SWIRL FLOW IN
DISPERSED FLOW FILM BOILING

by

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ABSTRACT

The heat transfer and pressure drop characteristics of two-
phase film-boiling nitrogen with tape-generated swirl flow through
a once-through boiler were investigated experimentally. The test
sections were electrically-heated inconel tubes of 0.40 in. I.D.
with tight fitting inconel tapes. Tape twist ratios of approximately
8.5 and 4.1 inside diameters per 180° of tape twist were considered.
Reference runs with straight flow tubes were made.

For equal flow rates, the maximum improvement in local film
boiling heat transfer coefficient with swirl flow was 200%. This
was achieved with the highest mass flux (100,000 lbm/hr ft²) used
for comparison with the 4.1 twisted tape.

On the basis of equal pressure drop, with equal mass flux, the
maximum improvement in overall film boiling heat transfer coefficient
with swirl flow was 70%.

Thesis Supervisor: Arthur E. Bergles

Title: Associate Professor of Mechanical Engineering
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A    Cross-sectional area of test section, square inches
f_T  Temperature correction factor - Eq. 3
G    Mass flux, lbm/hr ft^2
h    Heat transfer coefficient, Btu/hr ft^2°F
H_{fg} Latent heat of vaporization, Btu/lbm
I    Current, amperes
k    Thermal conductivity, Btu/hr ft°F
L    Length from inlet, inches
P    Pressure, psi
q''  Heat flux, Btu/hr ft^2
R    Resistance, ohms
r    Radius, inches
T    Temperature, °R or °F
TR   Twist ratio, diameters/180° of tape twist - used only in Fig. 2
V    Voltage, volts
W    Volumetric rate of heat generation, Btu/hr ft^3
w    Flow rate, lbm/hr
y    Twist ratio, diameters/180° of tape twist
Z    Axial distance along test section, inches
X_E  Equilibrium quality
Chapter 1
INTRODUCTION

In recent years there has been increasing interest in methods to augment heat transfer. This interest has been stimulated by the development of high power density thermal systems, and also the need to transfer heat in a zero gee environment, as in outer space.

The augmentation of heat transfer by turbulence promoters was recognized as early as 1921 by Royds [1]*. Since that time several basic augmentative techniques have developed such as the use of surface promoters, displaced promoters, vortex flows, fluid additives and electrostatic fields. Associated with these techniques are frequently increases in weight, fabrication cost, and/or pumping power. It is therefore necessary to look at the net improvement in a system and not just the increased heat transfer. A comprehensive survey and evaluation of the many augmentative techniques is given by Bergles and Morton [2].

Of the many techniques, those which produce vortex flows seem to be most attractive. Existing systems can readily be modified to accommodate vortex devices, and high heat fluxes are attainable. Gambill and Green [3] obtained a heat flux of almost $55 \times 10^6$ Btu/hr ft$^2$ with an inlet vortex generator on a very short tube; inlet vortices decay rapidly, however, and thus inlet vortex generators have limited application. Full length, twisted tape vortex generators have been used in many industrial applications and appear to be one of the most promising augmentative techniques.

*Numbers in parentheses designate references listed beginning on p. 36.
The heat-transfer and pressure-drop characteristics of single-phase flow with tape-generated swirl have been investigated rather extensively. There is considerable disagreement among investigators, as is pointed out in the survey papers by Gambill et al. [4] and Bergles [5]. The correlation recently suggested by Lopina and Bergles [6] seems to be as good as any. To be able to realize the overall effect of twisted tape inserts, the augmented channel is usually compared to the standard channel on the basis of equal pumping power or pressure drop.

The characteristics of two-phase flow with twisted tape inserts have been investigated less extensively than single phase. A recent study by Lopina and Bergles [6] with subcooled boiling shows that the boiling curve is essentially the same as in straight flow, but the pressure drop is substantially less. This, therefore, shows that twisted tape inserts can be very effective when compared on an equal pressure drop basis.

The critical heat flux (CHF) in subcooled boiling can be substantially increased over that for straight flow at the same pumping power. The data of Gambill et al. [8] show the CHF for swirl flow to be as high as twice that for straight flow at comparable flow conditions.

The performance of evaporators and high power boiler tubes has been substantially improved by the use of twisted tapes. Investigations have been made with water and liquid metals under conditions approximating those found in nuclear and space power plants [9,10].

The improvement in heat transfer by twisted tapes to post CHF flow has not been extensively investigated. Film boiling, or post CHF flow, is the dominant mode of heat transfer in many cryogenic systems, and is important in emergency shutdown of nuclear reactors. In cryogenic cooling systems, such as are used in many rocket engines, the vapor quality is
usually rather high so that liquid droplets are in a dispersed flow pattern. This suggests that the technique of swirl flow should be very effective in improving heat-transfer coefficients in this regime. The large centrifugal forces produced by a twisted tape will cause the liquid droplets to be thrown against the tube wall, thus improving wall to droplet heat-transfer.

