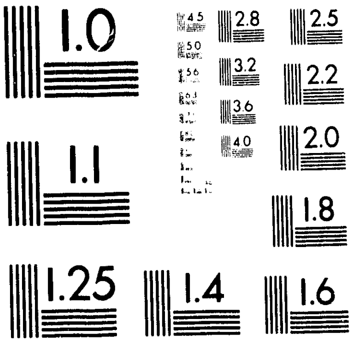
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BOILING HEAT TRANSFER IN A SMALL, HORIZONTAL, RECTANGULAR CHANNEL*

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Results of an experimental study are presented of heat transfer to the flow boiling of refrigerant-12 in a small, horizontal, rectangular channel. The channel height of 4.06 mm and width of 1.70 mm are representative of flow passages in compact heat exchangers. The flow channel material was brass with an overall length of 0.9 m. The channel wall was electrically heated, and thermocouples were installed on the channel wall as well as in the bulk fluid stream. End effects in small channel configurations can be appreciable, and the use of stream thermocouples eliminated the need to make temperature measurements near end fittings. Voltage taps were located at the same axial locations as the stream thermocouples to allow testing over a quality range of 0.15 to 0.8 and a large range of mass flux (50 to 400 kg/m^{2}s) and heat flux (4 to 34 kW/m^2). Saturation pressure was nearly constant during testing in the range of 760 to 950 kPa. Heat transfer coefficients were determined experimentally. Analysis of the data provided support for the conclusion, arrived at from a previous study of flow boiling in a small circular tube [1], that the nucleation mechanism dominates for flow boiling in small channels. The measured heat transfer coefficients were compared with

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predictions of correlation equations that were evaluated as part of the small circular tube study.

INTRODUCTION

Compact heat exchangers have traditionally found wide application in the transportation industry, where they are used as evaporators and condensers in vapor compression cycles for air conditioning and refrigeration. Such heat exchangers possess numerous attractive features including high thermal effectiveness, small size, low weight, design flexibility, and pure counterflow, and they can accommodate multiple streams. Today, there is a widespread interest in expanding the range of application of compact heat exchangers to include phase-change heat transfer in the process industries, among others. An overall objective of this effort is to provide the basis for establishing design technology in this area.

Compact heat exchangers are comprised of multichannel arrangements of small, typically noncircular, flow passages. Initial efforts of this work utilized a circular tube with a diameter of 2.92 mm which is representative of a compact heat exchanger flow passage [1]. Ten correlations were selected and evaluated for predicting the heat transfer in the small diameter tube [1]. The correlations evaluated were those of Chen [2], Chaddock and Brunemann (see Ref. 3), Pujol and Stenning [4], Shah [5], Stephan and Abdelsalam [6], Lazarek and Black [7], Liu and Winterton [8], Jung and Radermacher [9], Kandlikar [10], and Steiner and Taborek [11]. The ten correlations are presented in Table 2 of Ref. 1.

The heat transfer trends of the small diameter circular tube data indicated a dominance of the nucleation mechanism at all qualities tested which were as high as 60%. This result which is unlike results of experimental studies in larger tubes where the convective heat transfer mechanism typically dominates at qualities about 20% for refrigerants [9]. The small channel result was attributed to inherently high boiling numbers with small channels, and also to the fact that the slug flow pattern (based on flow pattern maps developed from air-water tests) occurred over a much larger parameter range than with larger tubes. As a consequence, the heat transfer correlations that predicted dominance of the nucleation mechanism, also predicted the data well if they also properly modeled the physical parameters. In general, the correlations of Lazarek and Black [7] and Stephan and Abdelsalam [6] predicted the small tube data very well where heat flux, and not mass flux, was the controlling parameter. However, neither correlation includes mass quality as a parameter.

In the present study, small channel flow boiling heat transfer was extended to a rectangular channel (4.06 x 1.70 mm) using refrigerant 12 (R-12). As with the circular tube studies, the flow channel wall was electrically heated providing a constant heat flux. Tests were performed over a quality range of 0.15 to 0.80, and large ranges of mass flux (50 to 400 kg/m^{2}s) and heat flux (4 to 34 kW/m^2). Heat transfer was measured and results are compared with correlation predictions.

EXPERIMENTAL APPARATUS AND TEST CHANNEL

The experimental apparatus is shown schematically in Fig. 1. Exiting the pump, the liquid refrigerant flows through a filter, an appropriate rotameter, a sight glass, and it enters the test section subcooled. The refrigerant is evaporated in the test channel to a quality of approximately 0.8 or less. The two-phase refrigerant leaving the test section then flows through a sight glass and is condensed in either the condenser or freezer, depending on the desired liquid temperature, before returning to the pump. A pump by-pass is provided to facilitate operation at low flow rates. System pressure is controlled using a high pressure nitrogen bottle with a pressure regulator and a bladder-type accumulator.

The flow channel is brass with an overall length of 0.9 m. The test channel is electrically resistance-heated by passing a DC current through the channel. A schematic of the test channel, showing the current clamps, and giving the locations of the various sensors (pressure, temperature, and voltage) is given in Fig. 2. Total electrical power supplied to the test channel was determined from a measurement of the overall voltage E between voltage taps and the current I as determined using a calibrated shunt resistor.

As shown in Fig. 2, in-flow temperature measurements of the bulk fluid temperature were made at four axial locations: the inlet and outlet, and in proximity to the two intermediate current clamps. Pressure ports and voltage taps also were provided at each of these four locations. Chromel-constantan thermocouples were used to make the temperature measurements. Inlet pressure was measured with a piezoresistive-type pressure transducer, and differential pressure between two pressure ports was measured with a strain-gauge-type transducer. Wall temperatures at selected axial locations along the length of the channel were measured with chromel-constantan thermocouples mounted to the surface with electrically-insulating, thermally-conductive epoxy; two or more thermocouples were installed at most axial locations (see Fig. 2).

The estimated uncertainty in measuring flow rate was ±3%. The pressure transducers were calibrated against a known standard; the estimated uncertainty in the pressure measurements was ±5%. Using isothermal tests, the thermocouple measurement systems were calibrated end-to-end, to include the multiplexer in the calibration, and correction equations were developed and input to the data acquisition system. It was estimated that the temperature measurements were accurate to within ±0.2°C.

TEST PROCEDURE AND DATA REDUCTION

As described subsequently, single-phase as well as flow boiling tests were performed in the test facility. In the performance of these tests, the establishment of steady-state conditions was verified by monitoring analog recordings of 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; this process took approximately 3 minutes. 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. In the establishment of test conditions, and in the subsequent analysis of the data, the fluid properties were determined from polynomial equations fit to R-12 fluid property data in the range of interest, as tabulated in the ASHRAE Handbook on Fundamentals [12].

Single-Phase Heat Transfer

Single-phase tests were performed and results used for (1) an overall system check of instrumentation, calibration, and data acquisition equipment and techniques, and (2) determining the heat loss to the environment. For a given test, single-phase heat transfer coefficients were computed from measured wall and bulk fluid temperatures, and input heat flux as determined from the enthalpy increase computed from measured inlet and outlet fluid temperatures. Single-phase pressure drop was also measured using the differential pressure transducer. The resulting heat transfer and pressure drop data were compared with correlations and data from other investigators. In the turbulent flow regime, the data were within -2% and +6% of the Dittus-Boelter prediction. These single-phase heat transfer data were also compared with data from a similar flow geometry (surface 4.00, p. 225 of Ref. 13), and the agreement was considered good. A comparison was made between isothermal, turbulent experimental friction factor and the Blasius prediction. Results were within 15 per cent.

In the single phase tests, the fluid enthalpy increase along the test section was measured via the inlet and exit stream thermocouples. Heat loss to the environment through 80 mm of insulation was determined by also measuring the input electrical power, Q_E. The heat loss factor η was then defined as

$$\eta = Q_T/Q_E \tag{1}$$

where

$$Q_T = \dot{m}c_p(T_{out} - T_{in}).$$

Despite substantial efforts to minimize the heat losses by insulating the test channel and associated piping, the heat losses determined from single-phase tests were significant, with η typically less than 0.8. This result was attributed to the fact that the heat loss problem is exacerbated in small channels because of the high heat transfer area density ratio (surface area divided by fluid volume) inherent in such channels. From the single-phase tests a functional relationship was developed for this heat loss which was then used to calculate the heat loss in the flow boiling experiments.

Flow Boiling Heat Transfer

The test procedure followed in the flow boiling experiments involved specifying a mass flux and heat flux or exit quality. These variables were chosen to achieve results with any one or two variables constant while the remainder varied in a sequence of tests. This procedure facilitated subsequent interpretation of the data and trends. As discussed previously, after steady state was achieved in a test, the sensor-output voltages were scanned 30 times and averaged using the computer-controlled Hewlett Packard data acquisition system. The results were than changed to engineering units in the computer and stored on computer disk for future processing.

