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Date of

Degree: July, 1995

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Title of Study: AN EXPERIMENTAL FACILITY FOR
STUDYING CONVECTIVE
HEAT TRANSFER IN BUILDINGS

Pages in Study: 125

Candidate for the

Degree of Master of Science

Major Field: Mechanical Engineering

Scope and Method of Study: An experimental facility for studying convective heat transfer and air flow in buildings has been built at Oklahoma State University. This facility has many beneficial features that no other experimental facility can offer. By using a removable panel scheme, the room can be configured to study different geometries, different inlet and outlet combinations, heat gain from office equipment and hydronic floor heating. An energy balance will be performed on the room and the experimentally determined film coefficient will be compared to film coefficients determined from published natural convection correlations and forced convection correlations.

Findings and Conclusions: The experimental room has achieved the energy balance criterion that was set. Final uncertainties in the air heat gain could be

higher than desired. This uncertainty can be reduced in the future with improvements in the measurement of the volumetric flow rate of air entering the room. The experimentally determined film coefficient compares favorably to the film coefficients that are derived from the published correlations, which are limiting cases, for natural convection (a lower bound of the film coefficient) and for forced convection (an upper bound of the film coefficient). A majority of the panels were controlled successfully to the desired setpoint of 105°F. The experimental facility at OSU is operational and capable of producing satisfactory results.

ADVISER'S APPROVAL:

AN EXPERIMENTAL FACILITY FOR STUDYING
CONVECTIVE HEAT TRANSFER
IN BUILDINGS

By

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Bachelor of Science

Oklahoma State University

Stillwater, Oklahoma

1992

Submitted to the Faculty of the
Graduate College of
Oklahoma State University
in partial fulfillment of
the requirements for
the degree of
MASTER OF SCIENCE
July, 1995

ACKNOWLEDGMENTS

A thesis could probably be written on all the people that have had some kind of influence on this research project. These people have all provided valuable knowledge and even more importantly, priceless friendship.

First of all, I would like to thank the University Center for Energy Research for providing the funding necessary for the construction of the Building Heat Transfer Laboratory and for recognize the need for such a facility at Oklahoma State University. In the same regards, I would like to thank Dr. Lowery for providing the perfect place to build the facility and allowing us to "ruin a perfectly good reverberation chamber".

A special thank you goes to my advisor Dr. Spitler. His patience on this research project was deeply appreciated and admired. He allowed me to collaborate with him on this project and provided the necessary guidance when it was needed. Most importantly, he believed in my abilities to get the job done.

My office mates: Mike Ramey, who allowed me to search through hundreds of articles that he had obtained for my literature search which saved me a lot of time in trying to find those articles for myself, and for teaching me all I ever wanted to know about UNIX and Jeff Ferguson for his cooperation in the design, construction and testing of this facility.

Three people who, if it was not for them, this project would still be on the ground Darrell Daugherty for designing the room supports. Tony Timko for designing the flow measurement box for the air conditioning system. And lastly Richard Paige for bringing his construction knowledge to this project.

I am also deeply in debt to the MAE Electronics Shop. Without Robert Taylor, Tom Crowell, and Don Lucas I would have been lost when it came to the electronic aspect of this project. They provided valuable assistance and allowed me to scavenge what ever I could find from the shop

The two people who had the most influence on this project do not even attend Oklahoma State. They are my parents, Ed and Pat Sanders. Mom always saw to it that I received the very best out of life and dad taught me the most important thing a person can have is common sense. If it wasn't for their weekend projects around the house, I would have never learned the skills that I did in order to construct this facility. I love you both.

Lastly, but will always be number one on my list, my wife Valerie and her daughter Tiara. Their love, smiles and patience always lifts me during hard times. If not for Valerie's valuable input to the drafts and proof reading of the drafts, it would easily be another year before graduation. I will always love you both.

There are several people that also have played a valuable importance in my life. I will not go on to name these people, (if I did, this section would be the longest part of this dissertation) but they know who they are and what positive affects they have had on my life.

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NOMENCLATURE

inlet - the point at which the working fluid enters the enclosure

outlet - the point at which the working fluid exits the enclosure

Nu - The Nusselt number

A_p or A - The aperture ratio

Ra - The Rayleigh number

h - the convective heat transfer coefficient or film coefficient (BTU/ft² - °F)

h_{mix} - the convective heat transfer coefficient or film coefficient for mixed convective heat transfer (BTU/ft² - °F)

$h_{natural}$ - the convective heat transfer coefficient or film coefficient for natural convective heat transfer (BTU/ft² - °F)

h_{forced} - the convective heat transfer coefficient or film coefficient for forced convective heat transfer (BTU/ft² - °F)

o.c.- on center

Q - Heat Flux(Btu/hr)

A - Area (ft²)

T - Temperature (°F)

F - The grey interchange factor

σ - The Stefan - Boltzmann constant

R_o - Wire resistance (ohm)

V - Line voltage (volts)

α - Coefficient of thermal resistance of the wire (ohms/ohm/°F)

U - Overall heat transfer coefficient (BTU/(hr-ft² - °F))

U_o - Fluid velocity at inlet (ft/s)

vfr - Volumetric flow rate (ft³/min)

β - Coefficient of thermal expansion (1/R)

g - gravitational acceleration (32.2 ft/s²)

L_c - Characteristic length (ft)

ν - kinematic viscosity (ft^2/s)

k - thermal conductivity of fluid ($\text{Btu/hr-ft-}^\circ\text{F}$)

u - uncertainty

ρ - density of fluid (lb_m/ft^3)

C_p - Specific heat of air at constant pressure ($0.24 \text{ Btu/lb}_m\text{-}^\circ\text{F}$)

Ar - Archimedes Number

Gr - Grashof Number

Pr - Prandtl Number

J - Jet momentum number

1 INTRODUCTION

1.1 Overview

Of the three fundamental heat transfer processes occurring in buildings, conduction, radiation and convection, convection is the least understood. In the analysis by simulation of a building's cooling and heating load, the conductive and radiative heat transfer can be predicted relatively accurately, but the prediction of the convective heat transfer is imprecise and relies on experiments performed 60 years ago using vertical, free standing flat plate geometry that is not typical of buildings. The inability to accurately predict the convective heat transfer component can lead to improper design of the heating and air conditioning system.

This thesis describes the design and development of an experimental facility for the study of convective heat transfer in buildings. The interior dimensions of the experimental room are 12 feet by 16 feet by 10 feet; these dimensions may be made smaller to study smaller enclosures such as attics and plenum spaces.

The goal of this project is to develop a versatile and operational facility for the study of natural and forced convection in buildings. The facility should be unique in design and not limited by the types of surfaces that can be studied, or only to convective heat transfer on surfaces.

1.2 Literature Review

To date, convective heat transfer in buildings has been studied in scale models and full size enclosures, with particular emphasis on natural convection heat transfer that occurs at Rayleigh numbers on the order of 10^{10} . Some of these test models and enclosures are discussed here.

1.2.1 Scale Models

Lawrence Berkeley Laboratory (LBL)
Fred Bauman et al.

Bauman's study (Bauman et al., 1980) used a table top size scale model utilizing water as the working fluid. This model was designed to measure natural convection within a single zone, or, with the insertion of a partition (to simulate a doorway), between two geometrically identical zones. The model had dimensions of 30 inches by 10 inches by 5 inches. The two opposing 30 inch by 5 inch walls were constructed from 0.2 inch copper plate. The remaining four surfaces were constructed from 0.5 inch plexiglass. The removable partition was constructed from 0.25 inch plexiglass.

One copper plate served as the hot wall while the other served as the cold wall. The hot wall had a constant heat input provided by six thermofoil heaters attached to the outside surface. The cold wall was chilled with 59°F tap water running through four horizontal 0.4 inch outside diameter cold rolled copper tubes glued to the outside surface of the wall with high conductivity epoxy. The entire model was covered with 0.5 inch polyethylene sheeting and a 2 inch shell of styrofoam. No measurements of temperature were made to calculate losses from the model to the surroundings.

Experiments performed covered the Rayleigh number range of 1.6×10^9 to 5.4×10^{10} . Data that was obtained agreed well with previous studies dealing with convective heat transfer in a single zone. Due to Prandtl number differences between water and air, strict similarity to full scale enclosures could not be obtained. This leads to possible limitations in use of the data obtained from the model in the development of correlations for full scale enclosures using air as the working fluid.

Lawrence Berkeley Laboratory (LBL)
Mark Nansteel et al.

Nansteel et al. (1981) expanded upon Bauman's work by using a model with dimensions of 33 inches by 12 inches by 6 inches. Again the two longest walls were a hot

wall - cold wall combination. The hot wall was manufactured from 0.2 inch copper plate and heated by 18 thermofoil heaters in three vertical columns of six each. The cold wall was 0.6 inch aluminum plate with two rectangular channels machined into the aluminum. These channels traversed the full breadth of the enclosure seven times. Tap water was used as the coolant for the wall.

The rest of the surfaces were made of 0.5 inch plexiglass. The joints between the plexiglass and the metal walls were sealed with cork gasket to minimize conduction. The ceiling, however, was removable and thermally isolated from the metal walls by air gaps. A removable partition made of 3.75 inch plexiglass could be inserted through the ceiling of the model so that partition lengths of 0, 1.5, 3, and 4.5 inches could be studied in the model. Two other partitions could also be used in the model. One of these was a highly conductive partition fabricated from aluminum. The other partition was an adiabatic partition fabricated from polystyrene foam encased in 0.004 inch stainless steel sheets.

The entire model was enclosed in a polyurethane foam shell and the outer surface was covered with aluminum foil to reduce radiative heat transfer to the surroundings. However, no estimate of the conduction losses or of the radiative heat transfer was apparently made in the experiment. Water was used as the working fluid in the model. The Rayleigh number range for Nansteel's experiments was 2.3×10^{10} to 1.1×10^{11} . Again the high Prandtl number limited the correlations that could be developed; however, two correlations were developed for conducting and non conducting partitions. These two correlations are:

$$Nu = 0.748A_p^{0.256}Ra^{0.226} \quad \text{conducting partitions} \quad (1.1)$$

$$Nu = 0.762A_p^{0.473}Ra^{0.226} \quad \text{non-conducting partitions} \quad (1.2)$$

where: Nu = the Nusselt number
 A_p = the aperture ratio, which is the product of the height and width of the partition divided by the product of the height and width of the enclosure
 Ra = the Rayleigh number

The effect higher Prandtl number had on the heat transfer could not be determined.

Solar Energy Research Institute (SERI)
Mark Bohn et al.

The model used by Bohn et al. (1983) was a cubical enclosure with interior dimensions of 12 inches. The model was constructed from eight aluminum plates that were 0.5 inches thick. The four inner plates overlapped one another and were screwed together with neoprene gaskets in the joints. The four outer plates provided heating and cooling to the four inner plates of the enclosure. These four outer plates were sealed and bolted to the enclosure walls. The aluminum walls were either heated or cooled by pumping water through channels that were machined in the outer plates of the enclosure. The top and bottom of the enclosure consisted of 0.5 inch Lucite plates. These plates were screwed to the enclosure walls and had neoprene gasket joints.

The aluminum walls were insulated with 3.25 inch thick urethane foam board insulation. Conductive heat losses were estimated based on wall temperature, outer surface temperature of the insulation and thermal resistance of the insulation. The top was uninsulated and the losses through it were estimated based on the highest temperature on the outside surface, ambient temperature and an assumed convection coefficient. The bottom plate was near ambient temperature, and both the top and bottom plates were assumed to be adiabatic.

The working fluid in the enclosure was de-ionized water. Experiments covered the Rayleigh number range from 0.3×10^{10} to 5×10^{10} . High Prandtl numbers still resulted; however, a correlation indicated a laminar boundary layer flow heat transfer mechanism. The correlation that was presented is:

$$\text{Nu} = 0.62\text{Ra}^{0.25} \quad (1.3)$$

where: Nu = the Nusselt number
 Ra = the Rayleigh number

however, a transition to turbulence was present at high Rayleigh numbers.

The Rayleigh number for this correlation used the temperature difference between the hot and cold walls, or the overall temperature difference for the model. However, when the film coefficient was calculated, the temperature difference between the wall temperature and the bulk temperature was used.

By rotating the model, studies on heat transfer on the floor and the ceiling as well as two walls could be performed. From these experiments, turbulent flow was produced when the floor was heated. The primary heat transfer from the floor was the release of thermals. This influenced the heat transfer of the vertical surfaces when it interfered with the wall boundary layers.

University of Tennessee
M. Keyhani et al.

Keyhani's model was made from a rectangular box (Keyhani et al., 1991) with interior dimensions of 5.5 inches by 5.5 inches by 6.7 inches. The box was fabricated from 0.5 inch plexiglass. The end walls were made out of 0.25 inch plate glass. A copper heat exchanger plate covered the top surface cavity and a constant temperature water bath was used to heat the copper plate.

The actual enclosure, however, was formed by placing a movable vertical plate and the copper plate inside the plexiglass box and glass end walls. To minimize losses, the movable plate, the copper plate, and the bottom plate were instrumented with silicone-rubber thermofoil guard heaters. Differential thermopiles were used to determine conductive heat losses.

In the model, ethylene glycol was used as the working fluid. The Rayleigh number range for the experiments was 2.5×10^6 to 4.6×10^{10} . The resulting Prandtl number was much higher than that for the experiments performed by other researchers using water as the working fluid. This would exacerbate the problem of applying the correlations to full size enclosures that use air as the working fluid. The correlation that was developed from this scale model is:

$$Nu = 0.296Ra^{0.223}A^{-0.53} \quad (1.4)$$

where: Nu = the Nusselt number
 Ra = the Rayleigh number
 A = the aperture ratio

The Rayleigh number in this correlation also used the temperature difference between the hot surface and the cold surface. The development of the film coefficient for the experiments performed also used the same temperature difference. The bulk temperature was not used in any of the calculations.

University of California-Berkeley
L. Neiswanger et al.

The experimental apparatus used by Neiswanger et al. (1987) was a model with dimensions of 7.9 inches by 10.8 inches by 7.9 inches. An inlet and an outlet, each with dimensions of 2.7 inches by 7.9 inches, were located opposite each other in the center of each of the two square walls.

The two end walls with the inlet and outlet along with the top and bottom of the enclosure were fabricated from 0.5 inch transparent acrylic plastic. These surfaces permitted flow visualization. Water was used as the working fluid in the model. The water's temperature was kept near the temperature of the laboratory to help in minimizing the conductive losses through all surfaces of the model. The two long vertical walls had a sheet of 0.1 inch thick Inconel foil stretched over 1 inch thick polystyrene foam insulation. An electric current was passed through the Inconel sheet to generate a uniform heat flux. Conductive losses were minimized by using the thin Inconel and the polystyrene foam insulation.

It was concluded that the results obtained from this experimental apparatus were only applicable for the Prandtl number and geometry studied. The Rayleigh number range that the experiments covered was 5×10^9 to 1×10^{11} . The primary use of this model was for studying

mixed convection heat transfer. The following correlation for mixed convection coefficient was presented:

$$h_{\text{mix}} = (h_{\text{natural}}^a + h_{\text{forced}}^a)^{1/a} \quad (1.5)$$

where a is a constant.

Natural convection could not be studied with this model since a cold wall was not present. Even though the two walls were heated, only one was studied due to symmetry of the model. This was the only model to use an inlet and outlet for the water, but the configuration for the inlet and outlet are not typical of full size rooms.

Massachusetts Institute of Technology (MIT)
D. A. Olson et al.

The model used here was an actual scale model of a full size enclosure (Olson et al., 1990), which will be discussed in the next section. The model used R-114 gas as the working fluid. The model used the same Rayleigh number and had the same dimensionless hot wall - cold wall temperatures as the full scale enclosure. Radiative heat transfer was not scaled between the two, however, but this had no apparent effect on the results obtained from the two facilities.

The model had dimensions of 53.5 inches by 26.8 inches by 18.5 inches. The vertical walls were fabricated from three vertical sections of 0.25 inch thick aluminum plates. Foil faced polyurethane strips insulated the wall sections and the assembled wall from the floor and ceiling. Electric resistance strip heaters were used to make one wall hot, while copper cooling tubes soldered to the back of the opposite wall created a cold wall.

The ceiling was a double pane plexiglass window with a 1 inch gap between the panes. The floor was also a plexiglass pane. The model was encased in 6 inches of polyurethane board. An insertable partition was fabricated from 0.6 inch thick polyurethane insulation.

Significant differences were found in temperature profiles and flow patterns as compared to the studies conducted in scale model enclosures using water as the working fluid. Experiments covered the Rayleigh number range of 1.3×10^{10} to 3.4×10^{10} . No correlation was developed from the data relating the Nusselt number to the Rayleigh number, however, the data did agree favorably with the correlations from previous studies conducted by Bohn and Nansteel.

1.2.2 Discussion of Scale Model Research

Scale models are advantageous in that they are relatively inexpensive to construct and do not require a large area in which to operate. Also, water is opaque to radiation, so the heat transfer that occurs in the model is convective. The higher Prandtl number that occurs with water and the other working fluids that were used allows Rayleigh number similarity with small scale models.

There are, however, many drawbacks to using them. The higher Prandtl number forces transition from laminar flow to turbulent flow at a higher Rayleigh number. The applicability of the laminar correlations obtained using water as the working fluid may be in doubt for the high end of the Rayleigh number regime. Also, in the mixed convection regime, flow is driven by both buoyancy and inlet/outlet flows. It is generally impossible to scale both phenomenon simultaneously.

1.2.3 Full Scale Enclosures

Massachusetts Institute of Technology (MIT)
D. A. Olson et al.

This is the full size enclosure (Olson et al., 1990) corresponding to the scale model discussed above. The room had dimensions of 12.8 feet by 25.9 feet by 8.2 feet. One end wall consisted of electric radiant heating panels with 0.06 inch thick aluminum sheeting to provide an isothermal, reflective surface. This served to reduce radiative heat transfer from the wall, which was estimated along with the conductive losses. It appears no attempt was

made to verify the estimates of the conduction losses or the radiative heat transfer to the surroundings. The opposing wall served as the cold wall and was cooled using copper solar collector panels mounted on the inside surface.

A partition constructed from 1 inch thick aluminum foil faced polyurethane insulation panel was mounted midway between the hot wall and cold wall and extended from the floor half way up the height of the room. Air was used as the working fluid in the full size enclosure and the results obtained from the room agreed well with the results from the model. However, the results from the room matched the results from the scale model research using water as the working fluid in Bohn's and Nansteel's experiments within thirty percent. The Rayleigh number range used in the experimental room was from 2.0×10^{10} to 3.1×10^{10} .

This facility can only be used to study natural convection. No air inlet or air outlet is present so forced convection cannot be studied in this facility.

Energy Monitoring Company (EMC)
Chris Martin et al.

The facility built by EMC (Martin et al., 1988) was a cube with internal dimension of 7 feet. The walls, floor and ceiling were made from brick with 5 inches of styrofoam insulation covering the outside surfaces of the cube.

A metered wall was installed inside the cube. The wall was divided into three horizontal sections, each heated by a tubular heater. The back of the wall was covered with 4.3 inches of styrofoam to reduce conductive losses. Highly polished stainless steel covered the wall surface to minimize radiative heat transfer. A uniform wall temperature was achieved by using fans along with a ventilation system to move air between the three wall sections. Back losses for the wall were nearly twice the design figure of 10%. This affects the uncertainties in the calculated back loss, but not to an unacceptable level.

This facility is useful for studying only one surface, a wall. Due to size limitations, a room cannot be studied inside the brick enclosure. No mention is made of the air inlet's location in respect to the surface of the wall, or of an air outlet from the room. The location

of these two could affect the heat transfer of the wall. No discussion of a Rayleigh number range was made and no correlations were developed.

Florida Solar Energy Center (FSEC)

S. Chandra

This facility was a 17.7 foot by 11.7 foot by 8.1 foot naturally ventilated room (Chandra et al., 1989) which was part of the Passive Cooling Laboratory. The room was located in the southeast corner of the lab. Two windows, which were actually unglazed holes in the wall, were located in the same wall. Two exterior wing walls were used to create a positive pressure on one window and a suction on the other, depending upon the wind direction. This created a cross-ventilation in the room.

The floor of the room was a slab-on-grade floor with a rubber pad and carpeting. The walls were lightly insulated, conventional stud frame walls. The ceiling was also lightly insulated and had a 4 foot ceiling fan hanging from it. All internal surfaces were unfinished bare 0.5 inch thick gypsum wall board.

Heat transfer was measured on the wall opposite the one with the windows. The windows were first closed and the room heated with two convective heaters while the ceiling fan was operating. The heaters and the ceiling fan were then turned off and the temperature in the room allowed to stabilize. The heaters were then removed and the windows opened to naturally ventilate the room.

Three panels of 4 foot by 7.87 foot gypsum wall board, with a thickness of 0.5 inches, were nailed on top of the existing wall board and covered with a thermocouple grid on the front and the back. These thermocouples were used to calculate the radiative component of the total heat transfer; no measurement of the conductive component was made.

This facility was used to measure the convective heat transfer on only one surface, the wall opposite of the window wall, which was a rough surface. Since no other ventilation system was connected to the room, only natural ventilation could be studied in this facility. No flow visualization work was performed to see how the air flowed across the wall or if the

boundary layer on the wall was laminar or turbulent. A Rayleigh number range for the experiments was not given and the only correlation developed was that the convective heat transfer coefficient is a function of local air velocity. It was shown that this correlation that was developed for a rough vertical surface does not agree with the equation or data presented in the ASHRAE Handbook of Fundamentals which was obtained from experiments performed on a copper plate.

Colorado State University (CSU)
J. Neymark

The facility built at CSU (Neymark et al., 1988) was a cube with internal dimension of 8 feet. This full size enclosure used air as the working fluid and was geometrically similar to the apparatus at SERI (see previous section on scale models).