Straight flow, forced convection film boiling has been investigated by Forslund [11] and others [12,13]. Sellers et al. [14] have reported data showing improvement in the first stage of once-through mercury boilers with vortex generators of the axial core type.

Burke and Rawdon [15] have reported film boiling heat-transfer data for a two-phase nitrogen stream with low flow rates in a 0.25 in. I.D. tube. Their test apparatus utilized a series of thermal capacitor rings whose inner boundaries constituted the heat-transfer surface. Their straight flow tests show a strong decrease in heat-transfer coefficient as quality is increased, the opposite of what one might expect. They report a limited amount of twisted tape data using a single, moderately twisted, full length tape \( y = 7 \) at a single flow rate. The flow rate used with the twisted tape was greater than any of the flow rates used for their straight flow, and with the twisted tape the heat transfer coefficient increased with increasing quality. An extrapolation of their straight flow data shows the heat-transfer coefficient for the twisted tape to be greater than 3 times that for straight flow at equilibrium qualities above 60\%. This is for a single flow rate and heat flux, and near atmosphere pressure.

The purpose of this study has been to obtain more comprehensive data on the augmentation of heat-transfer with film boiling of liquid nitrogen. The ranges of mass flux and heat flux studied were 20,000 - 180,000
1bm/hr ft\(^2\) and 1,000 - 18,000 Btu/hr ft\(^2\), respectively. Low pressures (5 - 10 psig) were used throughout the study.

The straight flow data used as a basis for comparison in this study were part of a present investigation [16], being performed in the Heat Transfer Laboratory, into the overall characteristics of film boiling of liquid nitrogen. The existing experimental equipment was modified to obtain twisted tape data over a range of flow rates with two twist ratio tapes.
Chapter 2

EXPERIMENTAL TEST APPARATUS

2.1 General

The experiments in this investigation were performed on the nitrogen test apparatus, which is located in the M.I.T. Heat Transfer Laboratory. The apparatus had been used in previous investigations [17,11,16] and only slight modifications were necessary. A schematic diagram is shown in Fig. 1.

2.2 Flow Apparatus

Liquid nitrogen was supplied from a 160 liter dewar as shown in Fig. 1. The dewar was pressurized to 100 psig by a regulated supply of pre-purified nitrogen gas. At the exit to the dewar, the flow passed through an electric solenoid valve which was connected to the emergency shutdown power switch. From the solenoid the flow passed to the first subcooler through a Fiberglas-insulated 3/8 in. copper tube. The first subcooler was a four ft section of the transfer tube enclosed in 0.5 in. copper tubing.

The flow then proceeded through the main subcooler to the inlet valve. The main subcooler was a 5 ft coiled tube-on-tube heat exchanger made of 3/8 in. O.D. and 1/2 in. O.D. type L copper tubing. At the entrance to the main subcooler, a small portion of the main flow was bled off through a 1/16 in. Hoke valve into the annulus of the subcoolers. From the subcoolers the bleed flow was steam-heated in the center of a tube-on-tube steam heat exchanger and then passed to a Welch #601-97
mechanical vacuum pump. Before entering the control valve, the main flow passed through a small brass block in which a pressure tap was mounted. The bleed flow also passed through a different channel in this block to insure that the block was below saturation temperature. The main control valve was a 1/8 in., 20 turns per in., precision valve. After the control valve, the main flow passed into a 1/4 in. threaded tee which turned the flow vertically into a 5 in. long inlet section upstream of the test section. The inlet section terminated at a steel flange to which the test section was secured. A fiber spacer was inserted between the steel flange and the test section flange to thermally and electrically insulate the two. The three bolts securing the two flanges together were electrically insulated by teflon washers.

On leaving the main test section, the flow passed through a short electrically heated glass section (approximately 5 in. long, 0.425 in. I.D.) through which visual observations of the flow regime could be made. The glass section was electrically insulated from the test section and downstream piping by oversize Swagelock fittings with teflon ferrules and nuts. Immediately downstream of the glass section, a stainless steel bellows was provided to allow for thermal expansion of the test section and any misalignment with the after-heaters.

The two after-heaters were tube-on-tube heat exchangers 6 ft long, made from 7/8 in. O.D., and 1 - 1/8 in. O.D. copper tubes. The nitrogen exit flow passed through the center of the heat exchangers and for high mass flux, and/or low heat flux runs, steam was passed through the annulus. The purpose of this was to bring the exit nitrogen as close as possible to ambient temperature before passing through the flowmeter(s). Toward
this end, the second after-heater could be bypassed by a 6 ft length of 7/8 in. copper tube exposed to the air.

A 3/4 in. valve was provided just upstream of the flowmeters to maintain the desired test section pressure.

The entire system from the solenoid to the glass sight section was thermally insulated with Santocel powder. The Santocel was contained by a 2-1/2 in. cardboard tube around the first subcooler, by a wooden box containing the main subcooler and valves, and by a 3 in. plexiglas tube around the test section.