Prior to data processing and analysis, multiple wall temperatures at each axial location were compared to detect any variance among the different surface temperatures measured. The purposes of making the comparisons were to identify trends that may be associated with flow patterns, and to identify possible malfunctioning instrumentation. In general, it was not possible to identify any flow-pattern-associated trends and the temperatures at a given cross-section typically agreed within 0.3°C. Consequently, once it was confirmed that all thermocouples were responding properly, the wall temperatures measured at a given axial location were averaged to obtain a single wall temperature value for that location. This detailed inspection throughout the tests ensured that an erroneous thermocouple reading was not incorporated into the data. Such readings although not frequently encountered were most likely to come from degradation in thermocouple contact with the heated surface due to thermal cycling between tests.

The temperature of the two-phase fluid at the axial locations of the wall temperature measurements were determined from the measured inlet and outlet temperatures and the measured two-phase pressure drop; it was determined that the inlet pressure measurement was not sufficiently accurate to use as one of the bases for the data analysis. The procedure followed was to determine the fluid saturation pressure at the outlet p_{out} from the measured outlet saturation temperature. Then, assuming the two-phase pressure drop to be linear (the test section pressure drop was small, typically less than 15 kPa), three equations were written relating (1) the test channel pressure at the start of boiling to the length of channel with subcooled liquid, (2) the length of test channel with subcooled liquid to the test channel temperature at the start of boiling, and (3) the pressure at the start of boiling to the temperature at the start of boiling. These three equations with three unknowns were solved for the two unknowns of interest, viz., the length of test channel with subcooled liquid L_{SB}, and the pressure at the start of boiling p_{SB}. Then, interpolation was used to obtain the saturation pressure at the wall temperature measurement locations, and the corresponding saturation temperatures were determined from the saturation temperature/ pressure relationship for R-12.

The local heat transfer coefficient was calculated as

$$h(z) = \frac{q''_T}{\left[T_w(z) - T_{sat}(z)\right]} \tag{2}$$

where

$$q''_T = \frac{\eta Q_E}{S(L_H - L_{SB})}$$

and

$$S = 2(a + b).$$

The quality at measurement location z was calculated as

$$x(z) = \frac{S(z - L_{SB})q''_T}{AGi_{fg}}. \quad (3)$$

Results from a typical test run, illustrating the calculational scheme, are shown in Fig. 3.

The experimental runs are summarized in Table 1. A total of 36 test runs are reported including 19 independent test runs, and 17 test runs which were a repeat of previous runs. The following parameter ranges were covered:

mass flux:	54 - 396 kg/m^2s;
heat flux:	4.1 - 33.7 kW/m^2;
mass quality:	15 - 76 %;
saturation pressure:	758 - 945 kPa.

RESULTS AND DISCUSSION

Following the procedures outlined, and as illustrated in Fig. 3, local heat transfer coefficients were computed from the measured wall temperatures and calculated fluid saturation temperatures and heat flux.

Typical results for heat transfer coefficient are presented in Fig. 4 over the full range mass flux and heat flux of the experiments. Repeatability tests were performed at different times or days. In general, excellent repeatability was demonstrated as illustrated by the results given in Fig. 5.

Chen [2] presented the concept of superposition of nucleation and forced convection heat transfer mechanisms in predicting the heat transfer coefficient in flow boiling upstream of CHF. Many correlations, developed subsequently with large data bases and/or including many different fluids, have followed this approach. For example, the correlations of Kandlikar [10] and Steiner and Taborek [11] contain two terms each, one for nucleation and one for convection, with appropriate suppression and enhancement factors to model the transition. These types of correlations have been derived from data on relatively large diameter pipes—diameters greater than 6 mm—and have been shown to predict the measurements reasonably well.

In the study of flow boiling of refrigerant R-113 in a small circular tube (2.92-mm diameter) [1], it was concluded, on the basis of data trends and a detailed evaluation of in-tube evaporation correlations, that the nucleation mechanism dominated over the range of qualities tested. In particular, the measured heat transfer coefficient was shown to depend primarily on heat flux with only a weak dependence on mass flux. Further, it was demonstrated that the correlations that predicted a dominance of the nucleation mechanism also best predicted the data.

Based on these results, the potential for nucleation dominance was investigated in the small rectangular channel of this study. Series of tests were performed, first over a range of mass flux at constant heat flux, and, secondly, over a range of heat flux at constant mass flux. Representative results are given in Figs. 6 and 7, for the cases of constant heat flux and constant mass flux, respectively. The results given in Figs. 6 and 7, indicate a strong heat flux dependence and only a weak mass flux dependence similar to the results of the circular tube study with R-113. These results indicate that the dominant heat transfer mechanism is nucleation in small passages – both circular and rectangular.

The circular tube data [1] were used to evaluate 10 selected in-tube evaporation correlations. The form and basis of the various correlations, together with information on the accuracy to which they predicted the data, were subsequently used to interpret the data relative to the heat transfer mechanism. These same 10 correlations were applied to the small rectangular channel data of this study. Typical results, corresponding to two different test runs, are given in Fig. 8. With the exception of the Chen [2] and Steiner and Taborek [11] correlations, both of which significantly overpredicted the data and, for that reason, were not included in Fig. 8, all of the correlations underpredicted the data. This is in contrast to the circular tube evaluation in which several of the correlations (in particular, Stephan and Abdelsalam [6], Lazarek and Black [7], and Liu and Winterton [8]) predicted the data well (mean deviation <15 per cent), and many of the correlations were centered in the data. Since the hydraulic diameters of the two channels were selected to be comparable, this difference in behavior between the circular and rectangular channels suggests a geometry effect. Neglecting the fact that the Kandlikar correlation significantly underpredicts the data, it is of interest to note that for the higher values of mass flux this correlation predicts the data trend of a sharp decrease in heat transfer coefficient at low values of quality; see, for example, Fig. 8b. The Stephen and Abdelsalam [6] and Lazarek and Black [7] correlations correlated well the circular tube data and are in close agreement with each other for several of the rectangular channel test runs; however, neither correlation includes a function of quality as seen in the data (Fig. 4). (See Appendix for a discussion of the relationship between these two correlations.)

In consideration of the evaluation of the various correlations for in-tube evaporation, it should be noted that the correlations were developed to predict heat transfer in larger diameter circular tubes. Also, many of the correlations include a term involving the single-phase heat transfer coefficient associated with the two-phase mixture flowing as a liquid. In small channel boiling, the flow regime associated with the two-phase mixture flowing as a liquid is typically laminar. It should be noted that these correlations were applied as developed using the turbulent form for single-phase heat transfer coefficient.

Nucleate boiling data is often represented by curves of heat flux as a function of wall superheat. Wadekar [14] and Robertson and Wadekar [15] have used this representation to separate purely convective heat transfer data from data containing nucleate boiling. An illustration of this type of plot is given in Fig. 9; the curves in Fig. 9 are for different values of mass flux. Following Wadekar [14], a line A-A was drawn to separate the two regions. The curves in the region to the left of the line A-A have slopes of unity indicating that the heat flux is proportional to wall superheat; that is, the heat transfer coefficient is independent of heat flux which is characteristic of convective heat transfer. To the right of line A-A, the curves are shown to be independent of mass flux with a slope greater than unity implying nucleate boiling domination.

El-Sallak et al. [16] performed an experimental investigation of the boiling flow characteristics of refrigerants R-11 and R-21 in a 10 mm diameter horizontal tube. They presented their data in the form of heat flux as a function of wall superheat for constant values of mass flux. For a given mass flux their data could be correlated, on a log-log representation, with a straight line of slope equal to approximately two. The data were shown to be a strong function of mass flux, as the data corresponding to different values of mass flux were well separated. These results serve to demonstrate the condition in larger diameter tubes in which both mechanisms, nucleate and convective, contribute significantly to the heat transfer.

In view of the results [14-16] discussed, the flow boiling data of this study were presented in the format of surface heat flux as a function of wall superheat. In Figs. 10a-10f, representing thermocouple locations b-g, respectively, these results are given for a series of tests with mass fluxes in the range 54 to 396 kg/m^{2}s. It is of interest to observe, first, that the correlations are approximately independent of mass flux (some of the apparent scatter may be attributed to experimental error) and, secondly, that the data correlate reasonably well with a straight line on the log-log plot. The first observation provides support for the conclusion that nucleation is the dominant heat transfer mechanism. The second observation suggests that a power function relationship exists between heat flux and wall superheat; this is in agreement with the form of the Stephan and Abdelsalam correlation.

