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FLOW BOILING WITH
ENHANCEMENT DEVICES FOR
COLD PLATE COOLANT CHANNEL DESIGN

SEMIANNUAL REPORT
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Ronald D. Boyd, Sr., Ph.D.
Department of Mechanical Engineering
College of Engineering
Prairie View A&M University
Prairie View, TX 77446

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PRAIRIE VIEW A&M UNIVERSITY
COLLEGE OF ENGINEERING

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Dr. Ronald D. Boyd, Sr., PI
and Professor

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DEVICES FOR COLD PLATE COOLANT CHANNEL DESIGN

Ronald D. Boyd (PI), Jerry C. Turknett,and
Alvin Smith (Graduate Students)
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.

This effort is intended to lead to the development of fundamentally-based heat transfer correlations which include effects of: (1) complex heat flux distributions, (2) enhancement

device configuration, and (3) basic flow parameters. This overall effort is intended to set the stage for the study of heat transfer and pressure drop in single-side heated systems under zero-gravity conditions.

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. Integrated averaged heat transfer coefficients will be obtained after the improved data reduction model has been implemented. In particular, this work will result in the inclusion of effects associated with circumferential variations in the effective heat flux at the inside wall.

The experimental progress has been interrupted over the last two months because the entire flow loop had to be moved from the "Old Engineering Building" (which is now being renovated) to the Agriculture Research Center. Testing is scheduled to resume in about one week.

Two graduate students (Jerry C. Turknett and Alvin Smith) will give presentations, based on the ongoing work, at the NASA-HBCU Space Science and Engineering Research Forum, in Huntsville (AL) on March 23, 1989.

CONTENTS

NOMENCLATURE

h   Local heat transfer coefficient, $W/m^2\ {}^\circ C$

h_∞ Mean heat transfer due to natural convection between the test section and the ambient, $W/m^2\ {}^\circ C$

q_c Heat loss from the test section due to convection, W/m^2

q_R Heat loss from the test section due to radiation, W/m^2

r   Radial coordinate for the data reduction model, m

T_r Bulk temperature of the flowing fluid, ${}^\circ C$

T_m Local measured outside wall temperature of the test section, ${}^\circ C$

Tsat   Saturation temperature (316 K at 0.19 MPa for freon-11), ${}^\circ C$

T_∞ Ambient temperature, ${}^\circ C$

Z, Z_i Axial coordinate for heated portion of the test section; in Figures 7 through 10, Zi represents Z_i, where $Z_i = 20.32(i-1)$, cm

Greek Letters

ϕ Circumferential coordinate; see Figure 4 for the datum. In Figures 7 through 10, ϕ is also referred to as "Phi."

π Half of a full rotation or 180°; in Figures 7 through 10 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 of experimental flow boiling heat transfer and pressure drop in the earth environment: (1) non-uniform heat flux distribution, (2) resulting local distributions of the heat transfer coefficient, (3) pressure drop and pumping power, (4) single and double enhancement devices, (5) the relative advantages of saturated and subcooled flow boiling regimes, (6) flow channel aspect ratio effects, (7) the relative effects of heat transfer enhancement techniques, and (8) correlations for mean (and local) heat transfer and pressure. Future multi-year research efforts, which will be applicable to both the earth and zero-g environments, will include basic phenomena such as: (1) the effect of orientation and Marangoni effects, (2) flow stability, and (3) identification of the threshold inertia (Froude and modified Froude numbers). Threshold inertia determination is necessary to identify when orientation (earth or reduced-g) and/or Marangoni (at zero-g) effects become important. Although not apparent, the development of improved data reduction models are essential to the accurate representation and interpretation of the heat transfer data.

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The objective 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 (and length-to-diameter aspect ratio) variations for the smooth wall channel, and (3) develop an improved data reduction analysis.

Thus far, freon-11 flow boiling experiments have been completed for the case of 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-11 temperature was 24°C, the absolute exit pressure was 0.19 MPa, and the mass velocity was 0.28 Mg/m²s. Work is proceeding on: (1) heat transfer enhancements due to internal devices, (2) the examination of the effect of diameter for smooth wall channels, and (3) the development of an improved data reduction technique.