The hot wall was constructed from eight electrical resistance heaters utilizing a graphite heating element with a uniform resistance over the entire surface. Aluminum foil covered the heater surface to provide a low emissivity and reduce the radiative component of the total heat transfer. A heated air gap on the back side of the wall was used to minimize losses to the surroundings.

The cold wall was fabricated from copper solar collector absorber plates. Water entered through a manifold in the bottom of the plate and exited from an upper manifold. Water flow was at a rate such that the water temperature difference between the two manifolds was less than 0.3°F. The remaining four surfaces of the enclosure were constructed from styrofoam sheets with aluminum foil facing. These remaining four surface were insulated from the surroundings with 18 to 36 inch fiberglass insulation. A partition used to divide the test cell into two zones was fabricated from two 1 inch thick aluminum foil faced styrofoam sheets set inside the enclosure, parallel to the hot wall - cold wall combination.

Conduction losses from the enclosure were measured by placing thermocouples on the outside surface of the fiberglass insulation and only running the hot wall of the enclosure.

Radiation heat transfer from the interior surfaces were measured using surface mounted thermocouples and the measured absorptivity of the six surfaces.

Since no ventilation system exists, this full size enclosure can only be used to perform studies of natural convection. The Rayleigh number range for the experiments was from 4×10^{11} to 1×10^{13} and this facility was used to make an important discovery in that at Rayleigh numbers on the order of 10^{12} , the lower half of the hot wall's boundary layer was laminar in the water filled model, while in the full size air filled model, a turbulent boundary layer was present. The correlation developed from the data obtained in the full scale enclosure was:

$$\text{Nu} = a\text{Ra}^b \quad (1.6)$$

where: Nu = the Nusselt number
 Ra = the Rayleigh number
 a and b = constants dependent upon the aperture ratio.

*SulzerInnotec, Research and Development
Andreas Schachenmann*

This experimental room (Schachenmann et al., 1990) with interior dimensions of 22 feet by 14.8 feet by 9.7 feet was constructed for verifying numerical methods of pure ventilation and free convection. An air inlet was located near the ceiling in one of the 14.8 foot walls, while the air outlet was located near the floor in one of the 22 foot walls. The heat transfer surfaces of the room, which were the floor and the 22 foot wall opposite the air inlet, consisted of plate heaters with many passes through them. The surface was made either hot (91°F) or cold (54°F) by running hot water or chilled water through the circulation system. The ceiling and the rest of the walls were insulated. All surfaces of the experimental room were covered with aluminum foil to reduce the radiative component of heat transfer. Removable partitions could be set up anywhere on the floor of the experimental room. The experimental room was located inside a building so that experiments could be performed year round; however, the experimental room itself was not enclosed inside a guard space.

Conduction losses into the building were measured, but apparently no attempt was made to reduce the losses from the two heat transfer surfaces.

A Rayleigh number range for the experiments was not given. The correlation developed was similar to that developed by Neiswanger for the convective heat transfer coefficient, except that Schachenmann applied it to the Nusselt number instead.

Only one inlet/outlet configuration is possible for this room. It can be used to perform experiments for natural convection, but only between the floor and one wall, not between two facing walls. The concept of a movable partition is a good idea in that the partition can be set at any position, at any height, and at any angle to the air inlet or air outlet. This concept along with a movable inlet/outlet combination would permit an unlimited number of flow fields to be studied.

University of Illinois - Urbana Champaign (UIUC)
Jeff Spitler et al.

The first facility built at UIUC (Spitler et al., 1987) was constructed from 8 inch concrete block. The room was enclosed by a guard space and both interiors were temperature controlled. The guard space was divided into two thermally separate L-shapes, one heated to simulate a typical summer design day, and the other space a free-floating temperature space.

The concrete block room had internal dimensions of 14.6 feet by 11.9 feet by 8.5 feet. Heat flux transducers were installed on the north and east walls (adjacent to the heated portion of the guard space), and the walls were then painted black. Depending on the experiment being performed, the remaining two walls either had aluminum foil covering them, or had a black painted surface.

The floor and ceiling were insulated with 3.9 inches of aluminum foil faced styrofoam. This made these surfaces nearly adiabatic and reduced the number of heat transfer processes occurring in the room. The insulation for the floor lay on the laboratory floor. 5.9 inches of concrete covered the ceiling's insulation and separated it from the guard space.

A diffuser located at the center of the ceiling provided conditioned air to the room. A return duct attached to the south wall was used as an air outlet from the room. The mass flow rate of air into the room could be varied using a damper.

Losses due to conduction were not calculated and cooling load calculations for the room showed that the assumption of an adiabatic floor and ceiling were incorrect. Radiative heat transfer was calculated and subtracted from the total heat flux to determine the convective heat transfer component.

Only two surfaces could be studied with this room, only one air inlet/outlet configuration could be used, and the ventilation system could only provide a low flow rate. The two passive walls had no control whatsoever. A range of Rayleigh numbers and a correlation were not presented; however, several regression analyses were performed to determine on what variables the convective heat transfer coefficient was dependent. The laboratory floor may have had an effect on the floor of the room, especially if it were a cold concrete floor, which would invalidate the adiabatic assumption.

The second facility constructed at UIUC (Spitler et al., 1991) had internal dimensions of 15 feet by 9 feet by 9 feet. The walls, floor and ceiling were well insulated with a typical R value of $57 \text{ }^\circ\text{F}\cdot\text{ft}^2/\text{Btu}$. The 6 surfaces were made up of 53 individually controlled wire resistance heater panels. Surface temperature measurements made with thermocouples were used to control the panels.

A fan system was used to deliver conditioned air to one of two inlets to the room. One inlet was located in the center of the ceiling, while the other was located in a side wall, in the center of the left side of the wall. The outlet from the room was located on the opposite wall from the air inlet to the room, in the bottom left corner.

Radiation heat transfer was minimized by maintaining the room surfaces at near-isothermal conditions. Conduction heat transfer was minimized by the high resistance value found behind all six surfaces and by enclosing the room in a temperature controlled guard space.

This facility had a number of beneficial features, one of which is that six surfaces can be studied at once. One limitation of this feature, though, is that the wire resistance panels were fixed, and could not be removed. Since the panels could not be removed, the dimensions for the experimental room were fixed. Another plus for this facility is the use of two different inlets; however, their positions in the ceiling and the side wall are fixed, as well as the position of the outlet, due to the fixed panel concept. This facility is useful for studying either forced or natural convection. One of the walls with heater panels has cold plates behind the panels, so that a hot wall - cold wall configuration can be used to generate a natural convection loop. The floor of the room sits on the laboratory floor and losses due to conduction through the floor cannot be controlled since the temperature outside the floor cannot be controlled.

1.3 Objectives

The objective of this project is to develop a versatile experimental facility for the study of convective heat transfer and air flow in buildings. The experimental room should be able to be set up for a multitude of air inlet/outlet configurations. Multiple room configurations should also be possible. This means that the room should be able to be configured as an attic or a plenum space, about which very little is known in the area of heat transfer. This demand also includes vertical partitions for the study of heat transfer between two zones.

The room should also be useful for studying different types of surfaces, such as building exteriors, different roughness of walls that are found inside buildings today, and fenestration. Losses due to conduction and the radiative heat transfer for all six surfaces of the room must be minimized as much as possible.

The experimental room should not be limited to the study of convective heat transfer on surfaces only. It should also be capable of being used to perform experiments for the measurement of radiative and convective fraction of heat gain from equipment such as computers, photo copying machines, printers and other modern office equipment.

Computer programs using computational fluid dynamics algorithms have been written to predict air flow and temperature inside a room. The experimental room can be used to validate these computer programs.

A heating system that has become popular recently is hydronic radiant floor heating. There is debate over the modeling and controlling of this type of system. The experimental room with its removable panel scheme is an excellent facility for conducting studies on hydronic radiant floor heating systems.

Accuracy is of utmost importance. In order to obtain excellent results, the temperature measurement is very important since all data reduction and any derivation of correlations will depend upon the temperature measurement. In the same regards, the control of the heated panels is very important. The use of a proportional controller should give better results than any other facility has obtained with previously used control algorithms.

2 EXPERIMENTAL FACILITY

2.1 The Experimental Room

The experimental room is a 12 foot by 16 foot by 10 foot (inside dimensions) enclosure with one hundred and eighteen "honeycomb" cells. A total of one hundred and eighteen panels may be used for natural convection experiments, while one hundred and sixteen panels may be used for a forced convection experiment (two "honeycomb" cells are for the air inlet to the room and the air outlet from the room). Provisions have been made to enclose the experimental room with a temperature controlled guard space with dimensions of 21 feet by 27 feet by 22 feet. An unlimited number of air inlet/outlet configurations can be set up due to a removable panel scheme in which one removable panel covers one "honeycomb" cell. This removable panel scheme also allows different heated panels or cooled panels to be studied. It will also permit the study of heat transfer in different types of room configurations.

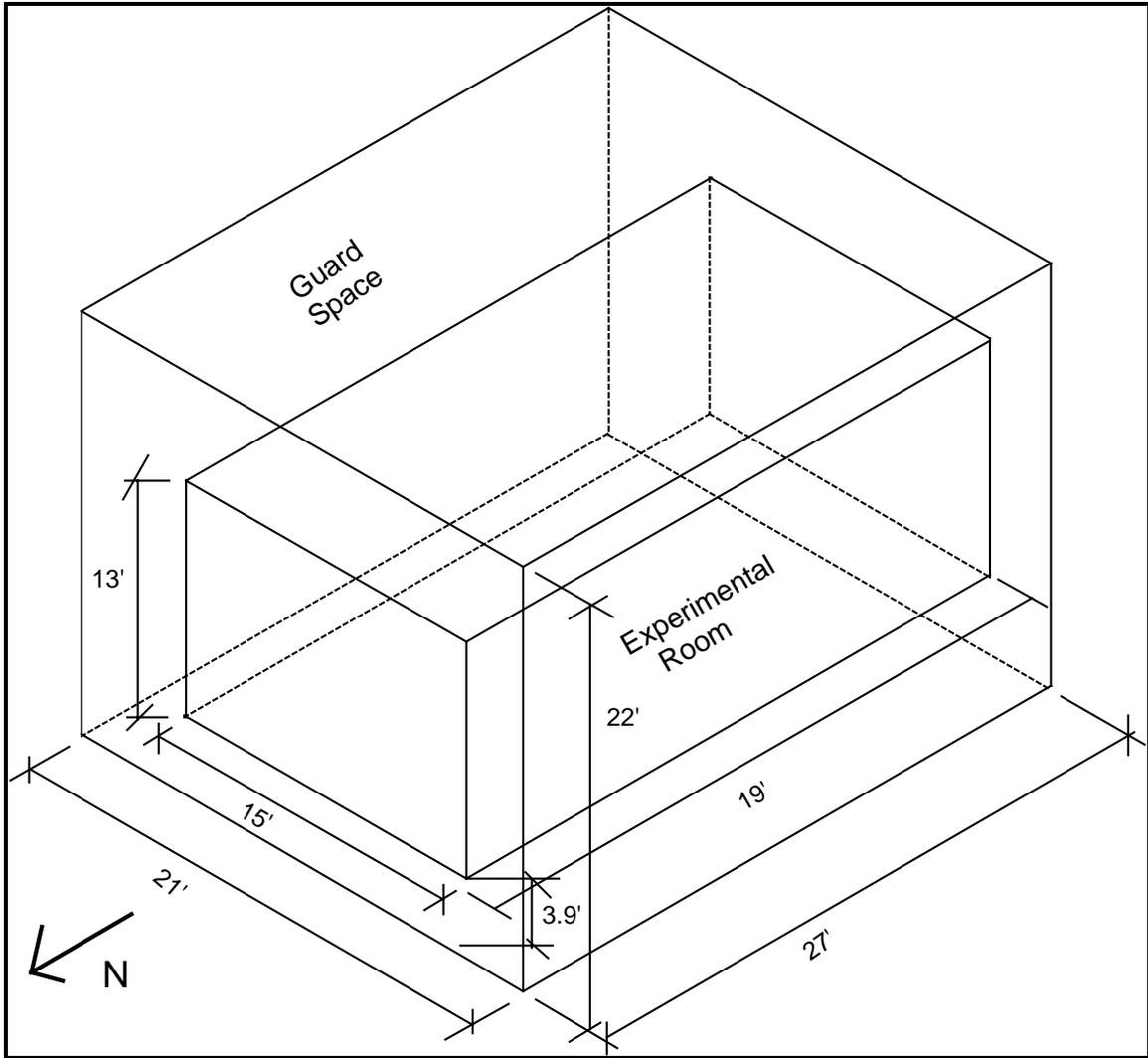


Figure 2.1: The experimental facility

2.1.1 The Floor of the Experimental Room

The floor of the experimental room is elevated above the floor of the laboratory with twenty six 2 inch by 10 inch by 5 foot planks placed around the perimeter of the floor. Four 2 inch by 10 inch by 5 foot planks are placed in the center of the floor to prevent any possible sagging of the floor. All of the legs for the room are attached to each other with 2 inch by 4 inch boards in order to stiffen the supports. All legs are attached to the floor of the experimental room with two 0.375 inch bolts per leg.

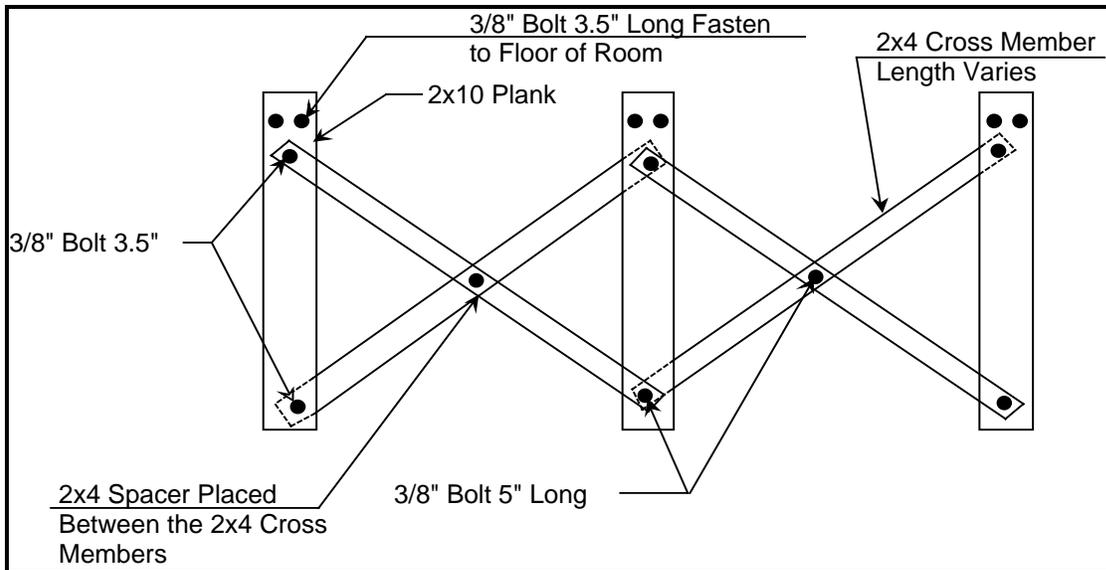


Figure 2.2: Leg supports for the floor (typical)

By lifting the floor of the experimental room off the laboratory floor, the temperature underneath the room can be controlled to a desired set point. This will assist in reducing the conduction losses through the floor, something no other facility has done.

The floor is constructed from 2 inch by 8 inch boards. The floor is essentially two layers of 2 inch by 8 inch boards assembled to create a 2 foot o.c. by 4 foot o.c. "honeycomb" of twenty four cells. Figure 2.3 shows the honeycomb layout of the floor as well as the leg placement.

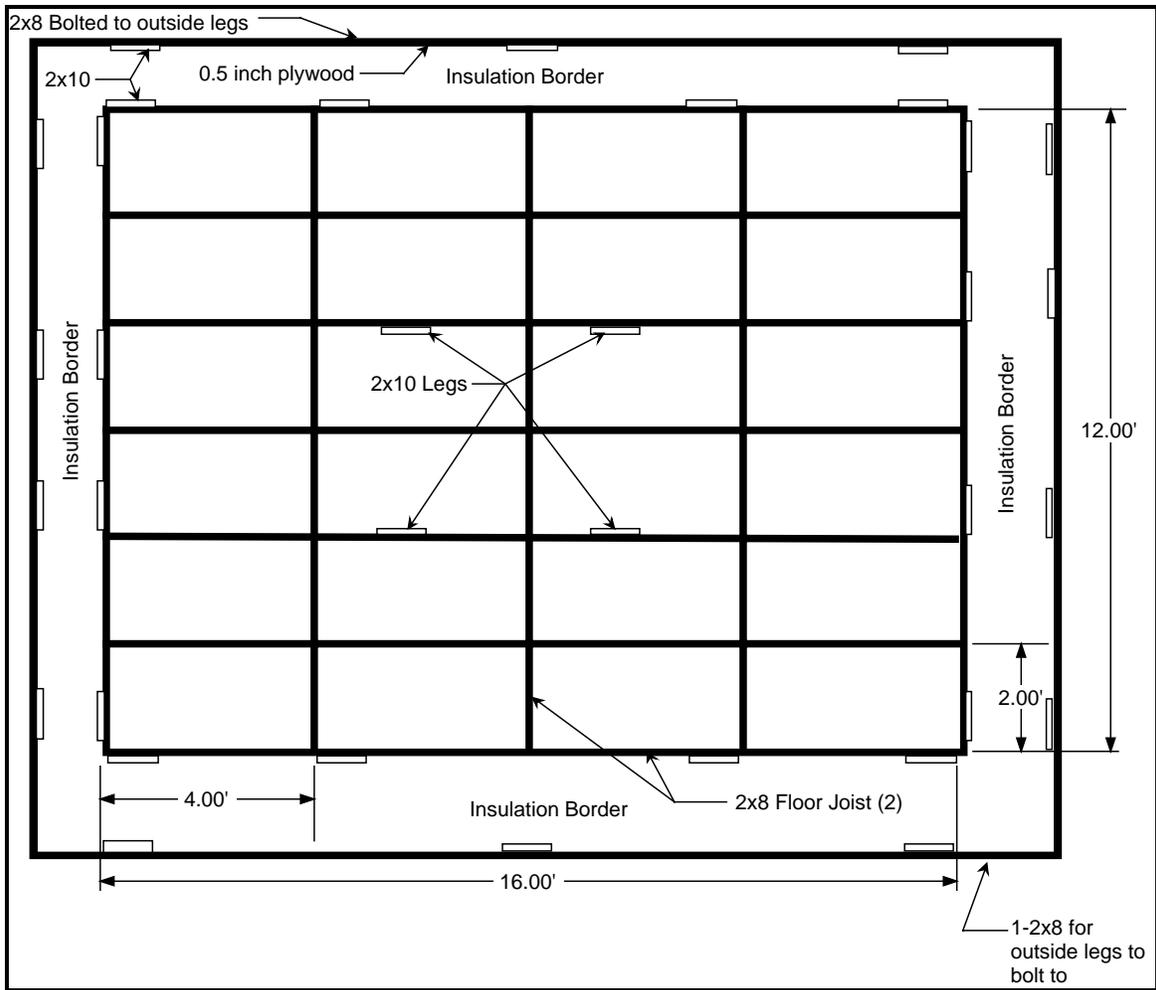


Figure 2.3: Plan view of the floor

The insulation border around the perimeter of the floor is filled with R-19 insulation. This insulation border is 18 inches wide and supports the walls.