The simple correlation of Stephan and Abdelsalam [6] performed well in correlating the small circular tube data [1]. The Stephan and Abdelsalam correlation is given as

$$h = c_4 q''^{0.745}, \tag{4}$$

where $c_4 = 2.5$ for refrigerant R-12. Equations (2) and (4) were combined and plotted in Figs. 10a-10f, where the results can be compared with the small rectangular channel data. While the correlation does not fit the data over the entire range, the plots indicate that a correlation of the Stephan and Abdelsalam form, that is,

$$h = C_1 q_T''^{C_2}, \tag{5}$$

is appropriate where coefficients C_1 and C_2 may be functions of quality. It is of interest to note from the plots of Fig. 10 that the data tend to approach the Stephan and Abdelsalam correlation as the quality increases (see Fig. 10f).

The measured heat transfer coefficients of this study could not be directly compared with the heat transfer coefficients from the small circular tube study [1] because the fluids were different in the two cases; R-12 was used in the small rectangular channel study, while R-113 was used in the small circular tube study. However, it should be noted that while many of the in-tube evaporation correlations (most notably, Lazarek and Black [7], Stephan and Abdelsalam [6], and Liu and Winterton [8]) did reasonably well in predicting the small circular tube data (mean deviation less than 16%), these same correlations significantly underpredicted the small rectangular channel data. This result indirectly implies that the heat transfer to flow boiling is better in a small rectangular channel of comparable hydraulic diameter than in a circular tube.

In Ref. 1, the dominance of the nucleation mechanism for flow boiling in small channels was attributed to two factors: (1) inherently high boiling numbers for such channels, and (2) the fact that, based on available flow pattern maps for small tubes, the slug flow pattern was determined to extend to high qualities. These flow patterns require further study for the channel and fluid of this investigation, and an alternative explanation for the dominance of the nucleation mechanism and the apparent higher heat transfer with the rectangular channel may be attributed to nucleate boiling in thin films. Nucleation in thin film heat transfer gives rise to high heat fluxes as noted by Marto and Anderson [17], and such nucleation has been presented in other studies, e.g., Jensen and Bensler [18] and Mesler [19, 20]. In small rectangular channels in particular, thinning of the liquid film in the higher quality region is expected on the "flats" due to surface tension causing liquid to remain in the corners. The data indicate nucleation in both circular and rectangular channels at high quality but thinner liquid films in the rectangular channel due to surface tension forces is the likely mechanism responsible for the increased heat transfer observed.

SUMMARY AND CONCLUSIONS

An experimental study of the heat transfer to the flow boiling of refrigerant R-12 in a small, horizontal, rectangular channel is reported. The data were used to evaluate state-of-the-art in-tube evaporation correlations. The applicability of the various correlations, together with the basis for their derivations, were used in analyzing the data and in comparing results obtained from previous experiments with R-113 in a small diameter tube. Representation of the data in plots of surface heat flux as a function of wall superheat provided further insights to the mechanism and a possible form for a correlation equation. Findings and conclusions derived from the study are as follow:

(1) The results of the study provide further support to the conclusion that the nucleation mechanism dominates in flow boiling in channels of small cross-sectional area. The heat transfer coefficients were shown to be a strong function of heat flux and only weakly dependent on mass flux over the large range of heat and mass fluxes tested.

(2) State-of-the-art correlations for in-tube evaporation that did well in correlating the small circular tube data, with the correlations well centered in the data [1], were shown to significantly underpredict the small rectangular channel data. This result led to the conclusion that heat transfer is enhanced in the small rectangular channel, most likely, as a result of surface tension effects in the small rectangular geometry which act to thin the film on walls.

(3) From plots of surface heat flux versus wall superheat, it was shown that a correlation of the Stephan and Abdelsalam form, that is,

$$h = C_1 {q''_T}^{C_2}, \tag{5}$$

is appropriate for correlating small channel flow boiling data. In the correlation, the coefficients C1 and C2 should be a function of quality (and geometry) as well as fluid properties.

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APPENDIX

Relationship between the Stephan and Abdelsalam [6] and Lazarek and Black [7] Correlations

The Stephan and Abdelsalam [6] and Lazarek and Black [7] correlations best predicted the circular tube data of the previous study [1]. While they both underpredict the data from the rectangular channel data, they, nevertheless, were in reasonable agreement with each other for many of the test runs. The two correlations were considered as follows.

The Lazarek and Black correlation is nondimensional and is given by

$$Nu = 30\ Bo^{0.714} Re_L^{0.857}, \qquad (A1)$$

where

$$Nu = \frac{hd_h}{k_L},$$

and a size effect is included through the use of the Nusselt number.

The Stephan and Abdelsalam correlation for refrigerants is typically given in the dimensional form

$$h = c_4 q''^{0.745}, \qquad (A2)$$

where h has dimensions W/m^2C and q" has dimensions W/m^2. If q" is nondimensionalized as $(q''d_h c_p/k_L i_{fg})$, Eq. A2 can be written in the form

$$Nu = \left(\frac{c_4 d_h}{k_L}\right)(Bo\, Re_L\, Pr_L)^{0.745}, \tag{A3}$$

To facilitate comparison with the Lazarek and Black correlation Eq. A3 can be written as

$$Nu = CBo^{0.745}Re_L{}^{0.745}, \tag{A4}$$

where

$$C = \left(c_4\, Pr_L^{0.745}\right)\left(\frac{d_h}{k_L}\right).$$

A comparison of Eqs. A1 and A4 shows that the two correlations are of the same form. The exponents on the boiling and Reynolds numbers are similar. Thus, it is not surprising that these two correlations produce similar prediction of the data. However, the Lazarek and Black correlation includes a size effect (d_h) and a weak mass flux effect ($G^{0.143}$) which are not included in the Stephan and Abdelsalam correlation (Eq. A2). Each includes a fluid property coefficient, although different in the two correlations.

Nomenclature

a	Channel height (m)
A	Channel cross-sectional flow area (m^2)
b	Channel width (m)
c_4	Property dependent coefficient in Stephan and Abdelsalam correlation, see Eq. (4)
C_1	Coefficient in Eq. (5)
C_2	Coefficient in Eq. (5)
c_p	Specific heat of liquid (J/kg C)
E	Voltage across heated length (V)
G	Mass flux (kg/m^2s)
h	Local heat transfer coefficient (W/m^2C); Eq. (2)
i_{fg}	Latent heat of evaporation (J/kg)
I	Electric current through test channel (A)
L_H	Heated length (m)
L_{SB}	Subcooled length (m)
$\dot{m}$	Mass flow rate (kg/s)
p_{in}	Pressure at inlet to test channel (kPa)
p_{out}	Saturation pressure at outlet form heated length (kPa)
q''_T	Surface heat flux (W/m^2); Eq. (2)
Q_E	Heat transfer rate based on electric power input (W); Eq. (1)

Q_T Heat transfer rate based on enthalpy increase (W); Eq. (1)

S Channel circumference (m); Eq. (2)

T_{in} Fluid temperature at the inlet to the heated length (C)

T_{out} Fluid temperature at the outlet from the heated length (C)

T_{sat} Saturation temperature (C)

T_w Wall temperature (C)

x Equilibrium mass quality; Eq. (3)

z Distance along channel from start of boiling

η Heat loss factor; Eq. (1)

Tables

Table 1. Summary of test runs

Mass Flux (G) (kg/m2-s)	Heat Flux (kW/m2)	Pin (kPa)	Approx. Exit Quality
54.6	4.1	840.8	0.8
63.2	4.8	838.4	0.8
71.8	5.8	838.6	0.8
88.5	6.9	841.2	0.8
104.7	8.0	845.1	0.8
128.1	10.7	841.5	0.8
150.4	12.6	849.4	0.8
171.5	14.7	841.5	0.8
204.4	17.7	850.7	0.8
204.4	17.8	947.4	0.8
204.5	17.6	849.9	0.8
204.5	17.5	749.3	0.8
204.5	17.5	757.1	0.8
204.6	17.4	783.4	0.8
216.3	18.8	845.6	0.8
216.8	18.7	846.3	0.8
277.2	23.1	813.0	0.8
277.4	23.2	850.6	0.8
277.4	23.1	812.1	0.8
277.5	23.1	773.3	0.8
277.5	23.0	808.4	0.8
277.6	23.2	836.8	0.8
336.4	27.6	853.3	0.8
395.8	33.7	865.5	0.8
104.6	6.9	838.4	0.6
104.8	7.0	837.1	0.6
153.3	10.2	835.5	0.6
204.5	14.0	839.0	0.6
277.4	17.8	850.2	0.6
277.4	17.8	846.7	0.6
277.9	17.7	823.1	0.6
104.7	5.2	842.0	0.4
104.8	5.2	831.3	0.4
150.4	7.5	836.8	0.4
204.5	10.2	841.1	0.4
278.1	13.0	819.0	0.4

