—Correlation of Data

In Part I of this paper, the authors reported an extensive series of heat transfer data for subcooled boiling of heptane in turbulent flow in an annulus, and in laminar flow past a coiled wire. These data plus some new measurements for laminar flow in the annulus were compared to the predictions of some 12 correlations from the literature. The applicability of these correlations to the present data is determined and a combination of correlations proposed to predict heat transfer with satisfactory accuracy.

Introduction

The calculation of convective and subcooled boiling heat transfer rates depends largely on available empirical or semi-empirical formulae [1]. The accuracy of predictions of these correlations is uncertain for conditions outside the range of the original data from which they were derived. This is especially true if the flow geometry or the fluid to be used is different from that of the original study. In Part I of this paper, the authors reported heat transfer measurements for subcooled boiling of heptane flowing in an annulus and past a coiled wire under a wide range of flow and thermal conditions. A comparison of these data to values predicted by correlations available in the literature seemed worthwhile for several reasons. The comparison would extend the range of fluids for which the correlations have been tested since subcooled boiling data for heptane were not available in the literature. A suitable correlation could be established for subcooled boiling on a cooled wire in cross flow. This configuration has been used for many heat transfer measurements or experiments [3, 4], yet a systematic investigation of heat transfer characteristics is lacking. Laminar flow data reported in this paper for the annular test section would permit a further verification of the subcooled boiling correlation of Shah [15], which is claimed to be superior to other correlations but has only been compared to few data for laminar flow. Shah himself, therefore, discourages the use of his correlation for Reynolds numbers less than 2300.

The geometry of the two test sections and the range of conditions covered in the experiments are given in Part I, Tables 1 and 2. Data correlated in this part were taken with pressurization by fluid expansion, as the nitrogen pressurization used for some runs in Part I resulted in varying and unknown degrees of nitrogen dissolution in the heptane.

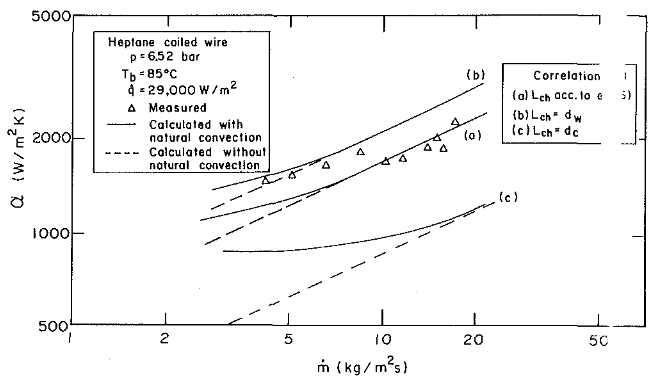

Fig. 1 Influence of the characteristic length definition and of natural convection on the calculated heat transfer coefficient past the coiled wire

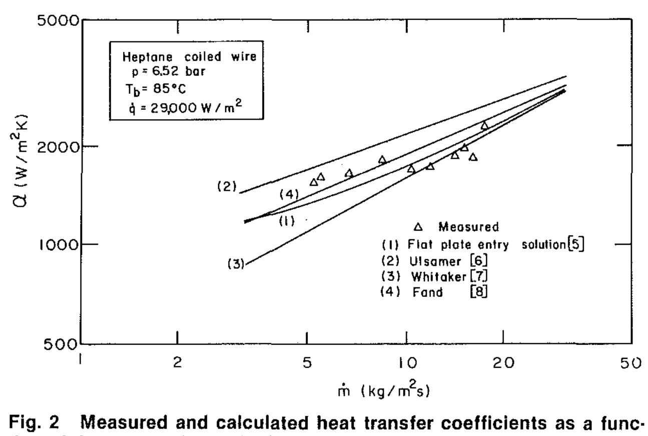

Fig. 2 Measured and calculated heat transfer coefficients as a function of the mass velocity for laminar flow past the coiled wire. For correlations (1) and (4), $L_{ch} = (\pi/2)d_w$.

Correlations Tested

The experimental data were compared with the predictions of the correlations listed in Table 1. Generally, the heat transfer coefficient for convective heat transfer is given as

$$\mathrm{Nu}_c = \frac{\alpha_c L_{ch}}{\lambda} = f(\mathrm{Re},\ \mathrm{Pr},\ \mathrm{Gr},\ x/L) \quad (1)$$

with the Reynolds, Prandtl, and Grashof numbers having their usual form. The (x/L) term usually holds for thermal and hydraulic entrance effects.

