Film Boiling of Nitrogen with Suction on an Electrically Heated Porous Plate

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Experimental equipment to study the film boiling of liquid nitrogen on a porous heat source with vapor suction was developed. The electrically heated element was a 3.69 in. sintered stainless steel screen, which was also used as a resistance thermometer in conjunction with an a.c. potentiometric circuit. In preliminary work, this mode of heat transfer was unstable, but stability was achieved by placing a porous flow control element on the liquid side of the heater. It was found that the Nusselt number was a function of only the Reynolds number; the heat transfer coefficient was increased by a factor of as much as 2.5 over the heat transfer coefficient in normal film boiling; it was possible to vary independently any two of the three variables, heat flux, flow rate through the plate, and temperature difference; and the generated vapor was considerably superheated. Fluctuations in the local surface temperature were measured in film boiling from both porous and nonporous flat plates, indicating that momentary solid-liquid contacts can occur, for all practical purposes, in film boiling.

To avoid some of the undesirable aspects of the usual modes of boiling heat transfer, a new mode of heat transfer (*1*) has been proposed consisting of film boiling from a porous heat source with the vapor being exhausted through the heat source itself. From a Taylor instability analysis one could predict the possibility of heat transfer coefficients several times greater than the usual film boiling heat transfer coefficients, with an additional advantage in obtaining a superheated vapor. One may further note that since there would be no need for gravity to remove the generated vapor, this mode of heat transfer would be advantageous in reduced gravitational fields. In the analysis, the stabilizing effects of surface tension and evaporation on small perturbations of the liquid-vapor interface were studied, with the assumption that a continuous vapor film separates the liquid from the solid, excess vapor being removed as fast as it is formed by passing through the hot surface. Phenomenologically, exhausting the vapor through the heat source allows liquid to approach closer to the plate, thereby increasing the temperature gradient.

Preliminary experimental work demonstrated that turbulence and pressure oscillations make stable operation of film boiling with suction difficult. Fluctuating local conditions in the heat transfer zone resulted in a portion of the heat transfer surface quenching to a temperature within 100°F. of the bulk liquid temperature, with incomplete vaporization of the fluid sucked through the heating element and extensive thermal stresses. A subsequent modification, which consisted of positioning a porous, nonheat generating element close to the porous heating element on the liquid side, eliminated this bistable condition by preventing the release of vapor into the bulk liquid and by controlling and restricting the flow of liquid to the heated surface.

EXPERIMENTAL APPARATUS

Figure 1 is a cross-sectional drawing of the heat transfer cell. The porous heating element (A) was a sintered, two-layer composite of Regimesh J with a composite thickness of 0.013 in. Regimesh J is a sintered, type 316 stainless steel screen of wire count 165 × 1,400. In Figure 2 the dimensions of the heat transfer surface, the portion of the porous heating element covered by gaskets, and the locations of the welded thermocouples are shown. The overall dimensions of the porous heating element were 1.91 × 1.94 in., with a center square 1.75 × 1.75 in., available for heat transfer to the liquid. Eleven, 30-gauge chromel vs. alumel thermocouples were welded to the bottom of the porous plate by an electric arc discharge, which made the thermocouple junctions an integral part of the heating element. To protect the junction from mechanical stresses and to insulate the leads, the junction and the leads were coated with cement to a diameter of 0.06 in. and for a length of 0.3 in. As a representative sample of all thermocouples made from the same roll of wire, five of the thermocouples were calibrated before being welded to the plate at the boiling points of nitrogen and water and at the freezing points of toluene and water. Two of these were recalibrated after being welded to a sample of the heating element; the calibration was unchanged. All the nonfluctuating, thermocouple electromotive forces were measured with a millivolt potentiometer to a limit of error of ± (0.05% + 3μv.). Copper leads (B, Figure 1) 0.013 in. thick were silver soldered to the heating element, contained between asbestos gaskets (C) 0.063 in. thick with center squares (1.75 × 1.75 in.) removed for the heat transfer area, and sealed with Saureisen cement. The transite flanges (D) connected to a 3-in. I.D. Pyrex exit pipe (E). The porous flow control element (F), 1.5-in. thick, was of sintered silica, with a porosity of 28.8% and an average

