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SUBJECT: A Study of Burnout Heat Fluxes Associated with Forced-Convection, Subcooled, and Bulk Nucleate Boiling of Water in Source-Vortex Flow

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A STUDY OF BURNOUT HEAT FLUXES ASSOCIATED WITH FORCED-CONVECTION, SUBCOOLED, AND BULK NUCLEATE BOILING OF WATER IN SOURCE-VORTEX FLOW

I. Summary

A preliminary experimental study has been made of nucleate boiling of subcooled water in forced-convection source-vortex flow in horizontal tubes of small diameter, with particular emphasis on the maximum or "burnout" heat flux obtainable. To the authors' knowledge, this is the first such study which has appeared in the literature. By "source-vortex flow" is meant fluid flow with both axial and tangential velocity components; more specifically, it refers in the present instance to the flow of a swirling or whirling fluid from one end of a tube to the other.

In this investigation, liquid water at room temperature was pumped through a spiral-ramp vortex generator at the inlet of an electrically-heated horizontal tube. Power dissipated across the test section was increased at constant flow rate until burnout of the tube occurred. This burnout point represents the maximum heat flux obtainable under boiling conditions and is caused by the local bubble population or coverage on the inner tube surface reaching some critical value which prevents removal of heat at a rate equal to that of generation within the tube wall. Since, with electrical heating, the heat flux is imposed independently of thermal resistance, the tube absorbs the excess heat and either undergoes mechanical rupture or fuses. The time required for burnout depends upon the level of heat flux at the burnout point and on the heat-storage capacity of the tube;

it is usually very short, of the order of milliseconds.

The tests culminated, after various refinements of experimental procedure, in a measured heat flux of just under 11,000,000 Btu/hr-ft^2, a value at least five times larger than any previously reported in the literature for comparable conditions of geometry, pressure, linear velocity, and degree of subcooling. Further considerable increases of heat flux are believed possible for reasons outlined in the closure.

II. Motivations for Study

Recent experiences with source-vortex flow of fluids (air and water) in forced-convection heating without boiling (to be published) led the authors to believe that this type of flow field might be especially advantageous in a boiling system. With source-vortex flow, steam bubbles formed at the inner surface of the heated tube wall are surrounded by a rapidly rotating body of liquid of much higher density, and the gravitational acceleration arising from the vortex motion would be expected to cause a much stronger inward radial displacement of bubbles away from the inner wall toward the tube center than in the case of linear liquid motion. Were this effect sufficiently strong, it would appear that the inner surface would be freer of bubbles at a given heat flux than in standard straight-through flow, thereby permitting the attainment of a greater peak flux before burnout. These considerations led the authors to carry out the preliminary study described herein. The large heat fluxes obtained empirically demonstrate the qualitative validity of these early concepts.

III. System Description

A schematic representation of the experimental system is given as Figure 1. The water, from a building supply line, was circulated by an Aurora Apco D4 turbine-type pump through a standard rotameter, and thence to the vortex generator, vortex tube, and mixing chamber (disc and dough-nut internal design). Discharge was through Tygon plastic tubing to a floor drain (free atmospheric discharge). The test-section inlet, shown as Figure 2, consisted of a spiral-ramp fluid-introduction zone (taken from a Model B Hilsch tube sold by Lucas Brenning, San Carlos, California) followed by a convergent-cone section which accelerates the rotation rate of the water through reduction of radius. This same vortex generator, with orifice diameter of 0.117 in., was used with all of the test-section tubes. The available flow rates, in conjunction with the design of the vortex generator, permitted the attainment of several thousand gravities (calculated) at the test-section inlet. The resistance-heated, thin-walled tubes (of copper or Inconel) were cooled by the vortex water flow within them.

Test-section outer wall temperatures as well as water inlet and exit temperatures were obtained with calibrated chromel-alumel thermocouples connected to a Honeywell strip-chart recording potentiometer of $\pm$ 1% accuracy. Periodic verification of the terminal temperatures of the water was made with mercury-in-glass thermometers. The voltage across the copper electrodes brazed at tube inlet and exit was measured with a precision vacuum-tube voltmeter.

