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SUBCOOLED FORCED CONVECTION BOILING OF TRICHLOROTRIFLUOROETHANE

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16. Abstract

Experimental heat-transfer data were obtained for the forced-convection boiling of trichloro-trifluoroethane (R-113 or Freon-113) in a vertical annular test section. The pressure ranged from 10.0 to 17.5 bar, mass velocity from 1.57×10^3 to 2.55×10^3 kg $m^{-2}s^{-1}$, and inlet sub-cooling from 10 to 50 C. The 97 data points obtained covered heat transfer by forced convection, local boiling, and fully-developed boiling. Correlating methods were obtained which accurately predicted the heat flux as a function of wall superheat (boiling curve) over the range of parameters studied.

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SUBCOOLED FORCED CONVECTION BOILING OF TRICHLOROTRIFLUOROETHANE

by R. S. Dougall and D. J. Panian

University of Pittsburgh

SUMMARY

An experimental investigation was made of subcooled forced-convection boiling of Refrigerant-113 (Freon). The flow was circulated through an annular geometry in an upward direction. The inner and outer diameters of the annulus were 1.90 and 3.18 cm respectively. The pressure range studied was from 10.0 to 17.5 bar, mass velocities from 1.57×10^3 to 2.55×10^3 kg $m^{-2}s^{-1}$, and inlet subcoolings of 10 to 50 C. Wall and bulk fluid temperatures were measured as the heat flux was increased through the forced convection, local boiling, and fully-developed boiling ranges. From these measurements, experimental boiling curves (i.e. heat flux versus wall superheat) were constructed.

The experimental data in the single-phase forced convection range could be correlated to $\pm$ 10 per cent by the conventional Dittus-Boelter equation. A correlation for the heat transfer coefficient during fully-developed boiling was obtained. The form of the equation was of the type commonly used by Russian investigators where the heat transfer coefficient is a function of heat flux and pressure. This correlation fit the data to $\pm$ 36 per cent. By using the forced convection and fully-developed boiling correlations and theoretical predictions for the point of bubble inception, the data in the local boiling range could also be correlated.

INTRODUCTION

Boiling heat transfer has been important in recent years because large amounts of heat may be transferred with only a modest temperature difference. In rocket motors and nuclear reactors, heat transfer rates from 500 to 5000 kw m^{-2} are common: In the cooling of high-speed digital-computer components, the heat fluxes are modest, but a fairly uniform temperature is required which can be achieved by operating under boiling conditions (ref. 1). Although much information is known about boiling, it is sometimes difficult to predict beforehand actual values. This requires the necessity of experimental tests of many boiling systems.

If an engineering system operates with water as the heat transfer substance, a great deal of information is available to the designer (e.g. see ref. 2). However, any required tests may still be very difficult and expensive since water systems operate at high pressures and heat fluxes. For a number of years, the Mechanical Engineering Department of the University of Pittsburgh has been studying the boiling properties of Refrigerant-113 (also known as Freon-113, Genetron-113, trichlorotrifluoroethane, etc.). The purpose of this program has been to improve our knowledge of the boiling characteristics of this fluid so that it may be used in place of other fluids to simplify boiling tests (refs. 3 to 7). These investigations have been primarily aimed at studying critical heat flux phenomena.

Other investigators have been working along similar lines. In reference 8, a scaling procedure to compare critical heat flux data with R-12 data was developed. Staub (ref. 9) further developed this procedure. Coffield et al (ref. 5) showed that this procedure could be applied to low quality and subcooled flows.

The primary aim of the references just cited was to compare other fluids to water. Berensen (refs. 10 & 11) has made experimental investigations with R-113 in order to apply the results to liquid metal flows. In particular he was interested in using the results for Rankine cycle space power systems using potassium as the coolant.

Based on these studies, information exists that enables engineers to predict the critical heat flux and stability of a R-113 flow. However, data on forced-convection boiling, bubble inception and bubble departure is not available in the literature. The aim of this report is to supply data on the heat transfer coefficients for forced-convection boiling of R-113. A limited amount of data for bubble inception is also included. Further data on bubble inception and departure has been obtained at the University of Pittsburgh and will appear in the near future.

Forced-convection boiling can be divided into two subdivisions: local boiling and fully-developed boiling. Fully-developed boiling is reached when heat transfer does not depend on the mass velocity or the bulk fluid temperature (subcooling). In this case, the heat transfer is determined by the wall superheat, pressure, and surface-fluid combination. In local boiling, the heat transfer does depend on the mass velocity and bulk fluid temperature. Local boiling is the transition region between single-phase forced convection and fully-developed boiling as is shown in Figure 1.