Only correlation (4) considers the contribution of natural convection to the heat transfer for the flow past obstacles. However, some authors, for example [18], recommend that a Reynolds number be evaluated by superposition of the forced convection Reynolds number and the Grashof number

$$\mathrm{Re} = (\mathrm{Re}_{fc}^2 + \mathrm{Gr}/2.5)^{0.5} \quad (2)$$

This method was applied to correlations (1) and (6).

Contributed by the Heat Transfer Division for publication in the JOURNAL OF HEAT TRANSFER. Manuscript received by the Heat Transfer Division March 13, 1985.

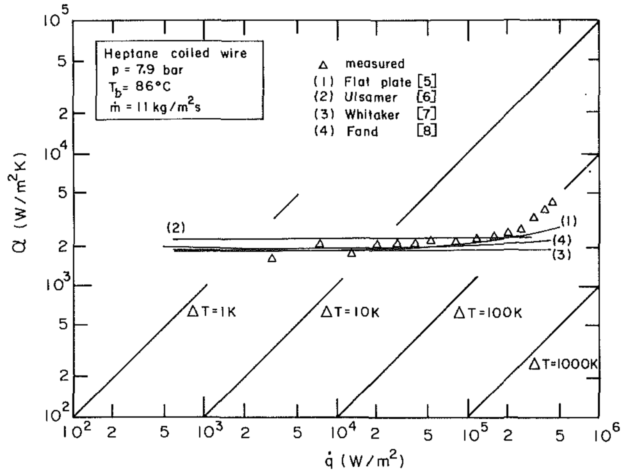

Fig. 3 Measured and calculated heat transfer coefficients as a function of the heat flux for laminar flow past the coiled wire

For flow inside ducts, the characteristic length in the Nusselt and Reynolds numbers is the equivalent diameter

$$d_{eq} = 4A_{cr}/f_w \tag{3}$$

For annular flow, Shah [15] suggests the use of equation (3) only if the annular gap, $(d_o - d_i)/2$, is larger than 4 mm. Otherwise, the heated perimeter should be used instead of the wetted perimeter in equation (3). For the annulus flow geometry given in Part I, the annular gap exceeds 4 mm and therefore

$$d_{eq} = \frac{4\pi(d_o^2 - d_i^2)}{4\pi(d_o + d_i)} = d_o - d_i \tag{4}$$

Chen [16] generally proposed the use of an equivalent diameter based on the heated perimeter. For the investigated conditions, this yields a significant decrease of the predicted heat transfer coefficients. Since the present results as well as about 1000 subsequently obtained but still unpublished data points for flow of heptane and of water indicate that equation (4) should be applied for the investigated conditions, the equivalent diameter according to equation (4) was used for all the correlations in this paper. For the flow past the coiled wire, no recommendations for the calculation of an equivalent diameter or a characteristic length could be found in the literature. However, as the coils used were not very tight, correlations for a straight wire should give satisfactory results. While correlations (2) and (3) are based on the wire diameter, Schlünder [18] suggests the use of correlation (1) with a characteristic length calculated from

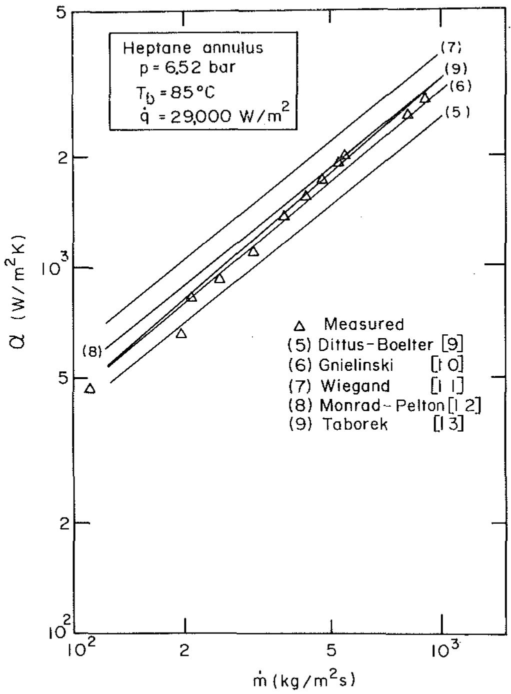

Fig. 4 Measured and calculated heat transfer coefficients as a function of the mass velocity, for turbulent flow in the annulus

$$L_{ch} = A_h/f_p \tag{5}$$

where A_h is the heated area and f_p the circumference of the projection in the direction of the flow. For a cylindrical body normal to the flow this leads to

$$L_{ch} = \frac{\pi d_w L}{2(L + d_w)} \approx \frac{\pi}{2} d_w \tag{6}$$

This definition was also applied to correlation (4). In the case of very narrow coils, L_{ch} should probably be calculated using a superposition of wire and coil diameters. The validity of this assumption will be the subject of further investigation.