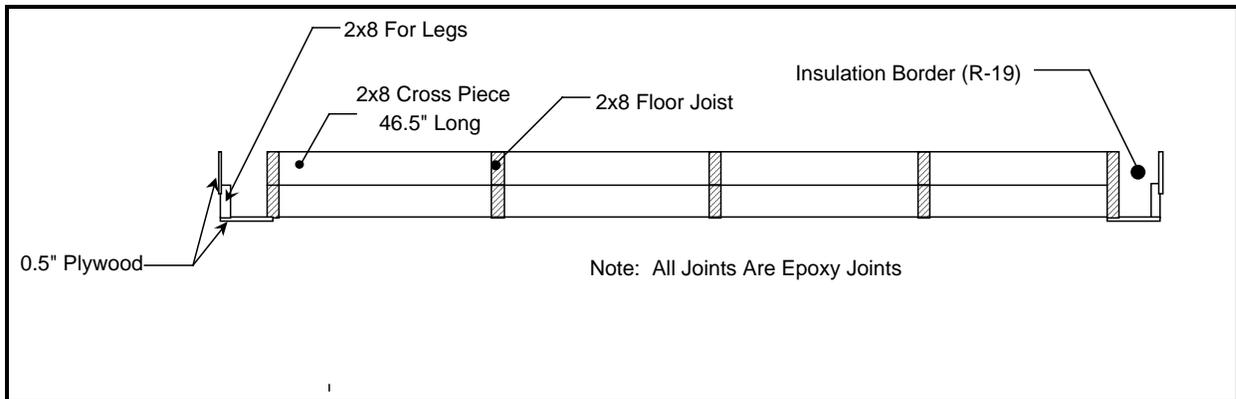


Figure 2.4: Profile view of floor

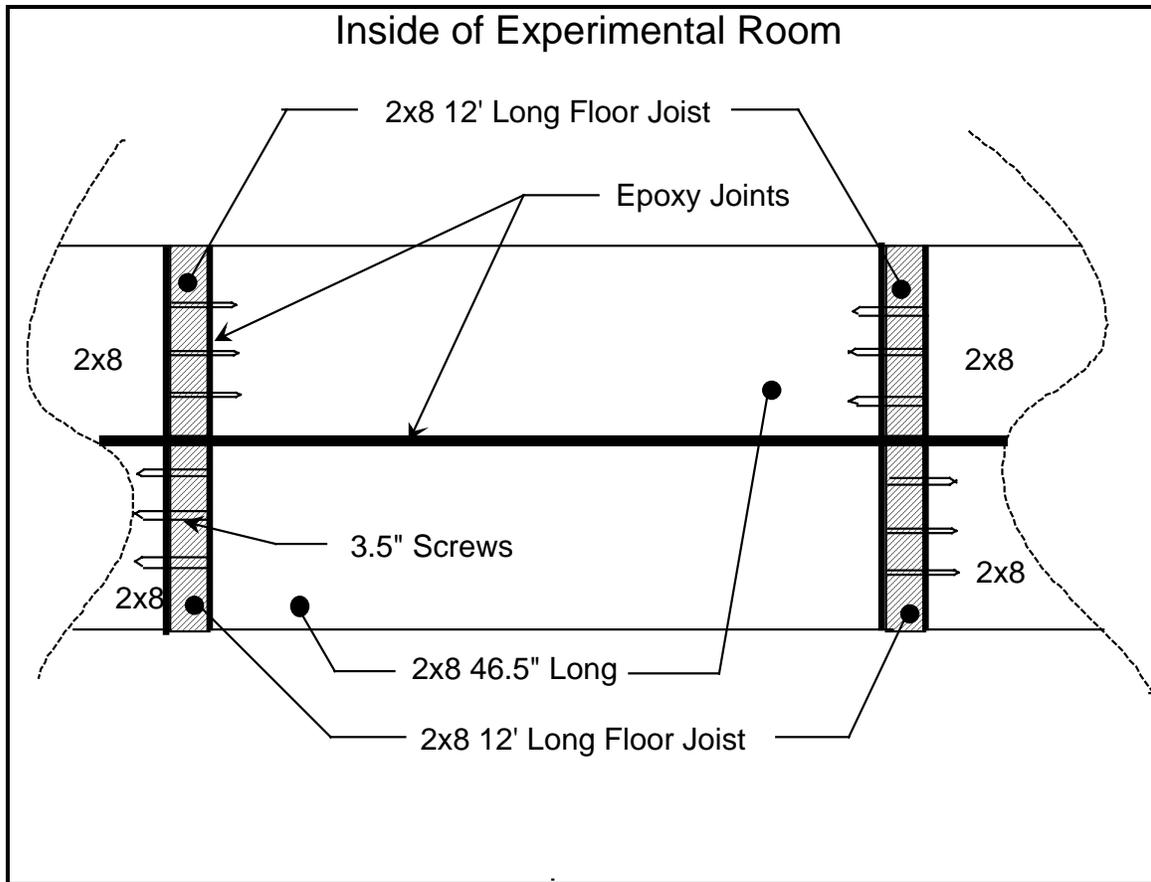


Figure 2.5: Floor joint detail

2.1.2 The Walls of the Experimental Room

The walls of the experimental room are constructed from 2 inch by 4 inch studs and are formed to create a 2 foot o.c. horizontal by 4 foot o.c. vertical "honeycomb" of sixty eight total wall cells. Two walls are 16 feet in length and the other two are 12 feet in length. All walls are 10 feet high. Figures 2.6 and 2.7 show the "honeycomb" and dimensions of the north, south and west walls respectively.

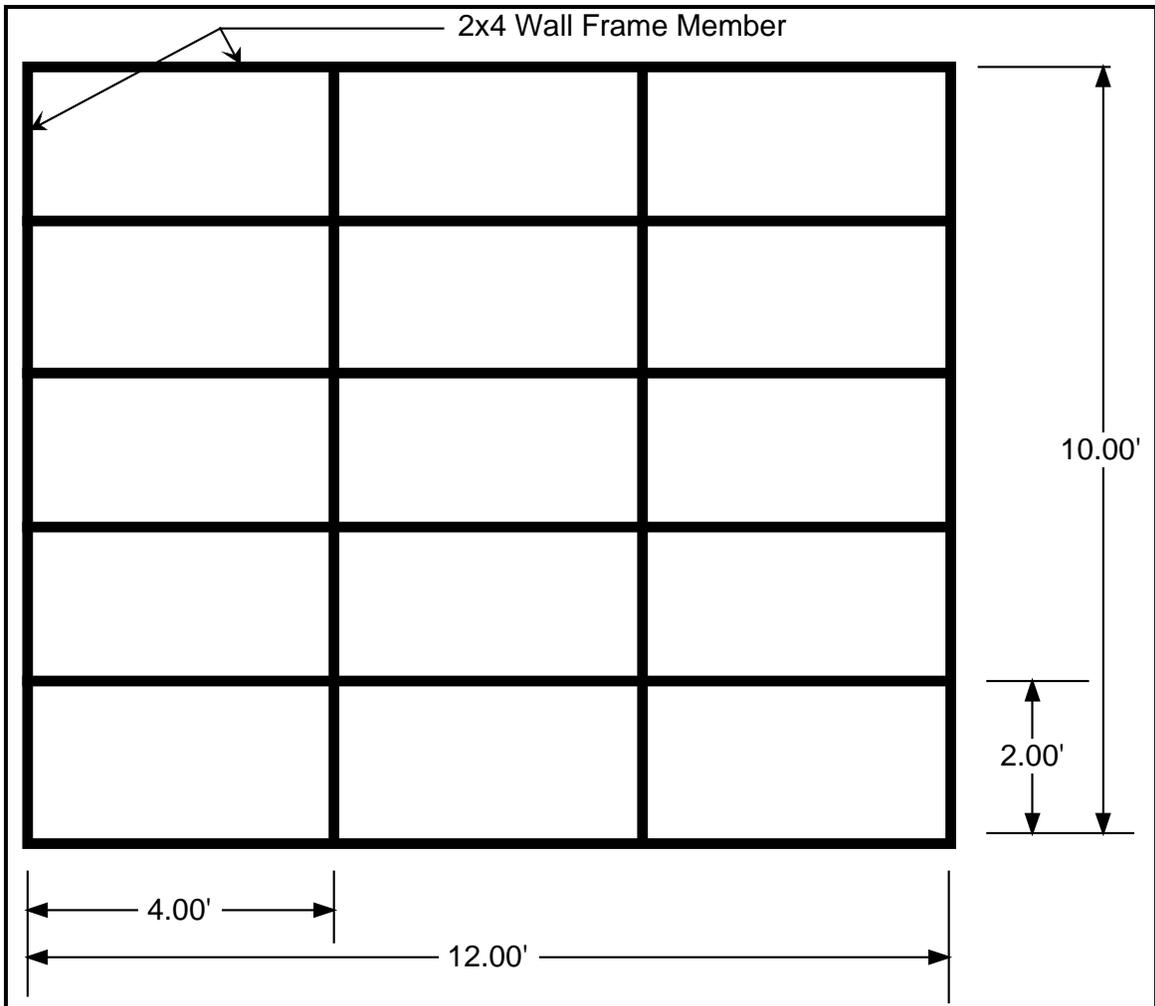


Figure 2.6: View of north and south walls

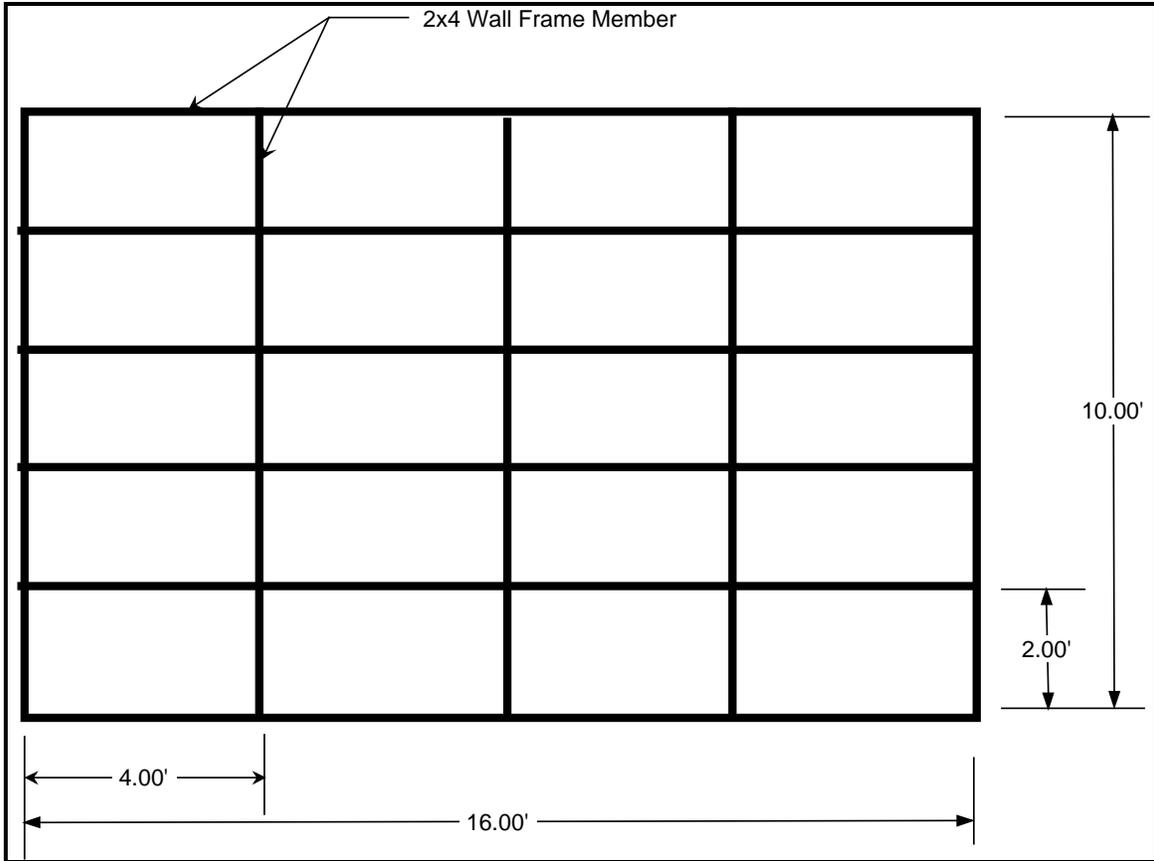


Figure 2.7: View of west wall

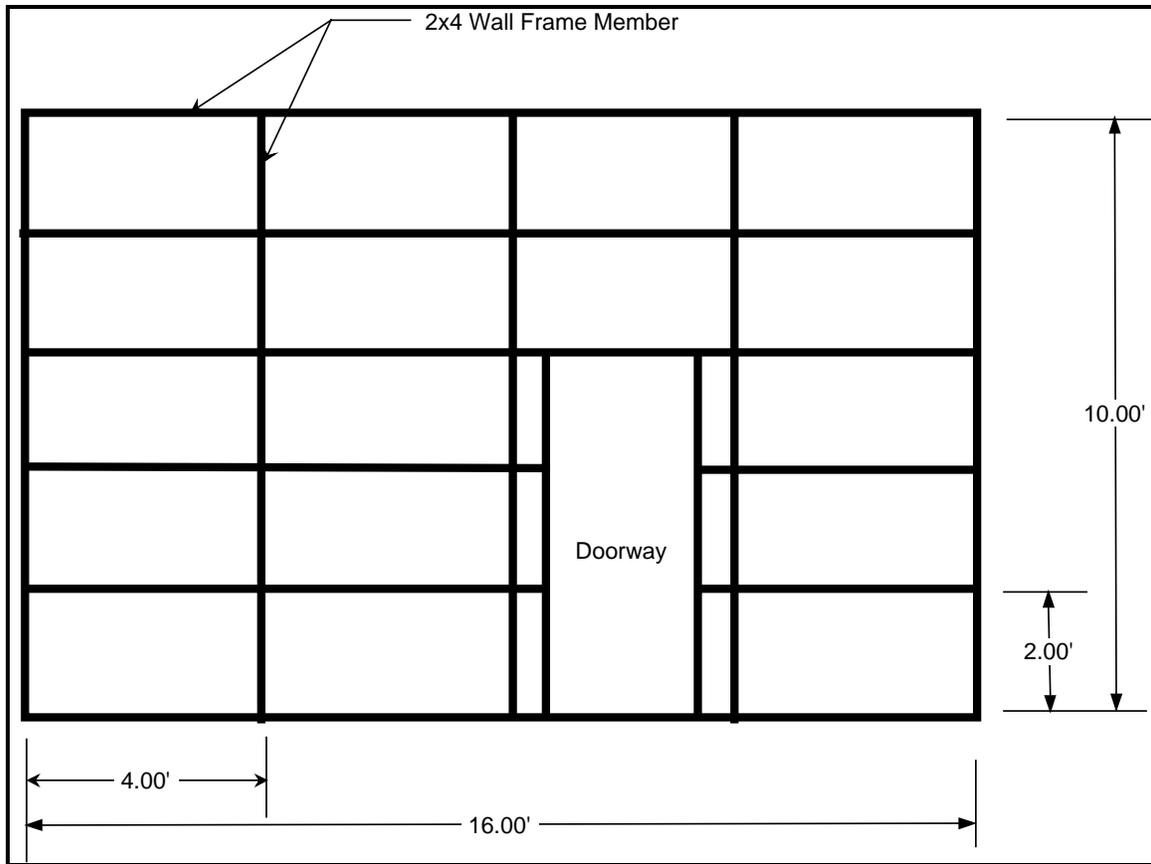


Figure 2.8: View of east wall

Figure 2.8 shows a 3 foot wide by 5.7 foot high doorway for access to the interior portion of the room. This doorway is filled with an insulation pillow and has three panels that cover it on the inside of the room. Access is gained by removing the plywood backing, then the styrofoam back and then the insulation pillow.

The thickness of each wall is 18 inches. This is created by setting two of the wall frames over the insulation border of the floor. This way the weight of the walls is concentrated on the leg supports of the room. The bottom of the walls are set 2 inches above the floor and back 2 inches from the floor edge to allow for the panels, which are 2 inches thick. The two wall frames are connected with 0.25 inch masonite that is glued and nailed to the 2x4's. The gap between the two frames is filled with two pieces of 0.75 inch thick styrofoam. The ends of the frames and the tops of the frames are fastened together with 0.5

inch plywood. The bottoms of the frames set on 0.25 inch masonite, which in turn is attached to the leg supports. Figure 2.9 shows a typical cross section of a wall.

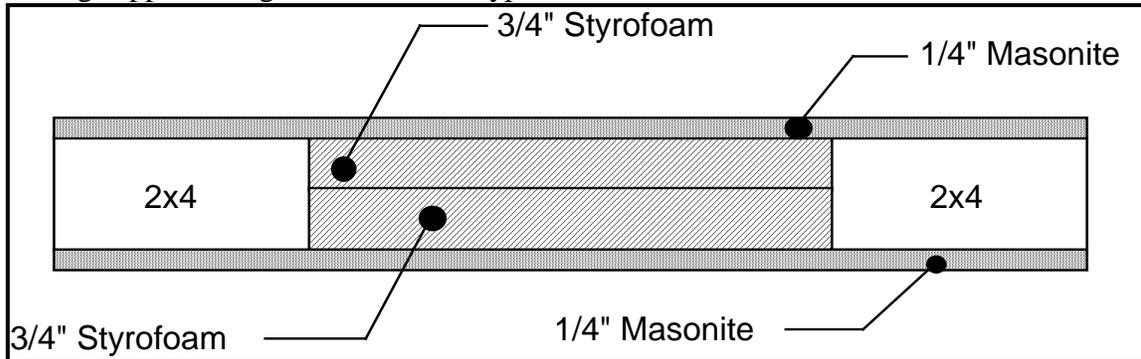


Figure 2.9: Typical wall cross section looking from side or from top of wall

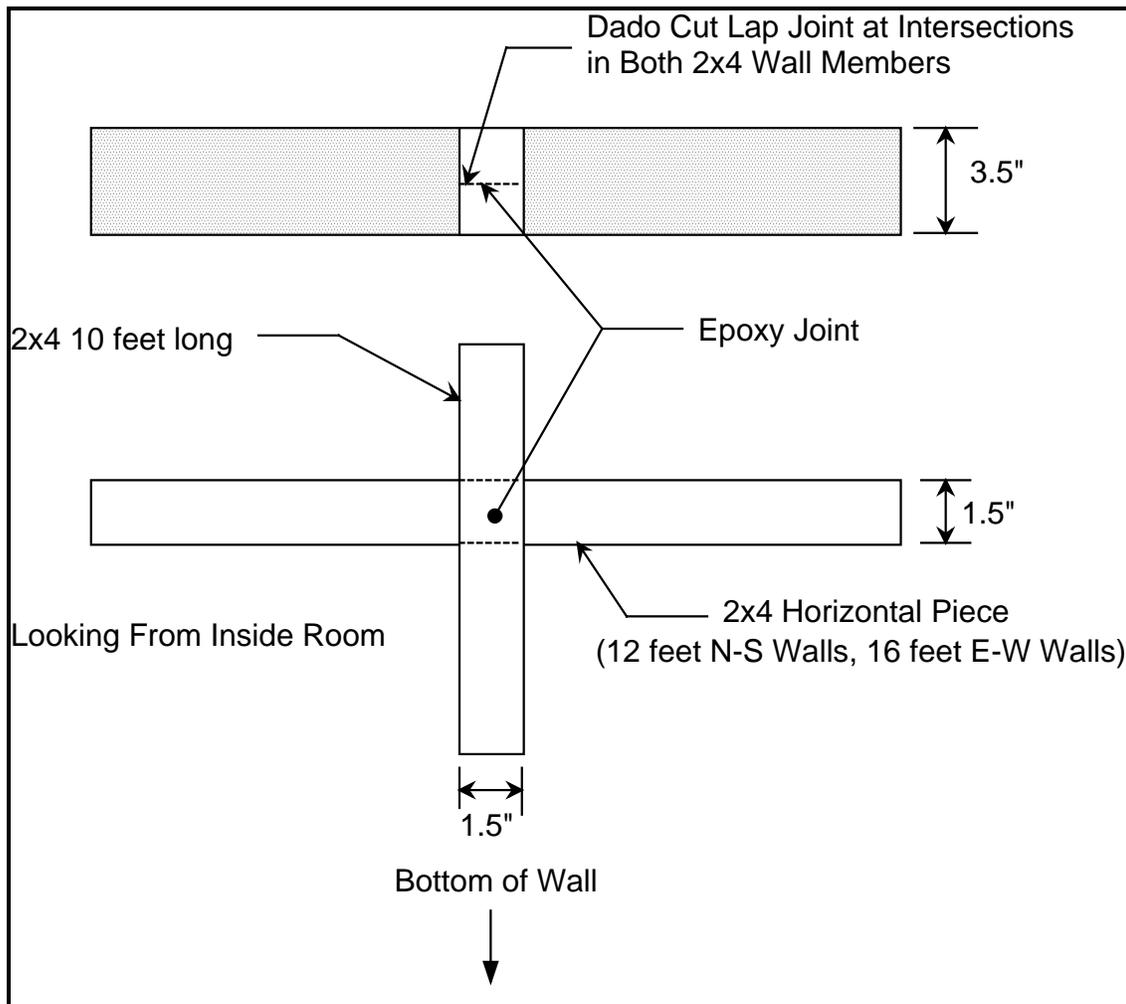


Figure 2.10: Wall joints detail

The corners of the walls match up as shown in Figure 2.11 so that the panels can be installed on the face of the walls. The corner space between the walls runs the full exterior height of the experimental room, and this space is filled with R-19 insulation from the bottom of the floor to the top of the ceiling. The outside frames of the walls are attached to each corner of the insulation border of the floor with 0.5 inch plywood. Figure 2.11 shows the corner details viewed from the top of the experimental room.

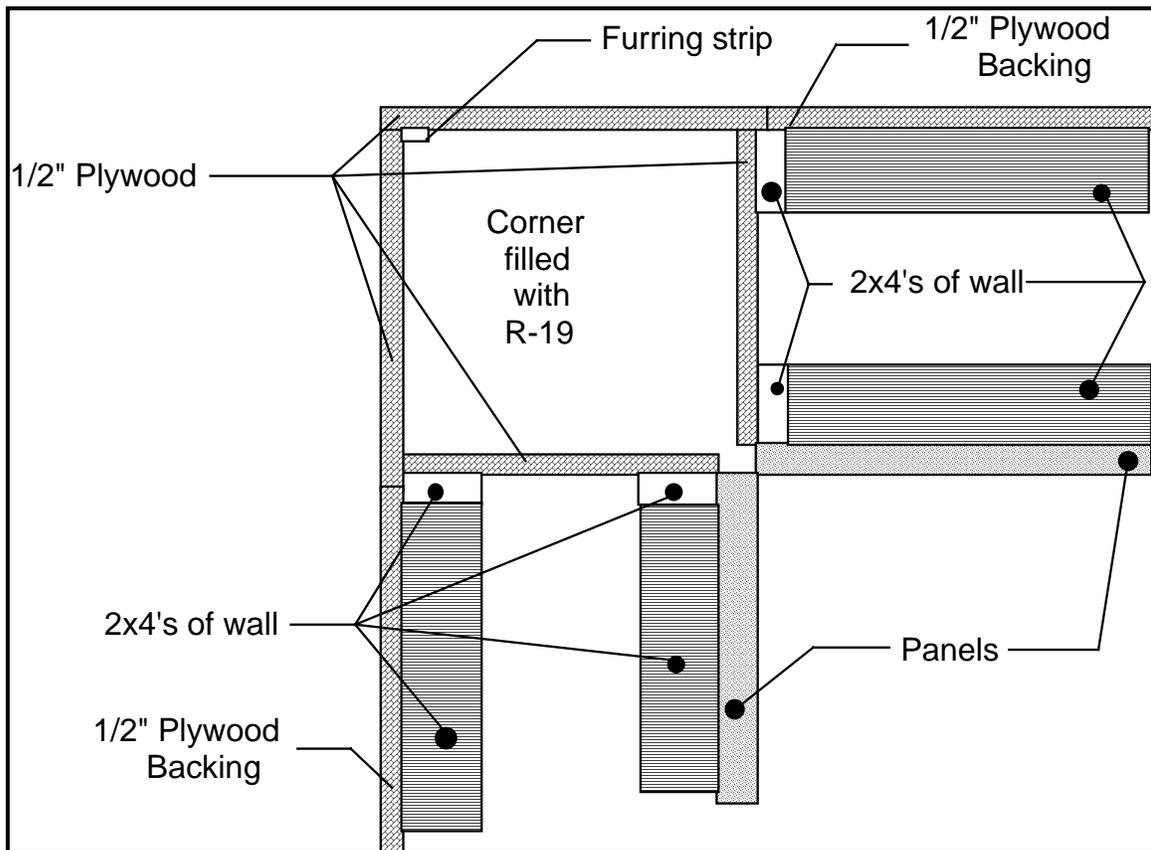


Figure 2.11: Corner details of the experimental room

2.1.3 The Ceiling of the Experimental Room

The ceiling of the experimental room is, in essence, the floor sitting on top of the walls. The 2 inch gap that is necessary for the panels is created by attaching 2 inch by 4 inch studs to the 0.5 inch plywood that is on top of the walls. Joists constructed from two 2 inch

by 8 inch planks glued together on their edges span across the 12 foot distance. These joists are spaced at 4 foot o.c. Two 2 inch by 8 inch planks that are glued together on edge are cut to 46.5 inch lengths to fit between the joists and are placed on 2 foot centers. This frame work creates twenty four "honeycomb" cells in the ceiling.

