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LOW HEAT-FLUX BOILING

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ABSTRACT

This report covers progress made in the design of a system to study boiling of water from the outer surface of tubes at low values of heat flux (from 5,000 to 100,000 Btu/hr ft^2) and pressures up to 2,000 psia for both natural and forced convection.

The primary and secondary electrical power systems are discussed. The various water piping systems (primary flow, purification, and pressure control loops) are described.

The results of a literature survey are presented. The survey covers effects of heating surface, pressure, gravity, and vapor quality on nucleate-boiling heat transfer. It also discusses heat transfer in two-phase flow.

NOMENCLATURE FOR LITERATURE SURVEY

A	heat-transfer area
a	acceleration
D	tube diameter
g	gravitational acceleration
h	heat-transfer coefficient
k	thermal conductivity
L	tube length
Nu	Nusselt number
Pr	Prandtl number
q	rate of heat transfer
r^*	minimum radius of thermodynamically stable bubble
ΔT	temperature difference
w_b	empirical constant
X_{tt}	Martinelli parameter
δ	laminar film thickness
λ	latent heat of vaporization
ν	kinematic viscosity
ρ	density
σ	surface tension
ϕ	square root of the ratio of the two-phase to single-gas-phase pressure drop for the gas mass velocity
χ	ratio of liquid pressure drop to gas-phase pressure drop if each phase were flowing alone

NOMENCLATURE (Concluded)

Subscripts

a	atmospheric pressure
f	liquid phase
g	gas at vapor phase
ℓ	liquid phase
sat	superheat above saturation

I. THEORETICAL PROGRAM

The theoretical study of the basic mechanisms of low heat-flux boiling has been deferred for the present period.

The literature survey, described separately, has given support to the design phases of the work and will provide the necessary background for the theoretical analysis which will be initiated in the next quarter.

II. EXPERIMENTAL PROGRAM

A. SYSTEM LAYOUT

The design of the experimental equipment, including test vessel, primary flow loop, etc., has proceeded along the lines described in the First Quarterly Progress Report.[1] Auxiliary power for preheating, maintaining loop temperature, and operating circulating pumps and controls will be supplied at 440 volts from the Fluids Engineering Building three-phase bus system. A distribution panel, consisting of circuit breakers, starters, meters, variable voltage regulators, lights, and switches has been designed and quotations have been requested.

The 3000-amp, 12-volt, d-c bus extension work began in December, 1961, and was completed in February, 1962. This permanent bus system will extend to within six feet of the point required for this research. The bus-electrode connection will be described in a separate section.

The flow sheet shown in the previous Quarterly Progress Report[1] is currently valid in most respects. Some piping revisions are anticipated around the condenser and degasifier. Additional sample taps, drains, and vents will be added as required.

The principal revision to the flow diagram will be the addition of a secondary circulating pump, which is a canned rotor pump of 30-gpm capacity at 30 feet head, Chempump Model CFT. It has been made available from surplus used equipment stocks of the USAEC. The secondary pump will be installed to facilitate the purification process during periods of pool-boiling studies when the main pump will not be required to produce flow. It will also provide a back-up to the main pump although at a considerably reduced flow rate.

B. TEST SECTION

1. General.—The test section size is a 6-in. length of 3/4-in. OD tubing, with an 18-gage (.049 in.) wall thickness. Electrode design calculations are based on these dimensions. Tube materials are to be Monel, Inconel, and carbon steel.

2. Electrode design.—The electrodes within the vessel will be fabricated of copper rod and double extra heavy copper pipe (0.840-in. OD x 0.252-in. ID), chrome-plated to avoid contamination of the water.

The electrodes have been designed for fabrication in sections which will be silver-soldered together to form a test assembly. The test section itself will have copper plugs silver-soldered to each end so the section can be immersed in the constant-temperature block for calibration of the thermocouples in place.

Differential thermal expansion of the test section and the test vessel will create a tensile load on the silver-soldered joints of the test section. To minimize this undesirable load, a short, heavy copper spring has been designed to be an integral part of the electrode assembly. Physical stops are incorporated on the electrodes to prevent ejection of the electrodes from the vessel in the event of breakage or burnout of the test section.

3. Bus Bar System.—Current for the resistance heating of the test section will be carried by bus bars from the laboratory bus system. The laboratory bus bars are copper, with a current density of 1000 amp/in.^{2} The connecting buses will be bus bar copper with a maximum current density of 1500 amp/in.^{2} This higher current rating can be tolerated for short sections and will permit less expensive and easier handled connecting buses. The connecting bus layout, providing for both vertical and horizontal electrode orientation, is shown in Fig. 1, Dwg. 308-C.