Many investigations, for example [1, 2], show that the heat transfer coefficient for subcooled boiling depends on the difference between surface and saturation temperature, $(T_s - T_{sat})$, rather than on the difference between surface and bulk temperature, $(T_s - T_b)$. Therefore, correlations for saturated boiling may be applied to calculate the subcooled boiling heat transfer coefficient by evaluating $(T_s - T_{sat})$ and adding this value to the difference $(T_{sat} - T_b)$. Thus

Nomenclature

A_{cr} = cross-sectional area, m^2
A_h = heated surface area, m^2
c_p = heat capacity, J/(kg)(K)
d_c = coil diameter, m
d_{eq} = equivalent diameter according to equation (3), m
d_i = inside diameter of annulus, m
d_o = outside diameter of annulus, m
d_w = wire diameter, m
f_p = perimeter of projection in flow direction, m
f_w = wetted perimeter, m
g = acceleration due to gravity, m/s^2
Δh_v = latent heat of evaporation, J/kg
L = length, m
L_{ch} = characteristic length, m
$\dot{m}$ = mass velocity, kg/m^{2}s
$\tilde{M}$ = molar mass, kg/kmol
p = pressure, bar
p^* = vapor pressure, bar
$\dot{q}$ = heat flux, W/m^2
S = suppression factor [16]
T = temperature, °C
u = flow velocity, m/s
x = length coordinate, m
α = heat transfer coefficient, W/m^{2}K
β = temperature coefficient of volumetric expansion, K^{-1}
ζ = friction factor
η = viscosity, kg/ms
λ = thermal conductivity, W/mK
ν = kinematic viscosity, m^2/s
ρ = density, kg/m^3
σ = surface tension, N/m

Subscripts

b = bulk
c = convective
crit = critical
fc = forced convection
g = gas
l = liquid
nb = nucleate boiling
s = surface
sat = saturation

Dimensionless numbers

Bo = $\dot{q}/\dot{m}\Delta h_v$ = boiling number
Gr = $g(T_s - T_b)L_{ch}{}^3\beta/\nu^2$ = Grashof number
Pr = $\eta c_p/\lambda$ = Prandtl number
Re = uL_{ch}/ν = Reynolds number