2.3 Test Section

The basic boiler test section was an electrically heated round tube of Inconel 600. This material was chosen since its electrical resistivity changes very little with temperature. The tube was 96 in. long, had an O.D. of 0.50 in. and an I.D. of 0.40 in. Both ends of the test section were fitted with brass block flanges, 2-1/2 x 1-1/4 x 3/8 in., which were silver-soldered in place. The flanges served as connectors for the semi-flexible copper power lead at the bottom and the flexible battery strap power leads at the top. A small nichrome wire auxiliary heater was placed at the very bottom of the test section to facilitate establishment of film boiling. Power to this heater was turned off when taking data.

The top of the test section was held in position by a spring loaded yoke which provided for thermal expansion. Pins protruding from both sides of the top flange were carried in the yoke so that no bending moment was transmitted to the tube.

Copper-constantan thermocouples were used for temperature measurement. There were 24 of these located at 4 in. intervals along the test section
starting 2 in. from the inlet, and one was located at the joint with the bottom flange. The thermocouples were spot-welded directly to the test section thus assuring good thermal contact. This necessitated a separate ice bath for each to insure electrical isolation from each other. No 60 cycle AC voltage from the test section was discernable on any of the thermocouples. The lead wires from the thermocouples were wrapped three times around the test section circumference and then cemented to the tube with Sauereisen cement.

The thermocouple wires were then led along the test section to approximately the center, where they were all led out of Plexiglas tube containing the Santocel insulation. The wires from the thermocouples terminated at groups of 24 prong Cinch Jones plugs.

The two twisted tapes ($y = 8.5$ and $4.1$) were fabricated from the same piece of Inconel 600 sheet, 0.018 in. thick. The sheet was first cut into four strips, 0.45 in. wide and 2 ft long. The strips were then welded end-to-end to form a continuous tape 8 ft long. This tape was hand-sanded to a width of 0.398 in. ± 0.002 in., and a thickness of 0.019 in. ± 0.001 in. at the three weld points. The twist was put in the tape by securing one end to the ceiling and hanging a 100 lb weight on the bottom. The weight was turned until the proper twist was obtained. The 4.1 twist ratio tape was formed by retwisting the 8.5 tape after data had been taken with it.

Insertion of the twisted tape was facilitated by first oiling both the tape and the inside of the test section. The tape was then inserted with a screw motion, with a small tension being applied to a wire leader. The completed assembly was then flushed three or more times with tri-
chloroethylene and then continuously with hot tap water for at least 1 hour. Both twisted tapes fit snugly with a maximum estimated clearance of 0.004 in. at any point. The tapes were silver-soldered to the test section at the inlet to insure good electrical contact and prevent the tape from moving. From the decreased resistance of the test section with the twisted tape, good electrical contact at the outlet was found to exist. Due to the observed tight fit of the tape in the test section, good electrical contact was assumed to exist all along the tape.

2.4 Power

The test section was heated by 208 volt, 60 cycle lab power. Primary voltage was controlled by two 110 volt variacs synchronized on a common shaft. The primary voltage was converted to low voltage by two step-down transformers connected in parallel with a ratio of 10 to 1. One transformer was a General Electric #61G76, capable of 5 kva, and the other a General Electric #9T51Y, capable of 3 kva. This power supply permitted a maximum heat flux of approximately 19,000 Btu/hr ft² with the particular test section used in these tests. Either 4 strands of #3 welding cable or 2 strands of #2 were used to carry the secondary current (up to 240 amperes) to the test section.

The power to the inlet flange heater and the heated glass sight section was supplied by 3 small Superior variacs capable of supplying up to 3 amperes.

2.5 Instrumentation

Test-section pressure was measured by five U-tube monometers connected to the five equally-spaced static pressure taps on the tube. The pressure lines each contained a 3 in. section of 0.010 in. I.D. capillary tube to
damp out small pressure fluctuations. The inlet pressure manometer contained mercury and was open to atmospheric pressure. The other four manometers contained either water or Meriam Fluid #3, and registered the pressure drop between successive pressure taps.

Bleed line pressure was measured at two places; by a 0-15 psi, 0-30 in. Hg vacuum gauge just after the bleed valve, and by a 0-30 in. Hg vacuum gauge after the subcoolers. Main flow pressure was monitored just upstream of both the solenoid valve and the main flow valve by 0-100 psi Helicoid gauges. Main flow pressure was also read just after the control valve by a 0-60 psi Helicoid gauge.

All thermocouple measurements were read on a 16 channel Brown recorder (Minneapolis-Honeywell Model 153x52V16). The 24 test section thermocouples were split into two groups of 12 odd and 12 even. Also recorded were the readings of thermocouples immersed in the flow at the inlet and outlet of the test section, the rotometers, the vacuum pump, the exit from the dewar, and just after the bleed control valve. The inlet flange temperature, and a reference atmospheric nitrogen temperature were also recorded on the Brown recorder.