The 12-volt d-c motor-generator set which provides power for the test section can be controlled best when operated near its rated voltage. The test section, electrodes, and bus bar system have a combined voltage drop of approximately 4 volts at 3000 amp, so an external variable resistor was designed into the d-c power system.

The resistor will consist of a 15-ft length of type 304 stainless-steel tube, 3/4-in. OD by 0.049-in. wall fabricated in a U-shape. It will be internally water-cooled and capable of dissipating 30 kw. Sliding clamps will be employed to furnish the adjustment required to bring the voltage drop of the system near 12 volts for control.

Also included in the bus bar system will be a precision high-current resistor for purposes of accurate current measurement. This measurement, along with the measurement of voltage drop across the test section, will allow the heat generated in the test section to be calculated.

C. TEST VESSEL

The test vessel was described and shown in the previous Quarterly Progress Report.[1] Minor design modifications have been made and quotations were requested for the fabrication of this vessel. It is expected to place the vessel and primary piping loop on order in the next quarter. Although the vessel is an unusual pressure vessel, fabricators have indicated that there will be no special problems with its construction.

D. PRIMARY FLOW LOOP

There have been no major changes in the design of this loop during the past quarter. The main circulating pump order has been held up pending final decision on whether or not a suitable pump can be borrowed or transferred from AEC stock elsewhere. This, in turn, has held up final detail design of the piping system, since the system must match with the pump connections.

E. WATER-PURIFICATION LOOP

The availability of a surplus 30-gpm canned pump, suitable for the pressure, temperature, and fluid being considered, has permitted the design of a more versatile water-purification loop. This smaller pump which has been supplied by the AEC, will be piped in parallel with the main-circulating pump, but with the primary function of producing circulation through the purification loop. For studies involving very low circulation rates, it may be used instead of the main pump, thereby improving control of the circulation rate.

Quotations have been received for the pH and electrical conductivity measurement instruments. Laboratory-owned indicators and recorders will be used with these instruments.

F. PRESSURE-CONTROL LOOP

The pressure-control loop design has progressed to the detailed design of the main condenser, Fig. 2, Dwg. 303 C. The longitudinal axis of the condenser will be inclined at a slight angle to the horizontal, with steam entering and condensate leaving countercurrently at the lower end of the inside tube. The upper end of this tube will be connected to the degassing vent.

The condenser will have compressed air as the cooling medium. This cooling air will pass through the annular space between the inner and outer tubes. Since the air side film coefficient is controlling and varies as the air mass flow rate raised to the 0.8 power, precise control of heat-transfer rates can be attained by varying the air flow rate. This flow rate, and corresponding heat-removal rate, will be slightly less than the net heat input less heat losses through insulation. The excess heat input will permit final pressure control by a small but finite bleed of steam from the Research Controls valve, actuated by a pressure controller.

The remainder of the pressure-control loop will be as previously described except that the make-up injection pump will be placed on manual operation instead of automatic high-level—low-level control. A low water-level alarm and safety shut-down switch will be included to permit unattended periods of safe pressurization. A suitable low-level switch could not be found commercially available. Therefore a special switch is being designed which utilizes the change in heat-transfer coefficient between a heated solid surface and a vapor as compared to a boiling liquid.

III. LITERATURE SURVEY OF NUCLEATE-BOILING HEAT TRANSFER

The number of variables which affect boiling is surprisingly large. These include, for example, viscosity, specific heat, density, thermal conductivity, and latent heat of vaporization, type of the liquid, type of the heating surface, gravity, vapor quality, and interface conditions, the latter being difficult to characterize. Among these many variables, some are fixed for specific experiments since the liquid is fixed by the purposes of the study. Several of the important variables and their influence on boiling will be discussed in this section.

A. EFFECT OF TYPE OF LIQUID ON BOILING HEAT TRANSFER

For water at 1 atm, many observations have been made.[2-7] Although the numerical observations vary widely, general agreement exists that the maximum heat flux in pool boiling is between 300,000 and 400,000 Btu/hr ft^2. This figure is greater than the maximum for organic liquids by a factor of about three to eight times and depends on the heating surface, its surface conditions, etc.

The reported values of the critical temperature difference vary from about 40° to 90°F, with most observations being close to 45° to 50°F.

For other common liquids, the nucleate-boiling behavior is much like that of water. Typical boiling curves at one atmosphere are available in the literature for ethanol, isopropanol, methanol, ether, benzene, n-butanol, kerosene, n-propane, n-pentane, n-heptane, isobutanol, ethyl acetate, trichloro-trifluoro ethane, dichloro-difluoro methane, methyl chloride, sulfur dioxide, and n-butane, etc.[8-11]

These boiling curves give maximum heat fluxes in the order of 50,000 to 150,000 Btu/hr ft^2, which are small compared with that of water. The critical temperature difference is about 40° to 90°F, essentially the same as for water. The heat-transfer coefficients for aqueous solution of organic substances are usually lower than that of water.

