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Nucleate pool boiling of refrigerant-oil mixtures from tubes

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NUCLEATE POOL BOILING OF REFRIGERANT-OIL
MIXTURES FROM TUBES

by

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ABSTRACT

This experimental investigation was conducted to provide design heat transfer data for applications involving the nucleate pool boiling of Refrigerants 11 and 113 from the external surfaces of tubes. Tests were conducted with 0.625-in. o.d. and 1.125-in. o.d. commercial copper tubing; at 1 and 2 atmospheres pressure; and with refrigerant-oil compositions from 0 to 10 percent oil by weight. Results show that each of these parameters can affect the heat transfer coefficient.

ACKNOWLEDGMENT

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I. INTRODUCTION

In vapor-compression refrigeration systems using oil-lubricated compressors, there is always some amount of oil transported through the system. In general, this oil affects the boiling process occurring in the evaporator of such a system.

Many investigations have been conducted in the field of boiling heat transfer; however, there remains a need for both additional and improved design heat transfer coefficients for the nucleate pool boiling of refrigerants from the external surfaces of tubes. "Pool boiling" is defined as vaporization occurring under the following set of conditions: (a) the depth of the liquid is large compared to the maximum bubble diameter; (b) any externally imposed velocity of the fluid is low enough to have negligible effect on heat transfer, and (c) a force field, usually gravity, exists to make the bubbles move away from the heated surface. It is customary to express heat flow by boiling from the heated surface to the surrounding fluid in terms of a heat-transfer coefficient, h, defined by the following relation:

$$h \equiv \frac{Q}{A(t_w - t_s)} \qquad (1)$$

where: Q = heat transfer rate

A = surface area

t_w = wall or surface temperature

t_s = saturation temperature of boiling fluid.

Currently available information applicable to the design of such refrigeration heat exchangers as flooded evaporators consists principally of the works of Stephan[1], Rohsenow[2], Myers and Katz[3], and Blatt and Adt[4] as summarized in the 1972 ASHRAE Handbook of Fundamentals[5]. However, these investigations have mainly dealt with pure refrigerants and with specially prepared test surfaces.

Some previous works related to the effect of oil on heat transfer during evaporation have been reported. Stephan[6] reports results for the boiling of R-12 and R-22 from a flat plate with varying concentrations of oil. Tschernobylski and Ratiani[7] performed experiments with oil-R-12 mixtures boiling from a 14 mm od horizontal tube. Worsoe-Schmidt[8], Green[9], and Green and Furse[10] have studied the effect of oil on the boiling of R-12 from the inside of horizontal tubes. Furse[11] also reports results for R-11 and R-12 boiling over a flat horizontal copper surface.

The investigation reported in this paper was directed toward extending the available heat transfer data on the boiling of refrigerant-oil mixtures. The results should provide additional thermal design data for the nucleate pool boiling of refrigerants from the external surfaces of tubes. The ranges of test parameters are as follows:

Refrigerants: R-11 and R-113

Oil and Concentrations: Paraffin base, 150 SSU @ 100°F 0 to 10% by weight

Boiling Surfaces: Commercial Copper Tubing,
Type L 0.625 in. o.d. and
1.125 in. o.d.

Evaporator Pressures: 1 ATM and 2 ATM

Heat Fluxes: 150 to 25000 Btu/hr sq ft.

II. EXPERIMENTAL APPARATUS

The equipment used for this study consisted principally of five basic systems: (a) test fluid and cylindrical test element, (b) boiling/condensing vessel system, (c) power supply, (d) auxiliary refrigeration unit for condenser, and (e) instrumentation and control system. Figure 1 shows schematically the arrangement of the various components. A more detailed sketch of the boiling/condensing system is given as Figure 2.

The boiling section consisted of a six-inch Pyrex glass tee containing the copper tube test section and the test fluid. Auxiliary cartridge heaters were inserted in the fluid and resistance heating tape was wrapped around the outside of the vessel. Heater tape was also wound around the condenser, and the complete boiling/condensing system was insulated. Power leads and thermocouples were brought into the chamber through Conax glands in each metal end plate.

The test specimen was constructed from commercial, type L, copper tubing. Except for cleaning with acetone, the surface finish was otherwise in "as received" condition. The two sizes of test sections used were 5/8 in. and 1-1/8 in. o.d., each 7-1/2 in. in length. A standard cartridge heater was used as the heating element inside each tube with machined copper sleeves filling the gap between the

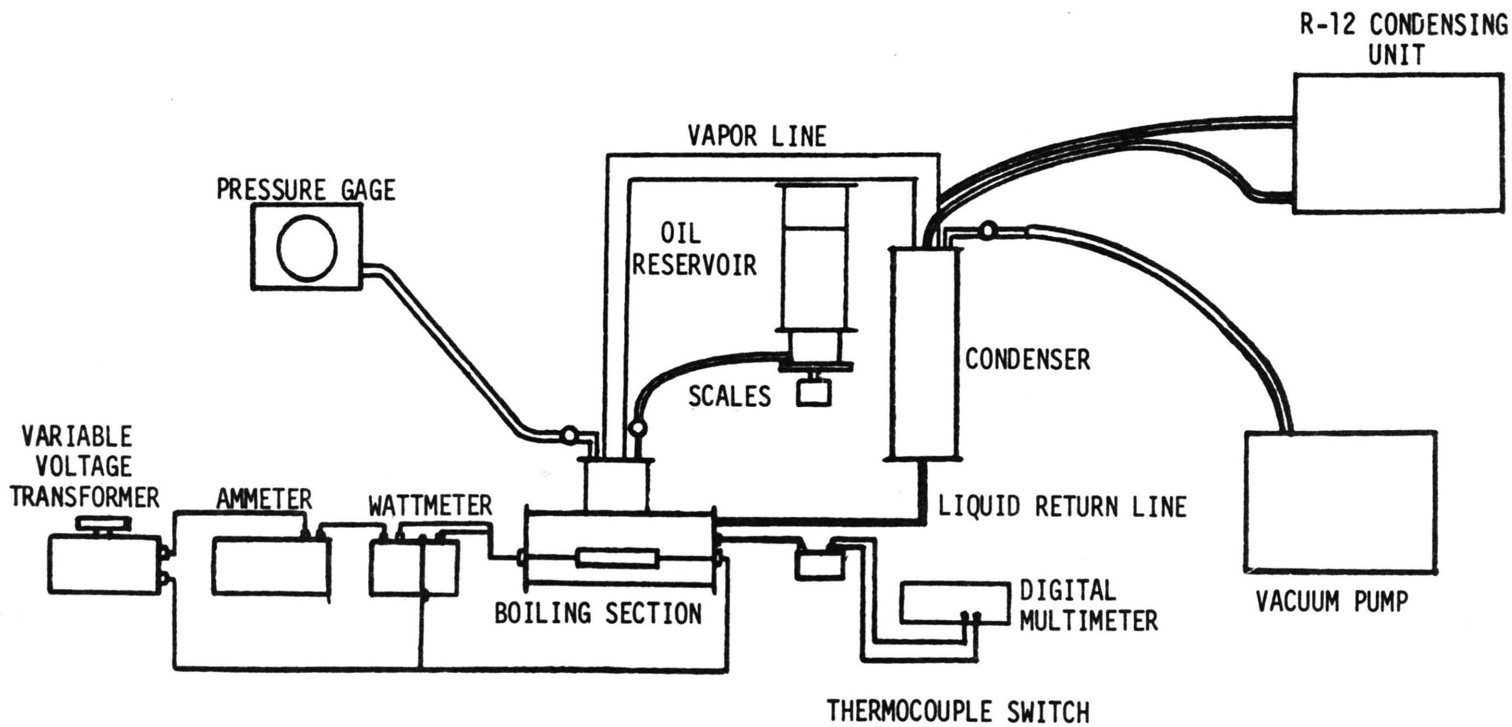

Figure 1. Schematic of Apparatus.

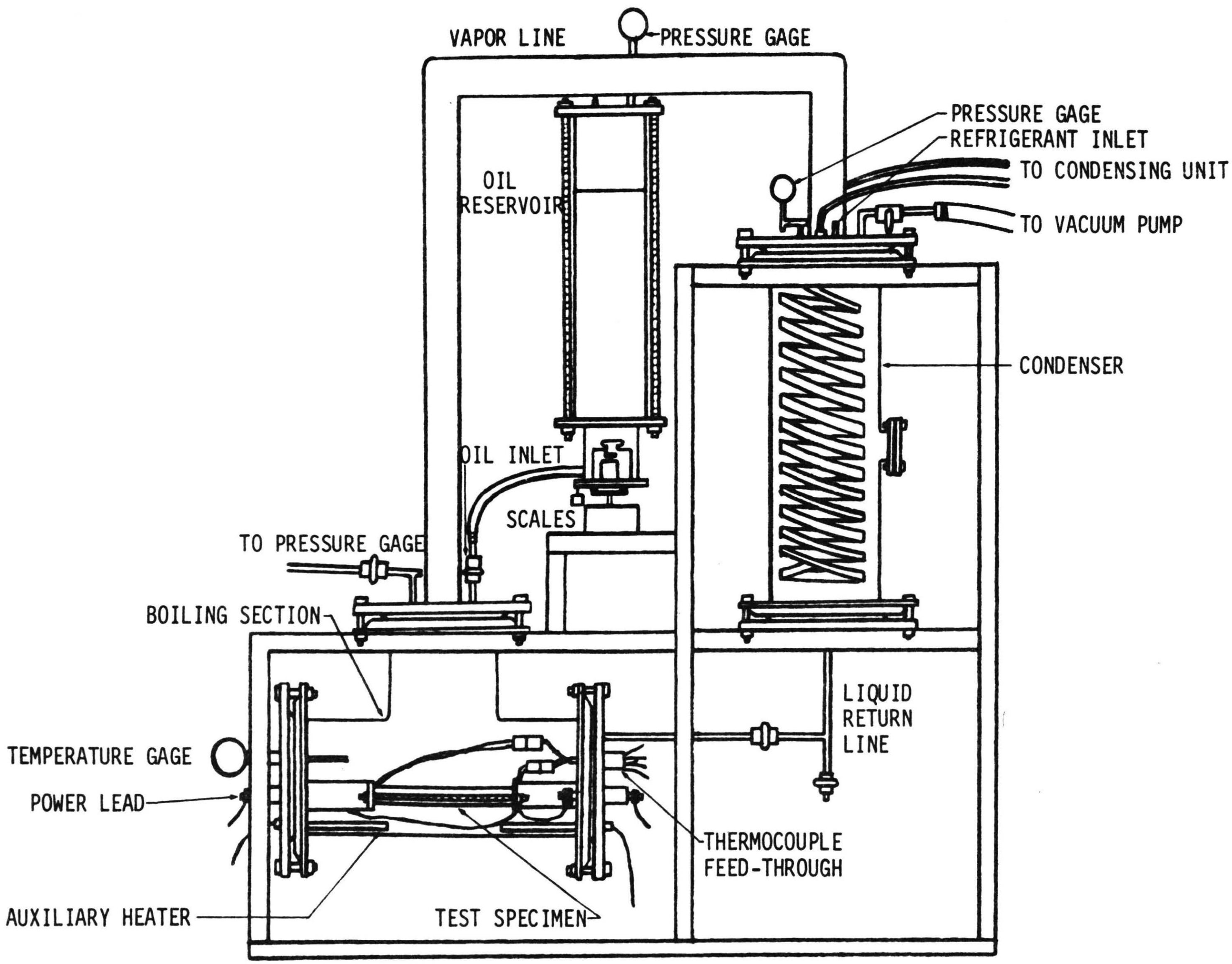

Figure 2. Boiling/Condensing System.

cartridge and the tube. Six copper-constantan, 32-gauge, thermocouples were soldered in slots between the sleeve and the tube. Reference junctions were submerged in the boiling fluid. Although the thermocouple measurements were obtained from the inside of the copper tube, these values were extrapolated to the outer surface using Fourier's equation for radial heat flow. The heater assembly was sealed on each end with Teflon caps to reduce axial heat losses. The Teflon caps were cemented to the assembly as well as mechanically clamped in place.

A pressurized reservoir mounted on scales contained the oil. A separate R-12 condensing unit provided cooling for the condensor section. AC power to the test heater assembly was provided through a large variable voltage transformer. A precision wattmeter was used for determining the energy supplied to the test section.

III. TEST PROCEDURE

The system was initially evacuated and then the boiling section was filled with a known (by weight) amount of the pure refrigerant. The auxiliary heaters were used to degas the refrigerant prior to actual test runs and to bring the test fluid to its saturation temperature. Test pressure was obtained and maintained by manually controlling the secondary refrigeration unit for the condenser, and/or through the power supplied to the heating and/or auxiliary heaters.

Power was supplied to the test cylinder at a low level for the first test point. The power level was increased for each succeeding data point until the arbitrarily pre-selected maximum heat flux of 25000 Btu/hr sq ft was reached. The power level was then reduced with data taken at intervals of decreasing power. At each power setting, the system was allowed to come to quasi-equilibrium and the following data taken: Wattmeter reading, tube thermocouple emfs, saturation temperature, and chamber pressure.

This procedure was followed for the oil-free runs at pressures of one and two atmospheres and then 1% by weight of oil was added to the refrigerant and mixed using the auxiliary heaters. Data points were obtained over the same general range as for the oil-free refrigerant at each pressure. Testing was similarly conducted for the other oil concentrations of 2, 3, 5, 7, and 10 percent by weight.

After completing this series of tests for the first refrigerant, the fluid was removed and the apparatus cleaned. The same procedure was then followed for the second refrigerant. Upon completion of these tests, the apparatus was disassembled, cleaned, another sized heater assembly installed, and the previous procedures again repeated.

IV. RESULTS

The results are presented in graphical form and are spaced throughout the DISCUSSION OF RESULTS for ease of reference.

Figures 3 and 4 are comparisons of (0% oil concentration) results of this experimentation to several boiling heat transfer correlations of the literature. Figures 5 through 12 present the experimental results, and Figures 13 through 20 are comparisons among the experimental results with respect to oil concentration.

Figures 5 through 12 are identified by the parameters: oil concentration, refrigerant, heater size, and nominal boiling pressure. The actual range of boiling pressures is given as follows:

Figure	Pressure Range (lbf/sq in, abs)
5	16.0-16.9
6,9,10	14.6-14.9
7	31.0-31.7
8,11,12	29.3-29.6

V. DISCUSSION OF RESULTS

For each data point, the six tube thermocouple readings were averaged and used to obtain the temperature difference between the tube surface and the saturation temperature of the boiling fluid. The heat transfer rate was obtained directly from the wattmeter measurement. Results of previous work have indicated that maximum error due to neglecting the axial heat flow through the Teflon end caps is less than 4 percent. Surface area was obtained from dimensional measurements of the test cylinder. Equation (1) then provides the value for the heat transfer coefficient, h. The oil concentration is defined as the ratio: Weight of oil divided by weight of oil and refrigerant.

Figures 3 and 4 present typical results of heat transfer coefficient versus heat flux density for oil-free Refrigerant-11 in comparison to the results of Stephan[(1)], Rohsenow[(2)], Forster and Zuber[(12)], McNeilly[(13)], Gilmour[(14)], Borishanskiy and Minchenko[(15)], and Kutateladze[(16)].

Figure 3 gives the results for R-11 at 1 atm pressure and with the 1.125 in. o.d. tube. Figure 4 is also for R-11 but at 2 atmospheres and for the 0.625 in. o.d. tube. As can be seen from both figures, the current experimental results fall within the range of the various predictive equations. Of these equations, the Borishanskiy-Minchenko correlation appears to agree best with the experimental data. The experimental data does not substantiate the value

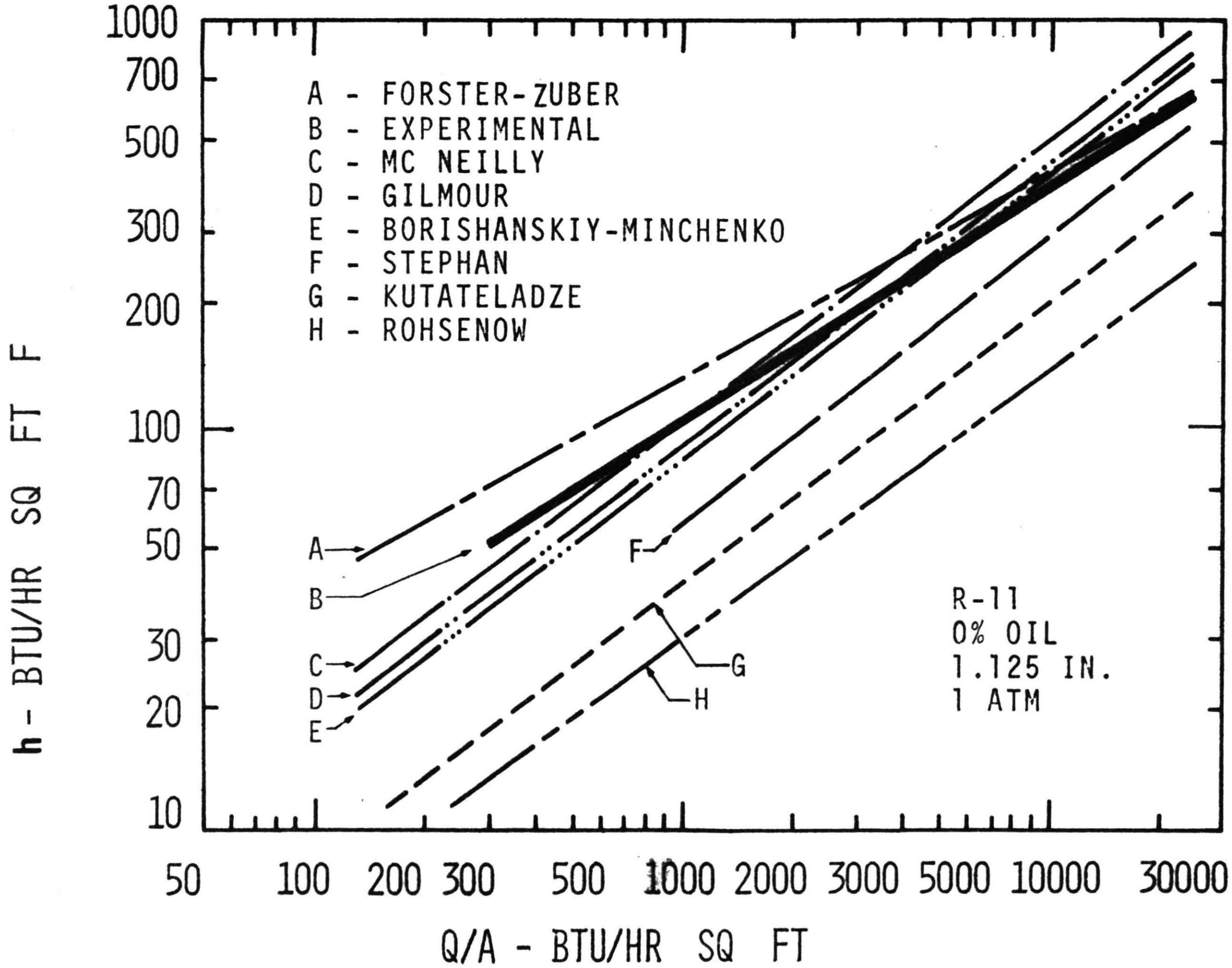

Figure 3. Comparison of Several Boiling Correlations to Experimental Data for Oil-free R-11 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -1 ATM).