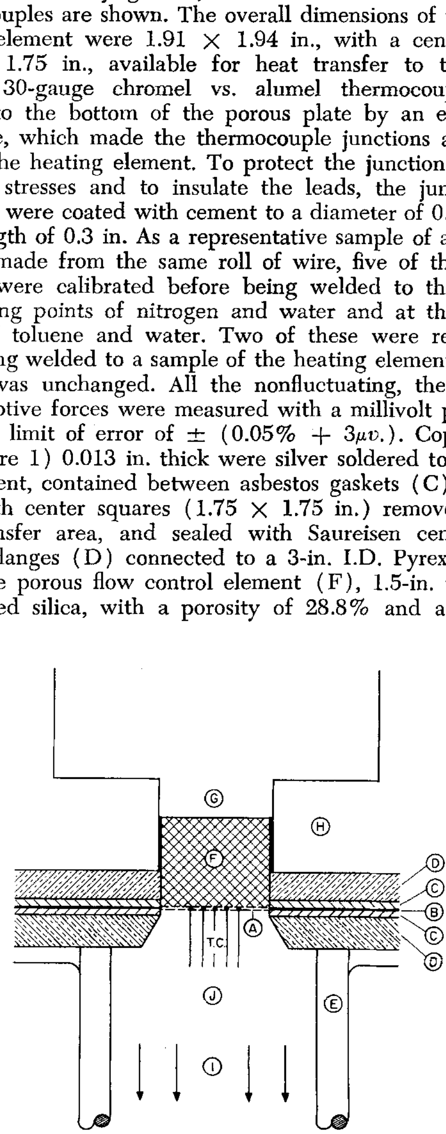

Fig. 1. Cross section of heat transfer cell.

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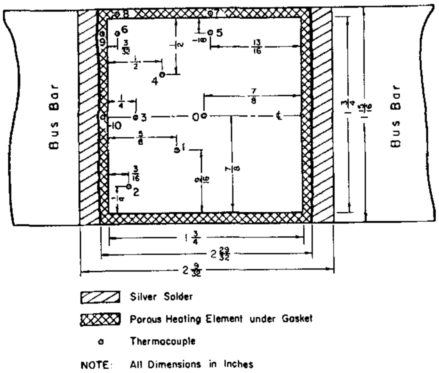

Fig. 2. Heat transfer surface.

pore diameter of 0.004 in. To insure uniform contact, a 0.125-in. layer of glass beads (diameter = 0.03 in.) was positioned between the heating element and flow control element. A pool of nitrogen (G), 5.4 in. in depth, at atmospheric pressure was maintained above the heating element in a styrofoam container (H). The temperature of the exhaust vapor (I), sucked downwards from the heating element through a rotameter by means of a vacuum pump, was measured with two chromel-alumel thermocouples (J) shielded from radiation effects by a bed of 0.25-in. Berl saddles. The pressure below the heating element was measured with a manometer with an 0.827 specific gravity liquid.

The 60-cycle power input to the electrically heated porous plate was controlled by a 5.75-kva. variac. Because of fluctuations in the line voltage, it was necessary to use, between the power lines and the variac, a 3-kva. voltage stabilizing transformer. A 15/1 step down, air cooled transformer rated at 20 kva. gave the necessary high current and low voltage needed by the low resistance heating element. The power input was measured with a voltmeter with a limit of error of ± 1% and an ammeter with a limit of error of ± 0.5%.

Since the heating element was to be used as a resistance thermometer, it was necessary to measure its resistance extremely accurately. The electrical resistance of the heating element was measured during operation by an a.c. potentiometric technique developed by Chiotti (*5*), wherein the potential developed across the heating element is balanced against the potential developed across a known resistance in the secondary circuit of a current transformer, whose primary is in series with the resistance heater. The balance point is found as a minimum on an a.c. null detector. Details of the circuit are given in reference *17*. The resistance vs. temperature curve of the porous plate stayed constant during the period when data for heat fluxes up to 20,000 B.t.u./hr. sq. ft. were taken. The higher power levels with their inherent higher temperatures caused a slow oxidation of the plate to occur, which caused a slight rise in the plate resistivity. The plate was recalibrated between each of the experimental runs at higher power levels. During the investigation of film boiling without the porous flow control element and film boiling with the flow control element at relatively high vapor flow rates through the plate, the local plate temperatures fluctuated considerably. These fluctuations made it necessary to record these points with a recorder in conjunction with a low level preamplifier.

