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LETTER TO THE EDITOR

Critical Heat Flux for Water in Swirling Flow*

The present trends in development of pressurized and boiling water nuclear reactors are toward high-power density systems. Critical heat flux (i.e., the burnout condition) is an important criteria in the design of these types of reactors. Since the critical heat flux limits the power density, increasing this flux would be advantageous.

Several experiments have recently been undertaken in an effort to show that critical heat flux can be increased by mechanical means. Oppenheimer (*1*) has reported about a dozen subcooled burnout data points, but has noted no significant increase in burnout heat flux with spinning flow. Gambill and associates (*2*, *3*) have carried out more extensive burnout heat-flux measurements with subcooled water in electrically heated horizontal tubes of small diameter. A substantial increase in critical heat flux was observed in these studies. The conditions under which these latter experiments were performed were much more severe than those of Oppenheimer. Reference *3* discusses advantages and possible applications of swirl flow and presents an extensive bibliography.

In this short note, a study of boiling burnout tests carried out at ANL with water in swirling flow is reported. Critical heat fluxes were determined at a system pressure of 2000 psia. Isothermal pressure drop was measured, and nonboiling friction factors were calculated.

The experiments were performed on a 2000-psia forced circulation boiling loop, which has been described in detail by Weiss (*5*). The vertical test section was a $\frac{5}{16}$ in. i.d., 21 in. long, Type 304 stainless steel tube. The two end fittings and electrodes were $1\frac{1}{2}$ in. long, leaving a heated test section of 18 in. The twisted Type 304 stainless steel strips, 0.005

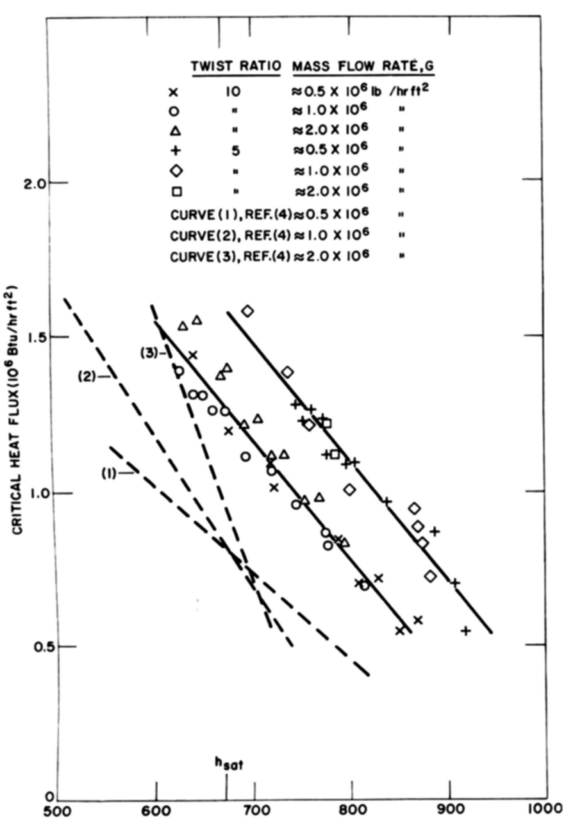

FIG. 1. Effect of twist ratio on critical heat flux at 2000 psia.

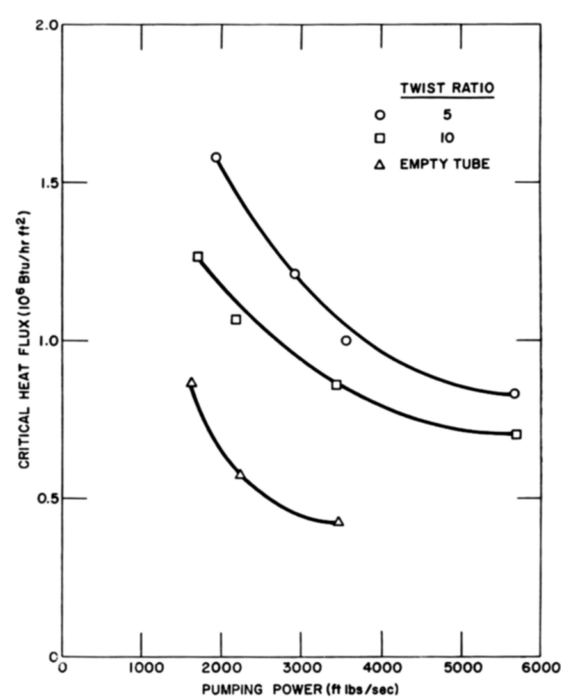

FIG. 2. Critical heat flux vs pumping power with net steam generation at 2000 psia and $G \approx 10^6$ lb/hr-ft².

* Work performed under the auspices of the U. S. Atomic Energy Commission.

in. thick, cut to fit the inner tube diameter snugly, were inserted in the tubes and welded at the ends. The twisted ribbons had twist ratios (internal diameters per 360° of twist) of 5 and 10, which correspond to a pitch of $1\frac{9}{16}$ and $3\frac{1}{8}$ in., respectively.