Power to the test section was obtained by measuring the voltage drop across the test section and the current passing through it. The voltage drop between the end flanges was measured with a Weston Model 433 ac precision voltmeter with a dual range of 0-10 and 0-20 volts. The current through the test section was measured with the use of a high precision Weston current transformer, Model 461 (ratio 80:1), and a Weston Model 370 ac ammeter with a dual range of 0-5 and 0-10 amperes. The voltmeter was calibrated and found to be ± 0.01 volts on the low range and ± 0.02
volts on the high range. The ammeter was calibrated and found to be ± 1/2 percent on the low range which was the only range used.

Nitrogen flow rate was measured by two Brooks Flowmeters used both singly and in parallel. One was a model 10-1110-10 with a tube size of R10M-25-1, and the other a model 10-110 with a tube size of R10M-25-3. The full scale readings at 70 °F and 14.7 psia were 80 and 118 lbm/hr, respectively. Both flowmeters were calibrated to ± 1 percent of full scale.

2.6 Experimental Limitations

In general, the upper limitations of heat flux and mass flux were dictated by the lab power available. The highest heat flux (approximately 18,000 Btu/hr ft\(^2\)) was obtained using 40 primary amps to the transformers which is the fuse limit of the variacs. This maximum heat flux also limits the mass flux since at mass fluxes higher than approximately 200,000 lbm/hr ft\(^2\), film boiling collapsed. At low mass fluxes and high heat fluxes the system was limited by the outlet gas temperature. Temperatures higher than 1000 °F were avoided to prevent melting of the solder in the exit lines and/or damage to the test section.
Chapter 3

EXPERIMENTAL PROCEDURE AND DATA REDUCTION

3.1 Experimental Procedure

A calibration check of the temperature recorder was made before the flow or power was turned on for a particular run. This was done by recording the output of a Leeds and Northrup potentiometer at -5, 0, +5, +10, and +15 millivolts. The vacuum pump was then started, and the bleed flow and a small amount of main flow were turned on to cool down the inlet transfer line. If film boiling was desired at the inlet to the test section, the tube was heated before liquid reached it; otherwise it was allowed to cool down before the power was applied. The desired heat flux, mass flux, and system pressure were then set and the system allowed to come to equilibrium. Then the switching of thermocouples and the marking of the temperature recorder was handled by one operator, while the other operator recorded the voltage, current, flow, and pressure readings. The time required for the system to reach equilibrium varied from approximately 5 to 20 minutes, and it usually took about 2 minutes to record the data.

3.2 Data Reduction

3.2.1 General

A Fortran IV computer program was written to facilitate the data reduction and present it in a useful form. It was run on the IBM System 360, Model 65 computer at the M.I.T. Computation Center. A sample print-out is shown in Fig. 2. Similar records of all data obtained in the investigation are on file in the M.I.T. Heat Transfer Laboratory.
3.2.2 Heat Flux

Heat input was calculated from the voltage and current measurements. The power factor of the system was measured on an oscilloscope and found to be 0.94. The heat generated in the tape was estimated from the knowledge of the current and resistance of the test section with and without the tapes, and this amount was about 8 percent of the total power applied to the test section. The enthalpy at any point was calculated on the basis of the total heat generated in the test section, but the heat flux was based on the area of the tube wall only. This has been done in most previous swirl flow investigations.

3.2.3 Temperatures

Thermocouple voltages above -5.379 millivolts (lowest value given in Leeds and Northrop conversion table) were converted to °R and °F by a subroutine that contains eleven fourth-order polynomial curve fits to the conversion tables. Voltages below -5.379 millivolts were converted to °R and °F by a different subroutine containing a fourth-order curve fit to low temperature thermocouple data supplied by the M.I.T. Cryogenic Laboratory. The measured temperature of saturated liquid nitrogen at atmospheric pressure was used as a reference point for the curve fit.

The inside wall temperature was calculated at each of the 24 locations on the test section by solution of the conduction equation

\[
\frac{d}{dr} \left( r \frac{dT}{dr} \right) + \frac{Wr}{k} = 0 \quad (1)
\]
with the boundary conditions

\[ T = T_0 \quad \text{at} \quad r = r_0 \]

and

\[ \frac{dT}{dr} = 0 \quad \text{at} \quad r = r_0 \]

For uniform \( w \) and constant \( k \), this results in the equation

\[ T_i = T_0 - \frac{w}{2k} \left( \frac{r_i}{r_o} \right)^2 \left( 1 - \ln \frac{r_i}{r_o} \right) \]  \hspace{1cm} (2)

The conductivity was evaluated at the outside wall temperature for each point. In this investigation \( T_0 - T_i \) was always less than 3 °F, so the assumption of \( w \) uniform and \( k \) constant across the tube wall is valid.