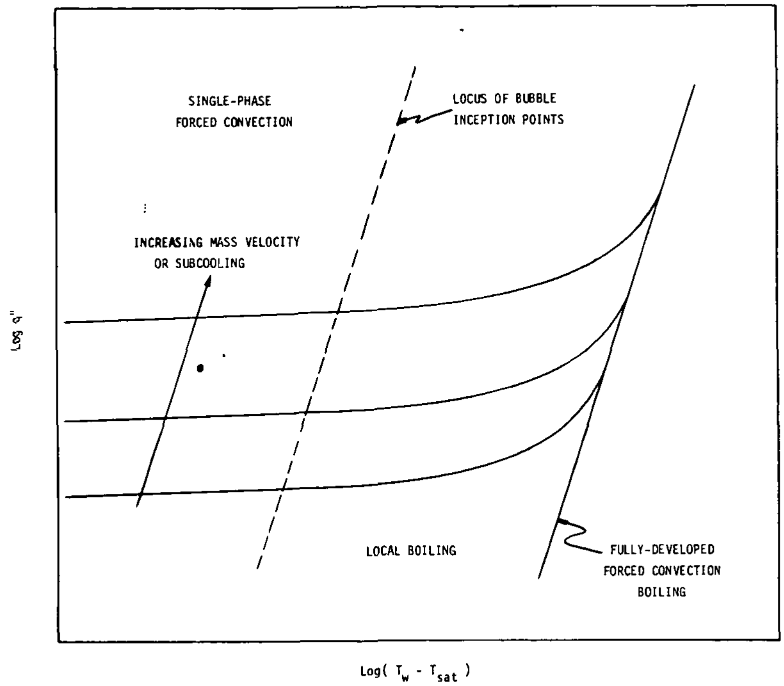

Figure 1. - Typical boiling curve for forced-convection boiling.

The principle aim of this paper is to experimentally determine boiling curves of the type shown in Figure 1 for R-113 in the pressure range from 10.3 to 17.2 bar. The tests were performed for the fluid flowing upward through an annular flow passage having a 32 mm O.D and 19 mm I.D. Heating was from the inside surface only. A correlation was obtained for the fully-developed forced-convection boiling process. Furthermore, the method of Rohsenow and Bergles (ref. 12) was shown to accurately predict the heat transfer during local boiling.

SYMBOLS

D_e	equivalent diameter, m
G	mass velocity, kg m^{-2} s^{-1}
h	heat transfer coefficient, W m^{-2} $^oK^{-1}$
k	thermal conductivity, W m^{-1} $^oK^{-1}$
Nu	Nusselt number, dimensionless
p	system pressure, bar
Pr	liquid Prandtl number, dimensionless

q''	heat flux, $W\ m^{-2}$
Re	liquid Reynolds number, dimensionless
T	temperature, ^{o}K
ΔT	temperature difference, ^{o}K
μ	liquid viscosity, $kg\ m^{-1}\ s^{-1}$

Subscripts:

b	bulk fluid property
fc	forced convection
fdb	fully-developed boiling
i	inception
lb	local boiling
ref	reference
sat	saturation value
sub	subcooling
w	wall

EXPERIMENTAL APPARATUS AND PROCEDURE

The experimental measurements for this investigation were made at the Thermal-Hydraulics Laboratory of the School of Engineering, University of Pittsburgh. The 20.4 atm (300 psi) R-113 Heat Loop was used to supply fluid to the test section at the desired flow rate, pressure and temperature.

The test section was in the form of a vertical annulus with a heated inner tube and upward flow of the fluid. The basic arrangement is shown in Figure 2. Heat was supplied by passing a D.C. electric current through the inner tube of the annulus. This tube was 19 mm O.D. and was made in three parts. The upstream section consisted of thick-walled nickel tubing which resulted in negligible heat being supplied to the fluid. This nickel section was approximately 127 cm long. The next section was 30.5 cm of thin-walled (0.089 mm) Type 304 stainless-steel tubing. This was the region of high heat flux and where temperature measurements were made. The last section was a thick-walled piece of Type 304 stainless steel and extended to the outlet of the test section. The three pieces

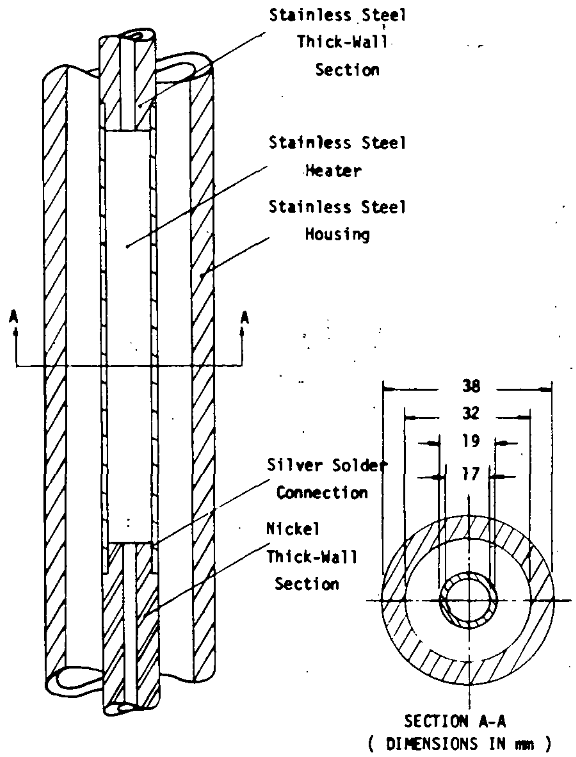

Figure 2. - Cross-sectional views of test section

of tubing were silver soldered together. The uninsulated outer wall of the annulus was made of Type 304 stainless steel and had a 38 mm outside diameter with a 3.0 mm wall thickness. The spacing between the heated inner tube and the outer tube was maintained by Micarta spacers at four elevations.

The inside wall temperature was measured at three elevations above the start of the high heat flux region of the test section. These thermocouples, plus a chromel-constantan thermocouple, were silver-soldered at the location shown in Figure 3. The chromel-constantan thermocouple located near the downstream end of the test section was used to sense the occurance of a critical heat flux (CHF) in the test section. The output of this thermocouple was connected to a Wheelco temperature controller whose reference junction was at room temperature. The Wheelco temperature controller triggered an alarm when a rapid and large temperature increase occured at CHF. The power to the test section was then manually tripped.