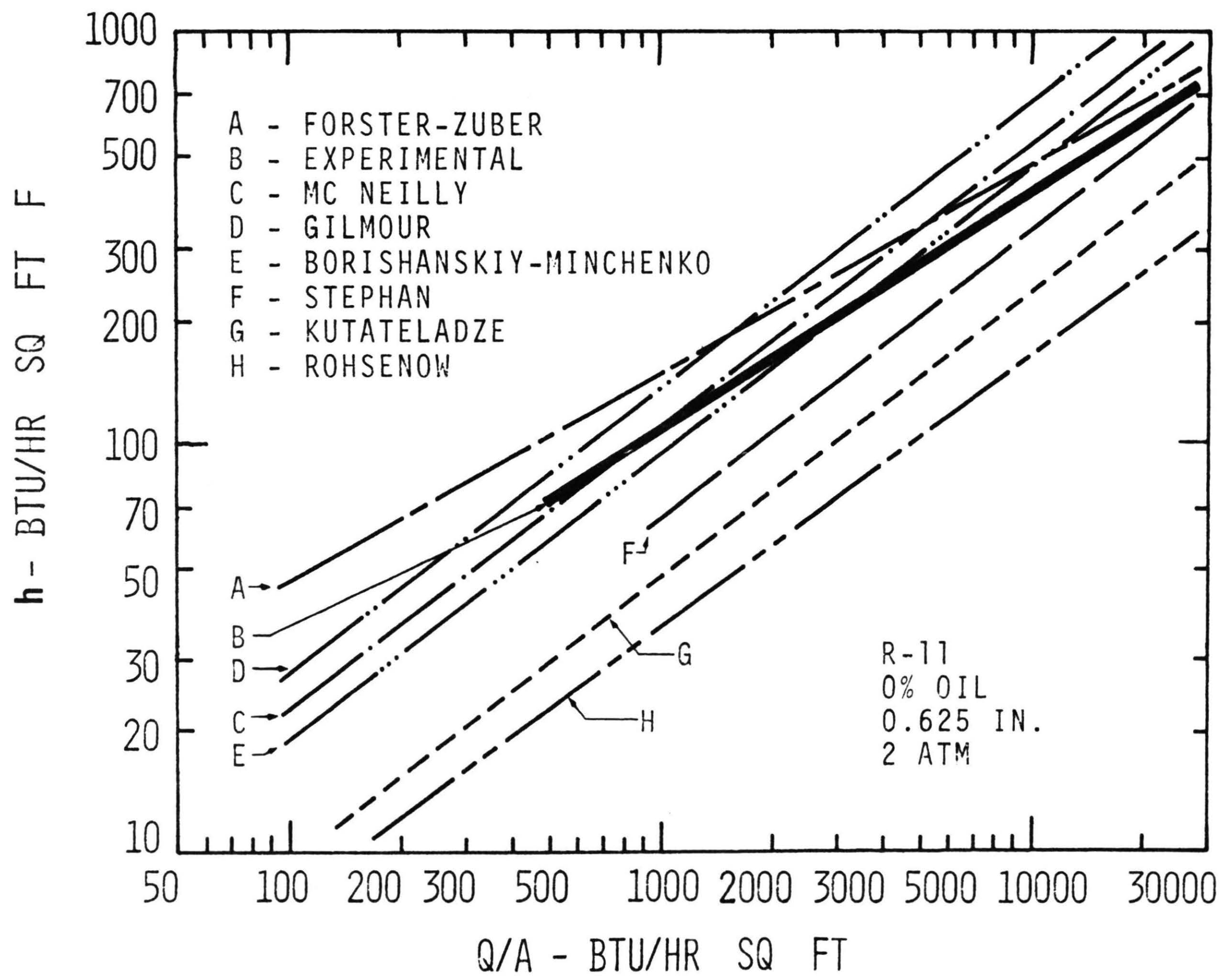

Figure 4. Comparison of Several Boiling Correlations to Experimental Data for Oil-Free R-11 Boiling from a 0.625 Inch o.d. Copper Cylinder (Saturation Pressure -2 ATM).

of C_{s_f} = 0.022 obtained from the data of Blatt and Adt[4] for the Rohsenow correlation but yields values ranging from 0.0067 to 0.009 for the R-11 copper combination of this investigation.

Figures 5-12 present the complete results of all tests covering the two refrigerants, R-11 and R-113; the two tube sizes, 5/8 in. o.d. and 1-1/8 in. o.d.; the two pressures, 1 atm and 2 atm; and oil contents of 0, 1, 3, 5, 7, and 10 percent by weight. Results are presented for each as heat transfer coefficient versus temperature difference. A summary of these figures is presented below:

Figure No.	Refrigerant	Pressure (atm)	Tube Size (in.o.d.)
5	11	1	1.125
6	11	1	0.625
7	11	2	1.125
8	11	2	0.625
9	113	1	1.125
10	113	1	0.625
11	113	2	1.125
12	113	2	0.625.

Each of the curves in these figures shows a strong hysteresis effect as the power level is decreased. This hysteresis effect is due to the greater heat fluxes needed to initiate boiling at sites along the tube as compared to the much lower heat fluxes needed to continue the boiling

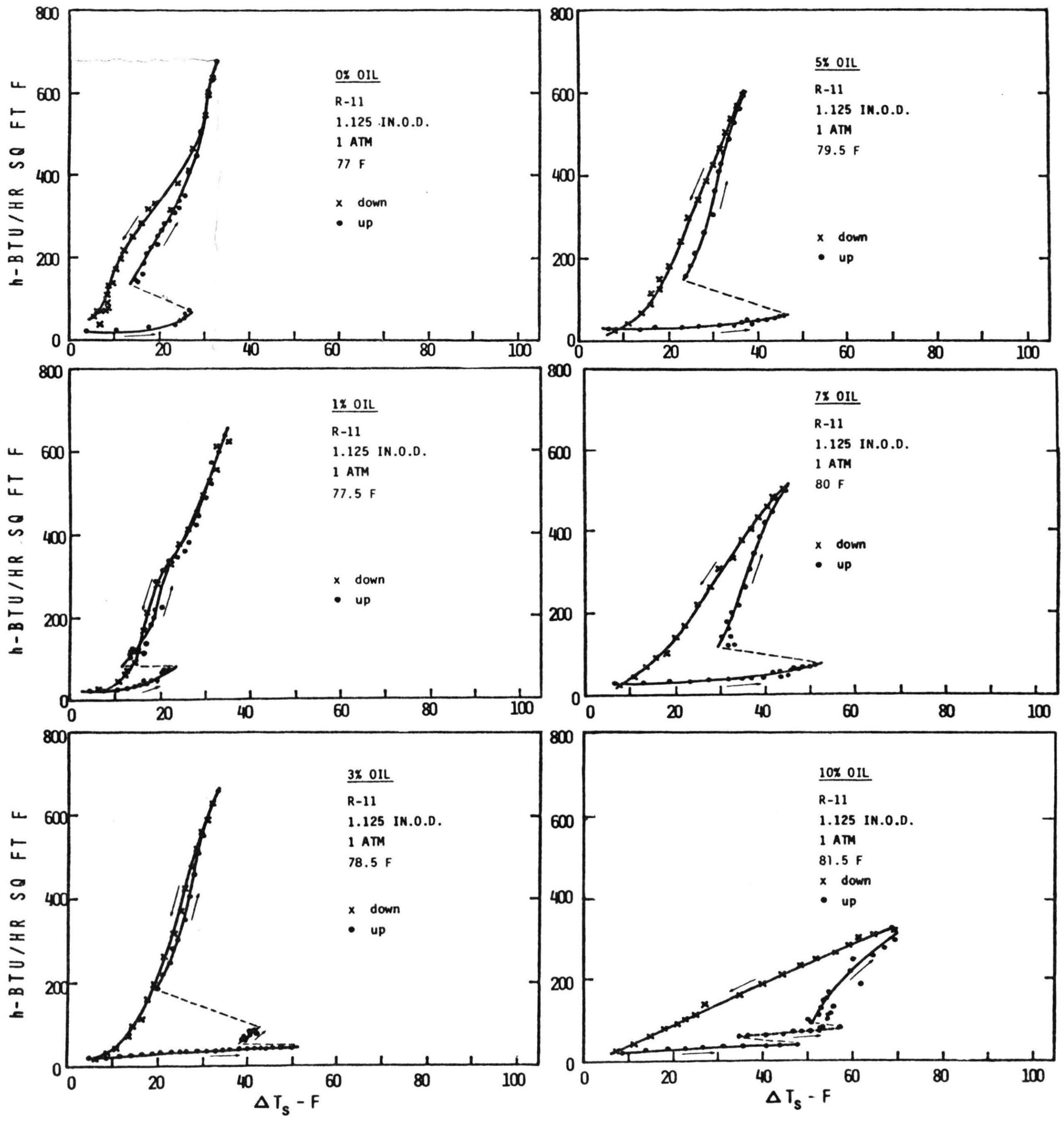

Figure 5. Effect of Oil on R-11 Boiling from 1.125 Inch o.d. Copper Cylinder at One Atmosphere (Saturation Pressure).

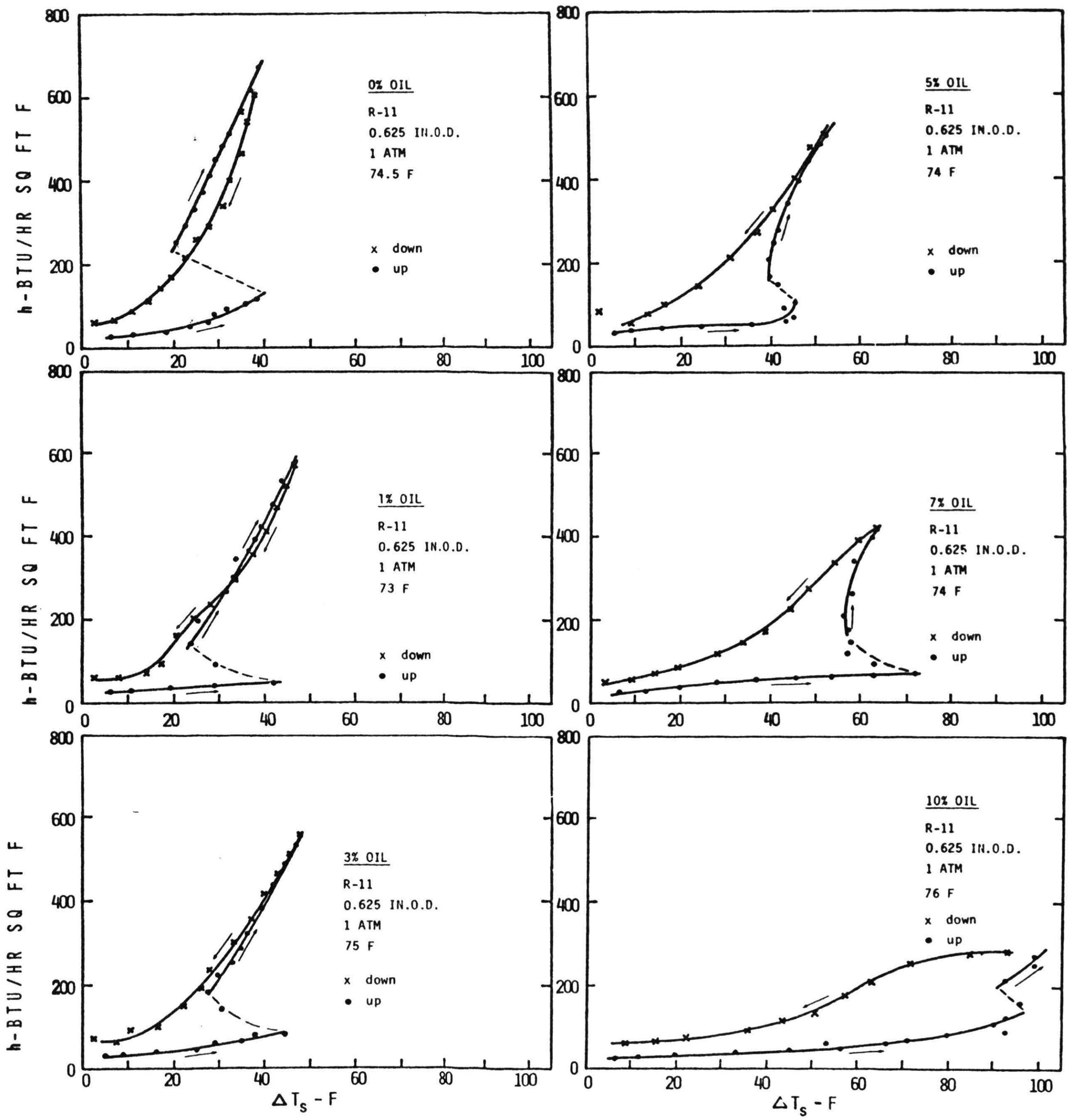

Figure 6. Effect of Oil on R-11 Boiling from 0.625 Inch o.d. Copper Cylinder at One Atmosphere (Saturation Pressure).

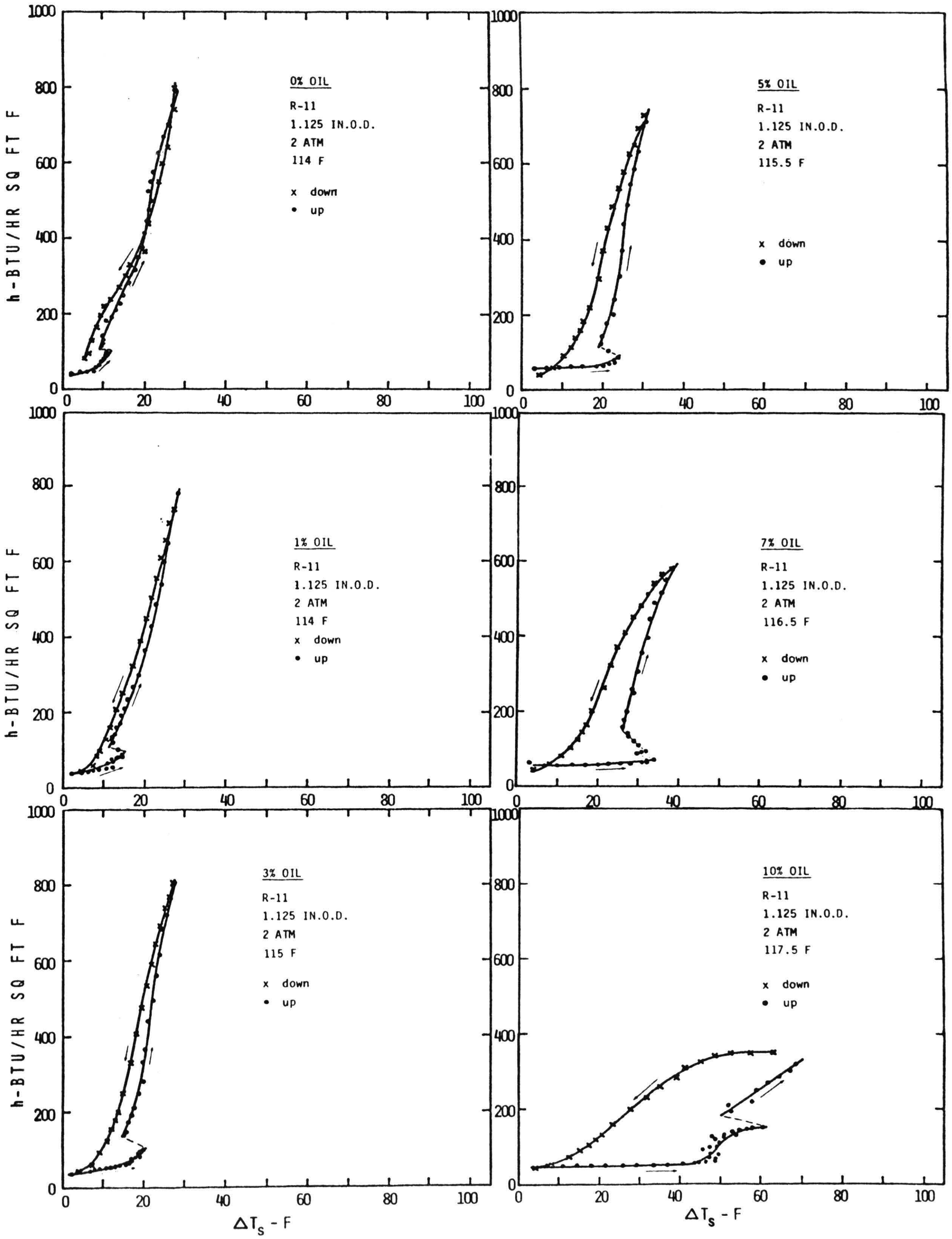

Figure 7. Effect of Oil on R-11 Boiling from 1.125 Inch o.d. Copper Cylinder at Two Atmospheres (Saturation Pressure).

