Local Boiling Coefficients on a Horizontal Tube

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Local boiling heat transfer coefficients were experimentally determined for nucleate boiling around the outer circumference of horizontal copper tubing. The tubes used were of 16 B.W.G. hard-temper copper with outside diameters of 1 1/4 and 2 in.; the liquids boiled were methanol and *n*-hexane. The maximum peripheral variation occurred with the 1 1/4-in. tube in methanol where an over-all ΔT of 30.2°F. gave local outside coefficients varying between 249 and 548 B.t.u./(hr.)(sq. ft.)(°F.). The minimum variation was found to occur in the same system, in which an over-all ΔT of 72.3°F. gave coefficients varying between 856 and 910 B.t.u./(hr.)(sq. ft.)(F.°). The results, plotted in polar coordinates, showed a cardioid configuration for methanol with the maximum coefficients occurring at the bottom of the tube. The *n*-hexane results had the general shape of horizontal ellipses with maximum coefficients occurring at the sides of the tube.

A considerable amount of experimental work has been described in the technical literature in which nucleate, boiling heat transfer coefficients have been measured on the outer surfaces of horizontal tubes, cylinders, and wires. In almost all these studies the coefficients reported were considered to be average values representing the entire outer tube surface. As it was known that artificial agitation could cause significant increases in boiling coefficients, it appeared likely that the vapor rising from the lower portion of a horizontal tube should have an effect on the heat transfer characteristics of the upper portion of the same tube. The present investigation was made to determine some of the conditions under which this variation might be significant.

The only reference that could be found describing prior work of this nature is that of Katz et al. (*2*), who used a single apparatus for studying both boiling and condensing processes. Using a system in which steam condensed inside a horizontal tube and *n*-hexane boiled on the outside, they noted temperature variations around the periphery of the tube wall; however, these were attributed in part, to the collection of condensate along the bottom of the tube.

The effects of external agitation on a boiling system are described by McAdams (*4*) and Beecher (*1*), who report that at low temperature differences the heat flux was thereby increased. Robinson and Katz (*6*) observed that the average boiling coefficient measured on a horizontal tube was appreciably raised by the introduction of vapor beneath the tube. Myers and Katz (*5*) measured the average boiling coefficient for each tube of a vertical tier of four tubes and found that at all temperature differences studied the boiling coefficients obtained from the upper three tubes were significantly greater than for the bottom tube.

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EXPERIMENTAL

The basic experimental apparatus consisted of a single copper tube passing horizontally through a vessel containing the liquid to be boiled. Hot water passing through the tube served as the heat source. The careful placing of thermocouples permitted measurement of the wall temperature, the bulk temperature of the fluid inside the tube, and the bulk temperature of the boiling fluid, all at the same longitudinal position. Carrying out a prior experimental program to determine a correlation between the inside film coefficient and the flow variables made it possible to predict the heat transfer coefficient for the inside surface at the point where the temperature measurements were made.

The following equation represents a heat balance at this point:

$$dq = h_i\, dA_i(T_i - T_w) = h_o\, dA_o(T_w - T_o) \qquad (1)$$

This equation may be solved for the local boiling coefficient (h_o) to give

$$h_o = \frac{h_i D_i}{D_o} \frac{(T_i - T_w)}{(T_w - T_o)} \qquad (2)$$

Among the simplifying assumptions employed when the equations above were used was the use of a tube-wall temperature measured midway between the inner and outer tube surfaces to represent the temperature of each. Because Equation (2) involves a ratio of temperature differences, the errors thus produced in the numerator and denominator are not additive but subtractive. The maximum error in h_o resulting from this procedure was found to be 1.1% and the average 0.7%.

Further assumptions were that the inside film coefficient of heat transfer was independent of angular and longitudinal position and that the bulk temperature of the fluid inside the tube was equal to the temperature measured at the tube axis. A fourth assumption was that the contribution of peripheral heat conduction in the tube was negligible. In this latter case, if the existence of a vertical plane of symmetry is assumed, then it would follow that the temperature gradient would be zero at both the top and bottom of the tube. At all other points, however, the existence of