The four chromel-alumel thermocouples plus the inlet and outlet thermocouples were calibrated in an Apex Scientific electrically-heated oil bath. A NBS thermometer with a range of 0 to 200 °C in 0.2 °C increments was used as the reference. The thermocouple emf's were read with a Model 8686 Leeds and Northrup potentiometer.

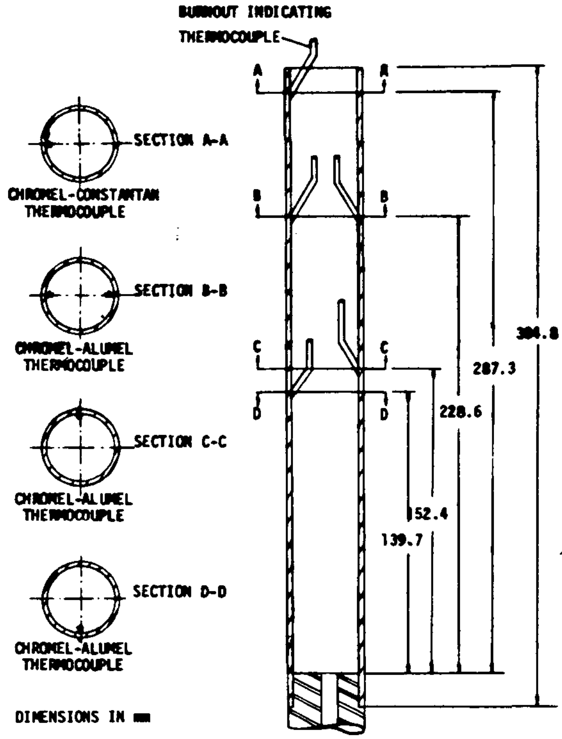

Figure 3. - Location of thermocouples.

More details of the experimental apparatus are given in refs. 7 and 13.

In general, the test procedure was to hold the pressure, inlet subcooling, and mass velocity constant while varying the power to the test section. Test data would be recorded for each variation in power.

Certain preliminary operations had to be performed prior to conducting an actual test. The more important of these involved pressurizing and heating the fluid and then removing any dissolved gases that may have been present.

To perform the test, the system was pressurized and the circulating pump was started. The energy from the circulating pump in conjunction with the preheaters was used to heat the R-113. A bleed-off at the uppermost elevation of the loop was employed to remove dissolved gases from the system while the R-113 was being heated. The loop heat-up cycle, including degasing, required approximately two hours. The pressure was then set at the test value and the mass velocity was adjusted to the desired value. Power was then turned on and adjusted to a level such that boiling of the R-113 occured and continued for from one-half to one hour. After the power-controlling rheostat was then adjusted to the minimum power position, the test would commence.

The initial set of data was taken with the rheostat in the minimum power position. Inlet and outlet temperatures were recorded first, followed by recording of the emf's of the Cr-Al thermocouples. This was then followed by the pressure, flow manometer, shunt millivolt, and test section voltage drop readings. Finally, the ambient temperature and pump parameters were recorded. The rheostat was then adjusted to produce a one millivolt increase in shunt millivolt value. Another set of data was then recorded. With data always taken for increasing power, this process continued until the Cr-Con thermocouple detected a rapid temperature increase i.e. CHF was reached. The power was then tripped and the test was considered complete. The pressure, inlet subcooling, and mass velocity were adjusted to the next set of operating conditions. However, no further degassing periods were conducted.

RESULTS AND DISCUSSION

Experimental Data

The range of values for the experimental data is summarized in Table I. Also shown in this table is the symbols that will be used to represent the various sets of data in the following figures. Based on an error analysis given in ref. 13, the maximum errors estimated for the main independent variables of this study are:

(a) Mass Velocity ± 1.7 per cent at 1.63×10^3 kg m^{-2} s^{-1}

± 1.0 per cent at 2.50×10^3 kg m^{-2} s^{-1}

(b) System Pressure $\pm$ 0.07 bar

(c) Inlet Temperature $\pm$ 1.0 $^{\circ}$K

(d) Heat Flux $\pm$ 10.7 per cent

TABLE I. - SUMMARY OF EXPERIMENTAL DATA

Run Number	Symbol	System Pressure bar	Bulk Temperature K	Mass Velocity kg m^{-2} s^{-1}	Number of Data Points in		
					Single-Phase Forced Convection	Local Boiling	Fully Developed Nucleate Boiling
1	○	12.6 to 13.9	382.0 to 403.3	2.47×10^3 to 2.54×10^3	5	5	12
2	□	10.0 to 10.5	386.0 to 407.7	2.37×10^3 to 2.50×10^3	3	2	8
3	△	17.1 to 17.5	380.5 to 401.2	2.40×10^3 to 2.50×10^3	16	8	0
4	●	17.1 to 17.3	397.6 to 414.5	2.42×10^3 to 2.48×10^3	0	2	11
5	■	12.7 to 13.3	381.0 to 408.0	1.57×10^3 to 1.67×10^3	5	6	8
6	▲	13.3 to 13.9	378.8 to 394.6	2.47×10^3 to 2.55×10^3	1	2	3

