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FLOW BOILING WITH ENHANCEMENT DEVICES FOR COLD PLATE COOLANT CHANNEL DESIGN

FINAL REPORT
August 28, 1989

Submitted to the
National Aeronautics and Space Administration (NASA)
Lyndon B. Johnson Space Center

by
Ronald D. Boyd, Sr., Ph. D.
Department of Mechanical Engineering
P. O. Box 397
Prairie View A&M University
Prairie View, TX 77446
Phone: (409) 857-4811 or 4023

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Dr. Ronald D. Boyd, Sr., Professor
and Principal Investigator (PI)

FLOW BOILING WITH ENHANCEMENT DEVICES FOR COLD PLATE COOLANT CHANNEL DESIGN

Ronald D. Boyd (PI), Jerry C. Turknett, and
Alvin Smith (Graduate Students)
Department of Mechanical Engineering
P. O. Box 397
Prairie View A&M University
Prairie View, TX 77446

EXECUTIVE SUMMARY

A multi-year research program has been launched to study the effects of enhancement devices on flow boiling heat transfer in circular coolant channels, which are heated over a fraction of ther perimeters. The objectives of this year's work were to (a) examine the variations in both the mean and local (axial, and circumferential) heat transfer coefficients for a circular coolant channel with either smooth walls or with both a twisted tape and spiral finned walls, and (b) initiate improvements in the present data reduction analysis. This continued effort is intended to lead to the development of fundamentally-based heat transfer correlations which include effects of (a) single-side heat flux, and (b) enhancement device configuration. This overall program will set the stage for future study of heat transfer and pressure drop in single-side heated systems under zero-gravity conditions.

Local (axial and circumferential) measurements [1] of the outside wall temperature have been made for horizontal freon-11 flow through a 1.37 cm inside diameter coolant channel with smooth walls, which is heated from the top side. The exit pressure, inlet temperature, and mass velocity were 0.19 MPa, 24 °C, and 0.281 Mg/m^{2}s, respectively. A preliminary data reduction model was used to relate the measured wall temperatures to the local heat transfer coefficients. For the above flow conditions, the preliminary results show that although both axial and circumferential heat transfer coefficient (h) variations are important, h increased by more than 50% at the onset of nucleate boiling. A detail description of the work completed is contained in the recently completed thesis (by Jerry C. Turknett) [2], which was sent to both NASA Headquarters (Dr. Yvonne B. Freeman) and Johnson Space Center (Mr. Russ Long). Enclosed in this report is a summary of that work [3].

The data reduction approach was improved slightly by developing a relationship between local wall temperature measurements and the mean value of h. In cases where circumferentially-averaged values of h were obtained, four local circumferential wall temperature measurements were used to obtain

a mean value of h at each of seven axial measurement locations. In other cases, an approach was developed to utilize the twenty-eight local wall temperature measurements to obtain a single heat transfer coefficient, which was based on both circumferential and axial averaging. For the case involving circumferential averaging only, axial variations in the mean heat transfer coefficient were obtained for conditions of subcooled freon-11 flow boiling in a top-heated finned coolant channel with and without a twisted tape. For these configurations, the channel diameter, exit pressure, inlet temperature and mass velocity were 1.0 cm, 0.19 MPa, 22.2 °C, and 0.21 Mg/m^2s, respectively. The results from this work indicate that in addition to enhancing the heat transfer, enhancement devices can reduce the axial variations in h. This work was summarized in a second paper [4], which is enclosed and which was presented at the 1989 NASA/HBCU Science and Engineering Research Forum.

Additional work is planned [5] to (a) further refine the data reduction procedure for the single-side heated flow channel, (b) expand the experimental matrix, and (c) make comparisons of the experimental data with existing correlations. The expanded experimental matrix will include the effect of other parameters (subcooling, diameter, and wall configuration) on the heat transfer coefficient at various levels of pumping power.

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   Att: Mr. Robert Effinger
   P. O. Box 3578
   College Station, TX 77843

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   Department of Mechanical Engineering
   P. O. Box 397
   Prairie View A&M University
   Prairie View, TX 77446

Presented at the 1989 NASA/HBCU Research Forum

FORCED CONVECTION AND FLOW BOILING WITH AND WITHOUT ENHANCEMENT DEVICES FOR TOP-SIDE-HEATED HORIZONTAL CHANNELS

Ronald D. Boyd (PI), and Jerry C. Turknett (Graduate Student)
Department of Mechanical Engineering
Prairie View A&M University
Prairie View, TX 77446