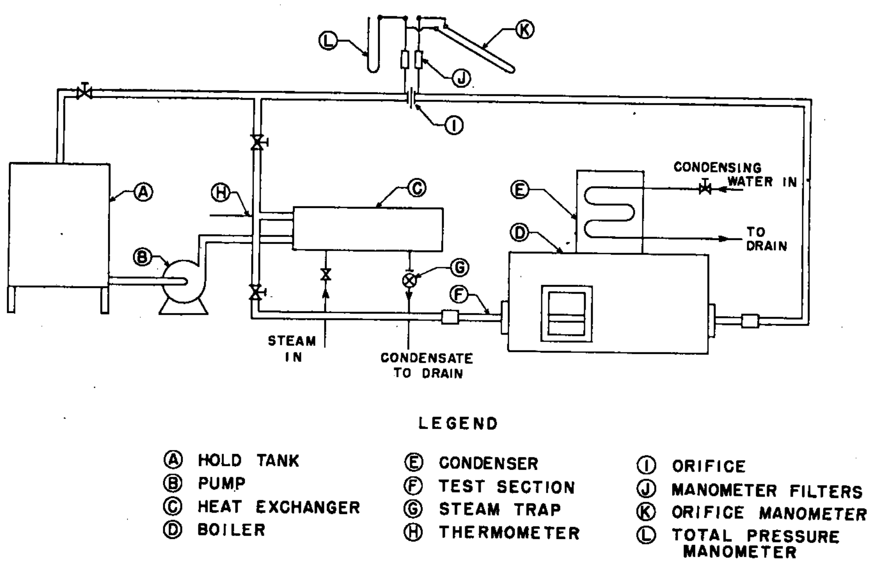

Fig. 1. Diagram of equipment.

unequal film coefficients would result in temperature gradients that would cause Equations (1) and (2) to be in error. To avoid this possibility, the section of the tube circumference being used for measurements was thermally insulated from the remainder of the circumference in a manner which will be described later.

APPARATUS

The apparatus, described above, is shown schematically in Figure 1. Distilled water was pumped from a 55-gal. drum into the tube side of a Ross (type BCF) heat exchanger, where it was heated by steam condensing on the shell side. The hot water leaving the exchanger flowed through the experimental boiler, then through an orifice, and back to the surge drum. Sufficient pressure was kept on the system so that vaporization was prevented at the orifice.

The boiler, shown in Figure 2, was an insulated rectangular tank 15 in. wide, 27 in. long, and 17 in. high, constructed from ⅛-in. sheet steel. It was fitted with a rectangular sight glass on each side and a multipass, finned-tube condenser. A drip pan placed beneath the condenser distributed the condensate to both sides of the boiler.

On each end of the boiler a packing gland was installed to permit easy rotation of the tube while a liquid seal was maintained. Copper tubes of two different diameters were employed as test sections. Circular cast-iron I-shaped flanges of identical outside diameter were constructed and soldered to each tube. The outer edges of these flanges rotated against the asbestos packing which provided the seal. The thermocouple leads were brought out through the flanges.

The test section of tubing was a 3-in. length shown in the detailed diagram, Figure 3. Two sizes were used, the dimensions of which are shown in Table 1.

TABLE 1—DIMENSIONS OF EXPERIMENTAL TUBES

	1¼-in. tube	2-in. tube
Wall thickness, in.	0.065	0.065
Inside diameter, in.	1.120	1.870
Outside area, sq. ft./ft.	0.327	0.523
Inside area, sq. ft./ft.	0.302	0.490
Inside cross-sectional area, sq. in	0.985	2.747
Total outside area for heat transfer, sq. ft.	0.681	1.091

The tube-wall thermocouple installation is illustrated in Figures 3 and 4. Two holes were drilled in opposite ends of the 3-in. test section and a single thermocouple wire, placed inside hypodermic tubing, was then soldered into each hole. The constantan thermocouple wire was connected to the upstream end of the test section and the copper wire to the downstream end. Because the small longitudinal gradient in the tube wall further diminished any minor error due to the voltage generated at the junction of the copper wire and tubing, the calibrated assembly thus gave the temperature of the wall at the point where the constantan wire made contact. After

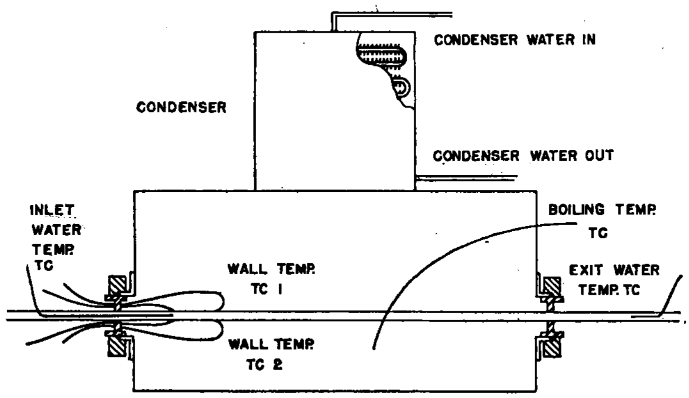

Fig. 2. Detail diagram of boiler.