TABLE II. - REDUCED EXPERIMENTAL DATA

Run Number	System Pressure bar	Saturation Temperature K	Bulk Temperature K	Heater Wall Temperature K	Heat Flux kW m^{-2}	Mass Velocity kg m^{-2} s^{-1}
1A	13.6	428.3	382.0	391.8	26.5	2.54×10^3
1B	12.6	424.2	382.2	396.5	36.3	2.54
1C	13.4	427.5	383.6	404.7	51.4	2.54
1D	13.4	427.5	386.1	409.2	58.4	2.52
1E	13.3	427.2	387.4	418.4	74.9	2.52
1F	13.0	425.7	389.3	431.3	107.3	2.50
1G	13.9	429.5	390.6	435.0	115.3	2.50
1H	13.9	429.5	393.2	436.6	130.1	2.49
1I	13.8	429.2	397.5	437.4	139.0	2.48
1J	13.9	429.5	389.0	436.6	152.5	2.22
1K	13.6	428.6	390.0	437.6	169.8	2.49
1L	13.6	428.3	391.1	437.6	180.6	2.49
1M	13.6	428.6	391.8	438.2	195.0	2.49
1N	13.6	428.6	392.7	438.6	209	2.49
1O	13.4	427.8	393.3	438.6	226	2.49
1P	13.4	427.8	394.5	439.0	243	2.48
1Q	13.7	429.0	396.4	440.1	258	2.48
1R	13.7	429.0	397.6	440.2	277	2.48
1S	13.9	429.5	399.2	441.1	296	2.48
1T	13.8	429.2	400.7	441.2	313	2.48
1U	13.8	429.2	401.9	441.5	332	2.47
1V	13.8	429.2	403.3	441.6	349	2.47
2A	10.2	413.2	386.0	396.1	26.6	2.50×10^3
2B	10.2	413.2	386.8	400.2	36.3	2.50
2C	10.3	413.5	387.6	408.0	50.7	2.50
2D	10.4	414.0	388.7	415.8	65.9	2.50
2E	10.4	414.0	390.3	420.3	83.0	2.50
2F	10.3	413.5	393.1	422.0	104.2	2.49
2G	10.4	414.0	396.3	423.3	127.5	2.49
2H	10.4	414.0	399.8	424.3	154.7	2.38
2I	10.3	413.5	403.9	424.8	162.7	2.37
2J	10.0	412.1	407.7	424.2	177.1	2.44
2K	10.4	414.2	298.1	425.8	193.0	2.48
2L	10.5	414.6	401.0	426.3	208	2.45
2M	10.5	414.6	404.5	426.8	223	2.45
3A	17.2	442.2	391.2	401.0	27.3	2.50×10^3
3B	17.2	442.2	389.9	401.5	30.6	2.50
3C	17.3	442.4	388.6	403.7	36.4	2.50
3D	17.3	442.4	388.7	408.0	37.8	2.48
3E	17.3	442.4	389.3	410.8	50.5	2.50
3F	17.3	442.4	389.8	414.5	57.7	2.50
3G	17.4	442.7	390.5	419.6	67.0	2.49
3H	17.4	442.7	391.3	424.8	76.0	2.49
3I	17.4	442.7	392.3	429.8	86.5	2.50
3J	17.4	442.7	393.5	435.1	195.8	2.49
3K	17.1	441.7	394.3	439.9	107.2	2.49
3L	17.1	442.0	395.9	442.8	110.7	2.49
3M	17.2	442.2	397.3	445.2	117.6	2.48
3N	17.3	442.4	399.8	447.3	142.9	2.48
3O	17.3	442.4	394.2	406.2	30.2	2.40
3P	17.4	442.9	394.2	411.4	43.2	2.49
3Q	17.5	443.1	394.7	418.9	58.8	2.49
3R	17.3	442.4	395.4	426.0	84.4	2.49
3S	17.3	442.4	396.2	434.7	94.9	2.49
3T	17.4	442.7	397.1	444.3	119.9	2.48
3U	17.4	442.7	398.1	447.6	132.0	2.48
3V	17.4	442.7	399.1	448.8	145.8	2.48
3W	17.4	442.7	400.1	449.5	157.1	2.48
3X	17.3	442.4	401.2	450.0	173.4	2.47
4A	17.2	442.2	402.8	450.4	183.3	2.47×10^3
4B	17.2	442.2	403.7	450.6	195.0	2.47
4C	17.1	442.0	404.8	450.8	211	2.47
4D	17.3	442.4	406.3	451.8	228	2.46
4E	17.3	442.4	407.5	452.2	245	2.44
4F	17.1	441.7	397.6	449.8	183.2	2.48
4G	17.1	441.7	399.5	450.6	216	2.47
4H	17.2	442.2	401.6	451.8	244	2.47
4I	17.3	442.4	406.6	452.2	265	2.44
4J	17.3	442.4	408.4	452.6	278	2.46
4K	17.2	442.2	410.7	452.7	296	2.43
4L	17.2	442.2	412.9	453.0	314	2.43
4M	17.2	442.2	414.5	453.1	352	2.42

TABLE II. - REDUCED EXPERIMENTAL DATA (Cont'd)