Some progress has been made to examine whether improvements are needed in the present data reduction analyses. Four models have been developed and formulated (e.g., [1]), including a preliminary finite difference model. Since all equations for each model must be verified before being used to reduce data, work is proceeding to "bench-mark" the developed models.
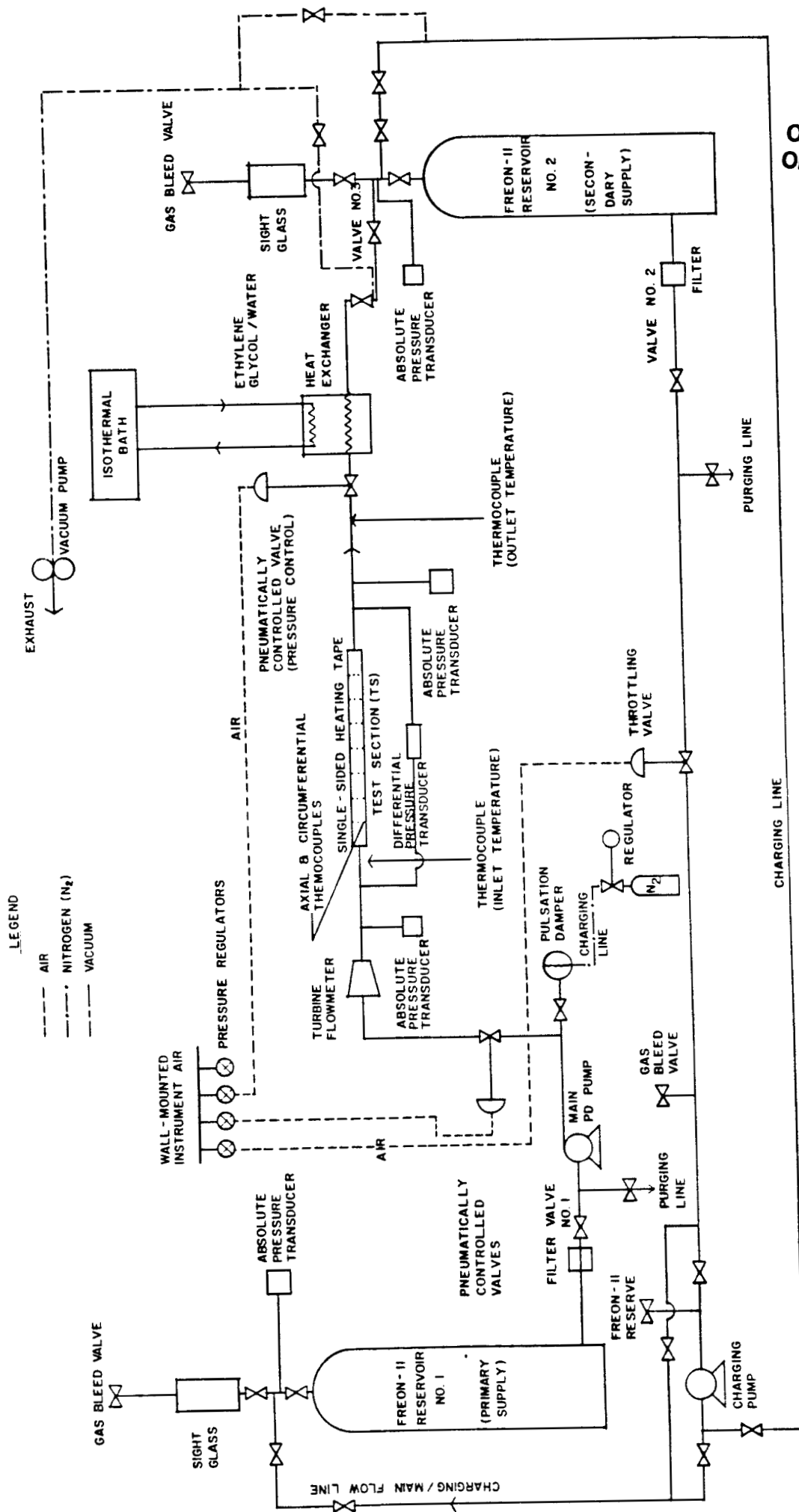
Two graduate students (Jerry C. Turknett and Alvin Smith) will give presentations at the NASA-HBCU Space Science and Engineering Research Forum, in Huntsville (AL) on March 23, 1989.

LITERATURE SURVEY

A survey of the literature is ongoing. Brief surveys of the related literature can be found in references [1 and 2].

EXPERIMENTAL INVESTIGATION

The reader is referred to references [1 and 2] for detail descriptions of the experimental flow loop, procedures, and data acquisition. Figures 1 and 2 show both the flow loop and test section configurations, respectively. Figure 3 displays the heated portion of the test section at a stage of the assembly where the axial and circumferential placements of the thermocouples (wires hanging below the horizontal tube), as well as the top-mounted heater tape, are clearly seen. The thermocouples are used to make temperature measurements of the outside wall of the heated coolant channel. Figure 4 is a schematic of the cross section of the heated portion of the test section [which is preceded by an 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 5), 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 approaches; e.g., finite difference for local h and analytical for mean h.

Figure 1: Freon-11 Flow Loop for Both Subcooled
and Saturated Flow Boiling Experiments.

223.5 CM

101.6 CM

UNHEATED FLOW DEVELOPING SECTION

HEATED SECTION

$\phi = 0$
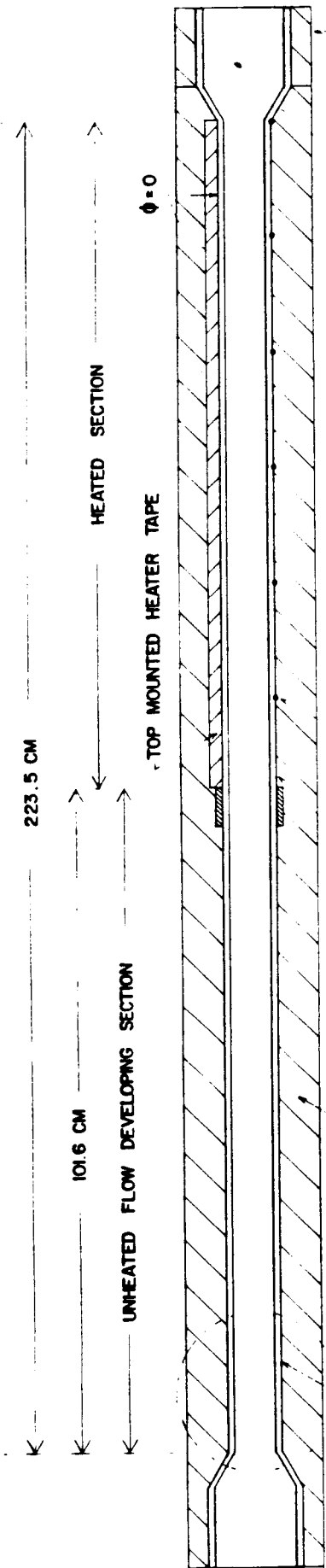
TOP MOUNTED HEATER TAPE

THERMOCOUPLES MOUNTED CIRCUMFERENTIALLY = $\pi/4$, $\pi/2$, $3\pi/2$ AND $7\pi/4$ AND AT SIX AXIAL LOCATIONS AT 20.32 CM INTERVALS