Table 1 Correlations used for comparison with the measured data

No.	Application and formula	Author	Reference
1	Laminar flow over flat plates $Nu_c = 0.664\ Re^{0.5}\ Pr^{0.33}$	Leveque as described by Drew	[5]
2	Laminar flow over tubes and wires $Nu_c = C Re^n Pr^{0.31}$ $C = 0.91,\ n = 0.385 \quad 0.1 < Re < 50$ $C = 0.6,\ n = 0.5 \quad 50 < Re < 10{,}000$	Ulsamer	[6]
3	Laminar flow over tubes and wires $Nu_c = (0.4\ Re^{0.5} + 0.06\ Re^{0.67}) \times Pr^{0.4}\ (\eta_w/\eta_b)^{0.25}$	Whitaker	[7]
4	Laminar forced and natural convection over cylinders $Nu_c = (0.255 + 0.699\ Re^{0.5} + 0.033 \times (Gr/Re^2)^{0.3} \cdot Gr^{0.25}) \cdot Pr^{0.29}$	Fand	[8]
5	Turbulent flow in pipes $Nu_c = 0.023\ Re^{0.8} Pr^{0.4}$	Dittus/ Boelter	[9]
6	Turbulent flow in pipes $\zeta = (1.82 \log Re - 1.64)^{-2}$ $Nu_c = \dfrac{\zeta/8(Re-1000)Pr}{1+12.7(\zeta/8)^{0.5}(Pr^{2/3}-1)} \cdot [1+(d/L)^{0.66}]$	Gnielinski	[10]
7	Turbulent flow in annuli $Nu_c = 0.023\ Re^{0.8}\ Pr^{0.4}(d_o/d_i)^{0.45}$	Wiegand	[11]
8	Turbulent flow in annuli $Nu_c = 0.02\ Re^{0.8}\ Pr^{0.33}(d_o/d_i)^{0.53}$	Monrad/ Pelton	[12]
9	Turbulent flow in pipes $Nu_c = 0.0143\ Re^{0.85}\ Pr^{0.5}$	Taborek	[13]
10	Pool boiling heat transfer to saturated liquids $\alpha_{nb} = C_1(\dot{q})^{C_2}$ with C_1, C_2 to be taken from diagrams	Stephan	[14]
11	Flow boiling heat transfer to subcooled liquids $Bo > 3 \cdot 10^{-5} \rightarrow \phi_o = 230\ Bo^{0.5}$ $Bo < 3 \cdot 10^{-5} \rightarrow \phi_o = 1 + 46\ Bo^{0.5}$ $\dfrac{T_{sat}-T_b}{T_s-T_{sat}} > 2 \rightarrow \phi = \phi_o + \dfrac{T_{sat}-T_b}{T_s-T_{sat}}$ $\dfrac{T_{sat}-T_b}{T_s-T_{sat}} > 63000\ Bo^{1.25} \rightarrow \phi = \phi_o + \dfrac{T_{sat}-T_b}{T_s-T_{sat}}$ Otherwise $\phi = \phi_o$ α_c according to correlation No. 5 if $Re > 2300$ $\dot{q} = \alpha_c \phi (T_s - T_{sat})$	Shah	[15]
12	Flow boiling heat transfer α_c according to correlation No. 5 if $Re > 2300$ $\alpha_{nb} = 0.00122 \left(\dfrac{\lambda_l^{0.79} c_{pl}^{0.45} \rho_l^{0.49}}{\sigma^{0.5} \eta_l^{0.29} \Delta h_v^{0.24} \rho_g^{0.24}}\right) \cdot \Delta T_{sat}^{0.24} \cdot \Delta p_{sat}^{0.75}$ $\Delta T_{sat} = T_s - T_{sat},\ \Delta p_{sat} = \dfrac{(T_s - T_{sat})\Delta h_v}{T_{sat}(\rho_g^{-1} - \rho_l^{-1})}$ $\dot{q} = \alpha_c(T_s - T_b) + \alpha_{nb}(T_s - T_{sat}) \cdot S = \alpha(T_s - T_b)$ The suppression factor S is given by Chen in a diagram as a function of the Reynolds number down to $Re = 13{,}000$ at which $S = 0.85$. It was extrapolated to have a value of unity for $Re \leq 3500$.	Chen	[16]

In the above correlations the physical properties should be evaluated for the
(*a*) Grashof number at $(T_s + T_b)/2$
(*b*) forced convective heat transfer correlations at T_b
(*c*) nucleate boiling correlation at T_{sat}

$$(T_s - T_{sat}) = f(\dot{q}) \tag{7}$$

and

$$\alpha = \frac{\dot{q}}{(T_s - T_{sat}) + (T_{sat} - T_b)} \tag{8}$$

This method was used for correlations (10) and (12) (Table 1). As the nucleate boiling heat transfer coefficient was found to be independent of the mass velocity [1, 19], pool boiling correlations could also be considered.

From the numerous correlations published for boiling heat transfer, only three correlations were chosen for this study, each originally representing a different application. While the Shah correlation was suggested recently [15] for subcooled flow boiling in annuli, the correlations of Chen [16] and of Stephan [14] were developed for saturated flow boiling and for pool boiling, respectively.

Comparison Between Measured and Calculation Values

Convective Heat Transfer

(*a*) *Laminar Flow Past the Coiled Wire.* Laminar flow

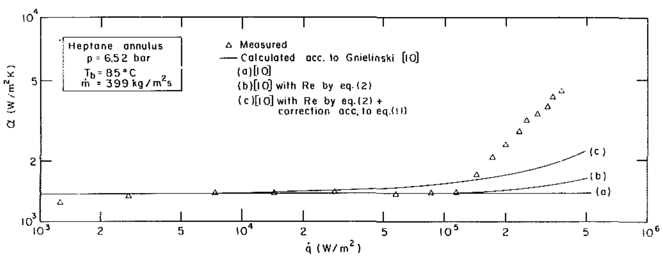

Fig. 5 Influence of the heat flux on the convective heat transfer coefficient for turbulent flow in the annulus

