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AuthorizedABSTRACT

A study has been made of burnout heat fluxes associated with forced-convection source-vortex flow of subcooled water. The maximum heat flux experimentally attained was 35.1×10^6 Btu/hr·ft² (11.1 kw/cm²). Vortex flow and straight flow heat-transfer rates have been compared.

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A PRELIMINARY STUDY OF BOILING BURNOUT HEAT FLUXES
FOR WATER IN VORTEX FLOW

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N. D. Greene

SUMMARY

A preliminary experimental study has been made of burnout heat fluxes for the new case of subcooled water in forced-convection source-vortex flow through electrically heated horizontal tubes of small diameter. These data, which cover a moderate range of velocity, L/D ratio, degree of subcooling, and power input, have been correlated empirically. The maximum burnout heat flux attained was 35.1×10^6 Btu/hr.ft² (11.1 kw/cm²).

A number of tests for low-pressure water in straight-through flow through similar test sections were also made for comparison purposes. Comparisons of burnout heat fluxes were made on the bases of equal superficial axial velocity and of equal flow power dissipation. These comparisons indicate a small superiority for the vortex case with low pumping power and a large advantage at high pumping power.

Local and over-all heat-transfer coefficients and their variation with heat flux were determined for one of the vortex tests.

NOMENCLATURE

a^*	ratio of cross-sectional flow area of orifice or slots of vortex generator to cross-sectional flow area of test section
A_1	inside heat-transfer surface, sq ft
C_p	specific heat at constant pressure, Btu/lb·°F
D_1, D_2	inside and outside diameters of test section, in.
E	modulus of elasticity in tension, lb/in. ² ; also voltage across test section
g_c	conversion factor in Newton's second law, ft·lb mass/sec ² ·lb force
h	heat-transfer coefficient, Btu/hr·ft ² ·°F
k	thermal conductivity, Btu/hr·ft ² ·°F/ft
L_h	heated test-section length, in.
L_v	latent heat of vaporization, Btu/lb
P	static pressure, lb/in. ²
P_f	flow power dissipated across test section, hp
P_q	heat-transfer rate at burnout point, hp
ΔP	total pressure drop across test section, lb/in. ²
q	heat-transfer rate, Btu/hr
$(q/A_1)_b$	burnout heat flux, Btu/hr·ft ²
R_1, R_2	inside and outside wall radii, ft
t_b	bulk water temperature, °F
Δt_{sub}	degree of subcooling, °F = ($t_{sat} - t_b$)
Δt_w	temperature drop across wall, °F
V	axial velocity for straight flow, ft/sec
V_{ax}	superficial axial velocity for vortex flow, ft/sec
V_t	tangential velocity, ft/sec
V_h	volume of heated test-section metal, ft ³

NOMENCLATURE (Continued)

W	weight rate of flow, lb/hr
x	steam quality at test-section exit, weight per cent
α	linear coefficient of thermal expansion, $^{\circ}\text{F}^{-1}$
σ_{tt_1}	tangential thermal stress at inner wall surface, lb/in. ²
Γ	circulation constant for a free vortex
ρ	fluid density, lb/ft ³
ν	Poisson's ratio
μ	fluid viscosity

INTRODUCTION

Modern high-performance devices such as nuclear reactors and rocket engines can be operated with very large power densities and heat fluxes. These are essentially constant heat-input systems, with the heat flux imposed independently of coolant thermal resistance. If cooling is inadequate, a rapid fusion failure or "burnout" of the heat-transfer surface ensues. This situation has prompted a large number of investigators over the past decade to study the boiling burn-out process, and especially to attempt correlation of such data for prediction purposes.

This study evolved from an earlier one, presently incomplete, of heat transfer to air and nonboiling liquids in vortex flow. The change of emphasis was motivated by the concept that the large radial contrast of fluid density associated with boiling systems might combine with the centrifugal acceleration arising from a rotating flow so as to establish a strong inward radial transport of steam bubbles. In this event, the inner surface would be freer of bubbles (lower fractional coverage) at a given heat flux. The maximum or burnout flux, which is believed to occur at some critical boundary-layer quality, would accordingly be larger. It was therefore decided to conduct, as simply as possible, a preliminary study of the gross variables of such a system.

Prior work in this area has apparently been quite limited. Many studies of fluid flow without heat transfer in vortex tubes of the Ranque-Hilsch type have been published; see, e.g., a recent paper by Hartnett and Eckert (1).¹ A number of other studies (2, e.g.) have dealt with low-flux heat transfer across annular air gaps with one or both boundary cylinders rotating, in simulation of

¹Underlined numbers in parentheses refer to the Bibliography at the end of the paper.

rotating electrical machinery. An analysis applicable to rotating atmospheric air masses has been published by Kuo (3). Jakob (4) and Eckert and Jackson (5) have reported results for natural-convection internal liquid cooling of rotating turbine blades. It was learned by the authors toward the end of their experimental program that two other preliminary investigations of boiling with whirling flow have been conducted (6, 7). In these unpublished studies, experimental conditions were milder than those reported here, and no significant increase of burnout flux ($< 20\%$ at equal pumping power) was observed. The earliest results of the present study were reported in a memorandum (8) and in a brief technical note (9). All results obtained through March 28, 1958 are included here.