OPERATING PROCEDURE

The following procedure describes the operation of a complete run, that is the operation at a particular power input starting with no flow through the heating element and ending when the flow rate caused unstable operation to occur. First the container above the heat transfer surface was filled with liquid nitrogen to a level of 5.4 in., and the plate was cooled to approximately saturated liquid nitrogen temperature by allowing vapor to exhaust through the plate. The exhaust was then closed and the power input set at the level of interest. All subsequent readings were made at steady state. It required from 2 to 3 hr. to attain steady state when no vapor was exhausting through the plate, which was the first point of a run. The slow drift towards equilibrium was probably due to small changes in the flow patterns in the flow control element. The next point was at approximately the same power input but with vapor being sucked through the plate at a fixed rate. It required from 1 to 2 hr. to attain steady state with vapor flowing through the plate. At the lower flow rates not all of the vapor exhausted through the plate, but at higher flow rates through the plate very little vapor could be observed coming from the top of the flow control element. The flow rate was increased for each succeeding point, and points were taken until a portion of the plate quenched, which was taken as the end of a run, recognized by the fact that a portion of the heating element went to approximately −320°F. At this point, uncontrolled two-phase flow through the plate with incomplete vaporization occurred, and a portion of the vapor again exhausted through the top of the flow control element. The quenched condition was easily detected, and operation terminated at this point. In the stable regime the exhaust was fully vaporized and, in fact, possessed considerable superheat. At higher power inputs, it was necessary to run the no flow point at 10,000 B.t.u./hr. sq.ft. and then increase the power input and flow through the heating element simultaneously until the desired power level was attained. The power input was then kept approximately constant for the rest of the run.

CALCULATIONS AND ERROR ANALYSIS

The calculated average heat transfer coefficient h is defined as

$$h = \frac{Q_o}{A\,(T_{sea} - T_o)} \tag{1}$$

The corrected heat flux was calculated by correcting the measured power input for conduction, convection, and radiation losses. There were two problems associated with calculating power input: the slightly distorted wave form and the appearance of the a.c.-d.c. calibration error. Since the measured a.c. resistance was only 1.6% higher than the measured d.c. resistance, a pure resistance was assumed for the power input calculations. An error of this size could result from the current transformer, calibration of the potentiometric circuit, or be due to reactive components.

To determine the conduction, convection, and radiation corrections, one of the plates was heated in air without the block, and power input was measured. With standard formulas used (*3*), the calculated combined radiation, conduction, and convection losses for a horizontal plate of the experimental design compared within ± 7% of the measured power input up to an average plate temperature of 950°F. Though these corrections are only approximations to a complex situation, this close agreement with data allowed similar corrections to be applied to the experimental data.

To attain as accurate a temperature measurement as possible, two independent measuring techniques were used in conjunction: the electrical resistance of the heating element was calibrated as a function of temperature and measured during operation, and the electromotive forces of eleven thermocouples welded to the bottom of the plate were measured. Over the entire range of operating conditions, the most accurate temperature measurement was the electrical resistance of the plate. The thermocouples were used to calibrate the plate, to help interpret the resistivity measurements, to define the end of stable operation, and to check on the operation of the heating element during no-flow conditions. Because of the extremely dense mesh structure, the element was considered to be a uniform porous material for computational purposes.

When thermocouple data were used, the average surface temperature was calculated in the following way. The surface was divided into eight concentric frames about the center. On the assumption that the thermocouple reading in each frame represented its isothermal temperature, these readings can then be area averaged to give the area-averaged surface temperature T_{sa}. The average d.c. resistance of the heating element can also be calculated from the thermocouples by determining the d.c. resistance of each frame from isothermal plate resistance calibrations and then adding the d.c. resistances of the eight frames, with Kirchhoff's laws. This calculated average d.c. resistance can then be converted to an electrically averaged surface temperature T_{set} with plate calibration data. The difference between these two average temperatures is due to the change in local resistance with temperature for a particular temperature distribution. As described below, this difference was used to convert the electrically measured average temperature to an area-average temperature.

In the calculation and presentation of the results, the a.c. potentiometric temperature measurement was used. This measured resistance was corrected for the portion of the heating element under the gasket and was then converted to the electrically measured, average surface temperature T_{se} with resistance calibration data.