An insulation border surrounds the ceiling and is filled with R-19 fiberglass insulation. This border is sealed on the top of the room with 0.5 inch plywood.

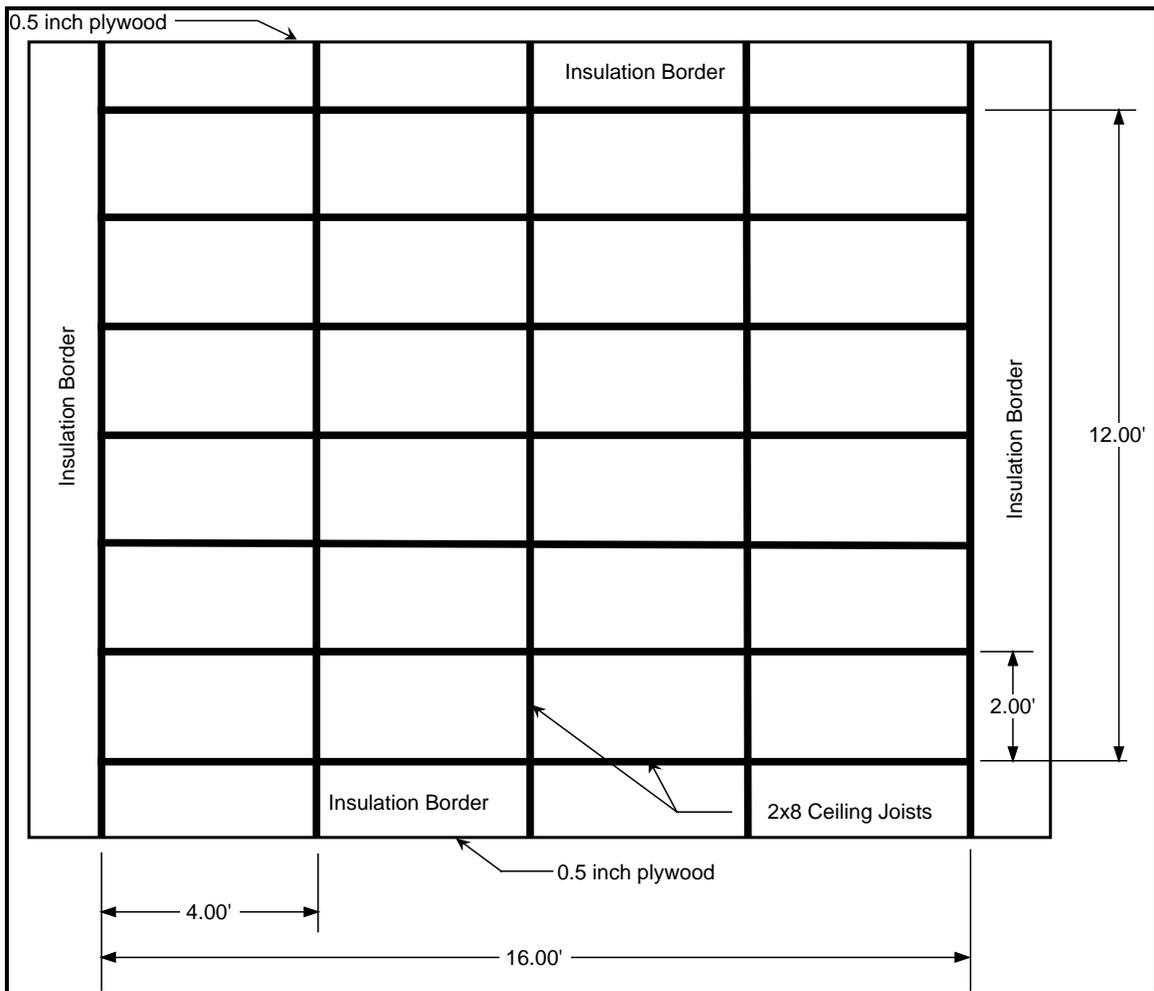


Figure 2.12: Ceiling plan view

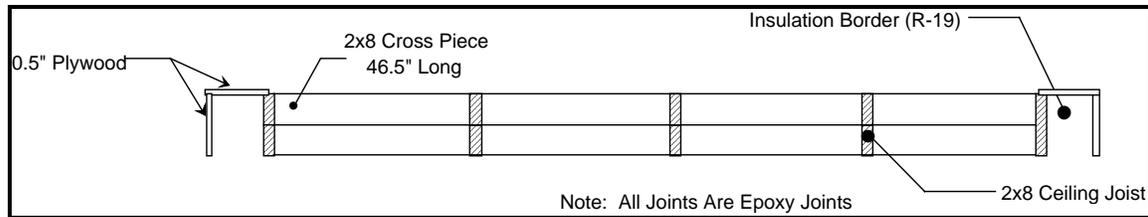


Figure 2.13: Ceiling profile view

2.2 The Guard Space

This section describes the concept of the guard space which is yet to be completed. Currently the north and east walls have been constructed, but have not been insulated. The south and west walls are completed. Since the walls are not finished, the heating system is non-existent. When future researchers wish to finish out the guard space, then this section will provide them with the information and ideas that were used to construct the guard space to its present condition.

The guard space is a 21 foot by 27 foot by 22 foot enclosure in which the experimental room sits. The south and west walls of the guard space utilize the existing concrete walls of the laboratory. These two walls are covered with 0.75 inch aluminum foil faced styrofoam with a 1.5 inch air gap between the concrete walls and the styrofoam.

The north and east walls of the guard space are constructed from 2 inch by 4 inch studs. The north wall has a 4 foot by 7 foot doorway to provide access to the guard space from the laboratory. The east wall has a 4 foot by 12 foot door across from the experimental room's door. This door provides access to the guard space and to the room for the movement of equipment into and out of the room. The two stud walls have provisions for R-11 fiberglass insulation between the studs, and both sides of the walls are to be covered with plastic. The ceiling of the guard space is the ceiling of the laboratory. Aluminum foil faced styrofoam should be glued to the ceiling for insulation purposes.

The guard space will be a temperature controlled environment. A possibility for the heating system is as follows. A fan is used to pull the air down from the top of the guard space through a header duct to the floor of the guard space. The air is passed through the fan

and exhausted through a duct on the floor and underneath the room. Two electric heaters placed on the floor of the guard space heats the air as it is exhausted from the fan. This is done to create a uniform 100°F temperature distribution throughout the guard space and minimize conduction losses from the wire resistance panel surfaces inside the experimental room.

2.3 Surface Heat Balance

The fundamental purpose of the experimental room is to study convective heat transfer from or to a surface. The convective heat transfer coefficient, or film coefficient, h , needs to be determined over a range of forced and free convection conditions. Figure 2.14 shows the heat balance for a typical surface of the experimental room.

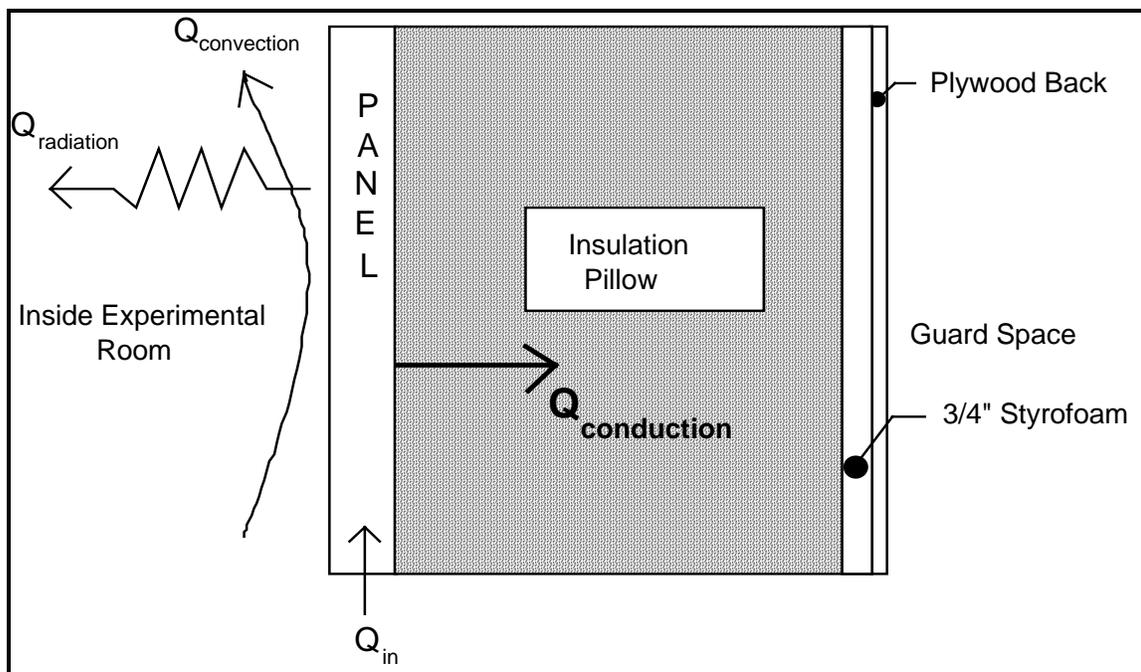


Figure 2.14: Surface heat balance for a typical surface

The surface heat balance equation is:

$$Q_{in} = Q_{convection} + Q_{radiation} + Q_{conduction} \quad (2.1)$$

$Q_{\text{conduction}}$ is minimized in two ways. The first way is by utilizing a high - thermal - resistance insulation. The second way is to keep the guard space temperature within a few degrees Fahrenheit of the panel surface temperature, thus minimizing back losses through the insulation.

If $Q_{\text{conduction}}$ is negligible, the surface heat balance equation reduces to:

$$Q_{\text{in}} = Q_{\text{convection}} + Q_{\text{radiation}} \quad (2.2)$$

$Q_{\text{convection}}$ and $Q_{\text{radiation}}$ are functions of the panel surface temperature and of the room conditions. The measurement and control of the panel surface temperature are of utmost importance since the two fundamental heat transfer components that have to be determined rely on this temperature.

The surface heat balance equation can be simplified further by minimizing $Q_{\text{radiation}}$. By covering the surface of the panel with a material (such as aluminum foil) with a low emissivity, the radiative heat transfer component is minimized. By operating at isothermal conditions, $Q_{\text{radiation}}$ can also be reduced. However, $Q_{\text{radiation}}$ will still have to be estimated since an isothermal condition cannot be met for all experiments. Also aluminum foil is not a good backdrop for flow visualization experiments; however, by implementing a removable panel scheme, the panel can be removed and the aluminum foil eliminated for a flow visualization experiment. By accurately measuring the power input to the wire resistance panel, Q_{in} may be accurately determined.

2.4 Panel Surface Temperature

The panel surface temperature is important in determining the heat transfer due to convection and radiation inside the experimental room. This temperature has to remain constant over the panel area. The convective heat transfer equation is:

$$Q_{\text{convective}} = hA(T_{\text{panel surface}} - T_{\text{air}}) \quad (2.3)$$

where: $Q_{\text{convective}}$ = the convective heat transfer (Btu/hr)
 h = the convective heat transfer coefficient (Btu/hr-ft²-°F)
 A = the area of the panel (ft²)

$T_{\text{panel surface}}$ = the temperature on the surface of the panel (°F)
 T_{air} = the temperature of the air at the outlet (°F)

The radiative heat transfer equation from surface 1 to all other surfaces j is:

$$Q_{\text{radiation}} = \sum_j A F_{1-j} \sigma (T_{1, \text{panel surface}}^4 - T_{j, \text{panel surface}}^4) \quad (2.4)$$

where: $Q_{\text{radiation}}$ = the radiative heat transfer (Btu/hr)

A = the area of the panel (ft²)

F_{1-j} = the grey interchange factor

σ = the Stefan - Boltzmann constant

$T_{1, \text{panel surface}}$ = temperature of panel surface 1 (°F)

$T_{j, \text{panel surface}}$ = temperature of panel surface j (°F)

Each panel is individually controlled to a specified surface temperature by a personal computer that receives data from a datalogger that reads the panel surface temperature with a thermocouple.

2.5 Unit Resistance for the Elements of the Experimental Room

The electrical resistance equivalent of the unit resistance of the floor is shown in

Figure 2.15.

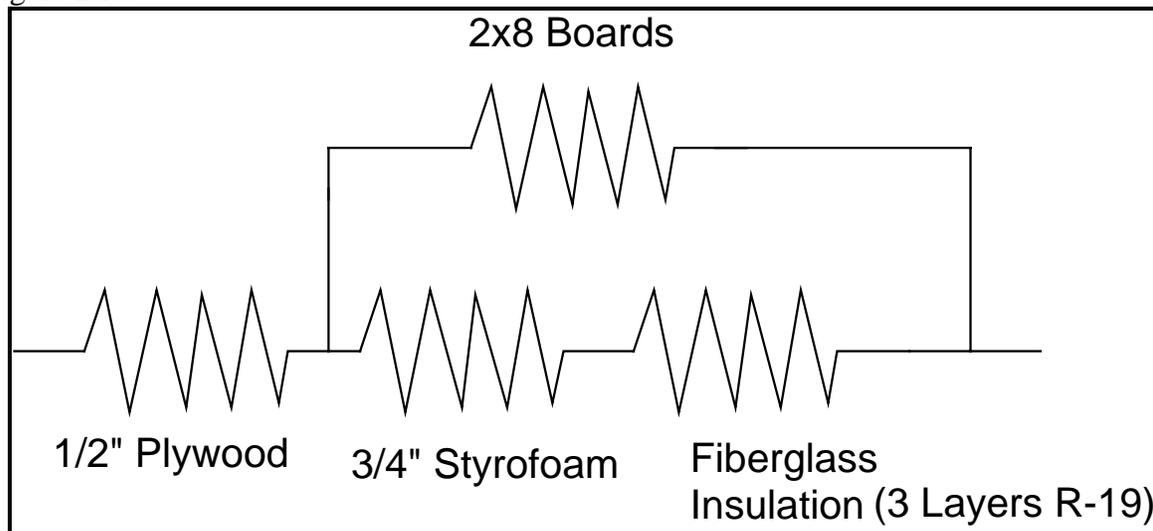


Figure 2.15: Electrical resistance equivalent of the floor's unit resistance

The resistance of the floor is calculated to be 46.7 hr-°F-ft²/Btu. Since the ceiling is the same construction as the floor, it has the same resistance. If the temperature difference between the panel surface and the guard space is 2°F, then the conductive loss from the panel

through the floor and the ceiling is 0.34 Btu/hr for each panel, or 8.2 Btu/hr for each surface. A total heat input of 700 Btu/hr for the floor and 210 Btu/hr for the ceiling is expected, so the conductive losses will not be significant in the overall heat balance on these two surfaces.

The equivalent resistance diagram for the walls is shown in Figure 2.16.

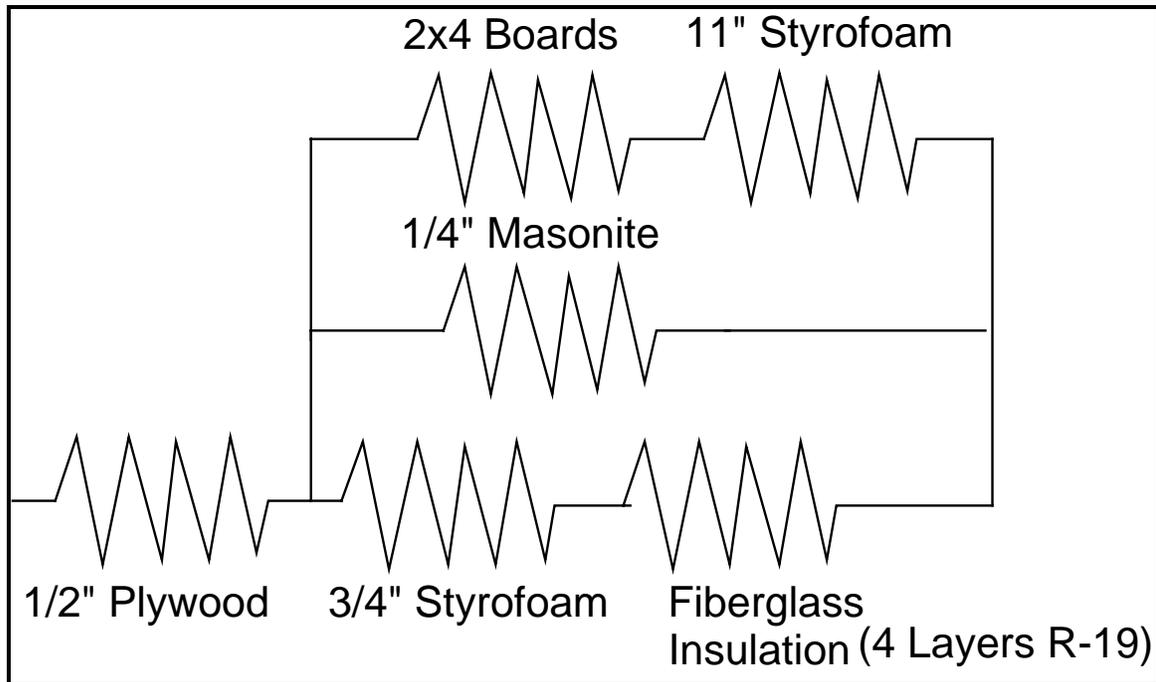


Figure 2.16: Electrical resistance equivalent of the wall's unit resistance

The unit resistance value for the walls is calculated to be $66.7 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$. With a 2°F temperature difference between the panel surface and the guard space, the conductive losses through the walls are 0.24 Btu/hr per panel, or 3.6 Btu/hr for the north and south walls and 4.8 Btu/hr for the east and west walls. A total heat input of 175 Btu/hr is expected for the walls, so the conductive losses will not be significant in the overall heat balance.

2.6 Panel Heating

This section discusses the design of the heated panel and the estimate of the conditions that will be encountered during an experiment.

Heat from the panel is generated by embedding nickel-chromium (Ni-Cr) resistance wire in a plastering material. Figure 2.17 shows a typical cross section for a panel.

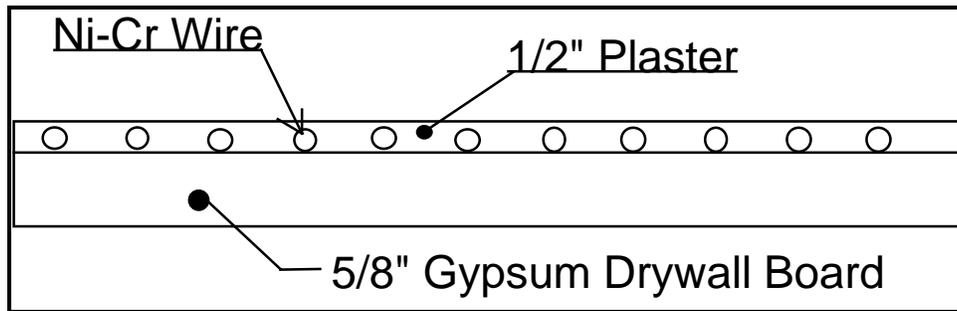


Figure 2.17: Typical cross section for a panel

A FORTRAN program (see Appendix A) was written to predict the panel surface temperature for a 0.5 inch thick gypsum plaster embedding material with a thermal conductivity of $5.6 \text{ Btu-in/ft}^2\text{-hr-}^\circ\text{F}$. A wire temperature and a surface type (ceiling, wall or floor) is inputted and the program calculates the surface temperature using a finite difference method for predicting temperatures. Figure 2.18 shows the boundary conditions used in the program for the finite difference method. The bottom and two sides are considered symmetry planes and are adiabatic surfaces. The top surface is exposed to the air and has to utilize an equation that has the film coefficient as a term to predict the surface temperature. The air temperature at a distance also has to be estimated to predict the surface temperature of the panel and for these cases, the air temperature at an infinite distance from the panel is assumed to be 75° F .