Four specific potential applications of vortex-boiling are presently being considered: cooling of particle accelerator targets, cooling of liquid-propellant rocket motor nozzle throats, application to high power-density nuclear reactors, and possible use in mild atmospheric re-entry situations.

APPARATUS AND PROCEDURE

A schematic diagram of the experimental system is shown in Fig. 1. Terminal water temperatures were measured with calibrated chromel-alumel thermocouples connected to a Honeywell strip-chart recording potentiometer with ice-bath cold junction. These water temperatures were periodically verified several times in each test and as close to the burnout point as possible with calibrated mercury-in-glass thermometers. Heat input from the pumps was important in some tests, causing a water temperature rise as large as 6°C . Inlet water temperature was always measured after a period of steady-state pump operation without heating. In vortex test No. 5, the exit mixing-chamber was maintained at 100 psig by a needle valve; in all other tests, heated water was discharged through Tygon tubing to the

atmosphere.

Test-section outer wall temperatures were measured in some tests with 36-gauge calibrated chromel-alumel thermocouples spot-welded to the outer surface and wound circumferentially for a short distance; these were printed on a multipoint recorder. The test section was generally left bare when burnout points only were being determined but was insulated with asbestos tape when wall temperatures were measured.

Voltage impressed across the copper electrodes was measured with a precision vacuum-tube voltmeter. Calculations indicated that heat loss by convection and radiation from a bare tube to the atmosphere and by axial conduction to the electrodes was generally of the order of 1/2% of total heat generation. Consequently, all burnout heat fluxes were calculated from water heat absorption rate and cold test-section dimensions.

The rotameters were directly calibrated over the full range of use with a large calibrated weigh tank and stopwatch. All static pressures were measured with new Bourdon-type gauges. As designated in Fig. 1, pressure P_1 was measured at the discharge of the progressive-cavity (Moyno) booster pump. Pressure P_2 was measured at the test-section inlet in all cases. With the spiral-ramp vortex generator (Fig. 2), the pressure tap was located halfway along the length of the tapered orifice (0.115-in. inlet to 0.093-in. exit), where the orifice diameter was 0.104 in. Wall pressure P_3 was measured for three vortex tests only at the portion of the test section brazed into the exit electrode.

Entrance regions used are depicted in Fig. 2. The spiral-ramp vortex generator, made of chrome-plated brass, was taken from a commercial Hilsch tube. A convergent cone transition section served to join generator and test sections and also to accelerate water rotation rate by conservation of angular momentum. About 4 tube diameters separated inlet orifice and initial heated portion of the

test section. Only three tests (12 to 14 of Table 1) have been made with the alternate tangential-slot vortex generator. For the spiral-ramp generator, a^* (see Nomenclature) varied from 12.0% to 17.3%; for the tangential-slot generator, a^* varied from 13.1% to 26.2%.

The first comparative straight-flow tests were made with straight, unaltered test-section inlets, but entrance region instability and apparent boundary-layer separation necessitated use of the convergent-cone entrance regions, which seemed to perform stably.

Test sections with tapered walls were used in some tests in an attempt to better utilize the greater circulation of the freshly formed vortex by generating more heat at the inlet and by reducing heat generation with length to qualitatively match the decreasing flow vorticity. Such a variation of heat generation rate with length is easily obtainable in a nuclear reactor by selective fuel placement. The effect of wall taper on burnout flux appears to be negligible, however, since the present results correlate independently of wall taper.

All test sections were horizontal and heating was with 60-cycle a-c throughout. It should be noted that the water used was not distilled, degassed, or deionized and contained a considerable amount of air, minerals, and, at times, even rust. Burnout was attained in all cases by slowly increasing heat-generation at constant flow rate. Test sections were limited to fairly thin wall, short tubes in an effort to balance the opposing requirements of very high heat flux, low thermal stress, and use of limited available voltage.

