AN EXPERIMENTAL INVESTIGATION OF NATURAL AND COMBINED CONVECTION FROM AN ISOTHERMAL HORIZONTAL PLATE

BY

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THESIS

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DEDICATION

To my loving family.
ACKNOWLEDGEMENT

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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. INTRODUCTION</td>
<td>10</td>
</tr>
<tr>
<td>2. THE WIND TUNNEL FACILITY</td>
<td>12</td>
</tr>
<tr>
<td>2.1 The Cryogenic Approach to Convective Heat Transfer</td>
<td>12</td>
</tr>
<tr>
<td>2.2 Design Concept</td>
<td>13</td>
</tr>
<tr>
<td>2.2.1 Flow Circuit Modifications</td>
<td>13</td>
</tr>
<tr>
<td>2.2.2 Cooling System Modifications</td>
<td>17</td>
</tr>
<tr>
<td>2.2.3 Other Modifications</td>
<td>18</td>
</tr>
<tr>
<td>2.3 Data Acquisition</td>
<td>19</td>
</tr>
<tr>
<td>2.3.1 Thermocouple Network</td>
<td>20</td>
</tr>
<tr>
<td>2.3.2 Data Storage Process</td>
<td>20</td>
</tr>
<tr>
<td>2.4 Flowfield Testing</td>
<td>20</td>
</tr>
<tr>
<td>2.4.1 Velocity Distribution</td>
<td>20</td>
</tr>
<tr>
<td>2.4.2 Turbulence Measurements</td>
<td>23</td>
</tr>
<tr>
<td>3. FLAT PLATE LITERATURE SURVEY</td>
<td>25</td>
</tr>
<tr>
<td>4. SIMILITUDE CONSIDERATIONS</td>
<td>28</td>
</tr>
<tr>
<td>5. FLAT PLATE TEST MODEL</td>
<td>30</td>
</tr>
<tr>
<td>5.1 Thermal Design</td>
<td>30</td>
</tr>
<tr>
<td>5.2 Dynamic Design</td>
<td>32</td>
</tr>
<tr>
<td>6. EXPERIMENTAL PROCEDURE</td>
<td>33</td>
</tr>
<tr>
<td>6.1 Prerequisites to Testing</td>
<td>33</td>
</tr>
<tr>
<td>6.2 Calculation of the Heat Transfer Coefficient</td>
<td>33</td>
</tr>
<tr>
<td>7. EXPERIMENTAL RESULTS</td>
<td>37</td>
</tr>
<tr>
<td>7.1 Natural Convection Results</td>
<td>37</td>
</tr>
<tr>
<td>7.2 Combined Convection Results</td>
<td>41</td>
</tr>
</tbody>
</table>
8. RESULTS AND CONCLUSIONS 44
REFERENCES 45
APPENDIX A 46
APPENDIX B 50
APPENDIX C 59
APPENDIX D 67
NOMENCLATURE

\( A \) Area (m\(^2\))
\( c_p \) Specific heat (J/kg-K)
\( f \) Defined by Eq. (4.2)
\( g \) Acceleration of gravity, 9.81 m/s\(^2\), or defined by Eq. (4.2)
\( h \) Convective heat transfer coefficient (W/m\(^2\)-K)
\( k \) Thermal conductivity (W/m-K)
\( L \) Characteristic length (m)
\( m \) Mass (kg)
\( p \) Pressure (N/m\(^2\))
\( q'' \) Convective heat flux (W/m\(^2\))
\( R \) Gas constant (J/kg-K)
\( t \) Time (s)
\( T \) Temperature (K)
\( T^* \) Dimensionless temperature, \((T - T_x)/(T_o - T_x)\)
\( u \) Horizontal component of the velocity vector (m/s)
\( v \) Vertical component of the velocity vector (m/s)
\( U_\infty \) Freestream velocity (m/s)
\( z \) Defined by Eq. (4.4)

Greek

\( \alpha \) Thermal diffusivity (m\(^2\)/s)
\( \beta \) Volume coefficient of expansion (K\(^{-1}\))
\( \delta \) Boundary layer thickness (m)
\( \Delta T \) Temperature difference, \((T_w - T_x)\) (K)
\( \varepsilon \) Emissivity
\( \rho \) Density (kg/m\(^3\))
\( \sigma \) Stefan-Boltzmann constant, 5.67\( \times \)10\(^{-8}\) W/m\(^2\)-K\(^4\)
\( \mu \) Dynamic viscosity (kg/m-s)
\( \nu \) Kinematic viscosity (m\(^2\)/s)
Subscripts

com Combined convection

\( f \) Properties based on the film temperature; \( T_f = (T_w + T_\infty)/2 \)

\( r \) Reference temperature

\( R \) Radiation

\( M \) Momentum

\( nat \) Natural convection

\( T \) Thermal or total

\( w \) Wall property

\( \infty \) Freestream or ambient value

\( o \) Initial condition

Superscripts

\(*\) Dimensionless quantity

\( \prime \) Time-dependent component

Dimensionless groups

\( Bi \) Biot number, \( hL/k \)

\( Gr \) Grashof number, \( g\beta\Delta T L^3/\nu^2 \)

\( Nu \) Nusselt number, \( hL/k \)

\( Pe \) Peclet number, \( Re Pr \)

\( Pr \) Prandtl number, \( \nu/\alpha \)

\( Ra \) Rayleigh number, \( Gr Pr \)

\( Re \) Reynolds number, \( \rho U_\infty L/\mu \)

\( Ri \) Richardson number, \( Gr/Re^2 \)

\( Tu \) Turbulence Intensity, \( \sqrt{u'^2}/U_\infty \)
1. INTRODUCTION

“It is not the brains that matter most, but that which guides them — the character, the heart, generous qualities, progressive ideas,” wrote Dostoevsky [1] in 1861. One such progressive idea lies in the field of convective heat transfer research: the realization that tremendous gains in buoyancy-driven and inertial forces can be achieved by lowering the temperature of the surrounding fluid. Details of these advantages are given in the following chapter and by Clausing [2]. To make use of these gains, a cryogenic heat transfer facility was built at the University of Illinois at Urbana-Champaign (UIUC). Many valuable data were gathered using this wind tunnel, but the results were often criticized because the flowfield in the test section was marred with appreciable turbulence levels and velocity gradients. Last year, extensive remodeling was performed on the facility to help alleviate these problems. For a complete description of the original facility, refer to Mueller, et al. [3], while the recent modifications are presented in detail in the following chapter.

The renovated wind tunnel was the host for an investigation of the natural and combined convective heat loss from a heated, square, upward-facing, horizontal flat plate to a surrounding gas medium. The following are the results of this experimental investigation.

(i) An improved correlation is provided for the natural convection regime by accounting for variable property effects. When the ratio of the absolute temperature of a heated object to the absolute ambient temperature, \( T_w/T_\infty \), is appreciably greater than unity, the physical properties of the fluid within the thermal boundary layer vary markedly. The density of air, for example, doubles in value when the temperature drops to 160 K. Most analytical and experimental research performed during the past few decades has ignored all property variations except the basic density differences which generate the buoyancy force. This approach, which will be referred to as the constant property case or the constant property assumption, is valid as long as \( T_w/T_\infty \) is near unity. Many applications exist, however, where \( T_w/T_\infty \) is much greater than unity. The solar receiver of the 10 MW\( e \) power plant in Barstow, California reaches temperature ratios as high as four. In these applications, the choice of the reference temperature at which the properties are evaluated in constant property correlations can strongly influence the values of the dimensionless parameters which govern the rate of convective heat transfer. In this investigation, a function in the heat transfer correlation accounts for variable property effects, providing a more accurate reckoning of the convective loss.

(ii) The cryogenic facility provides an excellent opportunity to extend the applicable range of the correlation over previous research. Dramatic changes in the Rayleigh number, \( Ra \), can be achieved by varying the ambient temperature, \( T_\infty \), without masking the results with radiative heat transfer. Thus, Rayleigh numbers presented here are up to 2000 times greater than those given under similar conditions by other investigators (See, e.g., [4], [5], [6], and [7]).
(iii) Nearly all available data for flat plates are for either pure free or for pure forced convection. Presented in this study are data for combined convection, or convection where both inertial and buoyant forces are of importance. A matter of concern for many researchers performing natural convection experiments is the presence of extraneous laboratory drafts or currents that could adversely influence their results. Experimenters often go to great lengths to ensure that drafts caused by doors opening or people moving about do not occur. The natural convection limit, or the Richardson number, $Ri$, at which combined convection becomes significant, is resolved in this analysis.

The ranges of the dimensionless parameters that are investigated in this study are: $2 \times 10^8 < Ra < 2 \times 10^{11}$, $1 < Tw/T_\infty < 3.1$, and $0.02 < Ri < \infty$. The pure natural convection, large Rayleigh number regime and the combined convection regime near the natural convection limit are emphasized.
2. THE WIND TUNNEL FACILITY

2.1 The Cryogenic Approach to Convective Heat Transfer

One problem that researchers performing convection experiments often encounter is the difficulty in obtaining sufficiently large Grashof numbers, $Gr$, and/or Reynolds numbers, $Re$. Large values of these parameters are often required to accurately model full-scale phenomena such as the convective heat loss from the solar receiver mentioned earlier. Large characteristic lengths and velocities are required in full-scale tests in order to achieve similarity conditions. This technique is often very expensive and also lacks the advantage of having controlled laboratory conditions. Rather than increasing the lengths and velocities to obtain large Grashof and Reynolds numbers, the other viable alternative is to change the thermophysical properties of the surrounding fluid. Assuming a temperature power law variation of the fluid properties, Clausing [2] shows that

$$Gr \sim p^2 \frac{(\Delta T)}{L^3} \frac{T^{4.8}}{T} \quad (2.1.1)$$

for air and nitrogen. Similarly, for the Reynolds number,

$$Re \sim p \frac{(U_\infty)}{L} \frac{T^{1.9}}{T} \quad (2.1.2)$$

The reduction of temperature greatly increases the values of these parameters, which can be seen graphically in Fig. 2.1.1. Constant values of $p$, $U_\infty$, $T$, and $L$ are assumed. Reducing the ambient temperature is an ideal way to generate high Reynolds numbers and an even better way to generate high Grashof or Rayleigh numbers. A variable ambient temperature facility also enables the researcher to cover large ranges of these dimensionless groups without changing models.

Consider next the reduction of radiative heat transfer at cryogenic temperatures. Room temperature tests at high Rayleigh numbers necessitates the extraction of the convective component of heat transfer from a large radiative component. The advantages of a cryogenic environment are apparent. For metallic surfaces, the hemispherical, total emissivity decreases linearly with decreasing temperature. Clausing [2] compares a test at 320 K to a cryogenic test at 80 K with equal values of $T_w/T_\infty$. The decrease in ambient temperature results in a reduction of radiated energy by a factor of approximately 1000. Thus, the radiative mode of heat transfer is virtually eliminated from tests conducted at cryogenic temperatures, providing a more accurate estimate of the convective loss.

Other advantages of an enclosed cryogenic facility include the ability to deduce the influence of variable properties by generating large ratios of $T_w/T_\infty$ and the elimination of extraneous convective currents which could affect the data.