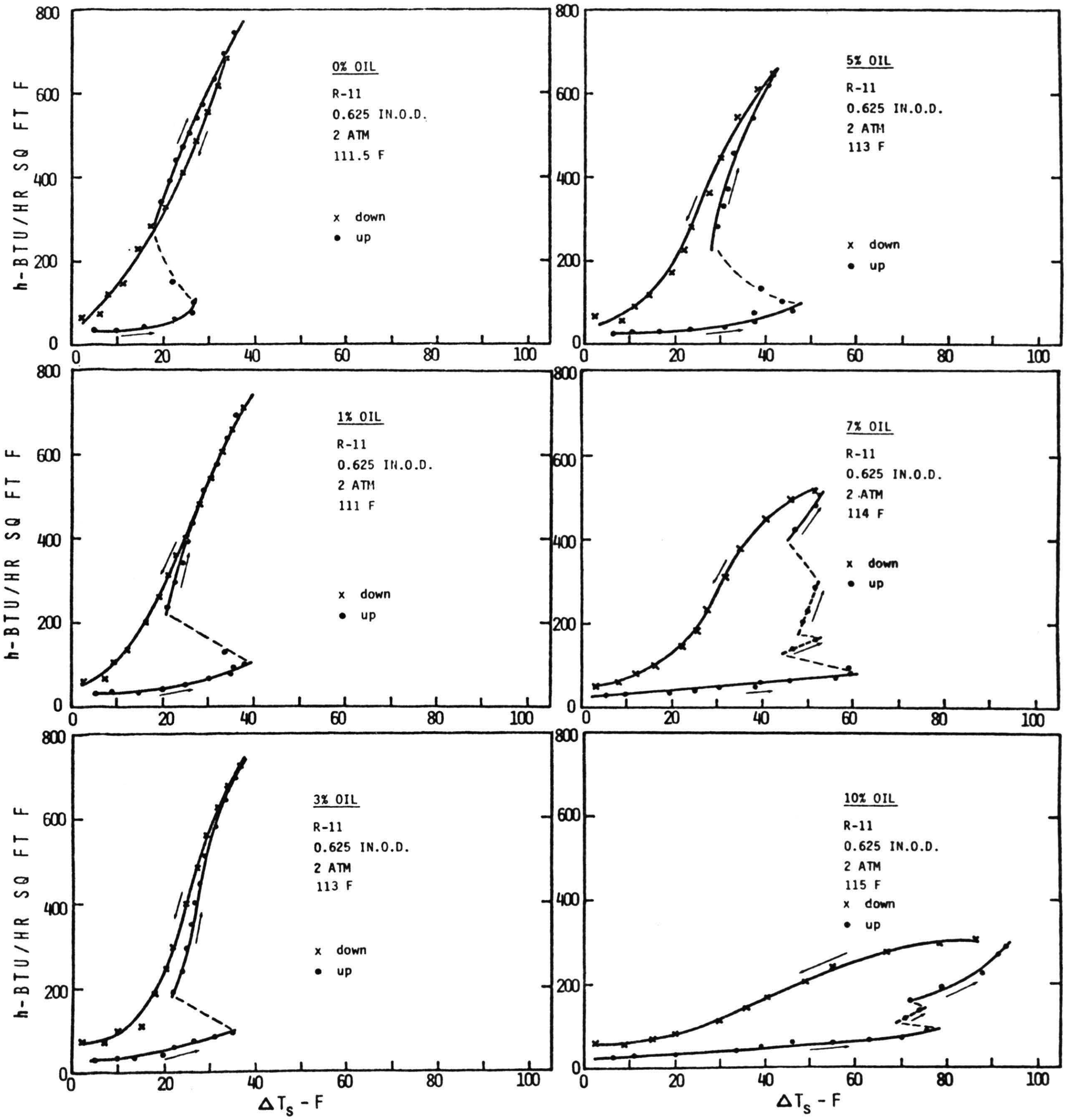

Figure 8. Effect of Oil on R-11 Boiling from 0.625 Inch o.d. Copper Cylinder at Two Atmospheres (Saturation Pressure).

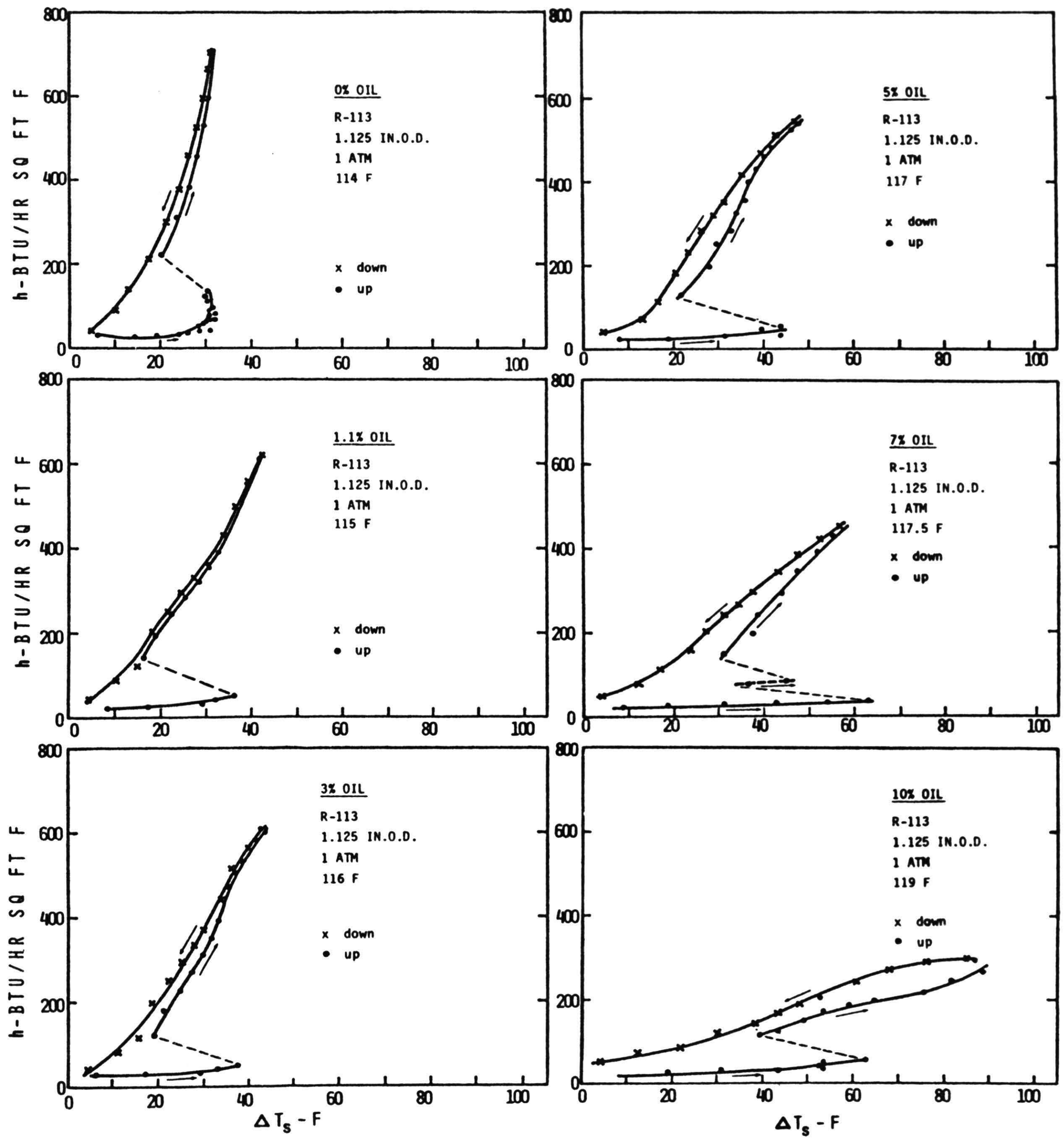

Figure 9. Effect of Oil on R-113 Boiling from 1.125 Inch o.d. Copper Cylinder at One Atmosphere (Saturation Pressure).

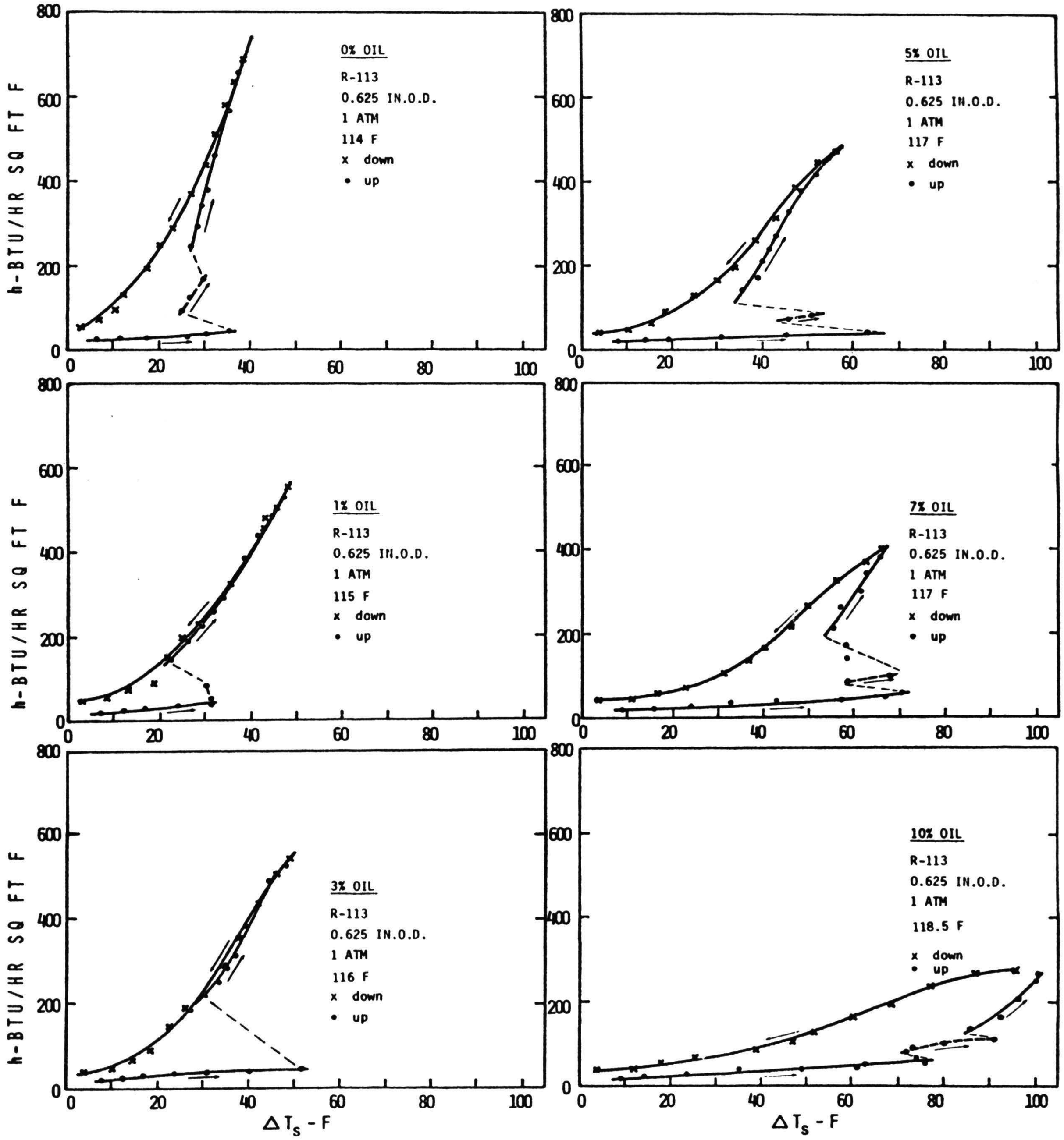

Figure 10. Effect of Oil on R-113 Boiling from 0.625 Inch o.d. Copper Cylinder at One Atmosphere (Saturation Pressure).

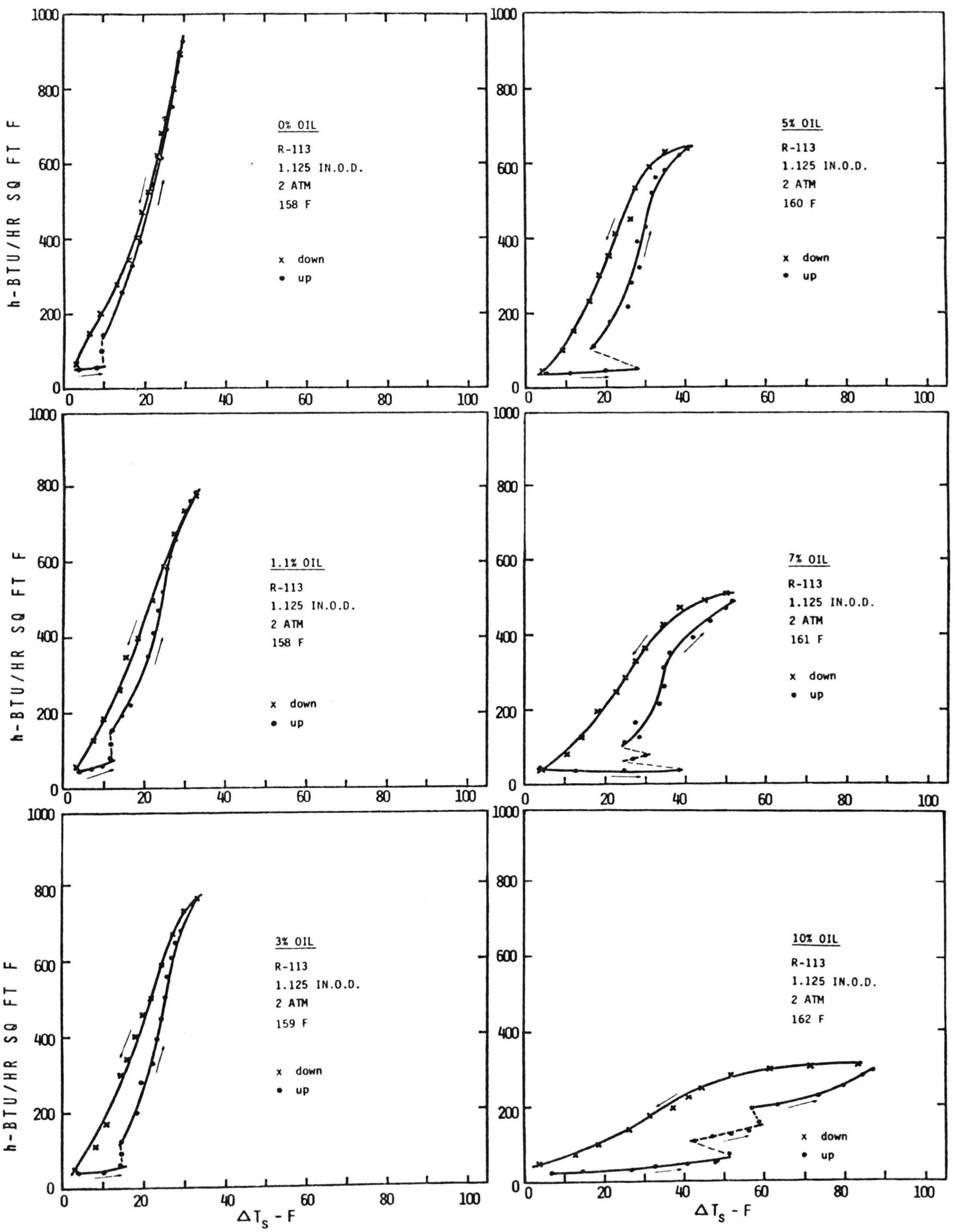

Figure 11. Effect of Oil on R-113 Boiling from 1.125 Inch o.d. Copper Cylinder at Two Atmospheres (Saturation Pressure).