TEST RESULTS AND THEIR CALCULATION

Results are summarized in Tables 1 and 2. In each table, $(P_f/P_q)\%$ represents pumping power across the test section as a percentage of heat extraction rate at the burnout point, as calculated from:

$$P_f = \frac{W\Delta P}{13,750 \rho} \quad (1)$$

$$P_q = \frac{q_b}{2,545} \quad (2)$$

and $(P_f/P_q)\% = \frac{100 P_f}{P_q} \quad (3)$

In all cases, the total pressure at the test-section inlet was used for ΔP . This assumes total head loss with no pressure recovery at exit and represents the most unfavorable condition. High-efficiency diffusers for velocity-pressure transformation of rotating flows are easily designed, however (10). For vortex tests with the spiral-ramp generator, the velocity head at the orifice pressure tap was added to the gauge reading; but with the other entrance regions, approach velocity in the outer concentric pipe was small enough not to require velocity head addition.

Superficial axial velocities in Table 1 correspond to the density at the exit temperature. The "bulk subcooling at exit" of Table 1 was taken as saturation temperature at exit pressure, minus bulk exit water temperature. This more nearly represents the local exit degree of subcooling at the air core -- water interface which existed in all vortex tests. Estimation of the actual local degree of subcooling at the exit wall is given in the Appendix.

Net steam was generated only in vortex tests 1 and 2; for these tests, exit quality was calculated from:

$$x = \frac{q_{\text{latent}}/L_v}{W} \times 100 \quad (4)$$

where $q_{\text{latent}} = q_{\text{total}} - q_{\text{sensible}}$

and $q_{\text{sensible}} = W C_p dt$

The burnout flux and q_{total} of equation (4) were determined by first running just below the point of net steam generation and calculating the heat absorption at this lower reference level from $q_R = W C_p dt$. For constant test-section resistance, power dissipation is proportional to the square of impressed voltage; accordingly:

$$(q_{\text{total}})_b = q_R \left(\frac{E_b}{E_R} \right)^2 \quad (5)$$

Calculations indicated that the small increase of tube-wall temperature caused by the power level increase from q_R to q_b was negligible in its influence on both heat loss and test-section electrical conductivity.

A review of the calibration errors indicates that the maximum error in the reported burnout heat fluxes is at most $\pm 5\%$, with a probable error of about $\pm 2\%$. It should be noted that burnout fluxes reported for the tapered-wall tubes were computed with total heat absorption at burnout and with total internal area; the heat flux at the exit where burnout occurs is therefore somewhat lower and may be estimated from a ratio of mean to exit electrical cross-sectional areas. The burnout fluxes reported for the untapered tubes are very nearly the values at the exit.

DISCUSSION

The goals originally defined for the experimental program were:

- (a) To determine the absolute magnitude of burnout heat flux attainable with a reasonable pumping system. This is of interest in applications to particle-accelerator target cooling, where potential heat fluxes would be very high, target tubes small, and pumping power relatively unimportant.
- (b) To devise an approximate correlation by which rough estimates of burnout flux could be made.
- (c) To compare vortex and straight-flow boiling on the basis of heat transfer at burnout with constant pumping power. This would be of importance in nuclear power reactor applications.

The discussion will deal with each of these points in turn, as well as with certain secondary information which was derived.

A. Magnitude

The burnout flux of 35.1×10^6 Btu/hr·ft² attained in vortex test No. 14 is, to the knowledge of the authors, the largest ever reported for any coolant in any geometry. A compilation of the broad range of heat fluxes encountered in modern industrial and scientific practice is given in Table 3 for comparison purposes. In considering absolute magnitudes of heat flux, one must distinguish between generated or input fluxes and those actually transferred to a coolant, and between steady-state and transient heat fluxes. In many shock tubes, e.g., enormous heat fluxes are generated for very brief periods but are not transferred to a coolant.

Conditions in vortex test No. 14 were also extreme in other respects.

The entire exterior surface of the test section was a bright red color for a

considerable period before burnout, and the wall temperature drop, approximated from:

$$\Delta t_W = \frac{(q/V_h)}{2k} \left[R_2^2 \ln \frac{R_2}{R_1} - \frac{R_2^2 - R_1^2}{2} \right] \quad (6)$$

was 1330°F, which corresponds to a radial thermal gradient of 133,000°F/in.

The power density or volumetric heat release rate, (q/V_h) in equation (6), was 419 kw/cc at burnout. The controlling thermal stress for the geometry under consideration is the tangential or circumferential stress at the inner wall; this was calculated from the following expression (11):

$$\sigma_{tt1} = \frac{E \alpha (q/V_h) R_2^2}{4k (1 - \nu) (R_2^2 - R_1^2)} \left[\frac{R_2^4}{R_2^2 - R_1^2} \ln \left(\frac{R_2}{R_1} \right)^2 - R_2^2 - \frac{R_2^2 - R_1^2}{2} \right] \quad (7)$$

to be 2.06×10^6 psi (elastic thermal stress), which shows that the Inconel was well beyond the elastic limit and in the region of plastic behavior. A photomicrograph of the tube wall for vortex test No. 14 indicated nothing abnormal, however.