The ideal facility to perform tests in natural, forced, and combined convection would be an elevated pressure, cryogenic wind tunnel. Since a sealed, pressurized chamber is more expensive to construct, a pure cryogenic ambient pressure wind tunnel was built here at the University of Illinois in 1977.
2.2 Design Concept

The wind tunnel facility prior to the recent modifications is illustrated in Fig. 2.2.1. It is a variable-speed, recirculating, liquid-nitrogen-cooled wind tunnel which uses gaseous nitrogen as the working fluid when operating at cryogenic temperatures. The use of liquid nitrogen permits the test section temperature to be varied between 310 K and 80 K. The test section measures 1.2 m in height and 0.6 m in width. The large vertical dimension of the test section was chosen to provide room for buoyant flow from the models. A 11.5 kW, variable-speed DC motor drives two 0.5 m diameter cast aluminum fans. Fan swirl in the flow is reduced by a honeycomb flow straightener located just downstream of the fans. The average turbulence Intensity in the original test section was 2.3 percent.

For a complete description of the initial design of the facility, refer to Mueller, et al. [3]. The high turbulence levels, velocity gradients, and temperature stratification In the test section prompted the redesign of the flow circuit and the cooling system.

2.2.1 Flow Circuit Modifications

Before entering the test section, the flow of the original facility passed through two sets of turning vanes and only one settling screen, which resulted in turbulent, non-uniform flow. This design was
improved by replacing the turning vanes with constantly-converging curved walls and by installing four additional settling screens. The converging walls decrease the magnitude of the adverse pressure gradient, thereby reducing the amount of flow separation along the inside wall. The remodeled facility is illustrated in Figs. 2.2.2 and 2.2.3. The path of the flow is further refined by the addition of two guiding walls mounted within the turn. These walls prohibit the flow from “hugging” the outside wall as it rounds the bend. The direction and the mass flux of each stream is further controlled by four adjustable hinged vanes fastened to the endpoints of the guiding walls (See Fig. 2.2.2). All walls are constructed of heavy-gauge sheet metal and are sealed at the seams with a silicone caulking compound to prevent the infiltration of outside air and moisture.

To accommodate the greater width of the test section, the flow is expanded in a diverging section after leaving the converging turn. The five settling screens serve the dual purpose of reducing the turbulence Intensity and to expand the flow to the dimensions of the test section.
Fig. 2.2.1 Cross-Sectional, Top View of Previous Cryogenic Facility
Fig. 2.2.2 Cross-Sectional, Top View of Remodeled Cryogenic Facility
2.2.2 Cooling System Modifications

In the original design, cooling was accomplished by filling the corrugated hollow walls which lined the perimeter of the tunnel with liquid nitrogen. This method had the advantage of cooling the mass of the tunnel very quickly. One problem occurred, however, when one attempted to run the tunnel at only moderately cold temperatures. To conduct a test with an ambient temperature of, say, 200 K required the walls to be only partially filled with liquid nitrogen. The lower portion of the walls would be at a much lower temperature, resulting in a large temperature stratification from top to bottom. A model with significant vertical dimensions would be subjected to a varying ambient temperature. To resolve this problem, the liquid had to be purged from the walls before an experiment could begin.

To simplify this procedure, the uneven corrugated walls were removed and were replaced with smooth sheet metal. This, incidentally, improved the quality of the flow in the circuit. A cascading, finned-tube heat exchanger was built and installed just downstream of the fans, as can be seen in Fig. 2.2.3. The heat exchanger is made of finned copper piping and all joints were sweat soldered into place.

Fig. 2.2.3 End View of Test Section and Liquid Nitrogen Vaporizer
The heat exchanger also serves as an additional flow straightener to reduce fan swirl. Nine copper-constantan thermocouples are mounted in the heat exchanger to monitor the level of liquid nitrogen within. The liquid nitrogen is supplied from pressurized dewars in the laboratory and is fed into the heat exchanger, where it is vaporized. Excess liquid is collected in a residue pan located behind the fans, where it boils into the tunnel environment. The level of liquid in the drip pan, which is monitored by two additional thermocouples, is kept to a minimum by regulating the flow from the dewars. The cooling process can be hastened somewhat by running the fans at a low speed, thus increasing the efficiency of the heat exchanger.

### 2.2.3 Other Modifications

An operating post was constructed next to the tunnel where the experimentalist has the controls of the model heating system within reach. Ten direct current model heating circuits are powered by two Sorenson and two Hewlett-Packard variable-output regulated DC power supplies. A typical heating circuit is schematically illustrated in Fig. 2.2.4. The power supplied to each heater can be determined by measuring the voltage drop across the heater and across the one-ohm shunt resistor. This design allows, if desired, steady-state heat transfer experiments to be conducted. Thirteen additional alternating current heating circuits are powered through wall outlets and are controlled by conventional dimmer switches. Combining these heating circuits with a set of regulating switches allows the model temperatures to be adjusted to within ±0.1 K of the desired temperature.

![Fig. 2.2.4 Schematic of Typical Model Heating Circuit](image)

The Kiel probe mounted in the test section wall (See Fig. 2.2.1) was removed to allow room for a solar cavity model. To take its place, a pitot tube was mounted in the ceiling of the tunnel 0.2 m downstream of the final settling screen. The pitot tube is connected to a Model 239 Setra 138 N/m² capacitance pressure transducer with an accuracy of ±0.2 N/m². Using the calibration standard provided by the Setra Corporation, temperature power law variations of density, and Bernoulli’s equation, the velocity transducer chart for various ambient temperatures is generated and is shown in Fig. 2.2.5.
Following the installation of a new galvanized sheet steel ceiling, the entire exposed exterior of the facility was sprayed with a layer of urethane foam insulation. In addition to providing protection against heat gain, the foam acts as an excellent vapor barrier to keep out unwanted outside air and moisture. Since the nitrogen entering the tunnel by way of the heat exchanger must somehow exit the tunnel, a two-inch diameter hole was drilled in the ceiling. Two-inch plastic PVC piping is attached to this opening, allowing the upper, warmer nitrogen gas to exhaust the tunnel. The nitrogen exhaust can be either routed outside the laboratory or channeled into the tunnel roof insulation, where it creates a helpful cooling effect. Caution is exercised when using the latter method by assuring adequate laboratory ventilation. The exhaust ports must be shut off when operating the fans, or the partial vacuum created by the flow will induce infiltration.

2.3 Data Acquisition

Convection data from the experimental models are gathered by recording their temperature as a function of time. A network of thermocouples is used in this process. The original data acquisition process required the manufacture of a new thermocouple arrangement for each new model to be tested. This process has been standardized and “hard-wired” for the latest series of experiments. A new data storage process has also been designed and is described below.
2.3.1 Thermocouple network

An isothermal area for electrical connections was built into which the thermocouple wires from any model can be plugged. This area is thermally insulated to prevent error-producing thermal gradients from forming between the electrical junctions. This isothermal connecting area greatly simplifies the thermocouple connecting process, consequently allowing more than one model to be tested. A permanent wiring trunk connects the isothermal area to two Kaye ice point units, where the cold junctions are referenced to 0°C. The wiring trunk then links to a Fluke Model 22408 datalogger, where up to sixty thermocouples are scanned at a continuous rate of twelve channels per second. The datalogger records the thermocouple voltage potentials as a function of time and sends them to the RS 232 port of a Texas Instruments Portable Professional Computer for further processing.

2.3.2 Data Storage Process

In a program written for the portable computer by A.M. Clausing, the voltage potentials of each channel are converted to temperatures via a fourth-order fit of the copper-constantan thermocouple data in Ref. [8]. These temperatures are averaged according to user specifications and are graphically displayed as a function of time on the screen of the computer. In this manner, the temperatures of a model consisting of several calorimeters can be displayed coincidentally. When all calorimeters reach the desired initial temperature, the test may begin. The temperature vs. time data for each calorimeter are written from RAM on a diskette and are subsequently uploaded via telephone link to the mainframe CYBER 175 computer for further processing and plotting.

2.4 Flowfield Testing

After the remodeling of the tunnel, the hinged vanes in the converging turn required adjustments to produce the most uniform test section flow possible. A TSI Model 1650 hot wire air velocity meter was used in this task. An iterative procedure was followed in which the four vanes were slightly adjusted and a series of velocity measurements in the test section were taken. This was done until the velocity measurements taken in the lateral direction achieved a minimum standard deviation. The vanes were then permanently secured into the “optimum” position.

2.4.1 Velocity Distribution

To test the uniformity of the flow in the test section, the probe of the TSI velocity meter was attached to a linear translating device capable of precise movement in both horizontal and vertical directions. The probe was inserted through a slit in the test section wall and flow was initiated. A minimum of two hours was allowed for the tunnel and the probe to reach thermal equilibrium. Each velocity scan consists of 420 individual velocity readings, providing a very accurate measurement of the velocity distribution. The velocity data were fed into a computer program which generated the contours of constant velocity shown
in Figs. 2.4.1 and 2.4.2. Fig. 2.4.1 shows the normalized contours for the upstream portion of the test section, just 0.3 in from the last settling screen. Fig. 2.4.2 shows a similar result taken at the same fan speed 0.4 m further downstream. The similarity of the contours measured at different locations in the test section illustrates the uniformity of the flow as it moves downstream. To determine the degree of velocity stratification in the test section, an area-weighted average of the velocity deviations from the mean velocity is used. The velocity uniformity of the flowfield is found to be 1.5 percent.

The time required for the flow to cease after cutting the power to the fans is approximately 15 seconds. The fading histories for initial velocities of 1, 2, and 3 m/s are recorded in Fig. 2.4.3 by using a TSI model 1640 omni-directional low velocity meter. Thus, the fans must be turned off at least 15 seconds prior to beginning a pure natural convection test.

![Normalized Velocity Contours](image)

**Fig. 2.4.1 Upstream Contours of Constant Velocity**
Fig. 2.4.2 Downstream Contours of Constant Velocity

Fig. 2.4.3 Velocity Decay in Test Section Following Power Cut-off
2.4.2 Turbulence Measurements

The turbulence measurements were made by using a TSI Model 1050 constant temperature anemometer and signal linearizer and a TSI Model 1210 hot film sensor. The constant temperature arrangement uses a feedback control circuit to vary the current supplied to the film sensor, thereby keeping its temperature constant. In this manner, variations in the heat transfer rate caused by turbulence are represented as voltage fluctuations which are observed on an RMS voltmeter. The turbulence intensity is calculated by dividing the RMS signal by the average voltage shown on a normal voltmeter. Fig. 2.4.4 shows the arrangement of the equipment. The linearizer is required so that the velocities can be represented as a linear function of the measured voltages.

Using a Wavetek-Rockland Model 5830A digital signal analyzer, the frequency distribution of the turbulence is deduced and is shown in Fig. 2.4.5. This figure shows that nearly all turbulence originates from frequencies less than 1 kHz.

The turbulence measurements involve the same traversing procedure of the test section as did the velocity measurements. A full scan of the test section 0.3 m from the final settling screen results in an average turbulence intensity of 1.7 percent.

Fig. 2.4.4 Sketch of Turbulence Equipment Arrangement
Fig. 2.4.5 Turbulence Frequency Distribution of Flow in Test Section
3. FLAT PLATE LITERATURE SURVEY

The previous natural convection research performed on isothermal horizontal flat plates did not consider the effects of variable properties in the thermal boundary layer. The results of these past experiments differ greatly. These discrepancies alone, as well as the desire to investigate the effects of high temperature ratios, warrant a more accurate analysis of the problem. Bosworth [9], for example, reports the following Nusselt number, $Nu$, correlations for the natural convection case of the title problem:

$$\begin{align*}
Nu &= 0.71 \, Ra^{1/4}, \quad \text{(laminar regime)} \\
Nu &= 0.17 \, Ra^{1/3}, \quad \text{(turbulent regime)}
\end{align*}$$

He gives no details about the applicable range, transitional Rayleigh number, model shape, characteristic length, or working fluid which is used in the experiment.