REMOVABLE INSULATION WHEN CHANGING TEST SECTION

DOWNSTREAM PART OF SECTION WITH OR WITHOUT ENHANCEMENT DEVICE

INSULATION

SMOOTH TUBE

Figure 2: Horizontal Test Section for Measuring Local Heat Transfer Coefficient Distributions.

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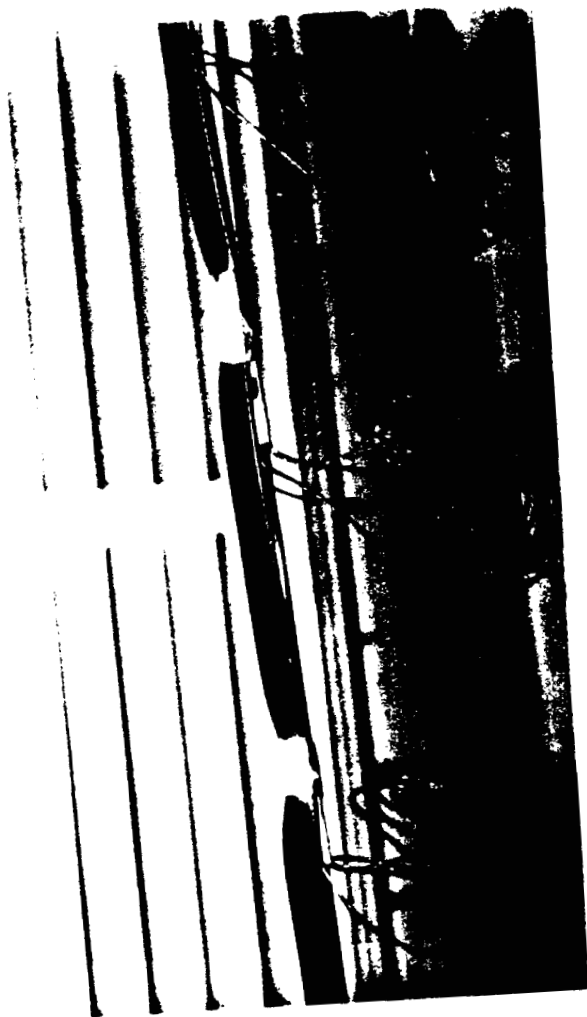
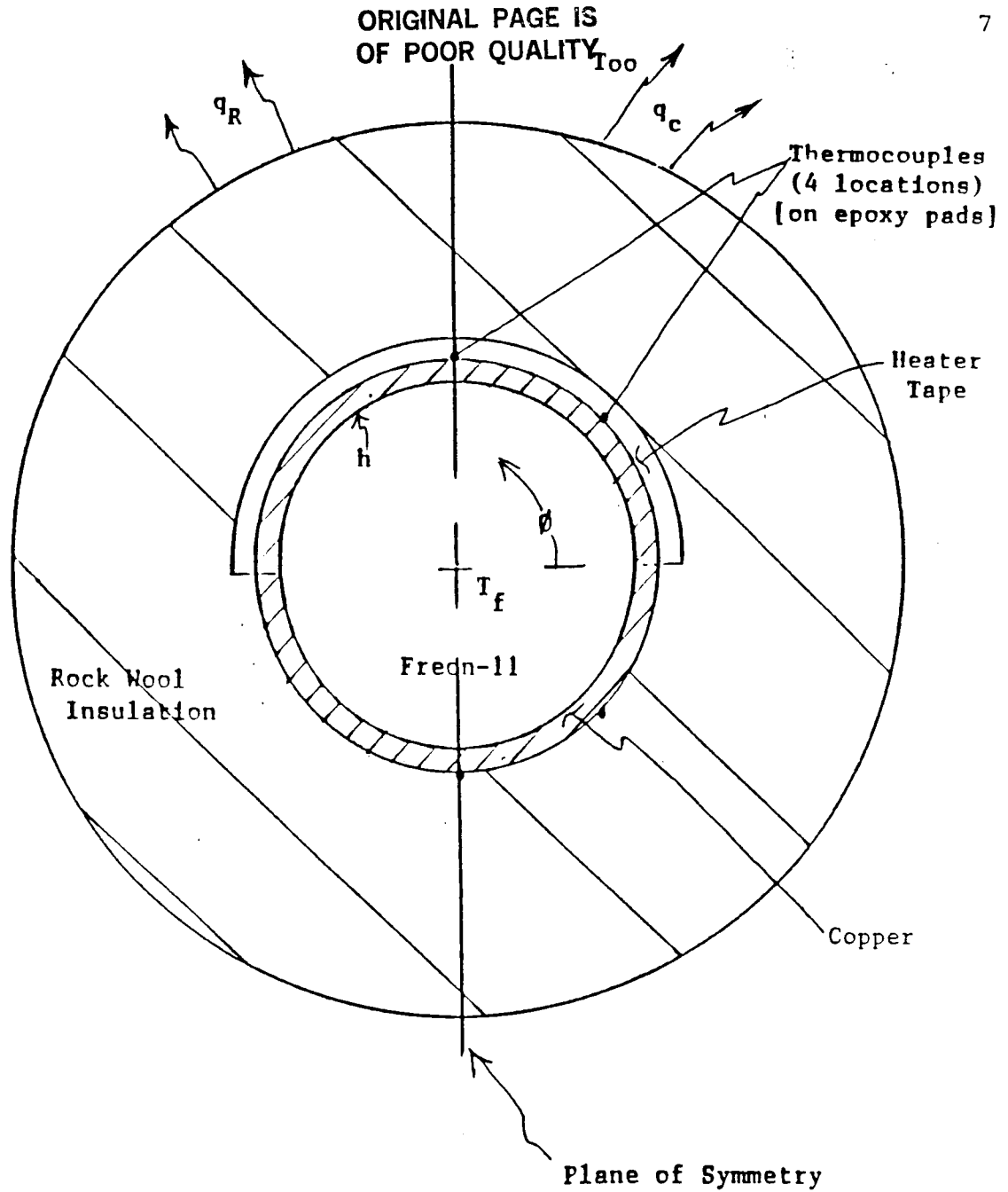Figure 3: Thermocouple and Heater Tape Installation.