An estimate was also made for vortex test No. 14 of the amplitude of the sinusoidally varying temperature at each surface of the test section. A complicated relationship for accomplishing this has been published in a University of California report (12). The equation, too lengthy to conveniently reproduce here, was derived for a flat plate insulated on one side and for uniform and constant thermal and electrical properties. Properties of the tube were taken in the present estimate at average wall temperature, and the flat plate condition is approximately fulfilled since the mean radius to wall thickness ratio was 12.75. The calculation indicates that the amplitude of outer wall temperature (at a frequency of 120 cycles/sec) was 180°F, and of inner wall temperature, 51°F. This causes thermal cycling of tube metal already under severe

conditions and would also be expected to cause earlier burnout than d-c heating because of the ripple in heat flux (roughly estimated to be at least $\pm 38\%$ about the mean for this test).

B. Correlation

The vortex burnout fluxes for tests with the spiral-ramp generator were empirically correlated on the basis of tests 1 to 6 by:

$$(q/A_1)_b = (15,750 V_{ax}^2 + 8.07 \times 10^6) \left[1.18 - 0.03 (L_h/D_1) \right] \quad (8)$$

which is shown graphically as Fig. 3. Burnout flux was apparently independent of degree of bulk exit subcooling over the range 0° to 150°F . Later vortex tests 7 to 10 followed equation (8) closely, but the very short L/D of test 11 caused the experimental burnout flux to exceed the predicted value by 18%. The results for tests 12 to 14 (tangential slot generator) differ from equation (8) by about 24%. For vortex tests 1 to 10, average and maximum deviations for equation (8) are 4.0% and 9.7%. For all tests, 1 to 14, average and maximum deviations are 8.6% and 24.2%.

The straight-flow burnout fluxes of Table 2 were calculated by plotting the ratio of burnout flux predicted by Gunther's equation (13) to the experimental burnout flux versus the (L_h/D_1) ratio. The points fell closely about a straight line, resulting in the following equation:

$$(q/A_1)_b = \frac{7000 V^{1/2} (\Delta t_{sub})_{exit}}{0.76 + 0.02735 (L_h/D_1)} \quad (9)$$

At an L/D of 6.0 (the length to equivalent diameter ratio used by Gunther), the denominator is 0.924, indicating that these burnout fluxes are about 8% larger than those reported by Gunther for the same conditions. Equation (9) correlates the limited data of Table 2 within average and maximum deviations of 2.6% and 5.2%.

C. Comparison with Straight Flow

An early comparison is shown in Fig. 4, where burnout fluxes for straight flow as predicted by the correlations of Gunther (13) and of McGill-Sibbitt (14) are plotted versus velocity with degree of subcooling as a parameter. The vortex curve was calculated from equation (8). The comparison is somewhat artificial to the extent that for any of the linear flow lines, exit pressure would have to be changed when velocity changes in order to maintain degree of exit subcooling constant as noted. Values calculated from the McGill-Sibbitt equation, which is based on data from a tube of $L/D = 21$, were approximately corrected to an L/D of 6 by the rule-of-thumb that halving L/D increases $(q/A_1)_b$ by $\sim 15\%$.

A more significant comparison is given as Fig. 5 where heat transfer versus flow power is plotted for straight and vortex flows in a given tube at the burnout point. See Appendix I for details concerning the construction of Fig. 5. Since the comparison is based on a particular tube, the ordinate is directly proportional to burnout heat flux. It is apparent that the small superiority of vortex flow at low pumping power (also noted in references 6 and 7) becomes very considerable at large values of flow power dissipation. While Fig. 5 is, of necessity, not general, the results are considered to be typical.

Still another comparison based on local conditions at the exit tube wall is given in Appendix II. This comparison, though limited in range, indicates that vortex boiling burnout heat fluxes are relatively independent of true local degree of subcooling at the wall as well as the "bulk degree of subcooling" previously defined.

D. Variation of Heat-Transfer Coefficient

Local and over-all heat-transfer coefficients and their variation with heat flux were calculated for vortex test No. 9 only. Thermocouples were

located on the test-section exterior surface at distances of 0.25 in., 1.17 in., and 2.08 in. from $L_h = 0$. Equation (6), a simplified form of the Kreith-Summerfield relation (15) for wall-temperature drop, was used in conjunction with a graphical procedure (see Appendix III for details) to calculate the coefficients. At the vertical line of Fig. 6, boiling just begins at the exit, using calculated inner wall temperatures and measured wall static pressure as a criterion. Boiling then occurs over an increasing fractional tube length at higher heat fluxes. The increase of heat-transfer coefficient with heat flux in the totally nonboiling region of Fig. 6 is probably caused by change of physical properties with temperature. For nonboiling, turbulent convection in linear flow at a given weight flow rate,

$$h \propto \left[c_p^{1/3} k^{2/3} \rho^{0.8} / \mu^{0.467} \right] \quad (10)$$

This factor, evaluated at the mean film temperature for each of the three points at the left of Fig. 6, undergoes a change of 36%, or about twice the variation of the calculated vortex coefficients. Fig. 6 also shows the sharp increase of film temperature drop which begins about 20% below burnout.