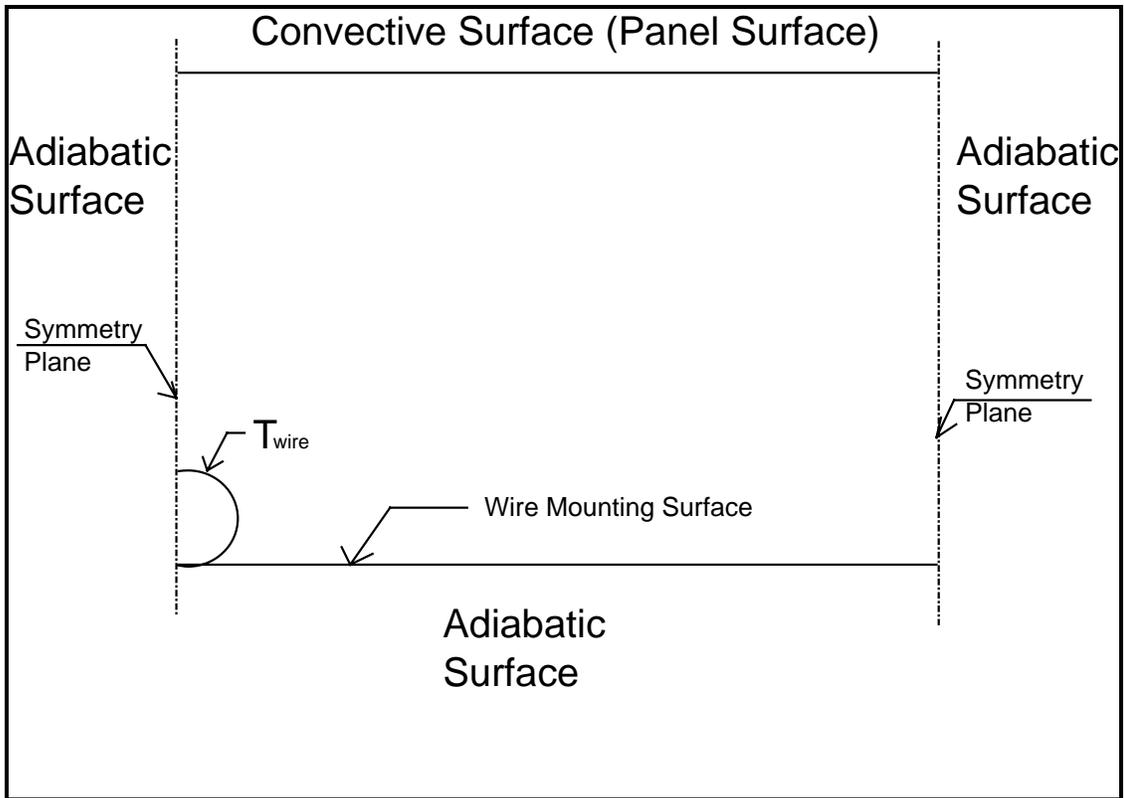


Figure 2.18: Boundary conditions for calculating the panel surface temperature

The temperature profile across the panel is sinusoidal, and it is desired to minimize the ΔT as much as possible so that a uniform temperature profile is seen across the panel.

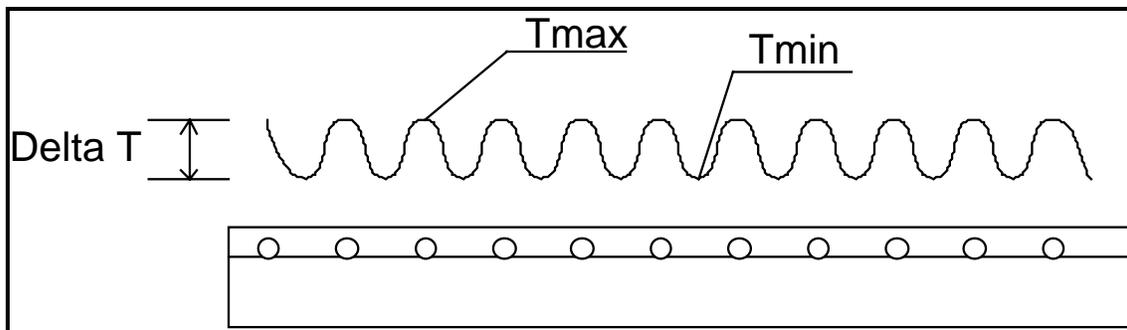


Figure 2.19: Temperature profile for a panel

Ultimately, a high thermally conductive epoxy would result in the best minimization of ΔT ; however, due to a limited budget for this project, an epoxy panel cannot be developed at this time. Table 2.1 shows the maximum temperature, minimum temperature and delta

temperature for the panel surface at different wire temperatures when 0.5 inch thick plaster is used as the embedding material.

Table 2.1: Panel Surface Temperatures(°F) at Different Wire Temperatures(°F)

	Ceiling			Wall			Floor		
Wire Temp	Max. Temp.	Min. Temp.	Delta Temp.	Max. Temp.	Min. Temp.	Delta Temp.	Max. Temp.	Min. Temp.	Delta Temp.
140°F	85.3°F	81°F	4.3°F	87.6°F	82.8°F	4.8°F	87.4°F	82.6°F	4.8°F
145°F	87.8°F	83.3°F	4.5°F	91.8°F	86.7°F	4.9°F	91.3°F	86.2°F	5.1°F
150°F	90.3°F	85.6°F	4.7°F	96.3°F	91.1°F	5.2°F	95.5°F	90.3°F	5.2°F
155°F	92.9°F	87.9°F	5.0°F	100°F	95.2°F	4.8°F	99.5°F	94.3°F	5.2°F
160°F	95.4°F	90.3°F	5.1°F						
165°F	97.9°F	92.6°F	5.3°F						
170°F	100.5°F	95.0°F	5.5°F						

It is desired to have a maximum temperature of 100°F on the panel surface. This is so that the heat pump that is used as the chiller for the ventilation system operates more efficiently. Once the surface temperature of 100°F was reached, calculations were terminated for that surface.

The Δt between the minimum and maximum temperature could be further reduced by covering the panel with aluminum foil. This would also help reduce the radiative component of the total heat transfer that is occurring on the panel. At this time, however, there is no plan to cover the panels with aluminum foil due to budget constraints and the amount of aluminum foil required.

Table 2.1 shows that on the ceiling a wire temperature of 170°F is required. The wall and floor require a wire temperature of 155°F. Using values for the convective heat transfer coefficient obtained by experimental work performed by Spitler at UIUC (Spitler, 1990), the convective heat transfer for a panel can be determined using the equation

$$Q_{\text{convective}} = hA(T_{\text{panel surface}} - T_{\text{air}}) \quad (2.5)$$

where: $Q_{\text{convective}}$ = the convective heat transfer (Btu/hr)
 h = the convective heat transfer coefficient (Btu/hr-ft²-°F)
 A = the area of the panel (ft²)
 $T_{\text{panel surface}}$ = the temperature on the surface of the panel (°F)
 T_{air} = the temperature of the air at the outlet(°F)

The temperature of the air is measured at the outlet and this measurement is used for the calculation of the convective heat transfer. The temperature at this point is chosen as a reference temperature due to previous research results, especially at low volumetric flow rates (Spitler, 1990).

Once the convective heat transfer has been estimated for each surface, the resistance required can be determined by using the equation

$$R_o = \frac{V^2}{Q} \{1 + \alpha(T_{\text{wire}} - T_o)\} \quad (2.6)$$

where: R_o = the wire resistance (ohm)
 V = the line voltage (volts)
 Q = the convective heat transfer for the panel (Btu/hr)
 α = the coefficient of thermal resistance of the wire (ohms/ohm/°F)
 T_{wire} = the wire temperature (°F)
 T_o = a reference temperature (77°F)

The convective portion of the total heat transfer is used in this equation since the facility had minimized the conduction losses and the radiation losses. Therefore, the total heat transfer in the room could be considered all convective heat transfer.

The following resistances are obtained and the current is found by dividing the line voltage by the resistance of the panel.

Table 2.2: Wire Resistance and Current for the Different Surfaces

Surface	Resistance (Ohms)	Current (Amps)
Ceiling	20.5	5.4
Wall	82.1	1.3
Floor	68.4	1.6

The wire spacing on the panel is at 1 inch. It takes 100 feet of wire to cover a 2 foot by 4 foot panel. The above resistance values are divided by 100 to obtain the resistance per foot value and these values are compared to numbers provided by the manufacturer. This determines the diameter of wire to use on each panel.

Table 2.3: Wire Diameter for the Different Surfaces

Surface	Diameter (inch)	A.W.G.
Ceiling	0.05707	15
Wall	0.02846	21
Floor	0.03196	20

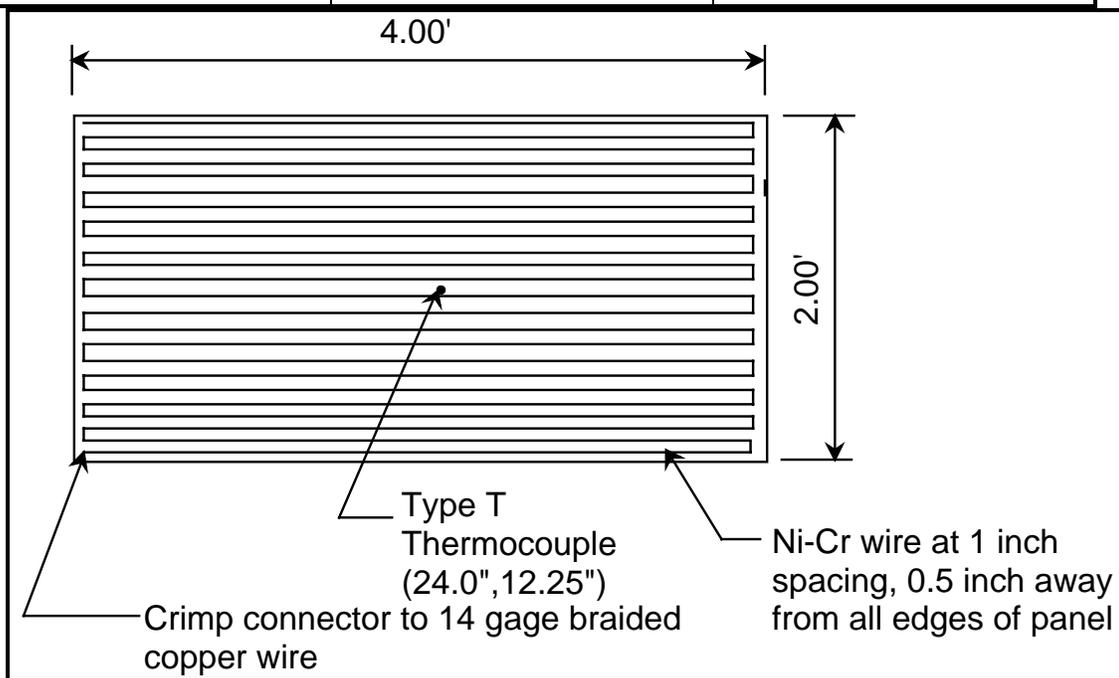


Figure 2.20: Panel plan view

2.7 Control of the Heated Panels

A control program (Appendix C) which uses a “pattern” control method (Fisher, 1989) is used to maintain the panel surface temperatures at 105°F. The “pattern” control method operates as follows:

1. The room is brought up to steady-state using a simple on/off control algorithm.
2. All panels are turned off for 10 cycles

3. All panels are turned on for 1 cycle, then off for 10 cycles. The temperature responses are recorded.

4. Step 3 is repeated with 2 cycles on and then with 3 cycles on.

5. The results from steps 3 and 4 are converted into three parameters:

DELTA1 is the maximum change in panel surface temperature due to 1 on cycle

DELTA2 is the maximum change in panel surface temperature due to 2 on cycles

DELTA3 is the maximum change in panel surface temperature due to 3 on cycles.

6. The control program switches into “pattern control” mode.

7. Surface temperatures are measured. Then, DELTA1, DELTA2 and DELTA3 are used to select an on/off pattern to control each panel. The on/off pattern that will result in the surface temperature staying closest to the setpoint is selected.

8. Step 7 is repeated until the experiment is finished.

Switching of the panels is done with solid state relays that receive either a high signal (5 VDC) or a low signal (0 VDC) from a Metrabyte digital i/o board located in the computer.

2.8 Experimental Configuration

For the initial shakedown tests of the facility, three experiments were performed, all at different flow rates and without the guard space being operational. The inlet was created in the south wall, midway up the wall, by attaching a flexible supply air duct to the back of a “honeycomb” cell. The other half of the cell was insulated. An inlet of 4 square feet was then formed (Figure 2.21). The outlet was formed by attaching a flexible return air duct to the back of the north wall’s bottom cell adjacent to the east wall “honeycomb” cell. The other half of this cell was also insulated to form an outlet of 4 square feet (Figure 2.22). Twelve heated panels were placed on the west wall, covering the first three levels from the floor. Ideally, more panels would have been used, but time constraints precluded the installation of

additional panels. The rest of the surfaces were covered with “passive” panels. The floor’s passive panel was made out of 0.5 inch plywood mounted on 2x4’s so that a person could walk around inside the room. Figure 2.23 shows the construction of the floor for the experiments. The walls and ceiling were covered with two layers of 0.75 inch aluminum faced styrofoam panels with a 0.5 inch air gap between the two panels. Figure 2.24 shows the wall passive panel construction. The ceiling is of similar construction.

