The Roles of Capillary Wicking and Surface Deposits in the Attainment of High Pool Boiling Burnout Heat Fluxes

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The problem of ascertaining a correct model for pool boiling burnout has occupied many researchers for a considerable time. Attempts to phrase the burnout problem analytically (*1, 2*) have not succeeded in explaining all of the experimentally observed phenomena (*3, 4*). This is particularly true in the case of surface effects; while most analytic studies are predicated upon models which would not admit surface effects, the effects have now been shown to be significant (*5*).

The use of capillary wicking material to supply coolant to a boiling surface has attracted attention recently (*6, 7, 8*). The attention is drawn largely by the fact that capillary feed systems may be able to perform as well or better in zero gravity environments than they do at ground level.

The experiments presented in this paper were initially designed solely to reveal the configuration of capillary wicking which permits the highest possible heat flux from a boiling surface at standard gravity. The authors felt that the results also shed light on the pool boiling burnout problem however, and for this reason pool boiling tests without wicking were conducted for comparison purposes. The test results and comparisons are presented in this paper.

APPARATUS

The test sections employed were made from type 304 stainless steel tubing, were 3 in. (plus or minus 0.25 in.) in active length, and had 0.016 in. wall thickness. Tests were run with both cylindrical and semicylindrical heaters (Figure 1). Silastic silicon rubber insulation or a combination of insulations (shown in Figure 1) was used to reduce energy losses from the semicylindrical heaters.

For tests in which temperatures were to be measured, a calibrated chromel alumel thermocouple was installed in semicylindrical heaters as shown in Figure 1. Temperature drop through the wall was calculated to permit the evaluation of surface temperatures. Correction was made for the effects of variable heater properties (*9*).

Tests were run with pure distilled water or with tap water, with liquid level held 1½ in. above the heater level (Figure 1). All metallic fittings in the tank (excepting the heat transfer surfaces) were Teflon plated to reduce contamination of the distilled water, and the Pyrex tank was thoroughly cleaned before each use (*9*). A stainless steel steam coil was used to degas the distilled water and to maintain saturation temperatures during tests with both distilled and tap water.

Direct current from a full wave rectifier was supplied to the heaters through a calibrated shunt. Voltage drop across the shunt was measured on a millivoltmeter which had been calibrated with a potentiometer. Voltage taps were installed on a great many of the heaters to provide a calibration of heater resistivity near the center of the heater where burnout most often occurred. The calibration permitted the calculation of heat flux fom the product of amperage squared times resistance. Frequent checks were made to certify that test sections employed had the proper resistivity. The probable uncertainty in burnout heat flux measurements for semicylinders is ±4.3%, including effects of heat transfer through the end fittings and insulation, geometry of the heaters, resistance uncertainty, meter calibration, and meter reading error. For cylindrical heaters the uncertainty is ±3%. The uncertainties in ΔT_{sat} (temperature at the surface of the metal minus saturation temperature of the fluid) are 3.9°F., largely due to the potential error caused by possible temperature gradients in the thermocouple cavity shown in Figure 1.

When capillary wicking* was used, it was arranged about the heater surface as shown in Figure 2C. Many other wicking configurations were tried (*9*), but none gave burnout heat fluxes as high as those obtained with the arrangement shown in Figure 2 which allows for relatively free escape of the vapor through a narrow channel, just above the uppermost point on the heater. Distilled water was used as the coolant in all tests in which wicking was employed.

As shown in Figure 2 the wicking was kept away from most of the heater surface so that the mechanisms of nucleation and vapor escape near the heater surface were not

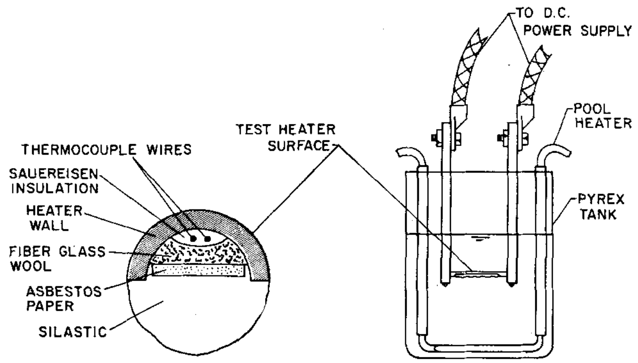

Fig. 1. Left: Section of semicylindrical heater. Right: Experimental apparatus.

