ANL/ET/CP--84414
Conf-941128--5

Boiling Heat Transfer in Compact Heat Exchangers*

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Prepared for presentation at the 1994 AIChE Annual Meeting, San Francisco, CA, November 14-18, 1994; session on Compact Heat Exchangers.

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

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ABSTRACT

Small circular and noncircular channels are representative of flow passages in compact evaporators and condensers. This paper describes results of an experimental study on heat transfer to the flow boiling of refrigerant-12 in a small circular tube of diameter = 2.46 mm. The objective of the study was to assess the effect of channel size on the heat transfer coefficient and to obtain additional insights relative to the heat transfer mechanism(s). The flow channel was made of brass and had an overall length of 0.9 m. The channel wall was electrically heated, and temperatures were measured on the channel wall and in the bulk fluid stream. Voltage taps were located at the same axial locations as the stream thermocouples to allow testing over an exit quality range of 0.21 to 0.94 and a large range of mass flux (63 to 832 kg/m^2s) and heat flux (2.5 to 59 kW/m^2). Saturation pressure was nearly constant, averaging 0.82 MPa for most of the testing; a few test data were also taken at a constant lower pressure of 0.52 MPa. Local heat transfer coefficients were determined experimentally. Analysis provided additional support for the conclusion, arrived at from previous studies, that a nucleation mechanism dominates for flow boiling in small channels; nevertheless, a convective-dominant region was identified at very low values of wall superheat ($<\approx 3$°C). Previous flow boiling studies in small channels, that did not include wall superheats this low, did not encounter the convective dominant mechanism. Conversely, cryogenic studies at very low wall superheats (≈ 1°C) did not encounter the nucleation dominant regime. The apparent discrepancy is explained by the results of this study.

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INTRODUCTION

Compact heat exchangers have been defined as having a surface-area density ratio greater than 700 m^2/m^3 (Shah 1986); for a circular tube, this translates to a diameter of <6 mm. The higher heat transfer surface-area density inherent in compact heat exchangers allows significantly higher heat flux levels to be attained relative to two-phase flows in conventional circular tube exchangers. An additional consideration with compact evaporators is the effect of flow passage geometry and size on the two-phase flow and heat transfer phenomena. For example, in the noncircular passages of compact evaporators, geometry may influence the liquid inventory (flow pattern) at a given cross section via surface tension and capillary force action.

Studies reporting in the open literature on vaporization in compact heat exchangers are relatively few. They can conveniently be grouped as covering exchangers with offset strip fin passages (Panitsidis et al. 1975; Galezha et al. 1976; Yung et al. 1980; Chen and Westwater 1984), exchangers with perforated fin passages (Panchal 1984, 1989), multichannel arrangements with offset strip fins (Robertson 1979, 1983; Carey and Mandrusiak 1986; Mandrusiak et al. 1988; Mandrusiak and Carey 1989), and multichannel arrangements with perforated fins (Robertson and Wadekar 1988; Wadekar 1992). Single-channel studies of flow boiling of refrigerant R-113 in a small-diameter circular tube (approximately 3 mm) have been reported by Lazarek and Black (1982) and Wambsganss et al. (1993). Boiling in single, small rectangular passages have been reported by Tran et al. (1993) and Peng and Wang (1993). In particular, Tran et al. (1993) studied flow boiling of refrigerant R-12 in a 4.06 x 1.70 mm rectangular channel, while Peng and Wang (1993) reported on flow boiling of water in an 0.6 x 0.7 mm rectangular passage.

Relative to the dominant heat transfer mechanism, results from tests on actual heat exchangers (Galezha et al. 1976; Chen and Westwater 1984; Panchal 1984) suggested a nucleation-dominant mechanism. Galezha et al. (1976) showed heat transfer coefficients to vary with heat flux, and Panchal (1984) showed heat transfer coefficients to be insensitive to flow rate. On the other hand, investigations with multipassage arrangements (Robertson, 1979, 1983; Carey and Mandrusiak 1986; Mandrusiak et al. 1988; Mandrusiak and Carey 1989; Robertson and Wadekar 1988; Wadekar 1992), all showed nucleation not to be an important mechanism. The heat transfer coefficients were independent of heat flux, dependent on mass flux, and increased with quality (all features of forced convective boiling). This apparent contradiction will be reconciled, in part, in this paper.

Investigators of boiling in small smooth channels (circular and rectangular) (Wambsganss et al. 1993, Tran et al. 1993, Peng and Wang 1993) all concluded that a nucleation mechanism dominates. For the range of parameters tested, the measured heat transfer coefficients were effectively independent of mass flux and quality and were dependent on heat flux.

It is recognized that the dominant heat transfer mechanism is determined, in part, by the range of test conditions employed and that this can be expected to explain, in part, the differences in conclusions reached by different researchers. However, it is also clear that in order to generate the technology base required for the development of design methods and standards for boiling in the flow passages of compact evaporators, there is a need to better understand these mechanisms and their transitions.

In the present study, flow boiling heat transfer with R-12 in a circular 2.46 mm i.d. brass tube was investigated. Tests were performed over a large range of mass flux and heat flux. The lower end of the range of heat flux (wall superheat) was also extended in an attempt to identify a convective boiling region and provide information on the associated convective/nucleate boiling transition.

TEST APPARATUS AND INSTRUMENTATION

The test apparatus and test procedure have already been described in some detail by Wambsganss et al. (1993) and Tran et al. (1993). Consequently, they are summarized only briefly here for completeness.

The test apparatus shown in Fig. 1 is a closed-loop system with system pressure controlled by high-pressure nitrogen via a pressure regulator and a bladder-type accumulator. The fluid enters the test section in a subcooled state and is evaporated in the test section to a quality of ≈94% or less, depending on mass flux and heat flux. (The critical heat flux is not exceeded in the test section.) The two-phase mixture leaving the test section is condensed and subcooled before entering the pump. Flow rate is measured with a variable area flow meter.

Based on results of previous investigations (Wambsganss et al. 1993; Tran et al. 1993; Peng and Wang 1993), nucleation is expected to be the dominant heat transfer mechanism. The flow channel shown in Fig. 2 has an overall length of 0.9 m. The channel is resistance-heated by passing a DC current through the channel wall. Heat input to the fluid was determined from the electric power input to the channel, accounting for heat losses to the environment.

In-flow temperatures of the bulk fluid were measured at four axial locations: the inlet and outlet, and near two intermediate current clamps. Pressure ports and voltage taps were also provided at each of these four locations. Both inlet pressure and a two-phase pressure drop were measured. Wall temperatures were measured at 18 locations along the length of the channel by means of surface-mounted RTDs. The locations at which the various measurements were made are given in Fig. 2. Liquid tests (isothermal and heat balance) were used to establish uncertainty in temperature measurements of ±0.25°C.