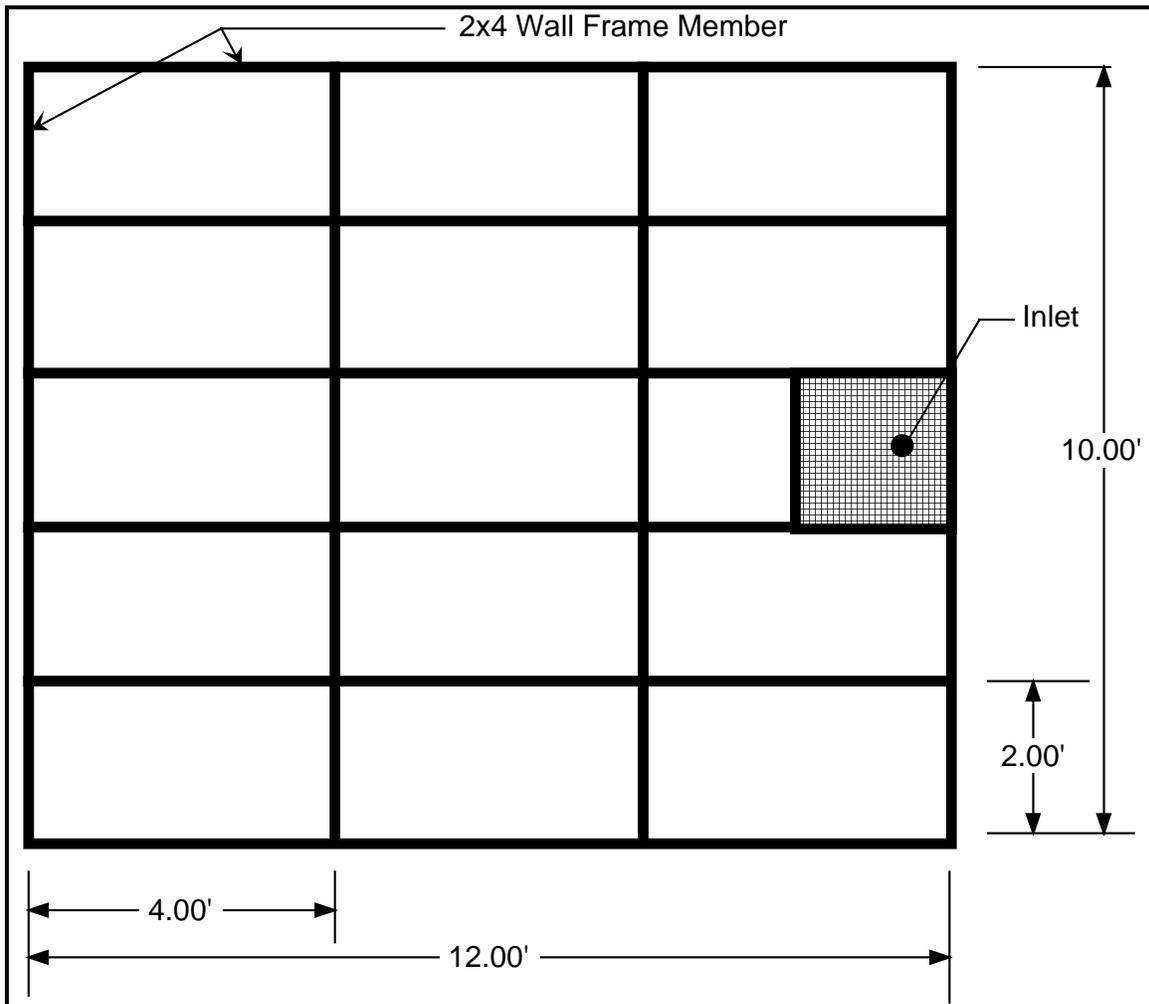


Figure 2.21: Air inlet location in south wall of experimental room

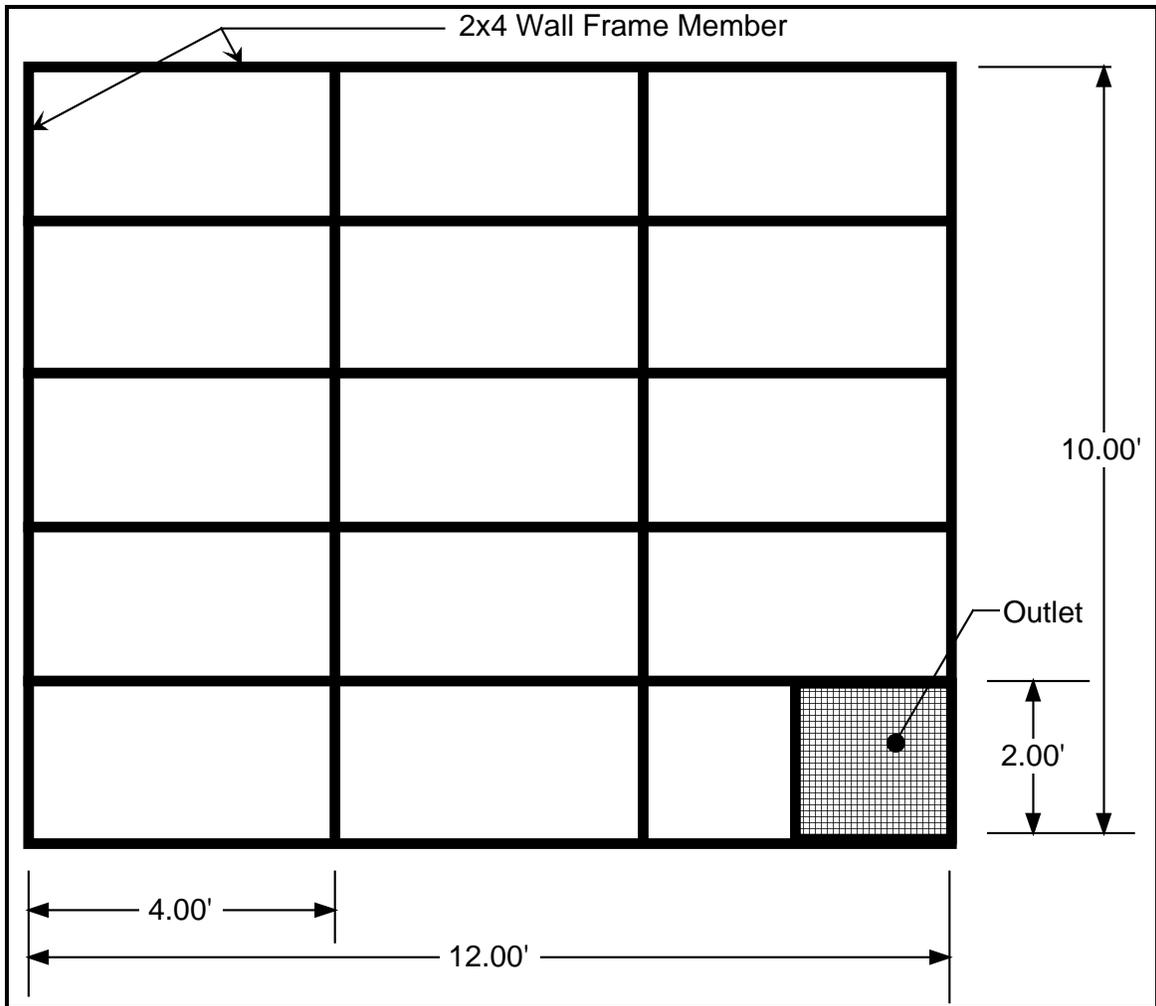


Figure 2.22: Air outlet location in north wall of experimental room

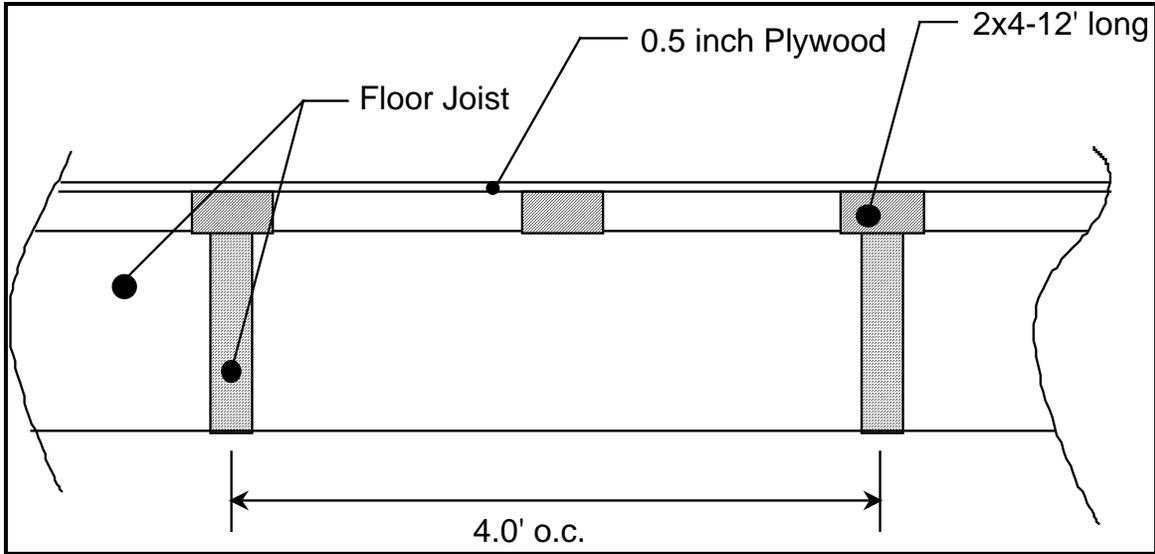


Figure 2.23: Floor passive panel cross section

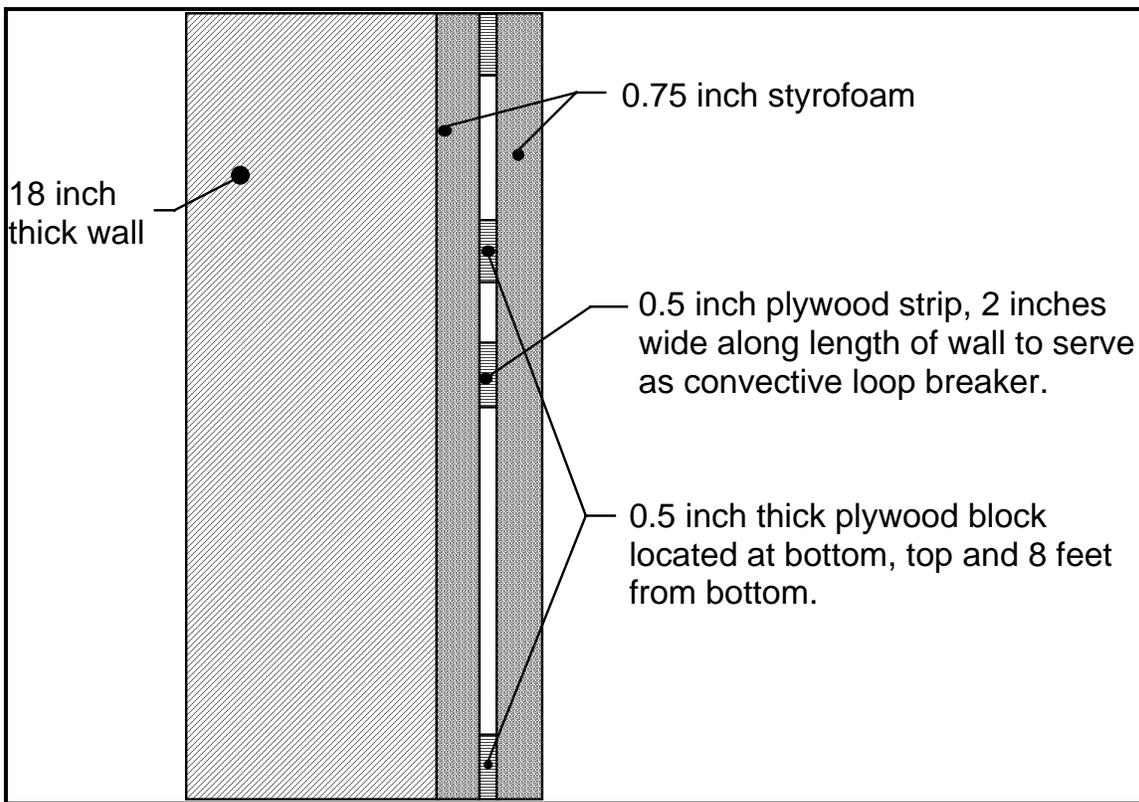


Figure 2.24: Wall passive panel cross section

Figure 2.25 shows the west wall configuration with a passive panel above the twelve heated panels.

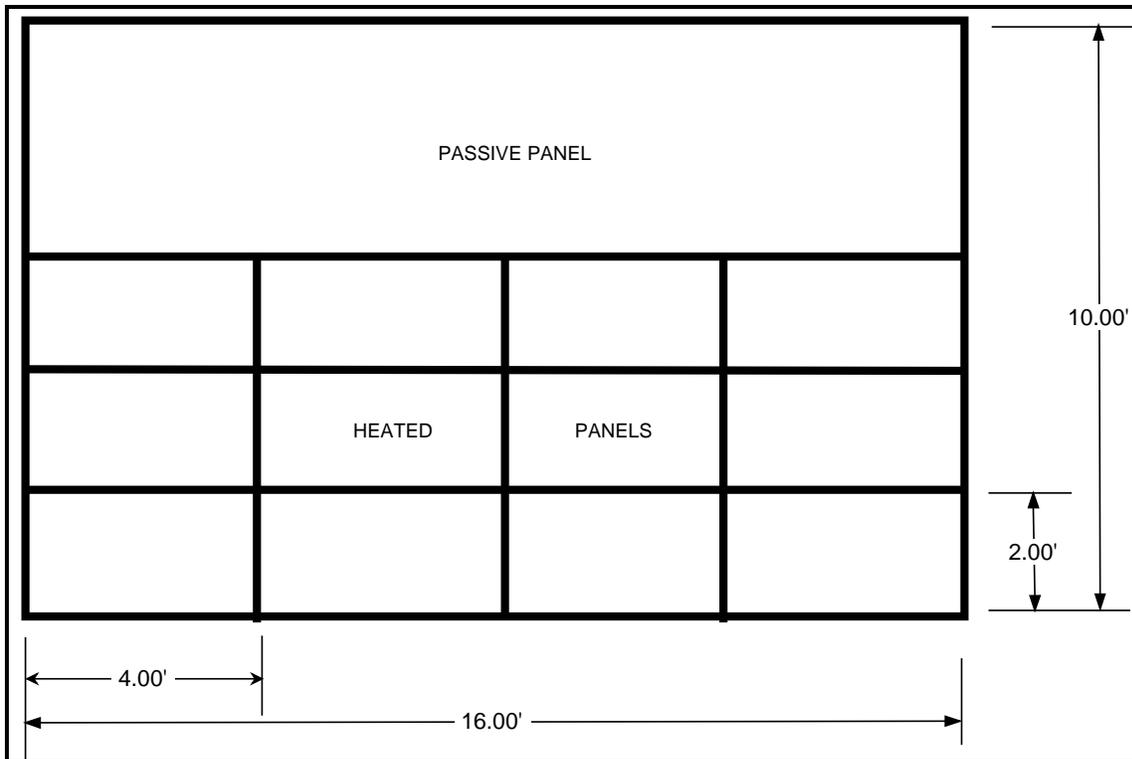


Figure 2.25: West wall configuration for the experiments

The guard space is not operational for these “shakedown” experiments. This will result in a “worse case” conduction loss from the panels on the west wall. If the temperature difference between the panel surface and the guard space is 25°F, then the expected loss is 3 Btu/hr, which is acceptable since the expected total heat input for the wall is 175 Btu/hr.

3 Analysis of Experimental Data

3.1 Data Analysis

This section describes the data analysis procedures. The data analysis procedures were implemented in a FORTRAN program, included in Appendix E. Each experiment generated two data files, one of which contains data recorded by the Fluke datalogger and one of which contains data recorded by hand..

A Fluke datalogger read the heated panel temperatures, the temperatures of the passive panel surfaces and the air temperatures at the measurement box, the room inlet and the room outlet. The control program, written in Basic (Appendix C), read the data from the Fluke and recorded it along with the time and the control bit for each heated panel. A scan was made and written to the hard drive approximately every 5 seconds. This data file of measured temperatures and control bits would contain approximately 900 scans, or 1.25 hours of experimental data.

The other datafile was an auxiliary data file that contained data recorded by hand during the experiment. This data included the line voltage, the static pressure in the measurement box, the pressure drop across the nozzle bank in the measurement box and which nozzles in the bank are open or closed.

3.2 Determining the Heated Panel Fluxes

3.2.1 Power Input to a Panel

The primary concern of the experiments is the determination of the convective heat transfer coefficient. In order to determine this coefficient, the power input to each panel has to be determined. The power input is simply:

$$Q_{\text{power}} = V^2 / R * 3.41 \quad (3.1)$$

Where: Q_{power} is the power input to a panel (Btu/hr)
 V is the line voltage
 R is the panel resistance (Ω)
3.41 is a conversion factor from watts to Btu/hr

There are times during the experiment, however, that a panel is off, so there is no current flowing to the panel. In order to determine the power input to a panel during an experiment, an average is determined by summing the power consumed by a panel during an experiment and dividing by the number of readings during the experiment.

$$\bar{Q}_{\text{power}, j} = \frac{\sum_{i=1}^N (Q_{\text{power}, j, i} * \text{control bit}_i)}{N} \quad (3.2)$$

Where $Q_{\text{power}, j, i}$ is the power input to the j^{th} panel at reading i (Btu/hr)
control bit $_i$ is 1 if the panel is on, 0 if the panel is off during the i^{th} reading
 N is the number of readings

3.2.2 Backlosses From a Panel

The backlosses, or conduction fluxes, from the panel are determined by taking the difference between the panel surface temperature and the temperature outside the room, and dividing this temperature difference by the thermal resistance of the wall. The surface temperature of the panel is used for this calculation instead of the temperature on the backside of the panel since no temperature measurements were made on the backside of the panel. This will be a safe assumption since the conduction fluxes are relatively small compared to the overall heat fluxes.

$$Q_{\text{cond}, j, i} = \frac{T_{\text{panel}, j, i} - T_{\text{outside}}}{R} \quad (3.3)$$

Where: $Q_{\text{cond}, j, i}$ is the backloss from panel j at reading i (Btu/hr)
 $T_{\text{panel}, j, i}$ is the temperature of panel j at reading i ($^{\circ}\text{F}$)
 T_{outside} is the temperature of the outside surface of the room ($^{\circ}\text{F}$)

R is the thermal resistance of the wall (hr-°F/Btu)

An average backloss for each panel for each experiment is determined by summing the backlosses from a panel during the experiment and dividing by the number of readings.

$$\bar{Q}_{\text{cond}, j} = \frac{\sum_{i=1}^N Q_{\text{cond}, j, i}}{N} \quad (3.4)$$

Where: $Q_{\text{cond}, j, i}$ is the backloss from panel j at reading i (Btu/hr)
N is the number of readings

3.2.3 Radiative Flux From a Panel

The radiative flux from the heated panels is calculated using Hottel's grey interchange method. The room is modeled simply as an 18 surface enclosure. Twelve of the surfaces are the heated panels with a surface emissivity of 0.9. The plywood floor is one surface and has a surface emissivity of 0.9. The north, south and east walls, ceiling and the passive panel on the west wall make up the other five surfaces, all with a surface emissivity of 0.25. The room air is considered a non-participating media (Spitler, 1990).

Direct view factors for all eighteen surface interactions and total grey interchange factors are determined using a FORTRAN program provided by Spitler (Spitler, 1990). Spitler's program incorporated Walton's (Walton, 1986) program for determining view factors and Pedersen's (Pedersen, 1989) program for computing grey interchange factors.

In order to use Hottel's method, black body emissive power must be determined using the Stefan-Boltzman law :

$$E = \sigma T_s^4 \quad (3.5)$$

Where: E is the black body emissive power (Btu/hr-ft²)
 σ is the Stefan-Boltzman constant (1.714*10⁻⁹ Btu/hr-ft²-R⁴)
 T_s is the surface temperature of each area (°F)

After the black body emissive powers are calculated for each panel at each temperature reading, the radiative flux for a panel can be determined by summing the individual net fluxes to each of the other surfaces.

$$Q_{i-j} = \sum_{i=1}^n \sum_{j=i}^n S_i S_j (E_i - E_j) \quad (3.6)$$

Where: $Q_{i,j}$ is the radiative flux from panel i to panel j (Btu/hr)
 $S_i S_j$ is the total grey interchange area from panel i to panel j (ft²)
 E_i, E_j are the black body emissivities of panel i and panel j (Btu/hr-ft²)

An average radiative flux for each panel is determined by summing the calculated radiative flux from each reading for each individual panel and dividing by the number of readings.

$$\bar{Q}_{rad, j} = \frac{\sum_{i=1}^N Q_{rad, j, i} / A_j}{N} \quad (3.7)$$

Where: $Q_{rad, j, i}$ is the radiation flux from panel j at reading i (Btu/hr-ft²)
 A_j is the area of panel j (ft²)
 N is the number of readings

3.2.4 Convective Flux for a Panel

The experimental average convective heat transfer can now be determined for each panel by using the following equation:

$$\bar{Q}_{conv, j} = \bar{Q}_{power, j} - \bar{Q}_{rad, j} - \bar{Q}_{cond, j} \quad (3.8)$$

3.3 Determination of the Convection Coefficient

Once the average convective heat transfer has been determined for each heated panel, the average convection coefficient, or film coefficient, can be determined by dividing the convective heat transfer value by a temperature difference. The temperature difference is the difference between the panel surface temperature, which is averaged over the experiment, and a reference temperature.

$$\bar{h}_j = \frac{\bar{Q}_{\text{conv}, j}}{A_j * (\bar{T}_j - T_{\text{ref}})} \quad (3.9)$$

For these experiments the reference temperature is selected to be the time average air temperature at the room outlet (Spitler, 1990).

3.4 Characteristic Parameters

There are certain characteristic parameters that are of interest for each experiment performed. Discussion of the calculation of each of these parameters is now presented.

3.4.1 Volumetric Flow Rate

Determination of the volumetric flow rate of air entering the room is of particular importance since it will lead to the calculation of several more characteristic parameters. The volumetric flow rate is determined by measuring the pressure drop across a nozzle bank and the static pressure entering the nozzle bank with two piezometer rings of pressure taps in the measurement box that are connected to an inclined manometer. Using the equations specified by ASHRAE Standard 51-1985, "Laboratory Methods of Testing Fans for Rating", the volumetric flow rate (VFR) is determined in units of cubic feet per minute (cfm). Further discussion of this measurement is reported by Ferguson (1995).

3.4.2 Inlet Velocity

Determination of several other parameters is dependent upon the velocity of the air entering the room. The inlet velocity, in feet per minute or fpm, is determined by dividing the volumetric flow rate by the area of the inlet.

$$U_o = \frac{VFR}{A_{inlet}} \quad (3.10)$$

3.4.3 Air Changes per Hour

Air changes per hour (ACH) for an experiment is calculated using the following equation.

$$ACH = \frac{VFR * 60}{VOL_{room}} \quad (3.11)$$

3.4.4 Archimedes Number

The Archimedes number is used to predict the trajectory of the inlet jet. The general form of the Archimedes number is:

$$Ar = \frac{\beta g L_c \Delta T_o}{U_o^2} \quad (3.12)$$

Where: β is the coefficient of thermal expansion (1/R)
 g is gravitational acceleration (32.2 ft/s²)
 L_c is a characteristic length, in this case the length of the wall that the jet washes (ft)
 ΔT_o is the temperature difference between the room outlet and inlet (°F)
 U_o is the fluid velocity (ft/s)

The most likely use of the Archimedes number is to characterize the flow regime rather than correlate to the heat transfer coefficient. In the same regard, the relationship would also be dependent upon whether the inlet location is in a wall, in the ceiling or even in the floor. The Archimedes number can be considered as a ratio of buoyant forces to momentum flux, so the Archimedes number's significance upon inlet location is important

since buoyant forces will act differently on each different type of inlet location. For the sidewall case which was used for the experiments, a low Archimedes number, since the buoyant forces are small, should indicate that the jet from the inlet is capable of transversing the room and impinging on the wall opposite from the inlet. A high Archimedes number, since the buoyant forces are large, should indicate that the jet drops upon entering the room and settles to the floor. Intermediate values would suggest that the jet travels a certain distance into the room, then falls to the floor of the room. In relation to the film coefficient, a low Archimedes number would suggest that the natural convection component of the overall convection flux is larger than the forced convection component. The opposite would hold true for a high Archimedes number in that the forced convection component is larger than the natural convection component. It is difficult to determine what the upper and lower bounds of the Archimedes number are without using a flow visualization technique inside the room.

3.4.5 Grashof Number

The Grashof number characterizes the ratio of the buoyancy force to the viscous force.

Written in equation form:

$$\text{Gr} = \frac{\beta g (T_{\text{surf}} - T_{\text{fs}}) L^3}{\nu^2} \quad (3.13)$$

where: β is the coefficient of thermal expansion (1/R)
 g is gravitational acceleration (32.2 ft/s²)
 T_{surf} is the temperature of the surface (°F)
 T_{fs} is the fluid free stream temperature (°F)
 L is the height of the wall(ft)
 ν is the kinematic viscosity (ft²/s)

It is difficult to estimate the fluid free stream temperature for the three experiments performed since air temperature was measured at the room inlet and the room outlet. The

assumption will be used then that the free stream air temperature is the same as the average temperature of the ceiling, floor and non-active walls.

3.4.6 Prandtl Number

The Prandtl number is the ratio of the kinematic viscosity to the thermal diffusivity. This can be defined as the diffusion of momentum divided by the diffusion of heat. The Prandtl number can be used to estimate the ratio of the velocity boundary layer thickness to the thermal boundary layer thickness. For a Prandtl number greater than 1.0, the thermal boundary layer is thinner than the velocity boundary layer. For a Prandtl number equal to 1.0, the respective boundary layer thicknesses are the same. When the Prandtl number is less than 1.0, the velocity boundary layer is thinner than the thermal boundary layer.

3.4.7 Rayleigh Number

The Rayleigh number is the product of the Grashof number and the Prandtl number. The Rayleigh number is typically correlated to the Nusselt number with the form:

$$Nu = C Ra_L^n \quad (3.14)$$

where: Nu is the Nusselt number
C is an empirically determined constant
n is an empirically determined exponent (typically 1/3 for laminar flows;
1/4 for turbulent flows)

3.4.8 Nusselt Number

The Nusselt number is defined as the ratio of the temperature gradient at the wall to the overall temperature difference. Written in equation form, the Nusselt number is:

$$Nu = \frac{h L}{k} \quad (3.15)$$

where: h is the film coefficient (Btu/hr-ft²-°F)
L is the height of the wall (ft)
k is the thermal conductivity of the fluid (Btu/hr-ft-°F)

The Nusselt number is often correlated to the Rayleigh number. Some of the correlations developed relating Nusselt number to Rayleigh number will be investigated in Chapter Four.

3.4.9 Jet Momentum

The jet momentum flux of the air at the room inlet is determined by multiplying the mass flow rate of the air (lb_m/min) by the velocity of the air entering the room (fpm). The jet momentum number is non-dimensionalized (Spitler, 1990) by using the following equation:

$$J = \frac{\text{VFR} * U_o}{g * V_{\text{room}}} \quad (3.16)$$

Where: VFR is the volumetric flow rate (cfm)

U_o is the velocity of the air entering the room (fpm)

g is gravitational acceleration

V_{room} is the volume of the room (ft^3)

3.5 Experimental Uncertainties

The difference between a measured result and the true value of that quantity is error. By assigning a value to that error, an uncertainty is defined. The uncertainties in each individual measurement lead to uncertainties in the experiment.

3.5.1 Uncertainties in Fluke Datalogger Temperature Measurements

Sources of error that cause uncertainty in the temperature measurements are the type T thermocouple wire properties, the cold junction compensation in the Fluke datalogger and the emf voltage measurement of the thermocouple. The thermocouple wire has an uncertainty of $\pm 0.9^\circ\text{F}$. The estimated error for the cold junction compensation is $\pm 0.1^\circ\text{F}$. The voltage measurement of the thermocouple by the Fluke translates into a $\pm 0.9^\circ\text{F}$ error at the heated panel setpoint of 105°F . The error, then, in temperature measurements made with the Fluke is:

$$u_T = \sqrt{(0.9)^2 + (0.1)^2 + (0.9)^2} \approx \pm 1.3^\circ\text{F} \quad (3.17)$$

3.5.2 Uncertainties in Panel Power Measurement

The errors associated with the panel power measurement are the voltage fluctuations and the resistance of the panel changing as the Ni-Cr wire heats up. Other errors are attributed to the digital multimeter used to measure the panel resistance and the line voltage.

Line voltage is measured once at the end of the experiment when the hand collected auxiliary data is being recorded. It is estimated that the voltage changes by a maximum of ± 2 volts during an experiment. Uncertainty in the voltmeter is ± 0.1 volts. The change in resistance of the panel due to temperature changes is negligible. The uncertainty in the measurement of the panel resistance is $\pm 0.1\Omega$. This is approximately 0.1% when using 75Ω

as the typical panel resistance. The total uncertainty in the panel power can be approximated by:

$$u_p = \sqrt{\left(\frac{2(2)}{120}\right)^2 + \left(\frac{0.1}{75}\right)^2} \approx 3\% \quad (3.18)$$

3.5.3 Uncertainties in Volumetric Flow Rate Measurement

ASHRAE Standard 51-1985 is used to measure the flow rate of air supplied by the fan to the room. The appendix of the standard presents a detailed error analysis for the error in volumetric flow rate. The variables used in the equation are as follows, with values assigned to them in parentheses.

