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LINE-FOCUS SOLAR CENTRAL POWER SYSTEM – PHASE 1

Subsystem Experiment: Receiver Heat Transfer

MASTER

By
Arthur J. Slemmons

April 1980

Work Performed Under Contract No. EY-76-C-03-0115-140

SRI International
Menlo Park, California

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Solar Energy

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**LINE-FOCUS SOLAR CENTRAL POWER SYSTEM—
PHASE I**

**SUBSYSTEM EXPERIMENT:
RECEIVER HEAT TRANSFER**

ARTHUR J. SLEMMONS

APRIL 1980

**SRI INTERNATIONAL
333 RAVENSWOOD AVENUE
MENLO PARK, CALIFORNIA 94025**

**PREPARED FOR THE
U.S. DEPARTMENT OF ENERGY
SOLAR ENERGY
UNDER CONTRACT EY-76-C-03-0115**

SUMMARY

Wind-tunnel tests confirmed that heat losses due to natural convection are negligible in the line-focus, solar-power receiver. Anomalies in the forced-convection tests prevented definitive conclusions regarding the more important forced convection.

Flow-visualization tests using a water table show much lower velocities inside the receiver cavity than outside, supporting the supposition that the forced-heat transfer should be less than that from a standard exposed cylinder. Furthermore, the water-table tests showed ways to decrease the low velocities in the cavity should this be desired.

Further wind-tunnel testing should be done to confirm estimates and to support advanced design. This testing can be done in standard wind tunnels since only the forced convection is of concern.

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I INTRODUCTION

Accurate calculations of solar receiver performance depend upon knowledge of the various energy losses. Of these, the convective heat loss is the most difficult to anticipate because

- Existing correlations do not go up to the high Reynolds and Grashoff numbers anticipated.
- The particular geometry had not been modeled or analyzed.

These difficulties apply not only to our design but also to the central-tower design.

Our approach has been to use engineering judgement to estimate the losses, and then to run tests to verify the estimates and to guide future design.

II MATHEMATICAL MODEL OF CONVECTION

Estimates of the heat-transfer coefficient for free convection were based on the worst case of a flat plate the size of the aperture. According to Holman,¹ this worst case, in which the plate faces upward, has a heat-transfer coefficient

$$h = 0.22 (\Delta T)^{1/3} \text{ Btu/hrft}^2 \text{ } ^\circ\text{F}$$

When an average temperature difference of 700°F is assumed, we have

$$h = 1.95 \text{ Btu/hrft}^2 \text{ } ^\circ\text{F} \quad (11.06 \text{ W/m}^2 \text{ } ^\circ\text{C})$$

When a correction is made for the difference between the area of a plate and the interior area of the receiver, and a factor of 1/2 is applied to reduce the degree of conservatism, the estimated free-convection, heat-transfer coefficient is

$$h = 0.276 \text{ Btu/hrft}^2 \text{ } ^\circ\text{F} \quad (1.56 \text{ W/m}^2 \text{ } ^\circ\text{C})$$

The estimate for forced convection was based upon an assumption of crossover from free to forced convection at a wind speed of 5 mph (1.708 m/s), and the use of the Holman exponent on the Reynolds number of 0.731. Thus, the convective heat-transfer coefficient was estimated to be

$$\begin{aligned} h &= 0.276 \left(\frac{v}{5 \text{ mph}} \right)^{0.731} \text{ Btu/hrft}^2 \text{ } ^\circ\text{F} \\ &= 1.56 \left(\frac{v}{1.708 \text{ m/s}} \right)^{0.731} \text{ W/m}^2 \text{ } ^\circ\text{C} \end{aligned}$$

¹J.P. Holman, Heat Transfer, 2nd ed. McGraw Hill, 1963, Table 7-4, p. 199.

The Nusselt versus Reynolds number interpretation of this forced-convection assumption is shown in Figure 1, together with the relationship for a cylinder in cross flow. The assumed heat-transfer coefficients are a factor of 4 to 5 lower than the cylinder in cross flow.

The heat-transfer coefficient is an average value for the receiver. The configuration is such that different circumferential positions would be expected to have different heat transfer coefficients. Furthermore, different temperatures would have an influence on the local heat transfer. These effects are all neglected in the average calculations.

Although the system performance computer model is capable of calculating a heat-transfer coefficient and a heat loss for every wind speed, only the average wind speed is used to arrive at an average performance. The immediate concern is to know the heat loss at 13 mph (1.708 m/s). The Reynolds number at this operating point is in the neighborhood of 6×10^5 . We will need to know details of local wind velocity and temperature for the final design.

FIGURE 1 DATA FOR HEATING AND COOLING OF AIR FLOWING NORMAL TO SINGLE CYLINDERS
(FROM HOLMAN, *ibid.*, p. 171)

III TEST STRATEGY

We elected to use the cryogenic wind-tunnel facility constructed by Professor Clausing at the University of Illinois:

- The facility had the capability of reaching high Grashoff numbers as well as high Reynolds numbers, so that both free and forced convection effects could be tested.
- A model of the central receiver was scheduled for a test in the same facility, so that comparative data would become available.

The drawback to using the University of Illinois facility was that the tunnel was not finished at the outset of the program. New experimental facilities typically require considerable debugging and verification before they become useful tools. This proved to be the case.

A 1/12-scale model for test in the tunnel was constructed at SRI. It was designed for both steady-state and transient tests. A drawing of the model and the test plan are included in Appendix A. Details of the wind tunnel itself, and the instrumentation, are included in Appendix B, the report of Dr. Clausing.

IV. TEST RESULTS FROM THE WIND TUNNEL

Dr. Clausen ran several series of tests from which he obtained forced- and free-convection data at both room and cryogenic temperatures. He ran both transient and steady-state tests for free convection but only transient tests for forced convection. The data, included in Appendix B, are rich in results of interest to heat-transfer specialists. The basic results are as follows:

- (1) The free-convection tests were repeatable and gave similar results for steady-state and transient tests. We have confidence in these results.
- (2) Free-convection heat transfer is much less than we had assumed in the math model,

$$h_{\text{assumed}} = 1.56 \text{ W/m}^2\text{°C}$$

$$h_{\text{measured}} = 0.35 \text{ W/m}^2\text{°C}$$

- (3) The forced-convection results were not as reliable. For example, the convective heat-transfer coefficient calculated from data at the beginning of a transient test was lower than that calculated toward the end.
- (4) The convective heat-transfer coefficients from cryogenic forced-convection tests were much higher than we had calculated, so unreasonably high that they fell about the extrapolated data line for an exposed cylinder in cross flow. At 13 mph (1.708 m/s),

$$h_{\text{assumed}} = 3.21$$

$$h_{\text{measured}} = 18.31$$

- (5) The cryogenic forced-convection heat-transfer coefficients varied with time over a tenfold range.
- (6) The room-temperature, forced-convection heat-transfer coefficients very nearly confirmed the heat-transfer coefficients assumed by SRI.

It was clear from the tests that natural convection was sufficiently low that it was of no concern. By contrast, forced convection needed further examination.

Two approaches were taken simultaneously. One was to review the test data in an attempt to explain the unanticipated results of the forced-convection tests. The second was to set up water-table tests to visualize the forced-convection flow through the receiver. (See Section V.)

Three major anomalies were identified in the test procedure that could significantly increase the measured heat-transfer coefficients:

- (1) The turbulence level of the tunnel was high because of its rectangular shape and small size. This turbulence resulted in all measured heat-transfer coefficients being increased, especially for cavities. When standard bodies are eventually run for calibration, a partial correction can be made.
- (2) The velocity distribution in the tunnel was significantly uneven, so that there was an axial air flow within the receiver cavity from the region of high-velocity impact to the region of low-velocity impact. Hence, more air passed through the cavity than would be expected.
- (3) The model had end plates to simulate the long receivers, but they were not large enough to extend into the entire flow field. Hence, even if the wind velocity had been uniform, air would have passed through the cavity due to the uncompensated flow-field distortion.

We concluded that the tests had excessive air flow through the cavity, invalidating the results. These conclusions are elaborated in Appendix C.

V WATER-TABLE TESTS

Since the measured forced-convection, heat-transfer coefficients appeared to be as high as those of a cylinder in cross flow, we became interested in the flow patterns and velocities to see whether they were really of the same magnitude as on the exterior of a cylinder. Appendix D describes the water-table experiments and results.

The results showed that the velocities in the cavity are much smaller than external flow velocities. This reinforces the supposition that the model losses were caused by flow anomalies along the axis of the model receiver.

During the water-table tests, means were developed to further reduce the internal circulation. A baffle was placed below the receiver so that the stagnation point moved down on the receiver to the opening region. Thus the wind stagnates, and the air barely circulates within the cavity, whether the wind is flowing toward the front or from the rear. The baffle could simply be the eyelid for the aperture. Instead of opening upward to expose the cavity of the receiver, it could open downward and further reduce the circulation in the cavity.

VI CONCLUSIONS AND RECOMMENDATIONS

We have concluded that

- (1) The heat transfer of natural convection is negligible.
- (2) There are ways of aerodynamically redesigning the receiver to reduce the loss from natural convection.
- (3) The forced-convection, heat-transfer tests were inconclusive, but the values used in the performance calculation are reasonable.

We recommend

- (1) Further forced-convection tests in standard wind tunnels, where turbulence levels and velocity profiles are controlled. There is no need for further tests in a cryogenic tunnel.
- (2) Final design of the receiver to incorporate aerodynamic principles.

Appendix A

CRYOGENIC WIND TUNNEL
HTLF RECEIVER TEST PLAN

Appendix A

CRYOGENIC WIND TUNNEL HTLF RECEIVER TEST PLAN

I Introduction

The heat losses due to wind and natural convection will affect the receiver efficiency for the High-Temperature Line-Focus Solar (HTLF) Central Receiver System being studied by SRI International for the Department of Energy under Contract EY-76-C-03-0115.

The complex geometry of the receiver (see Figure A-1) precludes analytical prediction of these heat losses. Wind tunnel tests in the University of Illinois Cryogenic Wind Tunnel under the direction of Professor Arthur Clausen will be performed to provide data that will give insight into the loss mechanisms and means for estimating the losses from the full size receiver.

Convective heat transfer is governed by two dimensionless parameters, i.e., Reynolds number and Grashof number. The cryogenic tunnel allows achievement of a high Grashof number, which is the dimensionless parameter governing natural heat convection.

II Test Objective

The objective of the test is to obtain a convection heat loss coefficient valid for various wind conditions. This coefficient will allow estimation of convective heat losses from the HTLF receiver.

III Method of Approach

A convective heat transfer coefficient can be obtained from a transient cooldown experiment. The principle of such an experiment is illustrated in Figure A-2. The thickness of the copper sheet is properly

A-3

FIGURE A-1 HIGH-TEMPERATURE, LINE-FOCUS CENTRAL RECEIVER

FIGURE A-2 TRANSIENT HEAT TRANSFER EXPERIMENTS

designed such that heat conduction in the transverse direction is negligible compared with the heat convection in the normal direction. With the copper sheet initially heated to higher than the tunnel gas ambient temperature, the decay in temperature is a measure of the heat transfer coefficient:

$$h = - \frac{\rho C_p \delta}{T_w - T_a} \frac{dT_w}{dt} , \quad (A-1)$$

where ρ , C_p , δ are the density, the heat capacity, and the thickness of the copper sheet, respectively; T_w is the temperature of the copper sheet and T_a is the tunnel gas ambient temperature. h , the local heat

transfer coefficient, is expected to be independent of time, or equivalently, of $(T_w - T_a)$.

IV Test Hardware and Instrumentation

The test facility and the model support and heating systems are shown in Figures A-3 and A-4. As seen in Figure A-4, the model and its support are an integral part of the tunnel door. The vertical support is rotatable to vary the yaw angle. The interior surface of the model is made of a thin (1/16") copper sheet with 24 copper-constantan thermocouples soldered to its back. These thermocouples are evenly distributed--four around a circumference and six along the length. They will have an operating range of 80 to 300 K and a sensitivity of ± 0.1 K. The output from these thermocouples is to be recorded and processed in the UIUC data acquisition and processing system shown in Figure A-5.

The heater is in the form of a sliding rod. When inserted into the model cavity, it will heat the interior of the cavity to the desired temperature of about 130 K.

V Test Procedure

After fabrication, the door and the model will be put in place. The thermocouple leads and other flow measuring devices will be connected to the data acquisition system. After appropriate calibrations, the following procedure will be used for the four test configurations to be tested:

A. Normal head wind:

1. Turn on the liquid nitrogen supply and the tunnel fan.
2. Monitor tunnel gas temperature until it reaches 80 K.
3. Insert heater and heat until the interior temperature of the model registers 130 K.
4. Retract heater.
5. Record $T(t)$ from all 24 thermocouples as well as the tunnel temperature.

FIGURE A-3 TEST FACILITY (TOP VIEW)

FIGURE A-4 MODEL AND SUPPORT

FIGURE A-5 DATA ACQUISITION AND PROCESSING

6. Repeat the above for three different wind velocities.
7. Turn off recording after sufficient data are collected (on the order of minutes).

B. Oblique wind:

Procedures same as in A except that the model will be rotated to the desired yaw angle after heater retraction and before the start of recording.

C. Normal tail wind:

Procedures same as in B. Angle of rotation is 180°.

D. No wind (natural convection):

1. Insert the heater and heat while letting the tunnel coast to a stop.
2. Retract heater after model reaches 130 K and start recording.

VI Expected Results and Data Reduction

Each temperature trace is expected to be an exponential function of time. A constant h value can then be calculated using Eq. (1). The h values along the length of the model are expected to be uniform near midspan and nonuniform near the two ends. The nonuniform end condition serves to indicate the adequacy of the aspect ratio of the model. The h values in the uniform midsection will be averaged in the circumferential direction to obtain an $\bar{h}$. This $\bar{h}$ can be used to estimate the heat loss from a prototype receiver under the corresponding wind condition using the relationship:

$$q = \bar{h}A(\bar{T}_w - T_a) , \quad (A-2)$$

where q is heat loss per unit time, A is the interior surface area of the prototype receiver, $\bar{T}_w$ is an average wall temperature of the prototype receiver, and $\bar{h}$ is the heat transfer coefficient for that wind velocity determined by test.

VII Documentation

A test report will be incorporated in the SRI final report documenting the results of the test, and will include the following:

- Test Objectives
- Test Hardware
- Test Procedure
- Test Data
- Copies of Photographs
- Analysis of Test Data
- Conclusions

Appendix B

**SUPPORTING ANALYSIS, PROCEDURE, TEST EQUIPMENT,
AND EXPERIMENTAL RESULTS**

Report of:

Supporting Analysis, Procedure,
Test Equipment and Experimental
Results

Test of:

SRI International
Line-Focus Solar Receiver

Prepared and
Submitted by:

Dr. A. M. Clausing
Engineering Consultant
613 Hessel Blvd.
Champaign, IL 61820
Telephone Numbers:
(217) 333-0366
or 352-7478

Submitted on:

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Report Number:

AMC 79-100

Part I

Analysis of Model for
Experimental Study of
the Convective Heat Loss
from the SRI International
Line-Focus Solar Receiver

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NOMENCLATURE

A	--	Area (m^2)
AZ	--	Azimuth angle of outward normal to cavity aperture (Deg)
C	--	Thermal Capacitance (J/K)
c_p	--	Specific Heat (J/kg-K)
h	--	Heat Transfer Coefficient (W/m^2-K)
k	--	Thermal Conductivity (W/m-K)
L	--	A Characteristic Length (m)
m	--	$[hL^2/(k\delta)]^{1/2}$, fin parameter
P	--	Power (W)
q	--	Heat Transfer Rate (W)
q''	--	Heat Flux (W/m^2)
t	--	Time (s)
T	--	Temperature (K or C)
$\bar{V}$	--	Velocity, Azimuth Angle of $\bar{V}$ is 180° (m/s)
α	--	Thermal Diffusivity (m^2/s)
ϵ	--	Emittance
δ	--	Thickness (m)
ρ	--	Density (kg/m^3)
τ	--	Time Constant (s)
Bi	--	Biot Modulus, hL/k_m
Gr	--	Grashof number, $g\beta(T-T_a)L^3/\nu^2$

Nomenclature (cont.)

Subscripts

a	--	Ambient condition
h	--	Heater
i	--	Insulation
m	--	Model or Active Area of Model
o	--	Initial Condition
t	--	Total
tc	--	Thermocouple reading

1. Objectives

- (i) Evaluate suitability of model for steady-state and transient tests
- (ii) Provide estimates of sources of errors and test procedures which minimize experimental error
- (iii) Provide a means of correcting the resulting experimental data

2. Properties and Physical Dimensions

2.1 Thermal Properties

- (a) Properties of test gas, gaseous nitrogen: The properties which were used are tabulated in:

Jacobsen, R. T. and Stewart, R. B., "Thermophysical Properties of Nitrogen from the Fusion Line to 3500 R (1944 K) for Pressures to 150,000 psia (10342×10^5 N/m²)," NBS Technical Note 648, 1973.

- (b) Properties of model and insulation

Property:	k	ρ	C_p	α	ϵ	Substance
Units:	$\frac{W}{m \cdot K}$	kg/m ³	J/kg-K	m ² /s	(—)	(—)
Model (Calorimeter) @ 0°C	13.85	7818	460	3.87×10^{-6}	0.7	Type 304 S.S.
Insulation (Behind Active Surface of Model)	.0231	28	1260	0.655×10^{-6}	N/A	1.75 lb/ft ³ Urethane Foam

2.2 Physical Dimensions

(a) Active Area of Model -- Calorimeter

(b) Heaters

76.2 mm (3") x 203.2 mm (8"); 118 Ω (nominal)

Wattage: 168 W_{max} in still air

T_{max} = 300F (6/18/79 -- Art Slemmons)

$$\frac{A_h}{A_m} = \frac{\text{Area of Heaters (Five total)}}{\text{Active Model Area}} = \frac{(5)(76.2)(203.2)}{0.1117} = 0.693$$

That is, 69% of Active Calorimeter Area (Active heat transfer Area) is directly heated. This presents a serious problem mainly because the thermocouples are placed midway between the heaters. The temperature difference between the area centered

under a heater and the area near a thermocouple is typically the maximum difference which occurs between any two points on the active heat transfer area.

(c) Insulation

2.3 Tunnel Dimensions and Blockage

(a) $AZ = 0$ or 180° (axis $\perp$ to Velocity Vector, $\bar{V}$)

$$\text{Blockage Ratio} \approx \frac{(0.147\text{m} \times 0.5)}{(0.61\text{m}) \times (1.22\text{m})} = \frac{0.0735\text{m}^2}{0.744\text{m}^2} = 0.0988$$

(b) $AZ = \pm 90^\circ$ (Natural Convection Studies or Axis to $\bar{V}$)

$$\text{B.R.} \approx \frac{\left(\frac{\pi D^2}{4}\right) = \left[\frac{\pi}{4} (.147\text{m})^2\right]}{0.744\text{m}^2} = \frac{0.0170\text{m}^2}{0.744\text{m}^2}$$

B.R. ≈ 0.0228 (Best Case)

2.4 Miscellaneous Dimensions

(a) Lip Length of Model: $\approx 56\text{mm}$

(b) Gap in Stainless Steel created by A. M. Clausing between lip and active calorimeter area to provide thermal isolation $\approx 3\text{mm}$.

(c) End Caps supplied by A. M. Clausing to minimize end effects and to simulate a receiver of a length, L , which is much greater than its diameter, D .

(d) Prototype Dimensions Supplied by A. Slemmons

I.D. = 6 ft

O.D. = 8 ft

Aperture Dia. = 4 ft

Length of Receiver: >> O.D. [0(1 mile)]

3.1 Thermal Performance Characteristics

3.1 Estimated Heat Transfer Coefficients for Pretest Analysis

(a) Radiative Heat Transfer

$\epsilon = 0.7$ (A. Slemmons)

$$\epsilon_{\text{effective}} = \epsilon \cdot \frac{A_a}{A_m} = (0.7)(.26) = 0.2 = \epsilon_e$$

If $(T_m - T_a)$ is small, $h_r \approx 4 \epsilon_e \left[\frac{T_m + T_a}{2} \right]^3$; hence

$$h_r \Big|_{100K} \approx 0.045 \frac{W}{m^2-K} ; \quad h_r \Big|_{293K} \approx 1.14 \frac{W}{m^2-K}$$

$$h_r \Big|_{210K} \approx 0.5 \frac{W}{m^2-K}$$

Conclusion: Radiative Heat Transfer is negligible at cryogenic temperatures and contributes little at room temperature.

(b) Convection: Natural: $h_n \approx 5 \text{ W/m}^2\text{-K}$

Forced: $h_f \approx 50 \text{ W/m}^2\text{-K}$ @ $\bar{V} = 8 \text{ m/s}$

Combined: $h_c \approx 55 \text{ W/m}^2\text{-K}$

These are estimates based on experience and published correlations. T_a is assumed to be near 80K.

3.2 Relative Importance of Resistances of Heat Flow Paths

(a) Biot Modulus, Bi

(a.1) Influence of resistance perpendicular to heat transfer area

$$Bi_{\perp} = \frac{\text{Internal Resistance of Calorimeter } \perp \text{ to Heat Transfer Surface}}{\text{External Resistance}}$$

$$\text{or } Bi_{\perp} = \frac{\frac{\delta_m}{k_m A_m}}{\frac{1}{h A_m}} = \frac{h \delta_m}{k_m} = \frac{h (0.794 \times 10^{-3} \text{ m})}{13.85 \frac{\text{W}}{\text{m}^2 \text{ K}}}$$

$Bi_{\perp}$ (maximum, based on h_c) = 0.0032 -- negligible internal resistance

(a.2) Resistance parallel to heat transfer area based on a characteristic length of (Cavity Circumference)/4 $\equiv L_c = 55.8 \text{ mm}$

$$Bi_{\parallel} = \frac{\frac{L_c}{k_m (\delta_m L_c)}}{\frac{1}{h L_c^2}} = \frac{h L_c^2}{k_m \delta_m}$$

OR

$$Bi_{\parallel} = \frac{h (55.8 \text{ mm})^2}{\left(\frac{13.85 \text{ W}}{\text{m}^2 \text{ K}} \right) (0.7938 \text{ mm})}$$

$$Bi_{||} = \left\{ 0.283 \frac{m^2 K}{W} \right\} h$$

$h \frac{W}{m^2 K}$	5	20	50
$Bi_{ }$	1.42	5.66	14.2

Conclusion: Energy is not effectively redistributed.

(a.3) Relative Resistance of Insulation -- Importance of Heat Exchange Between Insulation and Active Heat Transfer Area

(i) Steady-State Performance

$$P_t = q_1 + q_2$$

$$\frac{q_1}{q_2} = \frac{\frac{\delta_i}{k_i A_m}}{\frac{1}{h A_m}} = \frac{h \delta_i}{k_i} = Bi$$

$$\frac{q_1}{q_2} = \frac{h(0.0279m)}{\frac{0.0231 W}{m-K}} = 1.2 \frac{h}{\left[\frac{W}{m^2 K} \right]}$$

$h \left[\frac{W}{m^2 K} \right]$	5 (NATURAL CONVECTION)	10	50 (COMBINED CONVECTION)
$\frac{q_1}{q_2}$	6.0	12	60
$\frac{q_2}{P_e} \times 100\% \left\{ \begin{array}{l} \% \\ \text{LOST} \end{array} \right\}$	14.3 %	7.7 %	1.67 %

(ii) Transient Performance

Relative importance of energy stored in insulation.

Assume average temperature of insulation is $(T_m + T_a)/2$.

Then:

$$\frac{\text{Energy in Insulation at Steady-State}}{\text{Energy in Calorimeter (Model)}} = \frac{C_i/2}{C_m} = \frac{(\rho_i c_i \delta_i)/2}{\rho_m c_m \delta_m}$$

$$\frac{(28 \text{ kg/m}^3)(1260 \text{ J/kg-K})(27.94 \text{ mm})}{(2)(7818 \text{ kg/m}^3)(460 \text{ J/kg-K})(0.794 \text{ mm})} = \frac{995 \frac{\text{J}}{\text{m}^2\text{K}}}{2(2855 \frac{\text{J}}{\text{m}^2\text{K}})} = 0.1726$$

Conclusion: at steady-state conditions, the energy stored in the insulation is 17% of the energy stored in the calorimeter. Since the high temperature region is closest to the calorimeter portion of the model, the influence will be significant if the characteristic times are similar.

Characteristic time of Insulation, δ_i^2 / α_i

$$\frac{\delta_i^2}{\alpha_i} = \frac{(0.02794 \text{ m})^2}{6.55 \times 10^{-7} \text{ m}^2/\text{s}} = 1192 \text{ s} = 19.9 \text{ MINUTES}$$

3.3 Transient Characteristics of Model

(a) Time Constant of Calorimeter

$$\tau_m = RC = \frac{1}{h A_m} (\rho_m c_p A_m \delta_m) = \frac{(\rho_m c_p \delta_m)}{h} = \frac{(7810 \frac{\text{kg}}{\text{m}^3})(460 \frac{\text{J}}{\text{kg K}})(.7938 \text{ mm})}{h}$$

$$\tau_m = \frac{[2855 \frac{\text{W}}{\text{m}^2 \text{ K}}] \text{ s}}{h}$$

$h [\frac{\text{W}}{\text{m}^2 \text{ K}}]$	5	10	20	50	60
$\tau_m [\text{s}]$	571	285.5	142.8	57.1	47.6

Conclusion: That heat exchanged with the insulation in transient determinations of h is of the order of 10% of the energy exchanged in natural convection studies. This exchange is a small source of error in combined convection studies.