TEST PROCEDURE AND DATA REDUCTION

As with the test apparatus and instrumentation discussed above, the test procedure and data reduction methodology have already been described in detail by Wambsganss et al. (1993) and Tran et al. (1993) and are only summarized below for completeness.

Single-phase tests were first performed to provide (1) an overall system check of instrumentation, calibration, and data acquisition equipment and techniques, and (2) a determination of heat loss to the environment. Subsequently, a series of flow boiling tests was performed at constant values of mass flux and selected values of heat flux.

The local evaporative heat transfer coefficient was calculated as

$$h(z) = \frac{q''}{[T_w(z) - T_{sat}(z)]}, \qquad (1)$$

where $q'' = \eta Q_E/S(L_H - L_{SB})$. The quality at the measurement location z was calculated as

$$x(z) = \frac{S(z - L_{SB})q''}{AGi_{fg}}. \qquad (2)$$

In Eq. 1, the wall temperatures were measured directly, while the saturation temperatures were obtained indirectly from a two-phase pressure drop and exit saturation temperature measurement, following a procedure outlined in Tran et al. (1993). In some cases, a temperature measurement centered in the two-phase region served to verify the accuracy of this procedure.

For each of the test runs corresponding to constant mass and heat flux, local heat transfer coefficients were determined for a range of qualities. In virtually all cases, the heat transfer coefficients were effectively independent of quality for qualities greater than 20%. Typical results are shown in Fig. 3. As a consequence of this independence of heat transfer on quality, the test results of this study are average heat transfer coefficients—obtained as the average of the measured local heat transfer coefficients for qualities greater than 20%. Average wall superheats for given test runs are also calculated and used in the presentation of results. The product of the averaged heat transfer coefficient $\bar{h}$ and averaged wall superheat ΔT_{sat} is equal to the heat flux q''.

EXPERIMENTAL RESULTS

A total of 137 test runs are reported. The parameter ranges covered by these tests are given in Table 1.

Table 1. Test parameter ranges

Parameter	Range
Mass flux (kg/m^2s)	63-832
Heat flux (kW/m^2)	2.5-59
Mass quality	to 94%
Saturation pressure (kPa)	820 and 520

The heat transfer tests were performed so as to isolate the effects of heat flux, mass flux, and quality. In particular, tests were performed for selected values of mass flux with various heat flux levels, as well as for selected values of heat flux with various values of mass flux.

Results from all tests with the 2.46 mm circular tube are presented in Figs. 3-5. The local heat transfer coefficient is seen in Fig. 3 to be only weakly dependent on quality, allowing for computation of an average heat transfer coefficient. The results presented in Fig. 4, showing average heat transfer coefficient as a function of mass flux for various values of heat flux, clearly show that for the range of heat fluxes tested, the heat transfer coefficient is effectively independent of mass flux.

In Fig. 5, data from the tests are plotted in terms of heat flux and average wall superheat. The data can be correlated approximately with a straight line on log-log coordinates, indicating a power function relationship between heat flux and wall superheat (as shown in the figure).

Two additional test series were performed: the first at a different (lower), value of saturation pressure; and the second at very low values of heat flux for two different values of mass flux. Figure 6 shows the effect of saturation pressure. Figure 7 shows the results of tests performed at two different values of mass flux (approximately 75 and 150 kg/m^2s) to extend the data base to lower values of heat flux. At the lower values of heat flux, two distinct curves—each having a slope of approximately 1 on log-log coordinates—and corresponding to each of the two values of mass flux tested can be identified.

DISCUSSION

Heat Transfer Mechanisms

The two fundamental boiling heat transfer mechanisms are forced convection and nucleation. In forced convective dominant boiling, the heat transfer coefficient is independent of heat flux and dependent on mass flux and quality; heat transfer increases with increasing mass flux and quality. On the other hand, when nucleation dominates, heat transfer is independent of mass flux and quality and dependent on heat flux and saturation pressure.

With these definitions, the results shown in Figs. 3-7 lead one to conclude that over a broad range of heat flux, nucleation is the dominant heat transfer mechanism for flow boiling in the small passages considered in this study. This is in agreement with previous investigations (Lazarek and Black 1982; Wambsganss et al. 1993; Tran et al. 1993; Peng and Wang 1993) of flow boiling in small channels. The results of Fig. 6 show that saturation pressure has a measurable effect on heat transfer, decreasing the heat transfer coefficient with decreasing pressure. Because this sensitivity and trend is expected of nucleation-dominant heat transfer, it also serves to support the conclusion that a nucleation mechanism dominates.

It has been shown that a nucleation mechanism dominates over a broad range of heat flux values. Nevertheless, it was expected that at sufficiently low values of heat flux (very low wall superheat), forced convection will dominate. This was indeed shown to be the case, as illustrated in Fig. 7; at wall superheats of less than about 3°C, the boiling curve is a function of mass flux and the slope of the curve is approximately unity, implying that the heat transfer coefficient is independent of heat flux.

The transition from convective to nucleate boiling is well-defined and relatively abrupt. This was clearly evident in taking the data as the system became quasistable at transition, abruptly changing from convective to nucleation dominant. This behavior differs from that found with larger-diameter channels in which relatively broad transition regions occur, typically with contributions from both convective and nucleate boiling.

Identification of a convective-dominant region allows one to reconcile an apparent disagreement with Robertson and coworkers (Robertson 1979, 1983; Robertson and Wadekar 1988; Wadekar 1992), who concluded from their tests with multichannel arrangements that nucleation is not an important mechanism in small channel flow boiling. Their finding of convection dominance was based on data obtained for very low values of wall superheat (< 2.5°C), and indeed the results the present study for very low wall superheats would also lead to this conclusion. It remains, however, to reconcile differences in conclusions arrived at by Carey and coworkers (Carey and Mandrusiak 1986; Mandrusiak et al. 1988; Mandrusiak and Carey 1989) relative to the dominant mechanism.

Data Trends

The fact that the data shown in Fig. 5 can be reasonably correlated with a straight line suggests a power function relationship between heat flux and wall superheat. Therefore, a dimensional correlation equation can be written of the form

$$q'' = C_1(\Delta T_{sat})^{C_2} \tag{3}$$

where q" is heat flux with units kW/m^2, and ΔT_{sat} has units of °C. This is the correlation form for nucleate pool boiling, such as that developed by Stephan and Abdelsalam (1980). The constants in Eq. 3 obtained from curve fits to the experimental data, and for the Stephan and Abdelsalam correlation for R-12, are given in Table 2.