### 3.2.4 Mass Flux

Flow meter readings were converted to mass flow rate by calibration curves for both flow meters. A temperature correction suggested by the flow meter manufacturer, Brooks, was used. It was

\[ f_T = \frac{530}{T_r(°R)} \]  \hspace{1cm} (3)

Mass flux was obtained from mass flow rate by the usual equation

\[ G = \frac{W}{A} \]  \hspace{1cm} (4)

The cross-sectional area of the flow differed between the twisted tape and straight runs, the difference being the cross-sectional area of the tape itself.

### 3.2.5 Pressure

Pressure was evaluated at the 24 thermocouple points on the test
section from a fourth-order curve fit to the five local pressures. The slope of the pressure-length curve was calculated at each point to be sure that there were no inflections in the curve itself.

The test-section pressure was also evaluated at the calculated points where equilibrium qualities of 0.0, 0.5, and 1.0 existed so that total pressure drop between these points could be determined.

3.2.6 Equilibrium Quality

The equilibrium quality of the flow was calculated at each thermocouple location from the equation

\[ \chi_{Ez} = \frac{4q''u z}{\frac{D}{R}} - \frac{c_1 \Delta T_{sub}}{H_{fg}} \]  

(5)

\( H_{fg} \) and \( c_1 \) are evaluated at each point from fourth order curves fit to latent heat-pressure, and specific heat-pressure data [18] for nitrogen in the range of interest.

3.2.7 Heat-Transfer Coefficient

The pressure at each thermocouple location was used to calculate the saturation temperature at that point. The bulk temperature of the fluid was taken to be the saturation temperature for equilibrium qualities between zero and one. This assumes that equilibrium exists in this quality range and the assumption is made for calculation purposes only. Forslund [11] has shown that this is generally not the case. The bulk temperature was corrected for the specific heats of liquid and vapor for equilibrium qualities below zero and above one, respectively. The heat-transfer coefficient at each point was taken to be the heat flux divided
by the difference between the \textit{inside wall temperature} and the \textit{bulk temperature}.
Chapter 4

PRESENTATION AND DISCUSSION OF THE RESULTS

4.1 Straight Flow

"Burnout" as used in this paper does not mean physical distuction of the test section. Burnout was assumed to have occurred when the wall temperature was greater than 60 °R above the saturation temperature of the nitrogen (approx. 150 °R).

It was found that burnout with straight flow could be roughly classified into two categories. The first category, herein called Type I, is characterized by burnout at low equilibrium qualities (less than 20 percent). The location of this type of burnout was sometimes observed after the first thermocouple but more often the whole test section was considered to be in film boiling. The second category of burnout, given the name Type II, occurred at high equilibrium qualities (usually greater than 70 percent).

Two very similar types of burnout have been observed previously with water [15] and have been given the names fast and slow burnout, corresponding to Type I and Type II, respectively. The name fast burnout has been used with water systems since a burnout under sub-cooled or low quality conditions usually causes the surface temperature to rise rapidly to a very high value, generally resulting in actual physical burnout. The term slow burnout was used to describe the condition where the wall temperature rise, after CHF in the high quality region, is quite slow. This is due to the lower level of heat flux associated with a Type II burnout and probably a certain amount of
re-wetting of the tube wall by the annular liquid film. Vapor core velocity is also large at a high quality burnout and this tends to keep the wall temperature down also.

Type II, or high quality burnout, was not observed in an earlier investigation with nitrogen [11]. This was due to the use of a small heater around the test section at the inlet which caused burnout to occur always at the entrance to the test section. As mentioned previously, an inlet heater was sometimes used in this investigation during start up only; it was always turned off during a run.

The two types of burnout that were observed in this investigation were characterized by different temperature profiles. Figure 3 shows typical Type I temperature profiles; after burnout occurred, the temperature either stayed level or dropped to a minimum and then increased. This variation of temperature is with length along the test section and not time. It is presumed that the flow pattern after Type I burnout is that of a vapor annulus blanketing the wall and a dispersed liquid core. Figure 4 shows typical Type II temperature profiles; once the burnout occurred, the temperature kept rising. Before burnout, the wall temperature generally dropped to a slightly lower value than at the inlet. This is probably indicative of a thinning liquid film on the wall, the regime perhaps becoming forced convective vaporization with suppression of nucleation.

Both types of burnout are possible at the same mass flux and heat flux within certain ranges of both. This is analogous to, and can be explained by, a flowing Leidenfrost phenomenon. As in pool boiling, it is possible to have either film boiling or nucleate boiling at the
same heat flux as long as it is above the Leidenfrost point. This is shown on Fig. 5. Some work was done in this investigation on determining the analogous "flowing Leidenfrost point", and it was found to be only weakly dependent on flow rate as shown in Fig. 6. The temperatures shown in this figure are the temperatures of the first thermocouple on the test section, (2 in. from the inlet) just before transition from a Type I burnout to nucleate boiling at the inlet occurred. The transition from film boiling to nucleate boiling was made to occur by keeping the flow rate constant and reducing the heat flux. The inlet was always the location of transition with straight flow, thus the local quality at transition was always slightly subcooled.