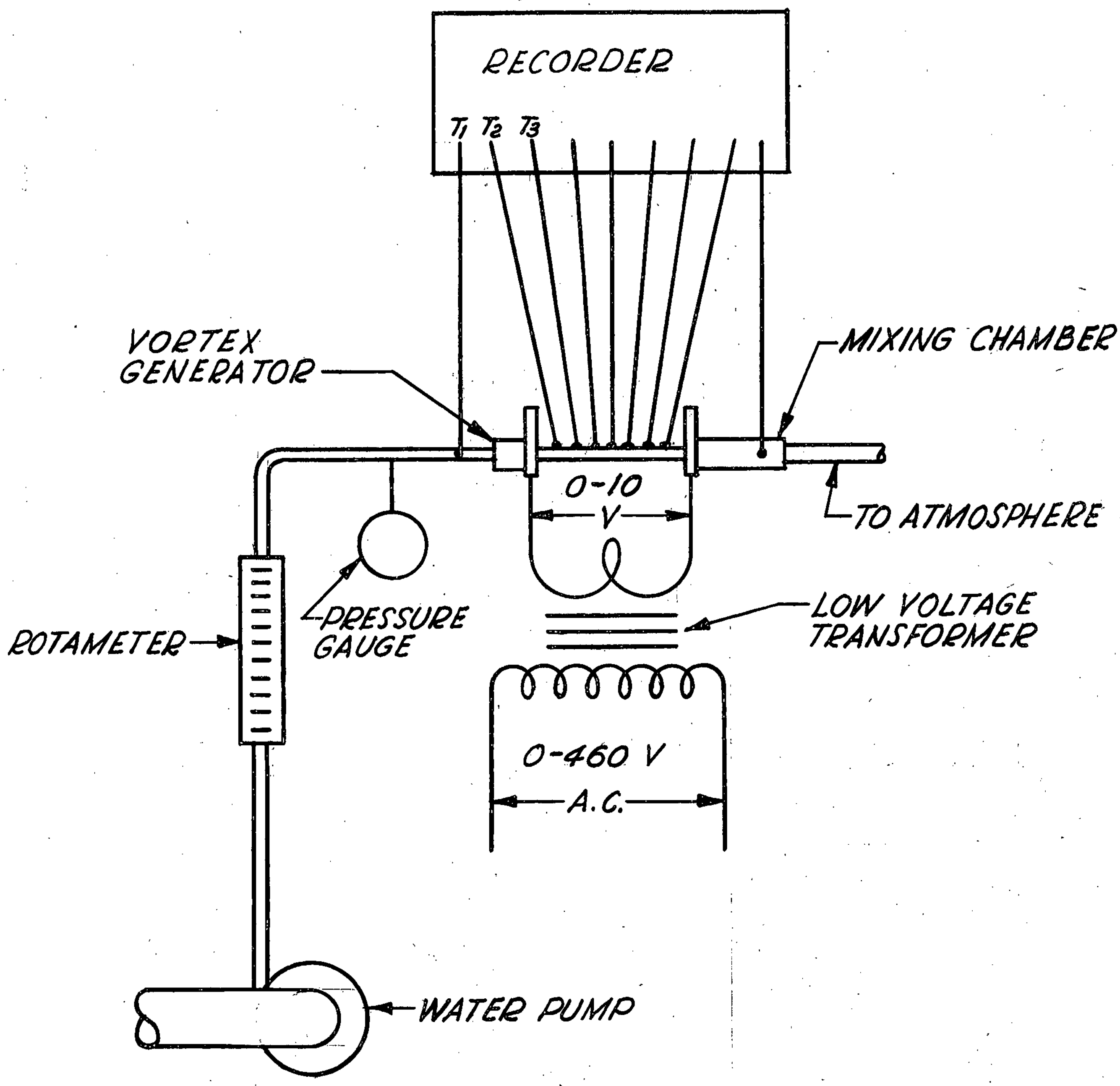

FIG. 1

THE EXPERIMENTAL SYSTEM

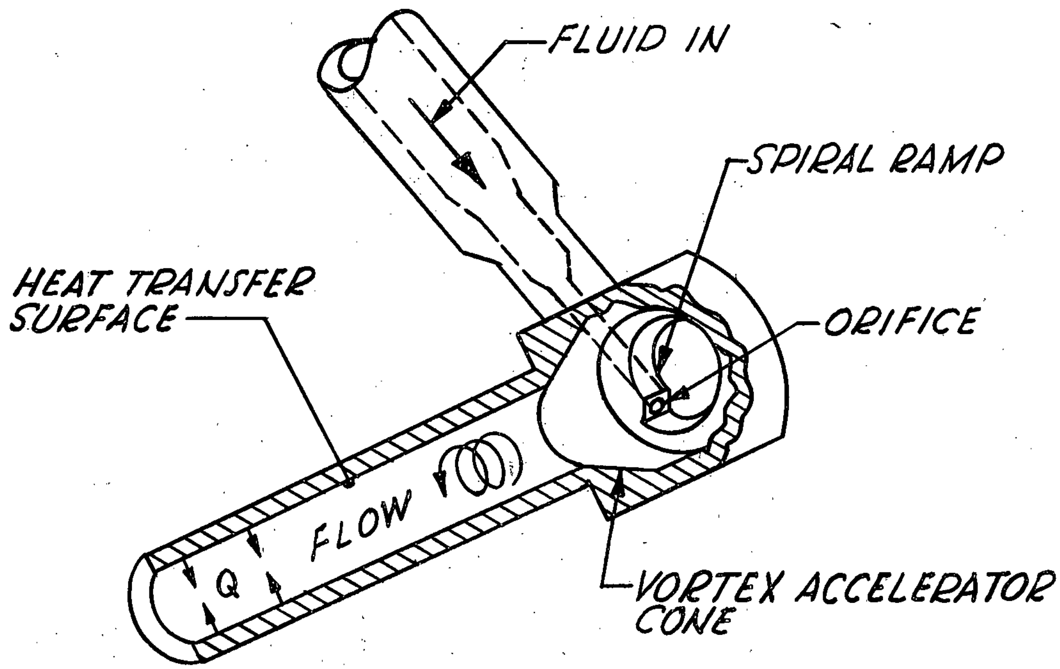

FIG. 2

THE VORTEX GENERATOR

IV. Derivation and Presentation of Results

The results of this investigation are summarized in Table 1 for the four experiments believed to be of highest accuracy (estimated over-all maximum error in burnout heat flux of $\pm$ 10%). Three other runs were made earlier, two with brass tubes and one with an Inconel tube, which yielded peak fluxes from 1.5×10^6 to about 6×10^6 Btu/hr-ft^2, but these tests were intended only for purposes of familiarization with general system behavior and for delineation of a desirable experimental procedure. The water for these early runs was low-pressure building-supply water, with no pump in the system.

The tubes used for the first three tests were all similar, as seen in Table 1; the tube for Run No. 3, however, was tapered, possessing a thinner wall at the inlet than at the exit. The use of this geometry was motivated by an attempt to more efficiently utilize the greater circulation in the freshly formed vortex at the test-section inlet by generating more power in this zone, and reducing the power generation with length to at least qualitatively match the decreasing vorticity of the water flow. This sort of variation of heat generation rate with length is similar to that obtainable by selective placement of fuel in a nuclear reactor. It might be noted here that in all tests about four tube diameters separated the inlet orifice and the initial heated portion of the test section, so that some vortex decay had taken place before heating began.