Run Number	System Pressure bar	Saturation Temperature K	Bulk Temperature K	Heater Wall Temperature K	Heat Flux $kW\ m^{-2}$	Mass Velocity $kg\ m^{-2}\ s^{-1}$
5A	13.0	426.1	381.0	403.4	37.0	1.67×10^3
5B	12.9	425.7	382.4	408.8	43.7	1.67
5C	12.8	425.1	383.5	414.7	51.0	1.67
5D	12.7	424.8	384.3	420.6	58.9	1.67
5E	12.9	425.7	385.5	424.2	63.1	1.65
5F	12.9	425.5	386.3	427.3	67.5	1.64
5G	12.9	425.7	387.2	430.6	74.6	1.65
5H	13.2	426.6	388.3	432.5	85.9	1.65
5I	13.3	427.2	389.8	434.1	106.8	1.65
5J	13.3	427.2	390.8	434.7	118.9	1.65
5K	13.2	427.0	392.0	434.4	129.1	1.63
5L	13.0	426.1	393.4	434.6	141.1	1.63
5M	13.0	426.1	394.8	435.0	153.9	1.61
5N	13.0	426.1	396.6	435.5	168.2	1.61
5O	12.9	425.7	398.4	435.5	179.1	1.60
5P	13.1	426.5	400.6	436.3	196.2	1.60
5Q	13.0	426.1	403.1	436.8	213	1.60
5R	13.0	426.1	405.4	437.0	227	1.57
5S	13.0	426.1	408.0	437.2	240	1.59
6A	13.3	427.2	378.8	410.1	74.9	2.55×10^3
6B	13.9	429.5	381.6	436.2	130.3	2.55
6C	13.6	428.3	384.1	436.2	169.6	2.54
6D	13.4	427.8	387.6	437.1	208	2.51
6E	13.4	427.8	390.3	437.6	258	2.50
6F	13.4	427.8	394.6	439.6	316	2.47

Table II presents the reduced experimental data for all the conditions investigated. The heater wall temperature presented in this table is the arithmetic average of the four values obtained from the thermocouples. The thermocouple readings were corrected to account for the temperature difference through the heater wall. It was felt this procedure gave the best value of temperature over the central portion of the heater surface.

The fluid properties were evaluated from tables developed in reference 13. In addition to these values, saturation fluid enthalpies as a function of temperature were obtained from reference 14.

Single-Phase Forced Convection

A number of experimental data conditions corresponded to single-phase forced convection for a turbulent flow. The number of points in this state are shown in Table I. The heat transfer coefficients were calculated using the values presented in Table II. The single-phase heat-transfer coefficient was defined by the following equation:

$$h_{fc} = \frac{q''}{T_w - T_b} \qquad (1)$$

where

h_{fc} is the local forced-convection heat-transfer coefficient

q'' is the local wall heat flux

T_w is the local wall temperature

T_b is the local bulk fluid temperature

The resulting heat transfer coefficients for turbulent forced convection can be correlated in the standard form of Nusselt number being a function of Reynolds number and Prandtl number. The well-known Dittus-Boelter correlation for fully-developed flow in smooth tubes agrees with this annulus data within the experimental error of the tests. The correlating equation is:

$$\frac{h_{fc} D_e}{k_b} = 0.023 \, (G D_e / \mu)^{0.8} \, Pr_b^{0.4} \qquad (2)$$

All the fluid properties are evaluated at the bulk liquid temperature. The equivalent diameter for an annulus is equal to twice the flow gap or the larger diameter minus the smaller diameter. A comparison of the forced convection data with equation (2) is shown in Figure 4.

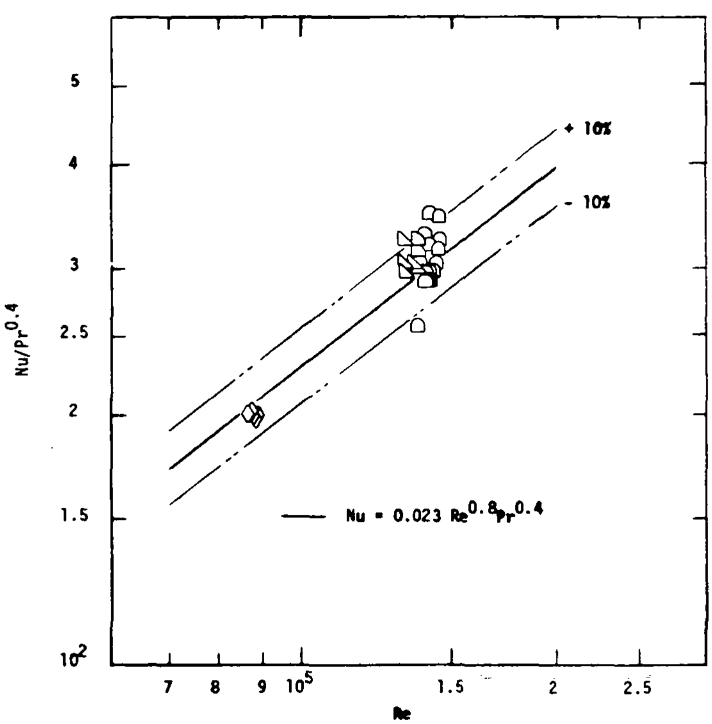

Figure 4. - Comparison of single-phase forced convection data with the Dittus-Boelter equation.

Fully-Developed Forced-Convection Boiling

Fully-developed forced-convection boiling is reached when the boiling heat-transfer coefficient no longer depends on the mass velocity or the subcooling of the flow. The number of data points taken in the regime of fully-developed forced-convection boiling are shown in Table I.

The heat-transfer coefficient for fully-developed forced-convection boiling is defined in terms of the amount the wall temperature is above the fluid saturation temperature (i.e. in terms of wall superheat). The defining equation is:

$$h_{fdb} = \frac{q''}{T_w - T_{sat}} \tag{3}$$

Correlations for pool boiling of a number of fluids including R-11, 12, and 22 are presented in reference 15. Correlations are also presented for the forced-convection boiling of R-11 and R-22 inside horizontal tubes. Danilova (ref. 16) presented correlations for pool boiling data of almost the entire "FREON" family including R-113. From these investigations, it was found that the fully-developed nucleate-boiling data could be correlated quite well with the following correlation for pool boiling and forced-convection boiling in horizontal tubes.

$$h_{fdb} = C \cdot (q'')^m \cdot p^n \tag{4}$$

where

h_{fdb}	is the fully-developed boiling heat-transfer coefficient
q''	is the local wall heat flux
p	is the system pressure
C,m,n	are experimentally determined constants which depend on the fluid.