Figure 4: Cross Section of the Heated Portion
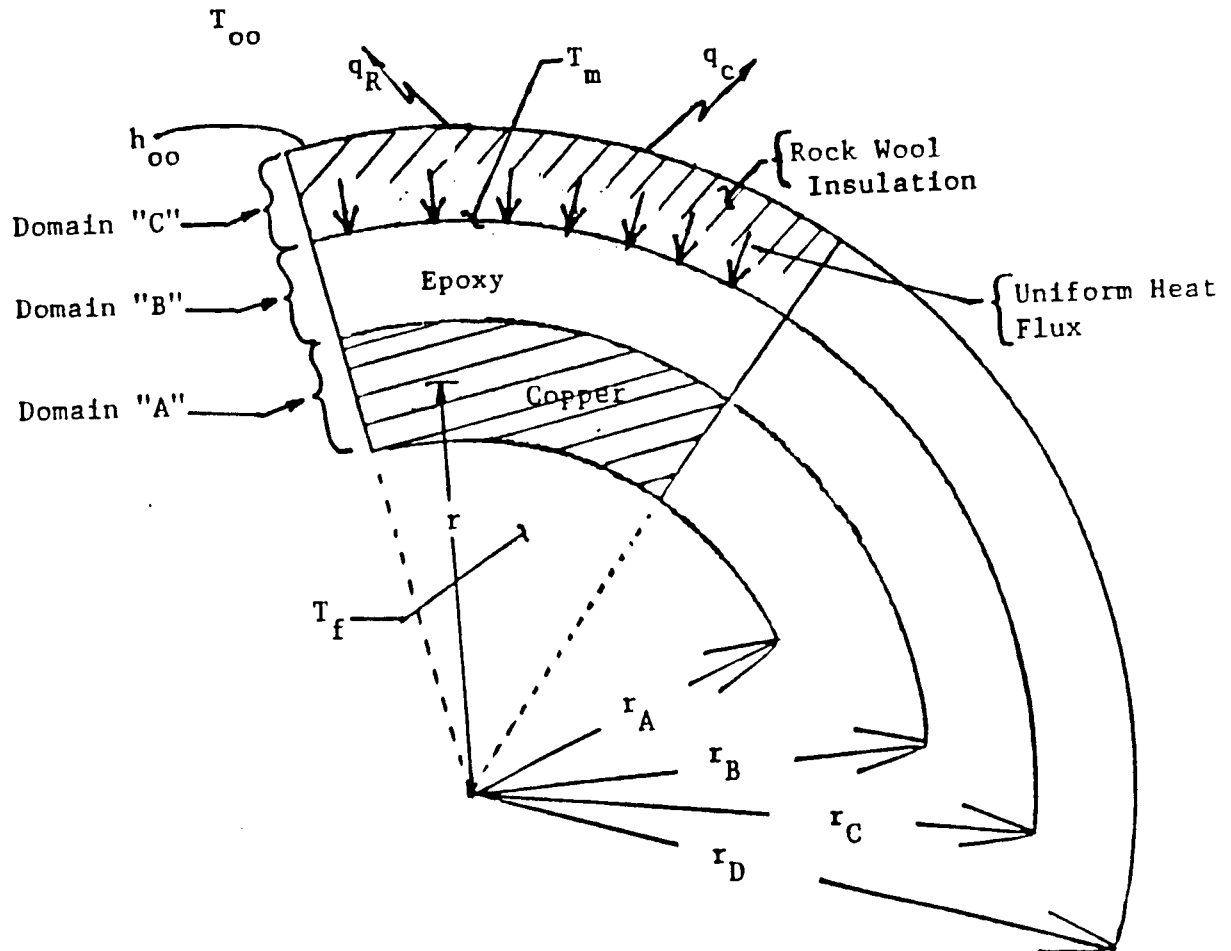of the Test Section.

Figure 5: Control Volume for the Heated Hydraulic Diameter Model.

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 a horizontal flow channel with smooth walls (1.37 cm inside diameter, and 122 cm heated length) which is heated from the top side. The flow rate and exit pressure of the freon-11 were maintained at 0.281 Mg/m²s and 0.19 MPa (316 K saturation temperature), respectively. The inlet freon temperature was held constant at 24.0°C.