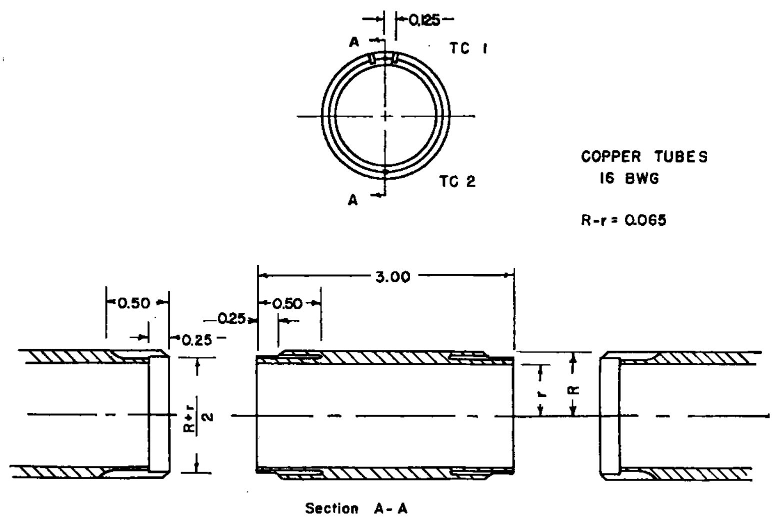

Fig. 3. Test-section details.

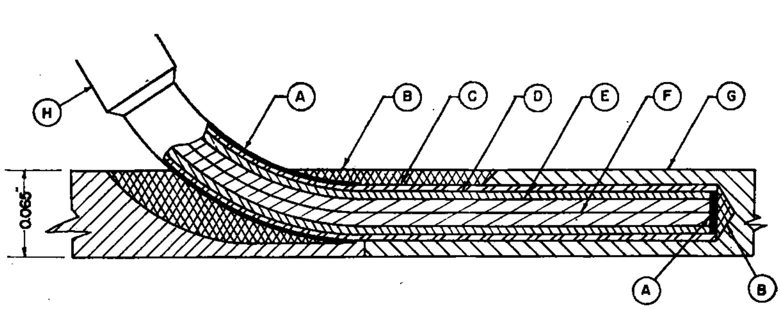

Fig. 4. Detail of thermocouple installation.

the thermocouple wires had been installed in the 3-in. test section, the assembly was soldered together and the surface of the copper tubing polished with size 0 emery paper. Temperature measurements indicated that the surface became stable after boiling for a period of 2 days.

To reduce the possibility of peripheral conduction the segment of the circumference containing the tube-wall thermocouple was thermally insulated by milling grooves on both sides of the thermocouple installation and filling those grooves with an insulating material. The grooves were 0.0312 in. wide and 0.005 in. deep. They were located a distance of 0.125 in. on either side of the thermocouple, thus insulating a 14-deg. segment of the 2-in. tube and a 23-deg segment of the 1¼-in. tube. The grooves were filled with sauereisen plastic porcelain 78, covered with a dilute solution of sodium silicate, and polished until smooth.

Copper-constantan thermocouples enclosed in stainless steel hypodermic tubing were used to measure water temperatures at the center of the copper tubing and the bulk temperature of the boiling fluid. The inlet water temperature was taken at a point 2 in. inside the boiling vessel at the same longitudinal position as the constantan wire of the tube-wall thermocouple. Thus the effective length of the tube was 25 in. The water and tube-wall thermocouples were calibrated by circulating hot water through the tube, which for this purpose was covered with heavy insulation.

Measurements were taken with two fluids used in the boiler: methanol of 99.85% purity, which was obtained from the Commercial Solvents Corporation, and *n*-hexane with a boiling range of 65° to 67°C., obtained from the United Fuel Gas Company.

DETERMINATION OF INSIDE FILM COEFFICIENTS

The film coefficients of heat transfer for the inside surface of the copper tubing were determined by the method of Wilson (7). The procedure involved keeping the boiling film resistance constant while varying the water film resistance. Since the boiling film coefficient of heat transfer has a unique value for each heat transfer rate in a given system, the criterion of constant heat flux was used as the method of obtaining constant-boiling film resistance.