Table 2. Correlation coefficients

	C_1	C_2	C_3	C_4
Small circular channel	0.479	2.60	0.763	0.620
Stephan-Abdelsalam Correlation (R-12)	0.0364	3.92	0.430	0.745

In Fig. 8, the developed correlation equation for the small circular tube is used for comparison with the Stephan and Abdelsalam correlation in the broad nucleation dominant region ($\Delta T_{sat} > 3$°C). The Stephan and Abdelsalam correlation successfully predicted previous small circular-tube data over the range of test conditions with R-113 as the boiling fluid (Wambsganss et al. 1993). The results show that, at low values of wall superheat, the Stephan and Abdelsalam correlation significantly underpredicts the R-12 data.

Data Correlation

Equation 3 was used to develop a dimensional correlation for the average heat transfer coefficient in the form

$$\bar{h} = C_3 q''^{C_4}, \tag{4}$$

where $\bar{h}$ has units W/m^2C, and q" has units kW/m^2. The coefficients C_3 and C_4 are given in Table 2. The circular tube correlation predicts 98% of the data within ±10% as shown in Fig. 9. This illustrates that a nucleate pool boiling correlation can be used to predict boiling heat transfer in small channels for wall superheats greater than 3°C.

The coefficients C_3 and C_4 in Eq. 4 may be functions of channel geometry and size, as well as of fluid properties and, possibly, surface conditions. Additional data will be required to develop generalized equations for these coefficients.

SUMMARY AND CONCLUSIONS

Boiling heat transfer measurements were made with R-12 in a small circular channel (d_h = 2.46 mm) over a substantial range of heat flux, mass flux, and quality. At all but the lowest wall superheats, heat transfer was found to be dependent on heat flux rather than on mass flux. This condition had been found previously in a small rectangular channel (Tran et al. 1993) with R-12 and in a small circular channel with R-113 (Lazarek and Black 1982; Wambsganss et al. 1993). This result implies that the nucleation mechanism dominates over the convective mechanism in small channels over the full range of qualities (precritical heat flux qualities <0.94); this is contrary to situations in larger channels where the convective mechanism dominates at qualities typically >0.2.

Experiments were conducted at very low wall superheats where, for ΔT_{sat} <≈3°C, the convective dominant region was measured. Here, the heat transfer was dependent on mass flux and not heat flux. The transition between regions of nucleate and convective dominance was found to be rather sharp. These are the first small channel results reported where both mechanisms were obtained in the same study. These results explain previous contradictory results in the literature of ΔT_{sat} above or below 3°C.

Data in the nucleation dominant region were correlated by the form of the Stephan and Abdelsalam equation (Stephan and Abdelsalam 1980) for pool boiling, where the heat transfer coefficient depends on the heat flux to the 0.62 power. The original Stephan and Abdelsalam correlation was found to predict R-113 data well in small circular channels, but it significantly underpredicted the R-12 data. Consequently, data from more fluids is required to resolve the property influence on the heat transfer.

ACKNOWLEDGMENTS

This work was supported by the U.S. Department of Energy, Energy Efficiency and Renewable Energy, Office of Industrial Technologies, under Contract W-31-109-Eng-38. The work also represents a U.S. contribution to the International Energy Agency (IEA) program on Research and Development in Heat Transfer and Heat Exchangers.

The authors thank Joseph A. Jendrzejczyk for technical contributions during the course of the work, Roger K. Smith for his contributions in fabricating and instrumenting the experiment, and Joyce A. Stephens for preparing the figures and overall manuscript for publication.

NOMENCLATURE

A	Channel cross-sectional flow area (m^2)
C_1	Coefficient in Eq. (3)
C_2	Coefficient in Eq. (3)
C_3	Coefficient in Eq. (4)
C_4	Coefficient in Eq. (4)
G	Mass flux (kg/m^2s)
h	Local heat transfer coefficient (W/m^2C); Eq. (1)
$\overline{h}$	Averaged heat transfer coefficient (W/m^2C)
i_{fg}	Latent heat of evaporation (J/kg)
L_H	Heated length (m)
L_{SB}	Subcooled length (m)
q''	Surface heat flux (kW/m^2)
Q_E	Heat transfer rate based on electric power input (W)
R	Correlation coefficient
S	Channel circumference (m)
T_{sat}	Saturation temperature (°C)
T_w	Wall temperature (°C)

ΔT_{sat}	Averaged T_w - T_{sat} (°C)
x	Equilibrium mass quality; Eq. (2)
z	Distance along channel from subcooled test section inlet
η	Heat loss factor

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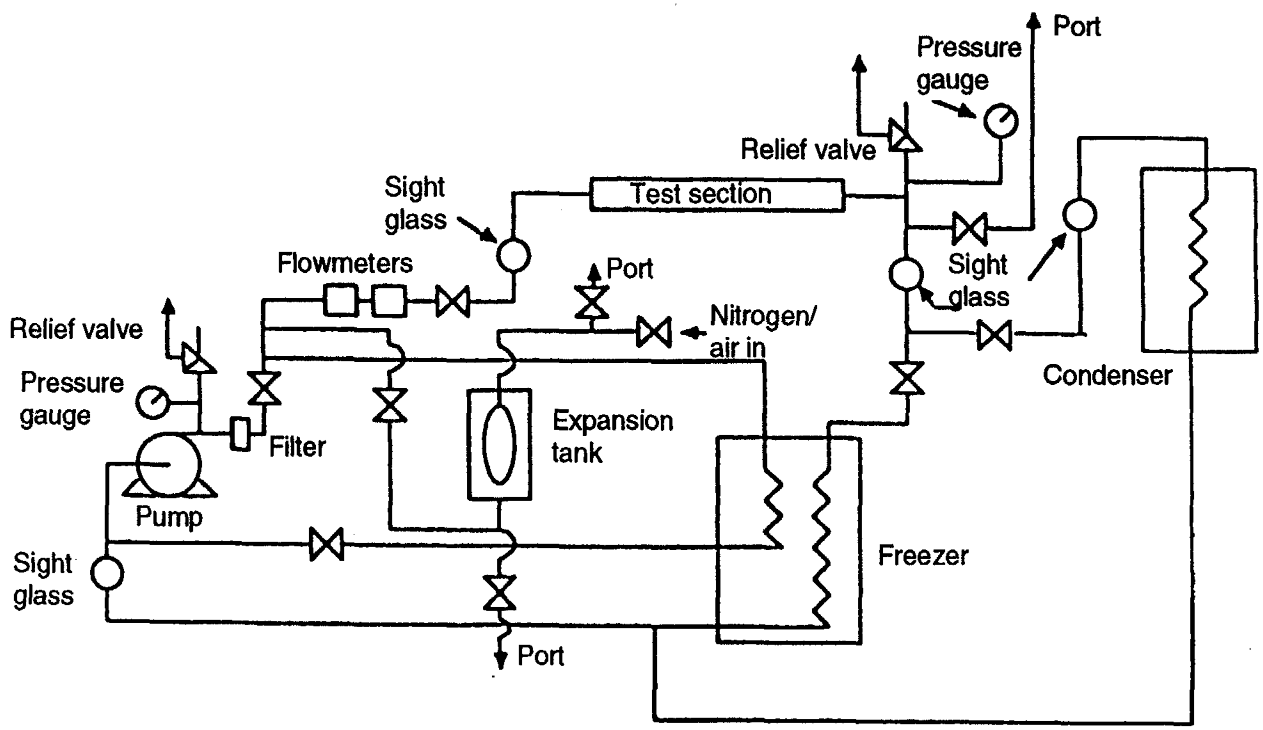