(b) Time required to spread energy in model

Characteristic time based on an assumed length of 25mm (≈ 1 in):

$$t_c = \frac{L_c^2}{\alpha_m} = \frac{(25\text{mm})^2}{3.87 \times 10^{-6} \text{ m}^2/\text{s}} = 161\text{s} = 2.7 \text{ min.}$$

Conclusion: The model will be relatively isothermal only in the pure natural convection case.

(c) Heating rate with an adiabatic calorimeter (no losses)

(Note: These results are used in preliminary tests of the model -- see Part II, Series 1)

$$P_t = C_m \frac{dT}{dt} = A_m \rho_m c_m \frac{dT}{dt} = (0.1117)(2855) \frac{\text{J}}{\text{K}} \frac{dT}{dt}$$

OR

$$\frac{dT}{dt} = \frac{P_t \quad 60 \text{ s/min.}}{318.9 \left[\frac{\text{J}}{\text{K}} \right]} = \frac{P_t}{\text{W}} \cdot 0.1881 \left[\frac{\text{K}}{\text{min.}} \right]$$

P_t [W]	10	20	40	80	160
$\frac{dT}{dt} \left[\frac{\text{K}}{\text{min.}} \right]$	1.881	3.76	7.53	15.05	30.10

(d) Steady-State power requirements

$$P_t = h A_m (T - T_a) \text{ where } A_m = 0.1117 \text{ m}^2$$

$h \frac{\text{W}}{\text{m}^2 \text{K}}$	5	10	55	5	10	55
$\Delta T [\text{K}]$	20	20	20	50	50	50
$P_t [\text{W}]$	11.17	22.3	123	27.9	55.8	307

4. Steady-State Temperature Distributions

(Note: the solution which follows is used in correcting the steady-state data)

Diagram of Calorimeter

⊗ -- THERMOCOUPLE LOCATION

Problem Analyzed

Governing Equations

Region 1, $-1 \leq x < 0$

$$\frac{d^2 T_1}{dx^2} - m^2 (T_1 - T_a) = 0 \quad (1)$$

Region 2, $0 \leq x < \infty$

$$\frac{d^2 T_2}{dx^2} - m^2 (T_2 - T_a) + \frac{q_0'' L_c^2}{k_m \delta_m} = 0 \quad (2)$$

where

$$m^2 = \frac{h L_c^2}{k_m \delta_m}$$

Let $\theta = (T - T_a)$. Solutions to (1) and (2) are

$$\theta_1 = c_1 e^{mx} + c_2 e^{-mx} \quad (3)$$

$$\theta_2 = c_3 e^{mx} + c_4 e^{-mx} + \frac{q_0''}{h} \quad (4)$$

BOUNDARY CONDITIONS:

$$\left. \frac{d\theta_1}{dx} \right|_{x=-1} = 0 \quad (5)$$

$$\left. \frac{d\theta_1}{dx} \right|_0 = \left. \frac{d\theta_2}{dx} \right|_0 \quad (7)$$

$$\theta_1(0) = \theta_2(0) \quad (6)$$

$$\theta_2(x \rightarrow \infty) = \text{finite value} \quad (8)$$

Boundary Conditions Give:

$$(8) \Rightarrow \boxed{c_3 = 0} \quad (9)$$

$$(6) \Rightarrow c_1 + c_2 = c_4 + q_0''/h$$

$$(7) \Rightarrow c_1 - c_2 = -c_4$$

$$(6)+(7) \Rightarrow 2c_1 = q_0''/h$$

$$\boxed{c_1 = \frac{q_0''}{2h}} \quad (10)$$

$$\text{AND } (6)-(7) \Rightarrow 2c_2 = 2c_4 + q_0''/h$$

$$\boxed{c_4 = c_2 - \frac{q_0''}{2h}} \quad (11)$$

$$(5) \Rightarrow 0 = c_1 e^{-m} - c_2 e^m$$

$$\boxed{c_2 = c_1 e^{-2m}} \quad (12)$$

Hence, (10) and (12) give:

$$c_2 = \frac{q_0''}{2h} e^{-2m} \quad (13)$$

and (11) and (13) give:

$$c_4 = \frac{q_0''}{2h} (e^{-2m} - 1) \quad (14)$$

If

$$T^* \equiv \frac{T - T_a}{\frac{q_0''}{2h}} \quad (15)$$

Eq. (3) becomes

$$\frac{T_1 - T_a}{\frac{f_0''}{2h}} \equiv T_1^* = e^{mx} + e^{-m(x+2)} \quad (16)$$

and (4) becomes

$$\frac{T_2 - T_a}{\frac{f_0''}{2h}} = T_2^* = (e^{-2m} - 1)e^{-mx} + 2 \quad (17)$$

checks:

$$T_2^*(x \rightarrow \infty) = 2 \quad \text{or} \quad f_0'' = h(T_{x \rightarrow \infty} - T_a)$$

$$T_2^*(0) = e^{-2m} + 1 = T_1^*(0)$$

$$\left. \frac{dT_1^*}{dx} \right|_{x=0} = m - m e^{-2m} = \left. \frac{dT_2^*}{dx} \right|_{x=0}$$

Maximum Temperature Difference in Model:

$$\Delta T_{\max} \equiv \left\{ T_1 \Big|_{x=1} - T_2 \Big|_{x \rightarrow \infty} \right\} = \left\{ \frac{T_1|_{x=1} - T_a}{f_0''/2h} - \frac{T_2|_{x \rightarrow \infty} - T_a}{f_0''/2h} \right\} \frac{f_0''}{2h}$$

or

$$\Delta T_{max} = \left\{ T_1 \Big|_{x=-1} - T_2 \Big|_{x \rightarrow \infty} \right\} \frac{q_0''}{2h}$$

(16) and (17) give

$$\Delta T_{max} = \left\{ e^{-m} - 1 \right\} \frac{q_0''}{h}$$

In terms of $(T_2|_{x \rightarrow \infty} - T_a)$:

$$\frac{\Delta T_{max}}{(T_2|_{x \rightarrow \infty} - T_a)} = e^{-m} - 1 = \frac{T_1(x=-1) - T_2(x \rightarrow \infty)}{T_2(x \rightarrow \infty) - T_a} \quad (18)$$

Maximum Error in Thermocouple Reading Due to Steady-State Gradients

$$T_1(x=-1) - T_2(x \rightarrow \infty) = \{e^{-m} - 1\} \{T_2(x \rightarrow \infty) - T_a\} \quad (19)$$

Note:

$$m \equiv \left[\frac{h L_c^2}{k_m \delta_m} \right]^{1/2} = 0 \text{ gives: NO ERROR}$$

AND

$$m \rightarrow \infty \text{ gives: } \Delta T = -\{T_2(x \rightarrow \infty) - T_a\}$$

h	$L_c = 10\text{mm}$		$L_c = 5\text{mm}$	
	m	% error in ΔT	m	% error in ΔT
$\frac{W}{m^2 K}$		$100 (e^{-m}-1)$		
5	0.2132	-19.2 %	0.1066	-10.1 %
20	0.4265	-34.7 %	0.2132	-19.2 %
30	0.5223	-40.7 %	0.2612	-22.9 %
2.5	0.1508	-14.0 %	0.0754	-7.26 %
10	0.3016	-26.0 %	0.1508	-14.0 %
60	0.7388	-52.2 %	0.3694	-30.9 %

5. Computer Simulations of Experimental Tests

5.1 Combined Convection with $\bar{V} = 8\text{m/s}$ and $T_a = 295\text{K}$

```

RHOM    = .782E+04,
CPM      = .46E+03,
AKM      = .1385E+02,
EMISS    = .25E+00,
D        = .914E-01,
XL       = .914E-01,
V        = .8E+01,
TA       = .285E+03,
TI       = .315E+03,
DTIME    = .4E+01,
BI       = .26792143875273E-02,
FO       = .24441226226081E+02,
FDP      = .18435409483964E-02,
VDA      = .7938E-03,
AIR      = F,
NMAX     = 60,
$END

```

TIME	T	RE	GR	RE**2/GR	HC	HF	HN	HR
0.	315.	.463E+05	.300E+07	715.19	45.2	44.8	6.1	1.5
4.	313.	.466E+05	.284E+07	762.76	45.2	44.8	6.0	1.5
8.	311.	.468E+05	.270E+07	813.67	45.2	44.9	5.9	1.5
12.	309.	.471E+05	.255E+07	868.15	45.3	44.9	5.8	1.5
16.	308.	.473E+05	.242E+07	926.45	45.3	44.9	5.7	1.5
20.	306.	.475E+05	.229E+07	988.84	45.3	44.9	5.6	1.5
24.	305.	.477E+05	.216E+07	1055.60	45.3	45.0	5.5	1.5
28.	304.	.479E+05	.204E+07	1127.05	45.3	45.0	5.4	1.4
32.	302.	.481E+05	.192E+07	1203.51	45.3	45.0	5.3	1.4
36.	301.	.483E+05	.181E+07	1285.33	45.3	45.0	5.3	1.4
40.	300.	.485E+05	.171E+07	1372.89	45.3	45.0	5.2	1.4
44.	299.	.486E+05	.161E+07	1466.59	45.4	45.1	5.1	1.4
48.	298.	.487E+05	.152E+07	1566.85	45.4	45.1	5.0	1.4
52.	297.	.489E+05	.143E+07	1674.14	45.4	45.1	4.9	1.4
56.	297.	.490E+05	.134E+07	1788.96	45.4	45.1	4.8	1.4
60.	296.	.491E+05	.126E+07	1911.32	45.4	45.1	4.8	1.4
64.	295.	.492E+05	.119E+07	2043.28	45.4	45.1	4.7	1.4

5.2 Combined Convection with $\bar{V} = 8\text{m/s}$ and $T_a = 80\text{K}$

```

RHOM    = .782E+04,
CPM     = .46E+03,
XKM     = .1385E+02,
EMISS   = .25E+00,
D       = .914E-01,
XL      = .914E-01,
V       = .8E+01,
TA      = .8E+02,
TI      = .315E+03,
DTIME   = .5E+01,
BI      = .29245727799362E-02,
FO      = .30551532782602E+02,
FOP     = .23044261854955E-02,
VDA     = .7938E-03,
AIR     = F,
INMAX   = 60,
I$END

```

TIME	T	RE	GR	RE+2/GR	HC	HF	HN	HR
0.	315.	.994E+05	.165E+09	59.71	50.4	49.1	11.4	.6
5.	294.	.110E+06	.194E+09	62.02	51.0	49.7	11.3	.5
10.	275.	.121E+06	.226E+09	64.60	51.5	50.3	11.2	.4
15.	257.	.132E+06	.261E+09	67.46	52.1	50.9	11.1	.3
20.	241.	.145E+06	.299E+09	70.64	52.6	51.5	10.9	.3
25.	226.	.160E+06	.339E+09	75.68	53.4	52.3	10.8	.2
30.	212.	.175E+06	.385E+09	79.75	54.0	52.9	10.7	.2
35.	200.	.191E+06	.432E+09	84.30	54.6	53.6	10.5	.2
40.	188.	.207E+06	.481E+09	89.39	55.2	54.2	10.4	.2
45.	178.	.224E+06	.528E+09	95.08	55.7	54.8	10.3	.1
50.	168.	.241E+06	.574E+09	101.45	56.3	55.3	10.1	.1
55.	159.	.259E+06	.617E+09	108.59	56.8	55.9	10.0	.1
60.	152.	.277E+06	.656E+09	116.58	57.2	56.4	9.8	.1
65.	144.	.294E+06	.689E+09	125.54	57.7	56.9	9.6	.1
70.	138.	.312E+06	.716E+09	135.58	58.1	57.3	9.5	.1
75.	132.	.329E+06	.736E+09	146.85	58.5	57.8	9.3	.1
80.	127.	.346E+06	.749E+09	159.49	58.9	58.2	9.1	.1
85.	122.	.362E+06	.755E+09	173.67	59.2	58.5	8.9	.1
90.	117.	.378E+06	.753E+09	189.59	59.5	58.9	8.8	.1
95.	114.	.393E+06	.744E+09	207.45	59.8	59.2	8.6	.1
100.	110.	.407E+06	.729E+09	227.50	60.1	59.5	8.4	.0
105.	107.	.421E+06	.709E+09	250.02	60.3	59.8	8.2	.0
110.	104.	.434E+06	.683E+09	275.29	60.6	60.0	8.0	.0
115.	101.	.446E+06	.654E+09	303.67	60.8	60.3	7.8	.0
120.	99.	.457E+06	.622E+09	335.54	60.9	60.5	7.6	.0
125.	97.	.467E+06	.588E+09	371.32	61.1	60.7	7.4	.0
130.	95.	.477E+06	.553E+09	411.51	61.3	60.8	7.3	.0
135.	94.	.486E+06	.517E+09	456.63	61.4	61.0	7.1	.0
140.	93.	.494E+06	.481E+09	507.30	61.5	61.1	6.9	.0
145.	91.	.502E+06	.446E+09	564.20	61.6	61.3	6.7	.0

5.3 Natural Convection with $T_a = 80K$

RHOM	= .782E+04,	TI	= .315E+03,
CPM	= .46E+03,	DTIME	= .2E+02,
XKM	= .1325E+02	BI	= .68575109287953E-03
EMISS.	= .25E+00,	FO	= .12220613113041E+03
D	= .914E-01,	FDP	= .92177047419612E-02
XL	= .914E-01,	VDA	= .7938E-03,
V	= 0.0,	AIR	= F,
TA	= .8E+02,		

TIME	T	RE	GR	RE♦♦2/GR	HC	HF	HN	HR
0.	315.	0.	.165E+09	0.00	11.4	0.0	11.4	.6
20.	295.	0.	.192E+09	0.00	11.3	0.0	11.3	.5
40.	278.	0.	.221E+09	0.00	11.2	0.0	11.2	.4
60.	261.	0.	.252E+09	0.00	11.1	0.0	11.1	.4
80.	247.	0.	.284E+09	0.00	11.0	0.0	11.0	.3
100.	234.	0.	.315E+09	0.00	10.9	0.0	10.9	.3
120.	222.	0.	.352E+09	0.00	10.8	0.0	10.8	.2
140.	211.	0.	.390E+09	0.00	10.7	0.0	10.7	.2
160.	201.	0.	.427E+09	0.00	10.6	0.0	10.6	.2
180.	192.	0.	.465E+09	0.00	10.5	0.0	10.5	.2
200.	183.	0.	.502E+09	0.00	10.4	0.0	10.4	.1
220.	176.	0.	.537E+09	0.00	10.2	0.0	10.2	.1
240.	169.	0.	.571E+09	0.00	10.1	0.0	10.1	.1
260.	162.	0.	.602E+09	0.00	10.0	0.0	10.0	.1
280.	157.	0.	.631E+09	0.00	9.9	0.0	9.9	.1
300.	151.	0.	.657E+09	0.00	9.8	0.0	9.8	.1
320.	146.	0.	.680E+09	0.00	9.7	0.0	9.7	.1
340.	142.	0.	.700E+09	0.00	9.6	0.0	9.6	.1
360.	138.	0.	.717E+09	0.00	9.5	0.0	9.5	.1
380.	134.	0.	.731E+09	0.00	9.4	0.0	9.4	.1
400.	130.	0.	.741E+09	0.00	9.2	0.0	9.2	.1
420.	127.	0.	.749E+09	0.00	9.1	0.0	9.1	.1
440.	124.	0.	.753E+09	0.00	9.0	0.0	9.0	.1
460.	121.	0.	.755E+09	0.00	8.9	0.0	8.9	.1
480.	119.	0.	.754E+09	0.00	8.8	0.0	8.8	.1
500.	116.	0.	.751E+09	0.00	8.7	0.0	8.7	.1
520.	114.	0.	.745E+09	0.00	8.6	0.0	8.6	.1
540.	112.	0.	.738E+09	0.00	8.5	0.0	8.5	.1
560.	110.	0.	.729E+09	0.00	8.4	0.0	8.4	.0
580.	108.	0.	.718E+09	0.00	8.3	0.0	8.3	.0
600.	107.	0.	.706E+09	0.00	8.2	0.0	8.2	.0
620.	105.	0.	.693E+09	0.00	8.1	0.0	8.1	.0
640.	104.	0.	.679E+09	0.00	8.0	0.0	8.0	.0
660.	102.	0.	.664E+09	0.00	7.9	0.0	7.9	.0
680.	101.	0.	.649E+09	0.00	7.8	0.0	7.8	.0
700.	100.	0.	.633E+09	0.00	7.7	0.0	7.7	.0
720.	99.	0.	.616E+09	0.00	7.6	0.0	7.6	.0
740.	98.	0.	.600E+09	0.00	7.5	0.0	7.5	.0
760.	97.	0.	.583E+09	0.00	7.4	0.0	7.4	.0
780.	96.	0.	.567E+09	0.00	7.3	0.0	7.3	.0
800.	95.	0.	.550E+09	0.00	7.2	0.0	7.2	.0
820.	94.	0.	.533E+09	0.00	7.2	0.0	7.2	.0

5.4 Natural Convection with $T_a = 295K$

```

RHOM    = .782E+04,
CPM     = .46E+03,
XKM     = .1385E+02,
EMISS   = .25E+00,
D       = .914E-01,
XL      = .914E-01,
V       = 0.0,
T1      = .285E+03,
T2      = .315E+03,
DTIME   = .3E+02,
BI      = .43666938693754E-03,
FO      = .18330919669561E+03,
FOP     = .13826557112973E-01,
VDA     = .7938E-03,
AIR     = F,
NMAX    = 60,
$END

```

TIME	T	RE	GR	RE♦♦2/GR	HC	HF	HN	HR
0.	315.	0.	.300E+07	0.00	6.1	0.0	6.1	1.5
30.	313.	0.	.281E+07	0.00	6.0	0.0	6.0	1.5
60.	310.	0.	.263E+07	0.00	5.8	0.0	5.8	1.5
90.	308.	0.	.247E+07	0.00	5.7	0.0	5.7	1.5
120.	307.	0.	.231E+07	0.00	5.6	0.0	5.6	1.5
150.	305.	0.	.217E+07	0.00	5.5	0.0	5.5	1.5
180.	304.	0.	.203E+07	0.00	5.4	0.0	5.4	1.4
210.	302.	0.	.191E+07	0.00	5.3	0.0	5.3	1.4
240.	301.	0.	.179E+07	0.00	5.2	0.0	5.2	1.4
270.	300.	0.	.168E+07	0.00	5.1	0.0	5.1	1.4
300.	299.	0.	.157E+07	0.00	5.1	0.0	5.1	1.4
330.	298.	0.	.148E+07	0.00	5.0	0.0	5.0	1.4
360.	297.	0.	.139E+07	0.00	4.9	0.0	4.9	1.4
390.	296.	0.	.131E+07	0.00	4.8	0.0	4.8	1.4
420.	296.	0.	.123E+07	0.00	4.7	0.0	4.7	1.4

6. Summary and Conclusions

1. The one piece lip/cavity construction is not acceptable; hence, the modification described in Section 2.3 was effected.
2. The mounting of the thermocouples between the heaters in the region where the temperature gradients in the model skin are the greatest is a serious problem which will drastically influence the accuracy of the results.
3. The error due to temperature gradients can be reduced by applying the theoretical values developed in Section 4. A transient analysis would give a correction which could include the influences of both the conduction in the skin and the conductive exchange with the insulation.
4. The steady-state heat loss through the insulation is significant in natural convection studies. It will be of the order of 10 percent of the total loss. The data will be corrected for this influence.

$h \left[\frac{W}{m^2 K} \right]$	5	10	20	40
correction (%)	16.5%	8.3%	4.1%	2%

5. Since the finite model is to approximate an essentially infinite prototype receiver, end caps were installed on the model to split the flow and in this manner simulate a long receiver. The end caps are thermally neutral (see description in Section 2.4).

Part II

**Data Reduction Procedure,
Description of Equipment,
and Experimental Results**

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Appendix A

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Test Series #	Temp	Steady or Transient	Forced or Natural
1	Room	both	N
2	Room	S	N
3	Room	T	N
4	Room	T	F
5	cryogenic	T	F
6			
7	cryogenic	S	N

1. General Assumptions and Experimental Procedures

1.1 General Assumptions

The data reduction is complicated by the following:

- (a) Heat exchange by conduction with the insulation and heaters which are located behind the active heat transfer area. It will be assumed that:

- (a.1) Transient data

The influences of the thermal capacitance of the heaters and insulation should be negligible. Unfortunately, this is not the case; however, it is difficult to accurately determine these influences in order to apply a meaningful correction. These influences must be minimized in the design of the model.

- (a.2) Steady-State data

The heat transfer through the insulation will be assumed to be

$$q_i'' = \frac{k_i (T - T_a)}{\delta_i} \quad (1)$$

Hence, the net flux through the active area is

$$q'' = \frac{P_t}{A_h} - q_i'' = \frac{12.92}{[m^2]} P_t - q_i'' \quad (2)$$

(b) The radiant heat exchange

Since the emittance of the active surface is low ($\epsilon = 0.7$, value obtained from A. Slemmons), since the ambient and model temperatures are low (generally below 293K), and since the cavity geometry further reduces the radiative losses, this influence is usually small. Specifically, the effective heat transfer coefficient due to radiation, h_r , will typically be $1 \text{ W/m}^2\text{-K}$ or less (see Part I, Section 3.1). Hence, no correction for radiant exchange is made. All reported heat transfer coefficients, therefore, are total values including the effects of both convective and radiative heat exchange.

(c) Temperature gradients within the model

Since the model was not supplied with a removable cover and since it was not designed to be injected into the tunnel, heat is always being exchanged by natural convection. The greatest variation in the heat transfer coefficient occurs in the circumferential direction; however, the five heaters are located (see Fig. 1) such that only an axial, not a circumferential, variation in the heat supplied is possible. Hence, an initially isothermal heat transfer surface cannot be established. Furthermore, the thermocouples are located in the areas between the foil heaters in the regions where the temperature

gradients are a maximum. (A scaled drawing of the heater and thermocouple locations is provided in Fig. 1). The consequences of these gradients follow.

- (i) It is not possible to obtain meaningful steady-state, combined convection data. The temperature in the unheated, cold area in which the thermocouples are mounted cannot be related to the power generated in the heated portion of the model.
- (ii) The steady-state natural convection data closely approximates a constant flux boundary condition. This is a meaningful and commonly studied problem; however, prescribed temperature boundary conditions are more representative of solar receiver applications.
- (iii) All data from the edges of the model are of dubious value. Specifically, the data from the thermocouples located along the lower, front edge, the $\theta = 0$ location as defined in Fig. 1, and along the cavity ends, the $z = \pm 5$ locations as defined in Fig. 1, are generally not meaningful. Hence, data from these locations is deemphasized. Some data from these locations are reported for completeness and to demonstrate these influences. These data should generally be ignored.
- (iv) All combined convection data are compromised.

In the acquisition and analysis of the data, the following assumptions are employed.

Fig. 1 Plane Projection of Cavity Showing Heated Areas and Thermocouple locations --
Drawn to scale

- (1) The energy generated by the foil heater is uniform over the entire heater area, 76.2mm x 203.2mm, as depicted in Fig. 1.
- (2) The thermocouples are centered between heaters and are at a distance of 13mm from the edges of the two adjacent heaters. See Fig. 1.
- (3) At steady-state, the temperature in the central portion of the heated area is

$$T = T_{tc} + (1 - e^{-m}) (T - T_a) \quad (3)$$

where T_{tc} is the temperature indicated by the thermocouple and $m = [h_t L_c^2 / k_m \delta_m]^{1/2}$. L_c is 13 mm. Note: Eq (3) is an implicit expression for T which was derived in Section 4 of Part I.

1.2 Experimental Procedures

1.2.1 Steady-State Experiments

The procedure for the steady-state experiments is relatively simple. All steady-state experiments are pure natural convection cases. Hence, the azimuth angle of the outward normal to the cavity aperture is set at 90 degrees to reduce the influence of the walls of the chamber. The procedure is:

- (i) Adjust the ambient temperature, T_a , to the desired value.
- (ii) Set the heater power, P_t , at the desired level. Each heater is set at identical power levels, $P_t/5$.

- (iii) Allow the system to reach steady-state.
- (iv) Record thermocouple readings (including the array of ambient temperature thermocouples), the time, the azimuth angle, and the power.

Note: The power is determined from measurements of the voltage drop across the heaters and the precision resistors in series with the respective heater. The time at which steady-state is attained is determined from intermediate readings. The variations between repeated runs and errors due to transients are both clearly indicated in Section 5.1 -- Test Series 1. Errors due to these influences are completely negligible.

1.2.2 Transient Experiments

Both natural convection and combined convection data can be obtained with the transient technique. Since steady-state techniques are not suitable for combined convection investigations, and since more accurate natural convection data is obtainable with the current model with steady-state experiments, the bulk of the transient tests were combined convection tests. The procedure is:

- (i) Adjust the ambient temperature in the cryogenic tunnel to the desired level.
- (ii) Heat the model with the fans off to the desired temperature level. During the heating mode, the power supplied to each of the five model heaters is identical.
- (iii) Shut off all heaters and wait approximately 30 seconds to reduce the magnitude of the temperature gradients between

heated and unheated areas (see Fig. 1).