A correlation of the type shown in equation (4) was used to correlate the fully-developed forced-convection boiling **data** obtained in this investigation. From a regression analysis the resulting equation is:

$$h_{fdb} = 1.67(q'')^{0.65}\ p^{\ 0.55} \tag{5}$$

where the heat transfer coefficient is in $W\ m^{-2}\ {}^oK^{-1}$; the heat flux is in $W\ m^{-2}$; and the pressure is in bars. This correlation is plotted with the data in Figure 5. The value of 0.65 for the exponent on the heat flux is in the range from 0.6 to 0.75

given in reference 15.

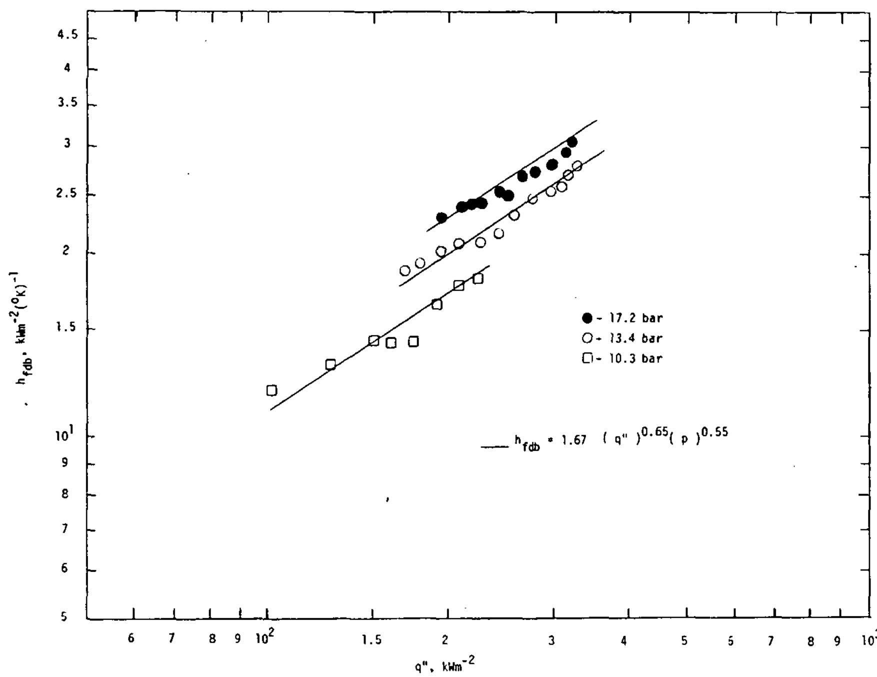

Figure 5. - Correlation of the data for fully-developed forced convection boiling.

The pressure effect shown in Figure 5 is typical of those obtained for the "FREONS." The effect can also be seen from consideration of equation (5). Given the same heat flux, the heat transfer coefficient increases with increasing pressure. From an error analysis presented in reference 13, the maximum error in the heat transfer coefficients determined from the data is ± 36 per cent. It is seen in Figure 5 that equation (5) correlates the experimental data well within this error band.

The point where the CHF occurred is not shown on Figure 5. However, this phenomena was experienced during each test series and was very close to the last value indicated. It was also found that on every test the temperature indicated on the Wheelco recorder decreased approximately 3°K just prior to CHF. A similar result was reported in reference 7.

Local Boiling

As shown in Figure 1, local boiling is the regime that occurs between the point of bubble inception and fully-developed boiling. In order to predict the complete heat transfer characteristics of a system during forced convection boiling, a method to determine the "boiling curve" in the region of local boiling must be available. There are a number of different procedures available to do this. The one that has shown good results for water is the one developed by Bergles and Rohsenow (ref. 12).

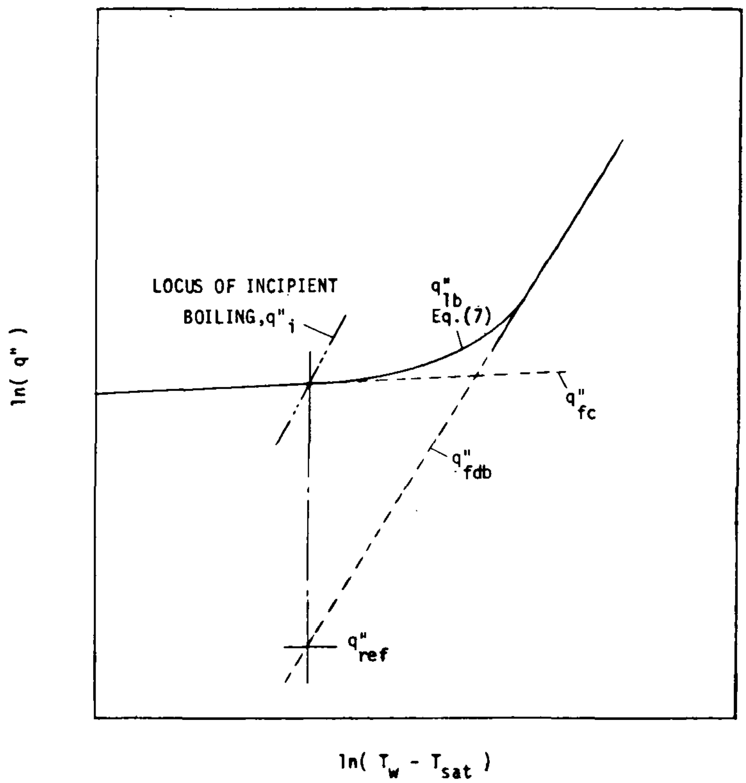

Figure 6. - Basic quantities used to determine boiling curve in region of local boiling.