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 [3] are shown in Figure 6. 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 $\pi/4$
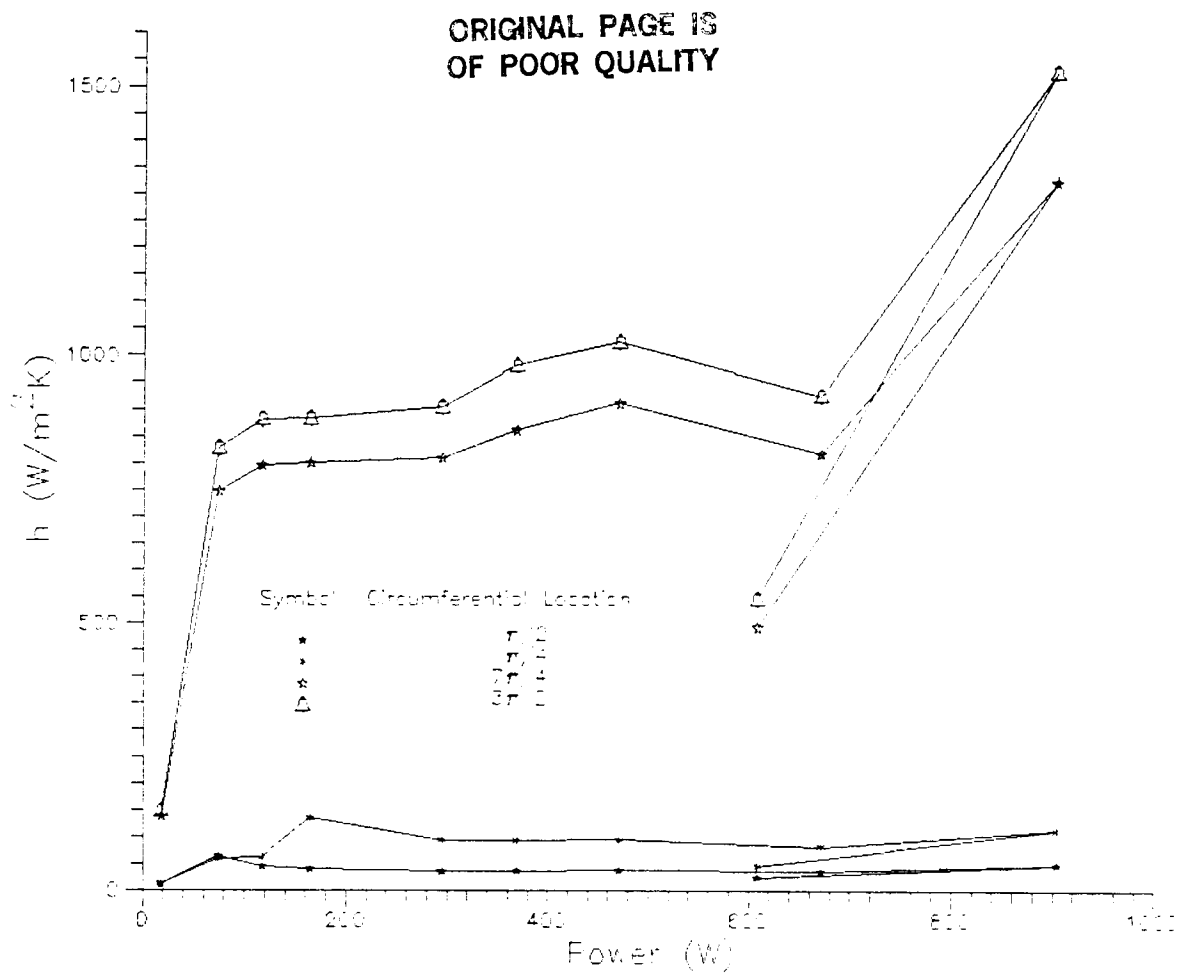
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Figure 6: Heat Transfer Coefficient Versus Power
Generation and Circumferential Location at
Z = Z4 = 61.96 cm (center of the test section)
for Top-Side-Heated Smooth Tubes for: 0.19 MPa
Exit Pressure, 0.281 Mg/m^2s mass velocity,
1.22 m Heated Length.

(see Figure 4 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. It should be noted that the catastrophic drop in

h for each of the curves is due to the heat tape becoming

overheated and as a result eventually malfunctioning.

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. In the hypocritical region, the increase in h appears to

be very minor. However, the actual increase in h is much larger

than that shown on the figure because Tsat was used for T_f rather

than its actual value. Hence, if further work is conducted in the

hypocritical regime, better estimates of the actual (local) fluid

temperature must be made.

Both axial and circumferential variations in h are found to

be significant. Comparisons of Figures 7 ($\phi = \pi/2$), 8 ($\phi = \pi/4$),

9 ($\phi = 7\pi/4$) and 10 ($\phi = 3\pi/2$) 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^{2}K). It is