(iv) Start the data logger in order to record:

$$\bar{V} = f_1 (\text{time})$$

$$T_{tc} = f_2 (z, \theta, t)$$

$$T_a = F_3(t) \text{ -- a weak function resulting from power input to fans}$$

(v) Start fans if a combined convection test, i.e., $\bar{V} \neq 0$.

(vi) Record data until $(T - T_a)$ is less than approximately 8K.

Note: The time required for a complete scan of all channels is approximately two seconds. The time required to attain a constant velocity is of the order of 8 seconds after switching on the fan motor.

2. Data Reduction Procedure -- Steady-State Technique

2.1 Procedure and Relevant Equations

Given: Thermocouple Readings (T_{tc}), P_t , and T_a

Find: $h = f(\theta, z)$

Step 1 -- Calculation of uncorrected values of total heat transfer coefficients, h .

$$h_t^o = \frac{P_t}{A_h (T_{tc} - T_a)} = \frac{(12.92) P_t}{(T_{tc} - T_a) \left[\frac{1}{m^2} \right]} \quad (4)$$

Step 2 -- Calculation of m

$$m^k = \left[\frac{h_t^{k-1} L_c^2}{k_m \delta_m} \right]^{1/2} = 0.124 \left[\frac{h_t^{k-1} (m^2 - K)}{W} \right]^{1/2} \quad (5)$$

Step 3 -- Calculation of temperature in central portion of heated area

$$T^k = T_{tc} + (1 - e^{-m})(T^{k-1} - T_a) \quad (6)$$

where $T^0 = T_{tc}$.

Step 4 -- Calculation of heat loss through insulation

$$(q_L'')^k = \frac{k_i (T^k - T_a)}{\delta_i} = \frac{0.828 W}{m^2 - K} (T^k - T_a) \quad (7)$$

Step 5 -- Calculation of corrected heat transfer coefficients

$$h^k = \frac{\frac{12.92}{m^2} P_t - (q_L'')^k}{(T^k - T_a)} \quad (8)$$

Step 6 -- Repeat steps 2 through 5. An iteration is used because Eq (6) is an implicit expression for T ; a single correction was found to be adequate. For example, compare the tabulated results from 1st and 2nd iterations given in Section 5.2.1, Test Series 2.

2.2 DATA REDUCTION PROGRAM FOR STEADY-STATE DATA

```
1      PROGRAM SRI (SRID,ANS,TAPE5=SRID,TAPE6=ANS,OUTPUT)
2      DIMENSION T(24),QL(24),H(24),TT(24)
3      NAMELIST /NAM/P,TT,TA,II,TEST
4      DATA II,P/16,10./
5 1     READ(5,NAM)
6       IF (EOF(5).NE.0.0) GO TO 999
7       PP=12.92+P
8       NN=II/4
9       K=1
10      DO 3 I=1,II
11         T(I)=TT(I)
12         H(I)=PP/(T(I)-TA)
13 3      CONTINUE
14      WRITE(6,102) TEST,P,TA
15      GO TO 13
16 15     WRITE(6,110)
17         DO 5 I=1,4
18            WRITE(6,100) H(I),H(I+4),H(I+8),H(I+12)
19 5      CONTINUE
20         DO 7 J=1,2
21            WRITE(6,104) J
22            DO 9 I=1,II
23               XM=SQRT(H(I))*0.124
24               T(I)=TT(I)+(1.-EXP(-XM))*(T(I)-TA)
25               QL(I)=.828*(T(I)-TA)
26               H(I)=(PP-QL(I))/(T(I)-TA)
27 9      CONTINUE
28         DO 11 I=1,4
29            WRITE(6,100) H(I),H(I+4),H(I+8),H(I+12)
30 11     CONTINUE
31 7      CONTINUE
32 13     WRITE(6,106)
33         IF (K.EQ.2) WRITE(6,108)
34         DO 17 I=1,4
35            WRITE(6,100) T(I),T(I+4),T(I+8),T(I+12)
36 17     CONTINUE
37         IF (K.GT.1) GO TO 1
38         K=2
39         GO TO 15
40 100    FORMAT(6F5.1)
41 102    FORMAT(///12H TEST NUMBER,F4.1/7H POWER=,F6.2,4X,3HTA=,
42 2 F4.1)
43 104    FORMAT(//14)
44 106    FORMAT(//25H TEMPERATURE DISTRIBUTION/)
45 108    FORMAT(18H CORRECTED VERSION/)
46 110    FORMAT(//27H HEAT TRANSFER COEFFICIENTS//19H UNCORRECTED VALUES
47 999    STOP
48      END
```

3. Data Reduction Procedure -- Transient Technique

3.1 Procedure and Relevant Equations

If the temperature gradients within the calorimeter portion of the model perpendicular to the heat transfer surface are negligible, and if the resistance parallel to this surface is large, the local heat flux q'' is:

$$q'' = - \rho_m c_{pm} \delta_m \frac{dT}{dt} \quad (9)$$

Since

$$q'' = h (T - T_a) \quad (10)$$

The local heat transfer coefficient follows from

$$h = \frac{- \rho_m c_{pm} \delta_m \frac{dT}{dt}}{(T - T_a)} \quad (11)$$

The conditions under which Eq (11) is valid are discussed in detail by Clausing. This reference is included in Appendix B for convenience. The criteria for the receiver model of interest are given in Part I.

3.2 Description of Data Reduction Program

The computer program used in the reduction of the data obtained with the transient technique is a lengthy and complex program. It uses several programs in the library of our computer system. It is not a program of general value, therefore, only a brief description is provided.

The program has the following function.

- (a) It converts the time and series temperature scans from the digital data acquisition system to a set of data in which all unknowns are provided at the same set of discrete times.
- (b) It provides the derivatives of these temperatures with respect to time, and calculates the respective heat transfer coefficient from these derivatives, the corresponding values of $(T-T_a)$, and the parameters of the model.
- (c) It prints out the resulting matrix of heat transfer coefficients and corresponding model temperatures using the specified time increment. The corresponding ambient tunnel temperatures are also printed in the table (see, e.g., the tabular results given in Section 5.3). The velocity in the test section is also printed.

4. Description of Apparatus

4.1 Equipment Used -- Steady-State Technique

The power for the steady-state tests is supplied by a well regulated DC power supply. Each heater is connected in parallel to the power supply as indicated in the schematic diagram, Fig. 2. The

B-43

Fig. 2 Schematic Diagram of Heater Circuit

five individual heaters, H25 through H29, have a nominal resistance of 120Ω . Since the actual resistance of the heaters range from 114Ω to 120.2Ω , the variable resistors R_1 through R_5 are utilized in order to supply equal power to the five heaters with a single power supply. Precision one ohm shunts are connected in series with each heater to enable the accurate determination of the current through each heater. The current in amperes is numerically equal to the voltage drop across the one ohm shunt. Two precision digital voltmeters are used to simultaneously measure the voltage drop across the heater, V_2 , and the current supplied, V_1 . A multi-channel data acquisition system is used to periodically record the model, chamber, and ambient temperatures. A cathode ray tube is used to display data during test runs. Relevant data are periodically recorded on magnetic tape with a digital cassette recorder. The cassettes are later read with an electronic data terminal to enable digital processing.

A list of the instrumentation used in the steady-state experiments follows.

Digital Voltmeters, V_1 and V_2 : Keithley Model 179 TRMS

Digital Power Supply: Sorensen Model DCR 150-10A

Data Acquisition System: Fluke Model 2240A

CRT Display: Hazeltine Model 1500

Cassette Recorder: Memodyne Digital Cassette Recorder

Electronic Data Terminal: Texas Instruments Model 733 ASR

4.2 Equipment Used -- Transient Technique

The equipment used with the transient technique is identical to the instrumentation used in the steady-state tests; although the

way it is employed is sometimes slightly different. For example, the circuit given in Fig. 2, and the instrumentation indicated in this figure, are utilized in heating the calorimeter portion of the model prior to the initiation of the transient tests. The power supplied during this heating mode is of secondary interest. The final temperature level is the only quantity of prime interest with the transient technique.

5. Results and Conclusions

5.1 Preliminary Experiments -- Test Series 1

A series of preliminary experiments was conducted in order to establish the accuracy of the instrumentation and to experimentally verify the characteristics of the model which were presented in Part I. This series of tests, Series 1, consisted of three experiments. The calorimeter portion of the model was insulated in all tests of Series 1 in order to eliminate convective heat exchange. The complete cavity was filled with fiberglass insulation in order to insulate the otherwise active, internal cavity area. The correspondence between the thermocouple number indicated in Fig. 1 (the same numbering scheme indicated on SRI drawing number 7896-610) and the channel number of the data acquisition system is given in Table 5.1.

Before test Series 1 was initiated, the electrical resistance at room temperature, 21.7C, was measured. These results are given in Table 5.2. These results were used to facilitate setting-up the heater circuits given in Fig. 2. The power dissipated in each heater was not based on these resistances. The electrical power was calcu-

Coordinate, z	-5	-3	-1	+1	+3
Channel No.	1-4	11-14	21-24	41-44	31-34
Thermocouple No.	1-4	5-8	9-12	13-16	17-20

Table 5.1 Correspondence Between Thermocouple Numbers and Channel Numbers

HEATER NO.	RESISTANCE [Ω]
H25	120.2
H26	120.0
H27	151.1
H28	119.9
H29	114.0

Table 5.2 Electrical Resistance of Heaters at Room Temperature (21.7°C)

lated from the voltage and current in order to eliminate the effects of variations in heater resistance with temperature.

Test 1.1 Equilibrium Tests

Objective and Procedure: The objective was to demonstrate the accuracy of the thermocouples and the data acquisition system used to record all temperature data. The procedure was simply to allow several hours for the unheated, insulated model to reach "equilibrium". Some edge effects are expected since the

model could not be perfectly insulated.

Test Conditions: $\bar{V} = 0$, $T_a = 21.7^\circ\text{C}$; $AZ = 90^\circ$

Results: See Table 5.3

Summary of Results:

- (i) Readings were essentially independent of time.
Fluctuations were 0.1K or less (I 0.05K).
- (ii) The maximum difference between any two thermocouples on the entire model was 0.7K.
- (iii) All thermocouples were reading properly, i.e., within the expected error band.

Test 1.2 Heating Test

Objectives and Procedure: The objectives were to check heaters and heater circuits, and to determine the characteristics of the calorimeter. The latter is done by comparing the variation of temperature with respect to time (specifically $\Delta T \equiv (T - T_0)$ where T_0 is the initial temperature at time $t = 0$) with the theoretical values obtained by assuming that all of the energy goes only into the 0.794mm thick calorimeter.

Test Conditions: $T_a = 21.7^\circ\text{C}$; $\bar{V} = 0$; $AZ = 90^\circ$; $P_t = 10\text{W}$

The results are given in Table 5.4 and are summarized in Table 5.5.

Test 1.3 Insulated Model -- Cooling Mode

Objective: To determine the rate of heat loss from the insulated calorimeter

Procedure: This test followed 612 seconds after the initiation

TEST 1.1 Isothermal, Insulated Model

44 21.7 t
43 21.7 t
42 21.7 t
41 21.8 t
34 21.1 t
33 21.1 t
32 21.1 t
31 21.2 t
24 21.4 t
23 21.4 t
22 21.5 t
21 21.5 t
14 21.7 t
13 21.7 t
12 21.7 t
11 21.8 t
4 21.4 t
3 21.3 t
2 21.4 t
1 21.4 t

125151
170:14:58:25

44 21.7 t
43 21.7 t
42 21.7 t
41 21.8 t
34 21.1 t
33 21.1 t
32 21.2 t
31 21.2 t
24 21.4 t
23 21.4 t
22 21.5 t
21 21.5 t
14 21.7 t
13 21.7 t
12 21.7 t
11 21.8 t
4 21.4 t
3 21.3 t
2 21.3 t
1 21.4 t

125151
170:14:58:35

44 21.7 t
43 21.7 t
42 21.7 t
41 21.8 t
34 21.1 t
33 21.1 t
32 21.2 t
31 21.3 t
24 21.4 t
23 21.4 t
22 21.4 t
21 21.5 t
14 21.7 t
13 21.7 t
12 21.7 t
11 21.8 t
4 21.3 t
3 21.3 t
2 21.3 t
1 21.4 t

125151
170:14:58:46

44 21.7 t
43 21.7 t
42 21.7 t
41 21.8 t
34 21.1 t
33 21.1 t
32 21.1 t
31 21.2 t
24 21.4 t
23 21.4 t
22 21.4 t
21 21.5 t
14 21.7 t
13 21.7 t
12 21.7 t
11 21.8 t
4 21.4 t
3 21.3 t
2 21.3 t
1 21.4 t

125151
170:14:58:56

44 21.7 t
43 21.7 t
42 21.7 t
41 21.8 t
34 21.1 t
33 21.1 t
32 21.2 t
31 21.2 t
24 21.5 t
23 21.4 t
22 21.5 t
21 21.5 t
14 21.8 t
13 21.7 t
12 21.8 t
11 21.8 t
4 21.3 t
3 21.3 t
2 21.3 t
1 21.4 t

125151
170:14:59:06

Table 5.3

TEST 1.2 Insulated Model $P_t = 10W$ -- Heating Mode

44	217	t
43	216	t
42	217	t
41	218	t
34	211	t
33	211	t
32	211	t
31	212	t
24	215	t
23	215	t
22	215	t
21	216	t
14	218	t
13	218	t
12	218	t
11	219	t
4	215	t
3	215	t
2	215	t
1	216	t

125151
170:15:08:45

$t = 0$

44	225	t
43	229	t
42	229	t
41	229	t
34	218	t
33	222	t
32	222	t
31	221	t
24	222	t
23	223	t
22	223	t
21	224	t
14	224	t
13	226	t
12	226	t
11	227	t
4	219	t
3	219	t
2	221	t
1	221	t

125151
170:15:09:33

$t = 57s$

44	233	t
43	240	t
42	240	t
41	239	t
34	226	t
33	232	t
32	232	t
31	230	t
24	229	t
23	232	t
22	233	t
21	231	t
14	232	t
13	236	t
12	235	t
11	235	t
4	224	t
3	226	t
2	228	t
1	227	t

125151
170:15:10:03

$t = 87s$

44	286	t
43	310	t
42	311	t
41	294	t
34	278	t
33	302	t
32	303	t
31	281	t
24	281	t
23	296	t
22	297	t
21	279	t
14	284	t
13	303	t
12	302	t
11	290	t
4	261	t
3	277	t
2	281	t
1	266	t

125151
170:15:13:07

$t = 271s$

296	t
321	t
323	t
303	t
287	t
314	t
315	t
289	t
289	t
307	t
308	t
287	t
293	t
314	t
313	t
299	t
267	t
285	t
289	t
271	t

125151
15:13:37

$t = 301s$

44	344	t
43	378	t
42	380	t
41	347	t
34	333	t
33	370	t
32	373	t
31	331	t
24	336	t
23	361	t
22	361	t
21	326	t
14	340	t
13	371	t
2	37	t
11	348	t
4	299	t
3	327	t
2	333	t
1	303	t

125151
70:15:16:24

$t = 468s$

349	t
384	t
386	t
352	t
338	t
376	t
379	t
336	t
341	t
367	t
367	t
331	t
345	t
377	t
376	t
348	t
303	t
333	t
338	t
307	t

125151
:16:45

$t = 489s$

Table 5.4

(T - T ₀) [°K]						
Channel Time	11/13	21/23	42/44	$\bar{\Delta T}$	ΔT theory	$\frac{\bar{\Delta T}}{\Delta T} \times 100\%$ theory
t = 0	--	--	--	--	0	--
57s	0.8/0.8	0.8/0.8	1.2/0.8	0.867	1.79	48.5%
87s	1.6/1.8	1.5/1.7	1.3/1.6	1.583	2.72	58.1%
271s	7.1/8.5	6.3/8.1	9.6/6.9	7.75	8.50	91.2%
301s	9.0/9.4	7.1/9.2	10.6/7.9	8.70	9.44	92.2%
468s	12.5/15.3	11.0/14.6	16.3/12.7	13.73	14.67	93.6%
489s	12.9/15.9	11.5/15.2	16.9/13.2	14.27	15.33	93.1%
57s-87s	0.8/1.0	0.7/0.9	1.1/0.8	0.883	0.941	93.8%
271s-301s	0.9/0.9	0.8/0.9	1.0/1.0	0.917	0.941	97.4%
468s-489s	0.4/0.6	0.5/0.6	0.6/0.5	0.533	0.658	93.8%

Table 5.5 Summary of Results for Insulated Model, Heating Test -- Test 1.2

of Test 1.2. The beginning of Test 1.3, the end of Test 1.2, was defined as the time the power was changed from 10W to zero. This is the definition of time, t = 0, for Test 1.3. Consequently:

- (i) large heat losses can be expected at small values of time. Note that the characteristic time of the insulation is $\delta_i^2/\alpha_i = 1190$ seconds.
- and (ii) The redistribution of energy in the calorimeter should result in an apparent heat gain at small values of time.

The results are given in Table 5.6 and are summarized in Table 5.7.

TEST 1.3 Insulated Model -- Cooling Mode

44	393	t	384	t	44	384	t	381	t
43	422	t	423	t	43	422	t	410	t
42	424	t	425	t	42	424	t	409	t
41	381	t	380	t	41	378	t	367	t
34	372	t	373	t	34	375	t	370	t
33	415	t	416	t	33	416	t	403	t
32	419	t	420	t	32	420	t	407	t
31	366	t	367	t	31	367	t	358	t
24	375	t	377	t	24	378	t	374	t
23	404	t	407	t	23	406	t	396	t
22	403	t	409	t	22	404	t	393	t
21	360	t	366	t	21	360	t	352	t
14	378	t	381	t	14	382	t	377	t
13	416	t	418	t	13	418	t	408	t
12	415	t	418	t	12	418	t	408	t
11	379	t	381	t	11	380	t	370	t
4	325	t	328	t	4	328	t	324	t
3	362	t	364	t	3	364	t	357	t
2	368	t	371	t	2	370	t	361	t
1	330	t	331	t	1	331	t	324	t

125151
170:15:18:57 $t=0$

5151
9:17 20s

125151
5
0:1 :19:38 41s

125151
5:21:44 167s

44	380	t	44	375	t	44	361	t
43	408	t	43	398	t	43	376	t
42	408	t	42	396	t	42	373	t
41	366	t	41	358	t	41	343	t
34	370	t	34	364	t	34	349	t
33	402	t	33	391	t	33	368	t
32	406	t	32	393	t	32	368	t
31	358	t	31	350	t	31	335	t
24	373	t	24	368	t	24	355	t
23	395	t	23	387	t	23	366	t
22	393	t	22	383	t	22	362	t
21	352	t	21	346	t	21	332	t
14	376	t	14	371	t	14	355	t
13	407	t	13	396	t	13	373	t
12	407	t	12	396	t	12	372	t
11	369	t	11	361	t	11	344	t
4	324	t	4	319	t	4	307	t
3	356	t	3	346	t	3	327	t
2	360	t	2	350	t	2	328	t
1	323	t	1	317	t	1	305	t

125151
170:15:21:54
 $t = 177s$

125151
170:15:23:35
 $t = 278s$

125151
170:15:27:43
 $t = 526s$

Table 5.6

$(T_0 - T)$					$\frac{dT}{dt}$	q_t^{**}
t	Channel:	11/13	21/23	42/44		
[s]		[K]	[K]	[K]	[K/min]	[W]
0		0	0	0	--	--
20		-0.2/-0.2	-0.6/-0.3	-0.1/-0.1	+0.75	-4.0
41		-0.1/-0.2	-0.0/-0.2	0.0/-0.1	+0.15	-0.80
167		0.9/0.8	0.8/0.8	1.5/0.2	-0.30	1.59
177		1.0/0.9	0.8/0.9	1.6/0.3	-0.31	1.65
287		1.8/1.9	1.4/1.7	2.8/0.8	-0.36	1.91
526		3.5/4.3	2.8/3.8	5.1/2.2	-0.41	2.18

** A negative value of q_t indicates a heat gain

Table 5.7 Summary of Results for Insulated Model, Cooling Test --
Test 1.3

5.2 Natural Convection Data

The natural convection data which is obtained with the steady-state technique is clearly the most accurate type of data obtainable with the current model designs. These data clearly establish the approximate magnitude of the convective loss from the SRI International Line-Focus Solar Receiver for the limiting case of no wind. The loss, that is, the effective average heat transfer coefficient, is very small for this condition. Of course, the data which follow need to be converted to dimensionless quantities, specifically to a variation of Nusselt number with Grashof number, in order to estimate the corresponding loss from the prototype cavity receiver. Three series of natural convection tests were performed. The results from these tests and conclusions follow.

5.2.1 Test Series 2 -- Steady-State Data with $T_a \approx 295K$

Test Series 2 consists of three experiments: Tests 2.1, 2.2 and 2.3. The ambient temperature, T_a , for these three tests is 21.4C, 23.8C and 25.3C, respectively, and the total power, P_t , is 10W, 10W and 7.5W, respectively. All other parameters are identical. Since the steady-state technique is used in Series 2, a constant flux boundary condition is being simulated. Tables 5.8, 5.9 and 5.10 give for tests 2.1, 2.2 and 2.3, the test conditions, the original data, and the reduced data.

It is concluded from these results that:

- (i) The variation in the local value of h at any given θ location is $0.2W/m^2-K$ or less.
- (ii) The variation in h between the repeated runs (test 2.1 and 2.2) is $0.1W/m^2-K$ or less.

- (iii) The θ variation is logical considering the nature of the convective flow. The data from the $\theta = 0$ location, however, could contain appreciable error for the reasons previously outlined.
- (iv) The corrections for the heat conducted through the insulation and the temperature gradients in the calorimeter are appreciable -- of the order of 50%.
- (v) The transients experienced during the "steady-state" experiments have a negligible influence (see Table 5.10_b).