For cryogenic liquids, Merte and Clark[12] made a study of the boiling heat transfer of liquid nitrogen at both standard and near zero gravity. The boiling curve shows that the maximum heat flux is between 40,000 and 50,000 Btu/hr ft^2 in a standard gravitational field, which is lower than those of common liquids. The critical temperature differences vary from about 20° to 30°F, which is also lower than water and other common liquids.

Drayer and Timmerhaus[13] have determined boiling heat-transfer coefficients of liquid hydrogen for a ΔT range of 0.06° to 1.17°F and a heat-flux range of 23 to 978 Btu/hr ft^2.

One can conclude that water is definitely superior from the standpoint of maximum heat flux to the other nonmetallic liquids, including cryogenic liquids, as a heat-transfer agent in nucleate boiling.

Liquid metals possess the desirable characteristics of heat-transfer media such as very low viscosities and high thermal conductivities. The high boiling points make liquid metals usable except at high temperatures. Lyon et al.[7] give measured values of heat-transfer coefficient h at a temperature difference of 10°F as: h for water is 500 Btu/hr ft^2 °F; h for mercury and a wetting agent is 3000; and h for NaK is 20,000. These results demonstrate that liquid metals are better than water as heat-transfer agents in nucleate boiling.

Chang[14] has drawn the following conclusions on the effect of the type of liquid on boiling heat transfer:

1. The high heat-transfer rate in nucleate boiling is due principally to the pumping action of a bubble.

2. Due to the intense turbulence caused by bubbles, transport properties with a molecular character of the liquid, namely, thermal conductivity and viscosity, have little effect on the heat transfer.

3. Liquids that possess larger values of density, specific heat, boiling point and low values of surface tension give high heat-transfer rate. Liquids which have large variations of vapor pressure with temperature are always desirable for boiling heat transfer.

B. EFFECT OF HEATING SURFACE ON NUCLEATE BOILING

Surface conditions, such as roughness, wetting character, contamination, and crystal structure are known to affect considerably the heat transfer in nucleate boiling. It controls (i) the effective solid-liquid interface area, (ii) the mean size of the originating bubbles, (iii) the frequency of bubble generation from each site, (iv) the population of active bubble sites, and (v) the superheat of incipient boiling. All these factors can be considered to affect nucleation.

Jakob and Fritz[15] studied the influence of heating surface upon the formation of steam bubbles. For this purpose, surfaces having different degrees of roughness were used for comparison. Three typical shapes of steam bubbles were obtained on the heating surfaces, one on the unwetted surface, one on the partially wetted surface, and one on the totally wetted surface. It was also found that for the same projected area, a grooved copper surface gave much higher heat-transfer coefficients in the range of moderate heat flux than those for the smooth chromium surface. The latter was also confirmed by the experimental results of Sauer[16] for boiling ethanol at atmospheric pressure with high flux and Deutsch and Rhode[17] for boiling distilled water at atmospheric pressure with high heat flux. Bonilla and Perry[10] found by boiling ethanol at atmospheric pressure on a horizontal flat plate that the heat-transfer coefficient is higher on a freshly polished surface than an aged surface. Mead, Romie, and Guibert[18] found that for boiling water at the same temperature difference the heat-transfer coefficient is higher on the copper than the other metals. Corty and Foust[19] boiled ether, pentane, and Freon-113 (trichlorotriflouro ethane) from horizontal, electrically heated copper and nickel surfaces. It was found that the heat-transfer coefficients increase with increasing roughness of the surface. This is partially due to the surface-area increase with an increase in surface roughness. The other reason is given by the authors in terms of gas-filled pits on the metal surfaces. A fine polish is presumed to give a narrower range of pit sizes than the coarse finishes so that a high superheat is required to get nucleation. An equation is established for the estimation of the heat transfer coefficient: $h = C(\Delta T)^n$ where C is a constant the exponent n varies from 12 for a smooth surface to 24 for the roughest surface.