Bosworth’s turbulent correlation is more than 25 percent greater than the horizontal plate correlation given by Hassan and Mohammed [4]:

$$\begin{align*}
Nu &= 0.135 \, Ra^{1/3}, \quad 10^3 < Ra < 10^8 \\
Nu &= 0.14 \, Ra^{1/3}, \quad 10^8 < Ra < 3 \times 10^{10}
\end{align*}$$

They conducted their experiment in air and used heat flux meters to measure the convective loss. Their experiments involved both heated and cooled rectangular plates at various angles of inclination. The radiative heat was always less than ten percent of the total loss. The most often-quoted correlation for horizontal plates is that given by Fishenden and Saunders [10]. They reported in 1950 the correlations for square plates in air:

$$\begin{align*}
Nu &= 0.54 \, Ra^{1/4}, \quad 10^5 < Ra < 10^8 \\
Nu &= 0.14 \, Ra^{1/3}, \quad 10^8 < Ra < 3 \times 10^{10}
\end{align*}$$

To achieve the high Rayleigh numbers for the turbulent relation, an elevated pressure chamber was used. Fishenden and Saunders cite tests with pressures as high as 60 atm and temperature differences greater than 1000°F. In comparison, the highest Rayleigh number achieved in the title study using the cryogenic technique is $2 \times 10^{11}$.

A study of the convective loss to air from plates of different shapes was conducted by Al-Arabi and El-Riedy [6]. They report for horizontal squares, rectangles, and circles the correlations:

$$\begin{align*}
Nu &= 0.700 \, Ra^{1/4}, \quad 2 \times 10^3 < Ra < 4 \times 10^7 \\
Nu &= 0.155 \, Ra^{1/3}, \quad 4 \times 10^7 < Ra < 3 \times 10^9
\end{align*}$$

Both the average and the local Nusselt numbers were determined.
The convective heat loss of rectangular plates submerged in water and at various angles of inclination is described in a study by Fujii and Imura [7]. For the case of the horizontal plate, they report the correlations:

\[ \text{Nu} = 0.16 \, \text{Ra}^{1/3}, \quad 7 \times 10^6 < \text{Ra} < 2 \times 10^8 \]  \hspace{2cm} (3.8)

\[ \text{Nu} = 0.13 \, \text{Ra}^{1/3}, \quad 5 \times 10^8 < \text{Ra} < 5 \times 10^{10} \]  \hspace{2cm} (3.9)

Two correlations are given because two models were involved in the investigation; the larger model accounts for the larger Rayleigh number data. All data were taken in the turbulent regime.

Mass transfer techniques were used to determine the convection rates from various horizontal planforms by Goldstein, Sparrow, and Jones [11] and by Lloyd and Moran [12]. In this method, the similarity of the equations governing heat and mass transfer allows an analogy to be drawn between the two different processes. One advantage of using a mass transfer method to determine convection losses is that property variations across the boundary layer are essentially negligible. Goldstein, Sparrow, and Jones found for very low Rayleigh numbers

\[ \text{Nu} = 0.96 \, \text{Ra}^{1/6}, \quad 1 < \text{Ra} < 2 \times 10^2 \]  \hspace{2cm} (3.10)

\[ \text{Nu} = 0.59 \, \text{Ra}^{1/4}, \quad 2 \times 10^2 < \text{Ra} < 10^4 \]  \hspace{2cm} (3.11)

where \( \text{Nu} \), the Nusselt number, is the heat transfer equivalent of the parameter used in mass transfer, the Sherwood number. Lloyd and Moran, using a similar mass transfer technique, obtained the following correlations:

\[ \text{Nu} = 0.54 \, \text{Ra}^{1/4}, \quad 3 \times 10^4 < \text{Ra} < 8 \times 10^6 \]  \hspace{2cm} (3.12)

\[ \text{Nu} = 0.15 \, \text{Ra}^{1/3}, \quad 8 \times 10^6 < \text{Ra} < 2 \times 10^9 \]  \hspace{2cm} (3.13)

Both studies report that by defining the characteristic length to be the ratio of the convecting area to the model perimeter, the above relations correlate the data from all of the planforms investigated. Since the Schmidt numbers encountered in these analyses were on the order of 2200, the relations given are only applicable to convection problems with comparably high Prandtl numbers.

Several analytical solutions for a horizontal plate in natural convection have been presented (See, e.g., [13] and [14]), but they do not account for the effects of turbulence within the boundary layer and consequently are only applicable to low Rayleigh number, laminar flow.

Turning our attention to past research on horizontal surfaces in combined convection, Mori [15] reports the analytical result for \( Pr = 0.72 \) and \( T_w > T_x \):

\[ \frac{\text{Nu}}{\text{Nu}_{\text{Re} \rightarrow 0}} \approx 1 - 1.202 \frac{Gr}{Re^{2.5}} \]  \hspace{2cm} (3.14)

Where \( \text{Nu}_{\text{Re} \rightarrow 0} \) is the Nusselt number for pure forced convection. The theory is only valid, however, for values of \( Gr/Re^{2.5} \) less than 0.195 and for laminar flow. Therefore, the results of this analytical study
cannot be used to find the natural convection limit that is deduced in this investigation. Eq. (3.14) is a determination of the forced convection limit.

The results of this investigation will verify the accuracy of some of these previous correlations by providing a more precise relation.
4. SIMILITUDE CONSIDERATIONS

The advantages of performing a dimensional analysis of the problem are to reduce the number of relevant variables that need to be studied and to render the results to a form which requires no special units of measurement. The set of dimensionless groups which govern the heat transfer from a horizontal plate is deduced from a dimensional analysis of the governing equations. The results of such an analysis, which is described in detail in Appendix A, shows that the average Nusselt number for natural convection from a square plate is dependent on

\[ Nu = Nu(Ra, Pr, T_w/T_\infty) \]  

Note that the effects of variable properties result in the addition of a single dimensionless group, \( T_w/T_\infty \), and that the Boussinesq approximation is not necessary. Experience has shown that for values of \( T_w/T_\infty \) near unity, the form of Eq. (4.1) is valid for both laminar and turbulent natural convection.

Following the procedure used by Clausing [16], the form of the correlation in the natural convection limit is hypothesized to be

\[ Nu = g(Ra) \cdot f(T_w/T_\infty) \]  

An excellent correlation for the experimental data is obtained by using this hypothesis. The function \( f(T_w/T_\infty) \) accounts for the influences of \( T_w/T_\infty \) on the Nusselt number, and \( f(1) \) is defined to be one. Thus, the function \( g(Ra) \) is the constant property Rayleigh number function.

Performing a similar dimensional analysis for convection in the presence of a freestream velocity, we find the average Nusselt number to be a function of

\[ Nu = Nu(Ra, Pr, Ri, T_w/T_\infty, Tu) \]  

Where \( Tu \) is the turbulence intensity of the freestream. This is obviously not a unique set of parameters. Since the data taken were near the natural convection limit, a combination of the Rayleigh number and the Richardson number is used. The Richardson number represents the ratio of buoyant to inertial forces and is defined as \( Gr/Re^2 \). The limit as \( Ri \to \infty \) is the natural convection limit. If the regime near the forced convection limit were of interest, the heat transfer would best be represented by using the Reynolds number in place of the Rayleigh number.

Again hypothesizing that the Nusselt number can be represented as a product of functions of the relevant groups, we arrive at

\[ Nu = g(Ra) \cdot f(T_w/T_\infty) \cdot z(Ri) \]  

where the functions \( g \) and \( f \) are assumed to be the same functions as in the case of natural convection. Therefore, the function \( z(Ri) \) is a function of the Nusselt number in combined convection divided by the Nusselt number in natural convection. That is,
\[ z(Ri) = \frac{Nu_{\text{con}}}{Nu_{\text{nat}}} \]  

(4.5)

Thus, the limit of \( z(Ri) \) as \( Ri \rightarrow \infty \) is defined to be one. The function \( z(Ri) \) accounts for inertial influences in the heat transfer rate as \( Ri \) decreases. It was again found that reducing the data in this manner works well.

The influences of the Prandtl number and the turbulence intensity are not resolved in this study. Therefore, the results are applicable to fluids with a Prandtl number of approximately 0.7. The properties in all dimensionless groups are evaluated at the film temperature, \( T_f \equiv \frac{(T_w + T_\infty)}{2} \), unless otherwise indicated with the appropriate subscript. The characteristic length used in the definitions of \( Ra \), \( Re \), and \( Ri \) is the side length of the square plate, \( L \).
5. FLAT PLATE TEST MODEL

5.1 Thermal Design

In order to generate larger Rayleigh numbers in the limited space of the test section, the square plate test model is represented by an $L$ by $L/2$ plate where $L = 0.6$ m. That is, the vertical line of symmetry which bisects the square plate is replaced by a vertical adiabatic wall. Sketches of the top view and the side cross-sectional view are shown in Figs. 5.1.1 and 5.1.2, respectively. Symmetry tests were conducted on this half-square model to verify the validity of this assumption in the design. These natural convection symmetry tests involved bisecting the plate a second time by temporarily placing an additional vertical adiabatic wall perpendicular to the permanent vertical wall. The comparison of natural convection tests conducted with and without this additional wall shows that the extra viscous shear induced by a wall of symmetry has a negligible influence on the overall heat transfer from the plate.

![Fig. 5.1.1 Top View of Calorimeter Arrangement](image-url)
The $L$ by $L/2$ model is divided into four thermally isolated calorimeters of equal mass and area in order to resolve areas of different convective heat transfer rates. The arrangement of these calorimeters is shown in Fig. 5.1.1. Each calorimeter measures $0.3 \times 0.15 \times 7.9$ m and has a mass of 0.98 kg. The calorimeters are highly polished, 6061-T6 aluminum alloy plates. Each plate is secured to its base with two low thermal mass, high thermal resistance threaded nylon bolts. Urethane foam insulation is sprayed into the gaps between the plates to thermally isolate them from one another. The Biot number of each plate based on a typical heat transfer coefficient is less than $10^{-2}$; therefore, thermal gradients within the model are negligible and a lumped capacitance analysis can be performed. The calorimeters are individually heated by thin foil resistance heaters of negligible thermal mass which are mounted on the back sides. Directly underneath the model and the heaters is a 25 mm board of urethane insulation, a heated copper guard, and a second 25 mm board of insulation (See Fig. 5.1.2). The guard is carefully maintained at the same temperature as the calorimeters during a test to eliminate conductive heat flow in that direction.

The model is instrumented with eighteen 30-gauge copper-constantan, special accuracy thermocouples. The small diameter thermocouples were chosen for their rapid time response and to minimize conductive losses through the wire. Four thermocouples are inserted into holes drilled midway into each calorimeter. These holes are packed with a high thermal conductivity paste and bits of aluminum foil are wedged into the remaining spaces to ensure good thermal contact between the model and the sensor. The temperature distribution in the test section of the tunnel and the temperature of the copper guard are determined by eight identical thermocouples. All thermocouples are plugged into the isothermal wiring area described in Section 2.3.1 and are continuously scanned by the data acquisition equipment. Care must be exercised to electrically insulate the foil resistance heaters from the model, or else erratic thermocouple readings will result.
A cover made of extruded polystyrene insulation is designed for the top of the plate. This cover is lowered onto the plate prior to a test and it aids in heating the model in its cryogenic environment. It can be raised or lowered by means of cables leading out through the tunnel roof. The cover is removed when the test begins.