was investigated for the coiled wire, with Reynolds numbers between 3.5 and 14.6 based on the wire diameter and between 5.5 and 23 based on the characteristic length calculated according to equation (6). Figure 1 shows measured heat transfer coefficients as a function of the mass velocity as well as values calculated according to correlation (1), using different definitions of the characteristic length, with and without inclusion of natural convection. As can be seen, the version with L_{ch} given by equation (6), including the influence of natural convection according to equation (2), yields the best agreement with the measured data. This result also holds for correlation (4). In correlations (2) and (3) the wire diameter d_w has to be used, according to [6, 7]. A comparison of the results calculated using correlations (1–4) with the measured data is given in Fig. 2. All four correlations provide roughly comparable values; however, correlation (2) predicts coefficients that are too large. Better agreement is achieved using correlations (1) and (4). The influence of the heat flux on the convective heat transfer coefficient is shown in Fig. 3. With increasing heat flux (or excess temperature), the growing contribution of natural convection leads to a slight enhancement of the heat transfer. However, for the present investigation, the effect of natural convection was found to be small in most cases, in accordance with the criterion given in [19], which states that the contribution of natural convection may be neglected as long as $Gr/Re^2 < 1$. In the present convective heat transfer experiments, $0.0001 < Gr/Re^2 < 4.4$

(*b*) *Turbulent Flow in the Annulus.* Figure 4 shows measured and calculated heat transfer coefficients for annular flow, plotted as a function of the mass velocity. For all measurements, the best results were obtained with the correlation of Gnielinski [10], which is based on all data for turbulent heat and mass transfer in pipes that Gnielinski was able to ex-

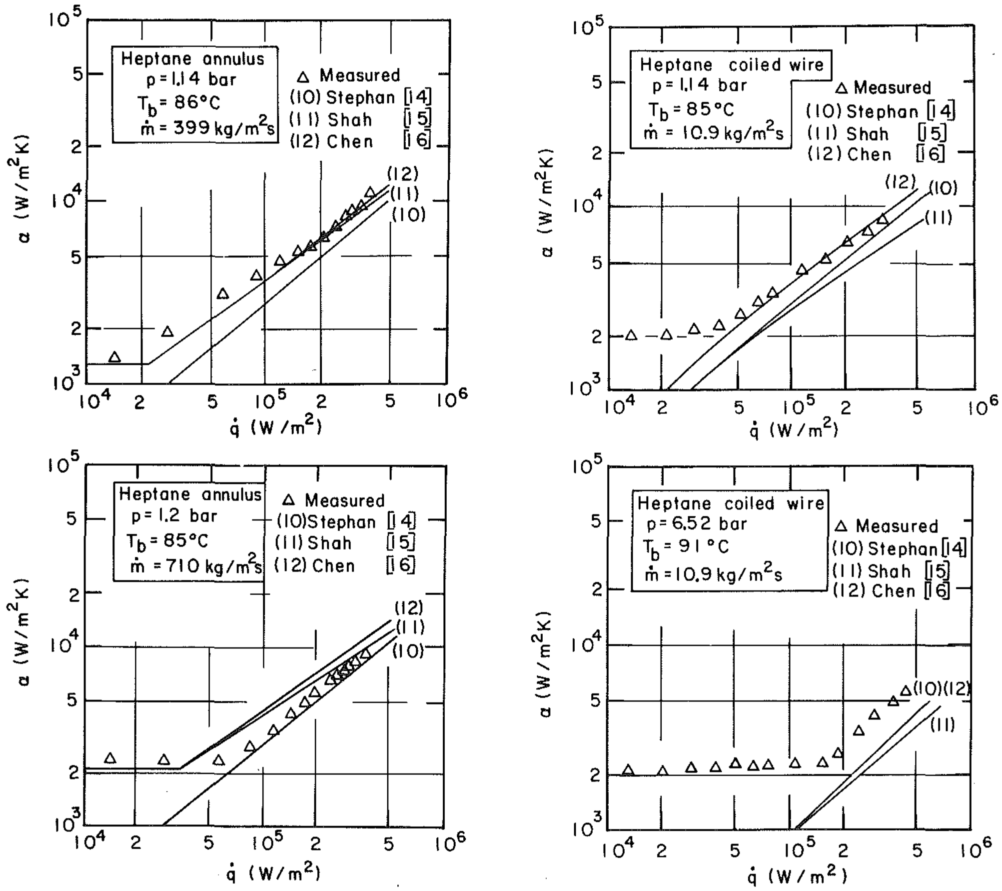

Fig. 6 Measured and calculated heat transfer coefficients for subcooled boiling as a function of the heat flux