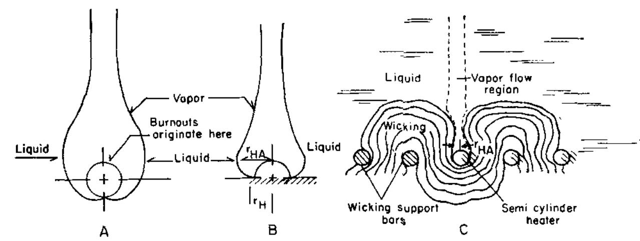

Fig. 2. Configurations of vapor around various types of ⅛-in. diameter heater surfaces. a. Full cylinders, b. Semicylinders, $r_{HA} \approx 0.19$ in., c. Semicylinder in wicking, $r_{HA} \approx 0.08$ in.

* The wicking used was packed to a density of 5.8 lb./cu. ft. and held in the configuration shown in Figure 2 by stainless steel rods running parallel to the heat transfer surface. (A complete description of the wicking material is given in reference 9, page 125). The wicking material was oriented such that the fibers tended to point toward the heater under test, since it was felt that this orientation would take maximum advantage of the wicking's ability to deliver liquid to the heater.

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directly influenced by the presence of the capillary material. Only the mechanism of liquid supply was affected, and this was primarily due to the fact that much of the liquid had to pass through the wicking fibers en route to the heater. The liquid carried foreign material (mostly silica) from the wicking and deposited it on the heater surface. This had a marked effect on the boiling process which will be described below.

When full cylinders were tested in wicking, the wicking and deposits served to insulate the bottom halves of the heaters. Energy had to be transferred to the top half, thence to the liquid (8, 9). Since the top half of the cylinders became only lightly speckled with deposits at the nucleation sites and was an effective heat transfer surface, it was desired to investigate boiling from the top half only. Hence semicylinders were employed in most of the tests in which wicking was used. The use of semicylinders also eliminated the complication of peripheral temperature variations.

PROCEDURE

Burnout Heat Flux

The tank was filled with the liquid to be used, which was heated to the saturation temperature and degased with the steam-heating coil. Saturation temperature was maintained with the steam coil, and the pool temperature was monitored by thermometer readings during the tests.

The heater to be tested was brought to a heat flux of 200,000 B.t.u./hr. sq. ft. This heat flux and all higher fluxes were held for time periods t given by the formula $t = 1.6 \times 10^6/q''$. The time to increase from one heat flux to another, t_i, was taken as $t_i = 4 \times 10^5/q''$. Heat flux was always increased by 10% of the previous flux.

Table 1. Burnout Heat Fluxes for Various Heater Sizes and Pool Conditions
(All burnout heat fluxes listed are in B.t.u./hr. sq. ft.)

Semicylindrical heaters

Heater diameter, in.		Distilled water	Tap water	In wicking, using distilled water
1/8 in.	Highest	449,000*	661,000*	1,230,000
	Lowest	407,000*	629,000*	980,000
	No. of tests	4*	5*	3
	Avg.	435,000*	641,000*	1,130,000
1/4 in.	Highest	425,000*	472,000*	None
	Lowest	379,000*	470,000*	
	No. of tests	2*	3*	
	Avg.	402,000*	471,000*	
1/2 in.	Highest	399,000	450,000	528,000
	Lowest	317,000	371,000	446,000
	No. of tests	3	3	3
	Avg.	372,000	407,000	528,000
Cylindrical heaters				
1/16 in.	Highest	476,000	567,000	890,000†
	Lowest	368,000	531,000	788,000†
	No. of tests	3	3	3
	Avg.	431,000*	522,000	844,000†
1/8 in.	Highest	433,000*		None
	Lowest	417,000*		
	No. of tests	3*		
	Avg.	425,000*	423,000	
1/4 in.	Highest	391,000	439,000	
	Lowest	340,000	322,000	
	No. of tests	3	3	
	Avg.	357,000	389,000	

* Data from reference 5. See Procedures section.
† Heat flux based on entire surface area. Actually lower half of heater is probably not effective as heat transfer surface.