Figures

1 Schematic diagram of heat transfer test apparatus

2 Test channel, showing locations of instrumentation

3 Data reduction scheme: $G = 204.4\ kg/m^2s$, $q''_T = 17.7\ kW/m^2$, $p_{in} = 783\ kPa$

4 Typical heat transfer coefficients for the experimental range of mass and heat flux

5 Experiment repeatability: Results from three independent test runs with $G = 204.5\ kg/m^2s$, and $q''_T = 17.5\ kW/m^2$

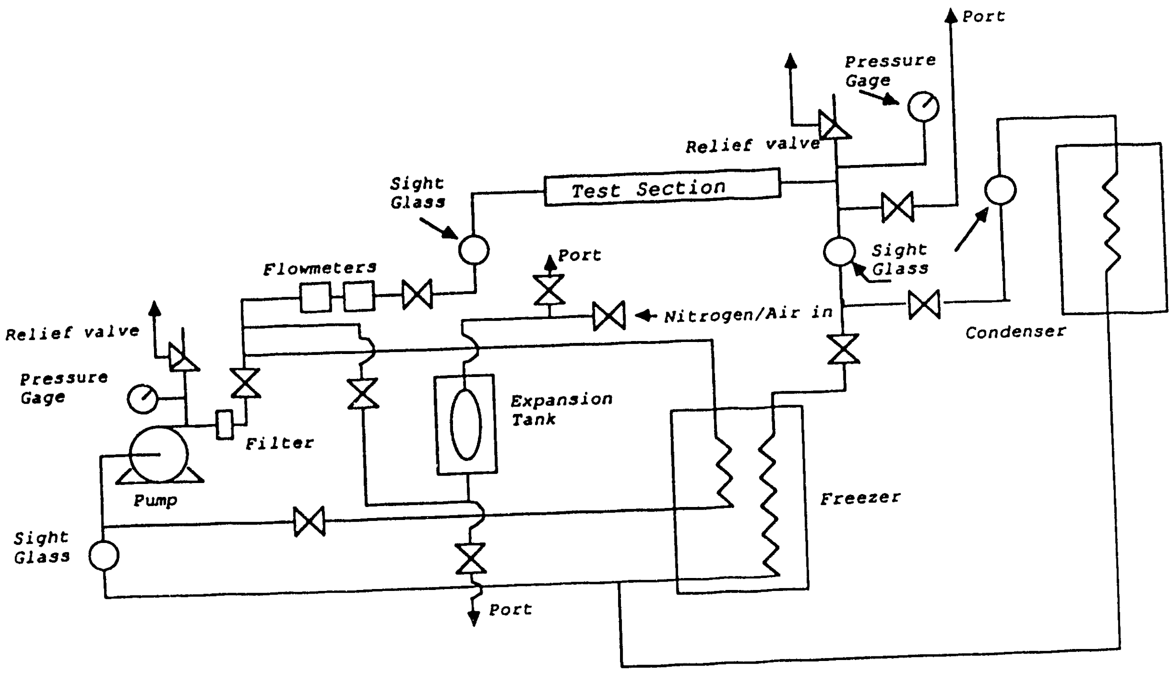
6 Mass flux effect

7 Heat flux effect

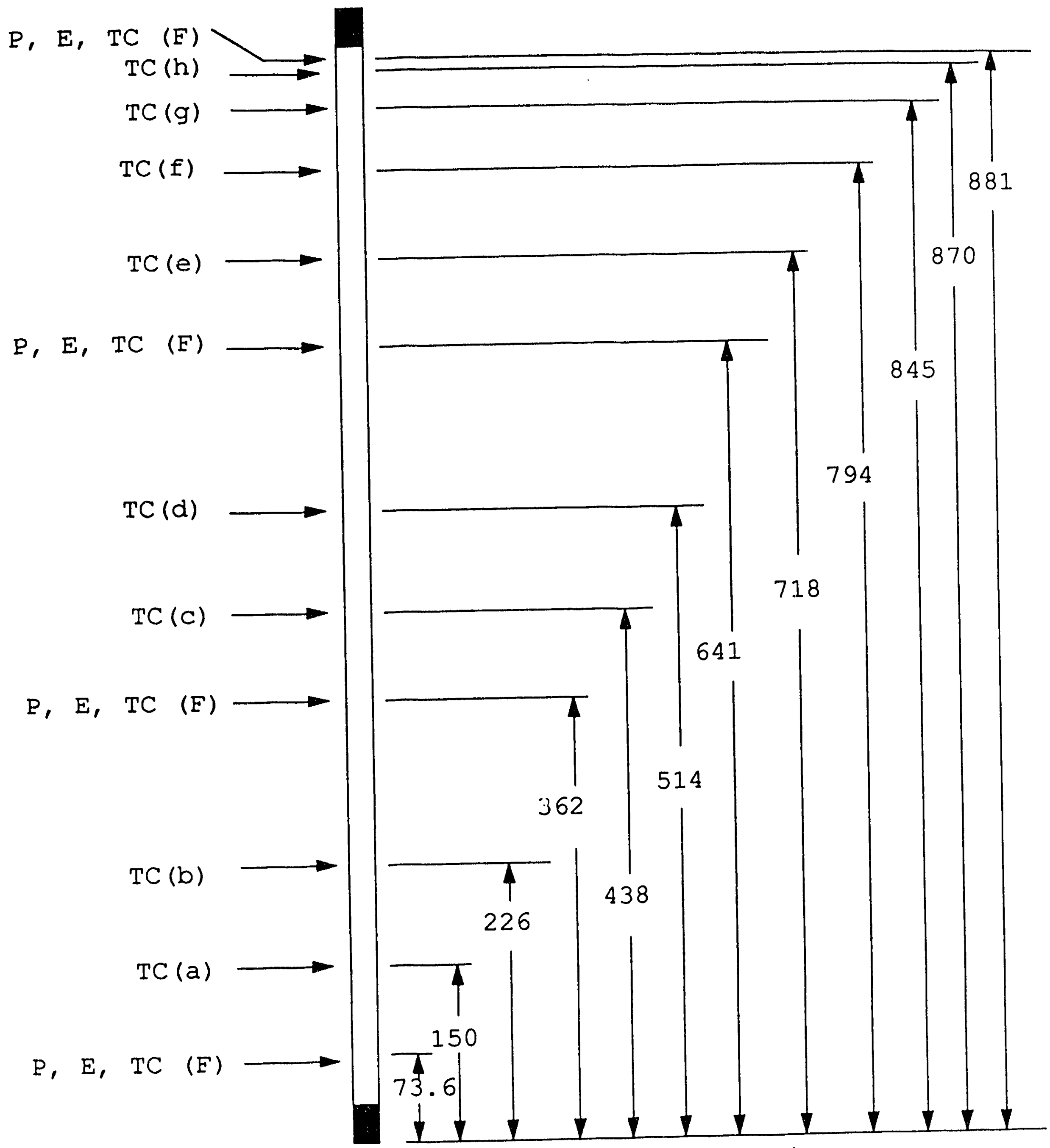
8 Correlation comparisons: (a) $G = 54.6\ kg/m^2s$, $q''_T = 4.1\ kW/m^2$; (b) $G = 204.4\ kg/m^2s$, $q''_T = 17.7\ kW/m^2$: (--•--) Experimental data; a - Chaddock and Brunemann (see Ref. 3); b - Pujol and Stenning [4]; c - Stephan and Abdelsalam [6]; d - Lazarek and Black [7]; e - Liu and Winterton [8]; f - Jung and Radermacher [9]; g - Kandlikar [10]; h - Shah [5]

9 Schematic diagram of surface heat flux as a function of wall superheat for various values of mass flux

10 Surface heat flux as a function of wall superheat; (a) thermocouple location b, (b) thermocouple location c, (c) thermocouple location d, (d) thermocouple location e, (e) thermocouple location f, and (f) thermocouple location g


Port
Pressure
Gage
Relief valve
Test Section
Sight
Glass
Sight
Glass
Port
Flowmeters
Nitrogen/Air in
Condenser
Relief valve
Pressure
Gage
Expansion
Tank
Filter
Pump
Freezer
Sight
Glass
Port