The procedure developed by Bergles and Rohsenow can be explained with the help of Figure 6. This figure shows the boiling curve for a given mass velocity and fluid subcooling. Basically, heat flux is plotted versus wall superheat. There are two asymptotes to the boiling curve. For low values of wall superheat, there is no boiling and heat transfer takes place by forced convection. The correlation presented in equation (2) is used to determine this portion of the boiling curve. This heat flux is designated as q''_{fc}.

For high values of wall superheat, fully-developed forced convection boiling takes place. Under these conditions, the correlation presented in equation (5) can be used. For this calculation, it is useful to combine equations (3) and (5) to eliminate the heat transfer coefficient. This results in the following equation.

$$q''_{fdb} = 4.33\ p^{1.57} (T_w - T_{sat})^{2.86} \qquad (6)$$

The units in equation (6) are the same as those used in equation (5).

A method of predicting the point of bubble inception is needed. Bergles and Rohsenow (ref. 12) give a method that has been reasonably successful in predicting the inception point under forced convection conditions. They graphically find when the temperature distribution in the liquid is tangent to the bubble equilibrium equation. A computer program was developed at the University of Pittsburgh that replaces this graphical solution. A listing of this program is given in reference 13. The heat flux at which bubble inception first occurs is designated as q''_i and is shown in Figure 6. At the same wall superheat that corresponds to q''_i, there is a heat flux for fully-developed boiling given by equation (6). This heat flux is designated as the reference heat flux, q''_{ref} and is also shown on Figure 6.

The method of Bergles and Rohsenow uses the curve of forced convection heat flux, q''_{fc}; the curve of fully-developed boiling, q''_{fdb}; and the reference heat flux, q''_{ref} to determine the curve in the region of local boiling. Their equation is:

$$q''_{lb} = q''_{fc} \left\{ 1.0 + \left[\frac{q''_{fdb}}{q''_{fc}} \left(1 - \frac{q''_{ref}}{q''_{fdb}} \right) \right]^2 \right\}^{\frac{1}{2}} \qquad (7)$$

where q''_{lb} is the heat flux for local boiling.

A comparison of equation (7) with the R-113 data obtained in this investigation is shown in Figures 7, 8, and 9 for pressures of 10.3, 13.4, and 17.2 bar respectively. The X's shown on the figures indicate the range of bubble inception values over the range of mass velocity and subcooling present in the experimental data. The curves drawn on these figures represent an average value of mass velocity and subcooling.

The data points vary around these average values. In particular, some of the subcoolings are ±15°K from the average values. This wide range in subcooling was due to the difficulty of controlling this property during the tests. Even with the range of data available, the boiling curves calculated at these three pressures appears to be in excellent aareement with the data. In particular, the method of Bergles and Rohsenow gives a very reasonable result in the region of local boiling.

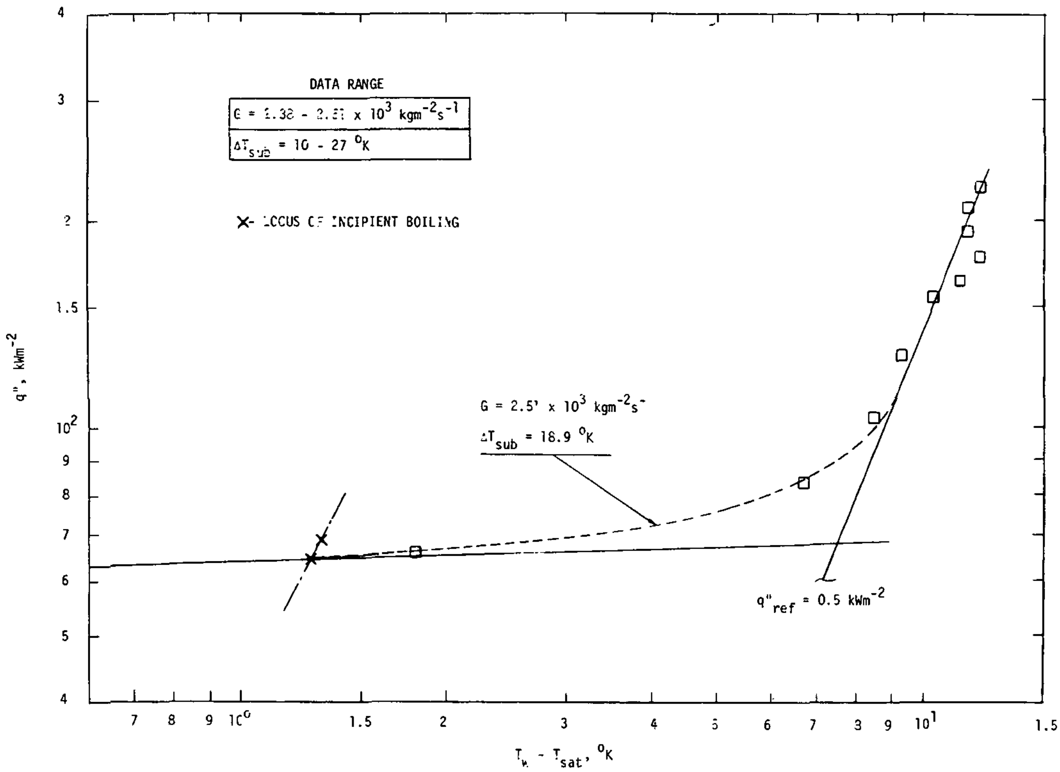

Figure 7. - Comparison of theoretical boiling curve of R-113 at 10.3 bar with experimental data.

