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FILM BOILING OF CRYOGENIC HYDROGEN DURING UPWARD AND DOWNWARD FLOW

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SUMMARY

The influence of buoyancy on a vertical flowing film boiling system was determined by comparison of upflow and downflow data under identical test conditions. Differences in the data were observed at two heat flux levels.

At low heat fluxes, the transition from nucleate to film boiling occurred prematurely for downflow at heat fluxes that supported nucleate boiling for upflow. At high heat fluxes, the inlet side of the downflow test section operated at a significantly higher temperature than for upflow. This difference in heat transfer was correlated by modification of an existing film boiling equation.

INTRODUCTION

Film boiling of cryogenic fluids occurs readily at low or moderate heat fluxes. Hydrogen, for example, when used as a coolant below the critical pressure operates almost exclusively in the film boiling regime. Consequently, studies aimed at understanding the characteristics of this mode of heat transfer can be of both scientific and practical interest. In particular, practical flow geometries could dictate upward and downward flow conditions that might be influenced by the buoyancy force.

Previous studies (refs. 1 to 4) have shown that both the hydrodynamic and heat transfer characteristics of liquid nitrogen film boiling can be considerably influenced by flow direction. However, only a limited amount of data is available and no generalization may be made regarding other fluids or test conditions.

The present paper is aimed at determining the influence of buoyancy on a boiling liquid hydrogen vertical flowing system. The apparatus consists of an instrumented tube test section as part of a flow system capable of being oriented so that fluid flow could be either in the direction of or opposed to the Earth gravity vector. By comparison of data obtained in both flow directions under identical test conditions it is possible to determine the influence of buoyancy on the flowing system.

NOMENCLATURE

Cp	specific heat
g	gravity acceleration
h	heat transfer coefficient
k	thermal conductivity
l	distance from inlet
T	temperature
ΔT	$T_w - T_s$
U	velocity
μ	absolute viscosity
ρ	density
λ	heat of vaporization

λ' $\lambda\left[1 + \frac{0.4(\Delta T)Cp}{\lambda}\right]^2$

Subscripts

b bulk

c calculated

d downflow

f film

L liquid

s saturated

u upflow

v vapor

w wall

Experimental Apparatus and Procedure

The liquid hydrogen heat transfer rig used in this study is described in detail in reference 2. The flow system operates in the blowdown mode by pressurizing a Dewar to force the fluid through an instrumented test section and then to discharge through a burn off torch. The main feature that makes this apparatus well suited for this study is its portability that allows rotation of the test section without making any changes in the piping system.

The test section was made of inconel X tubing having an inside diameter of 0.505-inch, a 0.010-inch wall thickness and a 12-inch heated length. Power was generated by a 1000 watt-400 hertz alternator using the test section as a resistance heater. Test section wall temperature profiles were measured with 12 copper-constantan thermocouples. Bulk temperature measurements were made with platinum resistance thermometers.

The test procedure was standardized by systematic variation of the controlled test variables that included pressure, flow rate, heat flux, and flow direction. With the test section positioned vertically and the pressure and flow rate set a data run was made with step increases of power over a range of heat fluxes from (0.02 to 0.32) Btu/sec in.^{2}. The fluid inlet conditions were saturated at 35 psia and data were obtained over a range of inlet velocities from 4.5 to 12.5 ft/sec.

Presentation of Data

The data are illustrated graphically so that direct comparison can be made between the upflow and downflow wall temperature profiles along the test section. Figure 1 presents data for an inlet velocity of 9.6 ft/sec. The range of heat fluxes used in this study produced wall temperatures that are indicative of both nucleate and film boiling as shown by the arrow of figure 1. Wall temperatures of about 50° R are representative of nucleate boiling while transition to film boiling occurs at higher temperatures. The upflow temperature profiles are represented by solid lines drawn through open symbol data points and the downflow profiles shown as dashed lines through filled symbols. Differences in the data are readily observed and will be discussed by examining individual pairs of curves.