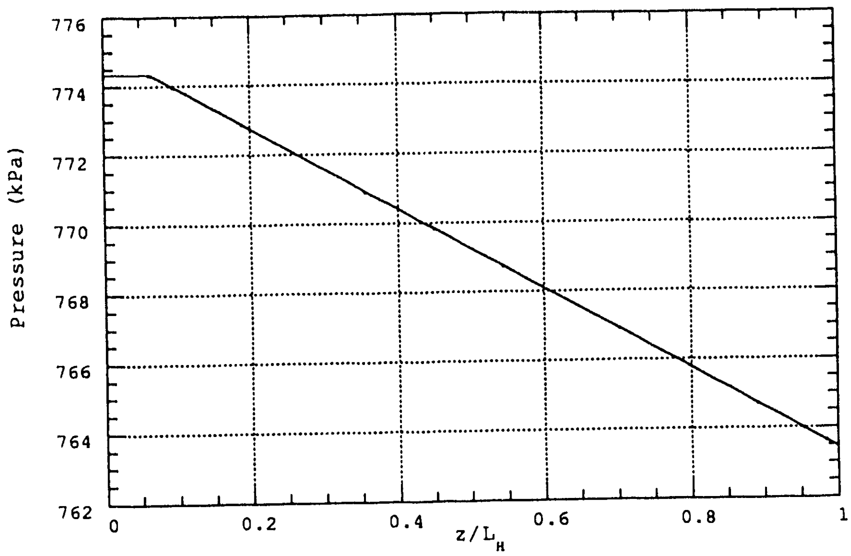
l



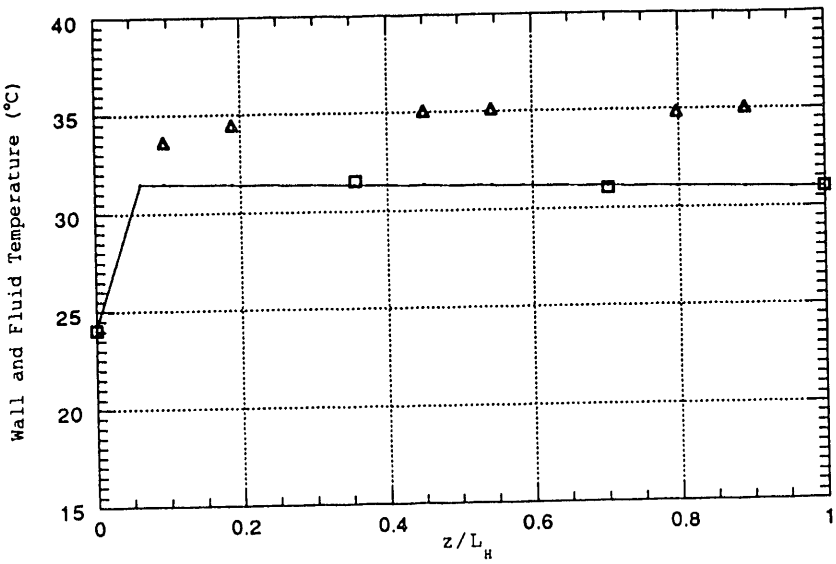
NOTE: All Dimensions in mm.

Nomenclature:

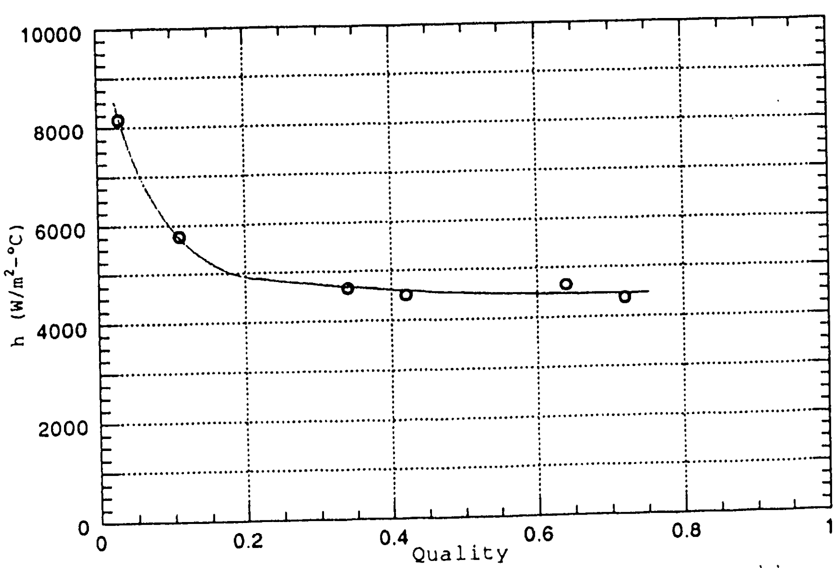
a) P - Pressure port
b) TC - Thermocouple
c) a, b, c...h - TC locations
d) F - In flow
e) E - Voltage taps



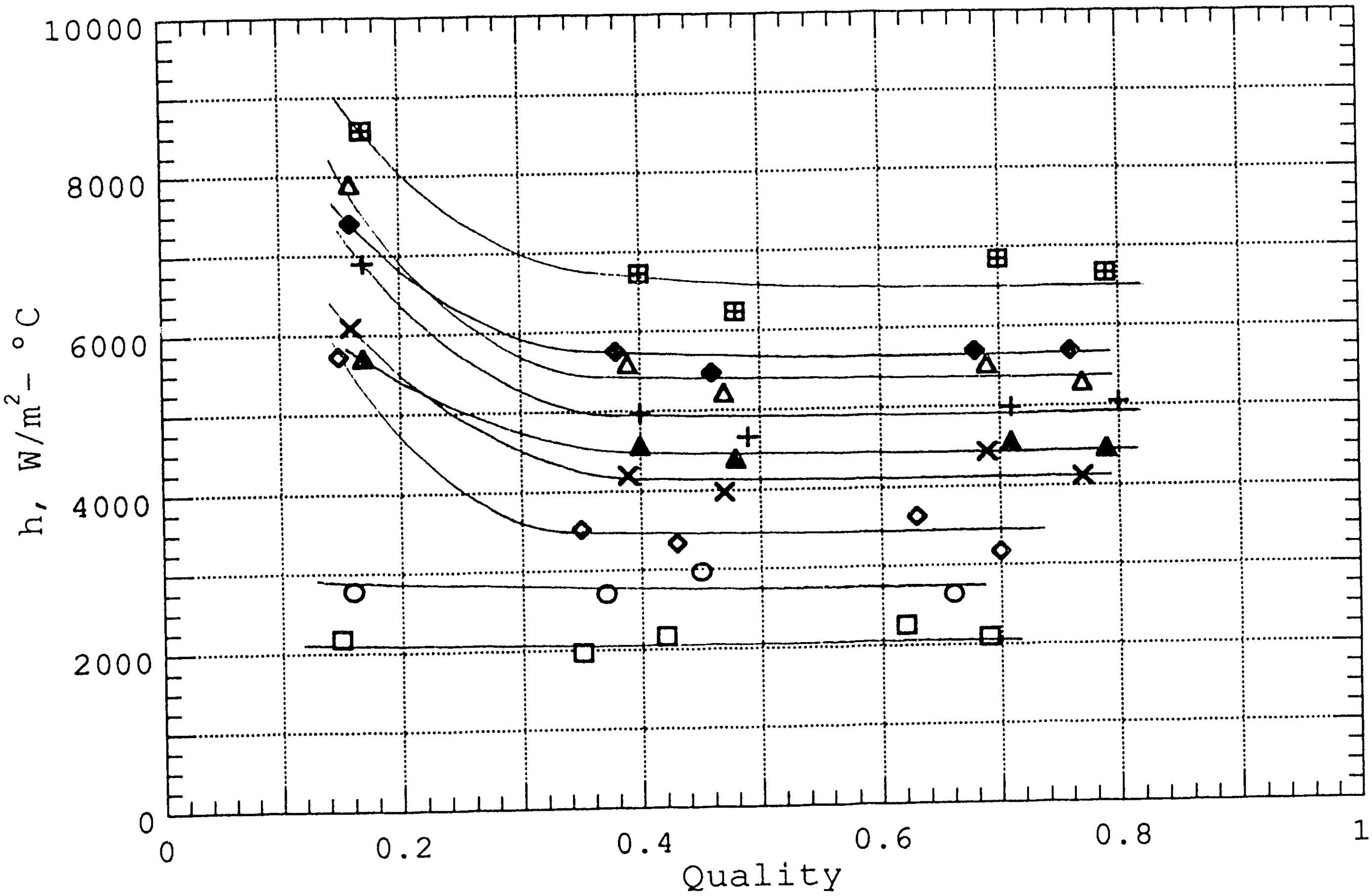
(a) R-12 Pressure Determination (calculation based on measured pressure drop)