Fig. 7 is a cross plot of the originally calculated local coefficients versus local heat flux. Coefficients at $L/D = 0$ are theoretically infinite, but values were calculated at this position by extrapolation of outer wall temperatures to $L_h = 0$ solely to show the general trend of h with (q/A) . The shapes of Figs. 6 and 7 are due, in part at least, to the large range of variables encountered over the length of the test section. Water is locally boiling over part of the length at the higher fluxes, but not at all at the lower fluxes; in addition, input heat flux decreases with length because of wall taper, while fluid vorticity is simultaneously decreasing with length. Thus, at a constant over-all flux, h first decreases with length and then tends to increase as

boiling begins, though the local heat flux at exit is lower than upstream. It is felt that h follows the general trend shown in Figs. 6 and 7 even though the exact shape may be somewhat different.

E. Miscellaneous Observations

In all tests, the flow exhibited excellent stability all the way to the burnout point, and the system could be operated for considerable periods a very few per cent below burnout.

No whistling or singing noises were noted as burnout was approached. This is a common phenomenon in straight-flow local boiling and may be absent in the vortex case because of the presence of a free liquid surface.

Pressure gauge P_3 of Fig. 1 (at exit wall of test section) fluctuated moderately under adiabatic conditions but immediately stabilized when the switch was thrown and heat was generated in the test section. With increase of heat flux, there was a small increase of exit wall pressure.

Large amounts of air were seen dispersed in the exit water stream as it flowed through a transparent hose. This would be expected, of course, from centrifuging and heating untreated water.

F. Possible Improvements

The burnout heat fluxes attained in this study could probably be increased by:

- (1) Using d-c heating to avoid heat flux ripple and thermal cycling of the test section.
- (2) Using distilled water. This change might not have much affect since with vortex flow, any gas or vapor is transported to the center of the tube where it will do the least harm.
- (3) Substituting another type of pump for the Moyno to avoid its slight pulsation and vibration.

- (4) Using one "floating" electrode to allow for test section thermal expansion.
- (5) Using other metals which generate lower thermal stresses and have higher strength at elevated temperatures. The Udimet "super-alloys", Nb, and Mo are being considered.

FUTURE WORK

Further studies are planned for:

- (1) Attaining heat fluxes above 35×10^6 Btu/hr·ft² in order to better define the limit on heat-transfer rates through metal walls as imposed by thermal stresses and other metallurgical limitations.
- (2) Studying variation of burnout flux with a^* at a given pumping power.
- (3) Extending tests into higher ranges of L/D and of exit quality.
- (4) Studying other geometries, such as curved tubes, cones, and tubes with tangential slots or holes distributed over their entire lengths.

CONCLUSIONS

1. Very large burnout heat fluxes can be simply attained by using vortex coolant flow.
2. At equal pumping power, vortex flow gives larger burnout heat fluxes than straight flow. The advantage is small at low pumping power but increases rapidly and becomes large at high pumping power.

3. Various comparisons indicate that the superiority of vortex flow cannot be wholly attributed to an increase of local velocity and degree of subcooling arising from the rotational motion. It is postulated that the advantage is connected with a mechanism whereby hot liquid or vapor at the wall is transported radially inward by the action of the centrifugal acceleration.

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APPENDIX

I. Construction of Fig. 5

For curve 1, equation (9) was used to calculate P_q , and P_f was calculated from equation (1), using the sum of velocity head and friction loss for ΔP . Friction loss was calculated from standard relations for linear flow without heat transfer. It is recognized that the ratio of pressure drop with heat transfer to the adiabatic pressure drop initially decreases with increase of heat flux, at equal liquid bulk temperature and flow rate. Upon inception of nucleate boiling, this ratio increases, and the maximum value of burnout may be above or below unity, but is usually not greatly different.^a In the absence of a specific correlation, the adiabatic friction loss was used for this calculation at the burnout point.

For curve 2, P_q was calculated from the McGill-Sibbitt equation (14):

$$(q/A)_b = 246,000 v^{1/2} (\Delta t_{sub})^{0.28} \quad (1)$$

corrected from $L/D = 21$ to $L/D = 6.5$ by the rule that halving L/D increases $(q/A)_b$ by $\sim 15\%$. P_f was calculated as for curve 1.