5.2 Dynamic Design

The entire model assembly is elevated 0.2 m from the floor of the tunnel, avoiding the higher levels of freestream turbulence which exist there. In order for the flow to uniformly pass over the plate during a combined convection test, an aluminum flow diverter is attached to the model assembly (See Figs. 5.1.1 and 5.1.2). This diverter is wedge-shaped at the leading edge of the plate, forcing the lower portion of the stream to pass underneath the model. The upper portion of the stream passes over the horizontal top of the diverter and allows a uniform boundary layer to develop over the plate.
6. EXPERIMENTAL PROCEDURE

6.1 Prerequisites to Testing

Various tasks must be completed before recording data for a typical cryogenic test. To begin a testing session, the facility must be cooled to its desired cryogenic temperature. Liquid nitrogen supplied from 160 liter dewars is fed into the heat exchanger, where it vaporizes. This process is described in greater detail in Section 2.2.2. The tunnel fans are run at a low speed to increase the efficiency of the heat exchanger and to minimize thermal stratification in the tunnel. A settling time of two to four hours is required for the mass of the facility to reach thermal equilibrium with the cold nitrogen gas. Once the desired ambient temperature is achieved, small amounts of nitrogen are occasionally injected into the tunnel to prevent the temperature from rising.

While the tunnel is cooling, the insulated cover is lowered onto the model and the copper guard and calorimeters are heated to their desired initial temperature. Recall that the guard, when heated to the same temperature as the plates, reduces conductive heat flow from the calorimeters. A heating time of approximately 45 minutes is required for the insulation between the model and guard to become isothermal. This time is determined by modeling the board of insulation as an infinite slab and performing a transient conduction analysis of the problem.

The temperatures of each calorimeter and the guard are graphically displayed as a function of time on the screen of the microcomputer. The experimentalist assumes the role of a “human thermostat” and uses the power supply controls and the heater switches to bring the calorimeters to their initial temperature. When the calorimeters are within 0.1 K of their desired temperature, the heaters are turned off and the flow, if desired, is initiated. The insulating lid is then raised from the model and data recording begins. A typical test, which generates a single data point, lasts approximately four minutes. During this time, the model cools approximately 10 K. A test schedule is designed such that each successive test has a lower model temperature, thus minimizing the heating time required.

6.2 Calculation of the Heat Transfer Coefficient

After the data files have been uploaded via telephone link to the mainframe CYBER 175 computer, they are further processed by the interactive Fortran V programs AMC30P, REPLAY, and HPLAY. These programs were written by A.M. Clausing specifically for data collected from the renovated facility. Their source codes are listed in Appendix C. A brief description of each of these programs follows.

The temperature of each thermocouple sensor is represented as a discrete function of time. Occasionally, the datalogger misreads the temperature signals, and a “glitch” results in the recorded data. AIC30P is a data smoothing program that processes these raw temperature data to rid each file of contaminated data points. AMC30P then generates a corrected, smoothed data file which the program REPLAY uses.
REPLAY is a program that reads the data files which were generated by the preprocessing program and calculates arrays of averaged, dimensionless temperatures for the various user defined ensembles. The dimensionless temperature is defined as

\[
T^* = \frac{(T - T_{\infty})}{(T_0 - T_{\infty})}
\]  

(6.2.1)

where \( T_0 \) is the initial temperature of the calorimeters. The definitions of the ensembles and the mass of each calorimeter are read by REPLAY from a parameter file. REPLAY generates a file containing dimensionless temperature as a discrete function of time which the program HPLAY uses.

HPLAY deduces the amount of convective heat transfer from each calorimeter by determining the rate of change of its internal energy. That is, the convective heat transfer coefficient is determined from the transient energy balance

\[
h = \frac{-\left(\frac{mc_p}{A} \frac{dT_w}{dt}\right) - \alpha \sigma \left(T_w^4 - T^4_{\infty}\right)}{T_w - T_{\infty}}
\]  

(6.2.2)

where \( m \) is the mass of the calorimeter, \( c_p \) its specific heat, \( A \) is the heat transfer area, \( \varepsilon \) is the emissivity for polished aluminum, and \( \sigma \) is the Stefan-Boltzmann constant. The temperature derivative with respect to time is approximated by a central finite difference quotient. The heat transfer surface is assumed to be gray, and the surroundings are assumed to be isothermal and black. The variations of both \( c_p \) and \( \varepsilon \) with temperature are taken into account.

Figures 6.2.1 and 6.2.2 show typical results for a 98 K, 150 s natural convection test for a Rayleigh number of \( 5.94 \times 10^{10} \). Figures 6.2.3 and 6.2.4 show similar results for a typical combined convection test. The symmetry and smoothness of the temperature versus time and the \( h \) versus time curves clearly attest to the accuracy of the experiment.
Fig. 6.2.1 $T^*$ vs. Time for a Typical Natural Convection Test

$U_\infty = 0.0$ m/s  
$T_\infty = 98.2$ K  
$T_w/T_\infty = 2.00$  
$Ra_f = 5.94 \times 10^{10}$  
$Nu_f = 650.2$

Fig. 6.2.2 $h$ vs. Time for a Typical Natural Convection Test

$U_\infty = 0.0$ m/s  
$T_\infty = 98.2$ K  
$T_w/T_\infty = 2.00$  
$Ra_f = 5.94 \times 10^{10}$  
$Nu_f = 650.2$
Fig. 6.2.3 $T^*$ vs. Time for a Typical Combined Convection Test

\[ U_\infty = 0.655 \text{ m/s} \]
\[ T_\infty = 106.3 \text{ K} \]
\[ T_w/T_\infty = 2.19 \]
\[ R_a_f = 3.78 \times 10^{10} \]
\[ R_e_f = 7.05 \times 10^4 \]
\[ R_l_f = 10.37 \]
\[ N_u_f = 582.0 \]

Fig. 6.2.4 $h$ vs. Time for a Typical Combined Convection Test

\[ U_\infty = 0.655 \text{ m/s} \]
\[ T_\infty = 106.3 \text{ K} \]
\[ T_w/T_\infty = 2.19 \]
\[ R_a_f = 3.78 \times 10^{10} \]
\[ R_e_f = 7.05 \times 10^4 \]
\[ R_l_f = 10.37 \]
\[ N_u_f = 582.0 \]
7. EXPERIMENTAL RESULTS

7.1 Natural Convection Results

The influence of the Rayleigh number on the Nusselt number for the natural convection data is illustrated in Fig. 7.1.1. A total of 56 data points were recorded. An examination of these results once the variable property effects have been removed shows that

\[ Nu = Ra^{1/3}, \quad 2 \times 10^8 < Ra < 2 \times 10^{11} \] (7.1.1)

over the specified Rayleigh number range. This particular Rayleigh number dependency is typical of turbulent natural convection data, as can be seen from the previous works described in the literature survey. The one-third exponent of the Rayleigh number accurately represents the heat transfer rates throughout the entire range of Rayleigh numbers investigated. The absence of a transition in the convective rates indicates that all data lie in the turbulent, natural convection regime.

It was assumed that the variable property influence could be accurately represented by a second-degree polynomial in \( T_w/T_\infty \). A least-squares, second-degree fit of the experimental data using the constraint \( f(1) = 1 \) yields

\[ Nu_c = 0.140 \, Ra^{1/3} \left[ a_1 + a_2(T_w/T_\infty) + a_3(T_w/T_\infty)^2 \right] \] (7.1.2)

where the constants \( a_i \) for three different reference temperatures are given in Table 7.1.1. The three variable property correlations and the corresponding experimental data are graphically illustrated in Fig. 7.1.2.

<table>
<thead>
<tr>
<th>Ref. Temp.</th>
<th>( a_1 )</th>
<th>( a_2 )</th>
<th>( a_3 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wall</td>
<td>0.433</td>
<td>0.626</td>
<td>-0.0581</td>
</tr>
<tr>
<td>Film</td>
<td>0.823</td>
<td>0.179</td>
<td>-0.00113</td>
</tr>
<tr>
<td>Ambient</td>
<td>1.212</td>
<td>-0.254</td>
<td>0.0405</td>
</tr>
</tbody>
</table>

Table 7.1.1. Constants of \( f(T_w/T_\infty) \) (See Eq. (7.1.2))
Fig. 7.1.1 Influence of the Rayleigh Number on the Nusselt Number

Fig. 7.1.2 Natural Convection Variable Property Correlation, $f(T_w/T_\infty)$
The coefficient of 0.140 in the constant property correlation, \( g(Ra) \), agrees with the coefficient reported by Fishenden and Saunders [10], who used the same turbulent Rayleigh number exponent. They used elevated pressures in order to obtain large Rayleigh numbers. The accuracy of the constant property correlation can be seen in Fig. 7.1.3, where the influence of variable properties is removed.

The choice of reference temperature has, by definition, no influence on the constant property correlation, \( g(Ra) \). The choice of reference temperature does, however, have a significant influence on \( f(T_w/T_\infty) \), which can be seen in Fig. 7.1.2. The function \( f \) is greater than unity if the wall or the film temperature is used as the reference temperature, but is it less than unity if the properties are based on the ambient temperature. This result contrasts with the correlations reported by Clausing [16] for natural convection from an isothermal vertical surface. For that case, the function \( f \) is greater than unity regardless of the reference temperature used. The variable property influence is also significantly greater for the case of the vertical plate.

\[ f(T_w/T_\infty) = 0.824 + 0.176 \left( T_w/T_\infty \right) \quad (7.1.3) \]
The linear nature of the function $f$ suggests the use of the reference temperature method of compensating for variable property effects. In this method, a reference temperature at which all properties are evaluated is chosen such that the function $g(Ra)$ achieves the maximum degree of correlation. If the reference temperature is chosen to be

$$T_r = T_w - 0.83(T_w - T_\infty) \quad (7.1.4)$$

then the value of $f(T_w/T_\infty)$ is found to be unity and no property corrections are necessary for the entire range of temperature ratios investigated. Although the reference temperature method is widely used, it should be stressed that for other geometries, particularly in the case of vertical plates, the technique is not successful.

The agreement between Correlation (7.1.2) and the experimental data is illustrated in Figs. 7.1.4 and 7.1.5. Figure 7.1.4 shows the degree of correlation obtained with the constant property correlation alone. This figure is a plot of $Nu_f$ divided by the function $g(Ra_f)$ versus $Ra_f$. The inability of the constant property correlation to predict the experimental results is evident. Deviations as large as 40 percent are present. On the other hand, Fig. 7.1.5 shows $Nu_f$ divided further by the variable property function, $f(T_w/T_\infty)$. A linear scale is used to show the excellent agreement between the 56 data points and Correlation (7.1.2). The maximum deviation of any data point is only 3.3 percent!