- u_c = Fractional error in nozzle discharge coefficient (0.012)
- u_A = Fractional error in nozzle area (0.005)
- u_N = Fractional variation in fan speed (0.01)
- u_f = Fractional error in pressure drop across the nozzle bank (0.1)
- u_p = Fractional error in static pressure at fan outlet (0.1)

Values for the fractional errors in nozzle properties are from the ASHRAE Standard and apply to nozzles built to the specifications. Pressure measurement errors are estimated based on the manometer having gradations of 0.2 inches. The fluid level fluctuates during an experiment and the pressure reading is estimated. It is assumed that the pressure reading is accurate within 0.1 inches. When substituting the values into the uncertainty equation:

$$u_{vfr} = \sqrt{u_c^2 + u_A^2 + u_N^2 + \frac{u_f^2}{2} + \frac{u_p^2}{2}} \quad (3.19)$$

an uncertainty in the volumetric flow rate of approximately $\pm 10\%$ is obtained.

3.6 Uncertainties in Derived Quantities

This section covers uncertainties in quantities calculated from the quantities measured during an experiment.

3.6.1 Uncertainties in the Calculation of the Air Heat Gain

An energy balance is performed on the room by comparing the power input to the panels to the air heat gain.

The air heat gain is given by:

$$Q_{\text{air}} = \rho(\text{VFR})C_p\Delta T \quad (3.20)$$

where: $\rho(\text{VFR})$ is the density of the air times the volumetric flow rate, which is equal to the mass flow rate of air entering the room (lb_m/hr)

C_p is the specific heat of air at constant pressure ($0.24 \text{ Btu}/\text{lb}_m\text{-}^\circ\text{F}$)

ΔT is the temperature difference between the room outlet and room inlet.

Assuming that density and C_p are determined with negligible error, then the error in the air heat gain can be stated in the form:

$$Q_{\text{air}} = \rho(\text{VFR})C_p \Delta T \pm \sqrt{(u_{\text{VFR}})^2 + (u_{\Delta T})^2} \quad (3.21)$$

u_{VFR} has already been determined to be approximately 10%. The error in ΔT is dependent upon the inlet and outlet temperature measurements. Using the $\pm 1.3^\circ\text{F}$ error in temperature measurements leads to the following equation:

$$Q_{\text{air}} = \rho(\text{VFR})C_p \Delta T \pm \sqrt{(0.1)^2 + \left(\frac{1.8}{\Delta T}\right)^2} \quad (3.22)$$

ΔT is the temperature difference between the room outlet temperature and the room inlet temperature. These two temperatures are averaged over the duration of the experiment, therefore ΔT is the experimental average temperature difference between the room outlet and the room inlet. For example, let the average ΔT for an experiment performed be equal to 10°F . When substituting this assumption into the equation above, the uncertainty is determined to be approximately 21%. Therefore the equation for the energy balance on the air is simply:

$$Q_{\text{air}} = \rho(\text{VFR})C_p \Delta T \pm 21\% \quad (3.23)$$

As shown in section 3.5.2, uncertainty in the total power input to the panels is assumed to be 3% since this uncertainty is dependent upon the measurement of the voltage and the panel resistance.

3.6.2 Uncertainties in the Calculation of Film Coefficients

The uncertainties in the film coefficient are dependent upon the uncertainties in the convective heat flux and the uncertainties in the temperature difference.

$$u_h = \sqrt{(u_{\text{conv}})^2 + (u_{\Delta T})^2} \quad (3.24)$$

where:
$$u_{\Delta T} = \frac{\sqrt{(u_{T, \text{panel}})^2 + (u_{T, \text{outlet}})^2}}{T_{\text{panel}} - T_{\text{outlet}}}$$

The uncertainties in the temperature difference are calculated in the same way the uncertainties in the temperature difference for the volumetric flow rate were calculated. We will assume that the temperature difference on average between the wall average temperature and the room outlet average temperature is 35°F. The uncertainties in the Fluke datalogger temperature measurements, ±1.3°F, will be used for the uncertainties in the temperatures measured. This results in $u_{\Delta T}$ of approximately 30% for typical cases.

The uncertainties in the convective flux are dependent upon the uncertainties in the power input, the backlosses and the radiation flux. The uncertainties in the backlosses are dependent upon the temperature difference between the panel temperature and the temperature outside the room. The temperature outside the room is not measured, and the

uncertainties in the temperature outside the room are assumed to be $\pm 5^\circ\text{F}$. This results in an uncertainty of 14% in the backlosses.

The uncertainty in the radiation is based on the uncertainty in the panel surface temperatures and the uncertainty in the value of surface emissivity. The assumption that uncertainty due to small errors in panel areas and view factors are negligible is made in order to simplify the analysis. The radiation from one panel to all other surfaces is given by:

$$Q_{\text{rad}, i - j} = \sigma \sum_{j=1}^{18} S_i S_j (T_i^4 - T_j^4) \quad (3.25)$$

This equation can be rewritten as:

$$Q_{\text{rad}, i - j} = \sigma \sum_{j=1}^{18} S_i S_j (T_i^4) - \sigma \sum_{j=1}^{18} S_i S_j (T_j^4) \quad (3.26)$$

If a constant, random uncertainty for each surface temperature is assumed, then the uncertainty in the second term on the right hand side is negligible compared to the first term on the right hand side. An uncertainty will be introduced into the first term.

An uncertainty of $\pm 0.9^\circ\text{F}$ exists in the panel temperature measurements according to section 3.5.1. An uncertainty in the temperature across the panel exists due to temperature variations, however, this uncertainty is impossible to calculate since only one thermocouple is used on each panel. This uncertainty will be ignored even though it probably plays a role in

the uncertainty of the radiation flux. A typical value of 565R is used for T_i . Therefore, the uncertainty in the radiation flux due to panel surface temperature is $\pm 0.9\%$.

The assumed surface emissivity of the panels is 0.9 ± 0.05 . Knowing that the $S_i S_j$ terms sum to the surface emissivity leads to an uncertainty in the radiation correction of:

$$u_\varepsilon = \frac{\pm 0.05}{0.9} = \pm 6\% \quad (3.27)$$

The total error in the radiation correction is approximately $\pm 6\%$.

The panel to room air radiation is neglected in this case. This assumption could be endangered if the room air is high in humidity since water vapor in the air can absorb and emit radiation. Spitler (1990) performed a “worse case” scenario for the panel to room air radiation. This scenario, when applied to the experiments, results in an uncertainty in the flux of 5%.

The uncertainty in the convective heat transfer can now be calculated knowing the uncertainties in the panel power, radiation and backlosses. These uncertainties are changed from percentages to numerical values based on fluxes. The uncertainties in the three fluxes are as follows for a given case:

Panel power input: $57.3 \text{ Btu/hr-ft}^2 \pm 1.7 \text{ Btu/hr-ft}^2$

Backlosses: $0.04 \text{ Btu/hr-ft}^2 \pm 0.006 \text{ Btu/hr-ft}^2$

Radiation: $3.43 \text{ Btu/hr-ft}^2 \pm 0.2 \text{ Btu/hr-ft}^2$

The above uncertainties are now added in quadrature to determine the uncertainty in the convective heat flux. This uncertainty is determined to be 1.7 Btu/hr-ft^2 ($\approx 3\%$). This value is substituted into equation 3.24 knowing that the uncertainty in the temperature difference has already been shown to be approximately 30%. Therefore, the uncertainty in the film coefficient is approximately $\pm 30\%$

$$h = h_{\text{exp}} \pm 30\% \quad (3.28)$$

where: h_{exp} is the experimentally determined film coefficient for the wall.

4 Experimental Results and Discussion

4.1 Overview

Three experiments were performed at 15, 20 and 25 Air Changes per Hour (ACH). The air heat gain for an experiment is compared to the total heat input for an experiment to see if an energy balance is obtained within the uncertainties determined in sections 3.5.2 and 3.6.1. The energy balance is of utmost importance. Without it, there is little confidence in the experimental results. If an energy balance cannot be achieved, then modifications will have to be made to the facility in order to reduce losses and bring the facility to a balance.

Another area of interest is the control of the panels. The panels have to be able to meet a desired set point and maintain that temperature within a reasonable range. Another question relates to the amount of time necessary to reach steady state conditions, or are steady state conditions ever achieved in the facility?

Finally as another check for the facility, the experimentally determined film coefficient will be compared to published correlations for the film coefficient. The correlations that are used for comparison do not apply to the conditions for which the tests were run, but might be thought of as limiting cases. Three of these correlations are for natural convection cases and can be considered as a lower bound to the experiments performed. The other two correlations that are used for comparison to the experimentally determined film coefficient are for forced convection cases and can be considered as an upper bound to the experiments performed.

4.2 Experimental Results

Table 4-1 shows the experimental results and the derived quantities for the three experiments of 15, 20 and 25 ach. Additional results from each experiment are included in Appendices F, G and H.

Table 4.1: Experimental Results and Derived Quantities

	<i>Air Changes per Hour</i>		
	<i>15</i>	<i>20</i>	<i>25</i>
Actual air changes per hour	15.26	20.31	24.30
Panel average temp (F)	103.97	103.24	103.80
Average temp west wall above panels (F)	73.07	70.15	71.04
Average temp east wall (F)	73.03	70.50	71.49
Average temp south wall (F)	72.91	70.91	71.92
Average temp north wall (F)	71.54	68.95	70.36
Average temp floor (F)	74.38	72.70	74.57
Average temp ceiling (F)	75.29	71.60	71.53
Average air inlet temp (F)	56.29	55.80	60.30
Average air outlet temp (F)	65.55	64.30	66.90
Average free stream air temp (F)	75.29	71.60	71.53
Average air inlet velocity (fpm)	122.07	162.48	190.50
Average air volumetric flow rate (cfm)	488.29	649.92	762.20
Average air mass flow rate (lb _m /hr)	2250.81	2993.88	3497.00
Average air density at inlet (lb _m /ft ³)	0.077	0.077	0.076
Average air kinematic viscosity (ft ² /s)	1.73e-4	1.73e-4	1.74e-4
Average air thermal diffusivity (ft ² /hr)	0.868	0.868	0.871
Average air coefficient of thermal expansion (1/R)	1.90e-3	1.91e-3	1.90e-3
Average jet momentum number	0.270e-3	0.470e-3	0.650e-3
Average Prandtl number	0.718	0.718	0.718
Average Archimedes number	2.2	1.1	0.64
Average Grashof number	58.6e9	64.8e9	65.2e9
Average Rayleigh	42.1e9	46.5e9	46.89

	<i>Air Changes per Hour</i>		
	<i>15</i>	<i>20</i>	<i>25</i>
number			
Average panel power input to room (Btu/hr)	5448.15	5634.86	5518.46
Average air heat gain (Btu/hr)	5003.62	6111.42	5527.50
Average convective flux for panels (Btu/hr-ft ²)	52.90	54.72	53.58
Average film coefficient (Btu/hr-ft ² -F)	1.39	1.41	1.46

The Rayleigh number is calculated as the product of the Prandtl number and the Grashof number. For all three experiments performed, the Rayleigh number is on the order of 10^9 . Generally, if the Rayleigh number is less than 10^9 the flow is laminar, whereas if the Rayleigh number is greater than 10^9 the flow is turbulent. These three experiments would be at the transition point of laminar to turbulent flow, if they were driven only by natural convection.

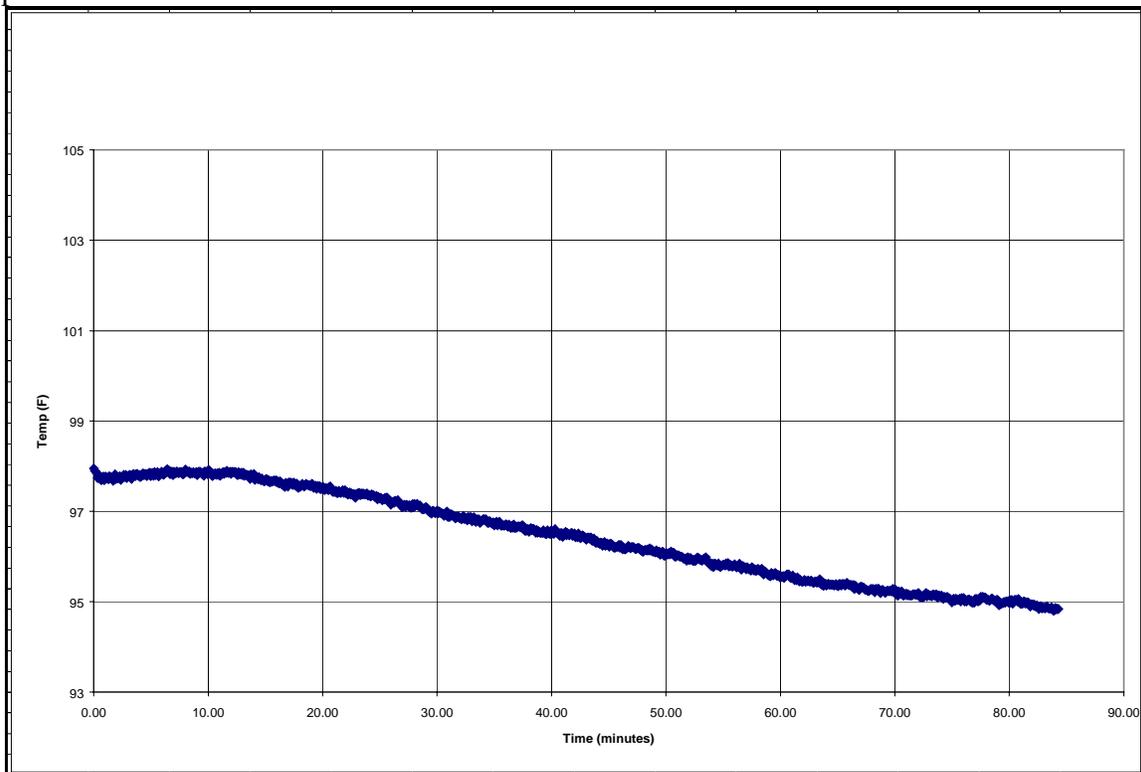
4.3 Panel Control Results

An important aspect of this experimental facility is control of the heated panels. As stated before, the heated panels must maintain a desired setpoint in order to achieve good experimental results. If the heated panels are constantly “hunting” the setpoint, then it will be nearly impossible to achieve steady state conditions in the experimental room. The “pattern” control method used does not completely eliminate “hunting”, but does reduce it greatly.

Figure 4.1 shows the temperature of panel number 1 during the 15 ach experiment. Time 0.0 is the start of the data recording for the experiment. The panel was allowed to reach the desired setpoint of 105°F before the fan was turned on. Immediately the temperature of the panel started to decrease. The panel temperature was allowed to drop until it seemed that the temperature of the panel had steadied. Panel number 1 was located next to the inlet of the room, and was not capable of maintaining the desired set point. Since panel number 1 was not

capable of maintaining the set point at the lowest flow rate that was studied, then it will not be able to maintain its setpoint at the two higher flow rates that were studied.

When looking at the data, the flux input for panel number 1 is lower than expected. Since the panel temperature is considerably lower than the desired setpoint, the flux input should be closer to 80% of the full duty cycle. This could be due to the incorrect “pattern” control being selected in the program and the panel is not “on” as much as it should be. This could be corrected by using a better control scheme like a proportional controller for the panels.



Figure

Figure 4.1: Panel number 1 temperature during 15 ach experiment

Figure 4.2 shows the temperature of panel number 8 during the 25 ach experiment. Panel number 8 is representative of panels 2 through 9 and panel 11. Panel number 8 is located half way up the heated surface and to the right of center for the wall. The center of this panel is located 2 feet below the centerline of the inlet and 10 feet away from the inlet. Figure 4.2 shows that panel number 8 is capable of maintaining the desired set point of 105°F

to within 0.5°F during the experiment. Since the panel is capable of maintaining the setpoint to within an acceptable allowance during the 25 ach experiment, then the panel should be able to maintain the desired setpoint at the lower flow rates.

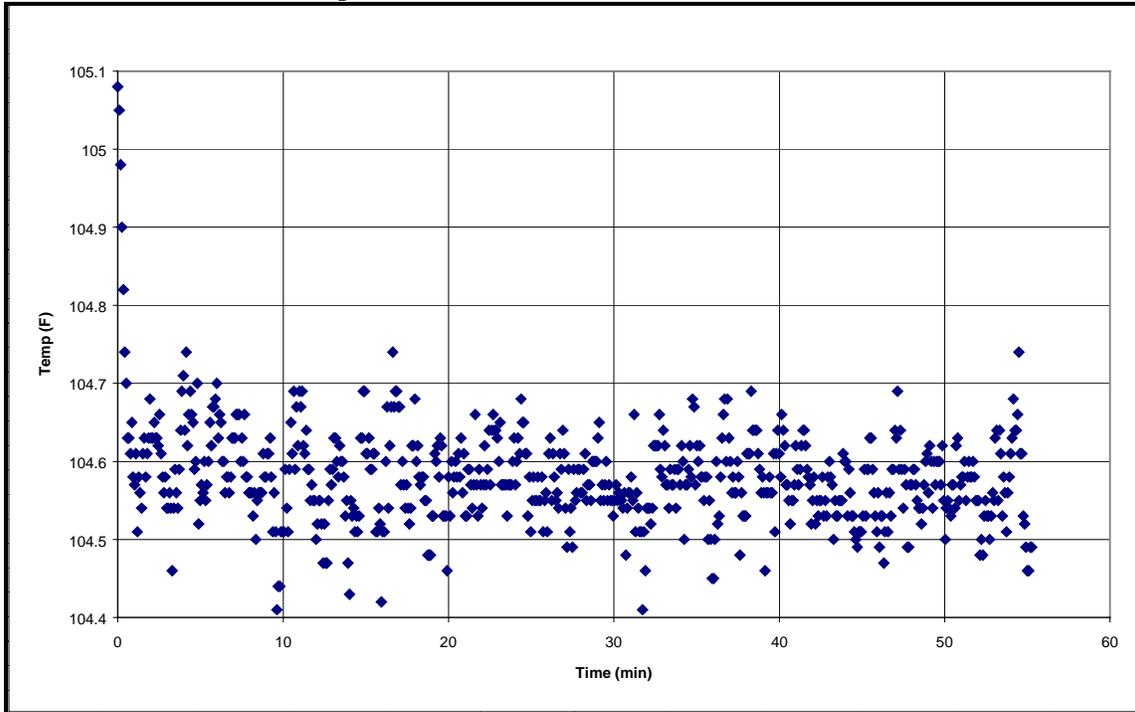


Figure 4.2: Panel number 8 temperature during 25 ach experiment

The temperature of panel number 12 during the 15 ach experiment is shown in Figure 4.3. Panel number 12 is located in the bottom right hand corner of the heated wall. It appears that the panel is under powered and requires a larger total heat flux. Since the panel is not capable of maintaining the desired setpoint at the low flow rate, then it will not maintain the setpoint at the higher flow rates that were performed. It is interesting to note though that the panel immediately above is capable of maintaining its desired setpoint. It could be possible that the north wall and the floor is interacting with panel number 12 somehow and that is the cause for the low temperature. Panel number 10 is behaving the same way as panel number 12, also. Possible explanations are that the wrong “pattern” control is being selected, or that the north wall in its passive condition is interacting somehow with the air flow over panels 10 and 12.

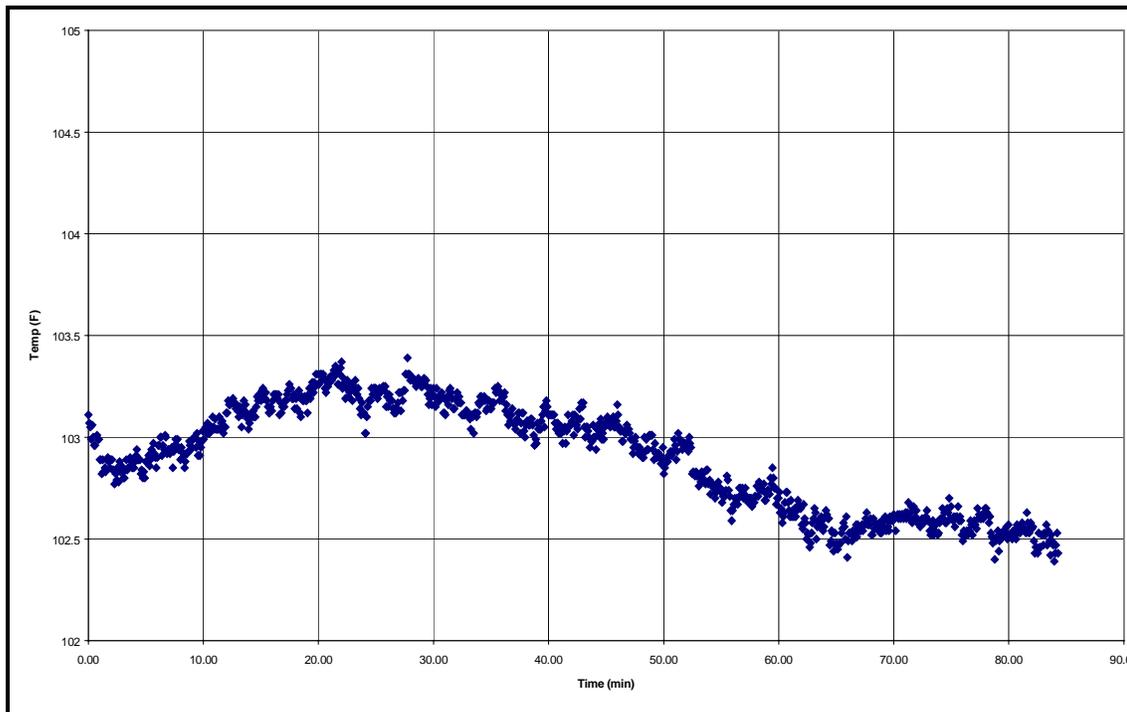


Figure 4.3: Panel number 12 temperature during 15 ach experiment

4.4 Transient Response of the Room

The transient response of the room was determined by allowing all the panels to come up to the setpoint temperature of 105°F, then turning on the fan to deliver air to the room. The temperature of the air inlet and outlet was monitored until the two temperatures became reasonably steady. It was then determined that steady state conditions had been obtained. Since these two temperatures were reasonable steady and the temperature difference between the inlet and the outlet is steady, then it can be concluded that the air heat gain has reached steady state. Figure 4.4a shows the temperatures of the air inlet and outlet during the 25 ach experiment. It took approximately one hour and forty five minutes for the panels to reach the setpoint temperature. After the fan was turned on, it took approximately 30 minutes for the room to reach steady state conditions. This was generally the case for the two experiments performed at 15 ach and 20 ach, also.