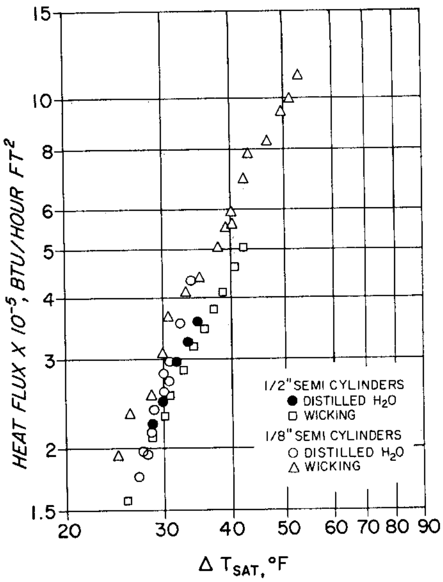

Fig. 3. Variation of ΔT_{sat}, metallic surface temperature minus saturation temperature of pool, for heaters in distilled water with and without wicking. Other sizes tested gave nearly identical results.

The timing schedule was adopted for reasons cited in reference 5 and also to insure that each heater tested would be subjected to exactly the same procedure. This also insured that the same amount of foreign material would be deposited on the heaters used with wicking or in tap water.* Control over the amount of deposit was required, since it has been shown that the deposit is influential in determining the burnout heat flux (5).

Increases in power level were continued until the burnout heat flux was reached; this was taken as the heat flux at which the heater melted. After burnout the heater was checked with micrometers near the burnout location to insure dimensional consistency. (Departures from dimensional consistency are included in the uncertainty estimates for heat flux, although they were invariably extremely small.)

The heat flux was based on the active surface area of the heater per unit length in the locality of the burnout. (Thus for a semicylinder of radius r_H ft. near the burnout, the surface area per unit length would be πr_H sq. ft. and for a full cylinder $2\pi r_H$ sq. ft. This procedure was used regardless of the nature of the surface deposit build up.)

Some data from reference 5 are used for comparison purposes in this paper. The data were taken with exactly the same time schedule as that outlined above, with minor exceptions.

Temperature Measurement

The thickness of the deposit on the heaters, and hence the thermal resistance of this layer, were functions of the time the heater was tested in tap water or in wicking. To obtain meaningful temperature readings with the wicking the heater was brought to a heat flux close to the anticipated burnout flux using the time schedule indicated above. Then the power levels were reduced and temperatures quickly recorded at a succession of heat fluxes. Only enough time was allowed for the heater to come to thermal

* The foreign material in the tap water consisted almost entirely of silica (9 parts/million), calcium (6 parts/million), and bicarbonate (20 parts/million) ions. Thus silicon or calcium carbonate was probably the major constituent of the surface deposits. The wicking contained a large amount of free silica, and this was carried to the heater by water passing through the wicking. It is felt that almost any contaminant which would improve surface wetability would be effective in increasing the burnout heat flux.

equilibrium during this phase of the testing.* Thus the curves of q'' vs. ΔT_{sat} (metallic heater surface temperature minus saturation temperature) are based on data taken with the heater rather uniformly coated with deposit (9).

Various checks were made to insure that the fluid used during distilled water tests was pure (9). These tests revealed no significant impurities in the distilled water.

RESULTS

Burnout Heat Fluxes

The burnout heat flux results for both cylindrical and semicylindrical heaters are listed in Table 1. It is evident that:

1. The heaters surrounded with wicking give the highest heat fluxes by far; those tested in tap water are next in order of heat flux, and those tested in distilled water gave by far the lowest values.
2. The heaters of the smallest size gave the highest burnout heat fluxes for any condition of operation.
3. Semicylindrical heaters invariably burned out at slightly higher heat fluxes compared with full cylindrical heaters of the same diameter.