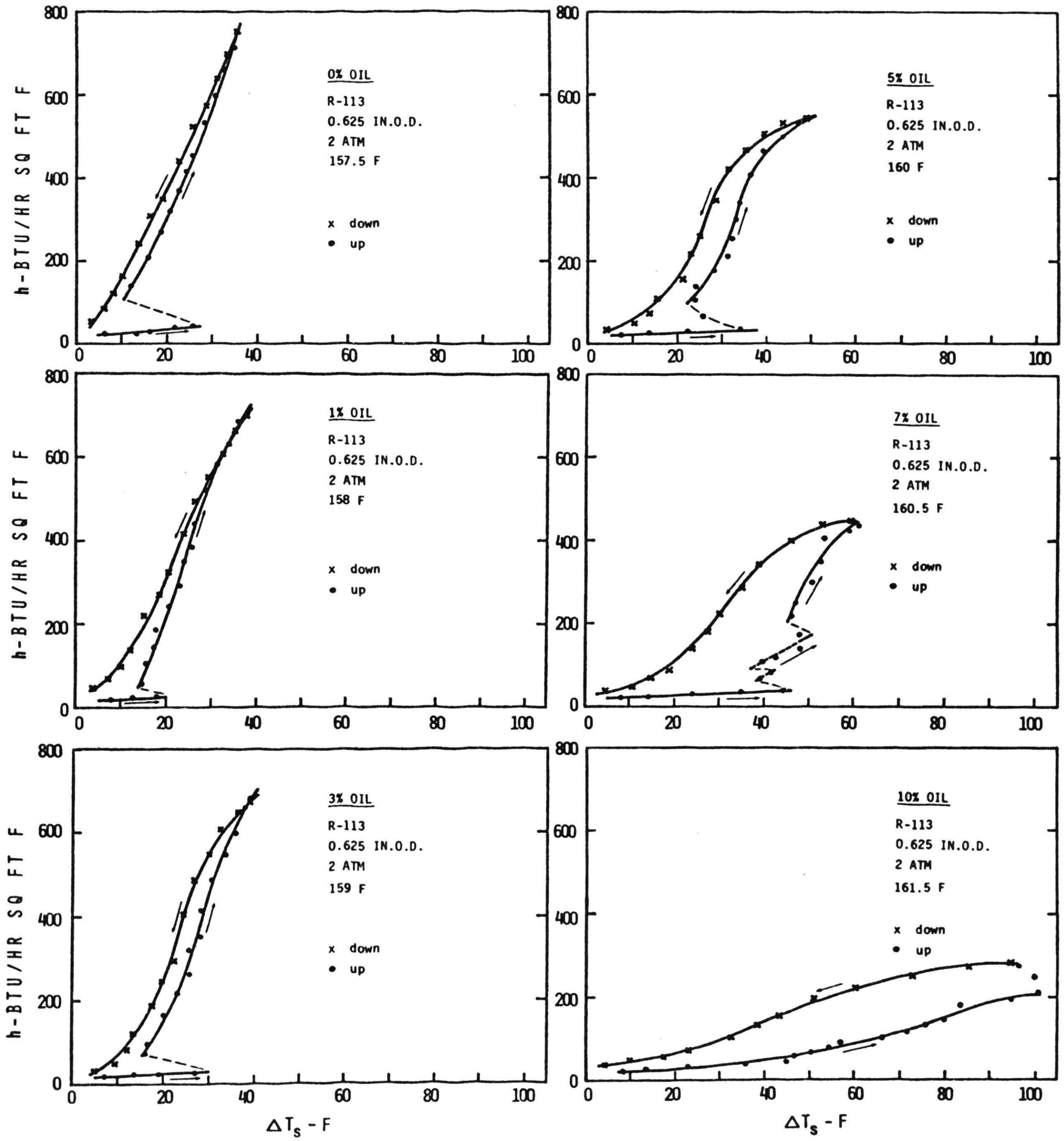

Figure 12. Effect of Oil on R-113 Boiling from 0.625 Inch o.d. Copper Cylinder at Two Atmospheres (Saturation Pressure).

once it has been initiated. In each of Figures 5-12 there is a discontinuity in the increasing power curve denoted by a dashed line. This discontinuity represents the transition from natural convection to boiling. Since, for the same heat flux, the heat transfer due to boiling occurs at a much faster rate per degree of temperature difference than that due to natural convection, a sudden decrease in the temperature difference and increase in the heat transfer coefficient accompanied the transition from convection to boiling.

From Figures 5 through 12 it can be seen that at low oil concentrations the transition occurred in a single "jump". However, at higher oil concentrations, the transition occurred in several small jumps. These small jumps are due to the fact that, at higher oil concentrations, boiling was initiated over less of the tube surface at one time than at the lower oil-refrigerant concentrations. Random sections of the tube began to boil at different times causing a discontinuity or jump each time a new section of the tube began to boil.

This resistance of the oil-refrigerant mixture to boiling was noted and explained by Stephan[6]. He stated that the surface tension and adhesion of the liquid oil-refrigerant mixture near a heated surface is much greater than that of the refrigerant-vapor bubbles forming on the surface. Therefore, the oil-refrigerant mixture resists bubble formation.

Figures 13 through 20 present the results as heat transfer coefficient versus heat flux density for the same parameters and in the same sequence as in Figures 5 through 12.

In Figures 13-20, the single line for each oil concentration shows the values for decreasing heat flux only. From these figures, it can be seen that oil concentrations of 3% or less do not greatly affect the heat transfer coefficient; and that in some cases, as Stephan[6] has also reported, the heat transfer coefficient may increase as oil concentration goes from 0% to 3%. Generally, however, oil concentrations larger than 3% cause definite decreases in the heat transfer coefficients and in the slopes of the curves for heat transfer coefficient versus heat flux density. These trends agree with Stephan's results for Refrigerants 12 and 22.

It was noted that the increase in heat transfer coefficient with increasing saturation temperature was less pronounced at high oil concentrations than at low oil concentrations. For increasing saturation temperatures at 0% oil concentration, the heat transfer coefficient increased a maximum of 55%. But over the same saturation temperature range at 10% oil concentration, the heat transfer coefficient increased a maximum of 40%. This indicates that, as Stephan[6] reported, at higher oil concentrations the heat transfer coefficient tends to be less affected by a change in the boiling temperature of the refrigerant-oil mixture.

In contrast to the results of Stephan[1] the cylinder diameter did affect the heat transfer coefficient. Over the range of conditions for this investigation, the increase in diameter from 0.625 in. o.d. to 1.125 in. o.d. resulted in increases in the heat transfer coefficient by

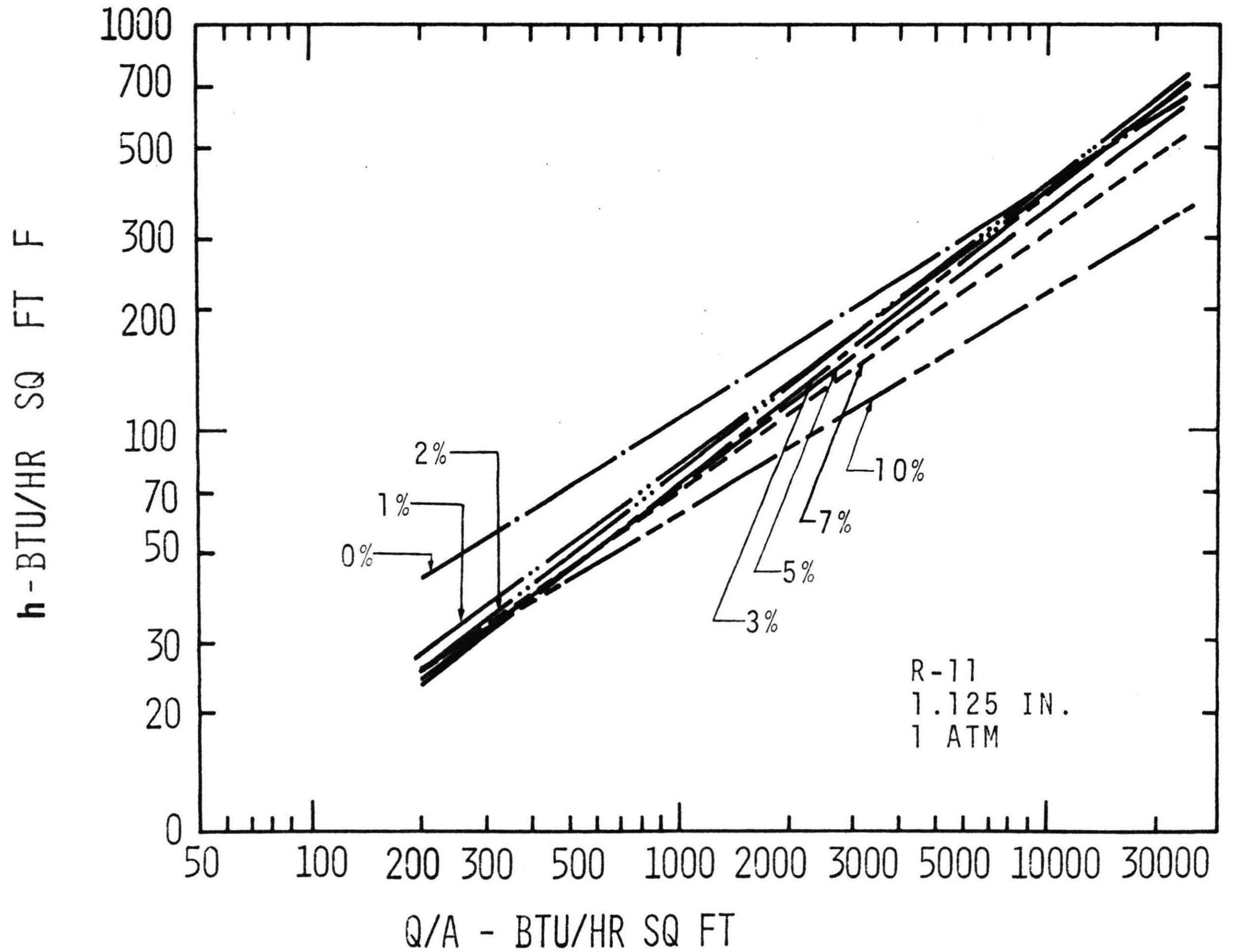

Figure 13. Effect of Oil on R-11 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -1 ATM).

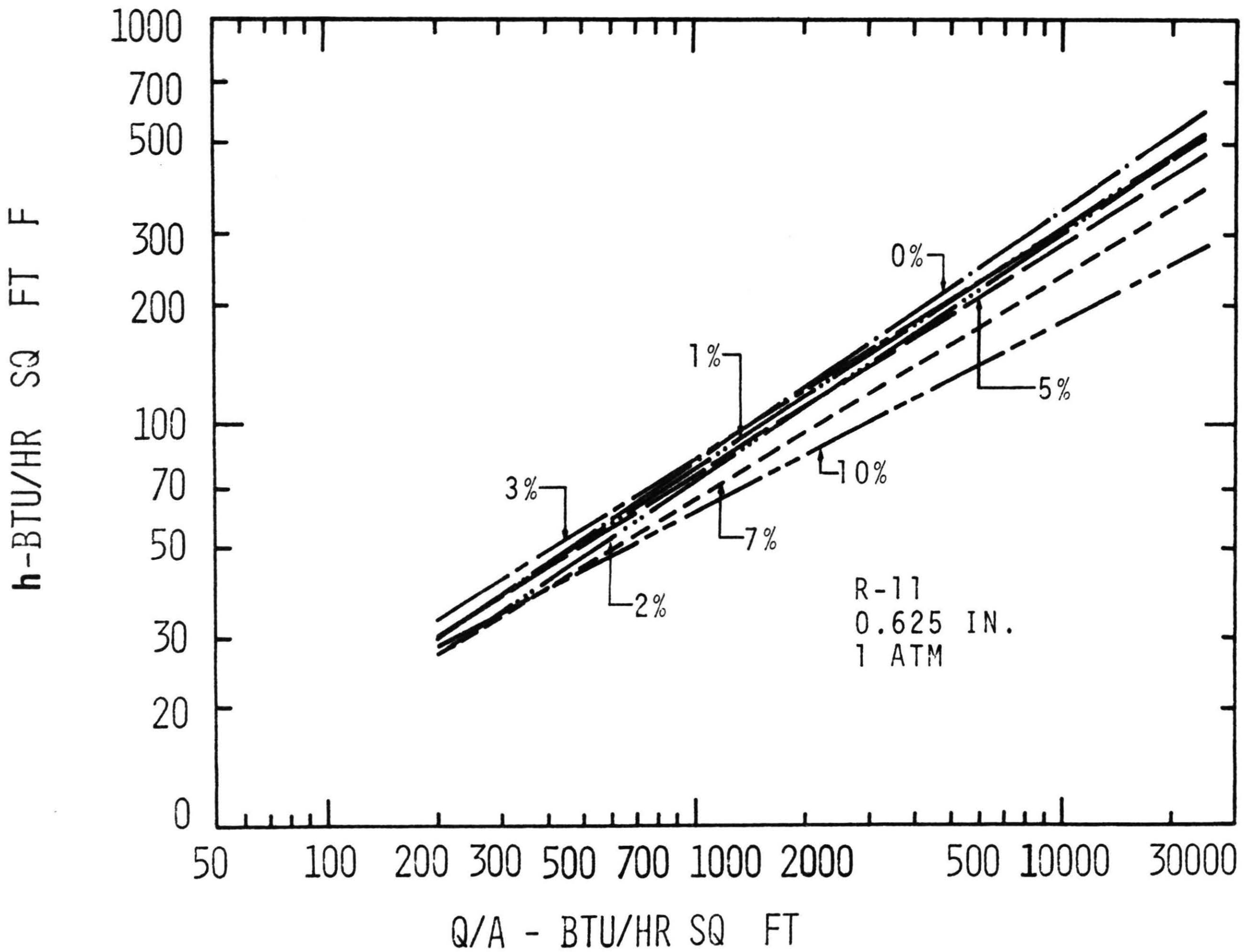

Figure 14. Effect of Oil on R-11 Boiling from 0.625 Inch o.d. Copper Cylinder (Saturation Pressure -1 ATM).

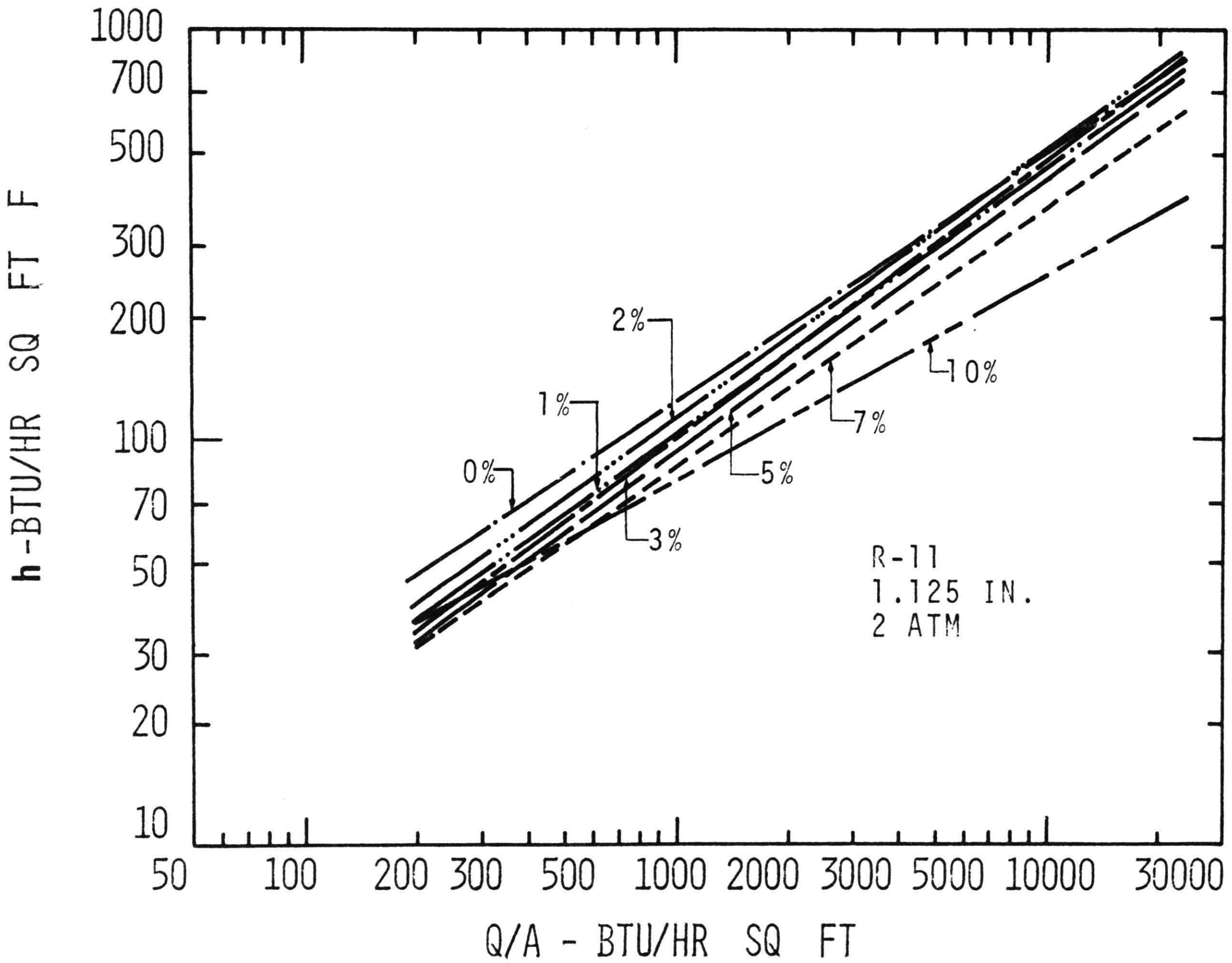

Figure 15. Effect of Oil on R-11 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -2 ATM).