The geometric arrangement and size of a heating surface also affects the boiling heat transfer. Jakob and Linke[20] found by comparing the boiling data for a given fluid boiled with horizontal and vertical surfaces that the results were nearly identical for both cases. The effect of diameter has been studied for horizontal platinum wires by Rinaldo.[21] The diameters he used were in the range of 0.004 to 0.024 in. It is reported that wire diameter affects substantially the heat flux in the region of natural-convection boiling and in the film boiling, but has little effect in the nucleate boiling. Meyers and Katz[22] found that for boiling refrigerants the heat-transfer coefficient of one horizontal tube was different from that of the other tubes in the same bundle. The effect of length of pipe on critical heat flux in forced motion of a steam and water mixture was studied by Styrikovich and Faktorovich.[23] It was found that the critical heat flux decreases with increase in the ratio of length and diameter.

C. EFFECT OF PRESSURE ON NUCLEATE BOILING

Pressure is a vital factor for nucleate boiling. For a large number of liquids boiling in pools at one atmosphere pressure and less, a decrease in saturation pressure causes a decrease in heat flux and vice versa for a given temperature difference. This is a conclusion obtained from the experimental data of Cryder and Finalborgo[11] and Braunlich[24] for water under reduced pressures, Addoms[3] for water at elevated pressures, Kazakova[25] for water at pressures from 14.7 to 2280 psia, Bonilla and Perry[10] for organic liquids such as n-propane, n-pentane, n-heptane, benzene and ethanol, Kaulakis and Sherman[26] Insinger and Bliss[9] for isopropanol, and Cichelli and Bonilla[27] for a number of organic liquids.

The maximum heat flux and the critical temperature difference are also functions of pressure. Cichelli and Bonilla[27] show that the pressure of maximum heat flux for water, ethanol, benzene, propane, pentane, and hexane is at about 35% of the respective critical pressures.

The maximum superheat for nonboiling liquids is also pressure dependent. The experimental results for nonboiling water by Mead et al.[18] show that the degree of superheat decreases with increase in pressure and the possible superheat (and therefore the temperature difference) decreases to zero as the critical pressure is approached.

For the empirical equations for correlation of heat-transfer data in boiling, Jakob[5] has proposed an equation which expresses the influence of pressure on heat transfer:

$$\mathrm{Nu} = \frac{h}{k}\sqrt{\frac{\sigma}{\rho_f - \rho_g}}$$

$$\mathrm{Nu} = 31.6\,\frac{\nu_a}{\nu}\left[\left(\frac{\rho_{f,a}}{\rho_f}\right)\left(\frac{\sigma}{\sigma_a}\right)\left(\frac{q/A}{\rho_{g,a}\lambda_a w_{b,a}}\right)\right]^{0.8} \qquad (1)$$

where the subscript a relates to atmospheric pressure, and $w_{b,a}$ is an empirical constant.

Bonilla and Perry[10] considered the influence of k and c_p and modified Eq. (1) so that it would correlate the results of their experiments with water, ethanol, butanol, acetone, and numerous binary mixtures of these substances boiling on horizontal plates. Their equation is:

$$\mathrm{Nu} = 16.6\,\frac{\nu_a}{\nu}\left[\left(\frac{\rho_{f,a}}{\rho_f}\right)\left(\frac{\sigma}{\sigma_a}\right)\left(\frac{q/A}{\rho_{g,a}\lambda_a w_{b,a}}\right)\right]^{0.73}(\mathrm{Pr})^{0.5} \qquad (2)$$

Both Eqs. (1) and (2) yield too small values at high pressure. Besides these two equations, Insinger and Bliss,[9] Cryder and Gilliland,[28] and Cryder and Finalborgo[11] have also proposed equations for the precalculation of the pressure influence.

Jens and Lottes[29] correlated a great number of test data for the boiling of high-pressure water into a single equation which shows that the maximum heat flux is proportional to an exponential function of pressure.

D. EFFECT OF GRAVITY ON BOILING HEAT TRANSFER[30]

Space applications of small nuclear reactors cooled by boiling liquid media has necessitated a better understanding of gravity effects on the heat-transfer process. Zero gravity environments create rather unusual conditions for processes which function due to density differences. It may be necessary to replace the normal gravitational forces with others, perhaps centrifugal, which will permit the mechanisms usually operative to function at or above their normal efficiency. Investigations using vortex tubes have already demonstrated tremendous increases in the maximum heat flux that can be transferred from surfaces to fluids without incurring burnout.

Merte and Clark[31] made a study of the influence of system acceleration on pool-boiling heat transfer in saturated distilled water, at approximately atmospheric pressure. The heating surface was a flat disc 3 in. in diameter, with acceleration vector (1-21 g's) away and normal to it. At low constant values of heat flux, ΔT_{sat} decreased as acceleration of the system increased. This is attributed to the increasing contribution of natural convection with acceleration. At high values of heat flux, ΔT_{sat} increased with increasing acceleration. In other words, during the transition from simple natural convection to nucleate boiling, higher acceleration gave higher heat flux, but a reverse trend took place in vigorous boiling. Some data are presented showing the influence of subcooling with the system under acceleration. Non-boiling data in the same range of a/g are presented.