On the assumption that the thermocouples gave the correct relative temperature, although not the correct absolute temperature, the difference $(T_{sa} - T_{set})$ was used to correct T_{se} to the area-average temperature T_{sea}. The difference between the two was usually small, but near the quench point at high heat fluxes the correction became appreciable. The maximum correction occurred at the end of stable operation with a heat flux equal to 44,800 B.t.u./hr. sq.ft., where the correction was 8.6% of the temperature driving force.

In summary, the estimated relative error in heat flux input was ± 3%. The conduction, convection, and radiation corrections add, at most, a relative error of about ± 2.5%, giving a maximum total relative error of ± 5.5% in the corrected heat flux. Not including the effect of temperature nonuniformity, the maximum error in the average temperature difference, $(T_{se} - T_o)$ was ± 14°F., except for 9-10 and 9-11 when it was ± 26°F., and for 9-14 when it was ± 20°F. The correction from electrical average to area average, which includes the effect of temperature distribution, adds an error, depending on the correction, of from 0 to 2% of the temperature driving force.

RESULTS AND DISCUSSION

In Figure 3, the corrected heat flux Q_c/A is plotted as a function of the area-average temperature difference, $T_{sea} - T_o$ for three different types of operation. Curve I represents data taken at the last stable point in film boiling with vapor exhausting through the plate and with the flow control element in place. Curve II represents data taken with this same experimental design, but with no flow through the plate. Curve III represents data taken in normal film boiling on a solid plate which was inserted into the equipment. Examination of curves I and III makes apparent the greater heat flux and rate of increase in heat flux of film boiling with suction compared with normal film boiling. With flow through the plate, the liquid approached closer to the plate, giving an increased temperature gradient and, therefore, heat flux. It should be noted that since the problem of temperature nonuniformity was not completely solved, the full potential of this mode of heat transfer has not yet been attained. The plate always became unstable at the center, the point of lowest temperature. If the temperature of the plate had been uniform at the quench point, the average temperature difference for a particular heat flux at the quench point would have

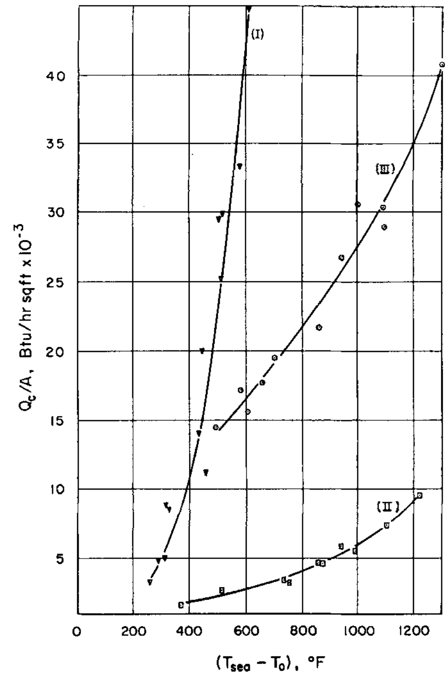

Fig. 3. Heat flux vs. temperature difference for three types of operation.

been considerably lower. Curve II represents the lower limit for heat transfer in the present experimental design and does not represent desirable operating conditions. When no vapor is flowing through the plate, the flow control element prevents the liquid from approaching the plate, thereby reducing the heat transfer coefficient.

Except for recent studies on horizontal flat plates by Berenson (*2*), in which transition boiling data on pentane and carbon tetrachloride extended into the film boiling regime for 150°F., and by Westwater and Hosler (*15*), in which film boiling of Freon-11 and water was studied over a temperature difference range of 280°F., film boiling work has been confined to wires, tubes, and spheres. Experimental heat transfer rates for natural convection, film boiling of nitrogen from horizontal tubes and wires have been reported by Bromley (*4*), Frederking (*6*), and Hanson and Richards (*7*). Hsu and Westwater (*8*) and Weil (*14*) did experimental studies on the film boiling of nitrogen outside of vertical tubes. There are extensive data on the film boiling of other liquids on wires and tubes. A survey of literature on heat transfer from solid surfaces to cryogenic fluids has recently been presented by Richards, Steward, and Jacobs (*11*). Although there are no experimental data for film boiling of nitrogen from flat plates, it is possible to estimate the temperature difference between film boiling on a tube and on a flat, horizontal plate and then to compare the present data with those of Bromley (*4*) for the film boiling of nitrogen on a tube. Berenson (*2*) derived the following relation between the heat transfer coefficient and temperature difference in film boiling:

$$h = 0.425 \left[\frac{k_{vf}^3 \, \Delta H \, \rho_{vf} \, g(\rho_l - \rho_v)}{\mu_f \, \Delta T \sqrt{\dfrac{g_o \, \sigma}{g(\rho_l - \rho_v)}}} \right]^{1/4} \qquad (2)$$