ABSTRACT

A three-year research program has been launched to study the effect of enhancement devices on flow boiling heat transfer in coolant channels, which are heated either from the top side or uniformly. Although the study will include other orientations and working fluids in subsequent years, the first years will involve studies of the variations in the local (axial and circumferential) and mean heat transfer coefficients in horizontal, top-heated coolant channels with smooth walls and internal heat transfer enhancement devices. Initially, the working fluid will be freon-11. The objectives of this fiscal year's work are to: (1) examine the variations in both the mean and local (axial, and circumferential) heat transfer coefficients for a circular coolant channel with either smooth walls or with both a twisted tape and spiral finned walls, (2) examine the effect of channel diameter (and the length-to-diameter aspect ratio) variations for the smooth wall channel, and (3) develop an improved data reduction analysis.

The case of the top-heated, horizontal flow channel with smooth walls (1.37 cm inside diameter, and 122 cm heated length) has been completed. The data has been reduced using a preliminary analysis based on the heated hydraulic diameter. Preliminary examination of the local heat transfer coefficient variations indicates that there are significant axial and circumferential variations in the local heat transfer coefficient. However, it appears that the circumferential variation is more significant than the axial ones. In some cases, the circumferential variations were as much as a factor of ten. The axial variations rarely exceeded a factor of three. Integrated averaged heat transfer coefficients will be obtained after the improved data reduction model has been implemented.

NOMENCLATURE

h	Local heat transfer coefficient, W/m^2 °C
h_∞	Mean heat transfer coefficient due to natural convection between the test section and the ambient, W/m^2 °C
q_c	Heat loss from the test section due to convection, W
q_R	Heat loss from the test section due to radiation, W
r	Radial coordinate for the data reduction model, m

T_f Bulk temperature of the flowing fluid, °C
T_m Local measured outside wall temperature of the test section, °C
Tsat Saturation temperature (316 K at 0.19 MPa for freon-11), °C
T_∞ Ambient temperature, °C
Z, Z_i Axial coordinate for heated portion of the test section; in Figures 3a through 3d, Zi represents Z_i, where Z_i=20.32(i-1), cm

Greek Letters

ϕ Circumferential coordinate; see Figure 1a for the datum. In Figures 3a through 3d, ϕ is also referred to as "Phi."
π Half of a full rotation or 180°; in Figures 3a through 3d, π is also referred to as "Pi."

INTRODUCTION

Space commercialization will require efficient heat transfer systems. The future success of many efforts will be based on our understanding of the behavior of two-phase flow boiling in both the space (zero-g or reduced-g) and earth environments. This three-year program is intended to focus on the following fundamental characteristics (e.g., nonuniform heat flux distribution, Marangoni effects, and single and double enhancement devices)of experimental flow boiling heat transfer and pressure drop in the earth environment [1].

The objectives of the first year's efforts are to: (1) examine the variations in both the mean and local (axial, and circumferential) heat transfer coefficients for a circular coolant channel with either smooth walls or with both a twisted tape and spiral finned walls, (2) examine the effect of channel diameter (length-to-diameter ratio) variations for the smooth wall channel, and (3) develop an improved data reduction analysis.

In this paper, we report on forced convection and flow boiling of freon-11 in a smooth-wall horizontal coolant channel (1.37 cm inside diameter, and 122.0 cm heated length) heated from the top side. The inlet freon temperature was 24°C, the exit pressure was 0.19 MPa absolute, and the mass velocity was 0.28 Mg/m^2s.

EXPERIMENTAL INVESTIGATION

The reader is referred to references [1 and 2] for detail descriptions of the experimental flow loop, procedures, and data acquisition. Figure 1a is a schematic of the cross section of the heated portion of the test section [which is preceded by an