As the purpose of the project was to determine the variation in local boiling coefficients, it may seem paradoxical that an "average" boiling coefficient should be assumed over the entire surface of the tube; however, as will be seen later the boiling coefficients were found to be nearly uniform at high boiling rates, and it was the information obtained from these runs which provided the key data for determining the inside film coefficients. The negligible effect of longitudinal variation is indicated by the fact that for nearly all runs the temperature decrease of the water passing through the tube was less than 5% of the over-all-temperature-difference driving force. Thus the concept of the average coefficient for

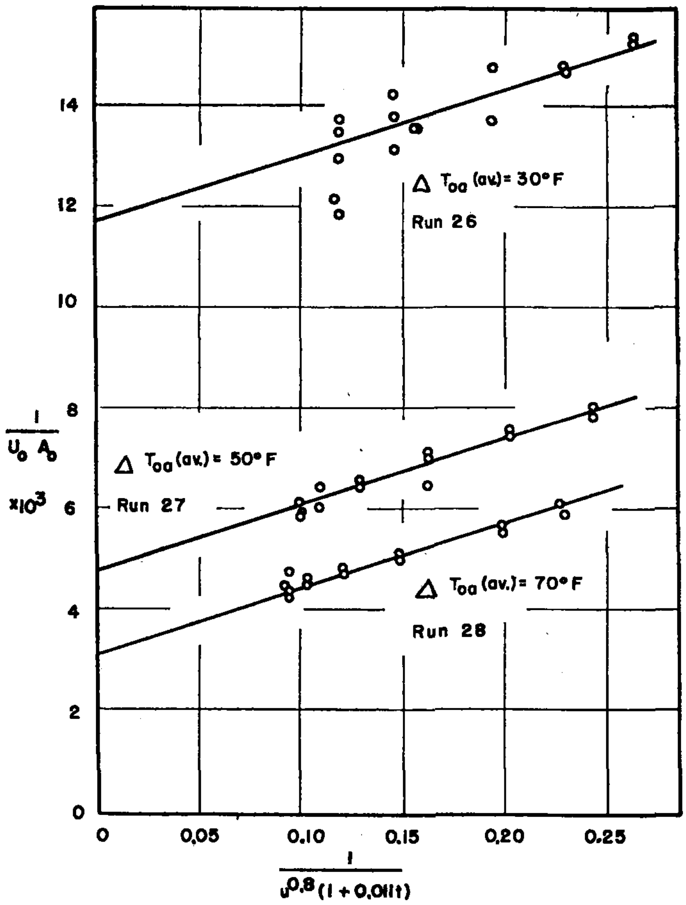

Fig. 5. Wilson plot for 1-1/4-in. tube.

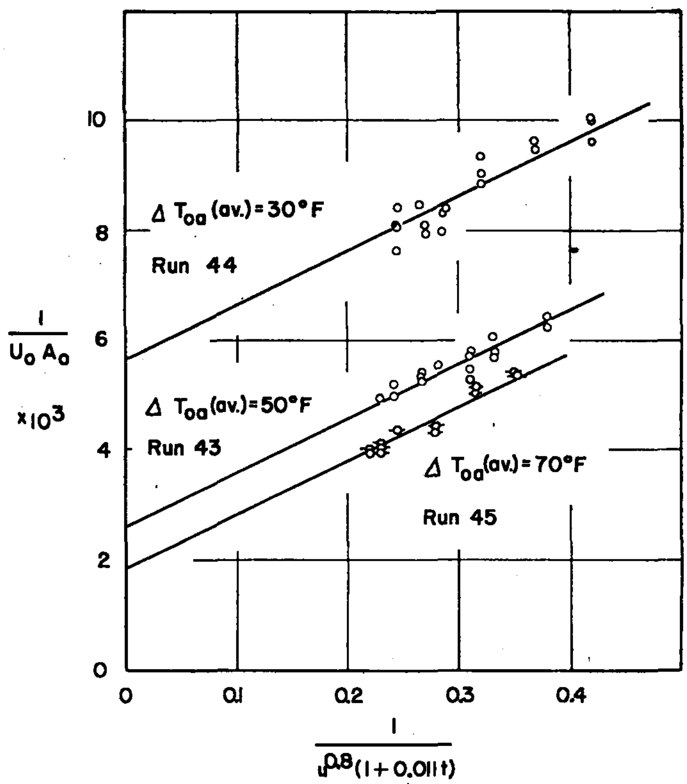

Fig. 6. Wilson plot for 2-in. tube.

purposes of the Wilson plot seems justified.