Berenson's equation was derived for a flat plate and was similar to Bromley's equation for natural convection film boiling on a horizontal tube (*4*):

$$h = 0.62 \left[\frac{k_{vf}^3 \, \Delta H \, \rho_{vf} \, g(\rho_l - \rho_v)}{\mu_f \, \Delta T \, D} \right]^{1/4} \qquad (3)$$

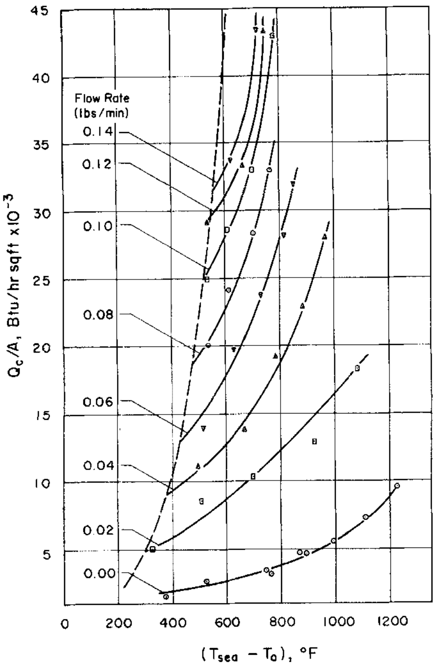

Fig. 4. Heat flux vs. temperature difference for various vapor flow rates.

The only differences are in the constant coefficient and the choice of characteristic length. The constant in Berenson's equation was evaluated from experimental data recorded at minimum heat flux, and the constant in Bromley's equation was evaluated from data recorded over a 1,600°F. temperature range in the film boiling regime.

Dividing Equation (2) by (3) and solving for the heat transfer coefficient for a flat plate one obtains

$$h = 0.685 \left[\frac{D}{\sqrt{\dfrac{g_o\ \sigma}{g(\rho_l - \rho_v)}}} \right]^{1/4} h_t \qquad (4)$$

Using the data for nitrogen and D = 0.350 in., which was the diameter of the tube used, one gets

$$h = 1.19\ h_t \qquad (5)$$

The heat transfer coefficients based on the data given in Curve II of Figure 3 are from 20 to 30% greater than the heat transfer coefficients measured by Bromley; therefore, these data agree quite well with past work. The in-

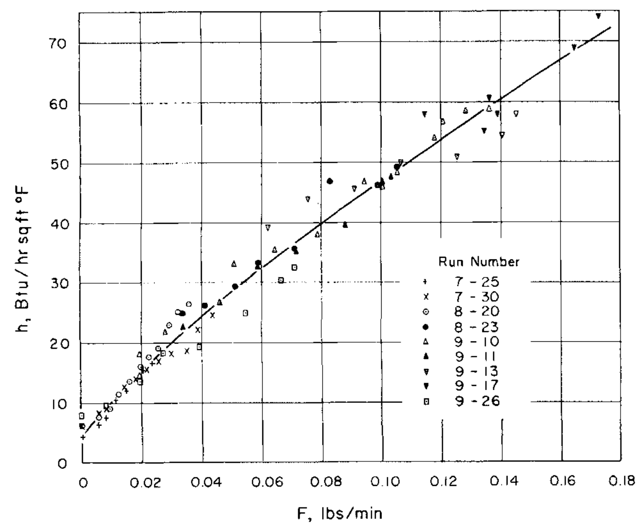

Fig. 5. Heat transfer coefficient vs. flow rate.