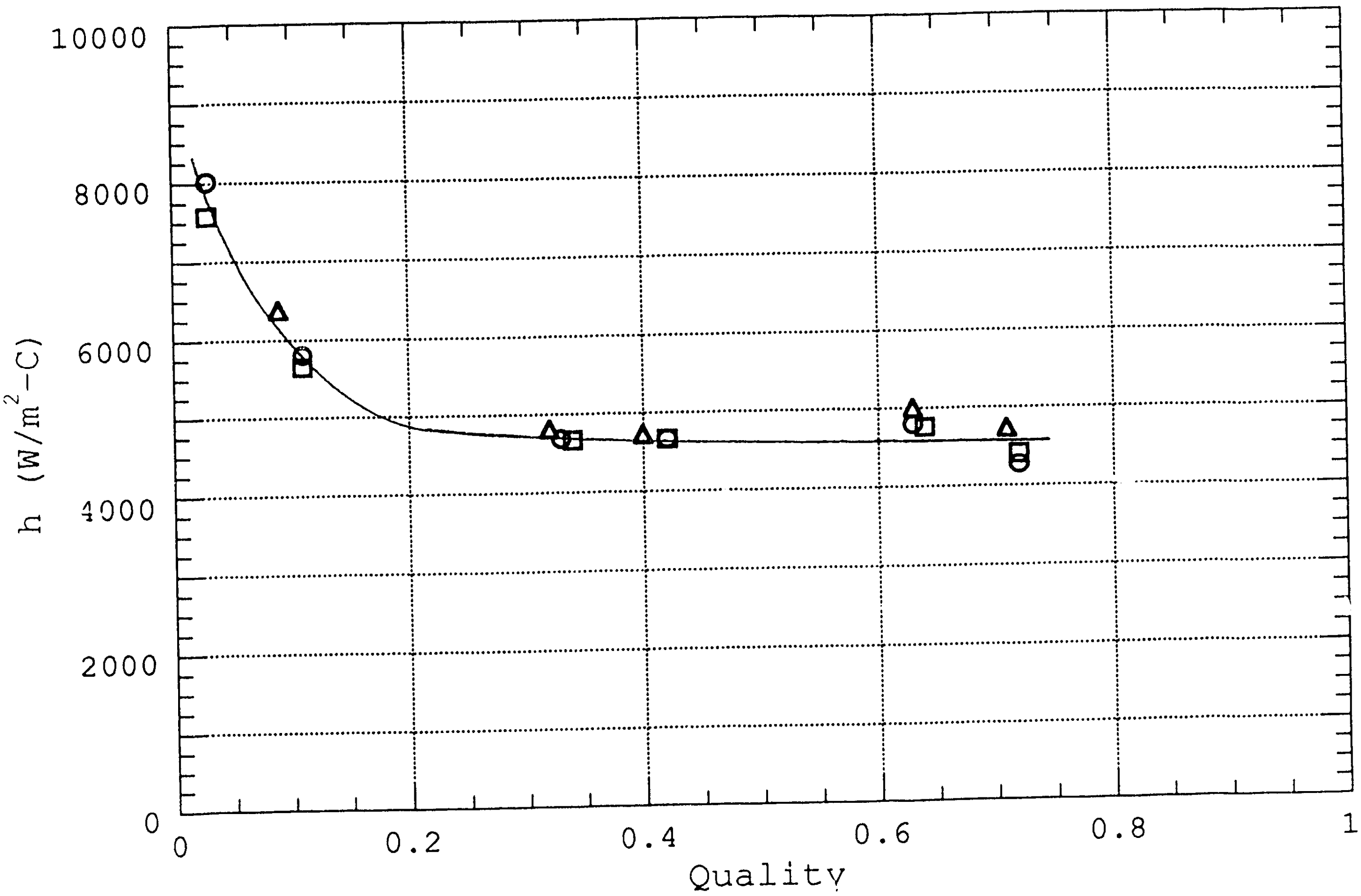
(b) Fluid and Wall Temperature Determination; (▲) measured wall temperature, (□) measured fluid temperature, (—) calculation based on saturation pressure