The Wilson plots, shown in Figures 5 and 6, represent the data taken solely for the determination of the inside coefficients of the two tubes. Methanol was used as the boiling fluid in both cases. Data are shown for three average over-all temperature differences, $\Delta T_{oa(avg)} = 30°$, 50°, and 70°F. The average heat loads corresponding to these temperature differences were 2,300, 7,900, and 13,900 B.t.u./hr. for the 1¼-in. tube and 3,600, 9,700 and 16,100 B.t.u./hr. for the 2-in. tube. Obviously, the over-all-temperature-difference driving force could not be held constant during a set of runs, in addition to the heat flux, and so the values given as $\Delta T_{oa(avg)}$ represent merely a convenient method of designation.

Both sets of data show considerable scattering at the lowest heat-flow rates, owing to the low temperature differences observed (0.41 to 1.28°F.). If a possible error of ± 0.1°F. is assumed for these readings and appropriate error values are taken for the other quantities entering the calculation, the possible errors in the ordinate values range between 11.1 and 29.8% for these two sets of data. The average possible errors are $\pm 20.3\%$ for run 26 (Figure 5) and $\pm 16.1\%$ for run 44 (Figure 6). However, the larger temperature differences observed for the other four runs result in much lower average possible errors of $\pm 7.7\%$ (run 27), $\pm 5.4\%$ (run 28), $\pm 6.9\%$ (run 43), and $\pm 4.7\%$ (run 45).

The best fitting straight lines on Figures 5 and 6 were established visually for the four runs at the high heat loads. Lines parallel to these were drawn for the two runs at low heat loads. This procedure is justified by the inclusion in the abscissa of the term representing the effect of water temperature on film coefficient.

The inside water film coefficients were obtained by the usual method of subtracting the ordinate intercepts from the ordinate values representing experimental runs. The values obtained for the 1¼-in. tube checked with the predicted coefficients by use of the Dittus-Boelter equation. Values obtained experimentally for the 2-in. tube were about 20% lower than the predicted values, possibly because of an entrance effect. The same external-flow system of 1-in. galvanized pipe was used for both tube sizes. The pipe-to-tubing connection was located 14 pipe diameters upstream from the boiler.

LOCAL BOILING COEFFICIENTS

Twelve sets of data were taken, from which local nucleate boiling coefficients were determined. The procedure involved all possible combinations of the two tubes, two boiling fluids, and three over-all temperature differences. The operating conditions are given in Table 2 and the

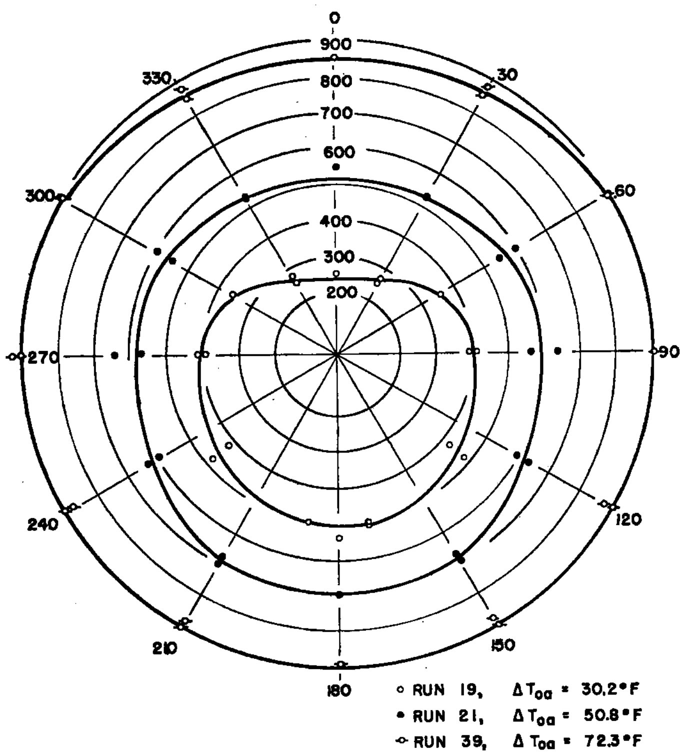

Fig. 7. Boiling coefficient profile—methanol; 1-1/4-in. copper tube.