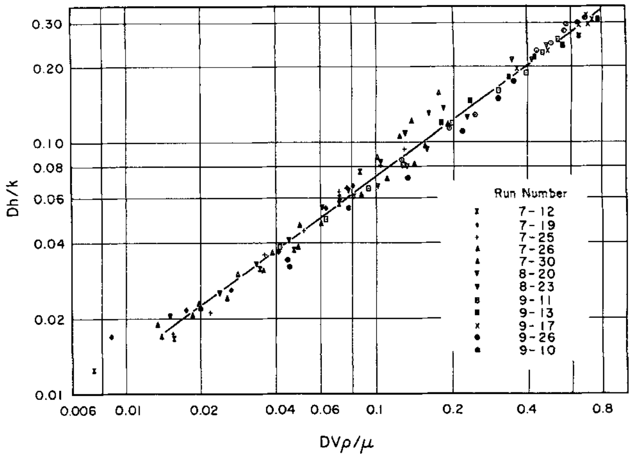

Fig. 6. Nusselt number vs. Reynolds number.

creased heat transfer in the present geometry is to be expected, since the turbulent flow patterns are more intense than would be expected for a tube suspended in a large volume of liquid.

In Figure 4, the heat flux Q_c/A is plotted as a function of area average temperature difference $T_{sea} - T_o$, with vapor flow through the plate F as a parameter. The data used in these curves are interpolated values of the recorded data. This figure shows how it was possible to vary any two of the three variables: heat flux, temperature difference, and flow. The dotted envelope on the left side was the limit of stable operation. The corresponding limit on the right-hand side would be the extension of the zero flow curve. Owing to the very low fluxes and temperatures, it was not possible to make reliable measurements in the region where these two envelopes meet. In normal boiling it is not possible to vary both the heat flux and temperature difference at the same operating pressure. In Figure 5, the heat transfer coefficient h is plotted as a function of the flow rate through the plate F, with a particular symbol used for each run. In Figure 6, the logarithm of the Nusselt number is plotted vs. the logarithm of the

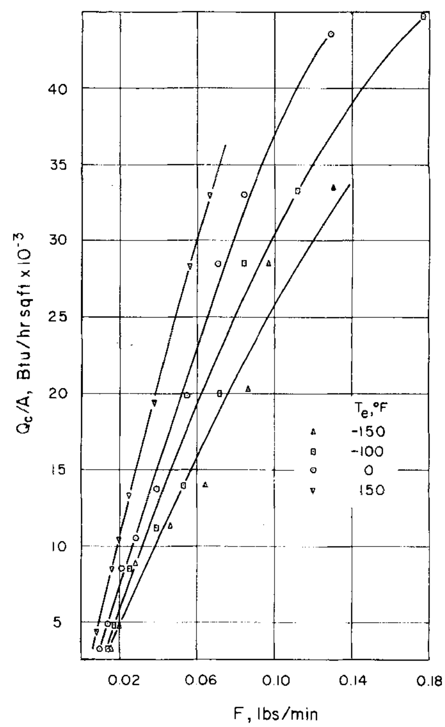

Fig. 7. Heat flux vs. vapor flow rate for various exhaust temperatures.

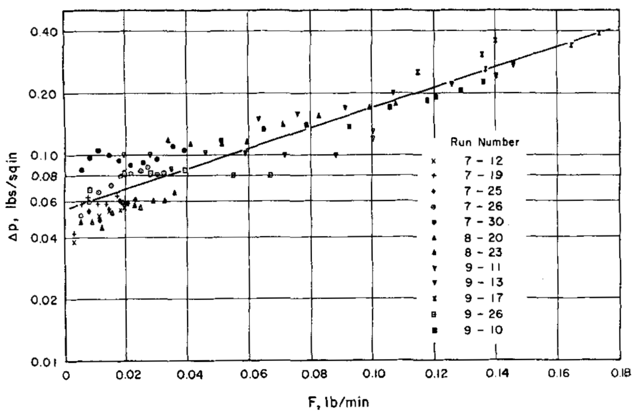

Fig. 8. Pressure drop vs. vapor flow.

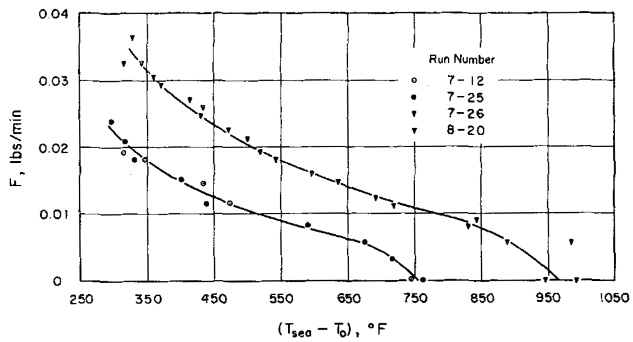

Fig. 9. Flow vs. temperature difference for two repeated runs.