Figure 1. Schematic diagram of test apparatus

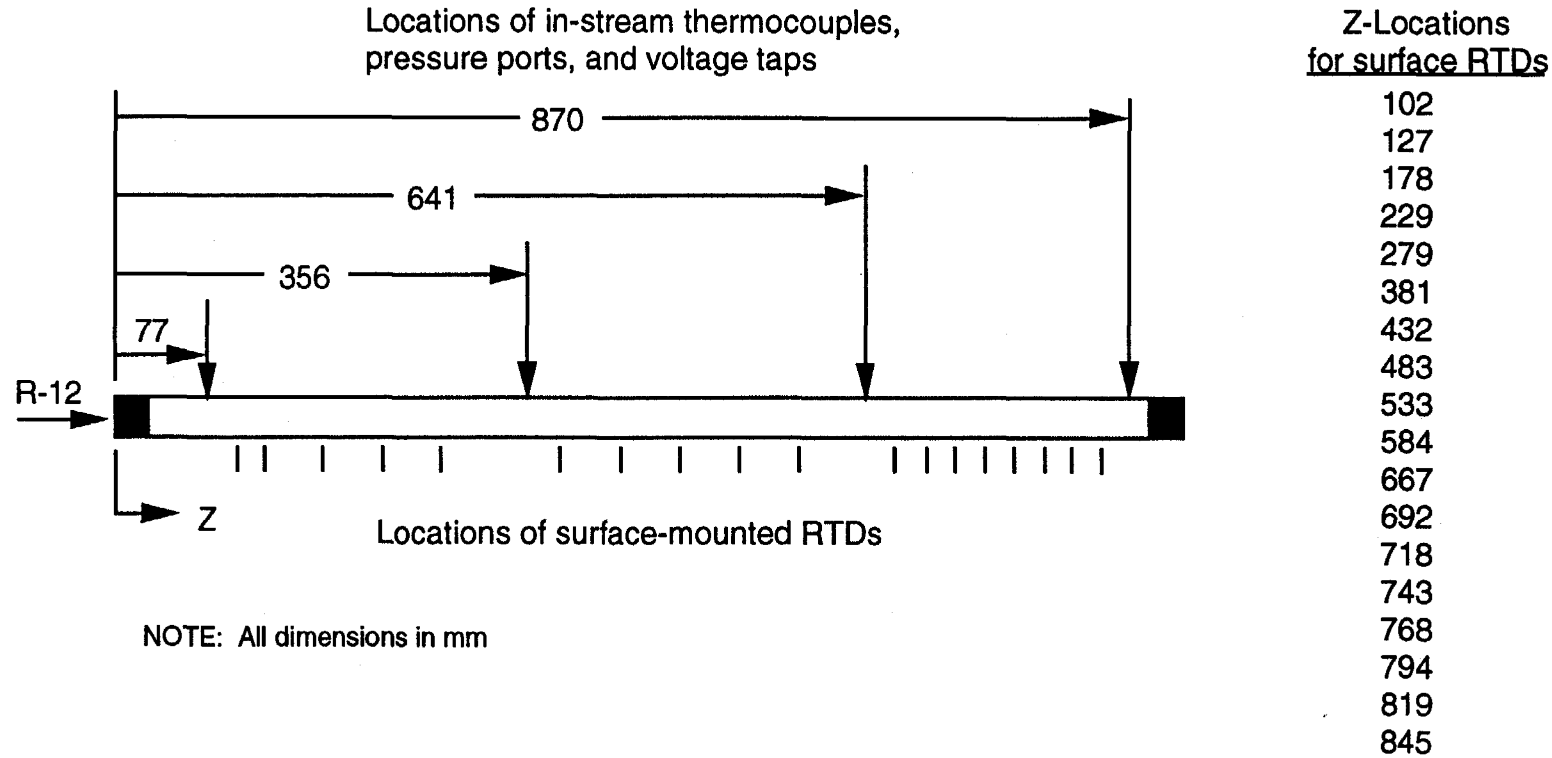

Figure 2. Test channel, showing locations of instrumentation

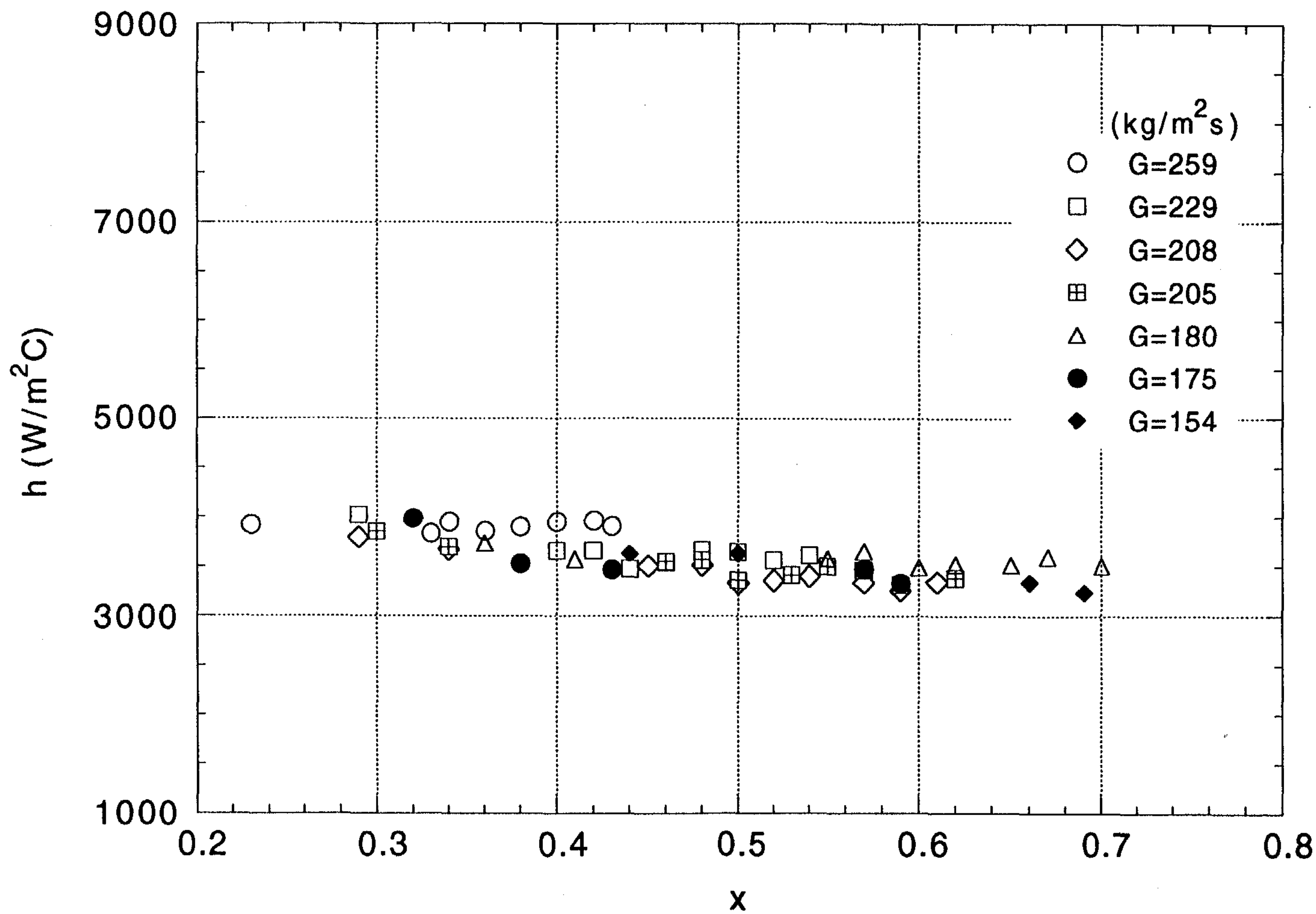

Figure 3. Local heat transfer results for range