TEST 2.1 ORIGINAL DATA

TEST NUMBER 2.1

POWER = 10.00 W $T_A = 21.4^\circ\text{C}$

TEMPERATURE DISTRIBUTION $[\text{C}]$

HEAT TRANSFER COEFFICIENTS

UNCORRECTED VALUES $[\text{W}/\text{m}^2\text{-K}]$

4.9	5.3	5.2	5.0
4.5	4.7	4.6	4.5
5.9	6.2	5.9	5.9
8.7	8.8	8.7	8.6

1st ITERATION -- CORRECTED VALUES

3.1	3.4	3.4	3.2
2.9	3.0	2.9	2.8
3.9	4.1	3.9	3.8
5.8	5.9	5.9	5.8

2nd ITERATION

3.1	3.4	3.3	3.2
2.9	3.0	2.9	2.8
3.8	4.0	3.8	3.8
5.7	5.7	5.7	5.6

TEMPERATURE DISTRIBUTION

CORRECTED VERSION $[\text{C}]$

54.1	52.1	52.5	53.8
56.4	55.2	56.3	56.6
49.2	48.1	49.2	49.4
41.3	41.1	41.2	41.4

44	20	36.4	t
43		43.4	t
42		50.0	t
41	17	47.4	t
34	16	36.2	t
33		43.2	t
32		49.7	t
31	13	46.2	t
24	12	36.1	t
23		42.3	t
22		48.7	t
21	9	45.9	t
14	8	36.3	t
13		43.2	t
12		49.8	t
11	5	47.7	t
4	4	33.4	t
3		39.6	t
2		45.0	t
1	1	42.6	t

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TEST CONDITIONS:

$$T_a = 21.4^\circ\text{C}$$

$$\vec{V} = 0$$

$$Az = 90^\circ$$

$$P_k = 10\text{ W}$$

Table 5.8

TEST 2.2 ORIGINAL DATA

TEST NUMBER 2.2

POWER= 10.00 TA=23.8

TEMPERATURE DISTRIBUTION

50.0 48.5 48.5 49.8
52.1 51.2 51.9 52.3
45.5 44.7 45.4 45.7
38.7 38.5 38.5 38.7

HEAT TRANSFER COEFFICIENTS

UNCORRECTED VALUES

4.9 5.2 5.2 5.0
4.6 4.7 4.6 4.5
6.0 6.2 6.0 5.9
8.7 8.8 8.8 8.7

1
3.1 3.4 3.4 3.2
2.9 3.0 2.9 2.9
3.9 4.1 3.9 3.9
5.8 5.9 5.9 5.8

2
3.1 3.3 3.3 3.2
2.9 3.0 2.9 2.8
3.8 4.0 3.9 3.8
5.7 5.7 5.7 5.7

TEMPERATURE DISTRIBUTION

CORRECTED VERSION

56.4 54.8 54.8 56.2
58.7 57.7 58.5 58.9
51.4 50.5 51.3 51.7
43.7 43.5 43.5 43.7

41		23.8	t
34	↑ 10	38.7	t
33		45.7	t
32		52.3	t
31	↓ 17	49.8	t
24	↑ 16	38.5	t
23		45.4	t
22		51.9	t
21	↓ 12	48.5	t
14	↑ 12	38.5	t
13		44.7	t
12		51.2	t
11	↓ 1	48.5	t
4	↑ 8	38.7	t
3		45.5	t
2		52.1	t
1	↓ 5	50.0	t

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TEST CONDITIONS:

$T_a = 23.8^\circ\text{C}$

$\vec{V} = 0$

$A_z = 90^\circ$

$P_t = 10 \text{ W}$

Table 5.9

TEST NUMBER 2.3
POWER= 7.50 TA=25.3

TEST 2.3 ORIGINAL DATA

TEMPERATURE DISTRIBUTION

45.3 44.2 44.0 45.2
47.0 46.2 46.7 47.1
42.2 41.5 42.0 42.3
37.1 36.8 36.7 37.0

HEAT TRANSFER COEFFICIENTS

UNCORRECTED VALUES

4.8 5.1 5.2 4.9
4.5 4.6 4.5 4.4
5.7 6.0 5.8 5.7
8.2 8.4 8.5 8.3

1
3.1 3.3 3.3 3.1
2.8 2.9 2.8 2.8
3.7 3.9 3.8 3.7
5.5 5.6 5.7 5.5

2
3.1 3.3 3.3 3.1
2.8 2.9 2.8 2.8
3.7 3.9 3.7 3.7
5.4 5.5 5.5 5.4

TEMPERATURE DISTRIBUTION

CORRECTED VERSION

50.1 48.9 48.7 50.0
52.0 51.1 51.7 52.1
46.7 45.9 46.5 46.8
41.0 40.6 40.5 40.9

41	Ta	25.3	t
34	↑18	37.0	t
33		42.3	t
32		47.1	t
31	↑17	45.2	t
24	↑16	36.7	t
23		42.0	t
22		46.7	t
21	↑13	44.0	t
14	↑12	36.8	t
13		41.5	t
12		46.2	t
11	↑9	44.2	t
4	↑8	37.1	t
3		42.2	t
2		47.0	t
1	↑5	45.3	t

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TEST CONDITIONS:

$$T_a = 25.3 \text{ C}$$

$$\bar{V} = 0$$

$$Az = 90^\circ$$

$$P_t = 7.5 \text{ W}$$

INPUT DATA -- TESTS 2.1 THROUGH 2.3

1 \$NAM TEST=2.1,TA=21.4,TT=47.7,49.8,43.2,36.3,45.9,48.7,42.3,36.1,
2 46.2,49.7,43.2,36.2,47.4,50.0,43.4,36.4 \$
3 \$NAM TEST=2.2,TA=23.0,
4 TT=50.0,52.1,45.5,38.7,48.5,51.2,44.7,38.5,48.5,51.9,45.4,38.5,
5 49.8,52.3,45.7,38.7,23.8 \$
6 \$NAM TEST=2.3,TA=25.3,
7 TT=45.3,47.0,42.2,37.1,44.2,46.2,41.5,36.8,44.0,46.7,42.0,36.7,
8 45.2,47.1,42.3,37.0, P=7.504, \$

Table 5.10_a

TEST 2.3 ORIGINAL DATA SHOWING DEGREE OF STEADY-STATE

41	250	t
34	366	t
33	420	t
32	468	t
31	449	t
24	363	t
23	417	t
22	463	t
21	438	t
14	365	t
13	412	t
12	459	t
11	438	t
4	367	t
3	419	t
2	467	t
1	450	t

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41	250	t
34	367	t
33	419	t
32	467	t
31	449	t
24	364	t
23	417	t
22	463	t
21	438	t
14	365	t
13	412	t
12	459	t
11	438	t
4	367	t
3	419	t
2	466	t
1	450	t

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41	250	t
34	368	t
33	421	t
32	470	t
31	451	t
24	366	t
23	419	t
22	466	t
21	439	t
14	366	t
13	414	t
12	461	t
11	440	t
4	368	t
3	421	t
2	469	t
1	452	t

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41	251	t
34	368	t
33	421	t
32	470	t
31	451	t
24	366	t
23	419	t
22	466	t
21	439	t
14	367	t
13	415	t
12	462	t
11	441	t
4	369	t
3	421	t
2	469	t
1	452	t

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41	252	t
34	369	t
33	423	t
32	471	t
31	452	t
24	367	t
23	420	t
22	467	t
21	440	t
14	368	t
13	415	t
12	462	t
11	441	t
4	369	t
3	421	t
2	470	t
1	453	t

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41	252	t
34	370	t
33	423	t
32	471	t
31	452	t
24	367	t
23	421	t
22	467	t
21	441	t
14	368	t
13	415	t
12	463	t
11	442	t
4	370	t
3	422	t
2	470	t
1	453	t

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Table 5.10_b

5.2.2 Test Series 3 -- Transient Data with $T_a \approx 295K$

The transient technique gives results for the pure natural convection case which are dubious. This is due to the energy exchanged with the insulation and the smoothing influence of the conduction within the calorimeter. The transient results cannot be directly compared with the steady-state results. The former are for a prescribed temperature boundary condition whereas the latter are for a constant flux boundary condition.

The results for Test 3.1, the only test in Series 3, are given in Table 5.11. It should be noted in interpreting these results that the temperature differences $(T-T_a)$ are small, especially at the $\theta = 0$ location. Even at $t = 5s$, $(T-T_a)$ for the leading edge ($\theta = 0$ locations) are 6.4K, 6.1K, 6.2K and 5.8K. This problem stems from: (i) the desire to reduce the possibility of heater damage if the thermal contact with the calorimeter were poor and (ii) the strong steady-state dependence of h on θ .

THIS IS THE DATA FOR TEST RUN NUMBER 2.1

TABLE 5.11 Natural Convection Results,
Transient Technique

HEAT TRANSFER V/A : .00079 m^3/s

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 360.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THERMOCOUPLE NUMBER (SEE FIG. 1)

THE AVERAGE VELOCITY : 0.03 M/S -- NATURAL CONVECTION

SURFACE TEMPERATURE [K]

(HEAT TRANSFER
COEFFICIENT [W/m²-K])

B-60

TIME [S]	TA [K]	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	298.6	310.8 2.6	311.2 1.8	308.0 2.8	305.0 2.1	310.2 2.1	310.8 1.9	307.5 3.0	304.7 3.6	310.6 3.0	311.0 3.0	307.7 2.1	304.8 4.1	309.8 1.8	310.5 2.3	307.4 2.3	304.4 1.4
15.0	298.6	310.7 2.3	311.1 2.0	307.9 2.7	305.0 2.1	310.1 2.1	310.7 2.1	307.5 2.5	304.7 3.0	310.5 2.4	310.9 2.7	307.6 2.2	304.7 3.3	309.7 2.0	310.4 2.3	307.4 2.5	304.4 2.3
25.0	298.6	310.6 2.1	311.0 2.2	307.8 2.5	304.9 2.1	310.0 2.1	310.6 2.3	307.4 2.2	304.6 2.5	310.4 2.0	310.8 2.4	307.6 2.3	304.6 2.6	309.6 2.2	310.3 2.3	307.3 2.6	304.3 2.5
35.0	298.6	310.5 2.0	310.9 2.3	307.7 2.3	304.9 2.0	310.0 2.2	310.5 2.3	307.3 2.0	304.6 2.2	310.3 1.8	310.7 2.2	307.5 2.2	304.6 2.1	309.5 2.3	310.2 2.3	307.2 2.6	304.3 3.0
45.0	298.6	310.4 2.1	310.8 2.4	307.7 1.9	304.8 1.9	309.9 2.2	310.4 2.4	307.2 2.0	304.5 2.1	310.2 1.8	310.6 2.1	307.4 2.2	304.5 1.8	309.4 2.2	310.1 2.4	307.1 2.3	304.2 2.4
55.0	298.6	310.3 2.2	310.7 2.3	307.6 1.9	304.8 1.9	309.9 2.1	310.3 2.4	307.2 2.0	304.5 2.1	310.1 2.0	310.5 2.1	307.4 2.4	304.5 2.2	309.3 2.1	310.0 2.3	307.1 2.5	304.2 1.5
65.0	298.6	310.2 2.0	310.6 2.1	307.5 2.0	304.7 2.1	309.7 2.1	310.2 2.2	307.1 2.4	304.4 2.1	310.1 2.1	310.4 2.1	307.3 2.3	304.5 2.0	309.3 2.1	310.0 2.3	307.0 2.1	304.2 2.2
75.0	298.6	310.2 1.8	310.6 1.8	307.5 1.9	304.7 1.9	309.6 2.2	310.1 2.2	307.0 2.3	304.4 2.1	310.0 2.1	310.3 2.3	307.2 2.1	304.4 2.4	309.2 2.2	309.9 2.3	306.9 2.4	304.1 2.1
85.0	298.5	310.1 2.0	310.5 1.9	307.4 2.3	304.7 1.7	309.5 2.2	310.0 2.1	307.0 2.7	304.3 3.1	309.9 1.6	310.2 2.2	307.2 2.6	304.4 3.2	309.1 2.0	309.8 2.1	306.9 2.4	304.0 3.2
95.0	298.5	310.0 1.7	310.4 1.9	307.3 2.5	304.6 2.1	309.5 1.9	310.0 2.1	306.9 2.6	304.3 2.9	309.8 1.7	310.2 1.9	307.1 2.5	304.3 3.1	309.0 2.1	309.7 2.5	306.8 2.8	304.0 3.2
105.0	298.5	309.9 1.7	310.3 1.8	307.3 2.0	304.6 1.4	309.4 1.9	309.9 2.1	306.8 2.3	304.2 2.6	309.8 2.2	310.1 2.1	307.0 1.9	304.3 2.0	309.0 1.9	309.6 2.6	306.7 2.8	303.9 2.2
115.0	298.5	309.9 1.5	310.2 2.0	307.2 1.7	304.6 1.4	309.3 1.7	309.8 2.1	306.8 2.6	304.2 2.4	309.7 2.4	310.0 1.9	307.0 1.7	304.2 1.1	308.9 2.0	309.5 2.4	306.6 2.5	303.9 1.5
125.0	298.5	309.8 1.8	310.2 2.2	307.1 2.3	304.5 2.1	309.3 1.9	309.7 2.2	306.7 2.2	304.1 2.0	309.6 2.5	309.9 1.9	306.9 1.8	304.2 1.1	308.8 2.4	309.4 2.3	306.6 2.0	303.8 2.2
135.0	298.5	309.7 1.7	310.1 1.8	307.1 2.1	304.5 1.4	309.2 1.9	309.6 2.1	306.6 2.6	304.1 2.4	309.5 2.2	309.8 2.1	306.9 1.8	304.2 1.1	308.7 2.4	309.3 2.3	306.5 2.0	303.7 2.2

5.2.3 Test Series 7 -- Cryogenic, Steady-State Data

Test Series 7 consists of nine experiments. The power and the model azimuth angle are identical in all tests in the series, 10W and 90° respectively. The velocity for these natural convection experiments is, of course, zero. The only parameter varied between tests is the ambient temperature which was changed from 87.7K (-185.5C) to 235K (-38.2C) in increments of approximately 20 degrees Kelvin.

The results for Test Series 7 are given in Table 5.12.* It should be noted in interpreting these results that: (i) they are for the constant flux boundary condition; (ii) the correction applied to account for the influences of conduction through the insulation and for the temperature gradients in the calorimeter is approximately 50% of the uncorrected value; and (iii) the correction for the lower edge, the $\theta = 0$ location, is especially dubious. The $\theta = 0$ location shows an appreciable increase in the heat transfer coefficients with decreasing temperature (increasing Grashof number). For example, an uncorrected temperature difference ($T - T_a$) of only 4 to 5K exists with $T_a = -185.5C$ whereas a difference of approximately 14K is required to dissipate the same power with $T_a = -38.2C$. The corresponding temperature differences at the top of the cavity, the $\theta = 0$ location, are approximately 30K in both cases; hence, this area of the cavity shows heat transfer coefficients which are relatively independent of Grashof number. These results need to be ex-

* The original data was recorded only on magnetic tape. It was reduced with a program which is slightly different than the one given in Section 2.2 on page 36 in order to account for the different source and form of the original data.

amined more closely including the influences of the design of the model.

**TABLE 5.12 Results of Cryogenic Steady-State Experiments --
Test Series 7**

TEST NUMBER 7.1

POWER = 10.00W TA = -185.5C

TEMPERATURE DISTRIBUTION (C)

-158.8	-160.7	-159.4	-160.8
-155.8	-156.7	-155.3	-154.5
-172.4	-172.4	-171.0	-171.0
-181.3	-180.7	-180.3	-180.5

**HEAT TRANSFER COEFFICIENTS W/(M²-K)
UNCORRECTED VALUES**

4.8	5.3	5.0	5.2
4.4	4.5	4.2	4.2
9.9	9.9	8.9	8.9
30.8	26.9	24.8	25.8

CORRECTED VALUES

3.1	3.4	3.1	3.3
2.7	2.8	2.6	2.6
6.4	6.4	5.8	5.8
18.0	16.0	15.0	15.5

**TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION**

-152.3	-154.6	-153.0	-154.5
-149.1	-150.0	-148.2	-147.7
-167.7	-167.7	-166.0	-166.6
-178.6	-177.8	-177.3	-177.6

TEST NUMBER 7.2

POWER = 10.00W TA = -163.4C

TEMPERATURE DISTRIBUTION (C)

-137.1	-138.9	-137.2	-138.0
-134.0	-135.2	-133.0	-135.0
-147.8	-148.0	-146.3	-145.3
-156.2	-155.7	-155.1	-155.4

**HEAT TRANSFER COEFFICIENTS W/(M²-K)
UNCORRECTED VALUES**

4.9	5.3	4.9	5.2
4.5	4.6	4.3	4.3
8.3	8.4	7.6	7.6
18.1	16.9	15.7	17.3

CORRECTED VALUES

3.1	3.4	3.1	3.3
2.8	2.9	2.7	2.7
5.4	5.5	4.9	5.1
11.3	10.6	9.9	10.9

**TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION**

-130.7	-132.7	-130.6	-132.3
-127.9	-128.6	-126.9	-126.7
-142.7	-142.9	-141.0	-141.3
-152.7	-152.1	-151.3	-152.3

TEST NUMBER 7.3

POWER = 10.00W TA = -146.8C

TEMPERATURE DISTRIBUTION (C)

-119.8	-121.3	-119.9	-121.1
-117.5	-116.0	-116.6	-116.3
-130.6	-130.0	-129.7	-129.6
-138.4	-137.9	-137.4	-138.1

**HEAT TRANSFER COEFFICIENTS W/(M²-K)
UNCORRECTED VALUES**

4.8	5.1	4.8	5.0
4.4	4.3	4.3	4.2
7.7	7.7	7.1	7.3
15.4	14.5	13.7	14.9

CORRECTED VALUES

3.0	3.3	3.0	3.2
2.8	2.8	2.7	2.6
5.0	5.0	4.6	4.7
9.7	9.2	8.8	9.4

**TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION**

-113.3	-115.2	-113.4	-114.7
-110.8	-111.3	-109.0	-109.5
-124.7	-124.7	-123.2	-123.6
-134.6	-134.2	-133.4	-134.2

TEST NUMBER 7.4

POWER = 10.00W TA = -127.0C

TEMPERATURE DISTRIBUTION (C)

-97.8	-99.5	-97.9	-99.0
-95.5	-95.3	-94.6	-94.3
-108.0	-108.1	-106.7	-107.2
-116.0	-116.2	-115.7	-116.3

**HEAT TRANSFER COEFFICIENTS W/(M²-K)
UNCORRECTED VALUES**

4.4	4.7	4.4	4.6
4.1	4.2	4.0	4.0
6.8	6.6	6.4	6.5
12.7	12.0	11.4	12.1

CORRECTED VALUES

2.8	3.0	2.8	2.9
2.5	2.6	2.5	2.4
4.4	4.4	4.1	4.2
8.1	7.7	7.4	7.6

**TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION**

-91.1	-93.0	-91.2	-92.4
-88.6	-89.5	-87.6	-87.3
-102.4	-102.5	-100.9	-101.3
-112.5	-111.9	-111.3	-112.0

TABLE 5.12 (cont.)

TEST NUMBER 7.5
POWER = 10,00W TA = -105.9C

TEMPERATURE DISTRIBUTION (C)
-76.6 -78.4 -76.7 -77.7
-74.5 -75.3 -73.7 -73.4
-86.3 -86.4 -85.0 -85.4
-95.0 -94.4 -93.9 -94.3

HEAT TRANSFER COEFFICIENTS W/(M2-K)
UNCORRECTED VALUES
4.4 4.7 4.4 4.6
4.1 4.2 4.0 4.0
6.6 6.6 6.2 6.3
11.9 11.3 10.8 11.2

CORRECTED VALUES
2.8 3.0 2.8 2.9
2.6 2.6 2.5 2.5
4.3 4.3 4.0 4.1
7.7 7.3 7.0 7.2

TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION
-69.9 -71.9 -70.0 -71.1
-67.6 -68.5 -66.8 -66.4
-80.6 -80.7 -79.2 -79.6
-90.7 -90.0 -89.4 -89.8

TEST NUMBER 7.6
POWER = 10,00W TA = -84.5C

TEMPERATURE DISTRIBUTION (C)
-55.4 -57.1 -55.5 -55.5
-53.4 -54.1 -52.5 -52.3
-64.2 -64.3 -63.1 -63.3
-72.8 -72.3 -71.8 -72.0

HEAT TRANSFER COEFFICIENTS W/(M2-K)
UNCORRECTED VALUES
4.4 4.7 4.5 4.6
4.2 4.3 4.0 4.0
6.4 6.5 6.1 6.1
11.1 10.8 10.2 10.4

CORRECTED VALUES
2.8 3.0 2.8 2.9
2.6 2.6 2.5 2.5
4.1 4.2 4.0 4.0
7.2 6.9 6.6 6.7

TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION
-48.7 -50.6 -48.0 -49.9
-46.5 -47.3 -45.0 -45.4
-58.4 -58.8 -57.2 -57.4
-68.3 -67.7 -67.1 -67.4

TEST NUMBER 7.7
POWER = 10,00W TA = -65.6C

TEMPERATURE DISTRIBUTION (C)
-36.5 -38.1 -36.5 -37.0
-34.4 -35.2 -33.0 -33.5
-44.6 -45.1 -43.7 -43.9
-53.1 -52.0 -52.2 -52.4

HEAT TRANSFER COEFFICIENTS W/(M2-K)
UNCORRECTED VALUES
4.4 4.7 4.4 4.6
4.1 4.3 4.0 4.0
6.2 6.3 5.9 6.0
10.4 10.1 9.7 9.8

CORRECTED VALUES
2.8 3.0 2.8 2.9
2.6 2.7 2.5 2.5
4.0 4.1 3.8 3.9
6.7 6.6 6.3 6.4

TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION
-29.8 -31.0 -29.8 -31.0
-27.5 -28.4 -26.7 -26.6
-38.7 -39.3 -37.7 -37.6
-48.5 -48.1 -47.4 -47.7

TEST NUMBER 7.8
POWER = 10,00W TA = -45.0C

TEMPERATURE DISTRIBUTION (C)
-16.0 -17.5 -16.1 -17.1
-14.0 -14.7 -13.3 -13.2
-23.6 -24.0 -22.0 -22.4
-31.7 -31.0 -31.1 -31.1

HEAT TRANSFER COEFFICIENTS W/(M2-K)
UNCORRECTED VALUES
4.5 4.7 4.5 4.6
4.2 4.3 4.1 4.1
6.1 6.2 5.8 5.9
9.8 9.7 9.3 9.3

CORRECTED VALUES
2.8 3.0 2.8 2.9
2.6 2.7 2.5 2.5
3.9 4.0 3.8 3.8
6.3 6.3 6.1 6.1

TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION
-9.3 -11.0 -9.4 -10.5
-7.2 -7.9 -6.4 -6.3
-17.7 -18.1 -16.0 -16.0
-26.4 -26.0 -26.2 -26.2

TABLE 5.12 (cont.)

TEST NUMBER 7.9

POWER = 10.00W TAs = 33.2C

TEMPERATURE DISTRIBUTION (C)

-9.3	-14.7	-9.3	-10.6
-7.3	-8.9	-6.6	-6.3
-10.5	-17.8	-15.8	-15.8
-24.7	-24.6	-24.1	-24.1

HEAT TRANSFER COEFFICIENTS W/(M²-K)
UNCORRECTED VALUES

4.5	4.7	4.5	4.7
4.2	4.3	4.1	4.1
6.6	6.1	5.8	5.6
9.6	9.5	9.2	9.2

CORRECTED VALUES

2.8	3.8	2.8	2.9
2.6	2.7	2.5	2.5
3.9	4.6	3.7	3.7
6.3	6.2	6.0	6.8

TEMPERATURE DISTRIBUTION (C)
CORRECTED VERSION

-2.0	-4.2	-2.6	-3.8
-4.5	-1.2	-4.3	-4.4
-18.6	-11.1	-9.8	-9.8
-19.9	-19.8	-19.2	-19.2

5.3 Combined Convection Data

Since the model is not suitable for steady-state experiments when the heat transfer coefficients are large, all combined convection data were obtained using the transient technique. Two basic series of tests were conducted: Series 4, $T_a = 295K$ and Series 5, Cryogenic Tests ($80K < T_a < 92K$). The values of the main parameters for these tests are given in Table 5.13. The groups indicated in Table 5.13, e.g., 4.1 and 4.2; 5.7, 5.8 and 5.9, are essentially repeated tests. The main parameters changed between tests were the velocity $\bar{V}$ and the direction of $\bar{V}$ relative to the model, i.e., the azimuth angle AZ .

The results for all of the tests in Series 4 and 5 are given in Appendix A which begins on page 27. The following points should be considered in interpreting these results.

- (i) Since it was not possible to begin the test with an isothermal model due to the natural convection during the heating mode, and due to the axial placement of the individual heaters, conduction of heat within the calorimeter has a significant influence on the results. The heat exchanged with the calorimeter substrate was also of importance, but this effect had a lesser influence. The heat transfer coefficient was calculated from Eq. (11) which is

$$h = \frac{-(\rho c_p \delta)_m \frac{dT}{dt}}{(T - T_a)} \quad (11)$$

Test Number	T_a [K]	$\bar{V}$ [m/s]	AZ [Deg]	$(T_5 - T_a)$ [K]	$(T_8 - T_a)$ [K]
				@ $t = 5_s$	@ $t = 5_s$
4.1	298.5 ± 0.1	3.03	0	19.2	10.1
4.2	292.5 ± 0.2	2.99	0	31.2	19.3
4.3	300.5 ± 0.7	7.93	0	17.0	7.5
4.4	296.0 ± 1.0	7.88	0	25.5	12.1
4.5	298.2 ± 0.6	7.86	180	26.7	16.6
4.6	290.8 ± 1.0	7.84	180	32.8	19.1
				@ $t = 0$	@ $t = 0$
5.1	88.9 ± 2.7	4.64	0	39.8	10.7
5.2	84.3 ± 0.8	4.84	0	49.8	12.4
5.3	84.2 ± 1.7	5.95	0	97.3	23.6
5.4	81.0 ± 1.6	8.78	0	56.4	16.0
5.5	91.4 ± 1.0	9.57	0	86.8	19.6
5.6	82.1 ± 0.7	5.39	180	48.9	13.0
5.7	84.0 ± 1.2	9.29	180	51.8	10.3
5.8	80.3 ± 0.1	9.36	180	56.9	16.2
5.9	84.8 ± 2.7	9.73	180	81.0	21.2
5.10	80.5 ± 0.1	9.25	45	89.6	22.5

TABLE 5.13 Parameters Used in Combined Convection Experiments
-- Test Series 4 and 5

If the heat exchanged by the calorimeter thermocouple area with the insulation and other portions of the calorimeter is denoted as q_L , h should be calculated from

$$h = \frac{-(\rho c_p \delta)_m \frac{dT}{dt} - \frac{q_L}{A}}{(T - T_a)} \quad (12)$$

Unfortunately, q_L is an unknown function of time. The portion of the calorimeter which is directly heated by the foil heaters is hotter than the unheated area where the thermocouples are located; hence, conduction within the calorimeter results in a heat gain. This gain provides a negative component to q_L . On the other hand, the heat transfer through the insulation results in a heat loss, i.e., a positive component at small values of time. It was shown in Part I that the time constants are:

$$\tau_m \Big|_{h=50} = 57s \quad ; \quad \tau_m \Big|_{h=20} = 143s$$

$$\frac{\delta_i^2}{\alpha_i} = 1192s$$

and
$$\frac{(25mm)^2}{\alpha_m} = 161s$$

Thus, at times less than approximately $100s$, h is underestimated. The degree of underestimation will decrease with increasing time.

The large values of the heat transfer coefficients at

at cryogenic temperature had two consequences. First, it was not possible to wait more than approximately 30 seconds before initiating the test if a sufficiently large temperature difference were to be retained. Second, the high combined convection values of h resulted in short test runs. Hence, the results from Series 5 show a marked increase in h with time. On the other hand, when T_a was near room temperature it was possible to wait approximately 90 seconds before initiating the convective flow. In addition, the calorimeter cooled much slower due to considerably smaller values of h ; thus, the results for Series 4 are relatively independent of time and are more accurate.

- (ii) The boundary condition represented by these results is a prescribed temperature boundary condition. However, the specimen is far from being isothermal as is clearly seen by a comparison of the last two columns of Table 5.13.
- (iii) There is some evidence that there may be a systematic error in T_a which is of the order of 1K (see Test Series 5.4 at a time of 30 seconds). This error is mainly of importance if ΔT is small and appears to only be a viable source of error in the cryogenic tests.