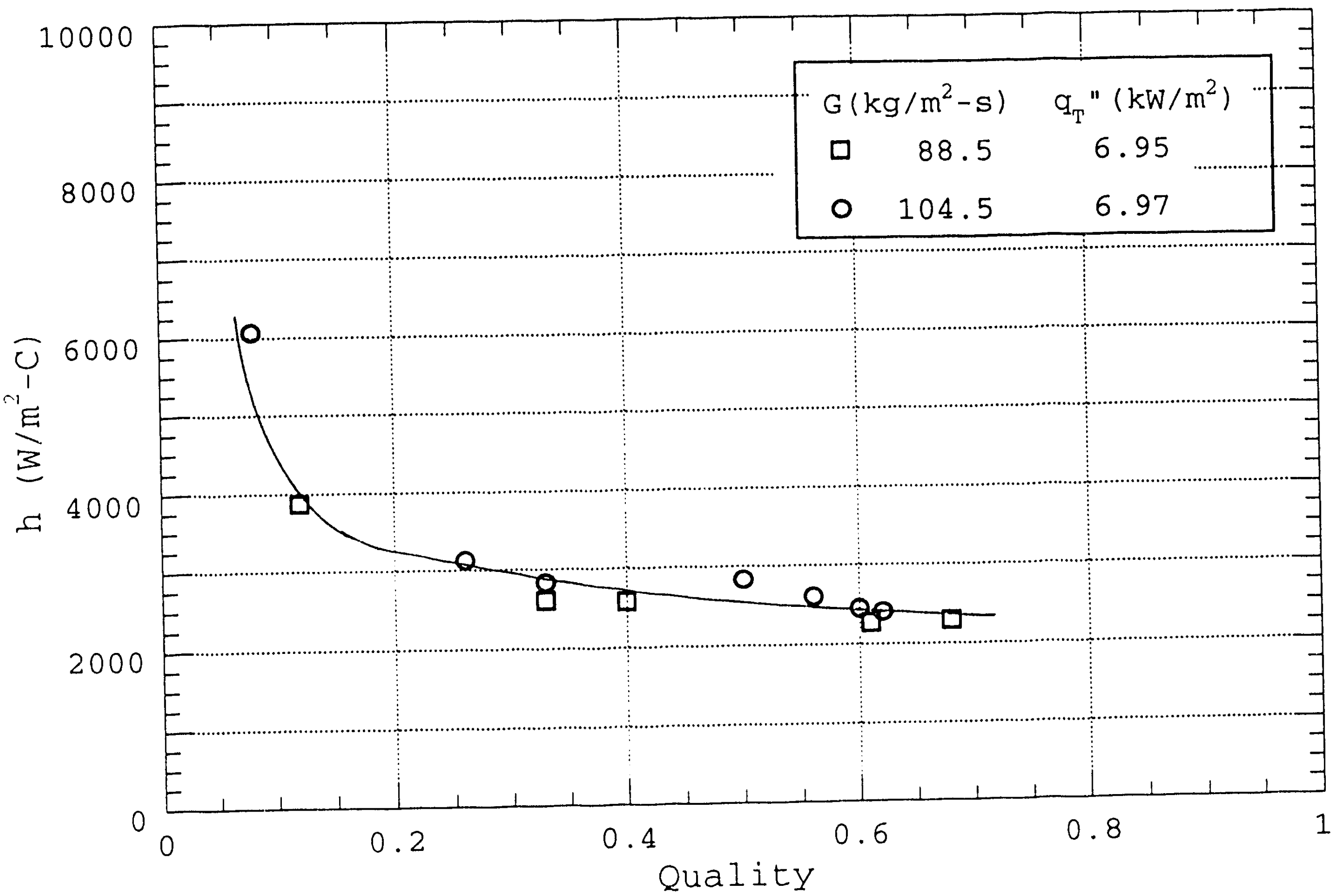


(c) Heat Transfer Coefficient Results; (O) measured h, (—) Trend

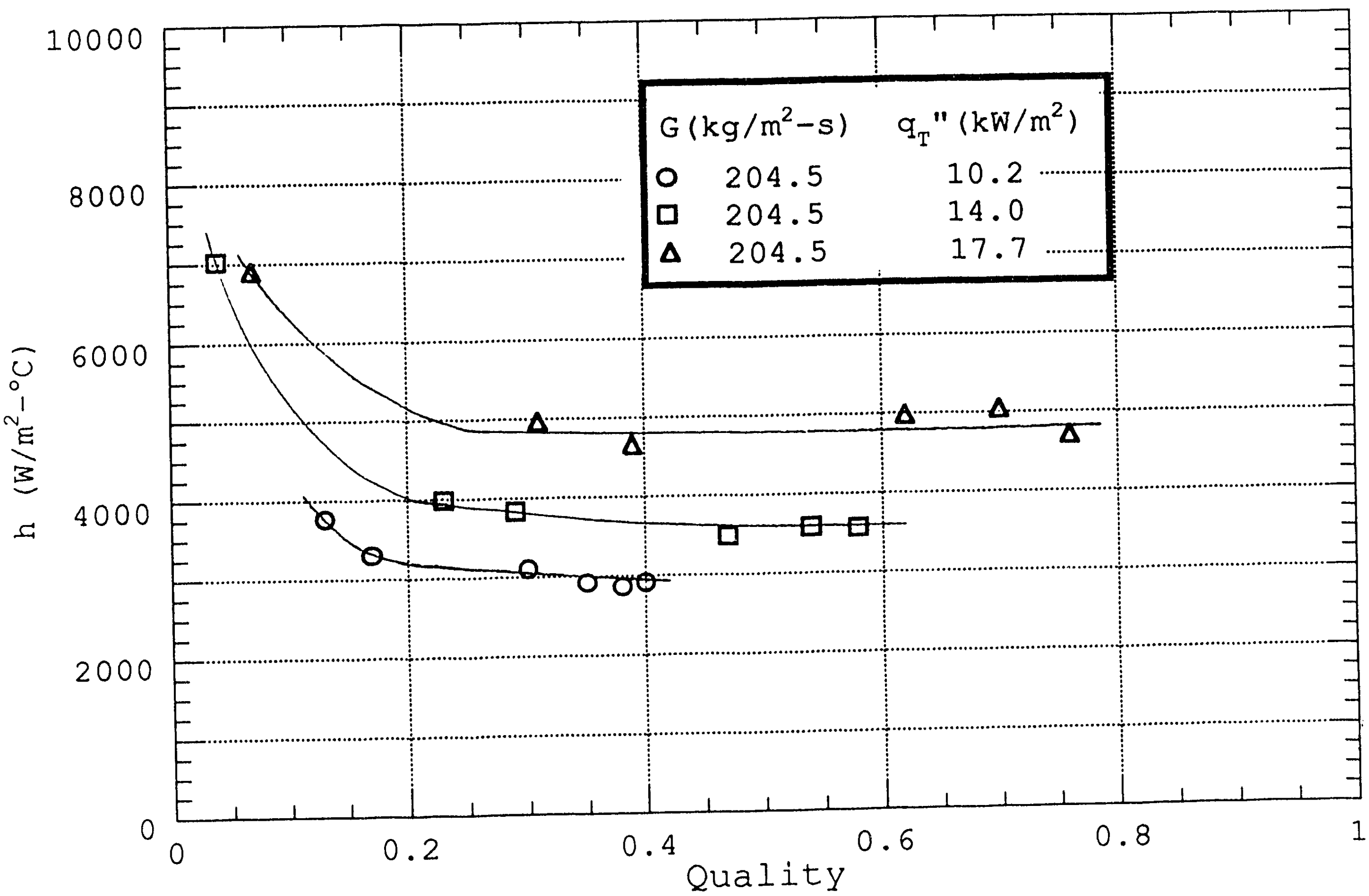


	G(kg/m^2-s)	q_T"(kW/m^2)	P_{in}(kPa)
□	54.6	4.1	840.8
○	71.8	5.8	838.5
◇	104.7	8.0	845.1
×	150.4	12.6	849.4
+	204.4	17.7	850.7
▲	216.8	18.7	846.3
△	277.4	23.2	850.5
◆	336.4	27.6	853.3
⊞	395.8	33.7	865.5

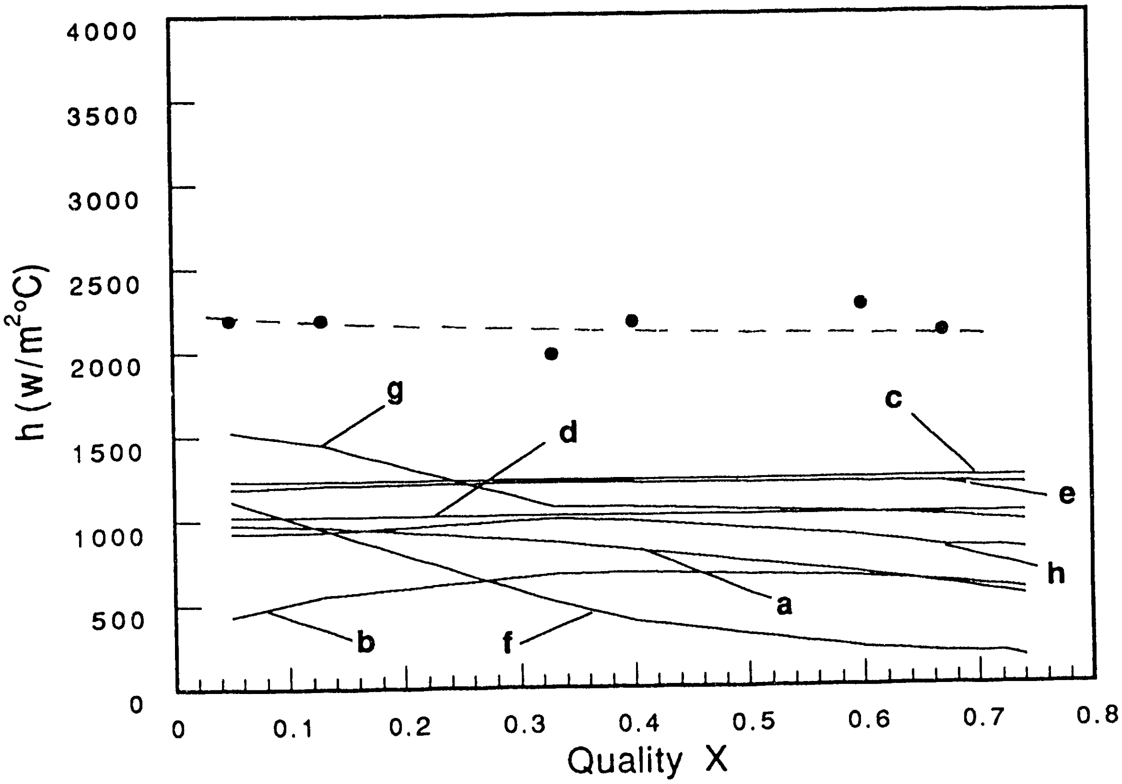

10000
8000
6000
4000
2000
0
h (W/m2-C)
0
0.2
0.4
0.6
0.8
1
Quality



10000
8000
6000
4000
2000
0
h (W/m^2-C)
0
0.2
0.4
0.6
0.8
1
Quality
G(kg/m^2-s)
q_T" (kW/m^2)
88.5
6.95
104.5
6.97

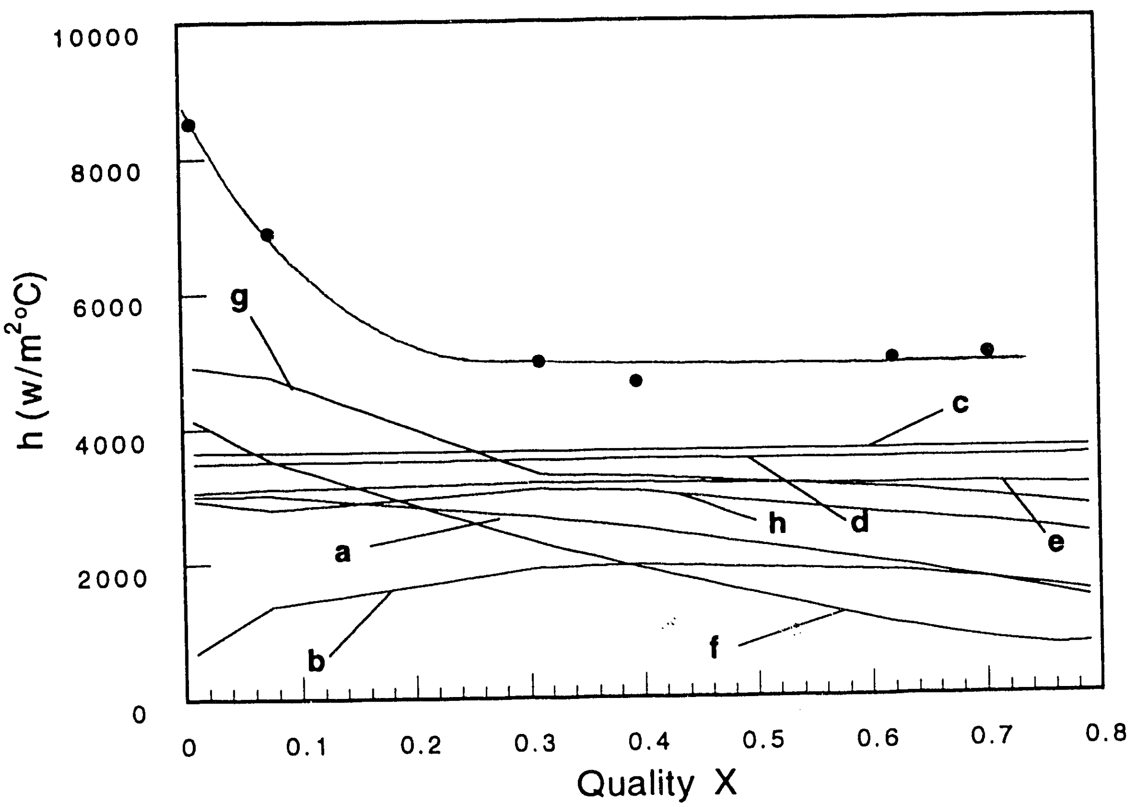


10000
8000
6000
4000
2000
0
h (W/m2-°C)
0
0.2
0.4
0.6
0.8
1
Quality
G(kg/m2-s)
qT"(kW/m2)
204.5
10.2
204.5
14.0
204.5
17.7