Reynolds number, giving a straight line with the corresponding equation:

$$\frac{hD}{k} = 0.402\left(\frac{DV\rho}{\mu}\right)^{0.73} \qquad (6)$$

In these calculations the thermal conductivity and viscosity were evaluated at the average temperature between the saturated vapor and exhaust temperatures. The diameter D was 3.28×10^{-5} ft., the minimum pore diameter of the sintered screen, which was estimated to be equal to the minimum-diameter particle filtered out by the sintered screen, while V was the superficial velocity. Over the experimental range the variation in the Prandtl number was small, so that this quantity did not appear in the correlation. Since the flow of vapor towards the heated surface increases the average temperature gradient for heat transfer, the increase in the heat transfer coefficient with vapor flow rate is an expected result. On a microscopic scale, heat is transported by conduction, convection, and radiation outside and within the porous heating element; therefore, the heat transfer coefficient is an average value for the combined transport mechanism. In Figure 7, heat flux is plotted as a function of vapor flow rate with exhaust temperature as a parameter. The data points are interpolated values of the experimental data. Only those data for which the exhaust temperature was between —150° and 150°F. are plotted because outside of this temperature range errors due to low flow rates and/or large temperature gradients were present. These curves show that the exhaust had considerable superheat, which is not present in normal boiling; therefore, the energy in the vapor generated by film boiling with suction can be used to produce work at a higher thermodynamic efficiency than that generated by normal film boiling. In Figure 8, the pressure drop across the porous heating element and vapor film is plotted as a function of vapor flow rate through the plate. There is a considerable spread in the data at some flow rates, which is caused by the combined effects of: pressure fluctuations, a slight change in permeability with temperature, local corrosion or plugging, and channeling of vapor and/or liquid. At low flow rates, some of the generated vapor channels up through the flow control element. This is in accord with the Saffman-Taylor study (*12*) of the stability of the interface between two fluids in a porous medium, where it was found that the displacing fluid penetrated the other in the form of rounded-ended fingers when the fluid being displaced was the more dense and viscous. The small pressure drop, between 0.04 and 0.40 lb./sq.in., should be noted.

The per cent of energy input recovered in the exhaust was the measure of the effectiveness of channeling the heat flow away from the bulk liquid. At zero exhaust flow rate, all of the heat was transferred to the bulk liquid. At low flow rates, some of the vapor was not trapped by the flow control element and was transferred into the bulk liquid. At moderate and high flow rates, since all of the vapor exhausts through the porous plate, the per cent energy recovered was an energy balance on the system and was usually between 100 and 125%. The values above 100% are believed due to heat leakage into the exhaust vapor or into the liquid nitrogen, near the heating surface.

To check reproducibility, two of the runs, 7-12 and 7-26, were repeated in runs 7-25 and 8-20, respectively, and are compared in Figure 9. As expected, the maximum deviations were at the beginning and the end of the runs. At low fluxes most of the vapor bubbled up through the block, while the onset of instability at the end of a run is influenced by small variations in conditions.

Because of the low energy content of the thin heating element per unit area of surface, the local surface temperature responded rapidly to any fluctuations in the local heat flux. In the normal film boiling experiments it was found that the local surface temperature fluctuated considerably, as shown by the recorder traces of surface thermocouple outputs. A summary is given in Table 1, and representative recordings of these temperature fluctuations may be found in reference *17*. Stock (*13*) measured fluctuations of ± 2.5°F. in the film boiling of Freon-11 on a horizontal tube. As noted previously, the vapor film on a flat plate is apparently subjected to more violent disturb-

Table 1. Fluctuations of the Local Surface Temperature in the Normal Film Boiling of Nitrogen on a Solid Horizontal Flat Plate