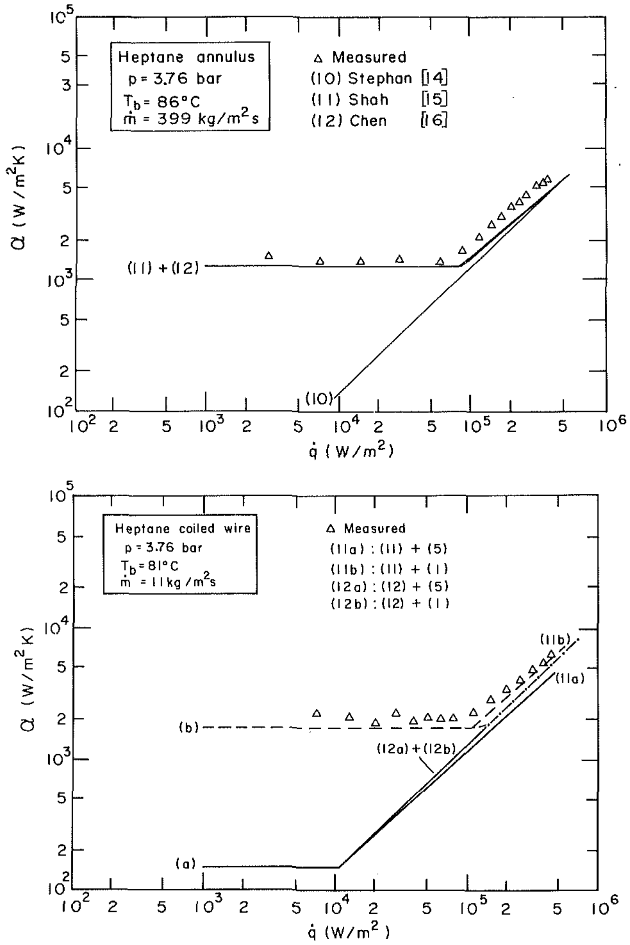

Fig. 7 Measured and calculated heat transfer coefficients for convective boiling in (*a*) turbulent and (*b*) laminar flow

tract from the literature. This equation is also recommended in [20].

The influence of the heat flux on the measured convective heat transfer coefficients for turbulent annular flow is given in Fig. 5. Again, the contribution of natural convection is negligible, as the measured heat transfer coefficients are independent of the heat flux as long as convective heat transfer predominates ($\dot{q} < 120{,}000$ W/m^2). Figure 5 also contains three curves calculated according to correlation (6), each showing a different influence of the heat flux on the calculated heat transfer coefficient. For the original correlation, curve (*a*), the heat transfer coefficient is independent of the heat flux, because the physical properties of the liquid were evaluated at the bulk temperature. Curve (*b*) shows the calculated results if the influence of natural convection is considered using equation (2). As the flow velocity is rather high, slight deviations between (*a*) and (*b*) are found only at high heat fluxes. Some authors suggest that the influence of the heat flow direction (e.g., cooling or heating) be included by multiplying the coefficient by a factor

$$F = \left(\frac{\mathrm{Pr}_b}{\mathrm{Pr}_s}\right)^n \tag{9}$$

with n being about 0.25. For the convective heat transfer measurements described in Part I, this procedure yields a small overprediction of the heat transfer coefficient at high heat fluxes as seen in Fig. 5, curve (*c*).

Subcooled Nucleate Boiling. Figure 6 shows measured and calculated subcooled boiling heat transfer coefficients plotted against the heat flux for the two different heater geometries. For the annulus, data are shown for two different mass

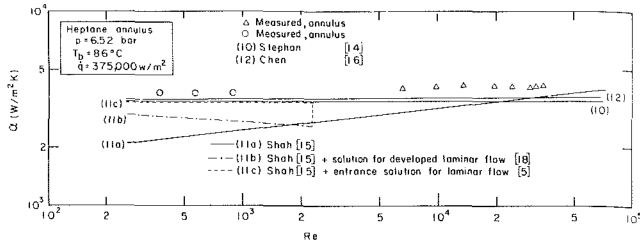

Fig. 8 Influence of the Reynolds number on the subcooled boiling heat transfer coefficient

velocities, while for the coiled wire two different pressures have been employed at the same mass velocity. All three correlations used for comparison, namely correlations (10), (11), and (12), yield results which generally agree with the measured data in the nucleate boiling region. Only at a pressure of 6.52 bar are some deviations between measured and calculated values to be noted. These deviations might be caused by minor amounts of residual nitrogen dissolved in the heptane (see Part I) and though they could also represent a more general failure of the three correlations at high pressure.