Heat Transfer Behavior

Results of tests to determine the variation of q'' with ΔT_{sat} for heaters surrounded by wicking are shown in Figure 3. Note that the ΔT_{sat} is based upon the temperature at the surface of the metallic heater. In the case of the tests with wicking there was considerable foreign matter on the surfaces, primarily concentrated at the nucleation sites. Although no correction has been made for temperature drops through the foreign matter, the temperatures compare closely with those obtained with distilled water. This indicates that possibly the boiling heat transfer coefficients are not adversely affected by the wicking, and the process is not significantly different than normal pool boiling heat transfer.

Reference 5 indicates that the deposit of foreign matter is much heavier in tap water than is the case with the wicking in distilled water combination. However if correction is made for the temperature drop through the contaminant deposited by tap water, the heat transfer coefficients are close to those obtained with distilled water (5).

It is apparent that although deposits do not markedly change the heat transfer process which takes place at the surface of the deposits, their presence does appreciably and beneficially alter the burnout heat flux.

Nature of the Deposit Effect

Reference 5 indicates that the deposits increase the wetability of the surface, and that this probably accounts for the increased heat fluxes with tap water. It is proven conclusively in (5) that alterations of properties of the water itself are not the cause of the improvement.

The question arises, is the different type of deposit caused by the liquid being supplied through capillary passages responsible for the very high burnout heat fluxes obtained with wicking present? Deposits are at least partially responsible for the improvement, but there must be still another effect of the wicking.

Tests were run in which a 1/8-in. diameter semicylinder in wicking was brought to a heat flux of roughly 1,000,000 B.t.u./hr. sq. ft. In one case the power was turned off, and the heater was then removed from the wicking and tested in distilled water. It burned out at 650,000 B.t.u./hr. sq. ft. In a second test the heater was left in the wicking, and heat fluxes of 200,000, 300,000, 400,000, and 650,000 B.t.u./hr. sq. ft. were applied. After a period of about 1 min. at each of these fluxes the wicking was caused to drop away from the heater. The heater burned out at 650,000 B.t.u./hr. sq. ft. as soon as the wicking was dropped (10). This clearly indicates not only that the deposits from the wicking permit the attainment of heat fluxes well in excess of the normal pool boiling burnout q'', but that the wicking must be present to attain the highest fluxes reported here. Thus the presence of the wicking itself has an additional, desirable effect other than causing beneficial deposits on the heater.

Recently Dr. W. R. Gambill of Oak Ridge National Laboratory has conducted extensive and excellent tests to determine the inherent scatter of pool boiling burnout data. In the course of his tests he has found an effect of surface deposits on the burnout heat flux which largely substantiates the finding reported in this paper and in reference 5 (11).

THEORY

Burnout Mechanisms

It appears that the rate of liquid inflow from the sides is a factor controlling burnout heat flux in most of the tests reported here. This explains why the smaller heaters give the higher heat fluxes. Liquid has a shorter path to the central parts of the smaller diameter heaters, and the resulting higher liquid supply rate simply permits the attainment of higher heat fluxes. This also explains why the semicylinders had slightly higher heat fluxes compared with full cylinders. Vapor coming from the bottom of full cylinders partially screens liquid inflow from the sides and restricts the heat flux (see Figure 2). The effect appears to be small however.

Consider heaters of a single geometry, say the 1/8-in. diameter semicylinders. Since the potential rate of inflow from the sides would be the same, at least for heaters in distilled water and tap water, and since the gross behavior of the vapor would be the same for all 1/8-in. diameter semicylinders, one might expect burnout to come at the same heat flux for distilled and tap water. Indeed this would have to be the case if factors external to the heater were controlling the burnout. Since the burnout heat fluxes are actually quite different, one must look for a local mechanism on the heater itself. Some possible local mechanisms are discussed below.

1. In some cases it is possible that a local instability condition might arise in accordance with the theory of Chang (12). Thus a vapor bubble may reach a critical Weber number and be washed back onto the surface causing burnout. This condition could actually override liquid inflow from the sides as the mechanism causing burnout. It is shown in reference 5 that alterations of bubble size and population caused by surface deposits make treated heaters less susceptible to burnout by this mechanism than is the case for clean heaters.
2. Bubbles growing on the heater surface might serve to restrict liquid inflow to the most central parts of the heater. Surface deposits serve to reduce the base radius of the bubbles by improving surface wetability (Figure 4). By thus reducing the area of the heater occupied by bubbles the deposits may enable more liquid to reach the central parts of the heater.* At the same time smaller bubble bases mean that liquid has a shorter distance to travel to reach the center of the area vacated by a bubble.