At a heat flux of 0.02 Btu/sec in.^{2}, data comparison shows a 200° R wall temperature difference with nucleate boiling for upflow and film boiling for downflow.

As the heat flux is increased to 0.08 Btu/sec in.^{2} the upflow data shows a transition from nucleate to film boiling near the inlet with the balance of the test section in transition and film boiling. The downflow data again shows wall temperatures that are indicative of film boiling. At the first measuring station downstream from the inlet (0.75 in.) a wall temperature difference of about 375^{o} R exists.

The two remaining pairs of curves at heat fluxes of 0.18 and 0.32 Btu/sec in.^{2} are completely in the film boiling regime with significant differences only appearing just downstream of the inlet. At the first measuring station the downflow wall temperature is up to 240^{o} R higher than for the upflow case at a heat flux of 0.32 Btu/sec-in.^{2}.

Although only this particular set of data has been presented in detail the results are quite representative of the data.

Buoyancy Effects on Heat Transfer

The examination of figure 1 has shown that the heat transfer characteristics of a turbulent flow liquid hydrogen system can be significantly influenced by the buoyancy force. Differences in upflow and downflow data appeared at low heat fluxes in the transition from nucleate to film boiling and just downstream of the inlet at high heat fluxes with film boiling established in both flow directions.

At low heat fluxes, the transition from nucleate to film boiling was found to occur prematurely for downflow in comparison to upflow. Due to buoyancy, the vapor tends to rise in opposition to the flow direction. The increased vapor accumulation decreases the heat transfer to permit the establishment of a film on the heat transfer surface. In the upflow case the buoyancy force would tend to aid the removal of vapor.

At higher heat fluxes, with film boiling established in both flow directions the shape of the temperature profiles are due to a change in flow model as the fluid progresses downstream. An examination of the pairs of curves at heat fluxes of 0.18 and 0.32 Btu/sec in.^{2} will show that the wall temperatures just downstream of the inlet decrease sharply and then level off downstream. The higher wall temperatures near the inlet are the result of a laminar like vapor flow that becomes turbulent as the fluid progresses downstream. Observations such as these have been made for liquid nitrogen in reference 3.

The only significant differences in the data appears in the laminar like vapor flow region ($l < 2$ in. from the inlet) with downflow wall temperatures higher than for upflow. These differences must be attributed to buoyancy on the film boiling mechanisms. It is well known, reference 5, that under conditions of laminar vapor flow the primary mode of heat transfer is conduction through the vapor film so that film thickness is a controlling heat transfer parameter. For the present study both the flow rate and the buoyancy force should influence the thickness of this vapor film. Since the data are compared at the same flow rates the difference in heat transfer is due to decrease vapor removal by buoyancy resulting in a thickening of the vapor film.

A correlation of this buoyancy effect was made by modification of an equation from reference 5. The results are presented in appendix.

APPENDIX - HEAT TRANSFER IN LAMINAR VAPOR FLOW REGION

Bromley, reference 5 derived an equation applied to film boiling off a horizontal rod in cross-flow. In the analysis, the derivation required the existence of laminar vapor flow. Although the geometry is not consistent with that used in the present study it was decided to attempt a correlation of the present laminar vapor flow data with the following functional grouping of parameters derived by Bromley:

$$h_{c,u} \approx \left[\frac{k_f^3(\rho_L - \rho_v)\rho_v g\lambda'}{l(\Delta T)\mu}\right]^{1/4} \tag{2}$$