For curves 1 and 2, a trial-and-error solution was necessary, since degree of exit subcooling, on which burnout flux and P_q depend, was not known in advance and had to be assumed.

For curve 3, P_q was calculated from equation (8), and experimental pressure drop data for the specific geometry was used for P_f . The vortex-flow over-all ΔP varied as the square of flow rate over a broad range and was slightly extrapolated on this basis for some of the points of curve 3.

^aBartz, D. R., Jet Propulsion, 28, No. 1, 46-53, 1958.

Curve 4 was plotted directly from vortex tests 12 and 14. P_f was taken as the observed value, since it was noted on several occasions that ΔP was essentially independent of test-section length, probably because of the presence of the free liquid surface adjacent to the air core. The experimental P_q value was corrected to the conditions of Fig. 5 by a surface-area ratio and the L/D function of equation (8).

II. Comparison of Vortex and Straight-flow Burnout on Basis of True Local Conditions at Exit Wall

When the superficial axial velocities and "bulk degree of subcooling" of Table 1 are used in burnout prediction equations for straight flow, calculated values average about 1/4 to 1/5 the burnout fluxes experimentally observed at higher velocities.

It is, however, possible to estimate the local exit velocity and subcooling for three vortex tests (7, 9, and 10) in which measurements were made of the static pressure at the exit wall and of the ratio of air core diameter to tube diameter.

<u>Vortex Test No.</u>	<u>P_a, psig</u>	<u>$(D_{\text{air core}}/D_1)$</u>
7	21	0.7
9	27	0.7
10	46	0.7

For any vortex, a radial force balance gives:

$$\frac{dP}{dR} = \frac{\rho V_t^2}{g_c R} \quad (1)$$

The tangential velocity distribution in a free vortex is:

$$V_t = \frac{\Gamma}{R} \quad (2)$$

Combination of equations (1) and (2) and integration yields:

$$(P_{R2} - P_{R1}) = \frac{\rho}{2g_c} (V_{t1}^2 - V_{t2}^2) \quad (3)$$

Let sub 1 denote the air core-water interface and sub 2 the water-tube wall interface. From the observation that $D_1/D_2 \approx 0.7$, equation (2) gives:

$$\frac{V_{t1}}{V_{t2}} \approx 1.428 \quad (4)$$

Setting $P_{R1} = 1$ atmosphere, equations (3) and (4) may be solved simultaneously for the tangential velocities at the extreme radii of the water annulus at the tube exit. Also, the true axial velocity (average) is approximately double the superficial value.

The exit vector velocity calculated from these tangential and axial components may be used along with the true exit subcooling, defined as $(t_{\text{sat, wall, exit}} - t_{b, \text{exit}})$, in equation (9) to calculate burnout flux for equivalent straight flow, as follows:

Test No. ^b	(L/D_1)	$V_{\text{exit, ft/sec}}^c$	$(\Delta t_{\text{sub}})^d$ °F	Calcd. ^e $(q/A_1)_b$	Exptl. ^f $(q/A_1)_b$	Ratio Exptl./Calcd.
7	12.00	86.1	66	3.93×10^6	13.10×10^6	3.33
9	8.46	96.6	109	7.53×10^6	15.15×10^6	2.01
10	4.73	112.4	169	14.10×10^6	18.60×10^6	1.32

^bTotal flow rate and tube diameter nearly identical for all three tests.

^cVector velocity from true average axial component and arithmetic average of V_{t1} and V_{t2} .

^dLocal value at exit wall.

^eFrom equation (9) for straight flow, units of Btu/hr·ft².

^fValue at exit, calculated from over-all measured burnout flux and ratio of mean and exit electrical cross-sectional areas, units of Btu/hr·ft².

Note especially that the experimental burnout fluxes vary much more nearly in proportion to the change of vector velocity than to the change of subcooling. The burnout flux predicted for straight flow is nearly as large as the vortex value of low L/D because the degree of subcooling is large, and dependence is on the first power of Δt_{sub} . At large L/D , Δt_{sub} decreases rapidly and so does $(q/A_1)_b$ for straight flow. The vortex burnout flux decreases more slowly, as does the exit vector velocity on which it depends.

This comparison furnishes indirect evidence in favor of the hypothesis of radial vapor transport.