It was found, however, that the system's capabilities prevented a transition back to full film boiling along the test section without first shutting down the flow. If the flow was kept constant and the heat flux raised, the wetted length of tube wall would decrease, but the outlet temperature of the flow would reach its maximum permissible value before film boiling would occur at the inlet. Film boiling at the inlet could be reinstated by using the nichrome wire inlet heater. However, this procedure was not used because it took longer afterwards for the test section to reach equilibrium than it would when a flow shutdown procedure was used.

4.2 Twisted Tape Results

4.2.1 Critical Heat Flux

As with the straight tube it was also found that two types of burnout were possible with the twisted tapes. Typical wall temperature profiles for Type I and Type II burnout with twisted tapes are shown
in Figs. 7 and 8, respectively.

In the process of investigating the transition from a Type I to a Type II burnout, it was found that a combination of the two types of burnout was possible at high mass fluxes. This combination of burnouts was characterized by wall temperature distributions as shown in Fig. 9. This state of operation was reached by keeping the flow rate constant with a Type I burnout and reducing the heat flux. Once the liquid film had wetted the tube wall in the high quality region the location of the upstream edge of the film was observed to move slowly toward the inlet. The rate of growth of the liquid film seemed to be rather large (on the order of 10 in./min.) just after transition (first 1/2 min.), in fact almost a step change. After this initial growth, however, the film seemed to creep very slowly, almost imperceptibly, toward the inlet. Due to the distance between thermocouples (4 in.) on the test section, it was almost impossible to measure this slow rate of growth (probably less than 8 in./hr ). It is possible that the film had actually reached a steady state condition and that the small changes in wall temperature that were noticed near the edge of the film were due to very slight deviations in heat flux or mass flux. It is thought, however, that after a considerable length of time (more than one hour) a full Type II burnout might have occurred. The amount of liquid nitrogen available at one time was not sufficient to allow waiting the time that this full transition might take.

The tube wall was wetted first in the high quality region due to the large centrifugal forces produced by the fast moving dispersed swirl flow. At high mass fluxes the centrifugal force on the dispersed
liquid droplets was sufficient to overcome the Leidenfrost force tending to keep the droplets from wetting the tube wall. This explains why the high quality transition was not observed at low mass fluxes.

After a partial transition from a Type I burnout at high quality, it was found possible, at moderately high mass fluxes, to cause a reverse transition back to a full Type I burnout by increasing the heat flux considerably. This was more easily accomplished within the outlet temperature restrictions with the 8.5 tape than with the 4.1 tape.

4.2.2 Heat Transfer Coefficient

The improvement in film boiling heat-transfer coefficient of the twisted tapes is shown in Figs. 10 through 13. Figures 10 and 11 show that at low mass fluxes \((G = 20,000 \text{ lbm/hr ft}^2)\) and the same equilibrium qualities, the 8.5 tape improved the local heat-transfer coefficients between 60 percent and 70 percent, and the 4.1 tape showed an increase of approximately 100 percent. In both cases the improvement was slightly greater at high qualities than at low qualities. This is due to the relative ineffectiveness of the twisted tapes in the slow moving liquid core at low qualities. These results agree very well with those of Burke and Rawdon [15].

For the moderately high mass flux \((G = 100,000 \text{ lbm/hr ft}^2)\), as shown in Fig. 12, the 8.5 tape produced a 100 percent increase in heat transfer coefficient in the quality range of 0.7-0.8 and the 4.1 tape an increase of 200 percent. Again, at low qualities the improvement was less, due to the ineffectiveness of the tape.

At the highest mass flux used for comparison \((G = 180,000 \text{ lbm/hr ft}^2)\) the 8.5 tape caused an increase in the heat transfer coefficient of
120 percent at the quality of 0.7, as shown in Fig. 13.

The discontinuities in the plots of Figs. 12 and 13 are the locations where transition from a Type I burnout at high quality occurred, as mentioned previously. The data of these discontinuous plots was taken after initial transition had occurred and the film was creeping very slowly.

At qualities greater than 0.8, the heat-transfer coefficients for the twisted tape, high mass flux runs, are seen to decrease quite rapidly and approach those of the straight flow. This shows that the general increase in heat-transfer coefficient with the twisted tapes is not due solely to the increase in local velocity of the swirl flow. It is possible that the increase in heat-transfer coefficient could be due to a decrease in the non-equilibrium of the flow [11,16] and the resulting increase in the volumetric flow rate. This, however, does not seem sufficient to explain such large increases in the heat-transfer coefficient. It is, therefore, most probable that the major contributing factor is the increased heat transfer between the wall and the liquid drops. This would explain the sharp decrease in heat-transfer coefficient after an equilibrium quality of 0.8 with the twisted tape at high mass fluxes; there are few and very small droplets present at higher qualities.