interesting to compare the magnitude of h for the three regimes:
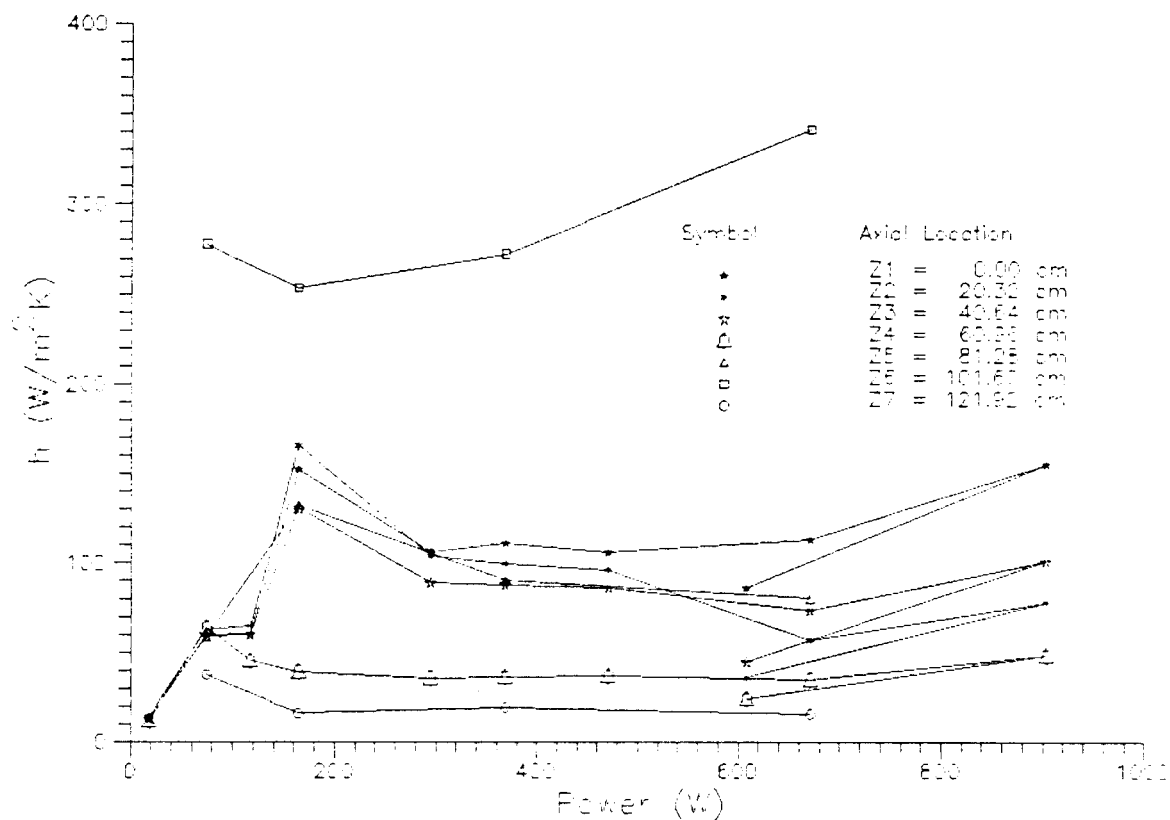
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Figure 7:   Heat Transfer Coefficient Versus Power
Generation and Axial Location at Phi = Pi/2 (top of
the test section) for Top-Side-Heated Smooth Tubes
for: 0.19 MPa Exit Pressure, 0.281 Mg/m^{2}s
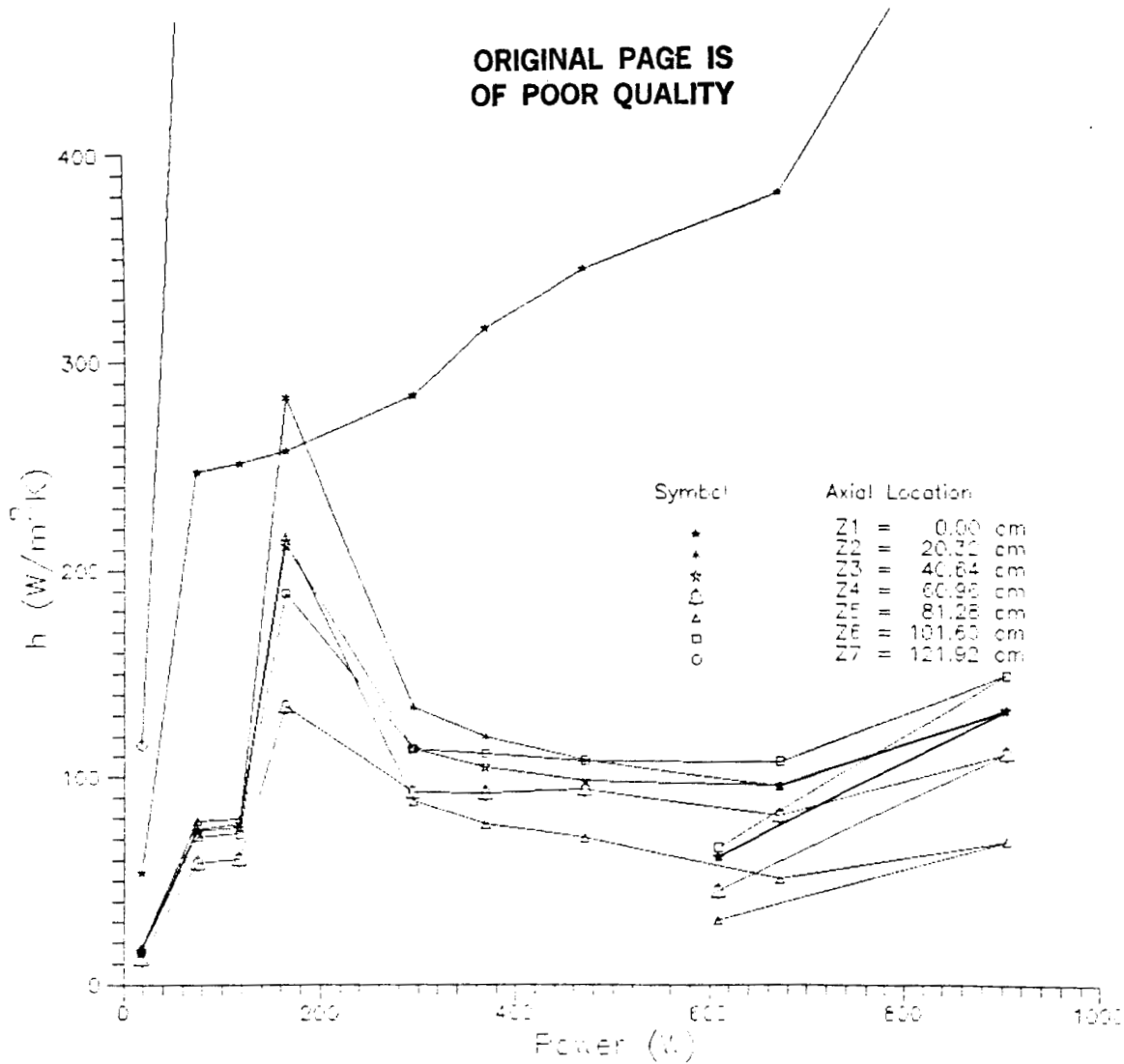mass velocity, 1.37 cm Channel inside Diameter,
and 1.22 m Heated Length.