* Even with wicking present the system came to thermal equilibrium within 30 sec. This would not have been the case if the heater were completely wrapped in insulation (8), but with the present configuration the thermal storage of the wicking is apparently not appreciably altered by changes in power level of the heater.

* The wetability increase does cause an increase in the bubble population also, and the increased population could tend to reduce liquid inflow (5). But in the present tests and at a heat flux of 293,000 B.t.u./hr. sq. ft. the population was roughly doubled (5) by the increased wetability, while the base area of an average bubble was less than 21% as large as that on a clean heater.

3. The surface deposits appear to improve the mechanics of bubble removal. If bubbles are not removed from the surface rapidly enough, the portion of the heater under the bubble may overheat. Bubble removal must also be rapid to prevent coalescence with subsequent bubbles and to permit liquid to rapidly occupy the portion of a heater formerly occupied by vapor.

A comparison of bubble geometries at a heat flux of 293,000 B.t.u./hr. sq. ft. was conducted with a high speed moving picture camera and the technique described in reference 5. It was noted that bubble residence times on the treated heaters were substantially less than those observed on clean heaters.

Some pertinent results are contained in Figure 4. The figure shows the largest bubble observed on a treated heater compared with the largest bubble on a clean heater (out of 127 bubbles). The base diameters for the bubbles shown in Figure 4 were computed by measuring numerous base diameters on various bubbles and obtaining a ratio of the base diameter to the largest bubble diameter for both treated and clean heaters (within about 40%). The ratio was used to estimate the base size for the bubbles shown in Figure 4: it was not possible to measure base radius directly for these particular bubbles because of the confused situation at their bases.

Bubbles with small radii are known to grow more rapidly than those with large radii, as experimental and analytic studies have revealed. The smaller bubbles on a treated heater thus reach their maximum size more rapidly than those on the clean heater. This fact was clearly observable in motion pictures. Moreover it is easier to accelerate the full-grown bubbles (or bubbles distorted into vapor columns) from the treated surface as the following analysis of forces indicates.

Forces Available for Vapor Removal

For the bubbles shown in Figure 4 the following forces were considered and evaluated at the instant of bubble departure.

Buoyancy: By integration, the upward force on a spherical bubble due to the pressure of the fluid surrounding the heater is

$$-2\pi\rho\frac{g}{g_c}r^3\left[\frac{\sin^2\theta}{2}+\frac{(\cos^3\theta-1)}{3}\right]$$

Regardless of the exact shape of the vapor leaving the heater the better wetability (and hence greater value of θ) for the treated heater gives the vapor greater buoyant forces. For the bubbles shown in Figure 4 the buoyant force 3.7×10^{-6} lb.$_f$ is almost equal for the two bubbles despite the greater radius of the bubble on the clean surface.

Surface Tension: If one regards the bubble departure as such as to sever the bubble from the heater as shown in Figure 4 (*C*), the surface tension force tending to hold

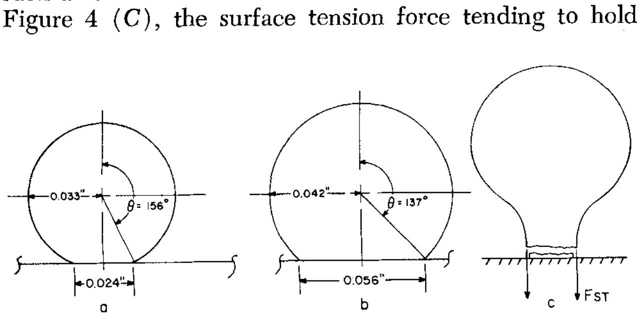

Fig. 4. a. Approximate geometry of largest bubble observed on treated surface at q'' = 293,000 B.t.u./hr. sq. ft. (at instant of departure). b. Approximate geometry of largest bubble observed on clean surface at q'' — 293,000 B.t.u./hr. sq. ft. C. Assumed method of bubble departure, used to calculate surface tension force F_{st}.