![Fig. 7.1.4 Degree of Correlation, Showing Influence of Only Ra](image-url)
7.2 Combined Convection Results

The Nusselt numbers for the combined convection data are shown in Fig. 7.2.1 as a function of the Reynolds number. The deviation from the analytical, pure forced convection solution is apparent. Even when the Nusselt number is corrected by the variable property function as shown in Fig. 7.2.2, the deviations are still large. Perhaps the buoyant effects which are not accounted for by the Reynolds number alone are influencing the rate of the convective loss. A trend in the data is obtained by dividing the Nusselt number in combined convection by Correlation (7.1.2), the natural convection correlation, and plotting the results against the Richardson number. The excellent degree of correlation is illustrated in Fig. 7.2.3. The function $f_f (T_w/T_\infty)$ is hypothesized to account for variable property effects in combined convection as well as in natural convection.

Recall that these results may be written mathematically as

$$Nu_f = g (Ra_f) \cdot f_f (T_w/T_\infty) \cdot z (Ri_f)$$  \hspace{1cm} (7.2.1)

Performing a third-order, least-squares fit in $\log(Ri)$ with the constraint $z(\infty) = 1$ yields
for the range $0.02 < Ri_f < 100$. In order for the freestream velocity to increase the convective heat transfer rate by more than one percent, the Richardson number must be less than 87. The increase is more than five percent if $Ri_f < 33$. 

Fig. 7.2.1 Influence of the Reynolds Number on the Nusselt Number
Fig. 7.2.2 Influence of the Reynolds Number on $\text{Nu}/f$

\[
\text{Nu}/f \propto \text{Re}_{f}^{1/2}
\]

Fig. 7.2.3 Comparison of Combined Convection Data and Proposed Correlation
8. RESULTS AND CONCLUSIONS

The following conclusions are drawn from the results of this investigation.

1) The variable property influence can be accounted for with a function of only $T_w/T_\infty$ over the Rayleigh number range $2 \times 10^8 < Ra < 2 \times 10^{11}$. This differs from the natural convection results for the vertical plate described in Ref. [16], where $f$ was found to be a function of both $Ra$ and $T_w/T_\infty$. For the vertical plate, it was found that $f(Ra, T_w/T_\infty) = 1$ until the transitional Rayleigh number is reached. $f(Ra, T_w/T_\infty)$ is then greater than one in the turbulent natural convection regime and reached values as high as 2 at $T_w/T_\infty = 2$. The dependency of $f$ on $T_w/T_\infty$ alone indicates that all data taken in the present study lie in the turbulent domain. This is substantiated by the turbulent natural convection data reported in Refs. [5], [6], [7], and [10].

2) The Nusselt number in the turbulent natural convection regime is strongly affected by property variations. Correlations proposed in the literature, such as the Fishenden and Saunders correlation (See Eq. 3.5), are acceptable only if $T_w/T_\infty$ is near unity. For example, the Nusselt number evaluated at $T_w/T_\infty = 3$ is 35 percent greater than the value predicted by the Fishenden and Saunders correlation.

3) Due to the linear nature of $f(T_w/T_\infty)$, a reference temperature can be selected which forces this function to be unity for the entire range of temperature ratios investigated. This reference temperature is $T_r = T_w - 0.83(T_w - T_\infty)$.

4) The combined convection data indicate that small to moderate drafts, such as doors opening or people moving about, can have an effect on the rate of heat transfer in “free” convection. Further evidence of combined convection influences are the great efforts taken by some investigators of natural convection phenomena to eliminate such influences in their experiments (See discussion in [16]).
REFERENCES

A.1 Governing Equations

Consider the flowfield and the temperature distribution for the case of the isothermal horizontal flat plate illustrated in Fig. A.1.1. If one makes the assumptions of

1) Two-dimensional flow,
2) A compressible fluid,
3) A perfect gas,
4) Stoke’s hypothesis,
5) Steady flow,
6) Laminar flow,
7) A radiatively non-participating gas,
8) Negligible work done by compression,
9) Negligible work done by viscous dissipation,
10) An isothermal surface and surroundings, and
11) Properties being power functions of the ratio \( T/T_r \),

then the Navier-Stokes equations can be written in the following simplified form:

Continuity:

\[
\frac{\partial}{\partial x} (\rho u) + \frac{\partial}{\partial y} (\rho v) = 0
\]  
(A.1.1)

x-Momentum:

\[
\rho \left[ u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right] = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[ \mu \left( 2 \frac{\partial u}{\partial x} - \frac{2}{3} \nabla \cdot \vec{u} \right) \right] + \frac{\partial}{\partial y} \left[ \mu \left( \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right]
\]  
(A.1.2)

y-Momentum:

\[
\rho \left[ u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right] = -\rho g - \frac{\partial p}{\partial y} + \frac{\partial}{\partial y} \left[ \mu \left( 2 \frac{\partial v}{\partial y} - \frac{2}{3} \nabla \cdot \vec{u} \right) \right] + \frac{\partial}{\partial x} \left[ \mu \left( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial x} \right) \right]
\]  
(A.1.3)

Energy:

\[
\rho c_p \left[ u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right] = \frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k \frac{\partial T}{\partial y} \right)
\]  
(A.1.4)
Equation of State:

\[ p = \rho RT \]  \quad (A.1.5)

Properties:

\[ \mu = \mu(T), \quad k = k(T), \quad C_p = C_p(T) \]  \quad (A.1.6)

Fig. A.1.1 Sketch of a Horizontal Plate in Convective Flow

These equations are to be nondimensionalized in order to determine the relevant dimensionless groups.

A.2 Natural Convection

Suitable choices can be made for the reference length and temperature. Specifically, we define

\[ x^* \equiv x/L \]  \quad (A.2.1)

\[ y^* \equiv y/\delta_T \]  \quad (A.2.2)

\[ T^* \equiv (T - T_w)/(T_w - T_\infty) \]  \quad (A.2.3)

where the asterisk denotes a dimensionless quantity.

Since \( U_\infty \) is zero in the case of natural convection, we lack an obvious choice for a reference velocity. We must look closer to find a suitable characteristic buoyant velocity to use as a reference. First,
we suppose the plate in Fig. A.1.1 is infinitesimally inclined to produce momentum and thermal boundary layers in the positive \( x \) direction. This precludes the possibility of plume flow rising near \( x = L/2 \). Inside the thermal boundary layer, we suppose that the variables \( x \) and \( y \) are of the same order as the dimensions \( \delta_T \) and \( L \), respectively. Using this assumption and for the moment considering constant properties, we have from the continuity and energy equations the characteristic buoyant velocities

\[
\begin{align*}
  u &\approx a/\delta_T \quad \text{(A.2.4)} \\
  v &\approx aL/\delta_T^2 \quad \text{(A.2.5)}
\end{align*}
\]

Thus, we define

\[
\begin{align*}
  u^* &\equiv u\delta_T/\alpha \\
  v^* &\equiv v\delta_T^2/\alpha L
\end{align*}
\]

The reference temperature mentioned in Assumption (11) is within the limits of the thermal boundary layer and is assumed to be of the form

\[
T_r = T_w - C (T_w - T_\infty)
\]

(A.2.8)

Note that if \( C = 0.5 \), the film temperature becomes the reference. Thus, it can be seen that the temperature dependent properties are general functions of \( T^* \) and the temperature ratio, \( T_w/T_\infty \). For example, the thermal conductivity may be written as

\[
k = a \left[ \frac{(T_w/T_\infty - 1)T^* + 1}{T_w/T_\infty (1 - C) + C} \right]^b
\]

(A.2.9)

where \( a \) and \( b \) are constants.

If we further assume a relatively thin boundary layer, we arrive at the relation \( \rho g + \partial\rho/\partial y = (\rho - \rho_\infty)g \). Defining the volume coefficient of expansion,

\[
\beta = -\frac{1}{\rho} \left( \frac{\partial \rho}{\partial T} \right)_p
\]

(A.2.10)

and expanding the density difference in a Taylor series, we ultimately arrive at the relation

\[
\rho - \rho_\infty \approx -\rho_\infty \beta (T - T_\infty) - \rho_\infty \beta^2 (T - T_\infty)^2/2! - \ldots
\]

(A.2.11)

Rewriting Equations A.1.1 through A.1.4 with the above variables and expressions, it can be seen that the flowfield solution, \( u \) and \( v \), and the temperature distribution, \( T^* \), are functions of the spatial coordinates and the dimensionless groups \( Ra_L, Pr \), and \( T_w/T_\infty \). Observing that the Nusselt number is a function of the temperature gradient at the wall, we have

\[
Nu = Nu (Ra_L, Pr, T_w/T_\infty)
\]

(A.2.12)
Note that the Boussinesq approximation is not necessary.

We can hypothesize that the Nusselt number is dependent on the product of a constant property Rayleigh number function and a variable property temperature ratio function. That is,

$$\dot{N}u = g(Ra_L) \cdot f(T_w/T_\infty)$$ (A.2.13)

Excellent agreement in the experimental data is obtained by using this hypothesis. Note that the influence of the Prandtl number is not resolved in this study.

### A.3 Combined Convection

In the case of combined convection, the obvious choice for a reference velocity is that of the freestream, $U_\infty$. This is appropriate only if the edge of the thermal boundary layer “sees” the freestream velocity and not buoyant flow. This occurs when the Prandtl number is near unity and the momentum and thermal boundary layers coincide. Since only gases with Prandtl numbers of approximately 0.7 are considered, the choice of $U_\infty$ as a reference velocity is valid. We may define

$$u^* \equiv u/U_\infty$$ (A.3.1)

$$v^* \equiv v/U_\infty$$ (A.3.2)

Keeping the same definitions for dimensionless coordinates and temperature and following the same procedure as before, we arrive at

$$\dot{N}u = \dot{N}u (Re_L, Ri_L, Pe_L, T_w/T_\infty)$$ (A.3.3)

Since the data taken at various freestream velocities did not correlate well with the Reynolds number alone, it was assumed that the parameters involved in natural convection were implicated. Thus, we hypothesize

$$\dot{N}u = g(Ra_L) \cdot f(T_w/T_\infty) \cdot z(Ri_L)$$ (A.3.4)

Where the functions $g$ and $f$ are the same as in the case of natural convection. Therefore, the function $z(Ri_L)$ is a function of the Nusselt number in combined convection divided by the Nusselt number in natural convection. That is,

$$z(Ri_L) = \dot{N}u_{com} / \dot{N}u_{nat}$$ (A.3.5)

The function $z$, in effect, accounts for inertial forces.

Reducing the data in the manner of Eq.(A.3.4) was found to work well.
B.1 Thermophysical Properties of Air, Nitrogen, and Al 6061-T6

*Power Law Relations for Air and Nitrogen:*

Each thermophysical property, \( y \), of the gases is represented by a constant, \( a \), times the absolute temperature, \( T \), in degrees Kelvin raised to the power, \( b \). Specifically, in SI units

\[
y = a_1(T/K)^b
\]

and in inch-pound units

\[
y = a_2(T/R)^b
\]

where the units of \( a_1 \) and \( a_2 \) for the respective properties and the relevant conversion factor, \( c \), are given in Table B.1.1. It is easily shown that

\[
a_2 = (a_1/c)(5/9)^b
\]

Five fundamental properties are considered: density, \( \rho \); dynamic viscosity, \( \mu \); thermal conductivity, \( k \); specific heat at constant pressure, \( c_p \); and the Prandtl number, \( Pr \). The constants \( a_1 \) and \( b \) were determined from published data by A.M. Clausing [17], who used a least-squares fitting procedure. The respective values of \( a_2 \) are then calculated from Eq. (B.1.3). The constants \( a_1 \), \( a_2 \), and \( b \), the applicable temperature ranges, and the maximum percent errors are given in Tables B.1.2 and B.1.3.