III. Calculation of Heat-Transfer Coefficients for Vortex Test No. 9

To take into account variation of wall thickness with length as well as radial and axial variation of wall thermal conductivity, the following procedure was used:

- (a) The over-all measured heat flux was assumed to occur as a local value at the length corresponding to the mean cross-sectional electrical area of the tube. Heat fluxes at other lengths were then calculated from ratios of electrical area.
- (b) Power density (q/V) was obtained by dividing local heat flux by local wall thickness. (q/V) was also plotted versus length.
- (c) Water temperature as a function of length was calculated by dividing the test section into three equal lineal zones and successively applying to each zone the relation:

$$t_{b \text{ out}} = t_{b \text{ in}} + \frac{(q/A_1)_{\text{mean}} A_1}{WC_p}$$

- (d) Equation (6) was then used for a trial-and-error solution of Δt_w . A mean wall thermal conductivity was assumed for each axial position, Δt_w calculated, and k at the arithmetic mean of measured outside wall temperature and calculated inside wall temperature compared with the assumed k . This process was repeated until Δk was less than 1%. It was necessary to use a desk calculator to compute the geometrical factor of equation (6) because of its numerical sensitivity. Thermal Conductivity values of "A" nickel were taken from International Nickel Company Engineering Bulletin T-15, "Engineering Properties of Nickel."
- (e) Local heat-transfer coefficients were calculated by dividing local heat flux by local film temperature drop. Local h was plotted versus L_h and graphically integrated to obtain the overall inside film coefficient of heat transfer.

This procedure approximately accounts for all variables except radial and axial variation of electrical resistivity, which changes negligibly for "A" nickel over the range of wall temperatures encountered in vortex test No. 9. The primary purpose of the calculation was to estimate the variation of coefficient with heat flux and length rather than to precisely determine its magnitude.

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TABLE 1. VORTEX-FLOW BOILING BURNOUT TESTS

Test No. [‡]	Superficial Axial Velocity ft/sec	Bulk Sub-cooling at Exit °F	Inlet Static Pressure psig	Test Section					x, Exit Quality %	$\left(\frac{P_f}{P_q}\right)^*$ %	Exptl. $(q/A_1)_b$ Btu/hr·ft ²
				ID in.	OD in.	Heated Length in.	L_h/D_1	Material			
1	11.80	0	94	0.275	0.300	3.187	11.60	Copper	1.24	0.28	8.24×10^6
2	12.35	0	99	0.275	Inlet 0.288 Exit 0.300	2.875	10.44	Copper	1.49	0.30	9.69×10^6
3	13.61	72	100	0.250	0.300	1.20	4.80	Inconel	0.0	0.60	10.98×10^6
4	13.18	135	248	0.300	0.375	3.05	10.16	Copper	0.0	0.48	8.63×10^6
5	16.30	150	248	0.250	Inlet 0.270 Exit 0.281	2.25	9.00	Inconel	0.0	0.48	10.81×10^6
6	24.00	49	285	0.250	Inlet 0.270 Exit 0.278	1.625	6.50	Inconel	0.0	1.45	17.25×10^6
7	26.40	17	415	0.275	Inlet 0.302 Exit 0.315	3.30	12.00	"A" Nickel	0.0	1.37	15.74×10^6
8	-----Premature burnout at inlet because of excessive wall taper-----										14.27×10^6
9	26.40	51	371	0.275	Inlet 0.305 Exit 0.314	2.33	8.46	"A" Nickel	0.0	1.63	17.19×10^6
10	25.80	87	365	0.275	Inlet 0.305 Exit 0.310	1.30	4.73	"A" Nickel	0.0	2.58	20.05×10^6
11	27.50	118	400	0.275	Inlet 0.305 Exit 0.307	0.54	1.963	"A" Nickel	0.0	5.33	26.50×10^6
12	26.42	28	495	0.245	0.265	1.87	7.63	Inconel	0.0	1.25	22.55×10^6
13	-Premature burnout because of heavily oxidized tube interior and wrinkle in tube wall near exit-										22.65×10^6
14	48.90	104	540	0.245	0.265	0.90	3.67	Inconel	0.0	3.39	35.10×10^6

[‡]Spiral-ramp vortex generator used for test 1 - 11; tangential slot generator for tests 12 - 14 (two forward slots open in tests 12 and 13, and all four slots open in test 14).

*Calculated for burnout condition, using total head loss for flow work term P_f .

TABLE 2. STRAIGHT-FLOW BOILING BURNOUT TESTS

Test No.	Velocity ft/sec	Degree of Subcooling at Exit, °F	L_h , in.	(L_h/D_1)	(N_{Re}) tube, mean	$(P_f/P_q)\%^\dagger$	Exptl. $(q/A_1)_b$ Btu/hr·ft	$(q/A_1)_b$ from Eqn. (9)	Ratio (Eqn./Expt.) for $(q/A_1)_b$
1	84.3	148	3.82	12.51	160,200	0.749	8.64×10^6	8.63×10^6	0.999
2	56.7	146	2.88	9.44	111,800	0.294	7.92×10^6	7.56×10^6	0.955
3	85.7	158	1.85	6.06	149,800	1.221	10.50×10^6	11.04×10^6	1.052
4	-----Burnout at test-section inlet, improper installation-----								
5	56.7	134	6.20	20.33	121,100	0.245	5.53×10^6	5.38×10^6	0.973
6	43.1	134	5.02	16.48	93,400	0.140	5.03×10^6	5.10×10^6	1.014
7	61.5	142	4.06	13.32	128,900	0.348	6.83×10^6	6.92×10^6	1.013
8	85.5	103	2.92	9.58	161,700	1.331	6.33×10^6	6.51×10^6	1.029

Notes - For all tests: $D_1 = 0.305$ in., $D_2 = 0.375$ in., tubes of "A" nickel, exit quality of 0%, atmospheric discharge.