Heat flux, B.t.u./hr. sq. ft.	14,500	15,600	17,800	21,700	24,400	26,800	27,400	30,400	31,400
Average surface temperature, °F.	175	289	341	546	563	621	640	770	686
Thermocouple with maximum fluctuation no.	1	1	4	0	4	3	3	1	3
Maximum fluctuations, ±°F.	11.5	11.6	14.0	13.0	14.0	17.0	13.5	17.0	16.0
Thermocouple with minimum fluctuation no.	2	6	2	6	2	6	6	2	6
Minimum fluctuations, ±°F.	8.1	8.2	9.0	8.0	6.0	7.3	4.5	11.0	4.2

ances from natural-convection currents than on a tube. These fluctuations indicate that it is possible for momentary liquid-solid contacts to occur, for all practical purposes, in film boiling on a horizontal flat plate. It is of interest to note that Stock (*13*) and Westwater and Santangelo (*16*) found essentially no liquid-solid contact in film boiling from tubes. With the flow control element in place, at low exhaust flow rates, strong fluctuations in the local surface temperature were recorded. Fluctuations also appeared in the resistivity measurement and in the exhaust pressure. Since the thermocouples represent blind spots to the vapor exhaust, they are not an accurate measurement of overall plate temperature fluctuations. Neither is the null indicator designed for such measurement; therefore, the measured fluctuations in film boiling with vapor flow through the plate are more a qualitative than quantitative measurement. The null indicator always started fluctuating before any of the thermocouples, which indicated that some liquid was momentarily touching the plate between the thermocouples. With vapor flow through the plate, the center thermocouple always showed the largest fluctuations and invariably quenched first. With heat fluxes below 9,000 B.t.u./hr. sq. ft., the plate quenched before measurable fluctuations occurred.

CONCLUDING DISCUSSION

The experimental results can be explained rather simply. The initial experiments demonstrated that fluctuating local conditions in the heat transfer zone, along with uncontrollable vapor flow, caused film boiling with suction to be unstable. The flow control element increased the stability of operation by controlling the flow of vapor. At low flow rates the heat transfer coefficient was reduced, although the exhaust vapor was considerably superheated. With increasing flow rate the heat transfer coefficient increased. When the vapor film was sufficiently thin, small fluctuations in the local conditions resulted in liquid momentarily touching or penetrating the heated surface, as shown by fluctuations in the average surface temperature, pressure drop, and/or flow rate. At low heat fluxes the plate quenched at this point. At higher heat fluxes, sufficient energy was present to vaporize these tongues of liquid, so that local fluctuations were observed in stable operation. The excess vapor generated during these brief contacts was forced into the flow control element, momentarily preventing liquid from approaching that portion of the heated surface. Depending on the power input, the operation remained stable until these bursts of vapor were removed, immediately upon generation, by a sufficiently large flow rate. At this point, a portion of the plate would quench and remain in this state. Since only a portion of the plate quenched in these runs, the exhaust vapor possessed considerable superheat, even at the quench point. A small reduction in the flow rate through the heating element would produce sufficient excess vapor to cause the surface to dry out, allowing film boiling to begin again.

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NOTATION

A = area of heating element not covered by gasket, sq. ft.
D = minimum pore diameter of sintered screen, ft.
F = flow rate, lb./min.
g = gravitational acceleration, ft./hr.^{2}
g_c = force-mass conversion factor, (lb.$_m$/lb.$_f$) (ft./hr.^{2})
h = heat transfer coefficient for flat element, B.t.u./hr. sq. ft. °F.
h_t = heat transfer coefficient for tube, B.t.u./hr. sq. ft. °F.
ΔH = average enthalpy difference between vapor at its arithmetic average temperature and saturated liquid, B.t.u./lb.$_m$
k = thermal conductivity, B.t.u./hr. sq. ft. °F./ft.
ΔP = difference between vapor-liquid interface and exhaust pressure, lb.$_f$/in.^{2}
Q_c/A = heat flux from uncovered heating element, corrected for radiation, convection and conduction losses, B.t.u./hr. sq. ft.
ΔT = temperature driving force, °F.
T_e = temperature of exhaust, °F.
T_o = temperature of saturated nitrogen at atmospheric pressure, °F.
T_{sa} = area average surface temperature from thermocouple data, °F.
T_{se} = electrically measured average surface temperature, °F.
T_{set} = electrically averaged surface temperature, from thermocouple data, °F.
T_{sea} = average surface temperature, electrically measured, converted to area average, °F.
V = superficial velocity, ft./hr.
μ = viscosity, lb.$_m$/ft. hr.
ρ = density, lb.$_m$ cu. ft.
σ = surface tension, lb.$_f$/ft.

Subscripts

f = film
l = liquid
v = vapor

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