Figure 8: Heat Transfer Coefficient Versus Power Generation and Axial Location at Phi = Pi/4 for Top-Side-Heated Smooth Tubes for: 0.19 MPa Exit Pressure, 0.281 Mg/m^2 mass velocity, 1.37 cm Channel inside Diameter, and 1.22 m Heated Length.
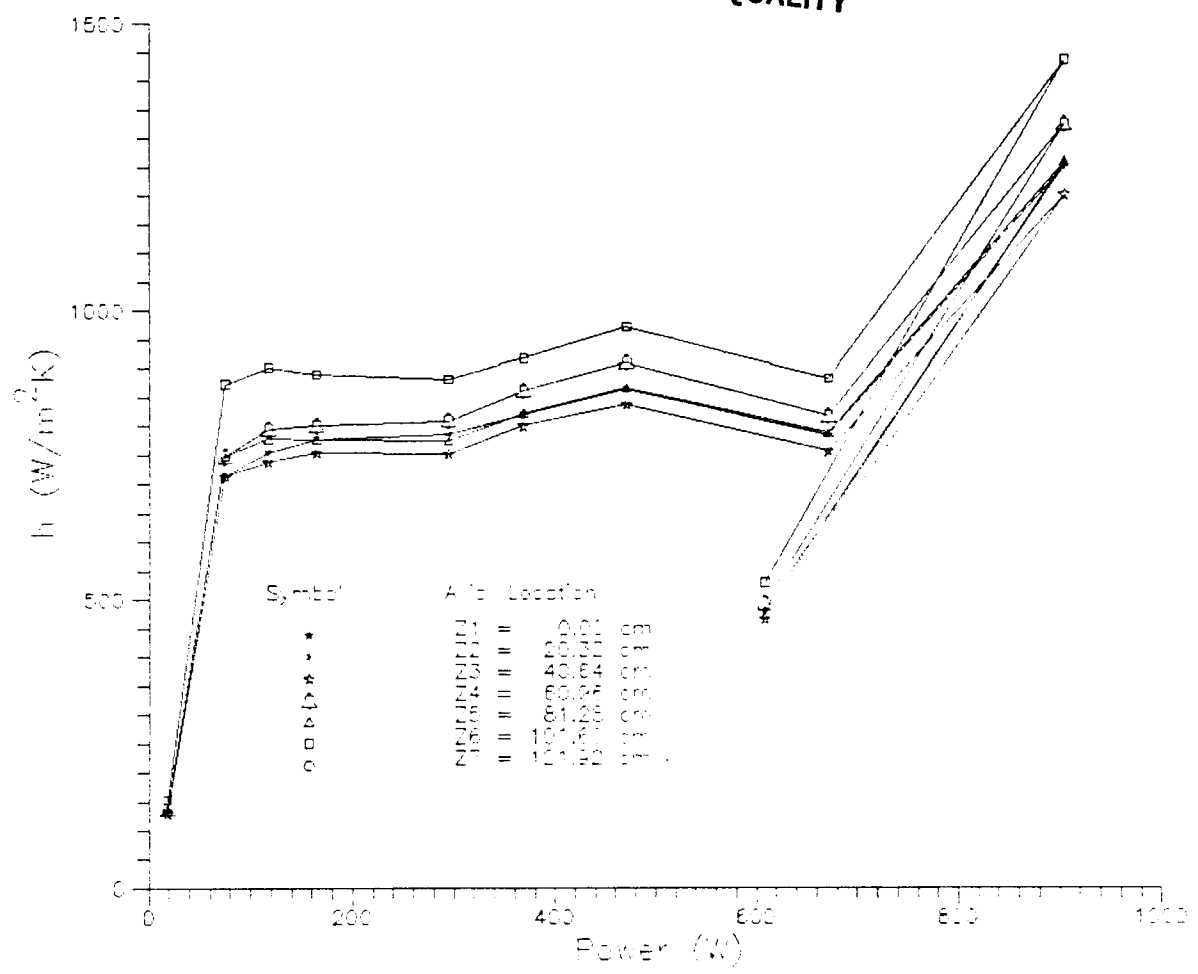
Figure 9: Heat Transfer Coefficient Versus Power
Generation and Axial Location at Phi = 7Pi/4 for
Top-Side-Heated Smooth Tubes for: 0.19 MPa Exit
Pressure, 0.281 Mg/m^{2}s mass velocity, 1.37 cm
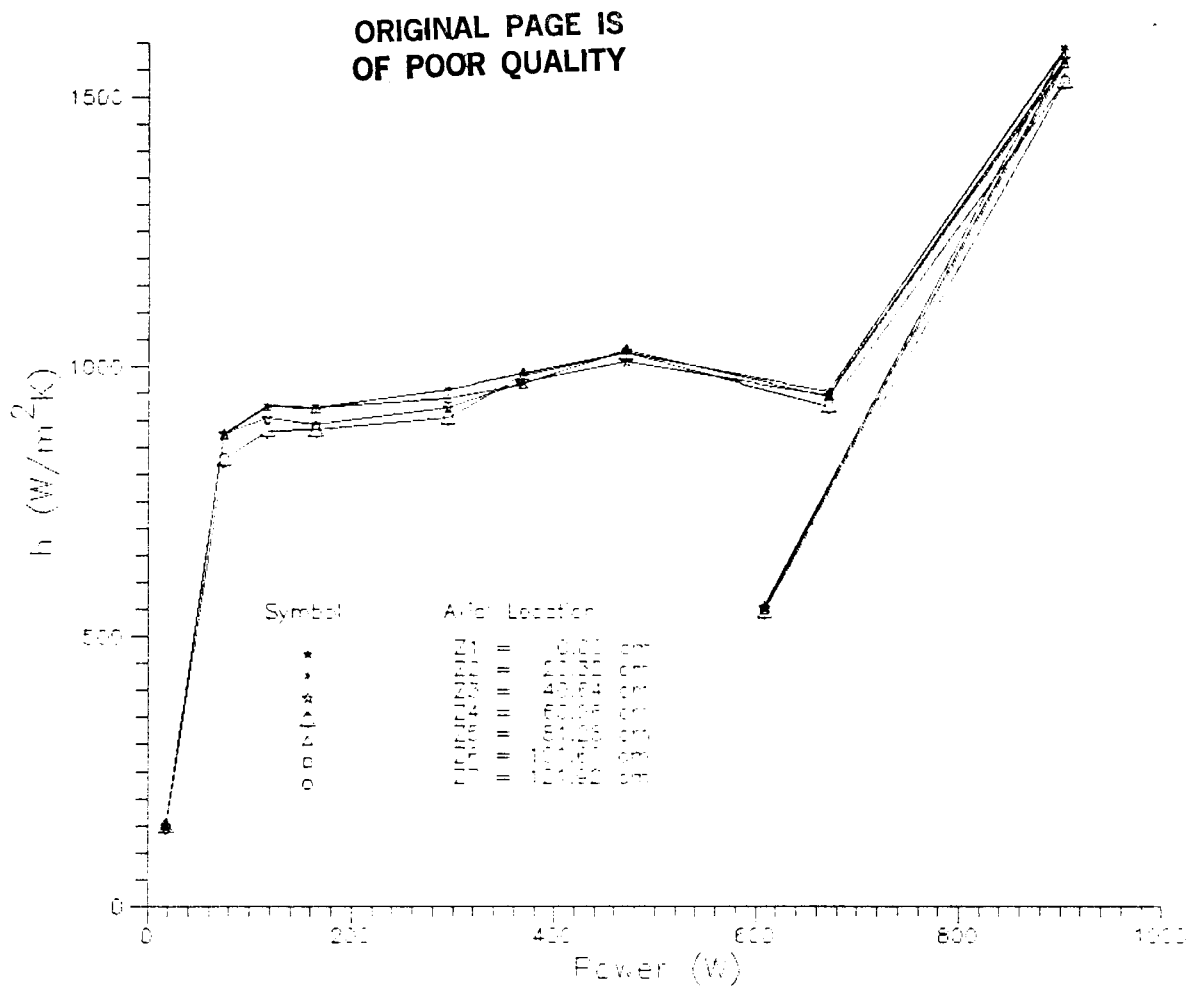Channel inside Diameter, and 1.22 m Heated Length.