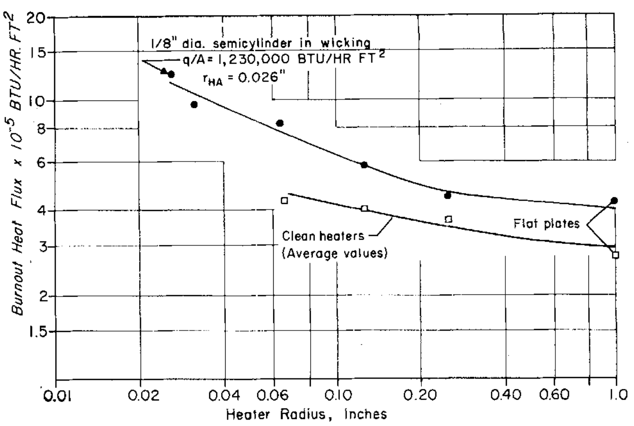

Fig. 5. Upper curve. Maximum values of burnout heat flux observed with treated heaters plotted vs. heater radius r_H. Apparent radius r_{HA} shown for ⅛-in. diameter semicylinder in wicking. Lower curve. Data for clean heaters, no surface treatment. Right-most points for both curves from (*10*).

the bubble on the heater is $2\pi r_b\,\sigma$. Because of poorer wetting the value of r_b is greater for the clean heater and the downward surface tension force is 5.9×10^{-5} lb.$_f$. This is over twice the force holding a bubble to the treated heater, 2.54×10^{-5} lb.$_f$

Pressure Forces on the Bubble: Pressure inside the bubble, acting on the base, tends to remove the bubble from the surface. Since the bubble bases were observed to be stationary when the bubble was ready to depart, neither expanding nor contracting, the liquid pressure at the base must balance the vapor pressure inside the bubble. (This is not true during the bubble growth time of course.) Calculating the liquid pressure at the bubble base [($\rho\, g/g_c\, 2r$) greater than the pressure acting at the top of the bubble] and multiplying by the base area one finds that the pressure force on the clean heater is seven times that on the treated heater. But the pressure force on the bubble shown in Figure 5 is only 7×10^{-6} lb.$_f$ for the clean heater.

Dynamic Forces: These forces are complex, comprising the inertia of bubble growth, the necessity of moving liquid from the vapor path and the probably overriding effect of liquid rushing in, tending to pinch the bubble off. It appears likely that dynamic forces assist in vapor removal, and because of their more rapid growth (preceding the instant of departure) the bubbles on the treated heater would receive stronger dynamic forces.

Even if one assumes equal dynamic forces, the summation of forces on the treated heater bubbles would exceed those on the clean heater (net force, F_D — 4.8×10^{-5} lb.$_f$ for clean heater, F_D — 2.0×10^{-5} lb.$_f$ for the treated heater). But the mass of the bubble on the treated heater (3.03×10^{-9} lb.$_m$ approximately for the bubble in Figure 4 is less than half that for the bubble on the clean heater (7×10^{-9} lb.$_m$ approximately). Thus if one considers the mass constant at the instant of departure, the acceleration of bubbles from the treated surface would be well over twice the acceleration of vapor from the clean surface.

The calculations are only indicative, because at high heat fluxes and steady state the bubbles are distorted into a partially columnar shape at the instant of departure. However it appears that because of more rapid growth and higher acceleration at departure vapor has a shorter residence time on a treated heater than on a clean one.

Thus a treated heater would be far less subject to burnout due to excessive vapor residence time as well as being less likely to fail owing to bubbles reaching a critical Weber number (5) or vapor blockage of liquid inflow. If this is so, the external limitations of liquid inflow from the sides might start to become the limiting factor on burnout heat flux. The rate of liquid inflow from the sides is strongly influenced by heater size, and one would expect heaters limited by inflow to show large effects of heater diameter. Such effects are clearly indicated in Table 1.