*Relations for Aluminum Alloy 6061-T6:*

A least-squares fitting procedure was used by A.M. Clausing to relate the density, emissivity, thermal conductivity, and specific heat of aluminum alloy 6061-T6 to various functions of the absolute temperature, \( T \), in degrees Kelvin. The following relations are given

\[
\rho = 2704 \exp[1.51 \times 10^{-2} - 3.79 \times 10^{-5}T - 4.59 \times 10^{-8}T^2]; \quad (273K < T < 483K)
\]

\[
\varepsilon = 0.04 + 6 \times 10^{-5}T; \quad (0 < T < 850K)
\]

\[
k = 86.6 + 0.261T; \quad (100 < T < 400K)
\]

\[
c_p = 689.6 + 0.6741T; \quad (273 < T < 500K)
\]

\[
c_p = -437 + 14.03T - 0.0592T^2 + 1.157 \times 10^{-4}T^3 - 8.221 \times 10^{-8}T^4; \quad (70 < T < 273K)
\]

where density, conductivity, and specific heat are in units of \( \text{kg/m}^3 \), \( \text{W/m-K} \), and \( \text{J/kg-K} \), respectively. The least-squares representations of the properties and selected values of the published data are plotted in Figs. B.1.1 through B.1.10.
Table B.1.1. Nomenclature, Units, and Conversion Factors (See Ref. [17])

<table>
<thead>
<tr>
<th>Property</th>
<th>Symbol</th>
<th>Units of a (SI)</th>
<th>Units of a (Inch-Pound)</th>
<th>c (Inch-Pound)=(SI)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>( \rho )</td>
<td>kg/m(^3)</td>
<td>lbm/ft(^3)</td>
<td>16.02</td>
</tr>
<tr>
<td>Viscosity</td>
<td>( \mu )</td>
<td>kg/m-s</td>
<td>lbm/ft-s</td>
<td>1.488</td>
</tr>
<tr>
<td>Conductivity</td>
<td>( k )</td>
<td>W/m-K</td>
<td>Btu/hr-ft-F</td>
<td>1.731</td>
</tr>
<tr>
<td>Specific heat</td>
<td>( c_p )</td>
<td>J/kg-K</td>
<td>Btu/lbm-F</td>
<td>4187.</td>
</tr>
<tr>
<td>Grashof No./(( \Delta T ) L(^3))</td>
<td>( \frac{g\beta}{\nu^2} )</td>
<td>m(^3)-K(^{-1})</td>
<td>ft(^3)-F(^{-1})</td>
<td>63.57</td>
</tr>
<tr>
<td>Prandtl No.</td>
<td>( Pr )</td>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
</tbody>
</table>

Table B.1.2. Coefficients for Power Curve Fits: Dry Air (See Ref. [17])

<table>
<thead>
<tr>
<th>Property</th>
<th>Temp. Range</th>
<th>( a_1 )</th>
<th>( a_2 )</th>
<th>( b )</th>
<th>Max. % Diff.</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \rho )</td>
<td>150K &lt; ( T ) &lt; 400K</td>
<td>364.1</td>
<td>41.04</td>
<td>-1.005</td>
<td>0.3</td>
</tr>
<tr>
<td>( \mu )</td>
<td>&quot;</td>
<td>1.764×10(^{-7})</td>
<td>7.349×10(^{-8})</td>
<td>0.814</td>
<td>1.4</td>
</tr>
<tr>
<td>( k )</td>
<td>&quot;</td>
<td>1.423×10(^{-4})</td>
<td>4.803×10(^{-5})</td>
<td>0.9138</td>
<td>0.9</td>
</tr>
<tr>
<td>( c_p )</td>
<td>&quot;</td>
<td>990.8</td>
<td>0.2362</td>
<td>0.00316</td>
<td>0.4</td>
</tr>
<tr>
<td>( \frac{g\beta}{\nu^2} )</td>
<td>&quot;</td>
<td>4.178×10(^{19})</td>
<td>1.004×10(^{19})</td>
<td>-4.639</td>
<td>3.1</td>
</tr>
<tr>
<td>( Pr )</td>
<td>&quot;</td>
<td>1.23</td>
<td>1.302</td>
<td>-0.09685</td>
<td>0.4</td>
</tr>
<tr>
<td>( \rho )</td>
<td>400K &lt; ( T ) &lt; 2100K</td>
<td>350.6</td>
<td>39.38</td>
<td>-0.999</td>
<td>0.4</td>
</tr>
<tr>
<td>( \mu )</td>
<td>&quot;</td>
<td>4.914×10(^{-7})</td>
<td>2.263×10(^{-7})</td>
<td>0.6429</td>
<td>0.5</td>
</tr>
<tr>
<td>( k )</td>
<td>&quot;</td>
<td>2.494×10(^{-4})</td>
<td>8.92×10(^{-5})</td>
<td>0.8152</td>
<td>3.2</td>
</tr>
<tr>
<td>( c_p )</td>
<td>&quot;</td>
<td>299.4</td>
<td>0.06372</td>
<td>0.1962</td>
<td>2.8</td>
</tr>
<tr>
<td>( \frac{g\beta}{\nu^2} )</td>
<td>&quot;</td>
<td>4.985×10(^{18})</td>
<td>9.726×10(^{17})</td>
<td>-4.284</td>
<td>1.5</td>
</tr>
<tr>
<td>( Pr )</td>
<td>&quot;</td>
<td>0.59</td>
<td>0.5815</td>
<td>0.0239</td>
<td>1.1</td>
</tr>
</tbody>
</table>

Table B.1.3. Coefficients for Power Curve Fits: Nitrogen (See Ref. [17])

<table>
<thead>
<tr>
<th>Property</th>
<th>Temp. Range</th>
<th>( a_1 )</th>
<th>( a_2 )</th>
<th>( b )</th>
<th>Max. % Diff.</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \rho )</td>
<td>83K &lt; ( T ) &lt; 160K</td>
<td>432.4</td>
<td>49.92</td>
<td>-1.046</td>
<td>0.5</td>
</tr>
<tr>
<td>( \mu )</td>
<td>&quot;</td>
<td>9.1×10(^{8})</td>
<td>3.52×10(^{8})</td>
<td>0.938</td>
<td>0.3</td>
</tr>
<tr>
<td>( k )</td>
<td>&quot;</td>
<td>1.239×10(^{-4})</td>
<td>4.103×10(^{-5})</td>
<td>0.9466</td>
<td>0.3</td>
</tr>
<tr>
<td>( c_p )</td>
<td>&quot;</td>
<td>155.3</td>
<td>0.3885</td>
<td>-0.079</td>
<td>0.8</td>
</tr>
<tr>
<td>( \frac{g\beta}{\nu^2} )</td>
<td>&quot;</td>
<td>4.379×10(^{20})</td>
<td>1.382×10(^{20})</td>
<td>-5.102</td>
<td>1.9</td>
</tr>
<tr>
<td>( Pr )</td>
<td>&quot;</td>
<td>1.137</td>
<td>1.179</td>
<td>-0.0872</td>
<td>0.9</td>
</tr>
<tr>
<td>( \rho )</td>
<td>160K &lt; ( T ) &lt; 450K</td>
<td>351.6</td>
<td>39.63</td>
<td>-1.005</td>
<td>0.2</td>
</tr>
<tr>
<td>( \mu )</td>
<td>&quot;</td>
<td>1.8×10(^{-7})</td>
<td>7.53×10(^{-8})</td>
<td>0.8058</td>
<td>1.4</td>
</tr>
<tr>
<td>( k )</td>
<td>&quot;</td>
<td>2.21×10(^{-4})</td>
<td>7.814×10(^{-5})</td>
<td>0.8345</td>
<td>1.4</td>
</tr>
<tr>
<td>( c_p )</td>
<td>&quot;</td>
<td>1031.</td>
<td>0.246</td>
<td>0.00239</td>
<td>0.4</td>
</tr>
<tr>
<td>( \frac{g\beta}{\nu^2} )</td>
<td>&quot;</td>
<td>4.08×10(^{19})</td>
<td>9.79×10(^{18})</td>
<td>-4.636</td>
<td>3.5</td>
</tr>
<tr>
<td>( Pr )</td>
<td>&quot;</td>
<td>0.841</td>
<td>0.854</td>
<td>-0.02652</td>
<td>0.5</td>
</tr>
</tbody>
</table>
Figure B.1.1  Density of Air and Nitrogen vs. Temperature

Figure B.1.2  Viscosity of Air and Nitrogen vs. Temperature
Figure B.1.3 Thermal Conductivity of Air and Nitrogen vs. Temperature

Figure B.1.4 Specific Heat of Air and Nitrogen vs. Temperature
Figure B.1.5 Grashof Number Base for Air and Nitrogen vs. Temperature

Figure B.1.5 Prandtl Number for Air and Nitrogen vs. Temperature
Figure B.1.7  Aluminum Density vs. Temperature

Figure B.1.8  Aluminum Emissivity vs. Temperature
Figure B.1.9 Aluminum Thermal Conductivity vs. Temperature

Figure B.1.10 Aluminum Specific Heat vs. Temperature
B.2 Gas Properties Subroutine

```
SUBROUTINE GASPT(NGAS,T,RHO,XMU,XK,CF,GRB,FR,IER)

PROGRAM NAME: GASPT

INPUT:
NGAS  = 1 IS AIR;  NGAS=2 IS NITROGEN
T      ---- ABSOLUTE TEMP. (K) OR NEGATIVE OF ABSOLUTE TEMP. (R)

OUTPUT:
RHO ---- DENSITY (KG/M3) OR (LB/FT3)
XMU ---- VISCOSITY (KG.M/SEC) OR (LB/FT-SEC)
XK ---- THERMAL CONDUCTIVITY (W/M-K) OR (BTU/HR-FT-R)
CF ---- SPECIFIC HEAT (J/KG-K) OR (BTU/LB-FT-3)
GRB ---- D*beta/Xmu*2 (1/(M3-K)) OR (1/(FT3-K))
FR ---- PRANDTL NUMBER (DIMENSIONLESS)
IER ---- ERROR PARAMETER

INFORMATIVE ERRORS:
IER=1 --- GAS NUMBER DOES NOT EXIST, GAS IS ASSUMED TO BE AIR.
IER=2 --- TEMPERATURE OUT OF RANGE OF PROPERTY SUBROUTINE

RESTRICTIONS:
NGAS ---- MUST BE 1(AIR) OR 2(NITROGEN)
T       ---- T MUST BE BETWEEN 1500K AND 21000K FOR AIR, AND BETWEEN
            83K AND 450K FOR NITROGEN. RANGES ARE SPECIFIED WITH ARRAY R.

DIMENSION A(15,2),B(15,2),R(3,3)
DATA A/364.1,1764E-6,1423E-3,990.8,.178E20,1.23,
      2 130.6,4914E-3,2394E-3,499E19,.893E20,2.32,
      3 432.4,91E-8,1.239E-3,1503,4.379E20,1.137,
      4 351.6,18E-6,221E-3,1031,.408E20,.841,380,
      5 1.005,814,.9138,.00164,-.6529,-.09455,
      6 1.199,.1429,.8152,1.632,-.4284,.03983,0,
      7 1.46,938,.9466,.079,.6102,.06772,
      8 1.005,.8058,.8345,.0039-.4.636,-.02652,31.0,
      9 .815,1,.2496,.8345,.0039,.6366,.02652,31.0/
DATA R/150.,400.,2100.,83.,160.,450./
IER=0
IF((NGAS,GT,0).AND.(NGAS,LT,3)) GO TO 1
IER=1
NGAS=1
1
IF(T,LE,0.9)
   T=P/T
IF(T,LTE,0.9)
   IF((T,GT,R(1,NGAS)).OR.(T,GT,R(3,NGAS))) IER=2
IF(T,GT,R(1,NGAS))=7
IER=7
RHO=A(I,NGAS)*TF*B(I,NGAS)
XMU=A(I,1,NGAS)*TF*B(I,1,NGAS)
XK=A(I,2,NGAS)*TF*B(I,2,NGAS)
CF=A(I,3,NGAS)*TF*B(I,3,NGAS)
GRB=A(I,4,NGAS)*TF*B(I,4,NGAS)
FR=A(I,5,NGAS)*TF*B(I,5,NGAS)
IER=7
RETURN
RHO=RHO/16.0
XMU=XMU/1.488
XK=XMU/1.731
CF=CF/4187.
GRB=GRB/63.57
RETURN
```