A more complete analysis of the results and comparisons of the results in terms of Nusselt number, Reynolds number and Grashof number should be effected. Much has already been learned, but the analysis of the data has only just begun. Conclusions should not be drawn until the analysis of the data is completed.

APPENDIX A -- Results of Combined Convection Experiments

Notes regarding results which follow in this appendix:

1. All results were obtained with the transient technique.
2. The test numbers on the top of each page identify the specific experiments and correspond to those used in Table 5.13. The test numbers range from 4.1 through 4.6 ($T_a \approx 295K$) and from 5.1 through 5.10 ($T_a < 93K$).
3. The column headings 5, 6, ...20 are the thermocouple numbers from which the corresponding data were obtained; hence, they specify the z and θ coordinates (see Fig. 1, page 31).
4. The first column, "Time", contains the time in seconds; the second column, "TA", is the ambient temperature in degrees Kelvin; and columns "5" through "20" give the temperature of the heat transfer surface, T , in degrees Kelvin followed by the uncorrected total heat transfer coefficient, h_t , in (W/m^2-K). The parameter given at the top of the page labeled "V/A" is the thickness of the calorimeter, δ_m ($\delta_m = 0.7938mm$).

THIS IS THE DATA FOR TEST RUN NUMBER 4.1
HEAT TRANSFER V/A : .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THE AVERAGE VELOCITY : 3.03 M/S $A2 = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	298.5	317.7 11.4	319.2 12.4	314.0 14.0	308.4 20.0	316.1 10.8	317.6 13.7	313.2 13.7	307.8 20.6	316.9 11.0	318.3 12.8	313.4 13.8	307.8 21.1	316.6 10.8	318.3 14.3	313.2 16.1	307.5 24.8
15.0	298.5	317.0 11.4	318.3 12.6	313.2 14.6	307.8 20.5	315.5 11.5	316.7 13.9	312.4 14.5	307.1 22.0	316.1 11.9	317.4 13.7	312.7 14.8	307.1 21.9	315.9 11.5	317.3 14.8	312.4 16.6	307.1 22.8
25.0	298.5	316.3 11.4	317.4 12.6	312.5 14.8	307.1 20.5	314.8 12.0	315.8 14.1	311.8 14.9	306.4 22.9	315.4 12.4	316.5 14.2	311.9 15.4	306.5 22.3	315.2 11.9	316.4 15.1	311.6 16.7	306.5 21.9
35.0	298.5	315.6 11.3	316.6 12.5	311.8 14.0	306.5 19.6	314.1 12.1	315.0 14.1	311.1 10.7	305.8 22.7	314.7 12.1	315.7 13.7	311.2 15.1	305.9 22.1	314.5 11.7	315.5 14.6	310.9 16.0	305.5 22.3
45.0	298.5	314.9 11.6	315.8 12.4	311.2 14.3	306.0 19.0	313.5 12.1	314.2 14.4	310.5 14.5	305.3 22.7	314.0 12.0	314.9 13.7	310.6 15.0	305.3 21.2	315.9 11.4	314.6 14.4	315.2 15.3	305.2 22.2
55.0	298.5	314.3 11.9	315.1 12.4	310.6 14.0	305.5 19.3	312.9 12.1	313.4 14.0	309.9 15.5	304.8 22.6	313.4 11.7	314.1 13.8	310.0 14.8	304.8 20.9	315.3 11.6	315.9 14.4	309.6 15.6	304.8 22.1
65.0	298.5	313.6 11.5	314.4 12.3	310.0 14.1	305.1 18.8	312.3 11.7	312.7 13.8	309.3 15.5	304.3 20.9	312.8 11.6	313.4 13.5	309.4 14.9	304.4 20.5	312.7 11.7	313.1 14.3	309.0 16.9	304.4 21.6
75.0	298.5	313.1 11.2	313.7 12.5	309.4 13.8	304.7 18.1	311.7 11.5	312.1 13.5	308.7 15.4	303.9 19.7	312.2 12.0	312.7 13.3	308.9 15.0	304.0 21.0	312.1 11.6	312.4 14.3	308.4 16.6	304.4 21.8
85.0	298.0	312.5 11.3	313.1 12.5	308.9 13.7	304.3 16.9	311.2 11.2	311.5 13.6	308.2 15.4	303.6 19.2	311.7 12.0	312.1 13.2	308.3 15.3	303.7 18.6	311.6 11.4	311.7 14.1	307.9 16.5	303.6 20.1
95.0	298.0	312.0 11.4	312.5 12.2	308.4 14.1	304.0 15.8	310.7 11.2	310.9 13.3	307.7 15.5	303.3 19.5	311.2 11.8	311.5 13.3	307.8 15.3	303.3 17.8	311.1 11.8	311.1 14.4	307.4 15.8	303.3 19.1
105.0	298.0	311.5 11.0	311.9 12.2	308.0 14.2	303.7 16.0	310.3 11.4	310.3 13.3	307.2 15.3	303.0 18.7	310.6 12.0	310.9 13.8	307.3 15.7	303.0 19.2	310.6 12.0	310.5 14.2	306.9 16.8	303.1 18.5
115.0	298.0	311.0 11.1	311.4 12.1	307.5 14.4	303.4 15.6	309.8 11.2	309.6 12.7	306.8 15.0	302.7 17.9	310.2 12.1	310.3 13.7	306.9 15.0	302.8 18.2	310.1 11.9	309.9 14.0	306.4 17.1	302.7 18.0
125.0	298.0	310.5 10.8	310.9 12.0	307.1 14.4	303.2 15.2	309.4 11.2	309.3 12.9	306.3 13.5	302.4 19.4	309.7 11.9	309.8 13.4	306.5 14.6	302.5 18.2	309.6 11.6	309.4 14.1	306.0 17.3	302.7 19.0

THIS IS THE DATA FOR TEST RUN NUMBER 41
HEAT TRANSFER V/A 1 .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THE AVERAGE VELOCITY IS 2.99 M/S $Az = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	292.4	323.6 10.1	327.1 11.2	320.2 12.5	311.7 20.8	321.2 10.7	325.5 13.5	319.3 13.0	311.0 22.5	323.0 10.9	327.3 12.5	320.5 14.1	311.2 21.0	322.3 9.1	321.3 13.5	320.5 15.8	311.3 20.1
15.0	292.5	322.5 10.5	325.7 11.4	319.0 13.4	310.3 21.4	320.1 10.8	323.9 13.5	318.1 13.7	309.6 22.8	321.9 11.2	325.8 12.4	319.1 14.7	309.4 21.4	321.3 9.4	323.8 13.9	319.0 15.9	310.2 21.0
25.0	292.6	321.4 10.8	324.4 11.5	317.7 13.9	309.0 21.6	319.1 11.0	322.5 13.6	316.9 14.3	308.3 23.0	320.8 11.4	324.3 13.2	317.8 15.0	308.6 21.7	320.3 10.5	324.2 14.2	317.6 16.0	309.0 21.6
35.0	292.6	320.3 10.6	323.2 11.4	316.6 14.1	307.4 21.4	318.1 11.3	321.1 13.6	315.7 14.6	307.1 23.0	319.7 11.7	322.9 13.4	316.5 14.7	307.4 22.1	319.3 10.4	322.7 14.0	316.2 15.8	307.0 21.4
45.0	292.7	319.3 10.9	322.0 11.7	315.4 13.4	306.8 20.4	317.1 11.5	314.8 13.8	314.5 14.9	306.0 22.0	318.6 12.0	321.5 13.4	315.3 14.9	306.3 22.0	318.3 10.9	321.3 14.0	315.0 16.1	306.7 21.7
55.0	292.7	318.3 11.0	320.8 11.4	314.3 13.9	305.8 20.2	316.1 11.3	318.5 14.8	313.4 15.2	305.0 22.3	317.5 11.8	320.2 13.3	314.1 15.2	305.4 21.5	317.4 11.1	314.4 14.0	313.7 16.4	305.7 21.0
65.0	292.7	317.4 11.0	319.4 12.3	313.3 14.3	304.9 20.2	315.2 11.1	317.3 13.0	312.4 15.1	304.1 22.3	316.5 11.5	319.0 13.4	313.0 15.3	304.4 21.0	316.4 11.6	318.6 14.1	312.6 16.4	304.8 20.5
75.0	292.7	316.4 11.1	318.5 12.1	312.5 14.2	304.1 20.3	314.4 11.2	316.2 13.4	311.4 14.7	303.3 21.4	315.6 11.5	317.8 13.8	312.0 14.8	303.6 21.2	315.5 11.6	317.4 14.2	311.5 16.2	303.4 20.8
85.0	292.7	315.5 11.2	317.5 12.1	311.4 14.0	303.3 19.3	313.5 11.1	315.1 13.1	310.4 14.4	302.5 20.6	314.7 11.8	316.6 13.5	311.0 14.9	302.8 20.9	314.6 11.5	318.2 14.3	310.4 16.3	303.1 21.1
95.0	292.7	314.7 11.0	316.4 11.7	310.5 13.7	302.6 18.4	312.8 11.0	314.1 13.1	309.6 14.5	301.8 20.2	315.8 12.1	315.5 13.4	310.1 15.4	302.1 20.1	313.7 11.4	315.1 14.1	304.5 16.7	302.4 21.6
105.0	292.7	313.8 10.8	315.5 11.7	309.7 13.4	302.0 17.4	312.0 11.1	315.1 13.5	308.7 15.5	301.2 20.7	315.0 11.9	314.5 13.2	309.2 15.7	301.5 19.7	312.9 11.3	314.0 13.8	300.5 16.4	301.7 19.0
115.0	292.7	313.7 10.6	314.5 11.7	308.9 13.7	301.5 17.0	311.3 11.3	312.2 13.5	307.9 15.9	300.6 20.4	312.2 11.5	313.5 13.1	308.5 15.5	300.4 19.1	312.2 11.0	313.0 13.4	307.6 16.1	301.1 19.7
125.0	292.7	312.3 11.1	313.7 12.1	308.1 14.4	301.0 17.6	310.5 11.2	311.3 13.0	307.1 15.6	300.1 19.8	311.4 11.1	312.6 13.0	307.5 14.9	300.3 19.1	311.4 11.1	312.1 13.4	306.8 16.1	300.6 19.0

THIS IS THE DATA FOR TEST RUN NUMBER 4.3

HEAT TRANSFER V/A : .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THE AVERAGE VELOCITY : 7.53 M/S $Az = 0^\circ$

TIME [S]	TA [K]	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	299.8	316.8 21.1	318.5 21.6	313.8 24.5	307.3 37.2	315.0 20.3	316.3 25.9	312.9 27.7	306.5 46.6	315.8 22.4	317.4 25.5	313.0 28.9	306.7 44.9	315.8 21.5	317.2 27.9	312.6 32.4	306.6 46.1
15.0	299.9	315.6 21.3	317.1 22.6	312.6 25.3	306.4 37.6	313.9 21.6	314.9 26.3	311.7 27.9	305.5 44.3	314.6 23.1	315.9 26.7	311.7 30.1	305.7 45.2	314.6 22.6	315.6 28.3	311.2 32.8	305.6 46.7
25.0	300.1	314.5 21.4	315.8 23.1	311.6 25.8	305.6 36.9	312.9 22.4	313.6 26.4	310.6 28.2	304.7 41.9	313.4 23.5	314.5 27.3	310.6 30.7	304.9 44.5	313.5 23.4	314.1 28.6	310.0 33.0	304.8 46.4
35.0	300.2	313.5 21.1	314.6 22.8	310.6 25.7	305.0 33.4	311.9 22.0	312.4 26.0	309.6 28.4	304.1 39.9	312.4 22.9	313.2 26.8	309.5 30.3	304.2 41.9	312.4 23.4	312.8 28.6	308.9 32.8	304.1 44.7
45.0	300.3	312.5 21.1	313.5 22.4	309.7 25.9	304.5 31.6	311.1 22.0	311.4 25.7	308.8 28.8	303.6 41.5	311.4 23.1	312.0 25.9	308.6 30.3	303.7 38.4	311.5 23.0	311.6 28.3	308.0 32.8	303.6 41.3
55.0	300.4	311.7 21.3	312.5 22.7	308.9 26.4	304.1 30.3	310.3 21.9	310.4 25.8	308.0 29.0	303.2 40.7	310.6 23.1	311.1 25.3	307.8 29.5	303.3 33.9	310.7 22.5	310.6 27.4	307.2 32.2	303.1 39.6
65.0	300.6	310.9 21.1	311.6 23.2	308.1 26.8	303.7 30.1	309.6 22.0	309.6 25.8	307.2 29.0	302.8 38.8	309.8 22.8	310.2 25.9	307.1 30.2	303.0 35.5	309.9 22.5	309.7 27.0	306.5 32.0	302.8 40.3
75.0	300.7	310.1 21.1	310.8 22.5	307.5 26.1	303.4 26.8	308.9 22.5	308.8 26.2	306.6 30.0	302.6 34.9	309.1 23.0	309.3 26.3	306.4 30.3	302.7 35.9	309.2 23.2	308.9 27.6	305.9 32.1	302.5 37.5
85.0	300.8	309.5 21.0	310.0 21.3	306.9 24.3	303.2 23.5	308.3 22.6	308.1 26.1	306.0 29.8	302.4 35.3	308.5 23.8	308.6 26.3	305.9 28.9	302.5 34.6	308.5 24.1	308.1 28.4	305.3 32.7	302.3 32.1
95.0	300.9	308.9 20.5	309.4 21.6	306.4 24.1	303.0 21.1	307.7 22.1	307.5 25.8	305.5 27.9	302.2 27.7	307.8 23.9	307.9 26.0	305.4 29.4	302.3 29.3	307.9 24.1	307.4 28.3	304.8 32.5	302.1 29.7
105.0	301.0	308.3 20.7	308.7 22.8	306.0 24.9	302.9 20.3	307.2 21.0	307.0 23.9	305.1 24.8	302.1 25.3	307.3 23.1	307.3 25.3	305.0 27.9	302.2 24.4	307.4 25.0	306.8 27.8	304.4 31.5	302.0 38.7
115.0	301.1	307.8 21.2	308.2 22.2	305.6 25.5	302.7 22.8	306.8 20.7	306.5 23.4	304.8 25.8	302.0 16.8	306.8 21.5	306.8 25.2	304.6 29.1	302.1 19.7	306.8 24.1	306.3 28.4	304.1 31.3	301.9 44.3
125.0	301.2	307.3 22.1	307.6 22.8	305.2 23.9	302.6 16.7	306.4 21.0	306.1 24.2	304.5 26.6	302.0 20.4	306.4 22.0	306.3 25.2	304.3 27.7	302.0 15.3	306.4 24.1	305.8 29.5	303.8 30.1	301.8 34.1

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THIS IS THE DATA FOR TEST RUN NUMBER 4.4

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THE AVERAGE VELOCITY : 7.88 M/S $Az = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	295.0	320.5 20.6	322.9 22.4	316.1 25.5	307.1 40.6	318.2 21.5	320.7 26.4	315.3 26.6	306.3 45.7	319.5 21.5	322.2 25.0	315.7 28.1	306.4 43.3	319.4 20.3	322.1 26.2	315.2 30.2	306. 43.
15.0	295.2	318.7 21.3	320.8 22.7	314.3 25.7	305.5 39.5	316.5 22.4	318.4 26.9	313.5 27.9	304.7 45.8	317.7 22.6	319.9 26.0	313.7 28.7	304.8 43.8	317.7 21.6	319.7 27.3	313.2 31.2	305. 44.
25.0	295.4	317.0 21.8	318.9 23.1	312.7 25.8	304.2 38.2	314.9 22.8	316.3 27.1	311.8 28.7	303.3 45.2	316.0 23.4	317.3 26.6	312.0 29.2	303.5 43.5	316.1 22.5	317.5 28.1	311.3 31.9	303. 44.
35.0	295.6	315.5 21.6	317.1 22.9	311.2 25.7	303.2 37.1	313.4 22.2	314.5 26.5	310.2 28.7	302.2 42.2	314.4 23.3	315.3 26.3	310.4 29.1	302.4 41.6	314.5 22.6	315.4 28.1	309.6 31.8	302. 44.
45.0	295.8	314.0 21.7	315.4 23.5	309.9 26.6	302.3 35.3	312.2 21.4	312.8 26.2	308.8 28.0	301.3 41.5	313.0 22.9	314.1 26.1	308.9 29.5	301.5 40.3	313.1 22.2	313.6 27.4	308.2 31.7	301. 42.
55.0	295.9	312.7 21.9	313.9 23.3	308.6 27.0	301.5 34.4	311.0 21.2	311.4 25.8	307.7 27.7	300.6 41.0	311.6 23.1	312.5 25.8	307.7 29.8	300.8 39.1	311.8 22.0	312.0 26.7	306.9 31.3	300. 40.
65.0	296.1	311.5 22.2	312.5 23.2	307.5 26.6	300.9 33.2	309.9 22.1	310.1 25.6	306.6 28.4	300.0 39.1	310.4 23.1	311.1 26.1	306.5 29.7	300.2 37.8	310.6 22.0	310.6 27.0	305.8 30.6	300. 39.
75.0	296.2	310.4 21.1	311.3 22.5	306.5 25.2	300.4 29.6	308.9 21.7	308.9 25.4	305.6 28.6	299.5 34.5	309.3 23.5	309.8 25.8	305.5 29.1	299.7 36.9	309.6 22.9	309.3 27.7	304.8 30.9	299. 37.
85.0	296.3	309.4 20.5	310.2 21.0	305.7 24.9	300.0 27.4	308.0 21.5	307.8 25.0	304.7 28.2	299.2 32.5	308.3 23.3	308.6 25.8	304.6 29.1	299.3 34.3	308.5 23.0	308.1 27.7	304.0 31.1	299. 36.
95.0	296.5	308.5 20.7	309.2 22.1	304.9 25.2	299.7 24.6	307.2 21.6	306.9 24.9	303.9 27.9	298.9 31.6	307.4 23.0	307.6 26.2	303.8 30.0	299.0 33.1	307.6 22.9	307.0 27.2	303.2 30.5	298. 35.
105.0	296.6	307.6 21.7	308.2 22.6	304.2 25.8	299.5 26.1	306.4 22.3	305.0 25.1	303.3 28.2	298.6 34.9	306.6 22.6	306.6 26.2	303.1 29.6	298.7 28.5	306.7 23.6	306.1 27.1	302.5 30.2	298. 35.
115.0	296.8	306.9 22.3	307.4 22.9	303.5 26.4	299.2 27.2	305.7 21.7	305.2 25.4	302.7 27.4	298.4 32.5	305.8 22.7	305.6 25.1	302.5 28.4	298.6 25.7	306.0 23.5	305.2 27.4	301.9 30.1	298. 33.
125.0	296.9	305.1 21.4	306.6 22.6	303.0 24.5	299.0 23.1	305.0 21.4	304.5 25.4	302.1 27.1	298.3 29.6	305.1 23.2	305.0 24.2	302.0 26.9	298.4 23.5	305.2 23.6	304.5 27.2	301.4 30.8	298. 33.

THIS IS THE DATA FOR TEST RUN NUMBER 4.5

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S)

THE AVERAGE VELOCITY : 7.86 m/s $AZ = 180^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	297.7	324.4 17.3	327.3 17.9	321.9 17.7	314.3 23.2	321.9 18.9	324.7 23.7	320.2 21.8	313.3 28.3	323.2 20.0	325.9 25.3	320.3 26.1	312.8 31.7	323.0 17.2	326.8 20.5	321.5 20.4	314.0 21.1
15.0	297.8	322.8 18.1	325.5 18.6	320.4 18.5	313.0 23.9	320.3 19.8	322.6 24.0	318.5 22.8	311.8 29.0	321.5 20.8	323.5 25.7	318.3 27.1	311.3 32.2	321.5 18.2	324.8 21.2	319.9 20.3	312.8 22.4
25.0	297.9	321.3 18.6	323.8 18.9	319.0 19.0	311.8 24.0	318.8 20.0	320.6 24.2	316.9 23.4	310.5 29.6	319.8 21.3	321.3 25.9	316.4 27.8	309.8 32.6	320.0 19.0	322.8 21.7	318.4 20.4	311.7 23.2
35.0	298.0	319.8 18.3	322.1 18.4	317.6 18.8	310.7 22.8	317.4 20.1	318.8 24.2	315.5 23.2	309.2 29.6	318.3 21.4	319.3 25.8	314.7 27.5	308.6 32.8	318.6 19.4	321.0 21.8	317.0 20.7	310.6 23.5
45.0	298.1	318.5 18.1	320.7 18.0	316.4 18.2	309.7 22.4	316.1 19.8	317.1 24.3	314.1 22.8	308.1 29.8	316.8 21.4	317.5 26.2	313.2 27.4	307.4 33.3	317.2 19.2	319.3 22.0	315.7 21.0	309.7 23.2
55.0	298.2	317.2 18.4	319.3 18.3	315.3 18.8	308.9 22.6	315.0 19.4	315.6 23.9	312.9 22.9	307.2 29.3	315.5 21.3	315.8 26.1	311.8 27.8	306.4 33.4	316.0 19.2	317.8 22.2	314.4 21.0	308.8 23.1
65.0	298.3	316.0 18.7	318.0 18.8	314.2 19.2	308.0 23.5	313.9 19.3	314.2 22.6	311.8 23.1	306.3 27.9	314.2 21.4	314.3 25.4	310.6 27.4	305.5 31.9	314.8 19.4	316.3 22.0	313.3 21.0	307.9 23.7
75.0	298.4	314.9 18.5	316.7 18.7	313.2 19.0	307.3 22.9	312.9 19.4	313.0 22.6	310.7 23.3	305.6 28.4	313.1 21.3	312.9 24.9	309.5 26.4	304.8 29.0	313.8 19.4	315.0 22.0	312.2 21.2	307.2 24.1
85.0	298.5	313.9 18.0	315.6 18.5	312.2 19.9	306.6 21.5	311.9 19.1	311.9 23.0	309.8 23.1	304.9 28.4	312.1 20.9	311.7 25.2	308.5 26.4	304.2 28.5	312.8 19.5	313.8 21.8	311.2 21.6	306.5 23.9
95.0	298.5	313.0 18.5	314.5 18.2	311.4 19.2	306.0 21.8	311.1 19.0	310.9 22.5	308.9 22.9	304.3 26.6	311.1 20.5	310.6 24.7	307.6 26.5	303.7 30.0	311.8 19.3	312.7 21.6	310.3 21.6	305.8 24.0
105.0	298.6	312.1 18.1	313.6 17.9	310.6 19.6	305.5 20.8	310.3 19.0	310.0 21.5	308.1 23.1	303.8 26.6	310.3 20.8	309.7 24.3	306.8 27.1	303.1 31.6	311.0 19.7	311.7 21.5	309.5 21.5	305.2 24.4
115.0	298.7	311.3 16.9	312.7 17.5	309.8 17.2	305.1 25.4	309.5 19.1	309.2 21.6	307.4 23.0	303.4 27.8	309.4 21.0	308.8 24.1	306.1 27.2	302.7 28.2	310.2 19.7	310.7 22.2	308.7 22.2	304.7 25.2
125.0	298.8	310.6 17.6	311.8 17.7	309.2 19.0	304.6 21.1	308.8 20.2	306.4 22.1	306.7 23.6	302.9 28.8	308.7 21.3	308.0 24.3	305.4 26.1	302.3 28.3	309.4 20.0	309.9 21.7	307.9 22.5	304.2 23.2

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THIS IS THE DATA FOR TEST RUN NUMBER 4.6

HEAT TRANSFER V/A : .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 5.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 125.0 SECOND(S) WITH A TIME INCREMENT OF 10.0 SECOND(S).