The pool boiling correlation of Stephan [14] is only applicable to the nucleate boiling region, whereas those of Shah [15] and Chen [16] are supposed to cover both the convective and the nucleate boiling regimes. According to Shah [15], there exists no experimental verification of the applicability of the Chen correlation to subcooled boiling. Figure 7 shows some measurements of the heat transfer coefficient as a function of the heat flux for both turbulent flow in the annulus (*a*) and laminar flow past the coiled wire (*b*). Both sets of measurements are compared with calculated values according to correlation (10), which only holds for fully developed nucleate boiling, and to correlations (11) and (12), which hold for both the convective and the subcooled boiling regimes. Both the latter authors use the Dittus–Boelter equation [9] for calculating the convective heat transfer; however, the increase in heat transfer as nucleate boiling occurs is accommodated by an additive expression in the correlation of Chen [16] and by a multiplication factor in the correlation of Shah [15]. Both correlations do well for the turbulent flow in the annulus (Fig. 7*a*). The flow past the coiled wire is a developing laminar flow; thus the Dittus–Boelter equation is not applicable (see Fig. 7*b*, curve (*a*)). Therefore, correlation (1) was used for the calculation of the heat transfer coefficients in the laminar convective heat transfer region. As Fig. 7(*b*) shows, this procedure gives reasonable agreement between measured and calculated values. However, while the developed boiling heat transfer coefficients according to Chen [16] are independent of the correlation used for convective heat transfer, the values calculated according to Shah [15] change if the convective heat transfer is varied.

To demonstrate the influence of the flow velocity on the heat transfer, measured and calculated heat transfer coefficients are plotted against the Reynolds number in Fig. 8. The data at low Reynolds numbers were not reported in Part I. As found by numerous authors [1, 19], the heat transfer coefficient for fully developed boiling is independent of the flow velocity. (The boiling heat transfer coefficients for the coiled wire, which were measured at characteristic length Reynolds numbers between 5 and 20, are very similar to those for the annulus.) This result is predicted only by the correlations of Stephan [14] (which was developed for pool boiling) and Chen [16], whereas Shah's correlation [15] shows a clear influence of the Reynolds number on the boiling heat transfer coefficient and considerable deviations between measured and calculated results. In Fig. 8, three possible variations of the Shah correlation were used to evaluate the heat transfer coefficients, namely, one for turbulent convective heat transfer (*a*) and the other two for different cases of laminar convective heat transfer (b and c). For the turbulent flow region ($Re > 2300$), the agreement between predicted values and measured values decreases considerably with decreasing Reynolds number. This trend would continue if the Dittus–Boelter equation [9] were also used in the laminar flow regime ($Re < 2300$), as Shah [15] tentatively suggested for saturated flow boiling. If, for $Re < 2300$, the Shah correlation is instead combined with a correlation for laminar convective heat transfer, a clear improvement of the predicted values can be seen in Fig. 8. Although the measured heat transfer coefficients were for developed flow, the combination of the Shah correlation with the correlation for developing laminar flow gives a fortuitously good agreement with the measured data, since the Reynolds number influence is just canceled in the above combination. If the "correct" combination for developed laminar flow were used for $Re < 2300$, the agreement between predicted and measured values would be poorer, but it would tend to improve for low Reynolds numbers.

Nevertheless, the influence of mass velocity is not properly accounted for in [15] and considerable deviations between predicted and measured heat transfer coefficients cannot be avoided for Reynolds numbers between 2300 and 10,000.

As described in Part I, increasing the system pressure at constant bulk temperature or decreasing the bulk temperature at constant system pressure yields a reduction of the heat transfer coefficient defined according to equation (8). This trend is identically predicted by the three correlations for boiling heat transfer.

Conclusion

Several correlations from the literature were checked as to their applicability to subcooled boiling of heptane flowing in an annulus and past a coiled wire. The best results were obtained using the Chen correlation [16] for convective boiling. However it is suggested that the convective term in the correlation be replaced either by Gnielinski's more up-to-date correlation for turbulent heat transfer in pipes [10] or by the entrance region solution for flow along flat plates [5], for turbulent flow in the annulus and laminar flow past the coiled wire, respectively. Similar accuracy with less evaluation of physical properties is achieved by using the higher of the two values produced by using either the Stephan correlation [14] or the appropriate correlation for convective heat transfer [5, 10]. In the case of lower Reynolds numbers, the application of the Shah correlation [15] may yield considerable errors in the prediction of the subcooled boiling heat transfer coefficient.