FNM
B.3 Aluminum Properties Subroutine

SUBROUTINE ALPT(T, CP, E, XK, RHO, IER)
C GIVEN THE ABSOLUTE TEMPERATURE T[K], THIS SUBROUTINE
C DETERMINES THE THERMAL PROPERTIES OF ALUMINUM ALLOY 6061-T6.
C CP---SPECIFIC HEAT [J/KG-K]; 70<T<500K
C RHO---DENSITY [KG/M3]; 273<T<483K
C E---HEMISPHERICAL TOTAL EMISSIVITY [-]: 0<T<850K
C XK---THERMAL CONDUCTIVITY [W/M-K]; 100<T<400K
C IER--ERROR PARAMETER: IER=1--TEMPERATURE IS OUT OF RANGE
DATA RHOZ, RZ, R1, R2, EZ, E1, XKZ, XK1, CPZ, CP1, AZ, A1, A2, A3, A4, A5/
& 2704.,-1.505E-2,3.79E-5,4.59E-8,.046E-5,86.6,.261,689.6,
& .6741,-437.,14.03,.05920033,.11568388E-3,.82212796E-7,.0/
C CHECK IF TEMPERATURE IS IN PROPER RANGE -- 70<T<500.
IF((T.LT.70.).OR.(T.GT.500.)) THEN
  IER=1
  RETURN
ENDIF
IER=0
C CALCULATE PROPERTIES
  RHO=RHOZ*EXP(-RZ-R1*T-R2*T**2)
  E=EZ+E1*T
  XK=XKZ+XK1*T
  IF(T.GT.273) THEN
    CP=CPZ+CP1*T
  ELSE
  ENDIF
  RETURN
END
APPENDIX C

C.1 Data Smoothing Program

```
C PROGRAM AMC30P; VERSION: 17 DEC. 1985
C PREPROCESSING OF DATA
C PROGRAMMED BY: A. M. CLAUSING
C
PARAMETER (III=60, NNN=400)
DIMENSION T(1NNN), TIME(1NNN), IDT(III), WK(2*NNN), NIT(III)
CHARACTER CDATE*12, CTIME*9, INFILE*7, OUTFILE*7, PARM*7, PINFILE*7,
& SMOOTH*1, CYN*1, DATE*10
DATA DIS,SF,SCK,MAXIT/1..20..04.45/, NIT/III*.0/

C READ NAME OF FILE FOR STORAGE OF DIFFERENCE TABLES
C
2 PRINT *, 'TYPE NAME OF FILE FOR STORAGE OF DIFFERENCE TABLES'
   READ III, OUTFILE
   OPEN(6, FILE=OUTFILE)
   PRINT *, 'IS DATA TO BE SMOOTHED? (Y/N)'
   READ III, SMOOTH
   IF(SMOOTH.EQ.'N') GOTO 1
   PRINT '(A)', 'TYPE THE FIRST AND LAST FLUKE CHANNEL',
   & 'NUMBERS OF THE DATA WHICH ARE TO BE SMOOTHED'
   READ *, ISF, IFF
   IF(IFF+1)
   IF=IFF+1
C START OF LOOP -- READ NAME OF DATA FILE
C
1 PRINT *, 'TYPE THE NAME OF THE INPUT DATA FILE'
   READ III, INFILE
   IF(INFILE.EQ.'END') GOTO 999
   CALL PF('GET', 0, INFILE)
   OPEN(5, FILE=INFILE)
   READ(5, 103) NC, CTIME, CDATE
   READ(5, 102) XMM, SS
   TIME=XMM*60.+SS
   N=1
   TIME(1)=.0
C READ AND COUNT ALL SCANS
3 READ(5, 104, END=99) (T(N, I), I=1, NC)
   N=N+1
   READ(5, 102) XMM, SS
   TIME(N)=XMM*60.+SS-TIMEZ
   IF(TIME(N).LT.0) TIME(N)=TIME(N)+3600.
   GOTO 3
99 NS=N-1
   DTIME=TIME(N)/NS
   CLOSE(5, STATUS='DELETE')
```
C SMOOTH DATA
   IF(SMOOTH.EQ.'N') GOTO 12
   SC=SCK/DTIME
   DO 11 I=IS,IF
      NIT(I)=MAXIT
      CALL ICSPRM(T(I,1),NS,DY,SC,NIT(I),WK,IER)
   11 CONTINUE
C WRITE CORRECTED DATA FILE IF (SMOOTH.EQ.'Y')
   PINFILE='P'/INFILE
   OPEN(8,FILE=PINFILE)
   WRITE(8,103)NC,CTIME,COATE,NS-3,DTIME,PINFILE,DATE()
   WRITE(8,105)TIME(1)
   DO 15 N=4,NS
      WRITE(8,107)(T(N,I),I=1,NC)
      WRITE(8,105)TIME(N+1)-TIME(1)
   15 CONTINUE
   CLOSE(8)
   CALL PF('REPLACE',0,PINFILE)
   CALL PF('CHANGE',PINFILE,PINFILE,'CT','PU','N','R')
C CALCULATE AND WRITE DIFFERENCE TABLES
   WRITE(6,110)DATE(),SMOOTH,SD,INFILE
   IF(SMOOTH.EQ.'Y')WRITE(6,112)SCK,MAXIT,IS-1,IF-1
   WRITE(6,109)(I,1=0,NC-1)
   DO 17 N=1,NS-1
      DO 19 I=1,NC
         IDT(I)=(T(N,I)-T(N+1,I))*SF
      19 CONTINUE
   WRITE(6,109)(IDT(I),I=1,NC)
   CONTINUE
   WRITE(6,109)(I,1=0,NC-1)
   WRITE(6,109)(NIT(I),I=1,NC)
   GOTO !
C END OF DATA FILE LOOP
C
102   FORMAT(7X,F2.0,1X,F2.0)
103   FORMAT(13,A9,A12,14,F:6.13X,A7,8X,A10)
104   FORMAT(15F5.2)
105   FORMAT(F6.0)
107   FORMAT(15F7.2)
109   FORMAT(1X,6512)
110   FORMAT('I',1X,A10,3X,'SMOOTHED: ',A,4X,'SF=',F3.0,13X,A7/)
111   FORMAT(A)
112   FORMAT(2X,'SCK=',F4.3,4X,'MAXIT=',13,3X,
      & 'IS-1=',12,3X,'IF-1=',12/)
113   FORMAT('DO YOU WISH TO PREPROCESS ANOTHER SERIES OF DATA'/
      & 'FILES (Y/N)? NOTE: MAKE SURE YOU SPECIFY A NEW FILE NAME'/
      & 'FOR STORAGE OF THE NEXT SET OF DIFFERENCE TABLES IF'/
      & 'YOU ANSWER WITH: Y')
999   CLOSE(6)
   CALL PF('REPLACE',0,OUTFILE)
C ASK IF ANOTHER SERIES OF DATA FILES ARE TO BE PREPROCESSED
   PRINT 113
C.2 Temperature Grouping and Averaging Program

```
READ III, CYM
IF(CYM.EQ.'Y') GOTO 2
STOP
END

PROGRAM REPLAY: VERSION: 17 DEC. 1985
C READES DATA FILES AND CALCULATES DIMENSIONAL AND DIMENSIONLESS
C TEMPERATURES OF DEFINED ENSEMBLES
C PROGRAMMED BY: A. M. CLAUSING
PROGRAM REPLAY
PARAMETER (II=50, NNN=400)
DIMENSION T(II, NNN), TIME(NNN), TE(20, 100), TEX(20, 100), R(20, 15)
& , JJ(15)
CHARACTER CDATE*12, CTIME*9, INFILE*7, OUTFILE*7, PFILE*7, ENFILE*7
& , CYN*1, DATE*10, INTEMP*6
REAL MASS(20)
C READ NAME OF PARAMETER FILE AND ITS CONTENTS
PRINT *, 'TYPE NAME OF PARAMETER FILE'
READ 105, PFILE
CALL PF('GET', 0, PFILE)
OPEN(5, FILE=PFILE)
READ (5, *) NA, NP, NG, IT
C CALCULATE VARIOUS CONSTANTS
NP=NA+1
NPF=NA+NP
NG=NPF+1
NGF=NPF+NG
NTOT=NGF+1
C READ PARAMETER FILE
C
DO 2 I=1, NGF
IF((I, LT, NP)) .OR. (I, GT, NPF)) THEN
READ(5, *) (I J(J, J), J=1, 15)
ELSE
READ(5, *) MASS(I-NA), (I J(J, J), J=1, 15)
ENDIF
2 CONTINUE
DO 4 I=1, NGF
DO 6 J=1, 15
IF (I J(J, J).EQ. 0) THEN
JJ(J)=J-1
GOTO 4
ENDIF
4 CONTINUE
CLOSE(5, STATUS='DELETE')
C END OF READING OF PARAMETERS
C
C READ USER SPECIFIED DATA FILE
PRINT *, 'TYPE NAME OF INPUT FILE'
READ 105, INTEMP
IF(INTEMP.EQ.'END') GOTO 999
INFILE='P'//INTEMP
CALL PF('GET', 0, INFILE)
OPEN(8, FILE=INFILE)
READ(8, 104) NC, CTIME, CDATE, NS, DTME
IF(NS+DTME.GT.696.) NS=696./DTME
DTME=DTME*IT
```
C READ DATA FILE
   DO 20 N=1,NS
   READ(8,*) TIME(N)
   READ(8,*) (T(I,N), I=1, NC)
   CONTINUE
   CLOSE(8, STATUS='DELETE')
C CALCULATE ENSEMBLE AVERAGE TEMPERATURES
C START OF MAJOR LOOP W.R.T. TIME
   N=1
   DO 40 N=1, NS, IT
   TIME(N)=TIME(N-I)
C CALCULATE AVERAGE TUNNEL AND CAVITY AMBIENT TEMPERATURES
   DO 22 I=1, NA
   TE(I,N)=T(I(J(I,J), NI)
   DO 24 J=2, JJ(I)
   24 TE(I,N)=TE(I,N)+T(I(J(J,I,J), NI)
   22 TE(I,N)=TE(I,N)/JJ(I)
C CALCULATE AVERAGE PLATE TEMPERATURES
   DO 26 J=1, NP1, NPF
   TE(I,N)=T(I(J(J,J), NI)
   DO 28 J=J1, JJ(J)
   28 TE(I,N)=TE(I,N)+T(I(J(J,J), NI)
   26 TE(I,N)=TE(I,N)/JJ(J)
C CALCULATE MASS-WEIGHTED TEMPERATURES OF DEFINED GROUPS OF CALORIMETERS
   G=NP1
   DO 30 G=NG1, NGF
   IG=G+1
   MASS(IG)=MASS(J(J(J(J, 1), 1), 1))
   TE(I,N)=TE(I(J(J(J(J, 1), 1), 1))+NA.N)*MASS(J(J(J(J, 1), 1), 1))
   DO 32 J=2, JJ(J)
   IG=IG+1
   MASS(IG)=MASS(IG)+MASS(J(J(J(J, 1), 1)
   32 TE(I,N)=TE(I,N)+TE(I(J(J(J(J, 1), 1)+NA.N)*MASS(J(J(J(J, 1), 1))
   30 TE(I,N)=TE(I,N)/MASS(IG)
   N=N+1
   CONTINUE
   N=N+1
C CALCULATE DIMENSIONLESS TEMPERATURES
   TINF=TE(I,N/4)
   TDENOM=TE(NGF,1)-TINF
   DO 50 N=1, NN
      DO 50 I=1, NFG
      TEX(I,N)=(TE(I,N)-TINF)/TDENOM
C WRITE RESULTS
   OUTFILE='E://INFILE(2:)
   OPEN(6, FILE=OUTFILE)
   WRITE(6,103) N1, CTIME, CDATE, N.N, DTIME, OUTFILE, DATE()
   DO 17 N=1, N1
      WRITE(6,109) TIME(N), (TEX(I,N), I=1, NFG)
   CONTINUE
      WRITE(6,106) (J(1,J=1, NA), I, J,J=1, NP), (K, K=1, NG)
   WRITE(6,107) TINF, TE(NGF, 1)
   CLOSE(6)
   CALL PF('REPLACE', 0, OUTFILE)
   CALL PF('CHANGE', OUTFILE, OUTFILE, 'CT', 'PU', 'M', 'R')
   GOTO 999
C ASK IF ANOTHER SERIES OF DATA FILES ARE TO BE PROCESSED
999 PRINT(2A1) 'DO YOU WISH TO PROCESS MORE DATA WITH A DIFFERENT'
      & '_PARAMETER FILE? (Y/N)' '
READ 105, CYN
IF(CYN.EQ.'Y') GOTO 5
102 FORMAT(7X,F2.0,1X,F2.0)
103 FORMAT(13,A9,A12,14,F7.3,13X,A7,8X,A10)
104 FORMAT(13,A9,A12,14,F16.12)
105 FORMAT(A)
106 FORMAT('//':TIME(S)',16I8)
107 FORMAT(3X,'AMBIENT TEMP.[K]='F6.1,4X,
      & 'INITIAL MODEL TEMP.[K]='F6.1)
109 FORMAT(F8.1,F8.4)
STOP
END
Sample Parameter File for Program REPLAY:

```
1 2 4 1 3
2 4 5 6 15*0
3 7 9 15*0
4 .972 9 10 11 12 15*0
5 .981 13 14 15 16 15*0
6 .979 17 18 19 20 15*0
7 .975 21 22 23 24 15*0
8 1 2 3 4 15*0
```