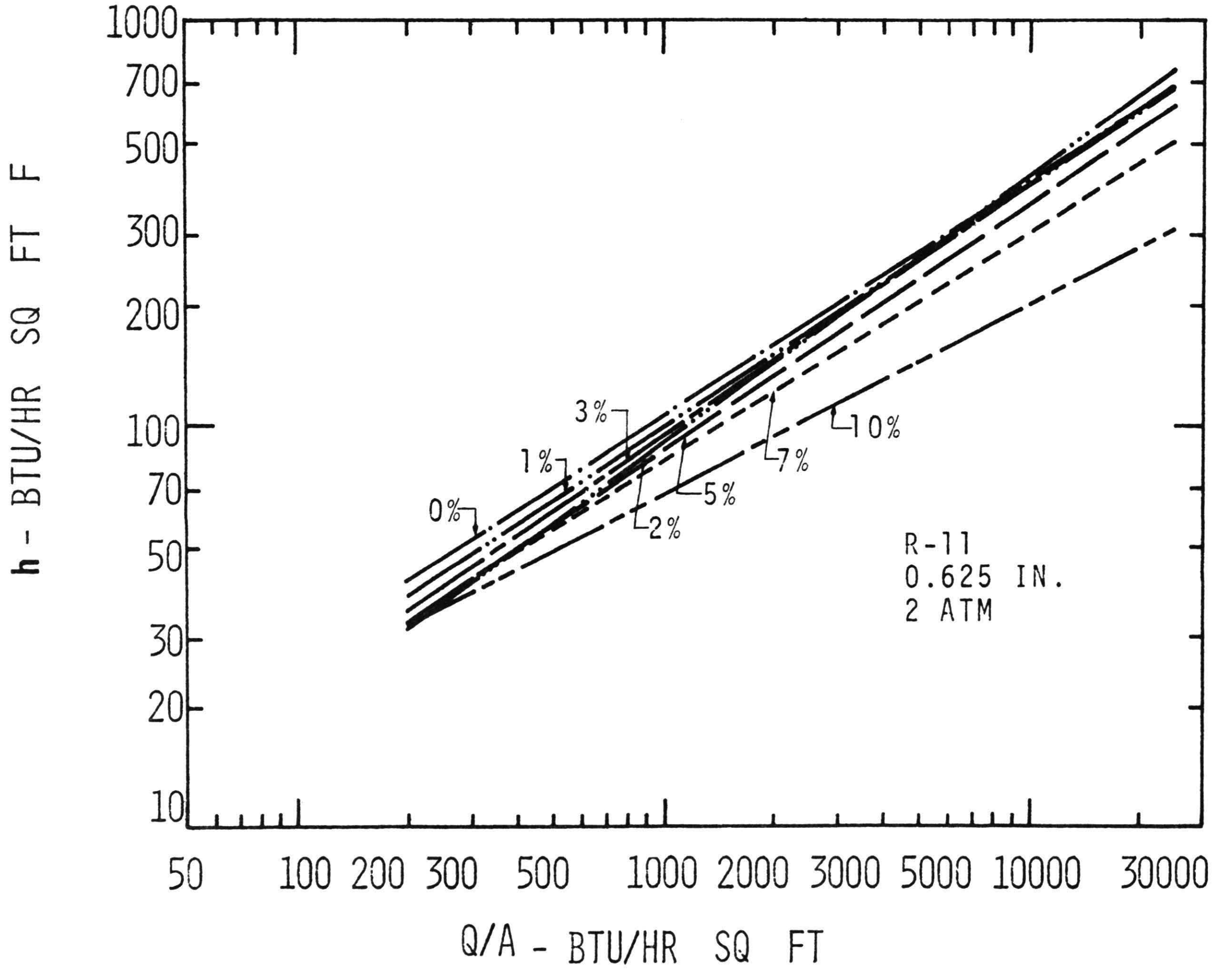

Figure 16. Effect of Oil on R-11 Boiling from a 0.625 Inch o.d. Copper Cylinder (Saturation Pressure -2 ATM).

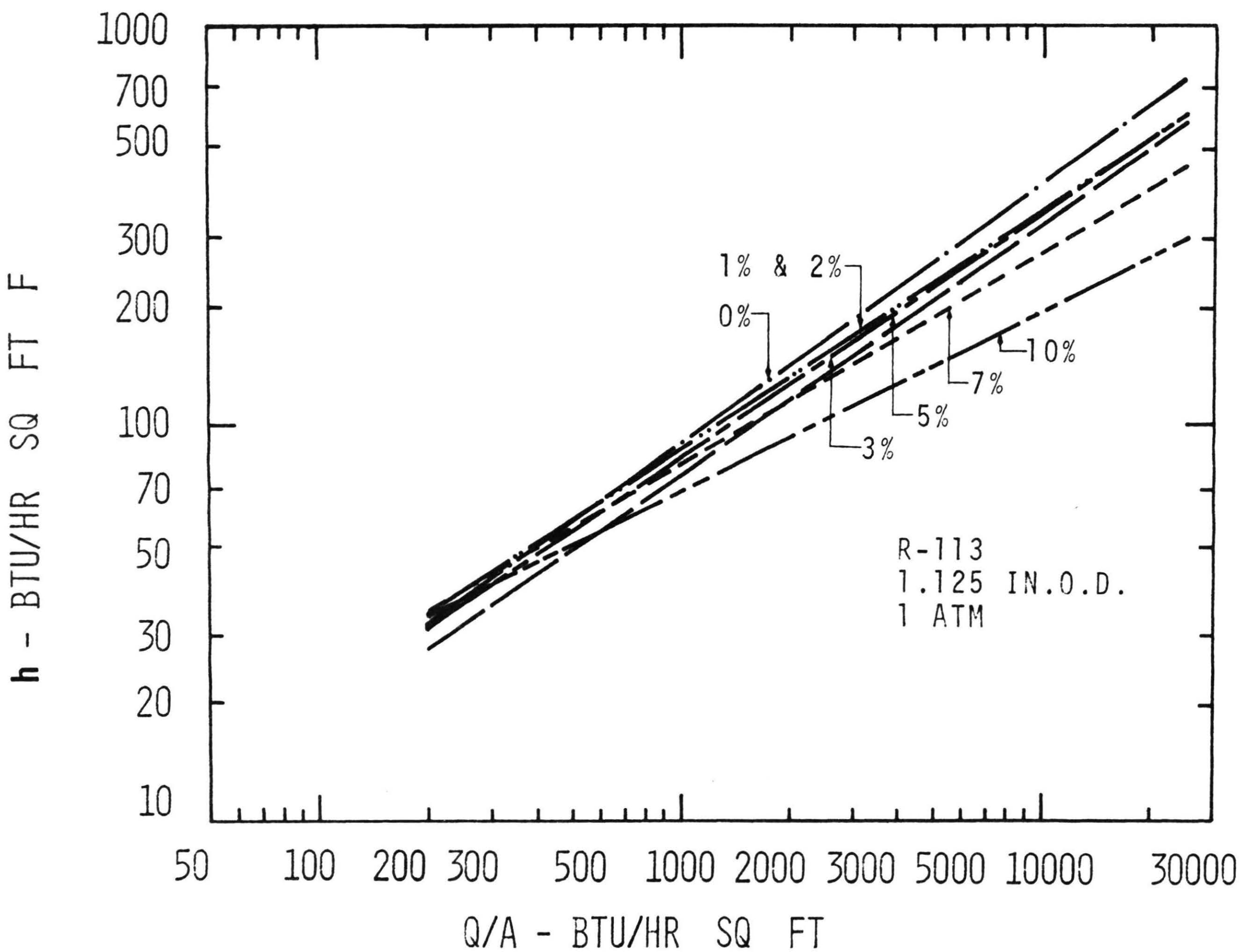

Figure 17. Effect of Oil on R-113 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -1 ATM).

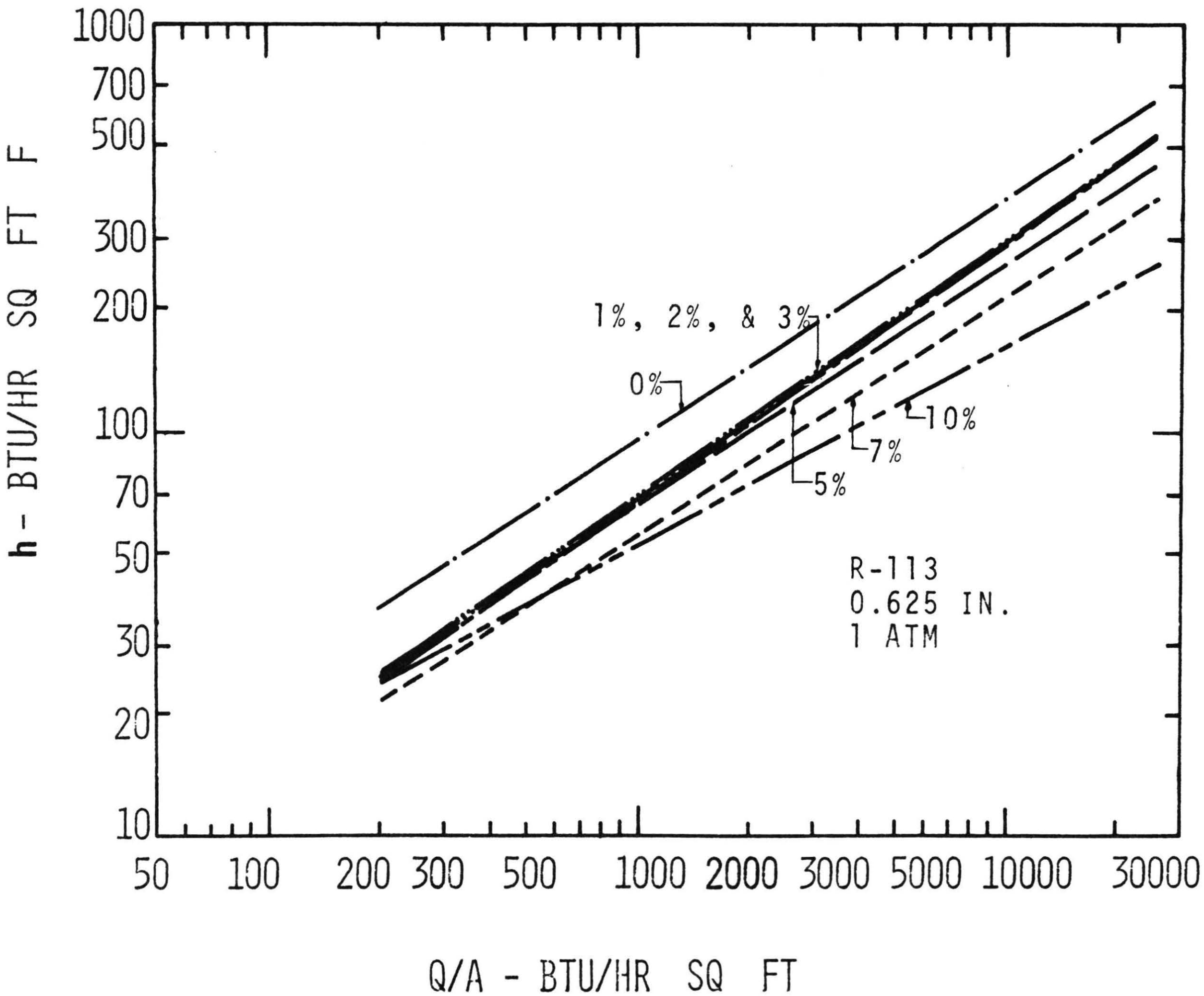

Figure 18. Effect of Oil on R-113 Boiling from a 0.625 Inch o.d. Copper Cylinder (Saturation Pressure -1 ATM).

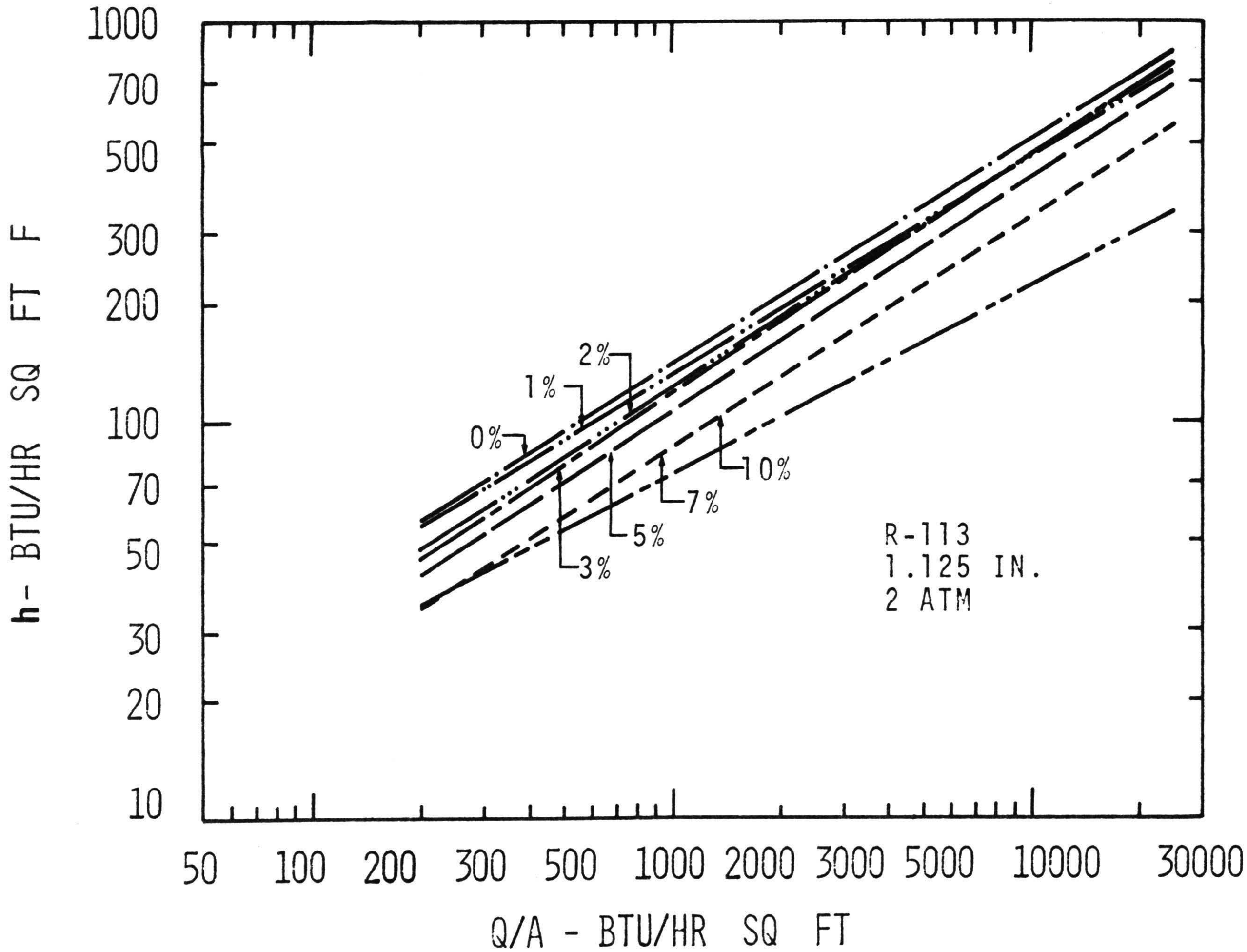

Figure 19. Effect of Oil on R-113 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -2 ATM).