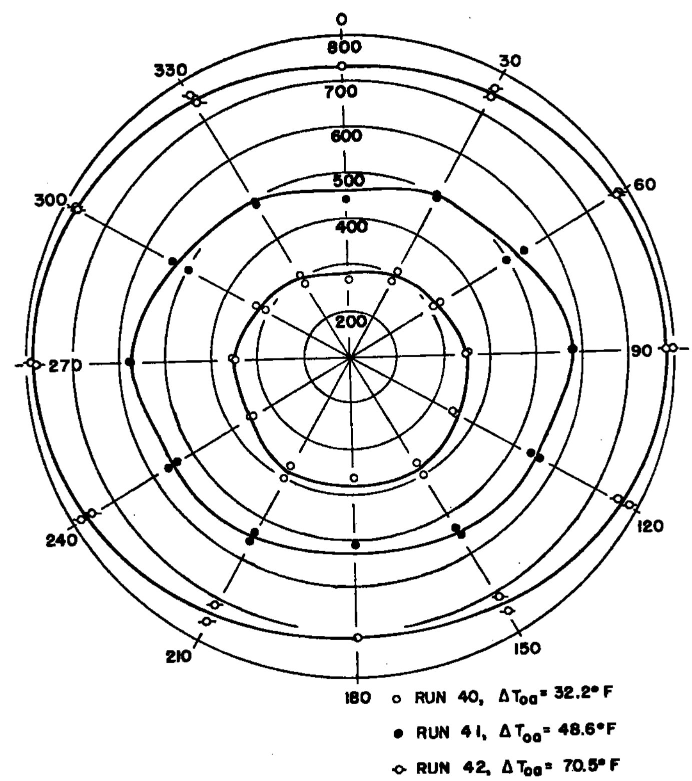

Fig. 8. Boiling coefficient profile—n-hexane; 1-1/4-in. copper tube.

experimental results in Figures 7 through 10.

The use of the Wilson plots described above was limited to obtaining inside film coefficients at only one water flow rate for each of the two tube sizes. This flow rate, chosen so as to be in the middle of the operating range—and at the same Reynolds number for each tube—was 58 lb./min. for the 1¼-in. tube and 97 lb./min. for the 2-in. tube.

The peripheral variation in heat transfer was determined by rotating the tube following each set of readings. The angular increment between reading positions was 30 deg., with the 0-deg. position taken as that in which the thermocouple was at the top of the tube. In each run two sets of readings were taken, with the tube being rotated 360 deg. in one direction and then 360 deg. in the opposite direction. The readings were averaged.

The experimental results are shown in Figures 7 through 10. Since there was little to distinguish one side of the tube from the other (the boiling vessel having a vertical plane of symmetry), the average coefficients referred to above were plotted on both sides of the graph and the best-fitting lines drawn through them. Each figure gives in polar coordinates the boiling coefficients plotted as a function of angular position at three values of the over-all-temperature-difference driving force.

EXPERIMENTAL RESULTS

The most obvious features of the results is the approach to radial symmetry as the temperature difference is increased. This probably indicates the dominant effect of local turbulence near the interface over the general agitation in the system when local heat transfer rates become high. Such a view would be substantiated by the results of Robinson (*6*), who found that the effect of artificial turbulence on heat transfer was negligible at high heat transfer rates.

The configurations measured by use of the two different fluids are quite unlike. For example, the ratio of maximum to minimum coefficients is greater for methanol than for *n*-hexane in all cases except one—that being the pair of runs (39 and 42) made on the 1.25-in. tube at the highest heat transfer rate. Even more obvious is the difference in the shape of the curves for the two fluids, the curves for methanol being of cardioid shape and the results from *n*-hexane of a generally elliptical shape, Table 2 contains a summary of the maximum and minimum coefficients measured for each run and the ratios of these extremes.

It had been anticipated that the maximum coefficients would occur at the 90- and 270-deg. positions as these would seem to be the points most affected by the vapor produced elsewhere on the tube. That this was the case for *n*-hexane