THE AVERAGE VELOCITY : 7.84 M/S $AZ = 180^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5.0	289.8	322.6	326.0	318.3	308.9	319.8	321.2	316.5	307.7	321.3	324.7	316.8	307.5	321.2	326.0	318.5	309.
		18.6	18.6	18.3	24.2	19.7	25.7	23.1	29.8	21.1	26.4	27.3	32.7	18.1	22.3	20.8	22.
15.0	290.1	320.6	323.8	316.5	307.3	317.8	320.4	314.4	306.0	319.1	321.6	314.3	305.6	319.3	323.4	316.5	307.
		18.6	18.8	18.5	24.4	20.0	25.1	23.1	30.2	21.3	26.5	27.2	32.6	18.6	21.9	20.4	22.
25.0	290.3	318.7	321.7	314.9	306.0	316.0	317.9	312.6	304.4	317.1	318.8	312.2	304.0	317.4	320.9	314.7	306.
		18.5	18.9	18.7	24.3	20.0	24.5	23.0	30.3	21.3	26.5	27.0	32.5	18.9	21.7	20.1	22.
35.0	290.5	316.9	319.7	313.3	304.7	314.2	315.7	310.9	303.0	315.1	316.4	310.2	302.5	315.7	318.7	313.1	305.
		18.5	18.7	18.6	23.8	19.7	24.0	23.0	29.6	21.1	26.1	26.9	32.2	19.0	21.6	20.0	22.
45.0	290.6	315.3	317.8	311.9	303.6	312.7	313.7	309.3	301.8	313.4	314.2	308.5	301.3	314.1	316.7	311.6	304.
		17.7	18.7	18.4	23.5	19.4	23.9	23.1	29.1	21.2	25.5	26.9	32.0	18.9	21.7	20.3	22.
55.0	290.8	313.8	316.1	310.6	302.6	311.3	311.8	307.9	300.7	311.8	312.2	306.9	300.2	312.6	314.8	310.1	303.
		18.5	19.0	18.9	23.2	19.4	23.8	23.5	28.9	21.3	25.4	26.9	32.1	18.9	21.6	20.9	23.
65.0	291.0	312.4	314.5	309.3	301.7	309.9	310.2	306.5	299.8	310.3	310.4	305.4	295.2	311.2	313.1	308.8	302.
		18.6	18.9	18.7	22.5	19.3	23.4	23.7	28.9	21.6	25.7	27.3	31.2	19.0	21.8	20.9	23.
75.0	291.1	311.1	313.0	308.2	300.9	308.7	308.7	305.3	299.0	308.9	308.7	304.1	296.4	309.9	311.5	307.5	301.
		18.5	19.0	19.0	21.5	19.4	22.9	24.1	28.3	21.1	25.4	27.1	30.5	19.3	21.5	20.9	22.
85.0	291.2	309.8	311.7	307.1	300.2	307.5	307.4	304.2	298.2	307.6	307.2	303.0	297.7	308.1	310.0	306.4	300.
		18.2	18.7	19.6	21.6	19.8	22.3	23.6	27.4	21.5	24.8	26.8	30.1	19.4	21.6	20.9	23.
95.0	291.4	308.7	310.4	306.0	299.6	306.5	306.1	303.2	297.6	306.5	305.9	302.0	297.0	307.6	308.6	305.3	299.
		18.2	18.3	19.8	21.6	19.8	23.0	22.8	26.3	21.5	24.6	26.7	30.5	19.4	21.9	21.1	23.
105.0	291.5	307.6	309.2	305.1	299.0	305.5	305.0	302.3	297.1	305.4	304.7	301.0	296.5	306.5	307.4	304.3	299.
		18.7	18.6	20.1	21.6	19.5	23.0	22.8	27.7	21.5	24.9	26.3	30.2	20.0	21.8	21.2	22.
115.0	291.6	306.6	308.1	304.2	298.5	304.6	304.0	301.5	296.6	304.4	303.8	300.2	296.0	305.5	306.2	303.4	298.
		18.7	18.6	19.4	20.3	19.5	23.1	23.8	26.3	21.0	25.2	26.7	27.9	20.0	21.8	21.7	21.
125.0	291.7	305.7	307.1	303.4	298.0	303.7	303.0	300.7	296.1	303.5	302.8	299.4	295.6	304.6	305.2	302.6	298.
		18.7	18.5	18.8	19.3	19.9	22.8	23.6	26.2	21.1	24.9	26.2	26.9	20.3	21.7	21.6	22.

THIS IS THE DATA FOR TEST RUN NUMBER 5.1

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0 SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 4.64 M/S

$A_2 = 0^\circ$

TIME [S]	TA [K]	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	86.2	126.0 18.9	130.2 21.5	108.0 25.6	96.9 40.8	123.8 20.8	129.9 26.5	108.3 26.6	97.2 44.3	125.4 20.6	131.7 25.0	109.9 30.8	98.1 40.7	123.6 15.5	132.0 24.2	109.7 31.3	97.9 39.0
2.5	86.9	125.4 20.4	129.4 23.1	107.5 27.1	96.5 43.0	123.2 22.2	128.9 28.0	107.8 28.0	96.8 46.4	124.7 21.9	130.7 26.4	109.2 32.2	97.7 42.4	123.1 16.9	131.0 25.6	109.1 32.7	97.5 40.8
5.0	87.5	124.7 21.9	128.5 24.6	107.0 28.7	96.1 45.4	122.5 23.5	127.9 29.5	107.3 29.5	96.4 48.8	123.9 23.2	129.7 27.9	108.6 33.6	97.3 44.3	122.6 18.3	130.1 26.9	108.4 34.1	97.1 42.8
7.5	88.0	124.0 23.5	127.7 26.2	106.5 30.3	95.8 48.2	121.7 24.9	126.9 31.1	106.8 31.1	96.0 51.4	123.2 24.6	128.6 29.4	108.0 35.1	97.0 46.2	122.0 19.7	129.0 28.3	107.8 35.6	96.8 45.0
10.0	88.6	123.2 25.1	126.7 27.9	106.0 32.0	95.5 51.4	121.0 26.4	125.8 32.7	106.3 32.7	95.7 54.3	122.4 25.9	127.6 30.9	107.4 36.6	96.6 48.4	121.4 21.1	128.0 29.8	107.2 37.2	96.4 47.4
12.5	89.1	122.4 26.8	125.8 29.6	105.5 33.8	95.2 54.9	120.2 27.9	124.7 34.4	105.8 34.4	95.4 57.7	121.7 27.4	126.5 32.5	106.8 38.2	96.3 50.7	120.8 22.6	127.0 31.2	106.6 38.8	96.1 50.0
15.0	89.5	121.7 28.4	124.9 31.3	105.1 35.7	94.9 59.1	119.5 29.4	123.7 36.1	105.3 36.2	95.1 61.5	120.9 28.8	125.5 34.1	106.2 39.9	96.0 53.3	120.2 24.0	126.0 32.7	106.0 40.5	95.8 53.0
17.5	90.0	120.9 30.2	123.9 33.1	104.6 37.7	94.6 64.0	118.7 30.9	122.6 37.9	104.8 38.0	94.8 65.9	120.1 30.3	124.4 35.8	105.6 41.7	95.7 56.2	119.5 25.5	124.9 34.3	105.5 42.3	95.5 56.3
20.0	90.4	120.0 31.9	122.9 34.9	104.1 39.8	94.4 69.3	117.9 32.5	121.5 39.8	104.3 40.0	94.5 71.1	119.3 31.8	123.3 37.5	105.1 43.5	95.4 59.3	118.9 27.0	123.9 35.9	104.9 44.2	95.3 60.1
22.5	90.7	119.2 33.7	121.9 36.7	103.6 42.0	94.2 77.0	117.2 34.1	120.5 41.7	103.9 42.0	94.3 77.3	118.5 33.3	122.3 39.2	104.5 45.5	95.2 62.8	118.2 28.6	122.9 37.5	104.3 46.2	95.0 64.5
25.0	91.1	118.4 35.5	120.9 38.6	103.2 44.3	93.9 86.0	116.4 35.7	119.4 43.6	103.4 44.1	94.0 84.8	117.7 34.9	121.2 41.0	104.0 47.5	94.9 66.8	117.5 30.2	121.8 39.2	103.8 48.2	94.8 69.5
27.5	91.4	117.5 37.4	119.9 40.6	102.7 46.7	93.7 97.6	115.6 37.4	118.3 45.6	102.9 46.3	93.8 94.4	116.9 36.5	120.1 42.9	103.5 49.5	94.7 71.3	116.3 31.7	120.8 40.8	103.3 50.4	94.6 75.4
30.0	91.6	116.7 39.3	118.9 42.6	102.2 49.2	93.5 113.3	114.8 39.1	117.3 47.7	102.5 48.7	93.6 106.6	116.1 38.1	119.1 44.8	103.0 51.7	94.5 76.5	116.1 33.3	119.7 42.6	102.8 52.6	94.4 82.5

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THIS IS THE DATA FOR TEST RUN NUMBER 5.2

EAT TRANSFER V/A: .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 4.64 M/S $AZ = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	83.5	133.3 13.9	139.3 14.5	110.9 19.2	95.9 22.4	129.8 19.5	138.8 22.9	111.0 20.6	98.0 62.1	132.7 19.1	141.7 20.6	113.3 25.5	98.5 63.2	129.7 12.5	141.4 17.4	112.8 25.7	98.2 95.6
2.5	83.7	132.6 18.4	138.5 19.2	110.4 23.5	95.7 31.3	129.0 22.7	137.6 26.8	110.4 24.8	97.2 67.2	131.9 22.4	140.5 24.6	112.6 30.1	97.7 67.2	129.1 15.9	140.4 21.3	112.1 29.9	97.0 98.8
5.0	83.8	131.7 22.8	137.5 23.7	109.8 27.7	95.3 40.1	128.1 25.9	136.3 30.7	109.8 29.0	96.4 72.6	130.9 25.7	139.2 28.5	111.8 34.6	96.9 71.4	128.4 19.1	139.3 25.1	111.3 34.1	95.9 102.1
7.5	84.0	130.7 26.9	136.3 28.1	109.1 31.8	94.9 49.2	127.0 28.9	134.8 34.5	109.1 33.1	95.6 78.5	129.8 28.9	137.8 32.3	110.5 39.1	96.1 76.0	127.6 22.3	138.0 28.7	110.4 38.3	94.9 105.8
10.0	84.1	129.6 31.0	135.0 32.3	108.4 35.9	94.4 58.5	125.9 32.0	133.2 38.2	108.4 37.1	94.8 84.9	128.6 32.1	136.2 36.1	110.6 43.6	95.3 81.0	126.7 25.3	136.6 32.3	109.5 42.5	93.9 109.7
12.5	84.3	128.3 34.9	133.5 36.5	107.6 39.9	93.8 68.2	124.7 34.9	131.6 41.9	107.6 41.2	94.1 91.9	127.3 35.2	134.5 39.8	108.9 48.1	94.5 86.4	125.8 28.3	135.1 35.8	108.6 46.8	93.0 114.1
15.0	84.4	126.9 38.8	131.8 40.5	106.8 43.9	93.2 78.5	123.5 37.8	129.8 45.6	106.7 45.2	93.3 99.6	125.9 38.2	132.8 43.5	107.9 52.7	93.8 92.3	124.7 31.1	133.4 39.3	107.6 51.0	92.2 118.9
17.5	84.5	125.4 42.5	130.1 44.4	105.9 47.8	92.6 89.6	122.1 40.6	126.0 49.2	105.8 49.1	92.5 108.3	124.5 41.1	130.9 47.0	106.8 57.2	93.0 98.8	123.6 31.9	131.7 42.6	106.5 50.3	91.4 124.3
20.0	84.6	123.9 46.0	128.3 48.1	105.0 51.7	92.0 101.6	120.8 43.4	126.1 52.8	104.9 53.1	91.8 118.0	123.1 44.0	129.0 50.5	105.7 61.8	92.3 106.0	122.4 36.5	129.9 45.9	105.5 59.7	90.7 130.3
22.5	84.8	122.3 49.5	126.5 51.8	104.1 55.6	91.3 114.9	119.4 46.0	124.2 56.2	104.0 57.0	91.1 128.9	121.6 46.8	127.0 53.9	103.8 66.4	91.6 114.0	121.2 39.0	128.1 49.0	104.4 64.0	90.0 137.1
25.0	84.9	120.7 52.7	124.6 55.2	103.2 59.3	90.7 129.7	118.0 48.6	122.3 59.6	103.0 60.7	90.4 141.5	120.1 49.4	125.0 57.2	103.4 70.9	91.0 122.9	120.0 41.4	126.3 52.0	103.3 68.3	89.4 144.0
27.5	84.9	119.0 55.7	122.7 58.4	102.2 62.9	90.0 146.5	116.6 50.9	121.3 62.7	102.0 64.4	89.7 156.0	118.6 51.9	123.0 60.3	102.3 75.3	90.3 132.9	118.7 43.6	124.4 54.8	102.2 72.6	88.9 153.8
30.0	85.1	117.3 58.7	120.7 58.5	101.2 63.5	89.2 149.9	115.1 51.1	116.3 62.7	101.0 64.9	89.0 163.8	117.0 52.0	120.9 60.5	101.3 76.4	89.6 138.6	117.3 44.0	122.3 54.8	101.0 73.2	88.2 157.3

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THIS IS THE DATA FOR TEST RUN NUMBER 5.3

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 5.59 M/S $\Delta z = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	82.5	179.8 39.3	191.6 40.9	133.2 45.6	106.1 92.4	172.4 37.3	188.3 42.0	132.8 41.1	106.9 91.7	176.8 37.1	193.2 41.5	136.2 48.7	107.4 90.0	173.8 29.4	194.8 38.5	135.9 45.5	107.0 83.5
2.5	83.0	176.5 40.6	187.8 42.2	131.2 47.2	104.3 95.7	169.5 38.0	184.5 42.7	131.0 42.3	105.0 94.6	173.8 37.6	189.2 42.4	133.9 49.9	105.5 92.3	171.5 30.3	191.1 39.1	133.9 46.5	105.3 85.6
5.0	83.4	173.2 42.0	184.0 43.4	129.3 48.8	102.6 99.1	166.6 38.8	180.7 43.5	129.3 43.6	103.3 97.7	170.8 38.2	185.3 43.3	131.7 51.1	103.8 94.6	169.2 31.2	187.5 39.7	131.8 47.6	103.7 87.7
7.5	83.8	170.0 43.4	180.2 44.6	127.3 50.4	101.0 102.7	163.9 39.5	177.1 44.2	127.5 44.8	101.7 100.9	168.0 38.8	181.6 44.1	129.6 52.4	102.2 96.9	166.9 32.1	183.9 40.3	129.9 48.6	102.2 89.8
10.0	84.2	166.7 44.8	176.5 45.9	125.5 52.0	99.5 106.5	161.2 40.2	173.6 45.0	125.9 46.1	100.2 104.1	165.2 39.4	177.9 45.0	127.6 53.7	100.7 99.2	164.6 32.9	180.5 40.9	128.0 49.6	100.8 91.9
12.5	84.5	163.5 46.2	172.9 47.2	123.6 53.7	98.1 110.4	158.5 40.9	170.1 45.7	124.2 47.4	98.8 107.5	162.4 40.0	174.2 45.9	125.6 54.9	99.3 101.3	162.3 33.8	177.1 41.6	126.1 50.7	99.5 93.9
15.0	84.8	160.4 47.6	169.3 48.4	121.8 55.4	96.9 114.6	155.9 41.6	166.8 46.4	122.6 48.7	97.5 110.9	159.8 40.6	170.7 46.8	123.7 56.2	98.1 103.4	160.0 34.6	173.8 42.2	124.3 51.7	98.4 95.8
17.5	85.0	157.3 49.0	165.9 49.7	120.1 57.1	95.7 119.1	153.4 42.3	163.5 47.2	121.0 49.9	96.3 114.4	157.2 41.2	167.3 47.7	121.8 57.5	96.9 105.3	157.8 35.4	170.6 42.8	122.5 52.8	97.3 97.7
20.0	85.2	154.2 50.3	162.3 50.9	118.4 58.7	94.7 123.7	150.9 42.9	160.4 48.0	119.5 51.2	95.3 117.9	154.6 41.7	163.9 48.6	120.0 50.7	95.9 107.0	155.5 36.2	167.4 43.5	120.9 53.8	96.3 99.4
22.5	85.4	151.3 51.7	158.9 52.1	116.7 60.4	93.7 128.7	148.5 43.6	157.3 48.7	118.0 52.4	94.3 121.5	152.1 42.3	160.6 49.4	118.3 60.0	95.0 108.5	153.3 37.0	164.4 44.1	119.2 54.9	95.4 101.0
25.0	85.6	148.3 53.0	155.7 53.3	115.1 62.1	92.8 134.0	146.1 44.2	154.3 49.5	116.5 53.6	93.4 125.0	149.7 42.9	157.5 50.3	116.6 61.2	94.1 109.8	151.2 37.7	161.4 44.7	117.6 55.9	94.5 102.4
27.5	85.7	145.5 54.2	152.5 54.5	113.5 63.7	92.0 139.6	143.8 44.8	151.4 50.2	115.1 54.8	92.6 128.6	147.4 43.4	154.4 51.1	115.0 62.4	93.3 110.9	149.0 38.4	158.5 45.4	116.1 56.2	93.8 103.6
30.0	85.8	142.7 54.7	149.3 55.0	112.0 64.6	91.3 144.6	141.6 45.2	148.5 50.9	113.7 55.6	91.9 130.4	145.1 43.8	151.4 51.6	113.4 63.3	92.6 111.0	146.9 38.8	155.7 45.9	114.6 57.8	93.1 105.0

THIS IS THE DATA FOR TEST RUN NUMBER 6.4

HEAT TRANSFER V/A : .00079 B

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 8.78 M/S $At = 0^\circ$

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TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	79.4	135.8 53.5	142.9 55.4	111.5 69.8	96.4 136.8	131.8 56.0	142.0 61.6	111.2 60.3	96.3 134.2	135.2 55.4	145.6 59.9	114.0 73.1	97.4 130.9	131.8 41.3	145.4 54.5	113.5 72.4	94.6 213.5
2.5	79.9	133.2 57.6	139.8 59.6	109.6 74.0	94.5 145.8	129.2 58.9	133.7 65.2	109.5 64.3	94.4 141.7	132.5 58.2	142.2 63.5	111.9 77.2	95.4 137.3	129.9 44.2	142.2 57.3	111.4 75.5	92.0 231.1
5.0	80.3	130.5 61.9	136.7 64.0	107.7 78.4	92.7 156.3	126.7 61.9	135.4 68.8	107.9 68.6	92.7 150.1	129.9 61.0	138.7 67.3	109.7 81.5	93.7 144.4	126.0 47.3	139.2 60.2	109.4 78.8	89.7 252.4
7.5	80.6	127.8 66.4	133.6 68.6	105.8 83.1	91.1 168.6	124.3 65.0	132.1 72.7	106.3 73.0	91.2 159.6	127.3 64.0	135.3 71.2	107.7 86.1	92.1 152.1	126.0 50.4	136.1 63.2	107.4 82.3	87.8 278.8
10.0	80.9	125.1 71.2	130.4 73.5	104.0 88.1	89.7 183.3	121.8 68.2	128.9 76.8	104.6 77.8	89.8 170.6	124.7 67.1	132.0 75.4	105.7 91.0	90.6 160.6	124.0 53.5	133.1 66.3	105.5 85.9	86.2 313.0
12.5	81.2	122.3 76.1	127.3 78.6	102.3 93.6	88.3 201.5	119.4 71.5	125.7 81.1	103.1 82.7	88.6 183.5	122.2 70.3	128.7 79.7	103.8 96.3	89.4 170.1	122.0 56.8	130.1 69.5	103.7 89.8	84.9 360.0
15.0	81.5	119.6 81.3	124.1 83.9	100.6 99.4	87.2 224.8	117.1 75.0	122.6 85.6	101.5 88.0	87.5 199.0	119.7 73.6	125.4 84.3	101.9 101.9	88.2 180.8	120.0 60.0	127.2 72.8	102.0 93.9	83.9 430.8
17.5	81.7	117.0 86.7	121.1 89.5	99.0 105.7	86.1 255.8	114.8 78.4	119.6 90.3	100.0 93.6	86.5 218.1	117.3 77.0	122.2 89.1	100.2 108.0	87.2 192.9	118.0 63.4	124.3 76.2	100.4 98.2	83.1 556.8
20.0	81.9	114.3 92.3	118.0 95.4	97.5 112.5	85.2 299.8	112.6 82.0	116.7 95.2	98.5 99.4	85.6 242.7	115.0 80.5	119.1 94.1	98.5 114.5	86.4 206.9	116.0 66.8	121.6 79.7	98.8 102.7	82.5 877.6
22.5	82.1	111.8 98.2	115.1 101.5	96.0 119.8	84.4 368.8	110.4 85.6	113.9 100.3	97.1 105.5	84.9 276.1	112.7 84.0	116.2 99.3	96.9 121.5	85.6 223.3	114.0 70.2	118.8 83.3	97.4 107.4	82.1 45232.8
25.0	82.2	109.3 104.3	112.2 107.9	94.6 127.6	83.7 495.3	108.4 89.1	111.1 105.6	95.8 111.9	84.3 325.3	110.5 87.5	113.3 104.6	95.4 129.0	85.0 243.4	112.1 73.6	116.2 86.9	96.0 112.4	81.9 618.5
27.5	82.4	106.9 110.5	105.5 114.5	93.2 136.1	81.1 816.0	106.4 92.0	108.6 111.1	94.5 118.6	83.7 407.3	108.4 91.1	110.5 110.1	93.9 136.9	84.5 269.0	110.2 77.0	113.7 90.6	94.7 117.5	81.8 116.0
30.0	82.5	104.4 111.2	106.6 115.2	91.8 138.8	82.5 *****	104.3 92.0	105.9 111.6	93.2 119.3	83.2 682.5	106.2 91.7	107.7 111.4	92.5 139.4	83.9 323.5	108.2 77.9	111.2 91.9	93.4 120.7	82.0 218.8

THIS IS THE DATA FOR TEST RUN NUMBER 5.5

HEAT TRANSFER V/A : .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 9.57 M/S $Az = 0^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	90.4	177.2 69.3	187.3 69.4	137.0 79.5	110.0 137.0	170.0 60.7	183.1 63.0	136.5 64.3	110.6 131.0	174.6 55.7	188.1 64.0	139.2 76.4	111.1 123.1	173.5 46.6	190.0 56.6	139.7 69.1	112.0 113.5
2.5	90.7	172.1 70.7	181.6 70.8	133.9 81.4	107.7 143.7	165.9 61.6	178.2 64.1	134.0 65.9	108.4 136.5	170.6 56.6	182.8 65.2	136.0 78.1	109.0 127.4	170.1 47.6	185.2 57.7	136.8 70.6	110.0 117.3
5.0	91.0	167.2 72.1	176.1 72.3	130.9 83.4	105.7 151.0	162.0 62.5	173.4 65.3	131.6 67.5	106.4 142.5	168.7 57.5	177.7 66.5	133.0 79.9	107.0 131.8	166.9 48.7	180.5 58.7	134.0 72.2	108.1 121.3
7.5	91.3	162.5 73.5	170.9 73.7	128.1 85.4	103.9 159.0	158.2 63.4	168.8 66.4	129.2 69.2	104.6 148.9	163.0 56.4	172.8 67.7	130.2 81.6	105.3 136.4	163.7 49.7	176.1 59.8	131.4 73.7	106.4 125.3
10.0	91.5	158.1 74.9	165.9 75.2	125.5 87.4	102.2 168.0	154.6 64.2	164.5 67.6	127.0 70.9	103.0 155.8	159.4 59.3	168.1 69.0	127.5 83.4	103.7 141.0	160.6 50.8	171.7 60.8	128.9 75.3	104.8 129.5
12.5	91.7	153.9 76.4	161.2 76.7	123.0 89.5	100.7 178.2	151.2 65.1	160.3 68.8	124.9 72.6	101.5 163.3	156.0 60.2	163.6 70.3	125.0 85.2	102.3 145.7	157.6 51.9	167.6 61.9	126.5 76.9	103.4 133.8
15.0	91.8	149.9 77.8	156.7 78.2	120.6 91.6	99.4 189.7	147.9 66.0	156.3 69.9	122.8 74.3	100.2 171.5	152.7 61.1	159.3 71.6	122.6 87.0	101.0 150.4	154.7 53.0	163.6 63.0	124.2 78.4	102.1 138.0
17.5	92.0	146.0 79.3	152.4 79.7	118.4 93.7	98.2 203.0	144.8 66.8	152.4 71.1	120.9 76.0	99.0 180.5	149.6 62.0	155.2 72.9	120.3 88.8	99.9 155.1	151.6 54.0	159.7 64.1	122.1 80.0	100.5 142.3
20.0	92.1	142.4 80.9	148.3 81.3	116.3 95.9	97.2 218.7	141.8 67.6	148.8 72.4	119.0 77.7	98.0 190.5	146.5 62.9	151.3 74.2	118.2 90.6	98.9 159.5	149.1 55.1	156.0 65.2	120.1 81.7	99.6 146.5
22.5	92.2	139.0 82.4	144.4 82.8	114.3 98.1	96.3 237.7	138.9 68.3	145.3 73.6	117.2 79.4	97.0 201.7	143.6 63.7	147.6 75.5	116.2 92.4	98.0 163.8	146.4 56.1	152.5 66.3	118.1 83.3	98.5 150.6
25.0	92.2	135.7 83.9	140.8 84.4	112.5 100.3	95.5 261.6	136.2 69.0	142.0 74.8	115.6 81.1	96.2 214.7	140.8 64.6	144.0 76.8	114.3 94.1	97.2 167.7	143.8 57.1	149.1 67.5	116.3 84.9	98.7 154.0
27.5	92.3	132.6 85.5	137.3 85.9	110.8 102.5	94.8 293.1	133.7 69.7	138.9 76.0	114.0 82.7	95.5 230.3	136.2 65.5	140.7 78.1	112.6 95.0	96.5 171.3	141.3 58.1	145.8 68.6	114.6 86.5	97.4 158.0
30.0	92.4	129.7 86.5	134.1 86.9	109.2 104.0	94.3 332.4	131.2 70.2	135.9 77.5	112.4 84.5	94.9 259.1	135.6 66.1	137.5 79.6	111.0 98.0	96.0 178.6	138.8 58.5	142.7 69.8	112.9 88.6	96.7 157.5

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THIS IS THE DATA FOR TEST RUN NUMBER 5.6