Acknowledgments

The authors are indebted to he German Research Council (D.F.G.), Bonn, Bad-Godesberg, for financing the fellowship of H. M. -St., and to the Natural Sciences and Engineering Research Council of Canada for continuing financial support.

References

1 Collier, J. G., *Convective Boiling and Condensation*, McGraw-Hill, New York, 1972.

2 Rohsenow, W. M., and Clark, J. A., "Heat Transfer and Pressure Drop Data for High Heat Flux Densities to Water at High Subcritical Pressure," 1951 Heat Transfer and Fluid Mechanics Institute, Stanford University Press, Stanford, CA, 1951.

3 Pike, F. P., Miller, P. D., and Beatty, K. O., "Effect of Gas Evolution on Surface Boiling at Wire Coils," *Chemical Engineering Progress Symposium Series*, No. 17, Vol. 51, 1955, pp. 13–19.

4 Fetissoff, P. E., Watkinson, A. P., and Epstein, N., "Comparison of Two Heat Transfer Fouling Probes," *Proceedings of the 7th International Heat Transfer Conference*, Munich, FRG, Vol. 6, 1982, pp. 391–396.

5 Drew, T. B., "Mathematical Attacks on Forced Convection Problems," *Transactions AIChE*, Vol. 26, 1931, pp. 26–80.

6 Ulsamer, J., "Wärmeabgabe von geheizten Drähten und Rohren," *Forschungsgebiet Ingenieurwesen*, Vol. 3, No. 94, 1932 (cited in McAdams, W., *Heat Transmission*, McGraw-Hill, New York, 1954.)

7 Whitaker, S., "Forced Convection Heat Transfer Correlations for Flow in Pipes, Past Flat Plates, Single Cylinders, Single Spheres and for Flow in Packed Beds and Tube Bundles," *AIChE Journal*, Vol. 18, 1972, pp. 361–371.

8 Fand, R. M., and Keswani, K. K., "Combined Natural and Forced Convection Heat Transfer From Horizontal Cylinders to Water," *International Journal of Heat and Mass Transfer*, Vol. 16, 1973, pp. 1175–1191.

9 Dittus, F. W., and Boelter, L. M. K., "Heat Transfer in Automobile Radiators of the Tubular Type," University of California Press, Vol. 2, No. 13, 1930.

10 Gnielinski, V., "Neue Gleichungen fur den Wärme- und Stoffübergang in turbulent durchströmten Rohren und Kanälen," *Forschung im Ingenieurwesen*, Vol. 41, No. 1, 1975, pp. 8–16.

11 Wiegand, J. H., "Discussion of Annular Heat Transfer Coefficients for Turbulent Flow," *Transactions AIChE*, Vol. 41, No. 5, 1945, pp. 147–152.

12 Monrad, C. C., and Pelton, J. F., "Heat Transfer by Convection in Annular Spaces," *Transactions AIChE*, Vol. 38, 1942, pp. 593–608.

13 Taborek, J., HTRI, Alhambra, CA, private communication, 1981.

14 Stephan, K., and Abdelsalam, M., "Heat Transfer Correlations for Natural Convection Boiling," *International Journal of Heat and Mass Transfer*, Vol. 23, 1980, pp. 73–87.

15 Shah, M., "Generalized Prediction of Heat Transfer during Subcooled Boiling in Annuli," Heat Transfer Engineering, Vol. 4, No. 1, 1983, pp. 24–31.

16 Chen, J. G., "A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow," *Proceedings of the 6th National Heat Transfer Conference*, Boston, 1963.

17 Razjevic, K., *Handbook of Thermodynamic Tables and Charts*, Hemisphere Publishing Company, Washington, 1976.

18 Schlünder, E. U., *Einführung in die Wärme- und Stoffübertragung*, Vieweg-Verlag, 1975.

19 Grassmann, P., *Physical Principles of Chemical Engineering*, Pergamon Press, New York, 1971.

20 Gnielinski, V., "Wärmeübertragung in Rohren," *VDI-Wärmeatlas*, 3rd ed., VDI-Verlag Düsseldorf, 1975.