of mass flux G at approximately constant heat flux (14.5 kW/m^2)

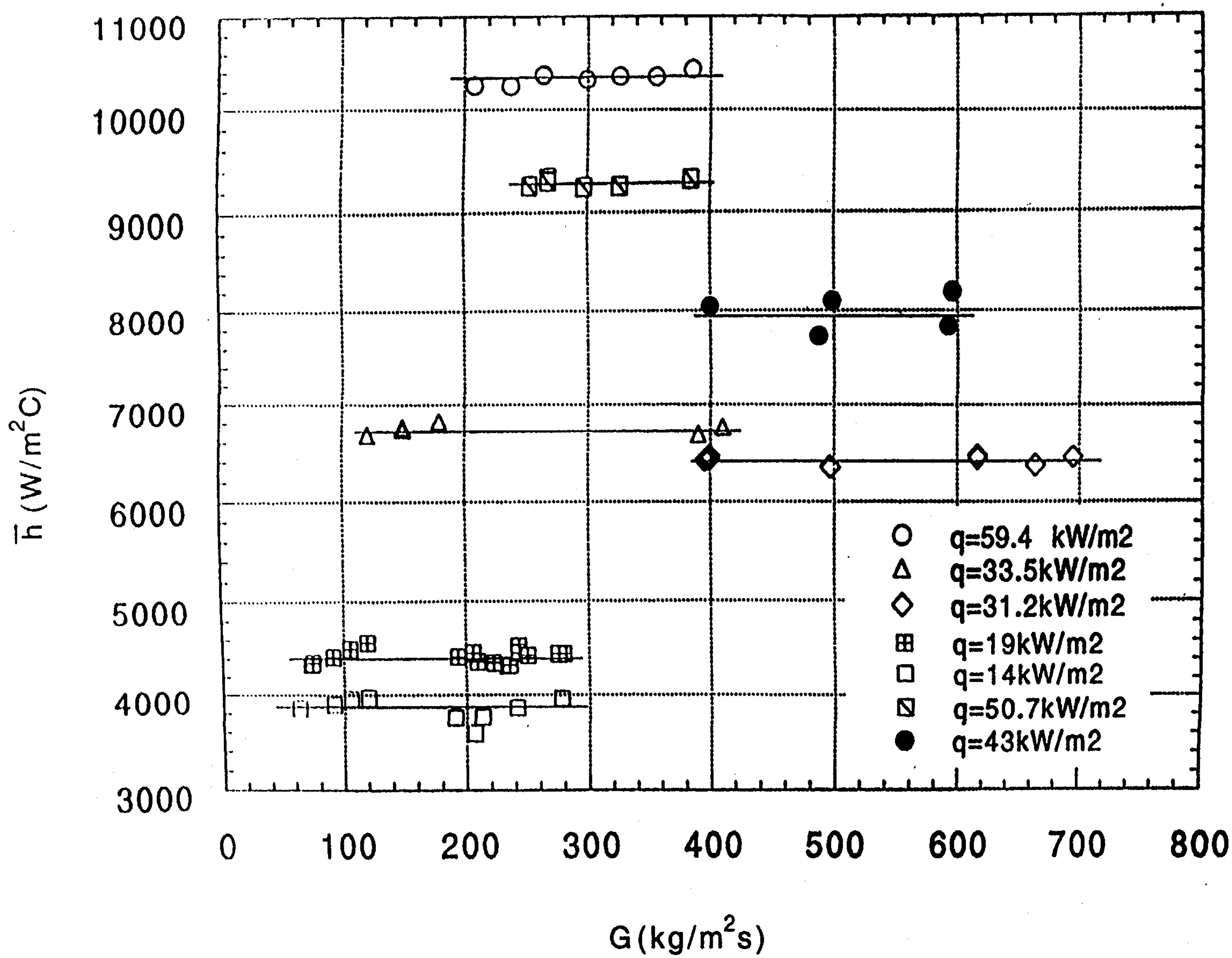

Figure 4. Average heat transfer coefficient as a function of mass flux G for select values of approximately constant heat flux q"