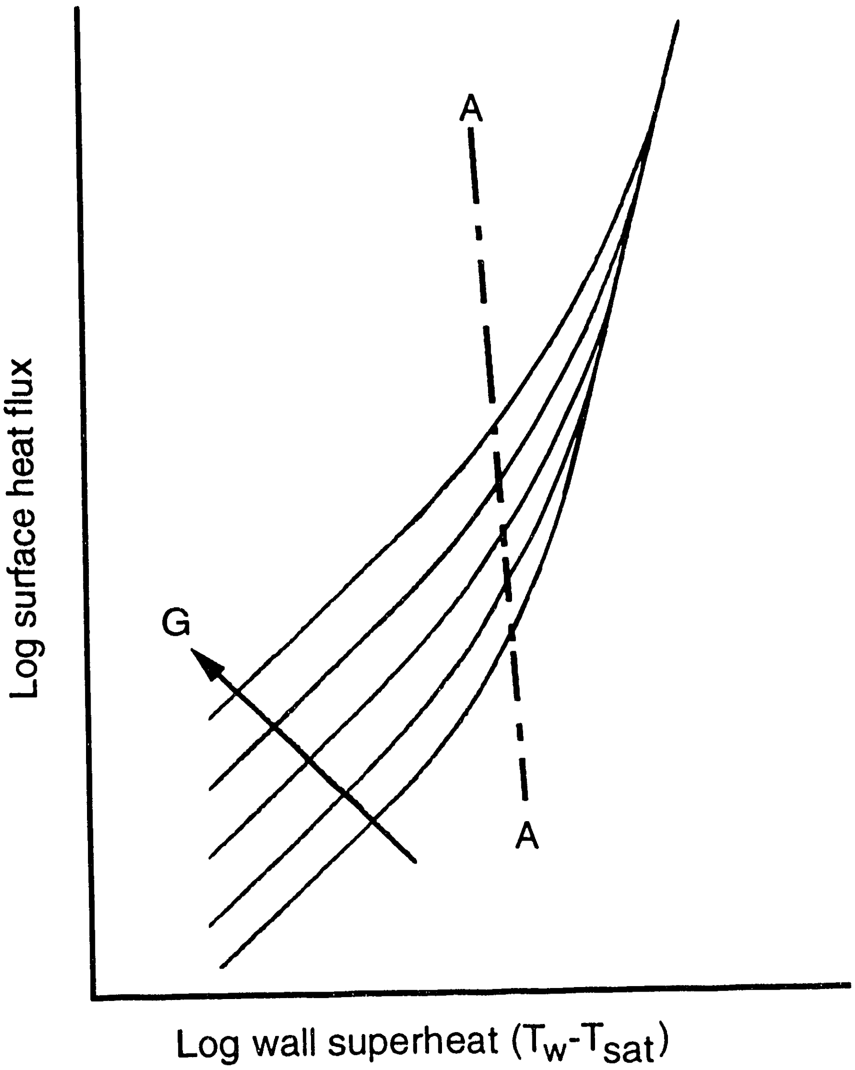


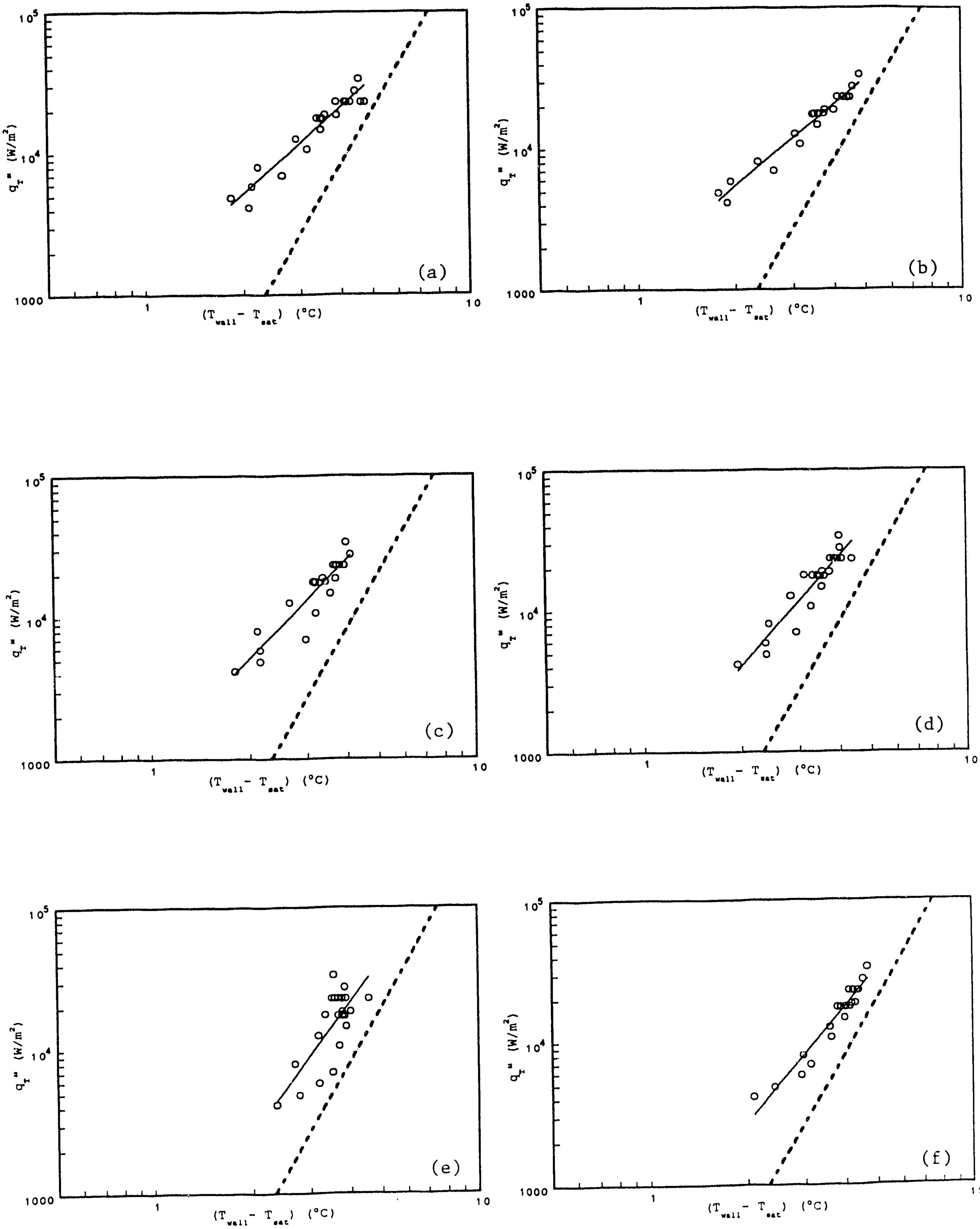
4000
3500
3000
2500
2000
1500
1000
500
0
h (w/m^2°C)
g
d
c
e
h
a
b
f
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
Quality X

8u



10000
8000
6000
4000
2000
0
h (w/m^2°C)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
Quality X
g
c
h
d
a
e
b
f


A
G
A
Log surface heat flux
Log wall superheat (T_w-T_{sat})



(a)
(b)
(c)
(d)
(e)
(f)
10^5
10^4
1000
1
10
q_r'' (W/m^2)
$(T_{wall} - T_{sat})$ (°C)

DATE

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END