4.2.3 Equal Pressure Drop Heat Transfer

To compare, on an equal pressure drop basis, the increase in heat transfer of the twisted tapes over the straight flow, two methods of presentation have been used. The first is shown in Figs. 14 through 19, where the heat-transfer coefficient is plotted versus the pressure gradient at constant mass flux and particular equilibrium qualities.
The pressure gradient for the points on these figures was calculated by taking the derivative of the fourth-order curve fit to the five local pressures. Typical local pressures and the curves fit to them are shown in Fig. 20.

In Figs. 14 through 19, it is seen that the twisted tapes result in a general increase in the heat transfer coefficient at equal pressure gradient. However, as can be seen from the figures, it is difficult to determine just what the amount of increase is. It does seem that the increase is greater at the quality of 0.5 than at 1.0. This might be expected in lieu of the wall-to-droplet heat transfer discussion given previously.

The most favorable comparison of equal pressure gradient heat-transfer coefficient, as shown in Fig. 16, is between the twisted tape data with low heat fluxes and the straight data with high heat fluxes. For these conditions the 8.5 twisted tape can be said to show an improvement of approximately 70 percent in the equal pressure gradient heat-transfer coefficient, and the 4.1 twisted tape has a slightly greater improvement.

In Figs. 14 through 19, comparisons are made with equilibrium qualities being the same. As mentioned previously, equilibrium quality is generally not equal to actual quality, and the difference between them should be less with the twisted tapes. Therefore, these comparisons might be considerably different if actual quality could be used. Using comparisons of heat transfer coefficients at the same equilibrium qualities does have one advantage; it gives an indication of the relative wall temperatures of the straight and twisted tape results. If,
for the same mass flux, pressure gradients, and equilibrium qualities, the twisted tapes have generally higher heat transfer coefficients, then the twisted tape wall temperatures will be generally lower than the straight flow wall temperatures. This is an important consideration in many design applications.

Another of the difficulties in presenting data as shown in Figs. 14 through 19 is the evaluation of the pressure gradient. Differentiating a fourth-order curve fit to the pressures tends to accentuate any irregularities in the curve and enlarge the errors. Also, local parameters such as pressure gradient are generally not sufficient for overall performance evaluation in two-phase flow; they are usually sufficient in single-phase flow evaluation.

For these reasons, a more basic approach for determining the augmentative value of the twisted tapes has also been used.

The second method of presentation of results is shown in Figs. 21 through 28. In this method, the pressure drop between two fixed qualities is plotted as a function of heat flux for the twisted tape and straight data at constant mass flux and pressure level. Considering a constant mass flux and constant change in quality is equivalent to considering a constant power input or total heat-transfer rate.

The purpose of representing the data in this fashion is to determine how much shorter a twisted tape test section may be than a straight flow test section for the same mass flux, pressure drop, and power input. For a given available pressure drop, mass flux, and power input ($\Delta X_E$), the heat flux with the twisted tapes can be higher, as is shown in Figs. 21 through 27. Then, from the First Law of Thermo-
dynamics, which can be represented as in Fig. 28, the resulting change in test-section length can be determined.

A sample of some of the results thus determined is shown in Table I. The top line of the table shows that for a mass flux of 20,000 lbm/hr ft
\(^2\), a pressure drop of 0.12 psi, and a change in quality from 0 to 0.5, the 4.1 twisted tape test section need only be 70 percent as long as the straight one. This can possibly be interpreted as a 30 percent improvement in heat transfer at equal pressure drop. As can be seen in Table I, the equal pressure drop improvement in heat transfer increases with increasing mass flux. At the highest mass flux, the length of the twisted tape test section need only be 28 percent that of the straight tube, which represents a heat transfer improvement of 72 percent at equal pressure drop. A disadvantage in presenting the data in this fashion is that it gives no information as to the relative wall temperatures of the twisted tape versus straight flow test sections. It is necessary to go back to the original data for this information.

A comparison of the type described above cannot be made for the 0 to 1 quality range at high mass fluxes due to the limitations of the system and the nature of the data required. To obtain equal pressure drops, either very high heat fluxes with the twisted tape or very low heat fluxes with the straight tube would be required. The high heat flux needed with the tape would cause an excessive outlet temperature. At very low heat fluxes the test section is not sufficiently long to reach an equilibrium quality of 1.0 and/or film boiling would collapse.

Some work was done in this investigation to determine the effect of twisted tapes on the critical heat flux. Much of the limited data
is somewhat scattered and Fig. 29 represents the most accurate comparison. As can be seen from the figure, the 4.1 twisted tape shows what seems to be a decrease in critical heat flux. This cannot be explained by the slightly higher mass fluxes for the twisted tape runs in this comparison since it has been found [16] that critical heat flux increases with increasing mass flux for this type of burnout. Figure 30 shows the wall temperature distributions for the two runs shown in Fig. 29 at the heat flux of 12,000. As can be seen, the wall temperatures after burnout are quite different.
Chapter 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The conclusions of this investigation on the effect of twisted tape swirl flow on forced convection film boiling are summarized as follows:

1) For both straight flow and flow with twisted tapes, there are two types of burnout, a high quality burnout and a low quality burnout.