Costello and Tuthill[32] used a flat ribbon electrically heated and mounted near the periphery of cylinder filled with distilled water at essentially atmospheric pressure. The system was spun about its axis producing effective accelerations normal to surface of from $a/g = 20$ to $a/g = 40$. The heat flux varied from $q/A = 100{,}000$ to $200{,}000$ Btu/hr ft^2. It was found for a given heat flux that ΔT_{sat} increased with increasing acceleration, resulting in a decrease in the "heat-transfer coefficient." This increase in ΔT_{sat} amounted to approximately 5-7°F for an increase in a/g from 1 to 40.

Costello and Adams[33] have measured the maximum heat flux for water from a carbon cylinder at approximately one atmosphere for a/g from 1 to 44. The acceleration was normal to the axis of the cylinder which was electrically heated. In other respects their test apparatus was similar to that previously reported by Costello and Tuthill.[32] The relationship between $(q/A)_c$ and a/g follows the 1/4 power law for a/g in the range from 10 to 44. Below a/g of 10 a power law representation between these quantities was also found but with an exponent somewhat less than 1/4.

Gambill and Greene[34] attained a critical heat flux of 55×10^6 Btu/hr ft^2 with water flowing in a vortex in an electrically heated tube. This was attributed to the effect of the centrifugal acceleration estimated to be 18,000 times normal gravity on the bubbles forming at the heating surface. However, the contribution of forced as well as free convection could not be isolated.

Siegel and Usiskin[35] performed a photographic study of boiling water at one atmosphere from several heater configurations in the absence of a gravitational field. No attempts were made to measure heat flux or temperature.

Usiskin et al.[36] found that the velocity of rising bubbles decreased as gravity was reduced while the bubble diameter was proportional to $g^{-0.285}$ whereas theoretical considerations would have predicted $g^{-0.5}$. Measurements of the critical heat flux from a platinum wire 0.0453 in. in diameter were made by Usiskin et al.[36] in saturated distilled water in various force fields of $0 \leq \frac{a}{g} \leq 1$. The burnout heat flux decreased with reduced forced fields but still had a finite value at $\frac{a}{g} = 0$. Measurements were also made of bubble sizes at departure and bubble rise velocities with reduced gravities.

Merte and Clark[12] have studied the boiling of saturated liquid nitrogen at atmospheric pressure from a 1-in.-diameter sphere for standard gravity and at near-zero gravity for 1.4 sec duration. The sphere is used as a dynamic calorimeter for continuous measurement's from film through nucleate boiling. In the nucleate boiling region, the characteristics are the same as at standard gravity, indicating perhaps that buoyant forces play a minor role in promoting the turbulence associated with boiling.

Li[37] studied various liquid configurations, based on the principle of minimum energy, for containers practically filled with a liquid and subjected to zero gravity. Consideration of tank outlet vents under the condition are examined. For liquids which wet the container wall, it is probable that the final zero gravity configuration is a wetted wall with an internally centered gas bubble. For nonwetting liquid roughly the opposite effect is anticipated.

Reitz[38] studied boiling and condensing mercury with zero gravity using parabolic flight of an aircraft. No quantitative heat-transfer measurements were made. The author discusses problems regarding slug motion of mercury in flow passages and undesired movement of condensed mercury back into boiler which he encountered in his study.

Tursela[39] discusses general problem areas of heat transfer, and those anticipated in future space vehicles. Tests are described of the behavior of gases released in fluids and in mercury condensing tests. Presentation is qualitative.

With an empirical modification of the nucleation theory developed by Volmer and Eyring from the Maxwell-Boltzmann distribution law, Chang[14] obtained an equation for the heat transfer in vigorous boiling. This equation includes effects of system acceleration and velocity of forced convection.

Its validity has been demonstrated by the correlation of test data for eighteen different systems under different pressures.

E. EFFECT OF VAPOR QUALITY ON NUCLEATE-BOILING HEAT TRANSFER

The influence of net vapor generation on the heat-transfer coefficient in the nucleate boiling regime parallels forced convection effects. In the low-quality regions, the vapor phase will likely remain dispersed in the liquid matrix, thus resulting in a reduction of the average fluid density. Under these conditions slip can be considered negligible and an increase in the flow rate will occur. The film at the heating surface will remain essentially the same, except that the boundary-layer thickness will decrease as the velocity is increased. Eventually a velocity will be reached such that the bubbles are sheared from the wall shortly after nucleating. At this point the film thickness has been reduced to a point where it no longer offers the resistance to heat transfer that it would at lower flow rates. A given heat flux can be sustained at lower temperature differences, and hence the surface temperature drops. This in turn deactivates sites and decreases the vapor generation at the surface. The effect of the growth and collapse of bubbles on the boundary layer becomes less significant.