In figure 2(a) the data correlation is presented in terms of local measured heat transfer coefficients divided by equation (2) plotted against $(U/\sqrt{gl})$. Circular symbols representing upflow and squares downflow are data from the first temperature measuring station downstream of the tube inlet. At this position ($l = 0.75$ in.), the vapor flow is still laminar like in nature. The data includes all the test runs over a range of inlet velocities from 4.5 to 12.5 ft/sec. The line drawn on the figure was obtained directly from Bromley's data (fig. 11) in reference 5 and represents the upflow data quite adequately. The equation of this line:

$$\frac{h}{h_c} = 0.14\left(\frac{U}{\sqrt{gl}}\right) + 0.6 \tag{3}$$

is presented herein as the correlation of the upflow data. An examination of the downflow data on figure 2(a) will show that the correlation overestimates the heat transfer by an average value of about 50 percent. The spread in the downflow data showed a consistent trend with wall temperature which made it possible to modify equation (2) with a temperature ratio parameter. The heat transfer relation for laminar vapor downflow follows:

$$h_{c,d} \approx \left[\frac{k_f^3(\rho_L - \rho_v)\rho_v g\lambda'}{l(\Delta T)\mu}\right]^{1/4}\left[1 - \left(\frac{T_w}{T_b}\right)^{-1/2}\right] \tag{4}$$

Figure 2(b) shows the correlation improvement using equation (4).

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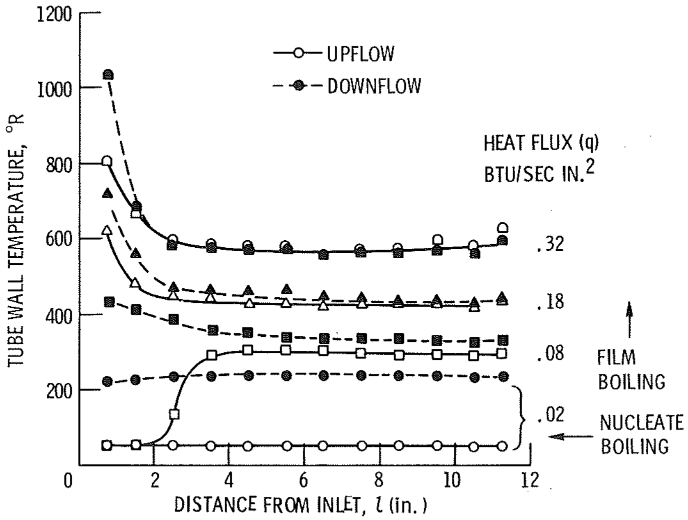

Figure 1. - Upflow and downflow temperature profiles for liquid hydrogen at saturated inlet conditions; bulk temperature, 43° R; pressure, 35 psia; flow diameter, 0.505 in.; fluid velocity, 9.6 ft/sec over a range of heat fluxes.

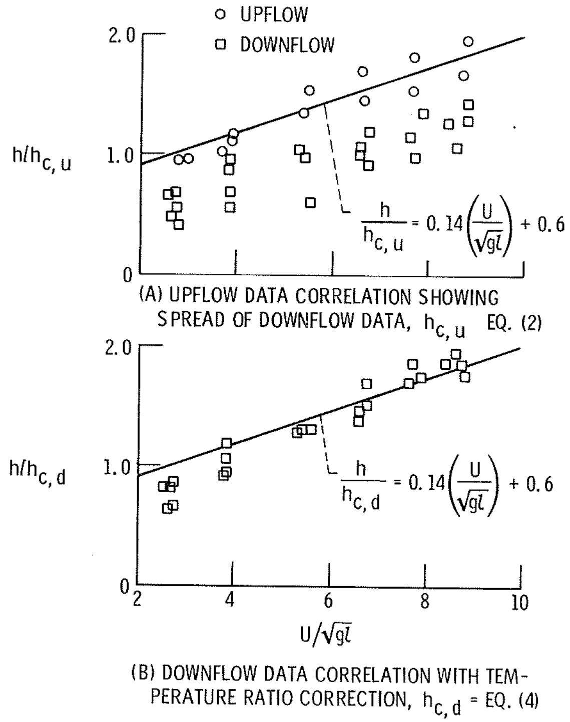

Figure 2. - Heat transfer correlations in inlet region for liquid hydrogen; saturated inlet; bulk temperature, 43° R; pressure, 35 psia; flow diameter, 0.505 in.; l = 0.75 in.