The axial velocity recorded in Table 1 is that of the water corresponding to the exit temperature (maximum linear velocity). The degree of

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subcooling listed is the minimum for the system (at exit conditions) and is equal to the saturation temperature of water at the exit static pressure (always just above atmospheric) minus the actual exit water temperature. The inlet static pressure was measured at the small orifice of the vortex generator at the tube inlet and approximates average static pressure at test-section inlet, since calculations indicate that most of the velocity head at the orifice location is dissipated across the orifice and in the inlet chamber. The exit quality, x, was calculated from:

$$x = \frac{q_{latent}/L_v}{W} \times 100 \tag{1}$$

where

$$q_{latent} = q_{total} - q_{sensible}$$

and

$$q_{sensible} = WC_p(100 - t_1) \times 1.8$$

The quantity designated $(P_{FF}/P_{HA})\%$ represents the ratio of calculated power required to force the water through the test section to the rate of heat absorption by the water, expressed in the same power units. This was calculated from:

$$P_{FF} = \frac{W\Delta P}{13,750\ \rho_L}\ \text{hp} \tag{2}$$

and

$$P_{HA} = \frac{(q_{total})_b}{2545}\ \text{hp} \tag{3}$$

The heat flux at burnout, $(q/A_i)_b$, was determined in the following fashion: For tests without net steam generation, q was obtained from

$q = WC_p dt$ and checked by voltage readings. For runs with net steam generation, a run was made at the highest power possible without net generation and the heat absorption at this lower reference level calculated from $q_1 = WC_p dt$. Following this, the exit mixing chamber was removed to prevent the possibility of choking when steam was formed, and the power level increased slowly to the burnout condition. For constant resistance, the power dissipation is proportional to the square of the impressed voltage; accordingly, the square of the ratio of the voltages for the burnout point and for the lower reference point was used as a multiplier of q_1 to obtain q_b:

$$q_b = q_1 \left(\frac{E_b}{E_1}\right)^2 \tag{4}$$

Calculations showed that the small increase of tube-wall temperature caused by the power level increase from q_1 to q_b was negligible in its influence on both heat loss and on test section electrical conductivity. The total heat loss from the heated test section to the ambient atmosphere and through the terminal electrodes was estimated to be approximately 1/2% of the generated heat.

The flow in all tests exhibited high stability right to the burnout point, and the system could apparently be operated indefinitely a few per cent below the burnout condition.

V. Discussion

The heat fluxes reported in this paper are extremely large for the conditions of operation; in fact, at least five times greater than any values previously reported for comparable system conditions. The largest experimental values of burnout heat flux that have been reported in the literature, to the knowledge of the authors, for flow through straight, round tubes, are those of Rohsenow and Clark[1] (3.1×10^6 Btu/hr-ft^2 for water at 2000 psia, 30 ft/sec, and 240°F average subcooling), and of Buchberg et al.[2] (3.8×10^6 Btu/hr-ft^2 for water at 507°F exit temperature, 44 ft/sec, 236°F subcooling, and L/D = 110). This latter value corresponds to about 5.3×10^6 Btu/hr-ft^2 for L/D = 7, according to the approximate rule that halving L/D increases $(q/A_i)_b$ by ~10%.[3] The highest value for any geometry or experimental conditions is apparently that of Gunther,[4] who obtained 11.4×10^6 Btu/hr-ft^2 for water at 114 psia exit pressure, 40 ft/sec, and 256°F subcooling at the exit. Gunther obtained his values with a very thin (0.004 in.) electrically-heated metal strip in a rectangular channel of $L/D_e = 6$; the strip was cooled from both sides. Since Gunther's values are consistently among the highest reported in the literature, a comparison of our Run No. 4 and his Test No. 5, for which conditions are almost identical, is revealing:

	Gambill and Greene Run No. 4	Gunther[4] Test No. 5
Water velocity, ft/sec	13.6	12.3
P_{exit}, psia	15.0	14.4
Subcooling at exit, °F	72	71
L/D_e	4.8	6*
$(q/A)_b$, Btu/hr-ft^2	10.98×10^6	2.0×10^6

*Based on hydraulic diameter of 1/2 in. given by Gunther.[4]

Were all of Gunther's experimental conditions identical with those of our Run No. 4, it is estimated that his $(q/A)_b$ would be 2-1/4 x 10^6 Btu/hr-ft^2, which gives a 390% higher peak heat flux for the vortex case. When comparisons are made with other literature data, the advantage becomes as great as 500% for the vortex case.