Sterman and Styushin[40] observed that the critical heat flux was increased by quality increases. Their observations with isopropyl alcohol in stainless-steel tubes showed that the critical flux was always approached first in the low-quality regions. They postulated that, since bubbles are removed from the surface at smaller diameters for increased flow rates, a greater number of sites can be activated before the growing bubbles begin to merge and blanket the surface. This requires a greater temperature difference at the critical point, and hence a greater heat flux.

Styrikovich and Faktorovich[23] investigated the effect of L/D and exit quality of steam on the peak heat flux with a vertical stainless-steel tube heated by a-c current. They observed that at a pressure of 382 psia, as the quality increases for constant mass velocities the peak (DNB) heat flux first increases and then decreases, passing through a maximum. At a pressure of 2640 psia, the difference in the specific volume of liquid and vapor is much less, and no maximum peak heat flux is observed.

Dolezhal <u>et</u> <u>al</u>.[41] conducted investigations to determine the burnout heat flux for water with forced convection in circular tubes over the ranges of pressure between 100 and 200 atmospheres (1470-2940 psi); of mass flow rate between 0.369×10^6 and 1.48×10^6 lb/hr ft^2, and of exit quality between 0 and 60%. It was shown that the critical heat flux decreases as the exit quality increases at a pressure of 2060 psia.

Bennett <u>et</u> <u>al</u>.[42] conducted experimental burnout heat flux measurements with a 0.375-in.-diameter heated tubular element 29 in. long centered inside a 0.546-in.-ID channel forming an annulus through which steam and water mixtures flowed upwards. The particular conditions tested were: pressure, 1000 psia; mass flow rate, 0.5-3.3 x 10^6 lb/hr ft^2; and steam quality, 5-65% steam by weight at exit. Their experiments showed that, at low qualities, the burnout heat flux exhibits a roughly constant value which falls away rapidly to a very low value when a critical range of exit steam qualities is reached. In general, the burnout heat flux was found to decrease with increasing steam quality and also increasing mass velocity (directly contrary to the Bettis burnout heat flux design equation at 2,000 psia).

Collier[43] showed that at burnout conditions the nature of the relationships between the main variables of mass velocity, steam quality and pressure are much more complex than has been suggested by DeBortoli et al.[44] particularly at the lower pressures below 2000 psi.

Mumm[45] observed that the heat-transfer coefficient increased with quality for qualities ranging up to 50%. For higher values a rapid decrease in the coefficient was observed with burnout resulting for qualities of about 70%. His correlation for the Nusselt number includes quality as a parameter. McAdams, Woods, and Bryan[46] and Rohsenow and Clark[47] observed an increase in the heat-transfer coefficient with quality increases.

Most investigators[30] agree that, for qualities below 50%, improved coefficients will be observed as quality increases. However, at higher qualities considerable disagreement exists about the exact behavior to be expected. An examination of the flow pattern sheds some light on the heat-transfer phenomena. At low qualities the flowing stream is essentially liquid with vapor dispersed as a discontinuous phase. At higher qualities the vapor coalesces, but liquid remains as the continuous phase. For sufficiently high vapor velocities, annular flow eventually develops such that vapor with dispersed liquid droplets moves along the tube axis while a liquid film flows along the tube wall. For liquids which wet the surface, high heat-transfer coefficients persist in this flow regime.

Forced convection effects have probably suppressed any surface boiling, but the high velocity of the gas phase through the core removes all but a thin liquid film at the tube wall, thus reducing the resistance to heat transfer. Eventually the liquid film is reduced to a point where it is difficult to detect. This stage is referred to as fog or mist flow. However, surface is still supplied with sufficient liquid to remove the necessary heat load by vaporization. As the quality continues to rise, a point is finally reached where insufficient liquid reaches the surface to dissipate the high energy input. This "dry wall condition" results in rapid temperature increases at the surface, and burnout occurs. Investigators refer to this type of critical condition as two-phase burnout.

Several investigators have measured high-quality, heat-transfer coefficients. McAdams et al.[46] observed for water-steam a drop in heat-transfer coefficient for qualities above 40% at 24 psi and 71 psi. Similar phenomena were also discovered for benzene-vapor. Dengler[48] observed three mechanisms operative over the quality range he studied. At low qualities nucleate boiling seemed to control; at higher qualities forced convection appeared to dominate. For qualities from 74% to 84%, or mass flow rate from 0.17×10^6 to 0.044×10^6 lb/hr ft^2, sharp decreases in the heat-transfer coefficient were observed. This phenomenon was attributed to "dry wall conditions."