It thus seems reasonable that there is a limit to the extent burnout heat flux can be increased by surface deposits and that the limit is imposed by liquid inflow from the sides. This was substantiated by treating some heaters for very long times in tap water prior to burning them out in the same tap water. A relatively heavy coating was thus deposited on the heater, compared with that deposit in tests conducted in accordance with schedule reported above.

Figure 5 shows the highest values of burnout obtained plotted vs. heater radius.* The heaters were generally not treated according to the schedule reported above but were more heavily coated. For this reason the authors believe the burnout values shown approach the maxima obtainable by altering surface conditions. (In fact too much deposit might serve to restrict rather than augment overall energy transfer.)

If the results shown in Figure 5 are extrapolated, it appears that for semicircular heaters 0.6 in. or more in diameter surface deposits are probably incapable of raising the burnout heat flux above the normally reported values (about 400,000 B.t.u./hr. sq. ft.). This is consistent with Berenson's findings for a large flat plate (*13*).

The Effects of Wicking

Surrounding a heater with wicking causes the surface to become coated with a deposit of foreign material. The material apparently increases the liquid-solid interfacial forces substantially, resulting in better wetability. From tests described above it appears that burnout due to local conditions on the heater are prevented owing to the beneficial effects of the surface deposit. (In this connection it would have been instructive to run with clean heaters in wicking, but this was not possible. Water always brings deposits to the surface from the wicking passages.)

The data shown in Table 1 support the contention that liquid inflow from the sides is responsible for burnout in the tests with wicking. With wicking around the heater the vapor is forced to rise in a narrow channel (Figure 2), while the liquid flow rate is not appreciably hampered by the presence of the wicking. (Indeed the wicking may augment this liquid inflow slightly.)

The constraint of the vapor column results in the liquid having a substantially reduced length-of-path through the vapor screen to the heater. Motion picture studies, in which an end view of the ⅛-in. diameter semicylindrical heater wicking combination was photographed, revealed the configuration of vapor shown in Figure 2. The distance r_{HA} is the approximate half width of the vapor column measured from the topmost point on the heater. (It was at the topmost point on the heater that burnout almost invariably originated.) For ⅛-in. semicylinders in wicking r_{HA} was about 0.08 in.

Figure 5 shows maximum values of burnout heat flux plotted vs. heater radius r_H. The radius may be regarded as proportional to the distance liquid from the sides must travel to reach the centermost portions of the heater. However motion picture studies of the end view of a ⅛-in. diameter semicylindrical heater at various values of q'' show that the vapor blooms out as shown in Figure 2 and requires the liquid to travel about 0.19 in. to reach the central portions of the heater. (The value appears to be only a weak function of heat flux). It seems reasonable to say that because the liquid must travel only 0.08 in. with the wicking in place, the wicking serves to make the heater have an *apparent* radius (compared with ⅛-in. semicylinders without wicking) of 0.08/0.19 (0.0625) or about 0.026 in.

Figure 5 shows that a radius of 0.026 in. might be expected to correspond to a maximum heat flux of 1,200,000 B.t.u./hr. sq. ft. using the extrapolated curve from the semicylindrical heaters without wicking. The ⅛-diam. semicylinder heater with wicking actually burned out at a maximum of 1,230,000 B.t.u./hr. sq. ft.

Of course it may be argued that the close comparison is merely fortuitous, but considering all of the data it would appear that the limiting condition on heat flux in nucleate boiling for treated semicylindrical heaters is imposed by liquid inflow from the sides. It also appears that heaters supplied liquid by capillary wicking arranged as shown in Figure 2 are subject to the same mechanism of burnout.

* A slightly higher heat flux than that shown in Figure 5 was reported for ⅛-in. semicylinders in tap water (*5*). The higher value, 1,180,000 B.t.u./hr. sq. ft., was obtained by holding liquid only 1/32 in. above the topmost point of the heater. This is now felt to reduce the drag on liquid coming from the sides, and thus the value is not comparable to those shown in Figure 5, which were obtained with liquid about 1½ in. above the heater.