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 5.39 M/S $\Delta z = 180^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	81.4	130.3 31.4	136.2 34.0	108.3 38.3	94.4 61.4	126.2 34.7	134.5 40.1	108.2 36.5	94.5 69.3	129.1 33.7	137.6 39.7	110.4 44.6	95.2 65.0	127.0 28.1	138.4 34.9	110.1 36.4	95.4 49.5
2.5	81.7	129.0 32.0	134.6 34.4	107.4 39.0	93.7 63.0	124.9 35.3	132.7 40.9	107.3 37.8	93.7 71.4	127.7 34.5	135.7 40.5	109.3 45.5	94.4 68.1	125.9 28.8	136.7 35.3	109.2 37.1	94.3 51.5
5.0	81.9	127.7 32.5	133.0 34.9	106.5 39.7	93.0 64.5	123.6 35.8	130.9 41.6	106.5 39.0	93.0 73.5	126.3 35.3	133.8 41.3	108.2 46.4	93.7 70.2	124.8 29.6	135.0 35.8	108.4 37.7	93.8 52.5
7.5	82.1	126.4 33.0	131.5 35.4	105.7 40.4	92.4 66.1	122.3 36.3	129.1 42.4	105.7 40.2	92.3 75.7	125.0 36.0	132.0 42.1	107.1 47.3	93.0 72.4	123.7 30.4	133.4 36.3	107.5 38.4	93.5 54.8
10.0	82.3	125.1 33.6	130.0 35.8	104.9 41.1	91.9 67.7	121.0 36.8	127.4 43.1	104.9 41.5	91.6 77.9	123.6 36.8	130.2 42.9	106.1 48.3	92.3 74.6	122.6 31.1	131.8 36.7	106.7 39.1	92.4 56.5
12.5	82.4	123.9 34.1	128.5 36.3	104.1 41.8	91.3 69.3	119.8 37.2	125.8 43.9	104.1 42.7	91.0 80.2	122.3 37.5	128.4 43.6	105.1 49.2	91.7 75.8	121.5 31.9	130.2 37.2	105.8 39.8	92.5 57.5
15.0	82.5	122.7 34.6	127.1 36.7	103.3 42.5	90.8 71.0	118.6 37.7	124.1 44.6	103.3 43.9	90.4 82.6	121.0 38.3	126.7 44.5	104.2 50.1	91.1 79.0	120.4 32.6	128.7 37.7	105.0 40.4	91.8 59.8
17.5	82.6	121.5 35.1	125.7 37.2	102.6 43.3	90.3 72.7	117.4 38.2	122.5 45.3	102.5 45.1	89.9 85.0	119.8 39.0	125.0 45.2	103.1 51.0	90.5 81.2	119.3 33.3	127.2 38.2	104.3 41.2	91.5 61.8
20.0	82.7	120.3 35.6	124.3 37.7	101.8 44.0	89.8 74.4	116.3 38.6	121.0 46.0	101.7 46.3	89.4 87.5	118.5 39.7	123.4 46.0	102.4 52.0	90.0 83.3	118.3 34.0	125.7 38.7	103.5 41.9	90.8 63.5
22.5	82.7	119.2 36.1	123.0 38.1	101.1 44.8	89.4 76.1	115.2 39.0	119.5 46.7	100.9 47.5	88.9 90.1	117.3 40.3	121.8 46.7	101.5 52.0	89.5 85.5	117.2 34.7	124.3 39.2	102.7 42.6	90.4 66.5
25.0	82.7	118.0 36.5	121.6 38.6	100.4 45.5	88.9 78.0	114.1 39.4	118.0 47.4	100.2 48.7	88.4 92.9	116.1 40.9	120.4 47.5	100.6 53.3	89.0 87.6	116.2 35.4	122.9 39.7	102.0 43.4	90.6 68.8
27.5	82.8	116.9 37.0	120.4 39.0	99.7 46.5	88.5 80.0	113.0 39.8	116.6 48.0	99.5 49.8	87.9 95.6	114.9 41.5	118.7 48.2	99.3 54.3	88.5 89.0	115.2 36.0	121.5 40.3	101.3 44.2	89.7 70.5
30.0	82.8	115.8 37.4	119.1 39.4	99.0 47.6	88.1 82.4	112.0 40.3	115.2 48.6	98.8 51.0	87.5 99.6	113.8 42.0	117.2 49.0	99.0 55.3	88.1 92.3	114.2 36.5	120.2 40.8	100.6 45.0	89.5 76.5

THIS IS THE DATA FOR TEST RUN NUMBER 5.7

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S).

THE AVERAGE VELOCITY : 9.29 M/S $AZ = 180^\circ$

B-83

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	82.8	134.6	141.0	108.1	93.1	129.7	137.5	107.7	92.4	131.9	140.7	109.2	93.0	131.5	142.7	109.9	93.6
		49.5	51.1	58.9	98.5	54.7	64.3	68.0	118.6	52.6	62.8	75.8	110.8	43.9	52.0	59.4	74.8
2.5	83.0	132.4	138.4	106.9	92.2	127.5	134.5	106.3	91.5	129.7	137.6	107.5	92.1	129.7	140.0	108.5	92.9
		50.2	51.9	59.7	100.7	55.4	65.3	69.1	123.4	53.6	63.9	77.1	114.3	44.9	52.9	60.1	77.2
5.0	83.3	130.3	136.0	105.6	91.5	125.4	131.6	105.0	90.6	127.6	134.6	106.0	91.2	127.9	137.5	107.2	92.2
		51.0	52.4	60.4	103.1	56.0	66.5	70.3	128.9	54.6	65.1	78.4	118.1	45.9	53.8	60.9	79.7
7.5	83.5	128.3	133.6	104.5	90.8	123.4	128.9	103.7	89.8	125.5	131.8	104.5	90.4	126.2	135.0	106.0	91.6
		51.7	53.1	61.3	105.8	56.7	67.6	71.5	135.4	55.6	66.4	79.8	122.4	47.0	54.7	61.7	82.6
10.0	83.7	126.3	131.4	103.4	90.1	121.5	126.3	102.4	89.1	123.5	129.0	103.1	89.7	124.4	132.6	104.8	91.1
		52.5	53.8	62.1	108.7	57.4	68.8	72.8	143.1	56.7	67.6	81.2	127.2	48.1	55.6	62.5	85.6
12.5	83.9	124.4	129.2	102.4	89.6	119.7	123.8	101.3	88.5	121.6	126.4	101.7	89.1	122.8	130.3	103.7	90.5
		53.2	54.5	63.0	112.1	58.1	70.0	74.1	152.5	57.8	68.9	82.6	132.7	49.2	56.5	63.4	89.5
15.0	84.2	122.5	127.1	101.4	89.0	117.9	121.5	100.2	87.9	119.8	124.0	100.5	88.5	121.1	128.0	102.6	90.0
		54.0	55.2	63.9	115.9	58.8	71.2	75.5	164.3	58.9	70.2	84.0	139.2	50.3	57.5	64.3	93.0
17.5	84.4	120.8	125.1	100.5	88.6	116.2	119.2	99.2	87.4	118.0	121.6	99.4	88.0	119.5	125.9	101.6	89.6
		54.8	56.0	64.9	120.5	59.6	72.5	76.9	179.8	60.0	71.5	85.5	146.9	51.4	58.5	65.3	98.3
20.0	84.6	119.1	123.1	99.6	88.2	114.6	117.1	98.2	87.0	116.2	119.3	98.3	87.6	118.0	123.8	100.7	89.2
		55.6	56.8	66.0	125.9	60.3	73.7	78.3	201.2	61.2	72.9	86.9	156.4	52.5	59.5	66.3	103.8
22.5	84.8	117.4	121.3	98.7	87.8	113.1	115.5	97.3	86.6	114.6	117.2	97.3	87.2	116.5	121.8	99.8	88.8
		56.4	57.6	67.1	132.8	61.0	75.0	79.8	232.9	62.3	74.2	88.4	168.6	53.6	60.6	67.4	110.3
25.0	84.9	115.9	119.5	98.0	87.5	111.6	113.1	96.5	86.2	113.0	115.2	96.4	86.9	115.0	119.9	98.9	88.4
		57.2	58.4	68.3	141.6	61.8	76.3	81.3	284.9	63.5	75.5	89.8	184.9	54.7	61.6	68.6	116.0
27.5	85.1	114.4	117.8	97.2	87.2	110.2	111.3	95.7	85.9	111.5	113.2	95.5	86.6	113.6	119.1	98.1	88.0
		58.1	59.2	69.6	153.5	62.5	77.6	82.9	306.9	64.6	76.8	91.1	200.2	55.8	62.6	69.7	117.0
30.0	85.2	112.9	116.1	96.5	86.9	108.5	109.6	95.0	85.6	110.0	111.4	94.7	86.3	112.3	116.3	97.3	87.4
		58.5	60.0	71.6	172.9	63.5	77.8	83.0	697.8	66.0	77.4	90.4	253.9	56.7	63.6	71.7	139.4

THIS IS THE DATA FOR TEST RUN NUMBER 5.8

HEAT TRANSFER V/A : .00079 M

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 9.36 M/S

$A_2 = 180^\circ$

B-84

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	80.3	137.2	143.6	111.7	96.5	132.9	141.5	111.7	95.4	135.3	144.4	112.9	95.9	134.3	145.2	113.1	97.1
		45.3	46.5	52.2	83.3	51.5	60.5	60.8	101.9	52.0	59.9	68.9	102.4	42.5	49.1	52.4	75.4
2.5	80.4	135.0	141.0	110.3	95.3	130.6	138.3	110.1	94.1	132.9	141.1	111.0	94.5	132.3	142.5	111.7	96.0
		46.5	47.6	53.5	85.8	52.4	51.7	62.1	105.4	52.9	61.1	70.2	105.3	43.5	50.2	53.6	77.0
5.0	80.4	132.8	138.6	108.9	94.2	128.3	135.3	108.5	92.9	130.5	137.9	109.2	93.3	130.3	139.8	110.2	95.0
		47.7	48.7	55.0	88.4	53.3	62.8	63.4	109.3	53.9	62.2	71.6	108.3	44.6	51.2	54.9	78.0
7.5	80.4	130.6	136.1	107.6	93.2	126.1	132.3	107.0	91.7	128.2	134.9	107.4	92.1	128.4	137.2	108.8	94.0
		48.8	49.8	56.4	91.1	54.2	64.1	64.8	113.4	54.9	63.5	73.0	111.4	45.7	52.3	56.2	80.2
10.0	80.4	128.5	133.7	106.2	92.2	124.0	129.5	105.5	90.7	125.9	131.9	105.8	91.0	126.5	134.7	107.5	93.1
		50.0	50.9	57.9	94.0	55.1	55.3	66.2	117.6	55.9	64.7	74.4	114.7	46.8	53.4	57.6	81.9
12.5	80.4	126.5	131.4	105.0	91.3	122.0	126.8	104.1	89.6	123.8	129.1	104.2	90.0	124.7	132.2	106.1	92.2
		51.2	52.1	59.4	97.0	56.0	66.6	67.6	122.5	57.0	66.0	75.9	118.0	47.9	54.6	59.0	83.0
15.0	80.4	124.4	129.1	103.7	90.4	120.0	124.1	102.8	88.7	121.7	126.3	102.6	89.1	122.9	129.8	104.8	91.4
		52.4	53.3	60.9	100.2	56.9	57.9	69.1	127.7	58.0	67.3	77.4	121.5	49.0	55.7	60.4	85.4
17.5	80.4	122.5	126.9	102.5	89.5	118.1	121.6	101.4	87.8	119.6	123.7	101.2	88.2	121.1	127.4	103.6	90.6
		53.7	54.8	62.4	103.5	57.3	59.3	70.5	133.3	59.1	68.2	78.3	125.0	50.1	56.9	61.9	87.2
20.0	80.4	120.5	124.7	101.3	88.8	116.2	119.2	100.2	87.0	117.6	121.2	99.8	87.4	119.3	125.1	102.3	89.6
		54.9	55.6	64.0	107.0	58.8	70.6	72.0	139.4	60.2	70.4	80.5	128.7	51.2	58.1	63.4	89.0
22.5	80.4	118.6	122.6	100.2	88.0	114.4	116.9	99.0	86.2	115.7	118.8	98.5	86.6	117.6	122.9	101.2	89.2
		56.1	56.8	65.5	110.7	59.7	72.0	73.5	146.1	61.3	71.5	82.0	132.4	52.2	59.3	64.9	90.8
25.0	80.4	116.8	120.6	99.1	87.3	112.7	114.6	97.8	85.5	113.9	116.4	97.2	85.9	116.0	120.8	100.0	88.5
		57.3	57.9	67.0	114.5	60.6	73.5	75.0	153.5	62.4	73.0	83.7	136.2	53.3	60.6	66.5	92.7
27.5	80.4	115.0	118.6	98.0	86.6	111.0	112.5	96.7	84.9	112.1	114.2	96.0	85.3	114.3	118.7	98.9	87.1
		58.9	59.0	68.5	116.5	61.5	74.9	76.5	161.8	63.5	74.4	85.3	140.0	54.3	61.8	69.0	94.6
30.0	80.4	113.3	116.7	97.0	86.0	109.4	110.4	95.6	84.3	110.4	112.0	94.9	84.7	112.8	116.6	97.8	87.3
		58.9	59.8	69.0	119.4	62.3	76.4	77.6	171.7	64.5	76.1	87.0	141.6	54.8	63.1	69.7	95.2

THIS IS THE DATA FOR TEST RUN NUMBER 5.9

HEAT TRANSFER V/A : .00079 H

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 9.73 M/S $Az = 180^\circ$

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	82.1	163.1 38.3	172.8 40.0	126.0 41.4	103.3 64.6	155.6 44.6	167.1 50.8	124.4 45.5	103.2 75.3	160.0 43.4	172.8 49.8	126.4 64.7	103.5 78.7	158.3 35.2	175.5 41.9	128.4 40.5	104.0 52.7
2.5	82.8	160.5 39.1	169.7 40.7	124.4 42.4	102.2 66.2	152.8 45.2	163.4 51.8	122.8 47.2	101.9 78.0	157.1 44.2	169.0 50.7	124.0 65.9	102.1 80.5	156.0 36.0	172.1 42.5	126.8 41.6	103.0 54.3
5.0	83.4	157.8 39.9	166.7 41.4	124.9 43.5	101.1 67.9	150.1 45.8	159.8 52.9	121.2 49.0	100.6 81.1	154.1 45.0	165.2 51.7	121.7 67.1	100.8 82.4	153.8 36.8	168.9 43.1	125.2 42.7	102.1 55.5
7.5	83.9	155.3 40.7	163.7 42.2	121.4 44.7	100.1 69.7	147.5 46.3	156.4 53.9	119.6 50.8	99.5 84.3	151.6 45.7	161.6 52.7	119.6 68.4	99.6 84.3	151.5 37.5	165.7 43.7	123.7 43.8	101.2 57.7
10.0	84.4	152.8 41.5	160.9 43.0	120.0 45.8	99.1 71.5	145.0 46.9	153.0 55.1	118.0 52.7	98.4 87.9	148.9 46.5	158.1 53.7	117.5 69.6	98.5 86.4	149.4 38.3	162.7 44.3	122.2 45.0	100.4 59.6
12.5	84.9	150.4 42.3	158.0 43.7	118.6 47.0	98.3 73.5	142.6 47.5	149.8 56.2	116.5 54.6	97.3 91.9	146.4 47.3	154.8 54.8	115.6 70.9	97.5 88.5	147.2 39.1	159.7 45.0	120.7 46.1	99.6 61.6
15.0	85.4	148.0 43.1	155.3 44.5	117.2 48.3	97.4 75.6	140.2 48.0	146.7 57.3	115.0 56.8	96.4 96.2	143.9 48.1	151.5 55.8	113.7 72.2	96.5 90.7	145.1 39.9	156.8 45.6	119.3 47.4	98.8 63.7
17.5	85.8	145.7 43.9	152.6 45.3	115.9 49.5	96.7 77.9	138.0 48.6	143.7 58.5	113.6 58.6	95.5 101.0	141.5 48.9	148.3 56.8	112.0 73.5	95.7 93.0	143.1 40.6	154.0 46.3	117.9 48.6	98.1 66.0
20.0	86.2	143.4 44.7	150.0 46.1	114.7 50.8	96.0 80.2	135.8 49.2	140.8 59.7	112.2 60.7	94.7 106.4	139.2 49.7	145.3 57.9	110.4 74.7	94.9 95.5	141.1 41.4	151.3 46.9	116.6 49.8	97.4 68.4
22.5	86.5	141.2 45.5	147.5 46.8	113.4 52.1	95.3 82.8	133.8 49.8	138.0 60.8	110.8 62.8	93.9 112.4	136.9 50.5	142.4 58.9	108.9 76.0	94.2 98.0	139.1 42.2	148.7 47.6	115.3 51.1	96.7 71.0
25.0	86.8	139.1 46.3	145.1 47.6	112.2 53.3	94.7 85.5	131.8 50.3	135.4 62.0	109.5 64.8	93.2 119.2	134.7 51.3	139.6 60.0	107.4 77.2	93.6 100.8	137.2 42.9	146.2 48.3	114.0 52.4	96.1 73.7
27.5	87.1	137.0 47.0	142.7 48.3	111.1 54.6	94.1 88.4	129.8 50.9	132.8 64.1	108.3 66.9	92.6 126.9	132.6 52.1	136.9 61.0	106.1 78.3	93.1 103.7	135.4 43.6	143.8 49.0	112.8 53.6	95.6 76.5
30.0	87.4	135.0 47.7	140.4 48.9	109.9 56.1	93.6 91.9	128.0 51.6	130.4 64.1	107.1 68.0	92.0 130.8	130.6 52.6	134.3 61.6	104.8 78.5	92.5 106.3	133.6 44.1	141.4 49.4	111.6 54.4	95.0 80.2

THIS IS THE DATA FOR TEST RUN NUMBER 5.10

HEAT TRANSFER V/A : .00079 W

THE STARTING TIME FOR EACH CHANNEL WILL BE 0.0
SECOND(S) AND THE ENDING TIME FOR EACH CHANNEL WILL
BE 30.0 SECOND(S) WITH A TIME INCREMENT OF 2.5 SECOND(S)

THE AVERAGE VELOCITY : 9.25 M/S $\Delta T = 45^\circ$

B-86

TIME	TA	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
0.0	80.4	170.0 86.0	131.5 78.4	128.0 104.0	102.9 122.4	162.6 94.5	177.1 93.4	128.6 91.2	102.1 154.5	165.6 100.0	181.4 95.0	130.3 99.2	101.7 144.9	166.2 85.8	182.0 103.1	128.7 118.5	101.3 178.8
2.5	80.5	163.5 88.1	174.8 80.4	123.9 106.7	100.6 126.9	156.0 96.6	169.5 95.8	124.9 92.9	99.3 162.5	158.5 102.3	173.3 97.6	126.1 102.1	99.1 154.0	160.0 87.9	173.2 105.8	123.9 121.9	98.3 188.0
5.0	80.5	157.3 90.3	168.4 82.4	120.0 109.6	98.5 131.8	149.9 98.8	162.3 98.4	121.4 94.7	96.8 171.5	151.8 104.7	165.7 100.3	122.2 105.3	96.7 164.5	154.1 90.0	164.9 108.6	119.5 125.4	95.5 196.2
7.5	80.6	151.4 92.6	162.2 84.5	116.4 112.6	96.5 137.1	144.2 100.9	155.5 101.0	118.2 96.7	94.5 181.9	145.5 107.1	158.5 103.1	118.5 108.6	94.5 176.8	148.5 92.2	157.3 111.5	115.4 129.0	93.1 209.4
10.0	80.6	145.9 95.0	156.4 86.7	113.0 115.7	94.7 142.8	138.8 103.2	149.2 103.8	115.1 98.7	92.4 194.0	139.7 109.6	151.7 106.0	115.0 112.2	94.4 191.6	143.2 94.5	150.1 114.5	111.7 132.6	91.0 221.7
12.5	80.6	140.7 97.6	150.8 89.0	109.9 119.0	93.0 149.2	133.7 105.4	143.2 106.7	112.3 101.0	90.5 208.3	134.3 112.2	145.4 109.1	111.6 115.1	90.6 209.6	138.2 96.8	143.4 117.5	108.3 136.4	89.1 235.4
15.0	80.6	135.7 100.2	145.5 91.3	106.9 122.3	91.5 156.1	129.0 107.7	137.6 109.7	109.6 103.4	88.8 225.7	129.2 114.7	139.5 112.3	108.8 120.2	88.8 232.1	133.5 99.2	137.3 120.7	105.1 140.2	87.5 250.5
17.5	80.6	131.1 102.9	140.5 93.8	104.3 125.9	90.1 164.0	124.7 109.9	132.3 112.8	107.1 106.2	87.3 247.4	124.6 117.2	133.5 115.6	105.9 124.6	87.2 261.5	129.1 101.6	131.6 123.9	102.3 144.0	86.1 267.3
20.0	80.6	126.7 105.7	135.6 95.4	101.8 129.5	88.8 172.8	120.6 112.1	127.5 116.1	104.7 109.4	86.0 275.7	120.3 119.8	128.8 119.0	103.3 129.4	85.8 301.4	125.0 104.0	126.3 127.2	99.7 147.8	84.9 286.0
22.5	80.6	122.6 108.6	131.3 99.0	99.5 133.3	87.6 182.9	116.3 114.2	122.9 119.5	102.5 113.0	84.8 314.5	116.3 122.3	124.0 122.6	100.8 134.6	84.5 359.4	121.1 106.5	121.5 130.5	97.4 151.6	84.0 306.7
25.0	80.6	118.8 111.6	127.1 101.7	97.4 137.3	86.6 194.7	113.4 116.3	118.7 123.0	100.4 117.4	83.7 371.4	112.7 124.7	119.5 126.2	98.6 140.3	83.4 452.5	117.5 109.0	117.1 133.9	95.3 155.2	83.2 329.9
27.5	80.6	115.2 114.6	123.1 104.5	95.5 141.5	85.6 208.7	110.3 118.2	114.8 126.6	98.5 122.6	82.8 404.4	109.4 127.1	115.4 129.5	96.5 146.5	82.4 623.8	114.1 111.4	113.0 137.0	93.4 158.6	82.5 355.3
30.0	80.6	111.9 117.3	119.4 106.0	93.8 143.9	84.8 222.6	107.3 119.7	111.2 130.4	96.5 131.7	81.9 671.2	106.4 129.4	111.7 133.6	94.5 154.3	81.5 3109.9	111.0 112.9	109.3 140.1	91.7 162.5	82.0 402.6

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Optimum Sensor Location in Transient Calorimeters

A. M. CLAUSING

Department of Mechanical Engineering,
University of Illinois at Urbana-Champaign,
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Transient calorimeters are simple and popular for heat flux measurement. However, erroneous readings are obtained whenever the temperature gradients within the calorimeter are such that the rate of change of the measured temperature is not proportional to the rate of change of the internal energy of the calorimeter. Optimum locations for the temperature sensor are determined for commonly used geometries. It is shown that the range of applicability of the calorimeter is vastly increased by placing the temperature sensor at the optimum location. The performance of the optimized calorimeter is established and compared with conventional calorimeters. A specialized analog circuit is also provided which makes a direct heat flux indicator out of a transient calorimeter.

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Optimum Sensor Location in Transient Calorimeters

A. M. CLAUSING

ABSTRACT

Transient calorimeters are simple and popular for heat flux measurement. However, erroneous readings are obtained whenever the temperature gradients within the calorimeter are such that the rate of change of the measured temperature is not proportional to the rate of change of the internal energy of the calorimeter. Optimum locations for the temperature sensor are determined for commonly used geometries. It is shown that the range of applicability of the calorimeter is vastly increased by placing the temperature sensor at the optimum location. The performance of the optimized calorimeter is established and compared with conventional calorimeters. A specialized analog circuit is also provided which makes a direct heat flux indicator out of a transient calorimeter.

NOMENCLATURE

A	Heat transfer area
Bi	Biot number, hL/k
C	A capacitor
c_p	Specific heat
E	Percent error in heat flux
e	Voltage
f	Frequency
h	Convective heat transfer coefficient
k	Thermal conductivity
L	Thickness of slab or outside radius of cylinder or sphere
m	Parameter which specifies geometry m = 0, plane slab; m = 1, cylinder; m = 2, sphere
q	Heat flux
Q	Heat flow rate
r	Independent spatial variable
R	Resistor
T	Temperature
t	Time
V	Volume
α	Thermal diffusivity
ρ	Density
τ	Time constant

Superscripts

* Dimensionless quantity

Subscripts

c	Calorimeter
i	Initial condition
L	Value at $r = L$
O	Value at $r = 0$
ref	Reference junction
s	Sensor location, $r = s$

= Ambient or fluid temperature
1 Some specified constant

INTRODUCTION

A transient technique is often used to determine the rate of heat transfer between a solid body and its surroundings. The variation of the temperature with respect to time is recorded at some point in the body. Then if the spatial gradients within the body are negligible, the net rate of heat transferred, Q , is given by

$$Q = \rho c_p V \frac{dT_s}{dt} \quad (1)$$

where ρ is the density of the object, c_p is its specific heat, V is its volume, and dT_s/dt is the time rate of change of the measured temperature. The average flux is

$$q = \rho c_p \frac{V}{A} \frac{dT_s}{dt} \quad (2)$$

where A is the surface area through which the heat is flowing. In boiling heat transfer studies and in single-phase convection studies, the complete model often is utilized as the calorimeter. Average heat fluxes and heat transfer coefficients are then obtained. Alternatively, local heat flux determinations are possible by using a thin-walled model or by inserting small calorimeters in large models. This general method of measuring heat flux is usually called a transient calorimeter. The popularity of the technique stems from its simplicity and its lack of interference with the velocity and temperature fields. In contrast, most other heat flux sensors are devices which must be installed in the model or on its surface. Hence, they introduce disturbances due to surface irregularities and changes in the conductive paths resulting from the insertion of the transducer.