C.3 Heat Transfer Coefficient Calculating Program

```c
C PROGRAM HPLAY; VERSION: 18 DEC. 1985
C THIS PROGRAM CALCULATES HEAT TRANSFER COEFFICIENTS FOR
C EACH PLATE AND USER DEFINED GROUPS OF PLATES
C PROGRAMMED BY: A. M. CLAUSING
C
C DIMENSION TX(20,150),TIME(150),C(20),A(20),I,J(20,150)
C CHARACTER DATE*12,TIME*9,INFILE*7,OUTFILE*7,PFILE*7,EFFILE*7
C & CYN*1,DATE*10,INTEMP*6,DATAF*5
C REAL MASS(20),H(20,150),HT(20,150)
C HFUN(T1,T2,E)=5.67*10E*(T1**2+T2**2)*(T1+T2)

C READ NAME OF PARAMETER FILE AND GLOBAL PARAMETERS
C PRINT *,'TYPE NAME OF PARAMETER FILE'
C READ 105,PFILE
C CALL PF('G',0,PFILE)
C OPEN(5,FILE=PF)
C READ (5,*)(NA,NP,NG,NAVES,NAVE,NGAS,XLC
C READ(5,*)(MASS(I),I=1,NP)
C READ(5,*)(A(I),I=1,NP)
C DO 2 I=1,NG
C READ(5,*)(I,J(I,J),J=1,15)
C CONTINUE
C OPEN SUMMARY FILE FOR APPENDING SUMMARY DATA
C IF(PFILE(2).EQ.'JACK')THEN
C DATAF='CDATA'
C ELSE
C DATAF='PDATA'
C ENDIF
C OPEN(10,FILE=DATAF)
C CALCULATE VARIOUS CONSTANTS
C NTON=NP+NG
C K=NP+1
```
DO 11 I=1,NG
  A(K)=A(IJ(1,1))
  MASS(K)=MASS(IJ(1,1))
  DO 7 J=2,15
    IF(IJ(I,J).EQ.0) GOTO 21
    A(K)=A(K)+A(IJ(I,J))
  7    MASS(K)=MASS(K)+MASS(IJ(I,J))
  21   K=K+1
  11  CONTINUE

C
C READ NEXT DATA FILE Specified IN PARAMETER FILE

1 READ(5,102)INTEMP,NUMAP,VEL
IF(INTEMP.EQ.'END')GOTO 999
IF(NUMAP.GT.20) THEN
  NAP=NUMAP/10
ELSE
  NAP=NUMAP
ENDIF
INFILE='E'/INTEMP
CALL PF('GET',0,INFILE)
OPEN(B,FILE=INFILE)
READ(B,103)NC,CTIME,CDATE,NN,DTIME
DO 20 N=1,NN
  READ(B,*)TIME(N),(TX(I,N),I=1,NC-1)
20  CONTINUE
READ(B,104)TINF,TC
CLOSE(B,STATUS='DELETE')
C
C DETERMINE CP AND E. CALCUULATE HR, HRF, AND COEFFICIENTS, C(I).
C
CALL ALPT(TC,CPM,E,XKM,RHOM,IER)
CZ=CPM/2/DTIME
IF(INTEMP(I:1).EQ.'P') GOTO 19
E=E/(1.03*(5*NAP-1.))
IF(NUMAP.EQ.0) E=.0
19 HRI=HFUNCTION(TC,TINF,E)
  TCF=TX(NC-1,NN-3)*(TC-TINF)+TINF
  HRF=HFUNCTION(TCF,TINF,E)
  HR=(2*HRI+HRF)/3.
  DO 9 I=1,NTOT
  9 C(I)=CZ*MASS(I)/A(I)
C
C CALCULATE HEAT TRANSFER COEFFICIENTS
DO 13 N=2,NN-1
DO 13 I=1,NTOT
IS=NA+I
13 HT(I,N)=C(I)*((TX(IS,N-1)-TX(IS,N+I))/(TX(IS,N)-TX(I,N))-HR
C SMOOTH HEAT TRANSFER COEFFICIENTS
DO 15 N=4,NN-3
DO 15 I=1,NTOT
15 H(I,N)=(HT(I,N-2)+2*HT(I,N-1)+3*HT(I,N)+2*HT(I,N+1)+HT(I,N+2))/9
C CALCULATE AVERAGE HEAT TRANSFER COEFFICIENTS
J=0
DO 23 N=NN-2,NN
NI=NAVES+J*NAVE
TIME(N)=TIME(NI)
DO 25 I=1,NTOT
H(I,N)=.0
DO 27 K=NI,NI+NAVE-1
27 H(I,N)=H(I,N)/NAVE
23 J=J+1
C CALCULATE RA, NU, T/TINF, ETC.
DT=TC-TINF
N=NAVES+NAVE
TC2=TX(NTOT,N)*DT+TINF
T12=TX(I,N)*DT+TINF
TF=(TC2+T12)/2
HRX=HR/H(NTOT,NN-1)
CALL GASPT(NGAS,TF,RHO,XMU,XK,CP,GRB,PR,IER)
RA=GRB*XLC**3*(TC2-T12)*PR
RE=VEL*XLC*RHO/XMU
IF(VEL.NE.0)RI=RA/PR/RE**2
XNU=H(NTOT,NN-1)*XLC/XK
Z=TC2/T12
C WRITE RESULTS
OUTFILE='H'/INFILE(2:)
OPEN(6,FILE=OUTFILE)
WRITE(6,103)NTOT+1,CTIME,DATE,NN-6,DTIME,OUTFILE,DATE()
DO 17 N=4,NN
17 WRITE(6,109)TIME(N),(H(I,N),I=1,NTOT)
WRITE(6,106)((I,I=1,NG),(J,J=1,NG)
WRITE(6,107)TINF,TC,TCF,HRI,HRX
IF(INTEMP(I,J).EQ.'C')WRITE(6,108)NUMAP
CLOSE(6)
IF(VEL.NE.0)THEN
WRITE(10,100)INTEMP,NGAS,NUMAP,VEL,TC2,T12,XNU,RA,RE,RI,Z,HRX
ELSE
WRITE(10,110)INTEMP,NGAS,NUMAP,VEL,TC2,T12,XNU,RA,RE,Z,HRX
ENDIF
DO 31 N=NN-2,NN
31 WRITE(10,109)TIME(N),(H(I,N),I=1,NTOT)
CALL PF('REPLACE',0,OUTFILE)
CALL PF('CHANGE',OUTFILE,OUTFILE,'CT','PU','M','R')
GOTO 1
C ASK IF ANOTHER SERIES OF DATA FILES ARE TO BE PROCESSED
999 CLOSE(10)
CALL PF('APPEND',DATAF,DATAF)
CLOSE(5,STATUS='DELETE')
PRINT '(2A)','DO YOU WISH TO PROCESS MORE DATA WITH A DIFFERENT' & '" PARAMETER FILE? (Y/N)' READ 105,CYN
IF(CYN.EQ.'Y')GOTO 5
100 FORMAT(/IX,A6,11,15,F7.2,3F7.1,2E9.3,F6.3,2F6.2)
Sample Parameter File for Program HPLAY:

```
2 4 1 4 3 2 0 6
 972 .981 .979 .975
 04351 .04356 .04336 .04336
1 2 3 4 15=0
P0527Q 1 2.87
P0527P 1 2.50
P0527G 1 0.00
P0527R 1 0.00
END 1 0.00
```
APPENDIX D

D.1 Raw Free Convection Data \((T_r = T_f)\)

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<th>(T_m)</th>
<th>(U_{NuL})</th>
<th>(Re_L)</th>
<th>(Re_{L'})</th>
<th>(RI_L)</th>
<th>(T_{w}/T_m)</th>
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*Velocities and temperatures are in units of m/s and K, respectively.*
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<th>$Re_L$</th>
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D.3 Raw Free Convection Data \( (T_r = T_\infty) \)

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