CONCLUSIONS

For horizontal, semicylindrical heaters in the size ranges tested (1/16- to 1/2-in. diameter) burnout is apparently caused by one of two mechanisms. If the heaters are clean, local conditions on the heater surface must determine the burnout heat flux. Since the surroundings are apparently capable of providing enough liquid to sustain higher heat fluxes, as tests with treated heaters indicate, local conditions on the heater surface must determine the burnout values. It is not yet known which local effects cause burnout, but treating heaters in tap water or wicking probably makes them less susceptible to burnout due to bubbles reaching a critical Weber number (*12*) by the liquid screening action of individual bubbles, or by slow vapor departure rates.

Treating semicylindrical heaters in tap water, or subjecting them to the foreign material entrained by liquid flowing through Fiberglas capillary material, improves surface wetability. Bubbles are more removed from the surface and heat fluxes are apparently limited by liquid inflow from the sides. This mechanism is strongly sensitive to heater diameter, and burnout heat flux may be increased by using smaller diameter treated semicylinders. Heat fluxes of at least 1,230,000 B.t.u./hr. sq. ft. are apparently attainable in pool boiling of water by restricting (with capillary wicking) the distance liquid must flow in from the sides.

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NOTATION

a = acceleration at instant of bubble departure, ft./sec.^{2}
F = force, lb.$_f$, F_D due to dynamics of bubble growth at instant of departure
g = acceleration of gravity, ft./sec.^{2}
g_c = constant, 32.17 lb.$_m$-ft./lb.$_f$-sec.^{2}
q'' = heat flux, B.t.u./hr. sq. ft.
r = radius of bubble at departure; r_b radius of bubble base, ft. or in.
r_H = heater radius; ft. or in.; r_{HA}, distance liquid must travel to reach centermost point on heater, that is apparent radius.

t = time for which heater held at given heat flux, min.

t_i = time to increase heat flux from one setting to another, min.

ΔT_{sat} = temperature at metallic surface of heater minus saturation temperature of liquid, °F.

θ = angle from topmost point of bubble to contact point on surface (Figure 4) radians or deg.

ρ = liquid density, lb.$_m$/cu. ft.

σ = liquid to vapor surface tension, lb.$_f$/ft.

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Gas Absorption Accompanied by a Large Heat Effect and Volume Change of the Liquid Phase

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Gas absorption is always accompanied by a heat effect and volume change of the liquid phase. The magnitude of the heat effect and volume change depends largely on the rate of transfer. In the case of absorption of a sparingly soluble gas these effects are usually negligible, and the process may be considered to be isothermal and isometric. On the other hand in many industrially important operations, such as absorption of ammonia by water, the heat effect and volume change are so large that they cannot be ignored. The problem to be solved in such transfer processes is the one involving simultaneous mass and heat transfer with a moving boundary.

In the past several papers (*1, 4, 7*) have appeared dealing with the moving boundary problem for either mass transfer or heat transfer but not for the coupled transfer process. In this paper a solution is obtained for the coupled process, and the results are compared with measurements of the rate of absorption of ammonia by water.

PHYSICAL MODEL AND ASSUMPTIONS

The physical model as shown in Figure 1, consists of a semi-infinite liquid phase in contact with a pure gas phase. As the gas phase consists of only one component and the liquid is assumed to be nonvolatile, no concentration gradient will exist in the gas phase. The concentration $c^*(\theta)$ at the liquid surface is assumed to be always in equilibrium with the gas phase at the liquid surface temperature $t^*(\theta)$. The liquid surface moves at a velocity $v(\theta)$, which is in direct proportion to the amount of gas absorbed.

Other assumptions involved in the present treatment are listed below:

1. The heat of solution (which may also include heats of reaction) is released instantaneously at the interface when the gas hits the liquid surface.
2. Physical properties and transport coefficients are constant at certain average values.
3. The volume change of the liquid phase is directly proportional to the quantity of dissolved gas; that is partial molal volumes are constant.
4. The Dufour and Soret effects are negligible.
5. The heat transfer between the gas and the liquid surface is negligible.

THE RATE EQUATIONS

Based on the physical model and the assumptions stated above the rate equations and boundary conditions may be expressed as follows:

For mass transfer

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