Parker and Grosh[49] studied the heat transfer characteristics for mist flow (steam and water) droplets flowing vertically upward in a tube. Heat flux was varied from 3,020 to 20,700 Btu/hr ft^2 with inlet qualities from 89 to 100%. Their results showed that equilibrium was not necessarily attained between the droplets and vapor and that considerable superheating of the vapor was possible in the presence of droplets. They also observed the heat-transfer coefficient to be a strong function of surface temperature. Above a certain critical temperature, spheroidal behavior was observed with coefficients approximately the same as for dry steam. Surface temperatures below this critical temperature produced coefficients 3 to 6 times greater than dry steam values. Flux and quality effects on this temperature appeared interrelated. Higher qualities and/or fluxes tend to promote the spheroidal state. Any method of directing the dispersed liquid phase toward the walls is likely to increase the heat-transfer coefficient in the very high quality regions.

Guerrieri and Talty[50] have attempted to separate the mechanisms of boiling and convection in high-quality heat transfer. They represent the following expression for the two-phase heat-transfer coefficient:

$$h_c = 3.4\ h_\ell \left[\frac{1}{X_{tt}}\right]^{0.45}$$

where h_ℓ is the single-phase liquid coefficient given by the Dittus-Boelter equation and X_{tt} is the Martinelli parameter. They relate boiling film coefficients when superimposed on convective effects by the following formula:

$$h = 0.187\ h_c \left[\frac{r^*}{\delta}\right]^{-5/9}$$

where r^* is the radius of a minimum-sized thermodynamically stable bubble and δ is the laminar film thickness.

These investigators and others all concur in the conclusion that a convective mechanism becomes controlling for high-quality systems.

IV. LITERATURE SURVEY OF HEAT TRANSFER IN TWO-PHASE FLOW

A. TWO-PHASE FLOW REGIMES

The flow patterns are correlated as a function of different flow parameters. Alves[51] correlates flow pattern regimes for horizontal air-water and air-oil flow in 1-in. pipe as a function of superficial gas and superficial liquid velocity. Bergelin and Gazley[52] correlate flow regimes for horizontal air-water flow in 1-in. tubes as a function of water and air flow rate. White and Huntington[53] correlate the flow pattern regimes for kerosene-natural gas flow in horizontal 1-, 1-1/2-, and 2-in. plastic tubes as a function of gas and liquid mass velocity (lb/hr ft^2). Johnson and Abou-Sabe[54] correlate flow patterns for horizontal air-water flow in a 0.87-in.-ID tube as a function of air rate and water rate. Baker[55] proposed a correlation which included parameters intended to account for different fluid properties. Vohr[56] compared the individual correlations cited above by constructing a table in which the flow pattern regimes were compared for a liquid velocity of 0.5 ft/sec and a gas velocity varying from 1 to 100 ft/sec. Similar investigations for vertical pipes were made by Govier[57] for vertical air-water flows in a 1-in. pipe.

B. HEAT TRANSFER

Verschoor and Stemerding[58] studied heat transfer to upward flowing air-water mixtures in 14-mm tube with a 400-mm test section. Their data show that the heat-transfer coefficient for a given mass flow rate of liquid first increases as the concurrent gas flow rate increases but then in some cases reaches a maximum. Similar results were observed by Johnson and Abou-Sabe[54] for horizontal flow.

Dengler[48] and Addoms[3] have reported heat-transfer data for the steam-water system in a vertical tube at low pressure over the entire steam quality range. Guerrieri and Talty[50] published a study of heat transfer to a number of boiling organic materials carried out in two single-tube vertical evaporators. In both these studies, heat transfer was primarily convective over the major part of the range corresponding to the dispersed flow regimes. The observed heat-transfer coefficients could be related to the single-phase heat-transfer coefficients in a way similar to that proposed by Lockhart and Martinelli for the prediction of pressure drop in two-phase systems. For two-phase flow of steam-water mixtures in horizontal tubes, Mumm[45] investigated the heat-transfer characteristics over the quality range between 0 and 60% and at operating pressures between 45 and 200 psia. Wood[59] and McAdams *et al.*[46] obtained heat-transfer data in a horizontal four-pass evaporator over the full range of steam quality at substantially atmospheric pressure.

Dukler[60] obtained numerical solutions for the two-phase annular, downward flow in tubes by the application of the classical von-Karman-Prandtl-Nikuradse work. Hewitt[61] studied the application of the Dukler analysis to upward concurrent flow in a tube, both fluid dynamics and heat transfer being considered.