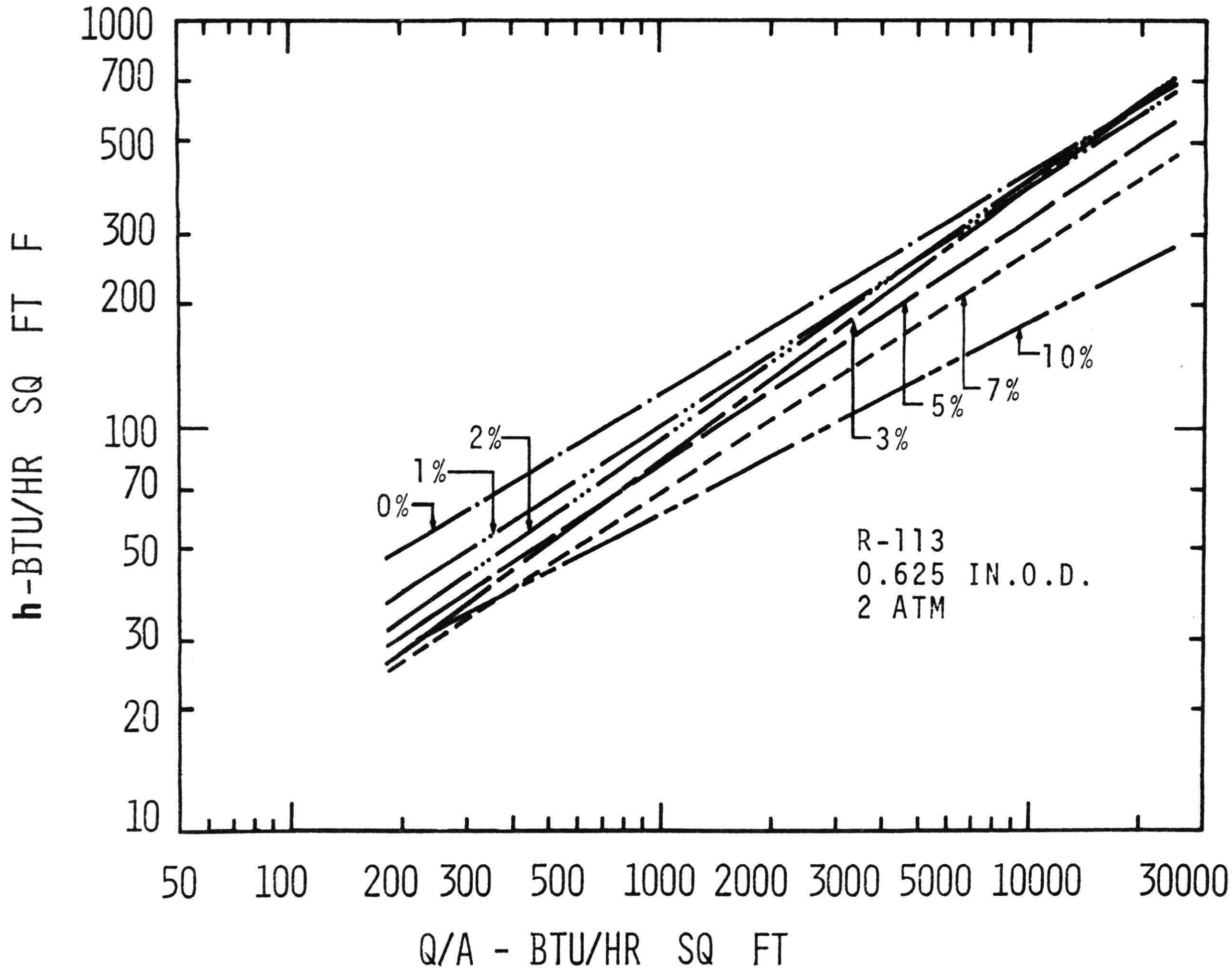

Figure 20. Effect of Oil on R-113 Boiling from a 1.125 Inch o.d. Copper Cylinder (Saturation Pressure -2 ATM).

20% to 60%. This, however, needs to be verified by tests with additional cylinders.

It might also be noted that as oil concentration was changed from 0% to 10%, the boiling temperature at a given pressure increased by approximately 4° F.

Since surface finish is known to have a decided effect on boiling heat transfer, the surface finishes of the two cylinders used in this investigation were measured. They were found to range over the surface as follows:

	RMS (μin)	AA (μin)
1.125 in.o.d.	8-20	6-25
0.625 in.o.d.	9-31	9-27

Since the maximum error due to instrumentation was less than 11% (see Appendix D), by far the maximum uncertainty in the results lies in the value of the surface temperature of the tube. As described previously, an average value was obtained from six local measurements. The temperature variation between local values was generally less than 6°F at low heat fluxes but became as high as 25°F when boiling was highly localized. This variation in temperature differences was most pronounced at high oil concentrations, and less noticeable at low oil concentrations. Boiling was also observed to occur more vigorously from the top of the tube than from the bottom indicating that the heat transfer coefficient does vary with angular position around the cylinder.

VI. CONCLUSIONS AND SUMMARY

In general, the addition of oil to boiling refrigerants causes a reduction in the heat transfer coefficient. However, for some conditions with small oil concentrations (3% or less), the heat transfer coefficient may even increase slightly. For oil concentrations greater than 3% the heat transfer coefficient always decreases with increased oil content, and for large oil concentrations (7%-10%) the heat transfer coefficient is greatly reduced, being as much as 60% less than that of the oil-free refrigerant.

In the design of flooded evaporators having 3% or less oil concentrations, the heat transfer coefficients for oil-free refrigerants could be used with only slight error. However, for larger oil-concentrations it is necessary to use the reduced values of heat transfer coefficients in the design to obtain reasonably accurate results.

Other factors which appear to affect the boiling heat transfer performance are the cylinder diameter and boiling pressure. The average heat transfer coefficient appears to increase with increasing cylinder diameter. However, more data is needed for verification. The average heat transfer coefficient does increase as the boiling pressure is increased over the pressure range studied.

Since nucleate boiling heat transfer coefficients are a strong function of surface condition, the results presented herein for commercial copper tubing should provide more representative values of the heat transfer coefficients than have previously existed for refrigeration system designers.

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VITA

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APPENDIX A

COMPUTER PROGRAM FOR DATA REDUCTION

The following computer program was used to reduce the experimental data to temperature differences, heat fluxes, and heat transfer coefficients.

Initially, identification data is read in and printed out. This data gives the values of the parameters: refrigerant, tube size, oil concentration, and saturation pressure. Also, the date of the data run is included.

The program then reads in the six thermocouple readings (temperature difference in milivolts), averages them, adds them to the saturation temperature of the boiling refrigerant, and corrects them with the zero reading of the voltmeter. Then the resulting tube wall temperatures in milivolts are converted to degrees F (using an approximating polynomial) and subtracted from the saturation temperature of the refrigerant. This average temperature difference between saturation temperature and wall temperature is then used with the heat flux to calculate the heat transfer coefficient.

The values of temperature difference, heat flux, and heat transfer coefficient are then printed out along with their logarithms (base 10) used for log-log plots.

The necessary input data is arranged as follows:

In the first five columns of a single card is the number (NTMS) of data sets (not the same as the number of data points-included in each set) to be reduced.

The following data is repeated in order for the number of data sets.

The next two cards contain the nine values of DATA(I). Each value is a decimal number typed on a field of ten columns and is defined as follows:

a.	refrigerant number	DATA(1)
b.	amount of refrigerant, lbm	DATA(2)
c.	diameter of the tube, inches	DATA(3)
d.	length of the tube, inches	DATA(4)
e.	oil concentration, %	DATA(5)
f.	saturation pressure, lbf/sq in ABS	DATA(6)
g.	month in which the data was taken	DATA(7)
h.	day of the month on which the data was taken	DATA(8)
i.	last two digits of the year in which the data was taken	DATA(9).

Then the following five values are read in from a single card. The first two are integers each taking five columns, and the last three are decimal numbers taking ten columns.

a.	number of data points in the data set	IPT
b.	number of data points to and including maximum heat flux	IPTCUT
c.	inner radius of the tube, i.e., the distance from the tube center line to the thermocouples, inches	DR1

d. thermal conductivity of the tube, btu/hr-ft-F — DK

e. value read when the voltmeter measuring thermocouple emfs is shorted across its terminals, milivolts — DCOR.

Next, the experimental data for each of the data points is read from two cards. Each number is in decimal form occupying ten columns.

a.-f. six thermocouple readings, milivolts — DVM(1)-DVM(6)

g. measured boiling temperature, F — DVM(7)

h. measured boiling temperature, milivolts — DVM(8)

i. power supplied to the test cylinder, kilowatts — DVM(9)

j. measured boiling pressure, lbf/sq in ABS — DVM(10).

```
C
C
C     THIS PROGRAM COMPUTES TEMPERATURE DIFFERENCE (DTFF) VERSUS HEAT
C     FLUX DENSITY (DQA) VERSUS HEAT TRANSFER COEFFICIENT (DH) FROM
C     EXPERIMENTAL DATA FOR A REFRIGERANT BOILING FROM THE OUTSIDE OF A
C     CYLINDER.
C
C
      DIMENSION DVM(70,10),DATA(9)
      NR=1
      NW=3
C
C           'NTMS' IS THE NUMBER OF DATA SETS TO BE REDUCED.
C
      READ (NR,30) NTMS
   30 FORMAT(I5)
      DO 1400 NTIMES=1,NTMS
C
C           'DCONST' IS A CONVERSION FACTOR FROM KILOWATTS TO BTU/HR.
C
      DCONST=3413.
      DPI=3.14159
C
C
C           'DATA(1)' IS THE REFRIGERANT NUMBER.
C           'DATA(2)' IS THE AMOUNT OF REFRIGERANT (LBM).
C           'DATA(3)' IS THE OUTSIDE DIAMETER OF TUBE (INCHES).
C           'DATA(4)' IS THE LENGTH OF TUBE (INCHES).
C           'DATA(5)' IS THE OIL CONCENTRATION (%).
C           'DATA(6)' IS THE NOMINAL BOILING (SATURATION) PRESSURE
C           (LBF/SQ INCH   ABS).
C           'DATA(7)' IS THE MONTH (NUMBER) IN WHICH THE DATA WAS TAKEN.
C           'DATA(8)' IS THE DAY OF THE MONTH ON WHICH THE DATA WAS TAKEN
C           'DATA(9)' IS THE LAST TWO DIGITS OF THE YEAR IN WHICH THE
C           DATA WAS TAKEN.
C
```

```
C
C
C     READ DATA(I) IN AND WRITE OUT AS A CHECK.
C
   40 READ (NR,50) (DATA(I),I=1,9)
   50 FORMAT(8F10.6,/,F10.6)
      WRITE(NW,75) DATA(1),DATA(2),DATA(3),DATA(4),DATA(5),DATA(6),DATA(
     17),DATA(8),DATA(9)
   75 FORMAT(T2,'R-',F5.0,T10,'AMOUNT:',F7.3,' LB.',T30,'TUBE:',F7.3,
     1 ' IN. O.D. X ',F7.4,' LONG',T70,'% OIL:',F5.1,T85,'PRESSURE RNG '
     2,F5.2,T110,'DATE TAKEN:',F3.0,'-',F4.1,'-',F3.0,//)
C
C           'DAREA' IS THE SURFACE AREA OF THE CYLINDER (SQ FT).
C
      DAREA=DATA(3)*DATA(4)*DPI/144.
      WRITE(NW,76) DAREA
   76 FORMAT(F10.5)
C
C           'IPTS' IS THE NUMBER OF DATA POINTS IN THE CURRENT DATA SET.
C
C
C           'IPTCUT' IS THE NUMBER OF DATA POINTS TO AND INCLUDING
C           MAXIMUM HEAT FLUX.
C
C
C           'DR1' IS THE INNER RADIUS OF THE CYLINDER (I.E., LOCATION OF
C           THERMOCOUPLES FROM CENTER OF CYLINDER).  (INCHES)
C
C
C           'DK' IS THE THERMAL CONDUCTIVITY OF THE CYLINDER, BTU/HR-FT-F
C
C
C           'DCOR' IS THE ZEROED VOLTMETER CORRECTION FACTOR ADDED TO
C           THERMOCOUPLE MILLIVOLT READINGS (MILLIVOLTS)
C
      READ(NR,80) IPTS,IPTCUT,DR1,DK,DCOR
```

```
   80 FORMAT(2I5,3F10.5)
C
C           'DR2' IS THE OUTSIDE RADIUS OF THE CYLINDER (INCH).
C
C
C     CALCULATE OUTSIDE RADIUS USING OUTSIDE DIAMETER.
C
      DR2=DATA(3)/2.
C
C
C     WRITE OUT THE NUMBER OF DATA POINTS, DATA POINTS TO MAXIMUM HEAT
C     FLUX, INNER RADIUS, OUTER RADIUS, THERMAL CONDUCTIVITY AND THE
C     CORRECTION TERM.
C
      WRITE(NW,90) IPTS,IPTCUT,DR2,DR1,DK,DCOR
   90 FORMAT(T5,'NUMBER OF DATA POINTS:',I5,T40,'NO. OF DATA PTS TO MAX.
     1 Q/A',I5,T75,'OUTER RAD. OF TUBE:',F10.5,T105,'INNER RAD. OF TUBE:
     2',/,F10.5,T20,'THERM. COND. OF TUBE:',F5.1,T55,'CORRECTION FACTOR
     3 TO MV READING',F10.5,//)
C
C           'DIVLN' IS THE CONGLOMERATION OF CONSTANTS USED TO MULTIPLY
C           BY THE HEAT FLUX TO FIND TEMPERATURE DROP FROM THEMOCOUPLES
C           (INNER RADIUS) TO OUTSIDE OF TUBE (HR-SQ FT-F/BTU).
C
      DIVLN=ALOG(DR2/DR1)/(2.*DPI*DATA(4)*DK)*12.
      WRITE(NW,95) DIVLN
   95 FORMAT(/,F12.8,//)
      IPTS1=IPTS+1
C
C           'DVM(L,J)' IS THE ARRAY OF L DATA POINTS HAVING J COMPONENTS
C           FOR EACH DATA POINT.
C
C     THE J COMPONENTS ARE:
C           'DVM(L,1)' THROUGH 'DVM(L,6)' ARE THE SIX THERMOCOUPLE
C           READINGS (MILLIVOLTS).
```

```
C          'DVM(L,7)' IS THE MEASURED TEMPERATURE OF THE BOILING
C          REFRIGERANT (F).
C          'DVM(L,8)' IS THE MEASURED TEMPERATURE OF THE BOILING
C          REFRIGERANT (MILLIVOLTS).
C          'DVM(L,9)' IS THE MEASURED POWER SUPPLIED TO THE TEST
C          CYLINDER (KILOWATTS).
C          'DVM(L,10)' IS THE MEASURED SATURATION PRESSURE OF THE
C          BOILING REFRIGERANT (LBF/SQ IN  ABS).
C
      DO 125 L=1,IPTS
      READ(NR,100) (DVM(L,J),J=1,10)
  100 FORMAT(8F10.6,/,2F10.6)
  125 WRITE(NW,100) (DVM(L,J),J=1,10)
      DO 130 L=IPTS1,70
      DO 130 J=1,9
  130 DVM(L,J)=0.
      WRITE(NW,140)
C
C     WRITE OUT COLUMN HEADINGS.
C
  140 FORMAT(T6,'AVG. TEMP. DIF. ',      T35,'HEAT TRANS.',      T60,
     1 'HEAT TRANS. COEF.',//)
C
C     IN THIS LOOP CALCULATE THE AVERAGE TEMPERATURE DIFFERENCE, THE
C     HEAT FLUX, AND THE HEAT TRANSFER COEFFICIENT FOR EACH DATA POINT
C     IN THE CURRENT SET.
C
      DO 350 L=1,IPTS
      DSUM=0.
C
C     AVERAGE THE SIX THERMOCOUPLE MEASUREMENTS.
C
      DO 150 I=1,6
  150 DSUM=DSUM+DVM(L,I)
      DAVG=DSUM/6.
```

```
C
C         'DTMV' IS THE AVERAGE TEMPERATURE OF THE OUTSIDE OF THE TUBE
C         (MILLIVOLTS).  IT IS THE SUM OF THE MEASURED BOILING
C         TEMPERATURE DVM(L,8), THE AVERAGE OF THE SIX THERMOCOUPLE
C         READINGS DAVG, AND THE ZEROED METER CORRECTION READING DCOR.
C
      DTMV=DVM(L,8)+DAVG+DCOR
C
C     THE NEXT SECTION CONVERTS THE TUBE WALL TEMPERATURE IN MILLIVOLTS
C     (DTMV) TO TEMPERATURE IN DEGREES F (DTC).
C
C     IF THE TEMPERATURE IN MILLIVOLTS (DTMV) IS GREATER THAN 1.8 GO TO
C     THE NEXT EQUATION BECAUSE THIS CONVERSION EQUATION IS GOOD ONLY
C     FOR DTMV IN THE RANGE OF -.67 MV TO 2.711 MV (0 F TO 150 F).
C
      IF(DTMV.GT.1.8) GO TO 175
      DTF=31.96749+46.75351*DTMV-1.29622*DTMV**2+.02593*DTMV**3
      GO TO 225
C
C     IF THE TEMPERATURE IN MILLIVOLTS (DTMV) IS GREATER THAN 3.8 GO TO
C     THE NEXT EQUATION BECAUSE THIS CONVERSION EQUATION IS GOOD ONLY
C     FOR DTMV IN THE RANGE OF 1.751 MV TO 3.941 MV  (110 F TO 199 F).
C
  175 IF(DTMV.GT.3.8) GO TO 200
      DTF=39.873672485+32.1386260986*DTMV+9.20127010345*DTMV**2-3.668966
     1293*DTMV**3+.6457173228*DTMV**4-.0437941+0219*DTMV**5
      GO TO 225
C
C     THE FOLLOWING EQUATION IS GOOD ONLY FOR TEMPERATURES (DTMV) IN
C     THE RANGE OF 3.712 MV TO 6.094 MV (190 F TO 280 F).
C
  200 DTF=-1.82405090+80.8410797119*DTMV-15.0294952392*DTMV**2+
     1  2.78997898*DTMV**3-.2697338462*DTMV**4+.01045963541*DTMV**5
C
C     COMPUTE THE HEAT FLUX DQA AT THE POWER OF DVM(L,9).
C
```

```
  225 DQA=DVM(L,9)*DCONST/DAREA
C
C     COMPUTE THE TEMPERATURE DROP (DTC) ACROSS THE TUBE WALL USING THE
C     POWER DVM(L,9).
C
      DTC=DVM(L,9)*DCONST*DIVLN
C
C     CALCULATE THE TEMPERATURE DIFFERENCE (DTFF IN F) BY SUBTRACTING
C     THE LIQUID REFRIGERANT TEMPERATURE DVM(L,7) (F), AND THE
C     TEMPERATURE DROP (DTC) ACROSS THE TUBE WALL FROM THE TUBE WALL
C     TEMPERATURE (DTF).
C
      DTFF=DTF-DTC-DVM(L,7)
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
      DH=DQA/DTFF
      DTFF10=ALOG10(DTFF)
      DQA10=ALOG10(DQA)
      DH10= ALOG10(DH)
C
C     WRITE OUT THE TEMPERATURE DIFFERENCE (DTFF), THE HEAT FLUX (DQA),
C     THE HEAT TRANSFER COEFFICIENT (DH), AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,250) DTFF,DQA,DH,DTFF10,DQA10,DH10
  250 FORMAT(T8,F10.5,T36,F12.5,T63,F10.5,T80,F10.5,T95,F10.5,T110,F10.5
     1,/)
  350 CONTINUE
 1400 CONTINUE
      STOP
      END
```

APPENDIX B

SEVERAL PREDICTIVE CORRELATIONS FOR BOILING HEAT TRANSFER

Nomenclature

A = area, sq ft

B = proportionality constant in Mikic-Rohsenow equation, dimensionless

B_L = proportionality constant in Levy equation, dimensionless

c_L = heat capacity of liquid, btu/lbm F

C_{sf} = proportionality constant in Rohsenow equation, dimensionless

d = characteristic dimension of heated surface, ft

g = earth's gravitational acceleration, ft/sq hr

g_o = conversion factor (gravitational constant), 4.17×10^8 lbm-ft/sq hr-lbf

G = mass velocity of liquid, lbm/hr-sq ft, defined with Gilmour equation

h = heat transfer coefficient, btu/hr-sq ft-F

h_{fg} = latent heat of vaporization, btu/lbm

k_L = thermal conductivity of liquid, btu/hr-ft-F

m = numerical exponent in Mikic-Rohsenow equation, dimensionless

P = absolute vapor pressure, lbf/sq ft

ΔP = difference in saturated vapor pressure of liquid corresponding to a change in liquid temperature of $\Delta T(F)$, lbf/sq ft

Q = heat transfer rate, btu/hr

T_L = saturation temperature of boiling liquid, R

ΔT = temperature difference between test surface and bulk liquid, R

v = vapor rate, lbm/hr

Greek letters

α_L = thermal diffusivity of saturated liquid, sq ft/hr

μ_L = viscosity of saturated liquid, lbm/ft-hr

ϕ = function defined in Mikic-Rohsenow equation

Φ = proportionality constant of Gilmour equation, dimensionless

π = constant, 3.14159 . . .