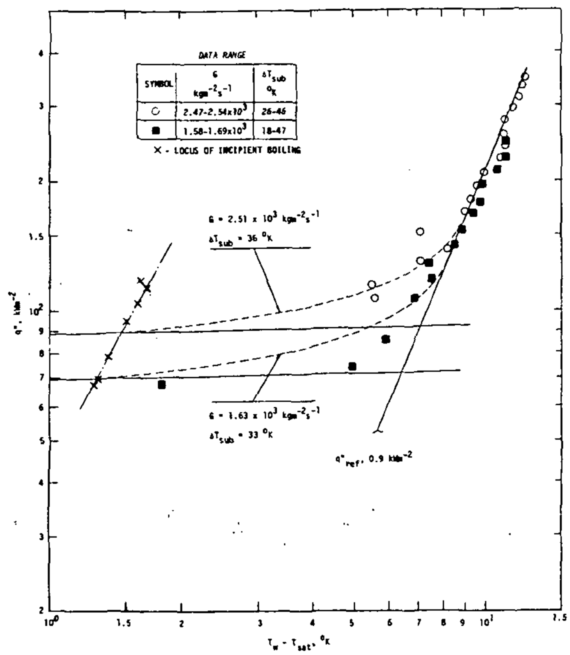

Figure 8. - Comparison of theoretical boiling curve of R-113 at 13.4 bar with experimental data.

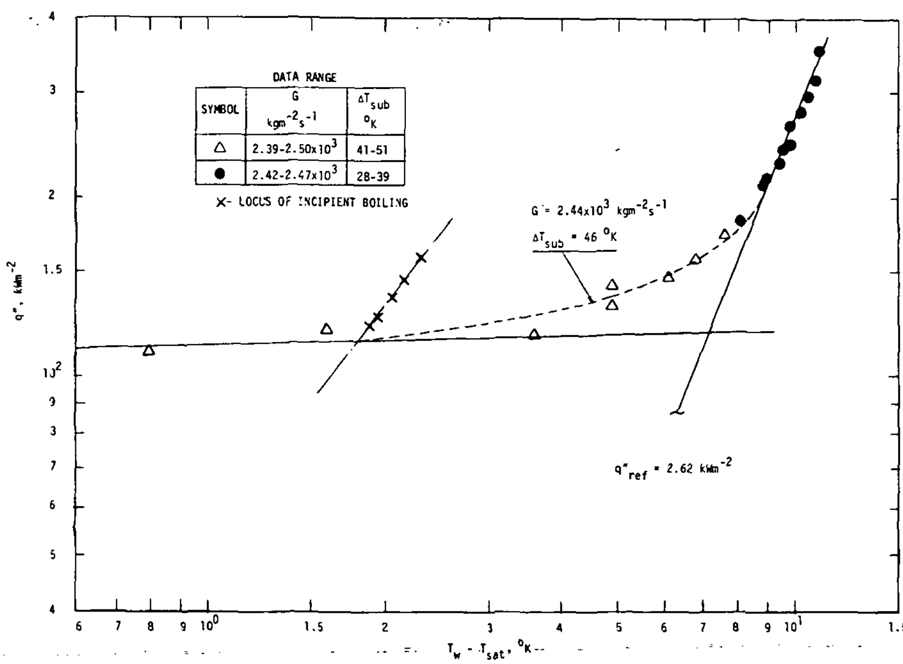

Figure 9. - Comparison of theoretical boiling curve of R-113 at 17.3 bar with experimental data.

CONCLUSIONS

Experimental data were taken for the subcooled forced convection boiling of R-113. An annular test section was utilized with the flow being in a vertically upward direction.

The non-boiling experimental data were found to correlate quite well with the conventional Dittus-Boelter relation

$$\frac{h_{fc} D_e}{k_b} = 0.023 \left(GD_e/\mu \right)^{0.8} Pr_b^{0.4} \qquad (2)$$

which correlates the data within $\pm$ 10 per cent.

The fully-developed nucleate boiling data were correlated by a relation similar to that used by Russian investigators and is given as

$$h_{fdb} = 1.67 \quad (q'')^{0.65} \; p^{0.55} \qquad (5)$$

where

h_{fdb} = heat-transfer coefficient for fully-developed boiling, W m^{-2} K^{-1}

q'' = heat flux, W m^{-2}

p = system pressure, bar

This equation correlates the data within $\pm$ 36 per cent.

Local boiling data were compared to that predicted by the method of Berles and Rohsenow (ref. 12) presented in equation (7). The agreement between the measured and preducted values in excellent.

Finally, it is important to note that the correlations are applicable only in the range for which they were derived. In particular, equation (5) applies for pressures from 10.0 to 17.5 bar and mass velocities from 1.57×10^3 to 2.55×10^3 kg m^{-2} s^{-1}. Furthermore, the average fluid enthalpy should be subcooled or only slightly above the saturated value. Extrapolation to values beyond these limits may lead to erroneous results.

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