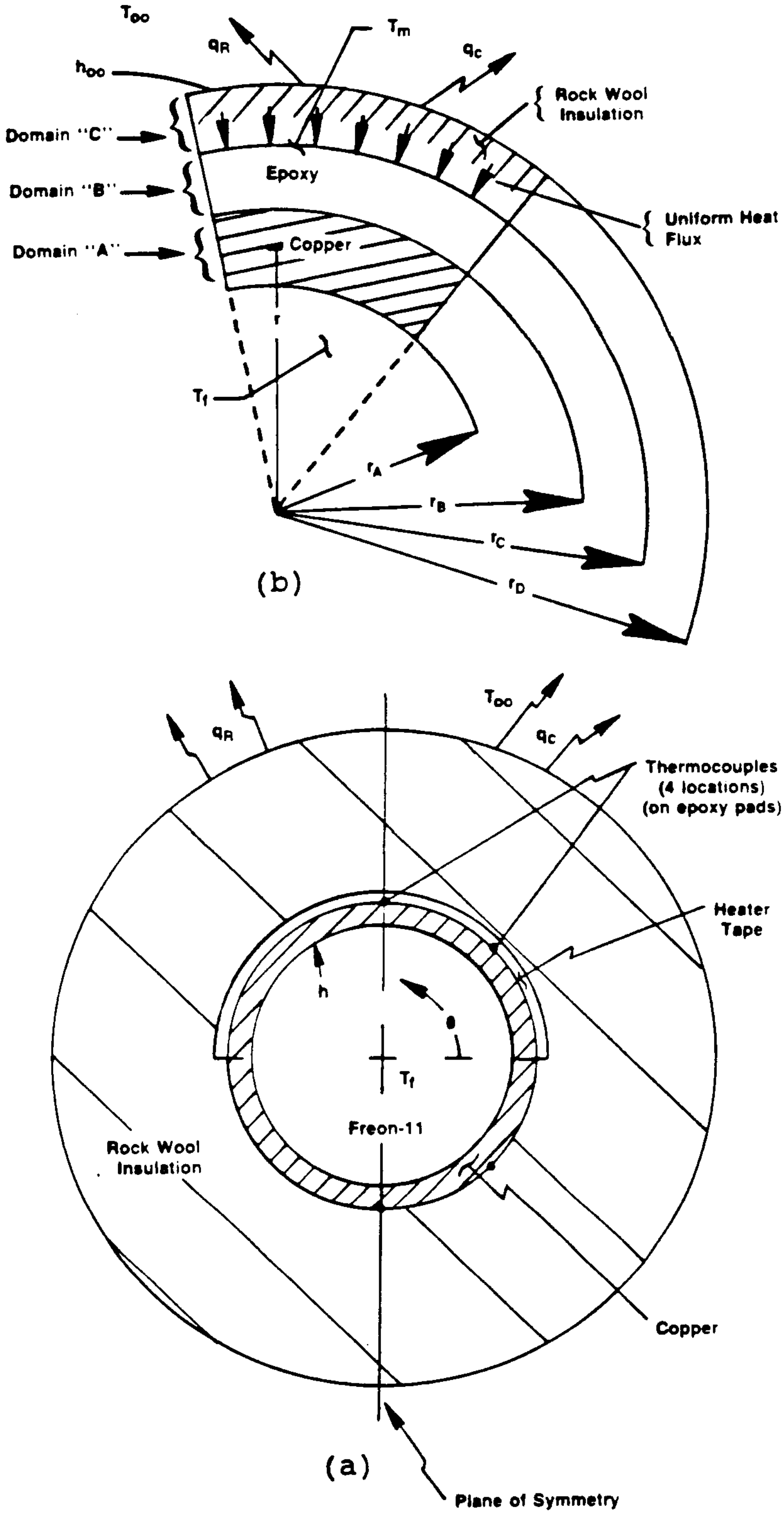

Figure 1: (a) Cross Section of the Heated Portion of the Test Section

(b) Control Volume of Heated Hydraulic Diameter Model upstream unheated portion for flow development]. The measured wall temperatures are used along with the data reduction analysis to determine the unknown heat transfer coefficient, h. Recently, a data reduction technique based on the heated hydraulic diameter [2] (see Figure 1b) was used to reduce the experimental data. This approach will result in, at most, a qualitative indication of the local distribution of h. Work is proceeding on more viable data reduction approaches; e.g., finite difference for local h and analytical for mean h.

In applying either model, knowledge of the fluid's bulk temperature must be used. An iteration scheme is used to compute the inside wall temperature, T_w, of the flow channel. The fluid's temperature is chosen based on the magnitude of the inside wall temperature relative to the wall temperature required to cause the onset to nuclueate boiling (T_{ONB}). If T_w is greater than than T_{ONB}, the fluid temperature is set equal to the saturation temperature. However, if the above condition is not satisfied, the fluid temperature is computed from the energy equation, using the measured inlet fluid temperature and the measured net thermal energy transfer to the fluid.

RESULTS

The results are presented for the case of forced convection and flow boiling with freon-11 flowing in a horizontal channel with smooth walls.

The variation in the heat transfer coefficient is more pronounced in the circumferential direction than the axial direction. The heat transfer coefficients, for the single-phase and hypocritical regions are shown in Figure 2. The fully developed nucleate boiling region does not appear in the figure but occurs somewhere in a narrow range between $\phi = 7\pi/4$ and $\pi/$ (see Figure 1a for the datum for ϕ). It is expected that the heat transfer coefficient in this narrow region to be greater than any of those shown.