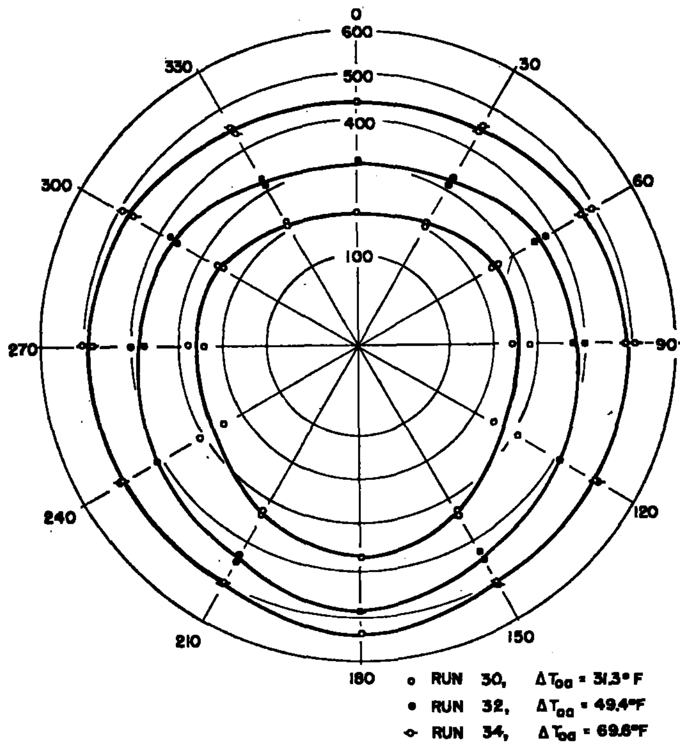

Fig. 9. Boiling coefficient profile—methanol; 2-in. copper tube.

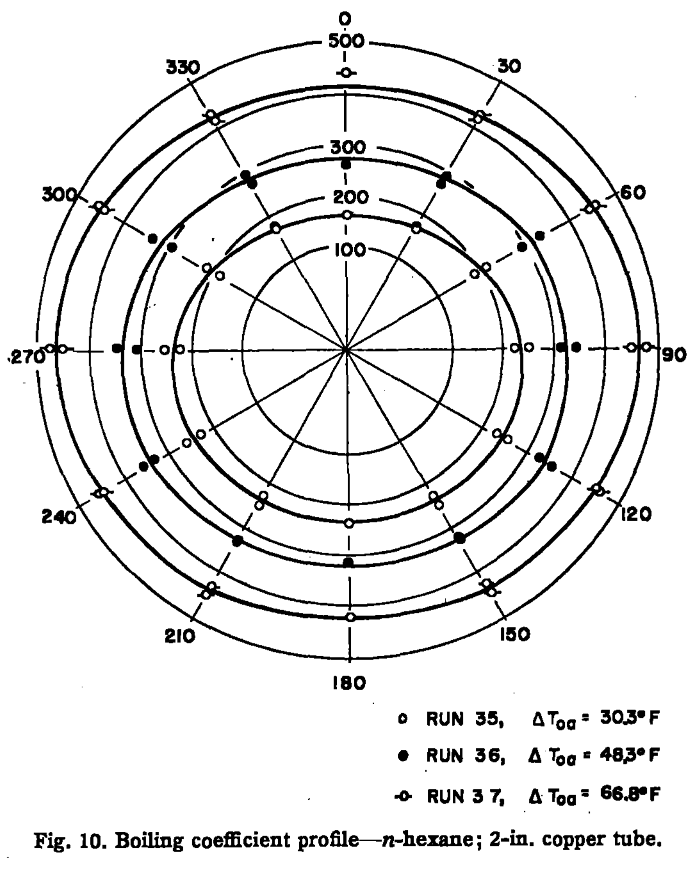

Fig. 10. Boiling coefficient profile—*n*-hexane; 2-in. copper tube.

and not for methanol may be due to the slightly lower coefficients found for *n*-hexane. This effect may also have been influenced by the fact that the volumetric latent heat for *n*-hexane of 26.9 B.t.u./cu. ft. was slightly lower than that for methanol, which was 33.0 B.t.u./cu. ft. Thus comparable heat transfer rates would produce a greater volume of *n*-hexane vapor than methanol vapor and accentuate the elliptical character of the results.

The boiling system was observed visually during the various runs and it was noticed that the bubbles formed on the bottom of the tube were somewhat larger than those formed on the sides. It might be expected from considerations of buoyancy that bubbles formed on the underside of the tube would be less easily dislodged than those on the remainder of the surface and would, as a result, grow larger. This would undoubtedly be another cause of lack of uniformity in the boiling coefficients around a tube. The experimental results at low heat loads all show a lack of horizontal symmetry, though why the coefficients are larger at the bottom than at the top is difficult to say.

No correlation expressing the variation in boiling coefficients around a tube is known to the authors, and because of the limited amount of such data available, no attempt was made to find one in this study. It is the authors' opinion that such a correlation, if found, might take the form of an expression for the boiling coefficient at high heat loads, modified by a coefficient expressing the angular variation at lower heat loads, where this variation becomes significant. The reference coefficient at high heat loads would be determined by the usual physical characteristics of the liquid-solid system, The second term would be an expression of the effects of the environment on local turbulence and would include such properties as would determine the volumetric rate of vapor evolution, velocity of bubble ascent, and the transport of momentum in the liquid-gas system.