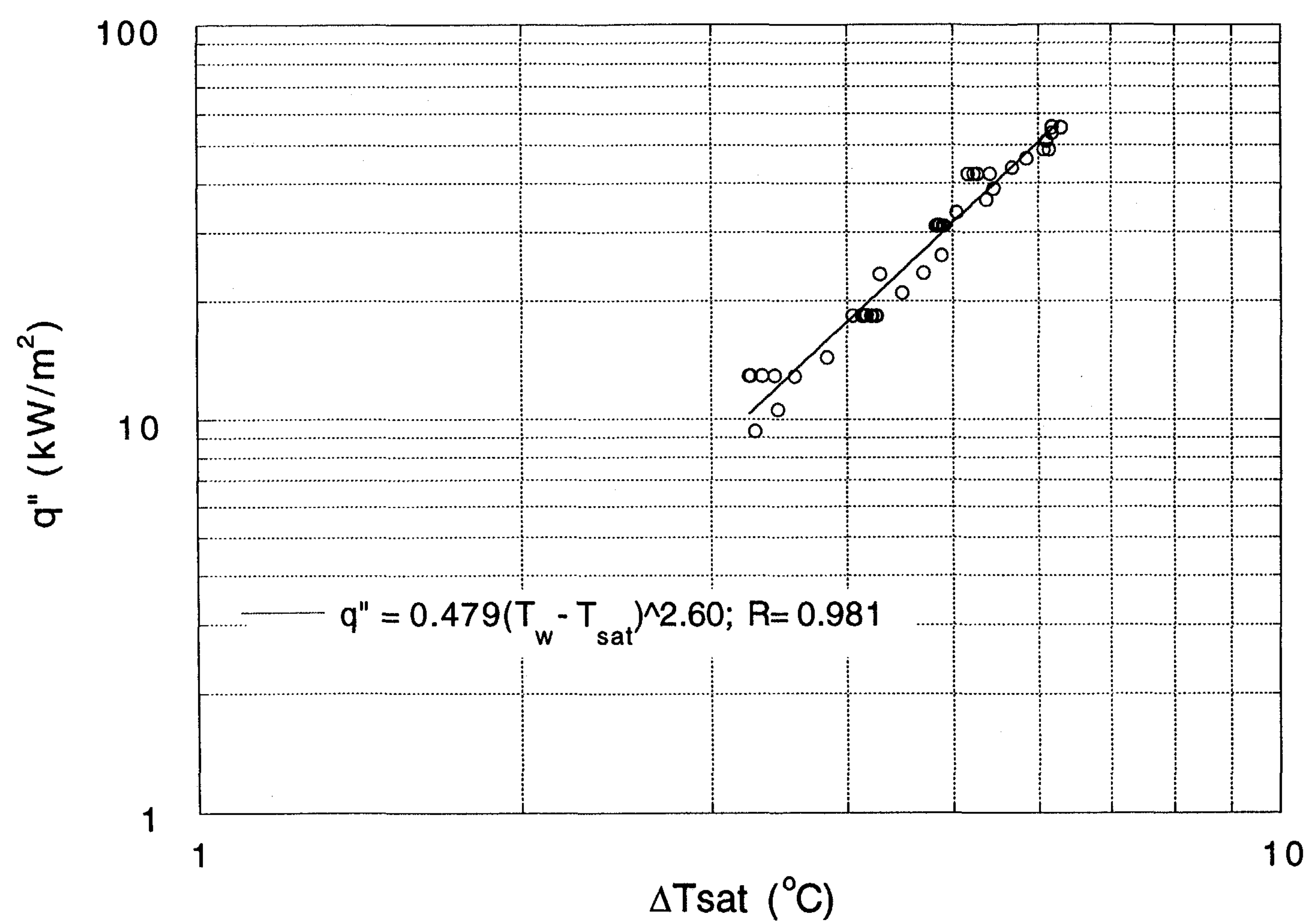

Figure 5. Heat flux q" dependence on average wall superheat ΔT_{sat}

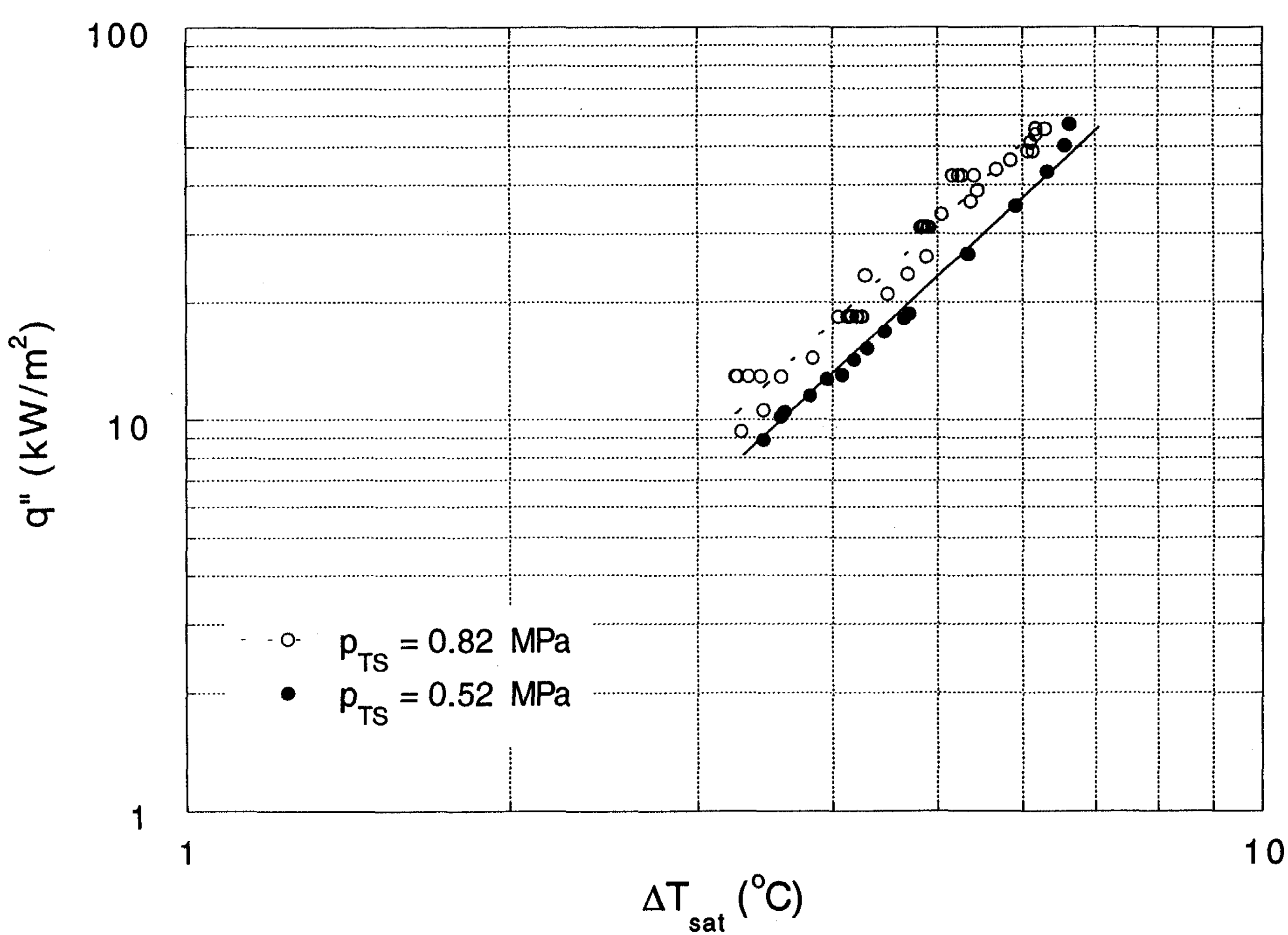

Figure 6. Effect of saturation pressure in fully developed nucleate boiling region; o - p_{sat} = 0.82 MPa; ● - p_{sat} = 0.52 MPa