The last two highest points on each of the curves show an increase in the heat transfer coefficient. For $\phi = 3\pi/2$ and $7\pi/4$, the increase is due to the onset of nucleate boiling which we see results in more than a fifty percent increase in h. For these two locations we see that there is only a secondary variation in h with ϕ.

Both axial and circumferential variations in h are found to be significant. Comparisons of Figures 3a ($\phi = \pi/2$), 3b ($\phi = \pi/4$), 3c ($\phi = 7\pi/4$) and 3d ($\phi = 3\pi/2$) reveal the complex nature of the variations. The variation in the local heat transfer coefficient increases from the bottom ($\phi = 3\pi/2$) to the top ($\phi = \pi/2$) of the test section at all axial locations. As noted earlier, the variation between $7\pi/4$ and $3\pi/2$ is small even at the locations where incipient nucleate boiling occurs (h = 1400 W/m^2K). It is interesting to compare the magnitude of h for the three regimes: (1) single-phase (800 W/m^2K), (2) incipient nucleate boiling (1400 W/m²K) and, (3) film boiling (10 to 100 W/m²K).

If one takes time to study the relative positions of the curves and use the reduced wall temperature, some of the character of the flow is revealed (e.g., See Figure 3a). In particular, notice from Figure 3a that: (1) h at Z_5 and Z_3 are almost identical at between 380 and 640 W, (2) h at Z_6 is much higher than all values of h at other locations, and (3) in some cases curves are crossing one another. These observations may imply a slug type flow. For example, at Z_6 the unusually large value of the heat transfer coefficient could be due to local cooling (slug flow). Contrasting the above description, Figure 3b ($\phi = \pi/4$) shows that between Z_2 and Z_5 the heat transfer coefficient decreases in the downstream direction. This is consistent with the previous observations made. That is, at $\phi = \pi/4$ the film boiling regime predominates.

CONCLUSIONS

Local (axial and circumferential) measurements of the outside wall temperature have been made for horizontal freon-11 flow (0.19 MPa exit pressure and 24°C inlet temperature) through a 1.37 cm inside diameter coolant channel with smooth walls and heated from

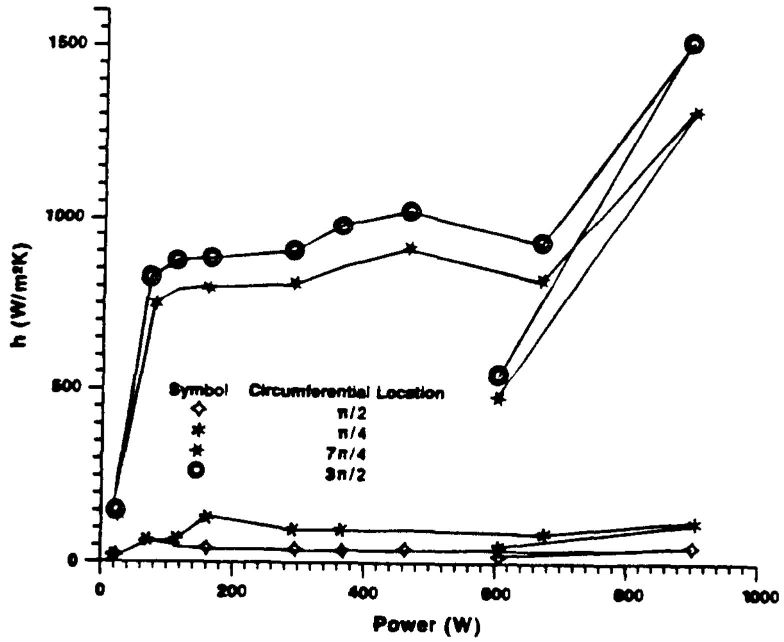

Figure 2: Heat Transfer Coefficient Versus Power Generation and Circumferential Location at Z = Z4 = 61.96 cm (center of the test section) for Top-Side-Heated Smooth Tubes for: 0.19 MPa Exit Pressure, 0.281 Mg/m^{2}s mass velocity, 1.22 m Heated Length.