Static pressure in test section at inlet electrode, adiabatic conditions: 16 - 27 psia.

Temperature of inlet water: 5.20 - 8.15°C.

Temperature of exit water: 12.54 - 26.15°C.

Test-section inlet regions: see Fig. 2.

† Calculated for burnout condition, using total head loss for flow work term P_f .

TABLE 3

HEAT FLUX RANGES
(q/A in Btu/hr·ft²)

I.	40	Normal black body radiation from earth to space on clear nights.
	300	Maximum direct solar heat flux at the earth's surface.
	440	Maximum direct solar heat flux at top of earth's atmosphere.
	7,000	Skin heating for a typical long-range cruise missile.
	35,000	Skin heating for a typical high-altitude, long-range hypersonic glider and for a typical high-speed surface-air defense missile.
	500 to 50,000	Process industry heat exchangers, 500 low for subcooling in condensers to 50,000 high for some vaporizers.
	100,000	Modern steam generator with combustion heating.
	160,000	Sodium-to-water steam generators, typical design.
	300,000	Reciprocating gas compressor jacket flux, higher value.
	400,000	Peak flux for water in saturated pool boiling at one atmosphere.
	700,000 to 900,000	Internal combustion engine jacket flux, higher range.
	1,000,000	Ballistic missile re-entry with a laminar boundary layer.
	216,000 to 1,150,000	Range for 12 thermal nuclear reactors of various types, maximum core heat flux (average = 1/3 to 1/6 of maximum).
	2,200,000	"Commercial" parabolic reflector solar furnaces, maximum at focus.
	3,000,000	British Dounreay fast fission reactor core heat flux; maximum reported for supercritical water heat transfer.
	3,000,000 to 5,000,000	Attained successfully in nozzle throats of operational liquid-propellant rockets.

TABLE 3 (Continued)

HEAT FLUX RANGES (q/A in Btu/hr·ft ²)		
II.	5,000,000	Maximum input flux from electric arcs and solar furnaces operating continuously.
	6,000,000	Commercial "plasma jet" (water stabilized arc).
	7,000,000	Multiply reflected arc image furnaces, maximum continuous input flux.
	8,300,000	Maximum at focus of solar furnace for nearly ideal optical properties.
	11,000,000	Maximum expected at nozzle throats of rockets in near future.
	14,000,000	Maximum reported in literature for burnout with water in linear flow. [†]
	20,000,000	Ballistic missile re-entry with a turbulent boundary layer, worst case. Maximum known to authors for heat flux with water in linear flow. ^{††}
	32,000,000	Subcooled nucleate boiling of water in a continuously consumed flow channel.*
	35,000,000	Highest vortex boiling burnout flux attained by authors.
	45,000,000	Approximate for satellite re-entry.
	1,160,000,000	Carbon sublimation cooling (air plasma heat input), short time.**
	3,200,000,000	Maximum input flux at electrode surface for high-current, high-pressure, short-time electric arcs.

[†]Glasstone, S., Principles of Nuclear Reactor Engineering, p 695, D. Van Nostrand, New York, 1955.

^{††}Private communication from D. E. Bloxsom, Jr., Marquardt Aircraft Company, February, 1958.

*Private communication from Dr. Peter Griffith, MIT, September, 1957.

**Bloxsom, D. E., Jr., Heat Transfer and Fluid Mechanics Institute Reprints, pp 159-72, 1957.

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Fig. 1. Experimental Heat-Transfer Apparatus.

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SPIRAL RAMP VORTEX GENERATOR

TANGENTIAL ENTRANCE VORTEX GENERATOR

STRAIGHT FLOW ENTRANCE REGIONS

Fig. 2. Test Sections.

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Fig. 3. Vortex Boiling Correlation.

Fig.4. Burnout Heat Fluxes for Vortex and Linear Flow.

Fig. 5. Comparison of Heat Transfer vs Power Input in Subcooled Boiling for Linear and Vortex Flow in Tubes.

Fig. 6. Variation of Over-All Heat-Transfer Coefficient with Heat Flux in Vortex Flow.

Fig.7. Local Heat-Transfer Coefficients vs Local Heat Fluxes.

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