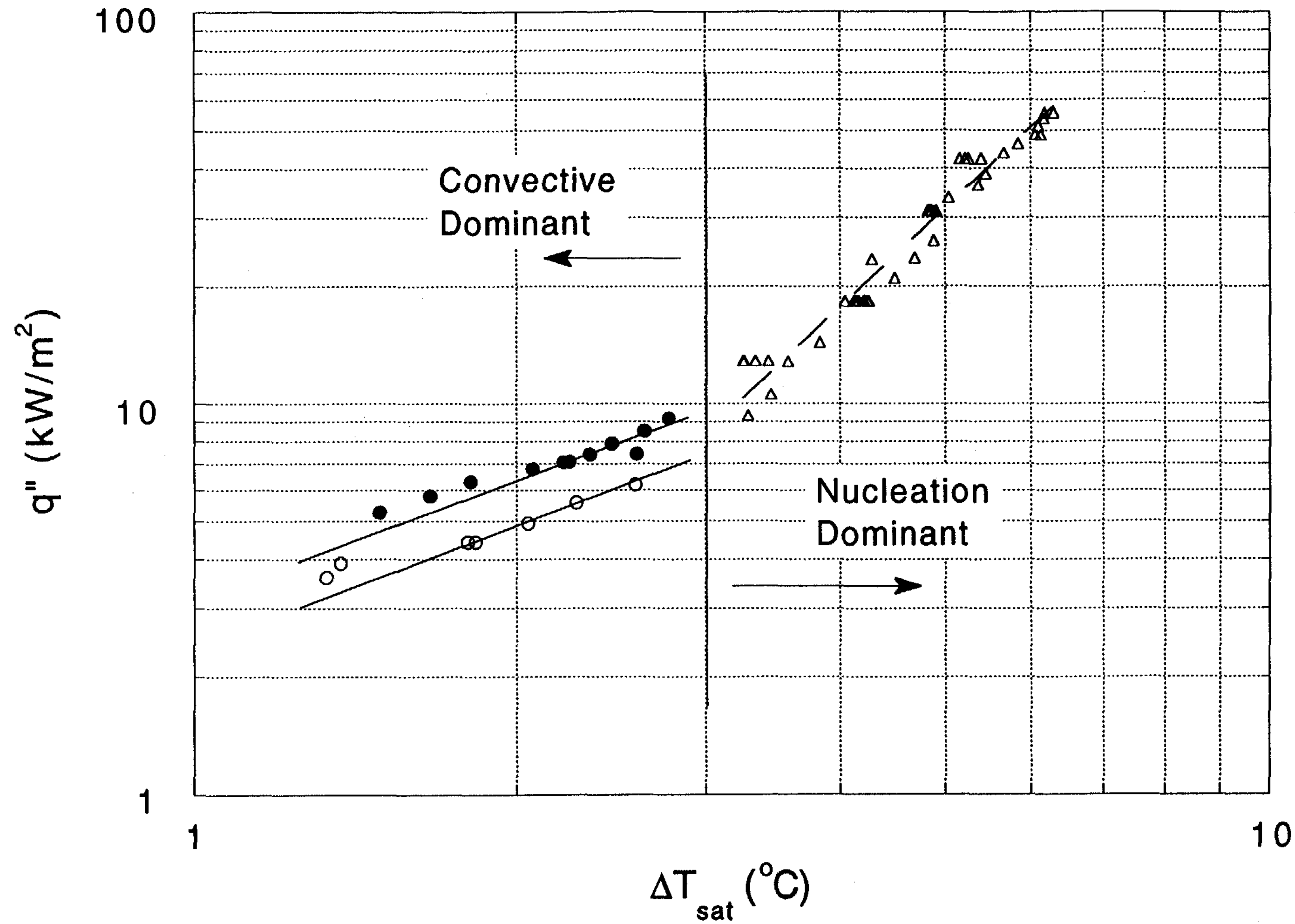

Figure 7. Heat flux q" dependence on average wall superheat ΔT_{sat}; convective boiling region: ○ - G = 75 kg/m^2s; ● - G = 150 kg/m^2s; ▲ - fully developed nucleate boiling region