2) For equal flow rates, the effect of the twisted tapes was to increase the local heat-transfer coefficient as much as 200 percent over the straight flow value. The increase was greatest at large mass fluxes.

3) On the basis of equal pressure drop, the length of a test section could be reduced over 70 percent for the same heat transfer.

5.2 Recommendations for Further Research

Only two twist ratio tapes were used in this study. To determine the full effect of twisted tapes on forced convection film boiling, it is recommended that the effect of at least one other twist ratio tape be studied. This tape should be of the tightest possible twist ratio \(y = 2\). An axial core swirl promoter with a tighter twist ratio than \(y = 2\) (maximum for a twisted ribbon) should possibly be studied also.

This study was limited by the maximum outlet temperature. It is recommended, therefore, that a shorter test section be used so that high heat fluxes are possible at moderate flow rates.

A third recommendation for future research is for the modification
of the primary power system to make generally higher heat fluxes possible.

Twisted tapes seem to be a very effective means of increasing heat transfer in the film boiling, dispersed flow regime. More work should be done with other fluids and different size test sections so that a fuller understanding of the augmentative value of twisted tapes in this regime is obtained.
REFERENCES


Fig. 1 Schematic Diagram of the Test Apparatus
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**Outlet Temperature:** 436.36 Degrees Rankine

**Total Pressure:** 4,613 PSI

**Dewar Pressure:** 3,724 PSI

**Outlet Pressure:** 2,142 PSI

Fig. 2 Typical Computer Output for a Run
Fig. 3  Straight Flow Wall Temperatures for Type I Burnout.
Fig. 4 Straight Flow Wall Temperatures for Type II Burnout
Fig. 5 Typical Pool Boiling Curve
Fig. 6 Transition Temperatures and Heat Fluxes for Straight Flow
Fig. 7 Twisted Tape Wall Temperatures for Type I Burnout
Fig. 8 Twisted Tape Wall Temperatures for Type II Burnout
Fig. 9 Twisted Tape Transition Wall Temperatures
Fig. 10 Effect of Twisted Tapes on Heat Transfer Coefficient - Low Mass Velocity
Fig. 11 Effect of Twisted Tapes on Heat Transfer Coefficient - Moderate Mass Velocity

\[ G = 40,000 \text{ lbm/hr-ft}^2 \]

\[ P = 21 \text{ psia} \]
$G = 100,000 \text{ lbm/hr-ft}^2$

$P = 22 \text{ psia}$

![Graph showing the effect of twisted tapes on heat transfer coefficient at high mass velocity](image)

**Fig. 12** Effect of Twisted Tapes on Heat Transfer Coefficient - High Mass Velocity
Fig. 13 Effect of Twisted Tapes on Heat Transfer Coefficient - High Mass Velocity
Fig. 14 Heat Transfer Coefficient versus Pressure Gradient - Low Mass Velocity

Fig. 15 Heat Transfer Coefficient versus Pressure Gradient - Low Mass Velocity
Fig. 16 Heat Transfer Coefficient versus Pressure Gradient - Moderate Mass Velocity
Fig. 17 Heat Transfer Coefficient versus Pressure Gradient - Moderate Mass Velocity
Fig. 18 Heat Transfer Coefficient versus Pressure Gradient - High Mass Velocity

G = 100,000 lbm/hr-ft²
Xₑ = 0.5
- 4.1 Tape
- 8.5 Tape
- Straight Flow

Fig. 19 Heat Transfer Coefficient versus Pressure Gradient - High Mass Velocity

G = 100,000 lbm/hr-ft²
Xₑ = 1.0
- 4.1 Tape
- 8.5 Tape
- Straight Flow
Fig. 22 Pressure Drop versus Heat Flux - Low Mass Velocity

G = 20,000 lbm/hr-ft²

ΔX_E = 1.0

X_E_in = 0.0

y = 8.5 Tape

y = 4.1 Tape

Straight Flow
Fig. 23 Pressure Drop versus Heat Flux - Moderate Mass Velocity
Fig. 24 Pressure Drop versus Heat Flux - Moderate Mass Velocity
Fig. 25 Pressure Drop versus Heat Flux - Moderate Mass Velocity

\[ G = 60,000 \text{ lbm/hr-ft}^2 \]
\[ \Delta x_E = 0.5 \]
\[ \chi_{E_{in}} = 0.0 \]
Fig. 27 Pressure Drop versus Heat Flux - High Mass Velocity
Fig. 28 First Law of Thermodynamics
Fig. 29 Effect of Twisted Tape on CHF
Fig. 30 Critical Heat Flux Temperatures
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Table I

Percentage Reduction in Length of Test Section

Possible with Twisted Tapes