ρ_L = saturated liquid density, lbm/cu ft

ρ_v = saturated vapor density, lbm/cu ft

σ = surface tension, lbf/ft

CORRELATIONS

Pool nucleate boiling:

1. Rohsenow[2]

$$\frac{c_L \Delta T}{h_{fg}} = C_{sf}\left[\frac{Q/A}{\mu_L\ h_{fg}} \left(\frac{g_o\ \sigma}{g(\rho_L-\rho_v)}\right)^{.50}\right]^{.33} \left(\frac{c_L\ \mu_L}{k_L}\right)^{1.7} .$$

2. Gilmour[14]

$$\left(\frac{h}{c_L\ G}\right) \left(\frac{c_L\ \mu_L}{k_L}\right)^{.6} \left(\frac{\rho_L\ \sigma}{P^2}\right)^{.425} = \frac{\Phi}{(D\ G/\mu)^{.3}}$$

where $G = \frac{V}{A}\ \frac{\rho_L}{\rho_v}$

and $v = Q/h_{fg}$.

3. Kutateladze[16]

$$\frac{h}{k_L} \left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5} = 7.0\text{x}10^{-4} \left[\frac{Q/A}{\alpha_L \rho_v h_{fg}} \left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5}\right]^{.7} .$$

$$\left[\frac{P}{\sigma} \left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5}\right]^{.7} \left[\frac{c_L\ \mu_L}{k_L}\right]^{-.35} .$$

4. Borishanskiy-Minchenko[15]

$$\frac{h}{k_L}\left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5} = 8.7\text{x}10^{-4}\left[\frac{Q/A}{\alpha_L\rho_v h_{fg}}\left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5}\right]^{.7}\cdot$$

$$\left[\frac{P}{\sigma}\left(\frac{\sigma}{g(\rho_L-\rho_v)}\right)^{.5}\right]^{.7}.$$

5. McNeilly[13]

$$\frac{h\ D}{k_L} = .225\left(\frac{D\ Q/A}{h_{fg}\mu_L}\right)^{.69}\left(\frac{P\ D}{\sigma}\right)^{.31}\left(\frac{\rho_L}{\rho_v}-1.0\right)^{.33}\cdot$$

$$\left(\frac{c_L\ \mu_L}{k_L}\right)^{.69}.$$

6. Forster-Zuber[12]

$$\frac{Q/A}{\Delta T\ k_L}\left[\frac{\Delta T\ c_L\ \rho_L\ \sqrt{\pi\ \alpha_L}}{h_{fg}\ \rho_v}\left(\frac{2\sigma}{\Delta P}\right)^{.5}\left(\frac{\rho_L}{\Delta P}\right)^{.25}\right] =$$

$$.0015\left[\frac{\rho_L}{\mu_L}\left(\frac{\Delta T\ c_L\ \rho_L\ \sqrt{\pi\ \alpha_L}}{h_{fg}\ \rho_v}\right)^{2.}\right]^{.62}\cdot$$

$$\left[\frac{\mu_L\ c_L}{k_L}\right]^{.33}.$$

7. Mikic-Rohsenow[17]

$$\frac{Q/A}{\mu_L\ h_{fg}}\left(\frac{\sigma\ g_o}{g(\rho_L-\rho_v)}\right)^{.5} = B(\phi\Delta T)^{m+1}$$

where

$$\phi^{m+1} = \frac{k_L^{.5}\ \rho_L^{2.125}\ c_L^{2.375}\ h_{fg}^{(m-2.875)}\ \rho_v^{(m-1.875)}}{\mu_L(\rho_L-\rho_v)^{1.125}\ \sigma^{(m-1.375)}\ T^{(m-1.875)}}$$

8. Levy[18]

$$Q/A = \frac{1}{B_L}\ \frac{k_L\ c_L\ \rho_L^2}{\sigma\ T(\rho_L-\rho_v)}\ (\Delta T)^3\ .$$

9. Forster-Grief[19]

$$Q/A = 4.3\text{x}10^{-5}\ \frac{\alpha_L\ c_L\ \rho_L(T-459.67)}{\sigma^{.5}\ (h_{fg}\ \rho_v)^{1.5}}\ [c_L\,(T-459.67)\alpha^{.5}]^{.25}\cdot$$

$$\left(\frac{\rho_L}{\mu_L}\right)^{.625}\ \left(\frac{\mu_L\ c_L}{k_L}\right)^{.33}\ (\Delta P)^2.$$

APPENDIX C

COMPUTER PROGRAM CALCULATING RESULTS FROM PREDICTIVE CORRELATIONS

The following computer program was used to compute temperature difference, heat transfer coefficient, and heat flux as predicted by several correlations in the literature. These correlations being those of:

a. Rohsenow[2]

b. Gilmour[14]

c. Kutateladze[16]

d. Borishanskiy and Minchenko[15]

e. McNeilly[13]

f. Forster and Zuber[12]

g. Mikic and Rohsenow[17]

h. Levy[18]

i. Forster and Grief[19].

Note: The last three correlations (g, h, i) are not yet debugged and, therefore, do not give reliable results.

Initially, refrigerant and cylinder properties are read in and then the computations for each correlation are made and printed out. The above order is the order in which the correlations are used and is the order of output.

The necessary input data to the program is as follows:

The number of data sets (NTIMES) is typed by itself in the first five columns of a single card. Each of these data sets is complete for calculations in and of itself with no extra data necessary between data sets.

The following data is then repeated for each data set.

For identification of the data set, the following values are read in on a single card. Each value is a decimal number in a field of ten columns.

a.	refrigerant number	REF
b.	nominal saturation pressure, atmospheres	ATM
c.	test cylinder outside diameter, inches	DIAM.

The next necessary data is read from two cards. These eleven pieces of data are decimal numbers in a field of ten columns.

a.	test cylinder outside diameter, inches	DOD
b.	test cylinder length, inches	DLEN
c.	surface tension of saturated refrigerant, lbf/ft	DSIGT
d.	density of saturated liquid, lbm/cu ft	DRHOL
e.	density of saturated vapor, lbm/cu ft	DRHOV
f.	latent heat of evaporation at saturation temperature, btu/lbm	DHFG
g.	viscosity of saturated liquid, lbm/ft-hr	DMU
h.	thermal conductivity of saturated liquid, btu/hr-ft-F	DK
i.	saturation pressure, lbf/sq in, absolute	DPSAT
j.	saturation temperature, F.	DTSAT.

Then in the first 20 columns of the next card is typed the Rohsenow constant C_{sf}, dimensionless CSF.

Next, for the Forster-Zuber correlation, it is necessary to use a series of saturation pressures. These are read in as DPRES(I), and there are 41 values, each in a ten column field. So, it is necessary to have nine cards to contain the data.

The first saturation pressure is the pressure at which the boiling actually took place. The next saturation pressures are those for a one degree increase in saturation temperature, a two degree increase, and so on until a 40 degree temperature increase is reached. The saturation pressures are in units of lbf/sq in, absolute pressure.

The next data to be read is for the Mikic-Rohsenow correlation. The necessary data are two decimal numbers, both read from a single card and each occupying a 20 column field.

a. variable exponent in the correlation, usually between 1 and 6 DMEXP

b. proportionality constant of the correlation DBCON.