Much of the previous work has involved only moderate heat flux. Bennett et al.[62] investigated the high heat-flux region up to 160,000 Btu/hr ft^2 to assess the suitability of two-phase coolants for nuclear power reactors. The two-phase steam-water mixtures in the liquid dispersed region at substantially atmospheric pressure was heated in an electrically heated test section which consists of a vertical glass tube fitted with an electrically heated co-axial stainless-steel tubular core. Two sizes of test section were employed, having 0.375- and 0.623-in. OD steel tubes inside 0.552- and 0.866-in. ID, respectively, glass tubes. Heat-transfer measurements have shown the existence of three regions of heat transfer as the steam quality was increased from zero to 100%, as reported in Section H2 of the last report. In the nucleate-boiling region, the heat-transfer data were correlated by the Rohsenow equation. The experimentally determined value of the constant describing the liquid-surface combination was 0.0132. In the forced-convection region, the experimental coefficients were correlated by the equation of the form proposed by Dengler[48] and Guerrieri and Talty.[50]

Wallis[63,64] analyzed the flow regimes which occur during one-dimensional vertical two-phase flow. Criteria for the transition between flow regimes are used to show that the maximum and minimum heat fluxes in pool boiling from a horizontal surface can be regarded as hydrodynamic phenomena. Experiments were conducted in which the dependence of the maximum heat flux on fluid properties was reproduced in a two-component system using various gas-liquid combinations.

Collier[65] studied the burnout problem and the hydrodynamic and heat transfer characteristics in two-phase flow systems. Collier and Lacey[66] measured heat transfer characteristics of steam or steam-water mixtures flowing in annuli, at pressure up to 1500 psi. The results fall generally below the Colburn relationship and with a high slope, particularly at the lower Reynolds numbers.

C. PRESSURE DROP

Boelter and Kepner[67] determined pressure gradients for two-phase flow of air-water and air-oil through 3/4 and 1/2 in. pipes. Martinelli et al.[68] proposed a method to correlate the value of Φ, which is the square root of the ratio of the two-phase to single gas-phase pressure drop for the gas mass velocity, using a two phase flow modulus X, which was a function of the fluid properties and flow rate. Later on Martinelli[69,70] presented an improved empirical correlation of Φ with a new parameter X, which was defined as the ratio of the pressure drop to gas phase pressure drop if each phase were flowing alone in the duct.

Gazley[52] developed a correlation predicting values about 25% below that of Martinelli's. For air-water systems, Richardson's correlation[71] does a satisfactory job, but as was demonstrated by Fohrman,[72] it has limitations for other systems. Fohrman also showed that the Martinelli correlation is valid for liquids having viscosities from 1.1 cp to 150 cp. As the liquid viscosity increased to 500 cp the data fall about 15% below Martinelli's prediction. Martinelli and Nelson[73] proposed a method of calculating pressure drop during forced circulation boiling of water. Isbin et al.[74] proposed a correlation for the steam-water system. Experimental data covered a wide range of flow and qualities. Armand[75] correlated the ratio of the two-phase pressure gradient to the liquid pressure gradient as a function of the volumetric steam content. He considered the ratio of the volumetric steam content to the fraction of pipe cross-section occupied by steam as a parameter. Bankoff[76] demonstrated the relationship between this parameter, the volume fraction and the slip ratio. This relationship when combined with the void fraction and density ratio will yield the quality. It is then possible to predict the two-phase pressure drop.

In England, as in the U.S.A. and the U.S.S.R., the civil and military power reactor programs have been based on water cooled reactor systems. As the result, the heat transfer, as well as pressure drop in two-phase flow, have been extensively investigated experimentally. Bennett and Thornton[77] have measured the pressure drop for the upward flow of air-water mixtures in vertical tubes and annuli. Collier and Hewitt[78] analyzed the pressure drop data presented by Bennett and Thornton, and the measurements of film thickness and entrainment rate (or film flow rate). The inter-relation of pressure drop-film thickness and pressure drop-entrainment rate is established. Hewitt et al.[79] have measured pressure drop and liquid holdup in air-water flow in a 1-1/4 in. diameter pipe for air rates of 200-500 lb/hr and water rates of 20-1000 lb/hr. The flow was, in all cases, in the annular regime. The results were analyzed in terms of Lockhart and Martinelli holdup and pressure drop calculations and also in terms of film velocity profile theories. With the assumption of homogeneous flow, and two phase friction factor being the same as for single phase liquid flow, Owens[80] has derived two equations representing two-phase pressure gradient; one for isothermal case and the other for the case with vaporization. Agreement between his theory and experimental results is good.

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