The burnout tests were made at a system pressure of 2000 psia for mass flow rates of approximately 0.5, 1.0, and 2.0×10^6 lb/hr-ft^2. Conditions for burnout were defined as those existing when a detector tripped the circuit breaker. The burnout detector circuit was a Wheatstone bridge, and the method of detection was the comparison for the electrical resistance of one inch of the test section nearest the downstream end to the electrical resistance of the rest of the test section. The burnout control system was designed to respond to a temperature excursion in this critical length of the heating tube and to disconnect the heating circuit whenever this temperature increase exceeded a predetermined value.

The procedure used in obtaining the data at burnout was as follows: The system was brought up to pressure, and the flow rate was set at the desired value by adjusting the throttling valve. The recording instruments were put into operation, and the test section inlet water temperature was reduced to as low value as possible by using no preheat and the maximum cooling afforded by the heat exchanger. A predetermined value of electric power was supplied to the test section. The inlet temperature was then raised slowly until the burnout detector tripped the circuit breaker. After the first burnout condition resulted, successive conditions were obtained by reducing the power to the test section and then slowly increasing the inlet water temperature.

The insertion of twisted strips not only produced swirling flow but also provided a parallel path for current flow. For instance, the coolant flowing with an axial velocity of 15 ft/sec through a $\frac{5}{16}$-in. tube, with a ribbon having a pitch of $1\frac{9}{16}$ in., would rotate at 6912 rpm and set up a rotational field which was a radial acceleration of 213 g. However, the actual rotational speed of the coolant is not 6912 rpm, and this value is only a ratio of the axial coolant velocity to the pitch. The heat generated in the ribbon was estimated from the knowledge of the total current and the electrical resistances of the tube as well as the twisted strip, and this amount was only about 3% of the total power applied to the test section. Thus, the exit enthalpy was calculated on the basis of the total heat generated in the entire test section, but the heat flux was based on the power produced in the tube only.

Some 60 test points were obtained, 52 of which are shown in Fig. 1. Several data points for which the mass flow rate measurements were questionable have not been included in the figure. The dashed lines in Fig. 1 represent Weatherhead's (*4*) data points for straight flow. The test sections used by Weatherhead in his tests were identical to those in the present experiments. The results indicate that a twisted ribbon inserted in a tube sets up centrifugal forces which are effective in breaking up vapor films and thereby increasing critical heat flux. Since with the available power supply a maximum heat flux of only 1.6×10^6 Btu/hr-ft^2 could be realized, no subcooled burnout data points could be obtained with the smaller twist ratio ribbon. There is some scatter of the data, particularly for the tube with the higher twist ratio ribbon; however, straight lines could be drawn through the data points for both cases. There seems to be some separation of the data with mass flow rate for the tube with a ribbon having a twist ratio equal to 10; still, not enough data are available to draw a definite conclusion.

The isothermal axial-flow friction factors obtained in the present experiments for the empty tube are a maximum of 15% higher than those for a smooth pipe. Insertion of a twisted ribbon in the tube increases pressure drop. The friction factors for the longer pitch ribbon are from 30 to 50% higher than for straight flow, and for the shorter pitch ribbon, they are only about 10% higher than for the former.

The friction factors obtained in the experiments with twisted ribbons inserted in the tube were correlated on a basis of an equivalent diameter of the test section. This was done by taking only half of the total flow area of the tube (allowance was also made for the thickness of the ribbon). The ribbon surface was also included in the calculation of the wetted perimeter. Thus, the friction of the twisted strip itself was accounted for.

A more significant comparison is given in Fig. 2 where critical heat flux is plotted versus pumping power for straight and spiral flows. The frictional, momentum, and hydrostatic head terms were included in the total pressure drop from which the pumping power was calculated. The points shown cover the entire experimental range of critical heat-flux data with net steam generation. Since tube geometry and mass flow rate are fixed, the pumping power increases only through an increase of steam quality, which increases the velocity of the two-phase mixture, and an increase in steam quality causes a decrease in critical heat flux. Figure 2 indicates that, for equal pumping power, the critical heat fluxes obtained with spiral flow are considerably higher than for straight flow. The data shown in Fig. 2 are for only one mass flow rate and are therefore not general; however, the conclusions are considered to be typical of other mass flow rates as well.

In conclusion, it can be stated that there are several factors which influence the heat transfer mechanism; however, the mixing due to the density difference in the presence of the centrifugal force field is probably the most important. The test results indicate that centrifugal forces are effective in breaking up the vapor film on the heated surface and increasing the critical heat flux in swirling flow. On the basis of constant pumping power, the critical heat fluxes obtained with swirl flow are a maximum of $2\frac{1}{2}$ times higher than those for straight flow.

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