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Figure 10: Heat Transfer Coefficient Versus Power
Generation and Axial Location at Phi = 3Pi/2 (bottom
of the test section) for Top-Side-Heated Smooth
Tubes for: 0.19 MPa Exit Pressure, 0.281 Mg/m^{2}s
mass velocity, 1.37 cm Channel inside Diameter, and
1.22 m Heated Length.

(1) single-phase (800 W/m²K), (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 7). In particular,

notice from Figure 7 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 8 ($\phi = \pi/4$) 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 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/m²K) 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.

ACKNOWLEDGMENTS

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Mr. Richard Parish (3)
NASA (JSC)
Mail Code FC-2
Houston, TX 77058

NASA Scientific & Information Facilities (2)
P. O. Box 8757
Baltimore-Washington Airport
Baltimore, MD 21240

Prairie View A&M Research Fountation
P. O. Box 3578
Attention: Robert Effinger

Dr. Wayne D. Perry, Dean
College of Engineering
Prairie View A&M University
Prairie View, TX 77446

Dr. Shield Lin, Chairman
Department of Mechanical Engineering
Prairie View A&M University
Prairie View, TX 77446

Dr. Ronald D. Boyd, Sr.
Department of Mechanical Engineering
Prairie View A&M University
Prairie View, TX 77446