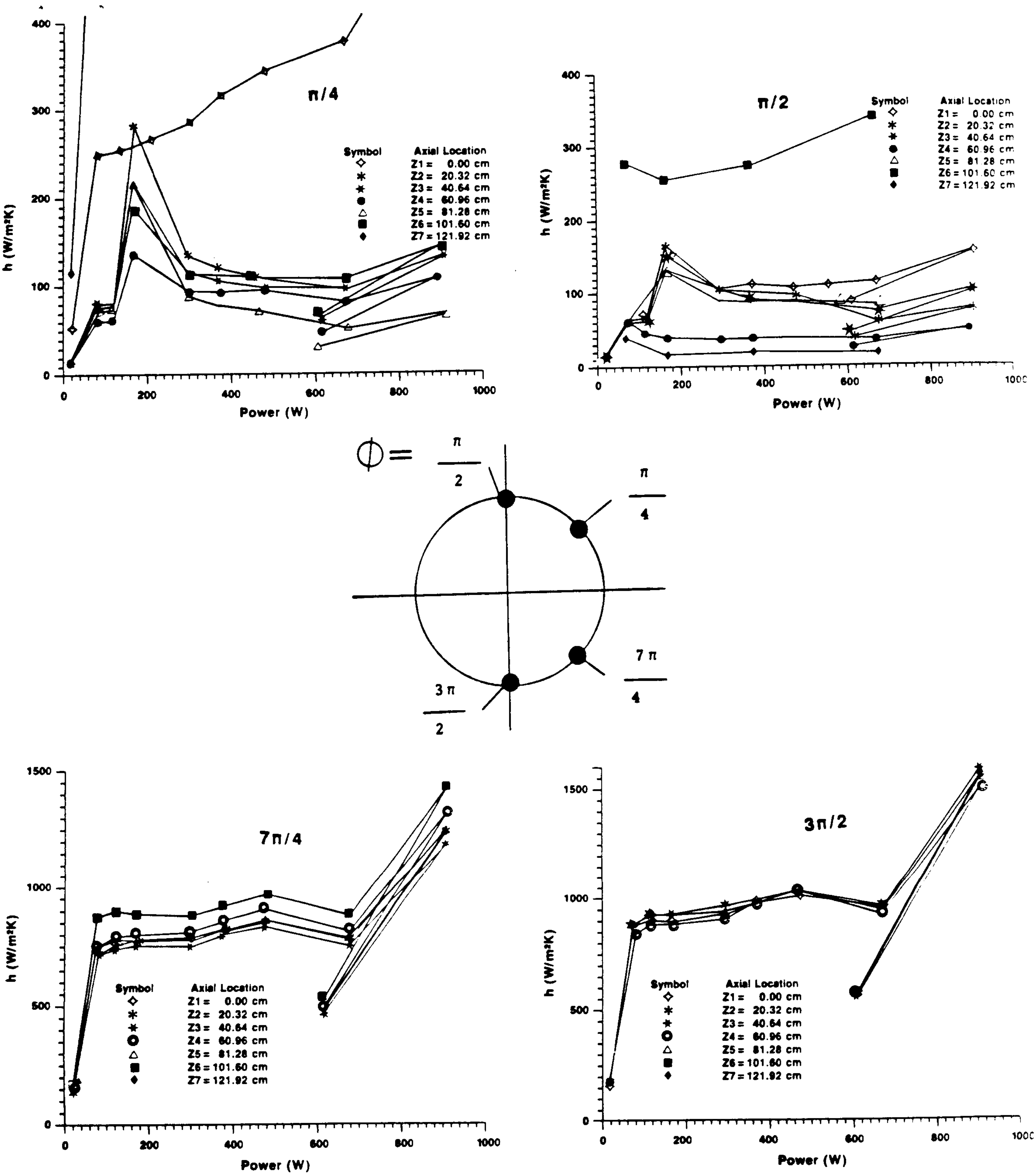

Figure 3: Heat Transfer Coefficient Versus Power And Axial Location at Phi = Pi/2(top), Pi/4, 7Pi/4, and 3Pi/2(bottom) for Top-Side Heated Smooth Tubes at 0.19 MPa and 0.281 Mg/m^2s. Channel Diameter = 1.27cm, Heated Length = 1.22m.

the top side. A preliminary data reduction model was used to relate the measured wall temperatures to the local heat transfer coefficient. Although the local temperature measurements are quantitative, the preliminary data reduction model results in what may be only qualitative local heat transfer measurements. Work is proceeding to evaluate and improve, if necessary, the existing data reduction model.

The preliminary heat transfer data indicates that there are significant axial and circumferential variations in the local heat transfer coefficient. However, it appears that the circumferential variation is more significant than the axial ones. The single phase heat transfer coefficient (near 900 W/m^2K) is increased by more than 50% at the onset of nucleate boiling. For the test performed, the circumferential heat transfer coefficient varied from the hypocritical to the single phase heat transfer regimes. This resulted [in some cases] in a factor of ten increase in the local heat transfer coefficient. The axial variations rarely exceeded a factor of three.

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