CONCLUSIONS

1. Local nucleate boiling coefficients have been measured around horizontal tubes 1¼ and 2 in. in diameter and have been found to vary substantially with angular position on both tubes. The greatest variation on a tube in a single run was such that the maximum coefficient was 120% greater than the minimum.

2. The variation in local boiling coefficients decreases as the heat load is increased. At the maximum over-all temperature differences employed, the variation was about 10 to 20%.

3. The configurations obtained when the boiling coefficients were represented in polar coordinates were found to be quite dissimilar for the two liquids. The curves for methanol were of cardioid shape with maximum coefficients occurring at the bottom of the tube, and those for *n*-hexane resembled horizontal ellipses with maximum coefficients at the sides of the tube.

4. It appears that boiling coefficients reported in the literature as being measured on horizontal tubes are average coefficients, depending on the method of measurement. This may explain some of the discrepancies in the literature on boiling heat transfer.

TABLE 2

Run	Boiling fluid	Temp. of boiling fluid, °F.	Tube size, in.	ΔT_{0a}, °F.	Water rate, lb./min.	Inside coefficient, B.t.u./(hr.)(sq. ft.)(°F.)	Outside coefficient, B.t.u./(hr.)(sq. ft.)(°F.) Min.	Max.	Ratio of outside coefficients, $\frac{h_{max.}}{h_{min.}}$
19	Methanol	148.8	1.25	30.2	58.0	756	249	548	2.20
21	Methanol	148.0	1.25	50.8	58.0	788	515	708	1.37
39	Methanol	149.2	1.25	72.3	57.7	828	856	910	1.06
40	*n*-hexane	152.0	1.25	32.2	58.0	754	264	403	1.53
41	*n*-hexane	151.4	1.25	48.6	57.8	792	445	582	1.31
42	*n*-hexane	151.6	1.25	70.5	57.9	829	714	796	1.12
30	Methanol	149.3	2.00	31.3	97.1	331	205	368	1.79
32	Methanol	148.4	2.00	49.4	97.0	355	309	490	1.58
34	Methanol	148.4	2.00	69.6	96.5	394	438	542	1.24
35	*n*-hexane	152.0	2.00	30.3	97.0	333	164	258	1.57
36	*n*-hexane	151.6	2.00	48.3	97.0	357	260	356	1.37
37	*n*-hexane	151.6	2.00	66.8	96.8	394	414	478	1.15

A comparison of the results to determine the effect of tube diameter indicates that the ratio of maximum to minimum boiling coefficients is slightly greater for the 2-in. than for the 1¼-in. tube in five of the six possible cases when the over-all temperature difference is approximately the same. However, there is no good reason for using the equal temperature difference as a basis for comparison, and it would seem that no conclusions regarding tube diameter are relevant beyond the statement that there is a considerable variation in the boiling coefficient around both 1¼- and 2-in. tubes. Perhaps experimental data measured on smaller tubes would provide further useful information on this factor; however, the experimental difficulties would be considerably greater in such a program.

ACKNOWLEDGMENTS

The authors wish to acknowledge the generosity of the Wolverine Tube Company in supplying the tubing used in this investigation.

NOTATION

dq = heat transfer rate through differential segment of tube, B.t.u./hr.
dA_i = area of differential element inside the tube, sq. ft.
dA_o = area of differential element outside the tube, sq. ft.
D_i = inside diameter of tube, ft.
D_o = outside diameter of tube, ft.
h_i = local film coefficient of heat transfer on the surface dA_i, B.t.u./(hr.)(sq. ft.)(°F.)
h_o = local film coefficient of heat transfer on the surface dA_o, B.t.u./(hr.)(sq. ft.)(°F.)
t = average of tube inlet and outlet water temperatures, °F.
T_i = bulk temperature of the water inside the tube at point of measurement, °F.
T_w = wall temperature at point of measurement, °F.
T_o = bulk temperature of the boiling fluid at point of measurement, °F.
ΔT_{0a} = average value of over-all-temperature-difference driving force $(T_i - T_0)$, °F.
u = bulk water velocity in tube, ft./sec,
U_0 = average over-all heat transfer coefficient based on total outside tube area (A_0), B.t.u./(hr.)(sq. ft.)(°F.)

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Manuscript submitted March 14, 1957; revision received August 19, 1957; paper accepted September 20, 1957.