Transient calorimeters, however, also have their limitations and disadvantages, mainly: (i) The use of Eq. (2) requires that the internal gradients be negligible or that

$$\left. \frac{dT_s}{dt} \right|_{\text{sensor}} = \frac{1}{V} \frac{d}{dt} \int_V T dV$$

The criteria which must be satisfied to enable this assumption have not been clearly established. (ii) The derivative dT_s/dt , i.e., the slope of the

temperature-time curve, must be determined accurately. Both graphical determinations of this slope and numerical differentiation leave much to be desired. (iii) Transient calorimeters do not give a direct indication of the variable of interest, the heat flux. The objective of this paper is to show how these limitations may be partially removed.

ERROR WITH CONVENTIONAL CALORIMETER

If a transient technique is employed to study a steady state process, an error may result due to the fact that the external conditions are not quasi-steady. This error can usually be eliminated by choosing a model with sufficient thermal capacitance and will not be considered here. An error can also be introduced if transient calorimeters are used to measure local heat fluxes in a thin-walled model due to conduction along the wall or heat transferred between the inside surfaces of the model. These influences are also usually secondary and will not be considered. The most common and important source of error associated with the use of transient calorimeters is caused by temperature gradients within the calorimeter. This is the error of interest.

In order to determine the parameters of importance and the magnitude of the errors resulting from internal gradients, consider the following problem: one-dimensional transient heat conduction in an initially isothermal region which is isotropic and homogeneous with constant thermal properties. A slab of thickness L ($m = 0$), a cylinder or radius L ($m = 1$), or a sphere of radius L ($m = 2$) will be considered. The temperature distribution in a calorimeter consisting of one of these three geometries is governed by

$$\frac{1}{\alpha} \frac{\partial T}{\partial t} = \frac{1}{r^m} \frac{\partial}{\partial r} \left(r^m \frac{\partial T}{\partial r} \right) \quad (3)$$

$$T = T_i \quad \text{at } t = 0 \quad (4)$$

$$\left. \frac{\partial T}{\partial r} \right|_{r=0} = 0, \quad t > 0 \quad (5)$$

where α is the thermal diffusivity and r is the independent spatial variable. Three different boundary conditions at the boundary through which the heat is flowing are considered:

$$-k \left. \frac{\partial T}{\partial r} \right|_{r=L} = h(T|_{r=L} - T_\infty), \quad (6a)$$

$$-k \left. \frac{\partial T}{\partial r} \right|_{r=L} = q_1, \quad (6b)$$

$$-k \left. \frac{\partial T}{\partial r} \right|_{r=L} = q_1 \sin(2\pi ft), \quad (6c)$$

a convective boundary condition with a constant heat transfer coefficient, h , and ambient temperature, T_∞ ; a constant flux; and a sinusoidal variation in the flux, respectively.

The temperature sensor is assumed to be free from error and located at $r = s$; hence, the heat flux determined from the transient calorimeter, q_c , is

$$q_c = -\rho c_p \frac{V}{A} \frac{dT_s}{dt} = -\rho c_p \frac{L}{(m+1)} \frac{dT_s}{dt} \quad (7)$$

The actual heat flux is

$$q_L = -k \left. \frac{\partial T}{\partial r} \right|_{r=L}$$

The percent error for the nonperiodic cases is defined as

$$E_p = 100 \frac{q_c - q_L}{q_L} \quad (8)$$

In order to avoid division by zero, the percent error is based on the peak-to-peak value for the periodic case, i.e.,

$$E_p = 100 \frac{q_c - q_L}{2q_1} \quad (9)$$

A dimensional analysis of Eqs. (3) through (9) (or an examination of exact solutions) gives the following results.

Convective boundary condition (6a):

$$E = f_a(t^*, Bi, m, \frac{s}{L}) \quad (10)$$

Constant flux boundary condition (6b):

$$E = f_b(t^*, m, \frac{s}{L}) \quad (11)$$

Sinusoidal variation in flux (6c):

$$E = f_c(t^*, \frac{fL^2}{\alpha}, m, \frac{s}{L})$$

where t^* , the dimensionless time, is defined as $\alpha t/L^2$, and the Biot number, Bi , is hL/k .

Graphical presentations of Eqs. (10) and (11) for the case of $s = 0$ are given in Fig. 1 for a plane slab, a cylinder, and a sphere. The variation of the percent

Fig. 1 Influences of $\alpha t/L^2$ and hL/k on Error in Heat Flux

error in the heat flux as a function of t^* is given with the Biot number as a parameter. The percent error in the heat flux for the sinusoidal boundary condition is not shown graphically. The error for the slab geometry with frequencies of (α/L^2) , $(1/4) \cdot (\alpha/L^2)$,

and $(1/16) \cdot (\alpha/L^2)$ varied between ± 44 percent, ± 12 percent, and ± 3 percent, respectively. The results for the other two geometries showed similar trends. Although the error with the sinusoidal boundary condition can be greatly decreased by effecting an appropriate phase shift, such corrections generally would not be possible in practice because the fluctuations would usually not be periodic or of known frequency.

If the sensor is located on the surface through which the heat is flowing, at $s = L$, the resulting error curves appear essentially as mirror images of those given in Fig. 1. At small times, the errors are positive, whereas the asymptotic errors as t^* approaches infinity are negative. Some numerical values for this sensor location are given later in Table 1 along with additional results.

It is concluded that if the temperature sensor is placed at the center or back face of objects similar to plane slabs, cylinders or spheres serving as transient calorimeters, the error in the heat flux will be less than approximately 3 percent if the following criteria are satisfied. The time period after any abrupt change in surface conditions must be greater than

$$t^* > 0.4, \quad (13a)$$

the Biot number must be less than

$$Bi < 0.1, \quad (13b)$$

and the frequencies of all significant harmonics must be less than

$$f < \frac{\alpha}{16L^2} \quad (13c)$$

The limitations imposed by criteria (13a) through (13c) severely reduce the usefulness of transient calorimeters; hence, a means of alleviating these restrictions is desirable. If a surface mounting of the temperature sensor is permissible, Eq. 2 could be ignored, and the resulting direct heat conduction problem could be solved. Although this procedure is used, it suffers from several disadvantages. Specifically: (1) The sensor must be mounted on the heat transfer surface; (2) The thermal properties of the substrate must be accurately known; and (3) The heat flux is derived only after considerable analysis; often a numerical solution is necessary.

An alternative procedure is to leave the sensor mounted inside the calorimeter and solve the resulting inverse heat conduction problem. A technique proposed by Burggraf (1) was considered. The first term of Burggraf's solution is Eq. (2). Unfortunately, the higher order terms contain the higher order derivatives d^2T_s/dt^2 , d^3T_s/dt^3 , etc. It was concluded from a series of numerical experiments that the higher order terms could not be determined with sufficient accuracy, due to inherent experimental error, to provide a significant change in criteria (13a) through (13c). In general, any inverse solution suffers from unknown errors which can be large if rapid fluctuations are occurring at the surface, as is clear if one considers the physics of heat conduction. In addition, inverse solutions require that the location of the temperature sensor be known with considerable accuracy.

OPTIMUM SENSOR LOCATION

The temperature sensor in calorimeters has traditionally been placed either at the surface,

$r^* = 1$, or at the center, $r^* = 0$. Unfortunately, these choices, which are probably based mostly on convenience, are the worst possible locations. They accentuate the error due to the temperature gradients within the calorimeter.

The determination of the best location for the temperature sensor in a specified calorimeter design may seem like a hopeless exercise since the location would appear to be dependent on Bi and t^* . In addition, the Biot modulus in practical problems is unknown, and it is typically not a constant. However, the results of this section will prove otherwise.

A brute force method was used to search for the optimum sensor location. Specifically, a set of error curves, analogous to those given in Fig. 1, were generated for different sensor locations ranging from the center of the calorimeter to its surface. The error curves were plotted with a Calcomp plotter and were calculated from numerical solutions. Several checks against available analytical results showed a negligible amount of numerical error. A family of seven error curves were typically generated for Biot moduli of 0.0 (the constant heat flux case), 0.2, 0.4, 0.8, 1.6, 3.2 and 6.4. Hence, the maximum modulus considered was essentially two orders of magnitude larger than allowed by criterion (13b). The scale used for the ordinate, the percent error, was ± 90 percent in all plots.

When the sensor location was near the optimum, a spectacular collapse of the family of curves occurred. The family of curves in this case was distinguishable from the zero percent error line on these machine plots only for $t^* < 0.2$. The numerous results which were obtained using boundary conditions (6a) and (6b) are summarized in Table 1. Almost all of the errors contained in this tabulation, i.e., $|E|_{\max}$ occurred with the Biot modulus of 6.4; however, the results include the complete range of this parameter, $0 < Bi < 6.4$. The results obtained with the sinusoidal boundary condition, Eq. (6c), are summarized in Table 2.

Table 1a Maximum Percent Errors, $|E|_{\max}$, for $0 \leq Bi \leq 6.4$ -- Slab Geometry ($m = 0$)

$r^* \backslash t^*$	0.0	.50	.54	.56	.6	1.0
-	40	9	4	1	-5	-70
0.2	-30	13	7	1	2	-60
0.1	-70	22	14	30	35	80

Table 1b Maximum Percent Errors, $|E|_{\max}$, for $0 \leq Bi \leq 6.4$ -- Cylinder ($m = 1$)

$r^* \backslash t^*$	0.0	.67	.69	.70	.72	1.0
-	80	5	1	-2	-5	-70
0.2	40	7	3	3	2	-60
0.1	-55	19	20	16	15	-50

Table 1c Maximum Percent Errors, $|E|_{\max}$, for
 $0 \leq Bi \leq 6.4$ -- Sphere ($m = 2$)

r^* t^*	0.0	.73	.75	.78	.80	1.0
∞	130	.8	3	-5	-9	-60
0.2	120	9	3	-4	-8	-60
0.1	-50	-15	10	7	7	-55

Table 2a Maximum Errors with Sinusoidal Boundary
Condition -- Slab ($m = 0$)

r^* f	0.0	.54	.55	.60	.62	1.0
a/L^2	± 44	± 9	± 8	± 7	± 8	± 90
$a/2L^2$	± 25	± 4	± 3	± 2	± 4	± 50

Table 2b Maximum Errors with Sinusoidal Boundary
Condition -- Cylinder ($m = 1$)

r^* f	.69	.70	.72	.73
$2a/L^2$	± 9	± 8	± 8	± 8
a/L^2	± 3	± 3	± 2	± 3

Table 2c Maximum Errors with Sinusoidal Boundary
Condition -- Sphere ($m = 2$)

r^* f	.73	.75	.78
$2a/L^2$	± 9	± 6	± 4
a/L^2	± 4	± 3	± 1
$a/2L^2$	± 2	± 1	± 0

The results showed:

- 1) The optimum location of the temperature sensor is essentially independent of t^* and the Biot modulus for $t^* > 0.2$ and $Bi \leq 6.4$.
- 2) The optimum location of the temperature sensor for a constant flux or a convective boundary condition is at r^* equal to 0.55, 0.69 and 0.75 for a slab ($m = 0$), cylinder ($m = 1$), and sphere ($m = 2$), respectively.
- 3) The increased error caused by the slight shift in the optimum location with increasing frequency for the sinusoidal boundary condition is insignificant.
- 4) The Biot modulus with a surface or centerline sensor location had to be less than 0.1 in order to obtain acceptable results. A Biot modulus almost two orders of magnitude larger is permissible if the sensor is at its opti-

mum location.

- 5) The time period, after an abrupt change, during which the transient calorimeter does not give meaningful results, has been essentially cut in half from $t^* < 0.4$ to $t^* < 0.2$.
- 6) The frequency of all significant harmonics with a surface or centerline sensor location had to be less than $a/16L^2$. Frequencies an order of magnitude higher can be tolerated if the sensor is located near the optimum location.
- 7) The error is a strong function of r^* only for r^* near 0 or 1. The error does not vary drastically from that obtained with the sensor located at its optimum location, as long as it is within approximately 0.05L of the optimum location.

The improved performance of the transient calorimeter with the sensor placed at its optimum location is drastic. Since the best location of the sensor is relatively independent of the Biot modulus, dimensionless time, and the frequency of periodic fluctuations, the results of this study should be of general value and applicable to more complex boundary conditions.

DETERMINATION OF dT/dt

Consider next the problem of accurately calculating the required derivative, dT/dt (see Eq. 2). Typically, a thermocouple is employed as the temperature sensor in the calorimeter. The EMF versus time recording is converted to $T = f(t)$ and then the desired derivative is usually determined by numerical differentiation. However, errors or noise in the experimental measurements coupled with those introduced in the differentiation often cause considerable scatter in the calculated values of q with this procedure. In addition, the investigator does not obtain a direct indication while taking the data of the quantity which is usually of greatest interest, i.e., the heat flux or dT/dt .

Since direct measurements of dT/dt are generally not possible, the main alternative to numerically or graphically differentiating $T = f(t)$ is to do it electronically. The specific problem is to generate from the EMF output of a thermocouple, a voltage which is proportional to the rate of change of the temperature of this thermocouple with respect to time.

Electronic or analog differentiation, like numerical differentiation, is much more difficult than integration. Some general purpose equipment is available, but an instrument which is specially designed to accurately and conveniently differentiate a thermocouple signal does not appear to be commercially available. However, high quality, low cost operational amplifiers are now available from which differentiators can be constructed using well known circuits (see e.g. Ref. [2]).

The specialized circuit which was developed for heat transfer studies is given in Fig. 2. The first stage of the circuit is an amplification of the thermocouple signal. This signal conditioner has an adjustable gain such that its output can be made independent of the type of thermocouple used. Furthermore, the ability to subtract off from the time varying temperature some constant reference or datum value such as the ambient fluid temperature, the initial temperature or, in a boiling study, the saturation temperature of the fluid, is provided. The datum or reference temperature, T_a , should not be confused with the thermocouple reference junction temperature, T_{ref} . The input resistor, R_1 , is a 1K Ω fixed resistor and a 10K Ω

assumption can be easily shown to be:

$$(T - T_{\infty}) = - \frac{\epsilon_{01} R_1}{a_1 R_{f1}} \quad (16)$$

Likewise, the relationship between the output of the differentiating stage, e_{O2} , and dT/dt can be shown to be:

$$\frac{dT}{dt} = \frac{e_{02}}{(C_2 R_{f2})} \frac{R_1}{a_1 R_{f1}} \quad (17)$$

It is suggested that the components R_1 , R_{F1} , C_2 and R_{F2} be chosen such that the output of the first stage in millivolts is numerically equal to $(T - T_0)$ in degrees ($^{\circ}\text{C}$ or $^{\circ}\text{F}$). Likewise, the output of the second stage in millivolts can be simultaneously made numerically equal to dT/dt in degrees/second or degrees/minute. It is noted that a single throw, double pole switch is provided to simultaneously change both C_2 and R_{F2} ; hence, the characteristic time of the differentiator is either one second or one minute. Specifically, if $a_1(R_{F1}/R_1) = 1 \text{ mV}/^{\circ}\text{C}$, then

$$(T - T_{\infty}) = -e_{01} (^\circ\text{C}/\text{mV}) \quad (18)$$

Likewise, with characteristic times of one second and one minute, one obtains from Eq. (17), respectively:

Fig. 2 Differentiator Circuit

potentiometer connected in series as shown in Fig. 2. The 1K Ω fixed resistor limits the gain of the amplifier to 100 to avoid saturating it accidentally. The 10K Ω potentiometer enables the gain to be set anywhere between 9.1 and 100. An internal adjustable voltage source is used to compensate for the thermocouple reference temperature T_{ref} and the ambient temperature T_a . A ten-turn 20K Ω potentiometer is provided. One turn corresponds to $\pm 100^\circ\text{C}^1$; hence, a temperature offset anywhere between 0 and $\pm 1000^\circ\text{C}^1$ is possible. Alternatively, the specimen can be allowed to come to thermal equilibrium with the reference or datum temperature, T_a , if it is available. The ten-turn potentiometer could then be employed to zero the output of the first operational amplifier, e_{01} . A switch is provided, so if an additional thermocouple is available, for measuring T_a , it can remain connected, in order to correct for small changes in T_a between runs.

The EMF of some thermocouples such as iron-constantan and chromel-alumel varies almost linearly with temperature. That is, $e = f(T)$ can be approximated by

$$e = a_0 + a_1 T \quad (14)$$

where a_0 is determined by the reference junction temperature. If the same reference junction temperature is used for both T and T_0 , $(T - T_0)$ follows from Eq. (14) as

$$e(T - T_{ref}) - e(T_{\infty} - T_{ref}) = a_1(T - T_{\infty}) \quad (15)$$

The relationship between the output voltage of the first stage, e_{01} (see Fig. 2), and $(T - T_0)$ with this

$$\left. \begin{aligned} \frac{dT}{dt} &= e_{02} \frac{^{\circ}\text{C}}{\text{s} \cdot \text{mv}} \\ a_1 \frac{R_{f1}}{R_1} &= \frac{1 \cdot \text{mv}}{^{\circ}\text{C}} \end{aligned} \right\} \quad (19)$$

$$\frac{dT}{dt} = e_{02} \frac{^{\circ}\text{C}}{\text{min mv}} \quad \left\{ \begin{array}{l} a_1 \frac{R_{f1}}{R_1} = 1 \frac{\text{mv}}{^{\circ}\text{C}} \\ c_2 R_{f2} = 1 \text{ min} \end{array} \right. \quad (20)$$

The temperature unit can be changed from $^{\circ}\text{C}$ to $^{\circ}\text{F}$ by an appropriate change in R_1 . Alternatively, the output of the second amplifier in millivolts could be made numerically equal to the heat flux; however, the properties of the calorimeter would then have to be introduced. This option is still possible with the circuit given in Fig. 2 with an appropriate adjustment in the gain of the first amplifier. However, one then forfeits having the output of the first amplifier numerically equal to $(T - T_0)$.

Three components, C_{R1} , C_{R2} , and R_2 , have been added to limit the gain at high frequencies and to provide dynamic stability. If any of the time constants $R_1 C_{R1}$, $R_2 C_{R2}$, or $R_2 C_2$ are of the same magnitude or larger than the reciprocal of the frequency of interest, $1/s$, a magnitude error and a phase shift will occur. In the circuit developed, $R_1 C_{R1}$ is either 0.5 or 0.05 times $C_{R2} R_2$; hence, its influence will be neglected. Table 3 gives the expected errors due to the combined effects of $R_2 C_{R2}$ and $R_2 C_2$. A double-pole, two position, center off switch is used (see Fig. 2) to simultaneously change C_{R1} and C_{R2} . The effective time constant of the filter can be varied in six steps between 0.002 and 2 seconds—a range suitable for accommodating most heat transfer processes.

¹See Eq. (18) and associated discussion.

Table 3 Errors Due to Noise Filters (2)
(With $R_{f2}C_{f2} = R_2C_2 \equiv \tau_c$)

Signal Frequency	$1/\tau_c$	$.1/\tau_c$	$.01/\tau_c$
Magnitude Error	-50%	-1%	--
Phase Error	-90°	-11°	-1°

The advantages gained with the differentiator can be partially deduced from Fig. 3. This figure shows two traces made with an X-Y plotter for pool

Fig. 3 Comparison of $T = f(t)$ with Direct Recording of Boiling Curve

boiling heat transfer from a 1/2 in. diameter copper cylinder in water at the saturation temperature. The one trace is time versus $(T - T_{sat})$ and the other is dT/dt versus $(T - T_{sat})$, that is, a direct recording of the boiling curve. $(T - T_{sat})$ was used as the abscissa for both curves; hence, the ordinate of the dT/dt is equal to the slope of the other curve for any given value of the abscissa. This figure shows the difficulty one would encounter in determining the maximum heat flux, and the value of $(T - T_{sat})$ which corresponds to this maximum, from the nearly horizontal line of the time versus $(T - T_{sat})$ curve.

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Appendix C

DATA INTERPRETATION

Appendix C

DATA INTERPRETATION

To aid in the understanding of the experiments, SRI asked Dr. Robert Moffat, Professor of Mechanical Engineering and Director of Thermosciences at Stanford University, to participate in the review. Dr. Moffat is a nationally recognized expert in the field of experimental methods and heat transfer.

As a result of discussions between Dr. Moffat and SRI, we concluded that it was questionable whether an internally heated cavity would behave like an externally heated cylinder. The method used to determine the heat-transfer coefficient did not account for losses through the insulation or into the copper-constantan thermocouple wire during the short-duration transient tests. Any heat loss would thus cause a higher heat-transfer coefficient to be predicted. The loss through the insulation was estimated to be on the order of 10 percent; however, the unsteady heat flow into the copper wire could account for considerably more.

We reasoned that the presence of nontypical axial flow within the receiver model could result in near-identical heat-transfer coefficients between forward and reverse flow conditions. End caps were placed on the model before the experiment to simulate long receiver geometry; however, the caps only covered the diameter of the cavity, and did not extend beyond the model to stop axial flow. The 5-cm clearance between the model and the wall of the wind tunnel would provide a flow channel past the model, which could encourage axial flow. Flow into the receiver cavity would stagnate at the center of the model and move axially until spilling into the side streams. During flow into the back of the receiver, the high-velocity side streams would create a low-pressure area in front of the receiver cavity. Flow into the

midspan of the receiver cavity, caused by entrainment of the high-velocity side streams, would generate stream paths similar to those for forward flow conditions. None of these conditions are typical, which could explain the data trends.

We agreed that of the two sets of data--cryogenic transient tests and room-temperature transient tests--the room-temperature tests are less likely to exhibit large errors from thermal losses. However, the nontypical flow conditions described above would still be present, and thus high heat-transfer coefficients would be predicted.

We recommend that further tests be performed using steady-state, room-temperature flows. Further, air temperature should be measured within the cavity, to accompany the cavity surface-temperature measurements.

Additional tests of the forced-convection heat loss will be required before going into final design.

Appendix D

RESULTS FROM THE WATER-TABLE TESTS

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SRI performed water-table flow tests to obtain a visual record of flow paths and relative velocities outside and inside the cavity of the receiver model. Both "upwind" and "downwind" experiments were performed. In addition, baffles were strategically placed to minimize circular flow within the receiver cavity. These panels can be rotated and will be deflected to a horizontal position during extreme wind conditions. The scale model of the receiver, shown in Figure 1(a) and (b), is the same size and shape as the one tested in the cryogenic wind tunnel at the University of Illinois. It has a single eyelid above the cavity. The radial thickness is greater in proportion to the receiver diameter than the actual receiver. The free stream velocity is ~ 6.75 in/sec. Velocity in the cavity is ~ 0.45 in/sec, about 7 percent of free stream velocity. Rotation is counterclockwise. (The two large streaks in the photographs are reflections of the camera tripod legs.)

Figure 1(b) shows the same model with a deflection vane attached below the cavity aperture. Flow traces show the free stream water velocity at the same ~ 6.75 in/sec, whereas the cavity velocity now approaches zero, about 0.1 in/sec, or about 1-1/2 percent of free stream water velocity. Note the movement of the flow stagnation point to near the center of the cavity aperture. It appears that this modification would reduce forced-convection losses to an acceptable level.

Figure 2(a) and (b) shows the scale model, closer to scale in thickness, of a two-eyelid design undergoing forward flow. In Figure 2(a), the free stream water velocity is ~ 6.3 in/sec. The cavity velocity is ~ 0.68 in/sec, or about 10 percent of free stream velocity. Rotation is again counterclockwise.

(a) WITHOUT DEFLECTOR

(b) WITH DEFLECTOR

FIGURE 1 SINGLE EYELID RECEIVER — FORWARD FLOW

(a) WITHOUT DEFLECTOR

(b) WITH DEFLECTOR

FIGURE 2 TWO EYELID DESIGN — FORWARD FLOW

(a) WITHOUT DEFLECTOR

(b) WITH DEFLECTOR

FIGURE 3 ORIGINAL MODEL — REVERSE FLOW

(a) WITHOUT DEFLECTOR

(b) WITH DEFLECTOR

FIGURE 4 TWO EYELID MODEL — REVERSE FLOW

In Figure 2(b), a deflection vane is added below the receiver. Flow inside the cavity approaches zero velocity. Free stream water velocity is ~ 6.75 in/sec. Note that the stagnation point is now below the aperture.

Figure 3(a) shows the original model in a reverse flow of 7.2 in/sec and a velocity in the cavity of ~ 0.23 in/sec, about 3 percent of free stream water velocity. When the deflection plate is added [Figure 3(b)], there is essentially no flow in the cavity with a free stream water velocity of ~ 9.45 in/sec.

In Figure 4(a) and (b), the two-eyelid model is seen in a reverse water flow of ~ 6.76 in/sec. In Figure 4(a), some slight motion (about 0.5 in/sec) is observed in the cavity, without the lower deflection vane (7 percent of free stream velocity).

Finally, when the deflection vane is added in a flow of 6.75 in/sec [Figure 4(b)], there is no movement in the cavity, and very little in a large space in front of the aperture.

These water-table tests indicate that the velocity in the cavity and the resultant convection heat losses in the line-focus central power receiver may be satisfactorily controlled by the correct aerodynamic design of the receiver and its surroundings. We hope to verify this with the proposed wind-tunnel tests.