Note that the burnout heat flux for Run No. 4 corresponds to a volumetric heat release rate within the tube wall of about 50 kw/cc.

The ascending magnitude of burnout heat flux with run number (Table 1) is primarily attributable to improvements in experimental technique as more was learned of the nature of the process of burnout in this new flow system. Run No. 2, e.g., gives a $(q/A)_b$ greater than that in Run No. 1 because a transition was made from the use of step transformer taps for voltage control to Variacs, by which power level could be increased gradually and smoothly rather than by finite increments only. The result is still higher in Run No. 3 because of slower Variac adjustment as a result of having learned the approximate burnout point from Run No. 2. Another factor tending to further increase burnout heat flux in Run No. 3 was the use of the tapered-wall test section, which provided an area for electrical current flow at inlet about half that at the exit. The calculated heat flux at the inlet for the burnout condition is 14.8 x 10^6 Btu/hr-ft^2 for this run. This value indicates the potential usefulness of utilizing the greater vorticity at the inlet by generating more heat in this region. The further increase in $(q/A)_b$ of Run No. 4 is primarily associated with the reduction of L/D from $\sim$10 to $\sim$5.

The large heat fluxes obtained for the case of net steam generation, Runs No. 2 and 3 of Table 1, indicate another possible advantage of vortex boiling - that any gas released or vapor generated will be transported to the center of the tube where it will do the least harm. This action occurs to some extent in linear flow because of Bernoulli forces, but many investigators have nevertheless concluded that greater amounts of dissolved gas decrease peak flux in subcooled boiling with linear flow.

VI. Closure

It has been shown that burnout heat fluxes for boiling heat transfer can be increased 400 - 500% by the use of a suitable source-vortex flow field. This is the first time such a study has appeared in the literature. The burnout values presented here can obviously be increased to considerably higher levels by the use of greater velocities, pressures, and degrees of subcooling, all of which decrease the maximum bubble size and increase the burnout heat flux. None of the conditions of the present study (except L/D) even approach the optimum for maximum $(q/A)_b$, which further emphasizes the immense heat fluxes possibly attainable.

An effort will be made to extend this study of heat transfer effects within a vortex flow field to other ranges of experimental variables. It is felt that a $(q/A)_b$ value of 40,000,000 Btu/hr-ft^2 is not unattainable. Also, further refinements of experimental technique may be expected to reduce the maximum error for $(q/A)_b$ to below $\pm$ 5%.

Further results will be published as they are produced.

VII. References

1. Rohsenow and Clark, Heat Transfer and Fluid Mechanics Institute Reprints, 193-207, 1951.

2. Buchberg et al., Heat Transfer and Fluid Mechanics Institute Reprints, 177-191, 1951.

3. Personal communication with P. Griffith, MIT, September 1957.

4. Gunther, Trans. A.S.M.E., 73, 115-123 (1951).

VIII. Notation

A_i	internal surface area of test section, ft^2
C_p	specific heat of water at constant pressure, Btu/lb-^oF
D_i	internal diameter of test-section tube, in.
dt	temperature rise of water across test section, ^oF
E	potential difference across test section, volts
L	test-section heated length, in.
L_v	latent heat of vaporization at normal boiling point, Btu/lb
P_{FF}	power required for fluid flow across test section, hp
P_{HA}	power corresponding to heat absorption rate q, hp
ΔP	test-section pressure drop, psi
q	rate of heat transfer, Btu/hr
t_1	inlet water temperature, ^oC
W	water weight flow rate, lb/hr
x	steam quality at test-section exit, % by weight
ρ_L	liquid density evaluated at exit temperature, lb/ft^3

Subscripts

1	at lower reference level of heat absorption
b	burnout condition
i	internal