The last piece of input data is the proportionality constant for the Levy correlation (DBL). It is read from the first ten columns of a single card.

```
C
C
C     THIS PROGRAM COMPUTES HEAT FLUX (DQA) VERSUS HEAT TRANSFER
C     COEFFICIENT (DH) VERSUS TEMPERATURE DIFFERENCE--HEATED SURFACE
C     MINUS SATURATION TEMPERATURE--(DTXL) FOR THE BOILING HEAT TRANSFER
C     CORRELATIONS OF ROHSENOW, GILMOUR, KUTATELADZE, BORISHANSKIY-
C     MINCHENKO, MCNEILLY, FORSTER-ZUBER, MIKIC-ROHSENOW, LEVY, AND
C     FORSTER-GRIEF; GIVEN THE PROPER DATA.
C
C
C     HOWEVER THE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS
C     BECAUSE OF SOME PROGRAMMING OR UNITS ERRORS NOT FOUND AS YET.
C
      DIMENSION DPRES(40)
      NR=1
      NW=3
C
C          'NTIMES' IS THE NUMBER OF DATA SETS.
C
      READ (NR,500) NTIMES
  500 FORMAT(I5)
      WRITE(NW,500) NTIMES
C
C          'DGO' IS THE GRAVITATIONAL CONSTANT IN UNITS OF LBM-FT/LBF-
C          SQ HR.
C
      DGO=4.17E08
C
C          'DCONST' IS A CONVERSION FACTOR FROM KILOWATTS TO BTU/HR.
C
      DCONST=3413.
      DPI=3.14159
      DO 5000 IJKL=1,NTIMES
```

```
C
C          'REF' IS THE REFRIGERANT; 'ATM' IS THE NOMINAL BOILING
C          PRESSURE (ATMOSPHERES); 'DIAM' IS THE TEST CYLINDER
C          DIAMETER (INCHES).
C
C     READ IN THE REFRIGERANT, PRESSURE AND DIAMETER, THEN WRITE THEM
C     OUT.
C
      READ(NR,700) REF,ATM,DIAM
  700 FORMAT(3F10.5)
      WRITE(NW,800)REF,ATM,DIAM
  800 FORMAT(T5,'R-',F5.1,T20,'PRESSURE RANGE',F3.1,T50,'TUBE DIAMETER',
     1 F10.5,/////)
C
C          'DOD' IS THE TEST CYLINDER OUTSIDE DIAMETER (INCHES);
C          'DLEN' IS THE CYLINDER LENGTH (INCHES);
C          'DSIGT' IS THE SURFACE TENSION (LBF/FT);
C          'DRHOL' IS THE DENSITY OF LIQUID REFRIGERANT AT SATURATION
C          PRESSURE (LBM/CU FT);
C          'DRHOV' IS THE DENSITY OF VAPORIZED REFRIGERANT AT SATURATION
C          PRESSURE (LBM/CU FT);
C          'DHFG' IS THE LATENT HEAT OF EVAPORATION AT SATURATION
C          PRESSURE (BTU/LBM);
C          'DMU' IS THE VISCOSITY OF SATURATED LIQUID REFRIGERANT
C          (LBM/FT-HR);
C          'DK' IS THE THERMAL CONDUCTIVITY OF SATURATED LIQUID
C          REFRIGERANT (BTU/HR-FT-F);
C          'DPSAT' IS THE SATURATION PRESSURE (LBF/SQ INCH   ABSOLUTE);
C          'DTSAT' IS THE SATURATION TEMPERATURE (F).
C
       READ(NR,1000)  DOD,DLEN,DSIGT,DRHOL,DRHOV,DHFG,DMU,DCP,DK,DPSAT,
     1DTSAT
 1000 FORMAT(8F10.6,/,3F10.6)
       WRITE(NW,1000)  DOD,DLEN,DSIGT,DRHOL,DRHOV,DHFG,DMU,DCP,DK,DPSAT,
     1DTSAT
```

```
C
C          'DCSF' IS THE CONSTANT OF THE ROHSENOW CORRELATION (UNITLESS)
C
      READ(NR,1005) DCSF
 1005 FORMAT(F20.10)
      WRITE(NW,1005)DCSF
C
C          'DB' IS THE BUBBLE DIAMETER OF THE ROHSENOW CORRELATION (FT).
C
      DB=(DSIGT/(DRHOL-DRHOV))**.5
C
C          'DPRL' IS THE PRANDTL NUMBER (UNITLESS).
C
      DPRL=DCP*DMU/DK
C
C          'DALPHA' IS THE THERMAL DIFFUSIVITY (SQ FT/HR).
C
      DALPHA=DK/(DCP*DRHOL)
C
C          'DAREA' IS THE SURFACE AREA OF TEST CYLINDER (SQ FT).
C
      DAREA=DPI*DOD*DLEN/144.
C
C          'DXQ' IS A CONVERSION FACTOR FROM POWER IN KILOWATTS TO HEAT
C          FLUX IN BTU/HR-SQ FT.
C
      DXQ=DCONST/DAREA
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DRHSN ) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE ROHSENOW CORRELATION.
C
      DRHSN=DHFG*DCSF*(DB/(DMU*DHFG))**.3333*DPRL**1.7/DCP
      WRITE(NW,1020) DB,DPRL,DAREA,DXQ,DRHSN
 1020 FORMAT(//,F20.10)
```

```
C
C      WRITE OUT COLUMN HEADINGS.
C
       WRITE(NW,1025)
 1025  FORMAT(//,T35,'ROHSENOW     CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
      1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',///)
C
C          'DPWR' IS THE POWER SUPPLIED TO TEST CYLINDER (KILOWATTS).
C
       DPWR=0.
       DCHNG=.005
       KINK=10
C
C      FIND THE HEAT FLUX, TEMPERATURE DIFFERENCE, AND HEAT TRANSFER
C      COEFFICIENT FOR SEVERAL POWER INCREMENTS.
C
       DO 1100 J=1,3
       DCHNG=DCHNG*J
       IF(DPWR.GT..145)  DCHNG=.05
       IF(DPWR.GT..145) KINK=22
       DO 1100 K=1,KINK
       DPWR=DPWR+DCHNG
C
C      CONVERT FROM KILOWATTS OF POWER TO HEAT FLUX (BTU/HR-SQ FT).
C
       DQA=DPWR*DXQ
C
C      COMPUTE THE TEMPERATURE DIFFERENCE DTX DUE TO THE HEAT FLUX DQA
C      FROM THE ROHSENOW CORRELATION.
C
       DTX=DRHSN*DQA**.33333
C
C      COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
       DH=DQA/DTX
       DQAL=ALOG10(DQA)
```

```
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 1050 FORMAT(T10,F18.5,T40,F18.5,T75,F12.3,T90,3F10.4)
 1100 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DGILM ) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE GILMOUR CORRELATION.
C
      DGLP=DRHOL/(DRHOV*DHFG)
      DGILM=.001*DCP*DGLP**.7*(DMU*12./DOD)**.3/(DPRL**.6*(DRHOL*DSIGT/
     1(DPSAT*144.)**2)**.425)
      WRITE(NW,1200) DGLP,DGILM
 1200 FORMAT(2F30.15)
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,1325)
 1325 FORMAT(//,T35,'GILMOUR      CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
C
C         'DPWR' IS THE POWER SUPPLIED TO TEST CYLINDER (KILOWATTS).
C
      DPWR=0.
      DCHNG=.005
      KINK=10
C
C     FIND THE HEAT FLUX, TEMPERATURE DIFFERENCE, AND HEAT TRANSFER
C     COEFFICIENT FOR SEVERAL POWER INCREMENTS.
C
      DO 1400 J=1,3
      DCHNG=DCHNG*J
```

```
      IF(DPWR.GT..145)  DCHNG=.05
      IF(DPWR.GT..145) KINK=22
      DO 1400 K=1,KINK
      DPWR=DPWR+DCHNG
C
C     CONVERT FROM KILOWATTS OF POWER TO HEAT FLUX (BTU/HR-SQ FT).
C
      DQA=DPWR*DXQ
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DUE TO HEAT FLUX USING THE
C     GILMOUR CORRELATION.
C
      DH=DGILM*DQA**.7
C
C     COMPUTE THE TEMPERATURE DIFFERENCE DTX.
C
      DTX=DQA/DH
      DQAL=ALOG10(DQA)
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 1400 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DKUTA ) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE KUTATELADZE CORRELATION.
C
      DKUTA=.0007*DK*DB**.4*(DPSAT*144./DSIGT)**.7/((DK/(DRHOL*DCP)*
     1 DRHOV*DHFG)**.7*DPRL**.35)
      WRITE(NW,1500) DKUTA
 1500 FORMAT(F30.15)
```

```
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,1625)
 1625 FORMAT(//,T35,'KUTATELADZE  CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
C
C          'DPWR' IS THE POWER SUPPLIED TO TEST CYLINDER (KILOWATTS).
C
      DPWR=0.
      DCHNG=.005
      KINK=10
C
C     FIND THE HEAT FLUX, TEMPERATURE DIFFERENCE, AND HEAT TRANSFER
C     COEFFICIENT FOR SEVERAL POWER INCREMENTS.
C
      DO 1700 J=1,3
      DCHNG=DCHNG*J
      IF(DPWR.GT..145)  DCHNG=.05
      IF(DPWR.GT..145) KINK=22
      DO 1700 K=1,KINK
      DPWR=DPWR+DCHNG
C
C     CONVERT FROM KILOWATTS OF POWER TO HEAT FLUX (BTU/HR-SQ FT).
C
      DQA=DPWR*DXQ
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DUE TO HEAT FLUX USING THE
C     KUTATELADZE CORRELATION.
C
      DH=DKUTA*DQA**.7
C
C     COMPUTE THE TEMPERATURE DIFFERENCE DTX.
C
      DTX=DQA/DH
      DQAL=ALOG10(DQA)
```

```
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 1700 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DBOMI ) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE BORISHANSKIY-MINCHENKO
C     CORRELATION.
C
      DBOMI=.00087*DK*(DPSAT*144./DSIGT)**.7*DB**.4/((DK/(DRHOL*DCP)*
     1 DRHOV*DHFG)**.7)
      WRITE(NW,1500) DBOMI
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,1925)
 1925 FORMAT(//,T35,'BORIS-MINCH  CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
C
C          'DPWR' IS THE POWER SUPPLIED TO TEST CYLINDER (KILOWATTS).
C
      DPWR=0.
      DCHNG=.005
      KINK=10
C
C     FIND THE HEAT FLUX, TEMPERATURE DIFFERENCE, AND HEAT TRANSFER
C     COEFFICIENT FOR SEVERAL POWER INCREMENTS.
C
      DO 2000 J=1,3
      DCHNG=DCHNG*J
      IF(DPWR.GT..145)  DCHNG=.05
      IF(DPWR.GT..145) KINK=22
```

```
      DO 2000 K=1,KINK
      DPWR=DPWR+DCHNG
C
C     CONVERT FROM KILOWATTS OF POWER TO HEAT FLUX (BTU/HR-SQ FT).
C
      DQA=DPWR*DXQ
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DUE TO HEAT FLUX USING THE
C     BORISHANSKIY-MINCHENKO CORRELATION.
C
      DH=DBOMI*DQA**.7
C
C     COMPUTE THE TEMPERATURE DIFFERENCE DTX.
C
      DTX=DQA/DH
      DQAL=ALOG10(DQA)
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 2000 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DMCNL ) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE MCNEILLY CORRELATION.
C
      DMCNL=.225*DK*(DPSAT*144./DSIGT)**.31*DPRL**.69*(DRHOL/DRHOV-1.)**
     1 .3333/(DHFG*DMU)**.69
      WRITE(NW,1500) DMCNL
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,2225)
```

```
 2225 FORMAT(//,T35,'MCNEILLY      CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
C
C           'DPWR' IS THE POWER SUPPLIED TO TEST CYLINDER (KILOWATTS).
C
      DPWR=0.
      DCHNG=.005
      KINK=10
C
C     FIND THE HEAT FLUX, TEMPERATURE DIFFERENCE, AND HEAT TRANSFER
C     COEFFICIENT FOR SEVERAL POWER INCREMENTS.
C
      DO 2300 J=1,3
      DCHNG=DCHNG*J
      IF(DPWR.GT..145)  DCHNG=.05
      IF(DPWR.GT..145) KINK=22
      DO 2300 K=1,KINK
      DPWR=DPWR+DCHNG
C
C     CONVERT FROM KILOWATTS OF POWER TO HEAT FLUX (BTU/HR-SQ FT).
C
      DQA=DPWR*DXQ
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DUE TO HEAT FLUX USING THE
C     MCNEILLY CORRELATION.
C
      DH=DMCNL*DQA**.7
C
C     COMPUTE THE TEMPERATURE DIFFERENCE DTX.
C
      DTX=DQA/DH
      DQAL=ALOG10(DQA)
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
```

```
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 2300 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DFORZU) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE FORSTER-ZUBER CORRELATION.
C
      DCON=DCP*DRHOL*(DPI*DALPHA)**.5/(DHFG*DRHOV)
      DFORZU=.0015*DK*(DRHOL/DMU*DCON**2)**.62*DPRL**.333/(DCON*(DSIGT/
     1 72.00)**.5*(DRHOL/(DGO*144.))**.25)
      WRITE(NW,2400) DALPHA,DCON,DFORZU
 2400 FORMAT(3F20.10)
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,2525)
 2525 FORMAT(//,T35,'FOR-ZUBER    CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
C
C     READ IN SEVERAL SATURATION PRESSURES DPRES(I) (LBF/SQ INCH  ABS)
C     OF THE REFRIGERANT BEGINNING WITH THE SATURATION PRESSURE AT THE
C     TEMPERATURE AT WHICH THE REFRIGERANT WAS ACTUALLY BOILING, AND
C     CONTINUING AT ONE DEGREE INTERVALS UNTIL THE NUMBER OF SATURATION
C     PRESSURES READ IN CORRESPONDS ROUGHLY TO THE MAXIMUM TEMPERATURE
C     DIFFERENCE EXPERIMENTALLY ENCOUNTERED.
C
      READ(NR,2550) (DPRES(I),I=1,41)
 2550 FORMAT(8F10.5)
C
C     NOW WRITE THESE PRESSURES OUT AS A CHECK.
C
      WRITE(NW,2550)(DPRES(I),I=1,41)
```

```
C
C     FOR EACH SATURATION PRESSURE EXCEPT THE FIRST, CALCULATE THE
C     DIFFERENCE BETWEEN IT AND THE FIRST SATURATION PRESSURE.  USE
C     THESE DIFFERENCES FOR CALCULATIONS IN THE FORSTER-ZUBER
C     CORRELATION.
C
      DPRESO=DPRES(1)
      DO 2560 I=1,41
 2560 DPRES(I)=DPRES(I+1)-DPRESO
      DTX=0.
C
C     CALCULATE HEAT FLUX AND HEAT TRANSFER COEFFICIENT FOR SEVERAL
C     (EQUAL TO NUMBER OF SATURATION PRESSURES READ MINUS 1) UNIT
C     (1 DEGREE) INCREMENTS IN TEMPERATURE DIFFERENCE USING THE FORSTER-
C     ZUBER CORRELATION.
C
      DO 2600 I=1,40
      DTX=DTX+1.0
C
C     CALCULATE HEAT FLUX USING TEMPERATURE DIFFERENCE, SATURATED
C     PRESSURE DIFFERENCE, AND THE FORSTER-ZUBER CORRELATION.
C
      DQA=DFORZU*DTX**1.24*DPRES(I)**.75
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
      DH=DQA/DTX
      DQAL=ALOG10(DQA)
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 2600 CONTINUE
```

```
C
C
C
C     THESE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS, BECAUSE
C     THESE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS, BECAUSE
C     THESE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS, BECAUSE
C     THESE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS, BECAUSE
C     THESE LAST THREE CORRELATIONS DO NOT GIVE VALID RESULTS, BECAUSE
C                    OF PROGRAMMING OR UNITS ERRORS.
C
C
C
C
C     READ IN THE VARIABLE EXPONENT (DMEXP) AND THE CONSTANT (DBCON)
C     FOR THE MIKIC-ROHSENOW CORRELATION.
C
      READ(NR,2850) DMEXP,DBCON
 2850 FORMAT(2F20.10)
C
C     NOW WRITE THEM OUT AS A CHECK.
C
      WRITE(NW,2850)DMEXP,DBCON
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DMIKRO) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE MIKIC-ROHSENOW CORRELATION.
C
      DPHIM=DK**.5*DRHOL**2.125*DCP**2.375*DHFG**(DMEXP-2.875)*DRHOV**
     1 (DMEXP-1.875)/(DMU*(DRHOL-DRHOV)**1.125*DSIGT**(DMEXP-1.375)*
     2(DTSAT+459.67)**(DMEXP-1.875))
      DMIKRO=DBCON*DMU*DHFG*DPHIM/DB
      WRITE(NW,1200) DPHIM,DMIKRO
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,2825)
```

```
 2825 FORMAT(//,T35,'MIK-ROH      CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
      DTX=0.
C
C     CALCULATE HEAT FLUX AND HEAT TRANSFER COEFFICIENT FOR SEVERAL
C     (GREATER THAN OR EQUAL TO THE MAXIMUM TEMPERATURE DIFFERENCE
C     EXPERIMENTALLY ENCOUNTERED) UNIT (1 DEGREE) INCREMENTS IN
C     TEMPERATURE DIFFERENCE USING THE MIKIC-ROHSENOW CORRELATION.
C
      DO 2900 I=1,40
      DTX=DTX+1.0
C     CALCULATE HEAT FLUX USING TEMPERATURE DIFFERENCE AND MIKIC-
C     ROHSENOW CORRELATION.
C
      DQA=DMIKRO*DTX**(DMEXP+1.)
C
C     COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
      DH=DQA/DTX
      DQAL=ALOG10(DQA)
      DHL=ALOG10(DH)
      DTXL=ALOG10(DTX)
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 2900 CONTINUE
C
C     READ IN THE CONSTANT (DBL) FOR THE LEVY CORRELATION AND WRITE
C     IT OUT AS A CHECK.
C
      READ(NR,2950) DBL
 2950 FORMAT(F10.5)
      WRITE(NW,2950)DBL
```

```
C
C      COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DLEVY ) EVALUATED
C      USING ALL THE CONSTANT TERMS IN THE LEVY CORRELATION.
C
       DLEVY=DK*DCP*DRHOL**2/(DSIGT*DHFG*(DTSAT+459.67)*(DRHOL-DRHOV)*
      1 DBL)
       WRITE(NW,1500) DLEVY
C
C      WRITE OUT COLUMN HEADINGS.
C
       WRITE(NW,2925)
 2925  FORMAT(//,T35,'LEVY          CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
      1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
       DTX=0.
C
C      CALCULATE HEAT FLUX AND HEAT TRANSFER COEFFICIENT FOR SEVERAL
C      (GREATER THAN OR EQUAL TO THE MAXIMUM TEMPERATURE DIFFERENCE
C      EXPERIMENTALLY ENCOUNTERED) UNIT (1 DEGREE) INCREMENTS IN
C      TEMPERATURE DIFFERENCE USING THE LEVY CORRELATION.
C
       DO 3000 I=1,40
       DTX=DTX+1.0
C
C      CALCULATE HEAT FLUX USING TEMPERATURE DIFFERENCE AND LEVY
C      CORRELATION.
C
       DQA=DLEVY*(DTX)**3
C
C      COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
       DH=DQA/DTX
       DQAL=ALOG10(DQA)
       DHL=ALOG10(DH)
       DTXL=ALOG10(DTX)
```

```
C
C     WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C     TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
      WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 3000 CONTINUE
C
C     COMPUTE (AND WRITE OUT) A SINGLE CONSTANT TERM (DFORGR) EVALUATED
C     USING ALL THE CONSTANT TERMS IN THE FORSTER-GRIEF CORRELATION.
C
      DFG1=DALPHA*DCP*DRHOL*DTSAT/(DSIGT**.5*(DHFG*DRHOV)**1.5)
      DFG2=(DCP*DTSAT*DALPHA**.5)**.25
      DFORGR=.0000430*DFG1*DFG2*(DRHOL/DMU)**.625*DPRL**.33333*144.**2
      WRITE(NW,2400) DFG1,DFG2,DFORGR
C
C     WRITE OUT COLUMN HEADINGS.
C
      WRITE(NW,3525)
 3525 FORMAT(//,T35,'FOR-GRIEF    CORRELATION',//,T10,'Q/A (BTU/HR-SQ.FT
     1)',T40,'H (BTU/HR-SQ.FT-F)',T75,'DELTA T (F)',//)
      DTX=0.
C
C     CALCULATE HEAT FLUX AND HEAT TRANSFER COEFFICIENT FOR SEVERAL
C     (EQUAL TO NUMBER OF SATURATION PRESSURES READ MINUS 1) UNIT
C     (1 DEGREE) INCREMENTS IN TEMPERATURE DIFFERENCE USING THE FORSTER-
C     GRIEF CORRELATION.
C
      DO 3600 I=1,40
      DTX=DTX+1.
C
C     CALCULATE HEAT FLUX USING THE FORSTER-GRIEF CORRELATION AND THE
C     SATURATION PRESSURE DIFFERENCES (DPRES(I)) FOUND BEFORE IN THE
C     FORSTER-ZUBER CORRELATION.
C
      DQA=DFORGR*DPRES(I)**2
```

```
C
C      COMPUTE THE HEAT TRANSFER COEFFICIENT DH.
C
       DH=DQA/DTX
       DQAL=ALOG10(DQA)
       DHL=ALOG10(DH)
       DTXL=ALOG10(DTX)
C
C      WRITE OUT THE HEAT FLUX DQA, HEAT TRANSFER COEFFICIENT DH,
C      TEMPERATURE DIFFERENCE DTX, AND THE LOG (BASE 10) OF EACH.
C
       WRITE(NW,1050)DQA,DH,DTX,DQAL,DHL,DTXL
 3600  CONTINUE
 5000  CONTINUE
       STOP
       END
```

APPENDIX D

UNCERTAINTY ANALYSIS

In any experimental work, uncertainties in measurements arise from the limitations of the equipment. The following estimates of instrumentation uncertainties are made to evaluate the accuracy of the results.

wattmeter:	±2.5 watts
pressure gauge:	±.25 lbf/sq in
thermometer (for saturation temperature):	±.5 F
ammeter:	±.05 amp
weighting scales (for oil):	±.002 lbm

digital multimeter and thermocouples:

range of ΔT_S (mv)	error in ΔT_S (mv)
.25 - .75	±.01
.75 - 1.4	±.03
1.4 - 2.2	±.04
2.2 - 2.9	±.05 .

These estimates are based on observations from all experimental runs and on the smallest division marked on the meter scales.

The multimeter and thermocouple errors are for local measurements (i.e., individual thermocouples) only and do not refer to temperature variation with location over the cylinder.

From observations made during the experimental work, the uncertainty in the pressure readings is assumed to cause an uncertainty in the measured temperature differences of 5% or less.

The uncertainty in surface area measurements is less than 3.5%.

Another source of uncertainty is the heat lost through the Teflon end caps. If they are treated as fins in a convective medium (no boiling was observed from them), it is found that the heat lost from them is less than 4% of the total heat supplied to the test cylinder.

On the basis of these sources of experimental uncertainties, a few data points have been analyzed for the probable uncertainty in the heat transfer coefficient h. (This analysis uses only data points occurring after boiling has begun.) Using the standard methods of uncertainty analysis, which find the total uncertainty by taking the square root of the sum of the squares of the individual uncertainties affecting the result, the uncertainty in h (defined by equation (1)) can be written as follows:

$$\Delta h = \left\{ \left[\frac{\partial h}{\partial Q} \Delta Q\right]^2 + \left[\frac{\partial h}{\partial A} \Delta A\right]^2 + \left[\frac{\partial h}{\partial (\Delta T_S)} \Delta(\Delta T_S)\right]^2 \right\}^{\frac{1}{2}}$$

or

$$\frac{\Delta h}{h} = \{[\frac{\Delta Q}{Q}]^2 + [\frac{\Delta A}{A}]^2 + [\frac{\Delta(\Delta T_S)}{\Delta T_S}]^2\}^{\frac{1}{2}}$$

where, in any consistent system of units:

Δh = uncertainty in the heat transfer coefficient

ΔQ = uncertainty in the heat transfer rate

ΔA = uncertainty in the surface area

$\Delta(\Delta T_S)$ = uncertainty in the temperature difference.

The uncertainty in h for the data points that were analyzed is then given in Table I. The uncertainty due to instrumentation is less than 11% for the data points analyzed and these should be representative of all the data taken.

TABLE I. Uncertainty in Heat Transfer Coefficient Due to Instrumentation

Refrigerant	Heater Size (in.o.d.)	Pressure (atm)	Oil Concentration (%)	Heat Flux (btu/hr-sq ft)	Uncertainty in h	
					Δh (btu/hr-sq ft-F)	$\Delta h/h$ (%)
11	1.125	1	10	2000	5.1	8.7
11	1.125	1	10	22100	25.2	7.9
11	1.125	2	10	5800	9.3	7.8
11	1.125	2	10	22100	24.7	7.7
11	1.125	2	0	2300	15.7	8.2
11	1.125	2	0	16500	56.7	8.5
11	0.625	1	1	5000	15.5	7.9
11	0.625	1	1	26600	45.5	8.0
113	1.125	1	0	1470	4.9	9.4
113	1.125	1	0	22100	59.5	8.5
113	1.125	2	10	4600	8.5	8.0
113	1.125	2	10	25700	22.1	7.5
113	0.625	1	10	2000	3.6	8.8
113	0.625	1	10	20000	15.8	7.6
113	0.625	2	3	1170	7.7	10.6
113	0.625	2	3	23300	54.8	8.4