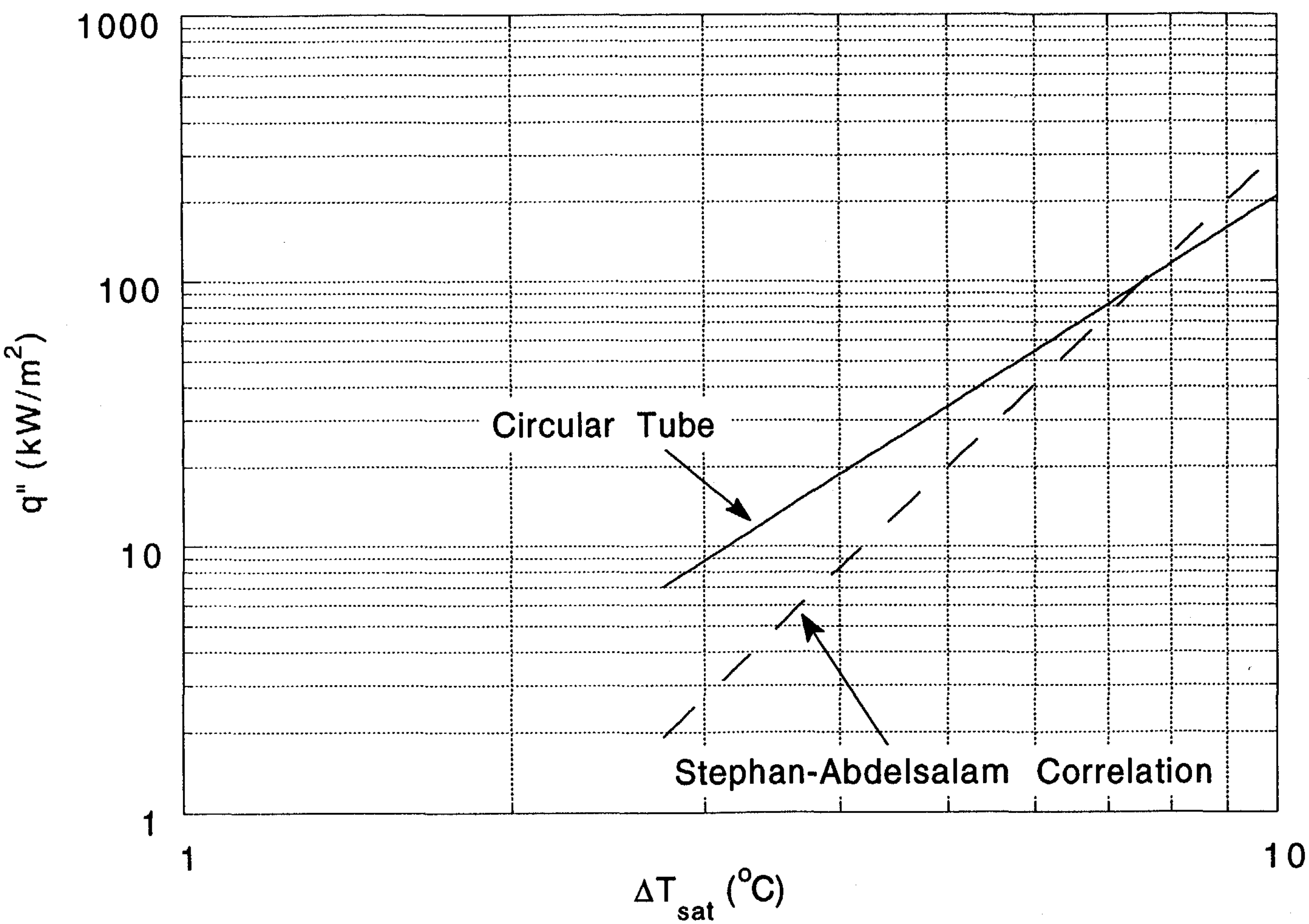

Figure 8. Heat transfer behavior of small circular channel and pool boiling prediction (Stephan and Abdelsalam 1980)

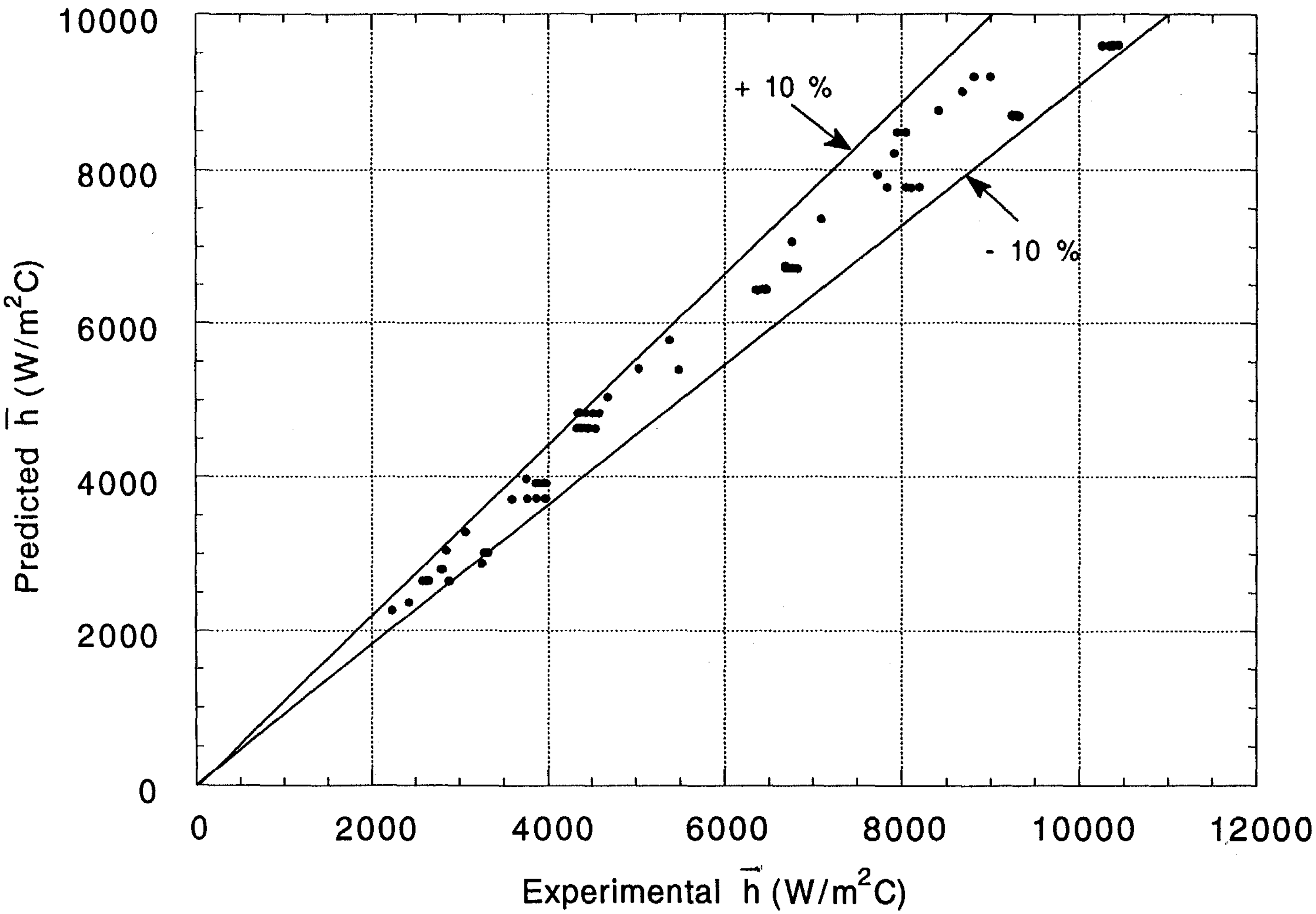

Figure 9. Heat transfer correlation of Eq. (4) for circular tube