Figure 4.4b shows the air inlet and outlet temperatures from the 30 minute mark to the 55 minute mark.

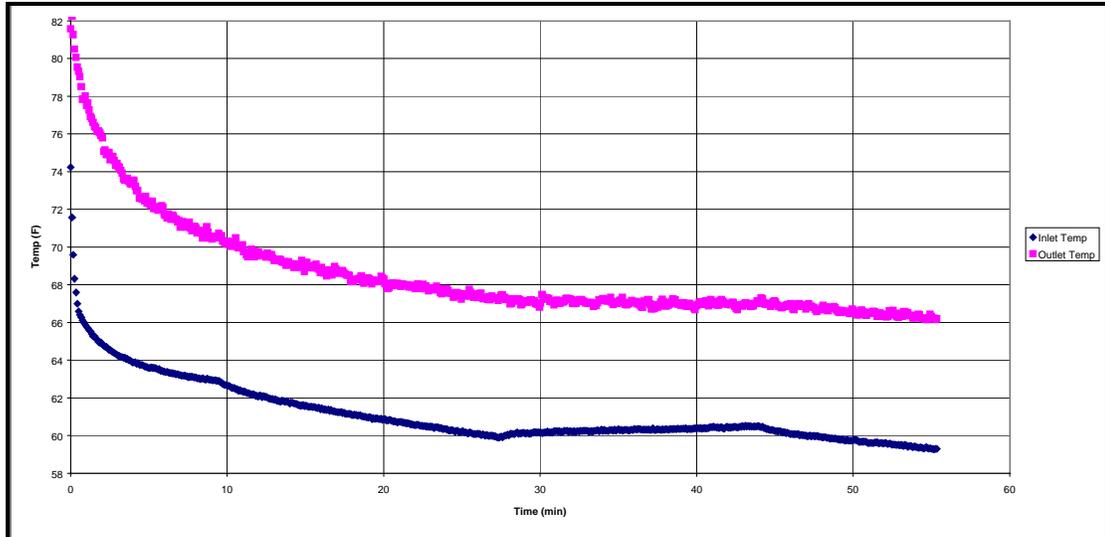
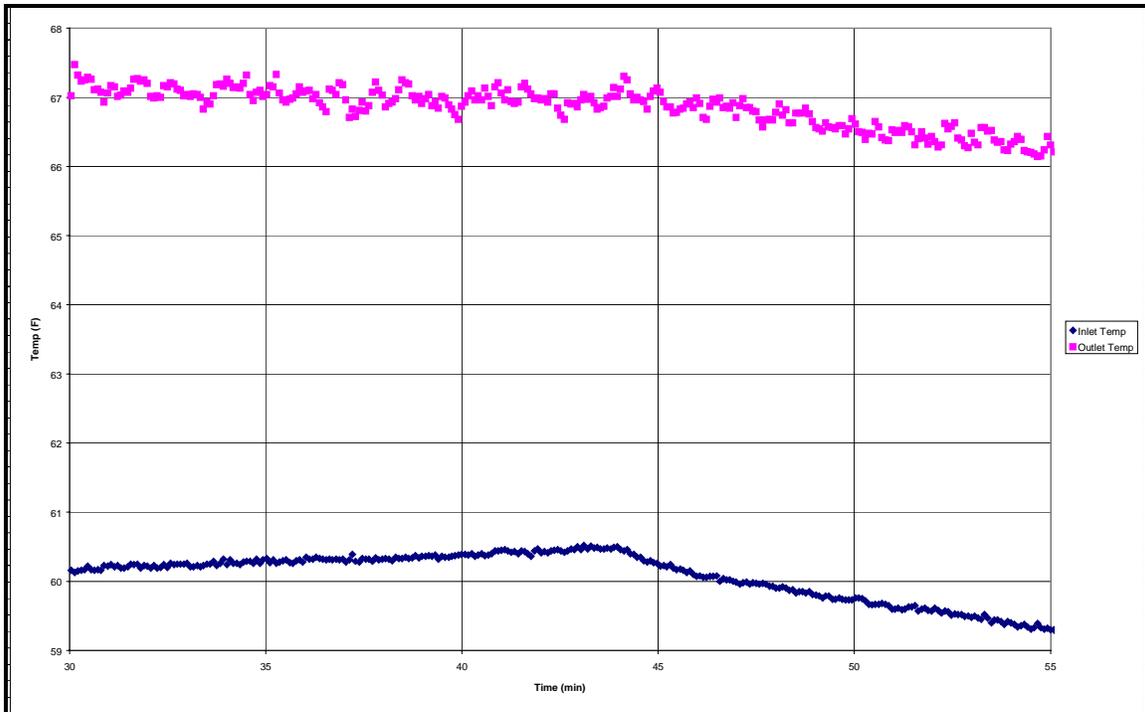


Figure 4.4a: Air inlet and outlet temperatures during the 25 ach experiment.

From the 30 minute mark to the end of the experiment, the inlet air temperature shows a slight rise and then a decrease. This is due to the cooling system in which a heat pump is cycled on and off to maintain a chilled water temperature in a storage tank. This water temperature cannot be kept constant during an experiment and it effects the temperature of the air at the room inlet. A means of controlling the inlet air temperature is needed.



Fig

ure 4.4b: Air inlet and outlet temperature during the last 25 minutes of the 25 ach experiment

4.5 Energy Balance on the Room

Ideally, if the room is perfectly balanced, the air heat gain will equal the panel power. Figure 4.5 shows the energy balances for the three experiments and all are within the predicted uncertainty. The diagonal line represents the ideal situation of when the panel power input equals the air heat gain. The horizontal bars represent the uncertainty in the panel power while the vertical bars represent the uncertainty in the air heat gain. The uncertainty in the air heat gain is due primarily to the uncertainty of the volumetric flow rate measurement. This uncertainty can be greatly decreased by improving the measurement of the pressure drop across the nozzle bank of the flow measurement box. The uncertainty in the panel power is due to the frequency of measurements of the voltage during the experiments and could be improved by continuous measurement of the AC line voltage.

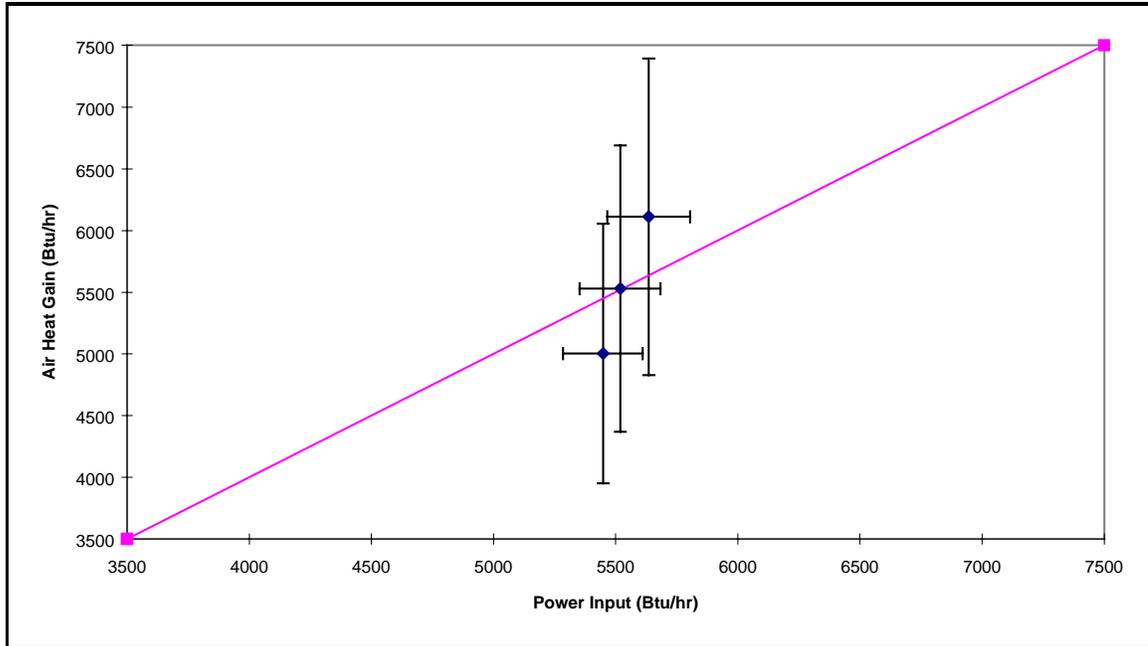


Figure 4.5: Energy Balance for the Three Experiments Performed Showing the Uncertainty in Panel Power and Air Heat Gain

4.6 Previously Published Correlations

The film coefficient determined from previously published correlations are used for comparison to the experimentally determined film coefficient to see if the results that are obtained from the experimental room are reasonable and “in the ballpark”. These correlations that are used for comparison do not apply for the conditions in which the three tests were run and are to be thought of as limiting cases only.

4.6.1 Natural Convection Correlations

By determining the Rayleigh, Grashof and Prandtl numbers for each experiment, the experimentally determined film coefficient can be compared to the film coefficient determined from correlations relating the Nusselt number to the Grashof and Prandtl numbers, and from correlations relating the Nusselt number to the Rayleigh and Prandtl numbers. The natural convection correlations that will be used to determine the film coefficient for comparison to the experimentally determined film coefficient are as follows:

The first correlation was presented by Eckert and Drake (1951). It is based on an approximate solution of the laminar boundary layer equations for the vertical isothermal flat plate.

$$\text{Nu} = 0.508\text{Pr}^{1/2}(0.952 + \text{Pr})^{-1/4}\text{Gr}^{1/4} \quad (4.1)$$

The second correlation was presented by Churchill and Chu (1975) for laminar free convection on a vertical isothermal surface over a wide range of Prandtl numbers. Note that this is the mean value of the Nusselt number, which is equivalent to $4/3 \text{Nu}_{x=L}$ for any case where $\text{Nu} = f(\text{Gr})^{1/4}$.

$$\overline{\text{Nu}} = 0.68 + \frac{0.67 \text{Ra}^{1/4}}{\left[1 + (0.492 / \text{Pr})^{9/16}\right]^{4/9}} \quad (4.2)$$

The third correlation for the mean Nusselt number developed by Churchill and Chu (1975) is:

$$\overline{\text{Nu}} = \left[0.825 + \frac{0.387 \text{Ra}^{1/6}}{\left[1 + (0.492 / \text{Pr})^{9/16}\right]^{8/27}}\right]^2 \quad (4.3)$$

This correlation is valid in both the laminar and turbulent flow regimes, however, it is not as accurate in the laminar region as the correlation presented for laminar free convection.

4.6.2 Forced Convection Correlations

The ASHRAE Handbook of Fundamentals (HOF) (1989) lists the following correlation for vertical plane surfaces for velocities less than sixteen feet per second:

$$h' = 0.99 + 0.21V \quad (4.4)$$

In this case though, the film coefficient is based on an “initial temperature difference”. The HOF does not define the “initial temperature difference”, but it appears to be the difference between the adjacent air temperature and the wall temperature.

All of the correlations listed above deal with free standing vertical surfaces. The last correlation that will be used for comparison was developed by Spitler (1990) for enclosures. The correlations developed by Spitler related the film coefficient to the inlet jet momentum

number. The correlation that is of interest for these particular experiments is Spitler's correlation for the north wall when an east wall inlet was used.

$$h = 1.6 + 127J^{0.5} \quad (4.5)$$

This correlation was developed for Archimedes numbers less than 0.3 and a range of jet momentum numbers of 0.002 to 0.011. The three experiments that were performed all have Archimedes numbers greater than 0.3 and jet momentum numbers less than the lower bound set on the correlation. The correlation gives the film coefficient in SI units. This is easily converted to IP units for comparison to experimentally derived film coefficients.

4.6.3 Comparison

The values in Table 4.2 for the ASHRAE HOF (eq. 4.4) correlation are obtained using the inlet velocity, as if it was maintained over the entire wall, and not the local velocity. This value of the film coefficient can be viewed as a maximum film coefficient for the experiment performed. Measurements of local air velocity were not made and it can be concluded that the film coefficient from equation 4.4 would be significantly less if the local velocity is used for the calculation. Notice that the film coefficients determined from the natural convection correlations are substantially lower than the experimentally determined film coefficient. Data from Table 4.2 is shown in Figure 4.6 with the uncertainty in the experimentally determined film coefficient shown also.

Table 4.2: Comparison of Experimental Film Coefficient to the Correlation Determined Film Coefficient

<i>Method of Determining Film Coefficient (Btu/hr-ft²-F)</i>	<i>Air Changes per Hour</i>		
	<i>15</i>	<i>20</i>	<i>25</i>
Experimental	1.39	1.41	1.46
Eckert and Drake correlation (eq. 4.1)	0.31	0.32	0.32
Churchill and Chu laminar correlation (eq. 4.2)	0.29	0.30	0.30
Churchill and Chu laminar and turbulet correlation (eq. 4.3)	0.50	0.52	0.52
ASHRAE HOF correlation (eq. 4.4)	1.42	1.56	1.66
Spitler's jet momentum correlation (eq 4.5)	0.65	0.77	0.85

The only correlation presented above that was developed in an enclosure similar to the one used for these experiments is Spitler's correlation (eq. 4.5). However, for the three experiments performed, the Archimedes number was greater than the 0.3 limitation on Spitler's correlation and the jet momentum number was less than Spitler's lower bound on the jet momentum number for the correlation. Spitler (1990) shows that at the higher Archimedes number, the film coefficient will increase, possibly even double that of the film coefficient calculated from the correlation. Figure 4.7 is reprinted from Spitler (1990) and shows this increase in the film coefficient.

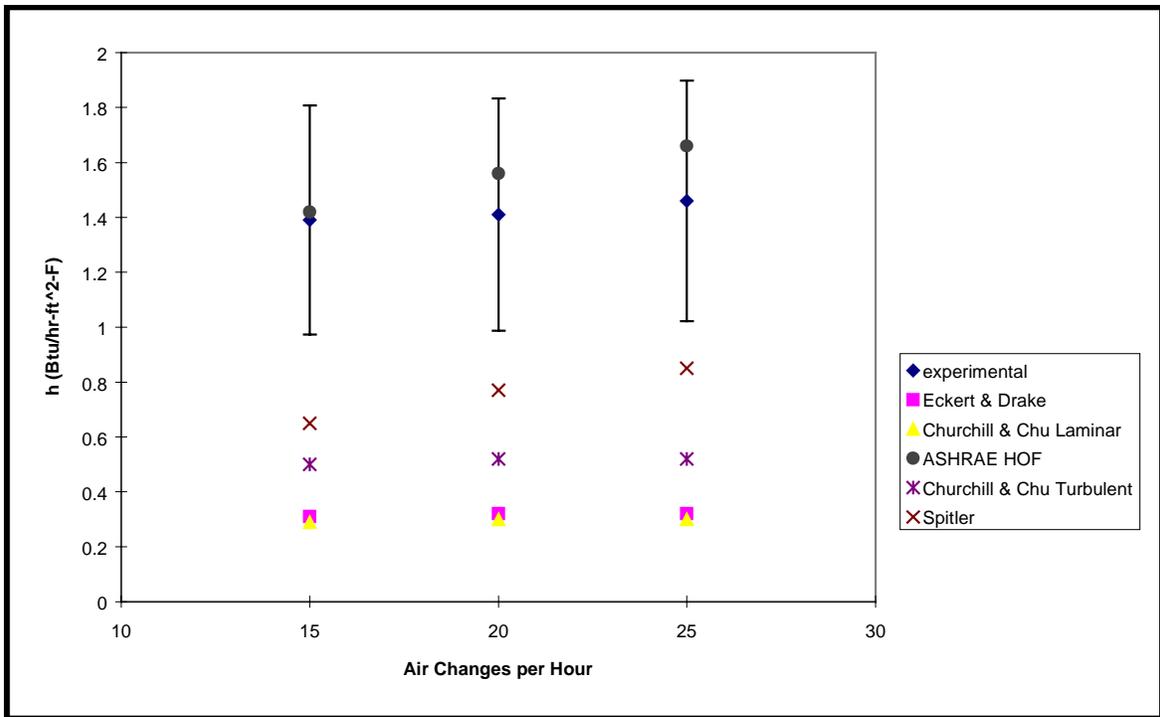


Figure 4.6: Film Coefficients Determined Experimentally and by Correlations at Each Flow Rate

Figure 4.7: h versus Bulk Air Velocity for Floor with East Wall Inlet, 22 Tests
(Size is proportional to Archimedes number)

To summarize, the experimentally determined coefficients were considerably higher than those estimated from the natural convection correlations. This is to be expected, because of the significant effect of the inlet jet. The experimentally determined film coefficients were significantly higher than Spitler's correlation, which might also be expected since the experiments were performed outside the range of Archimedes number and jet momentum numbers for the correlation. In this case, the Archimedes number was much higher than Spitler's, implying more significant buoyancy effects. The experimentally determined film coefficients were slightly lower than the ASHRAE HOF correlation, which is expected since the assumption that the inlet velocity remains constant over the entire wall was made. This assumption set an upper bound, or maximum value, for the film coefficient, which it did not exceed.

5 Conclusions and Recommendations

The following conclusions can be made about the Building Heat Transfer Facility constructed at Oklahoma State University:

1. A unique and versatile facility for the study of convective heat transfer and air flow in buildings has been developed. The experimental room utilizes a “honeycomb” frame with removable panels. This feature allows a multitude of configurations and phenomena to be investigated. Examples include:
 - Vertical or horizontal partitions can be added to the experimental room to allow for the study of heat transfer between zones, such as plenums or attic spaces.
 - The removable panel system allows for the study of the effect on the heat transfer due to different types of surface roughnesses that are found in buildings today.
 - The experimental room is not limited only to the study of convective heat transfer and air flow. It can also be used to study radiative and convective fraction of heat gain from equipment.
 - It can also be configured to study different types of heating systems, such as hydronic floor heating systems.
2. Losses due to conduction have been minimized by heavily insulating the floor, ceiling and walls. The losses due to conduction through the floor have further been minimized by supporting the experimental room off of the concrete floor of the laboratory. It has been shown that the energy balance criterion for the facility has been satisfied.
3. The experimentally determined film coefficients compare favorably to the film coefficients that are derived from the published correlations. These published

correlations should be thought of as setting an upper bound (forced convection) and a lower bound (natural convection) for the experimental situation.

4. The majority of the twelve panels were controlled successfully to within $\pm 1^\circ\text{F}$ of the desired setpoint. Three panels were not capable of reaching the desired setpoint and will require either a greater total heat flux or better control in future experiments. The present panels are capable of yielding a heat flux of 84.5 Btu/hr-ft^2 at full duty cycle and this may not be sufficient. The “pattern” control method is a simplistic control method and could also be the cause for the low flux on three of the twelve panels.
5. The final uncertainty in the air heat gain may be higher than what is desired. This can be reduced in future experiments with improved measurement of the volumetric flow rate of air entering the room.

Since these experiments were the first performed as “shakedown” tests, there is still a lot of work to be done to the facility. Some recommendations are:

1. A proportional controller for the panels so better control of the panel temperature can be achieved. This was an objective that could not be met at this time.
2. Investigation of using an epoxy - filled panel with a high thermal conductivity and a low thermal mass. The time to reach steady state conditions would be reduced and the temperature variation over the panel would be minimized. A low emissivity surface would reduce the radiative component of the heat transfer.
3. A larger air handling system, minimum of 3200 cfm, with a fan speed controller so larger ventilation rates can be achieved. The fan speed controller would allow for small ventilation rates to be used. The larger air handling system will allow

for a higher pressure differential across the nozzle bank, thus reducing the uncertainty in the volumetric flow rate. The present system is capable of providing 800 cfm of air to the room.

4. The installation of a high accuracy manometer and/or electronic pressure transducers connected to the datalogger to measure the air flow through the measurement box. This would greatly reduce the uncertainties in the air flow measurement.
5. The completion of the heated guard space to further reduce the conduction losses from the room. Once the guard space is completed, the conduction losses will be negligible in the calculations.
6. The placement of thermocouples inside the room to measure the air temperature at different locations inside the room. An even better solution would be an automated device inside the room to measure the air temperatures. This will allow for the determination of a bulk air temperature inside the room for calculating different dimensionless parameters.
7. The installation of a reheat coil connected to a controller in order to better control the room air inlet temperature and the ΔT between the wall surface temperature and the room air inlet. Currently, the air inlet temperature cannot be controlled.
8. Perform experiments at Archimedes numbers less than 0.3 and compare the results to Spittler's jet momentum correlation, and development of a correlation for the film coefficient at Archimedes numbers greater than 0.3.

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Appendices

Appendix A: Conduction Program for Predicting the Panel Surface Temperature

```
C      CONDUCTION PROGRAM FOR PANEL DESIGN
C      SCOTT SANDERS FALL 1993
C      THIS PROGRAM SIMULATES THE HEAT CONDUCTION THRU A
MATERIAL
C      WITH A WIRE EMBEDDED IN IT.  THE GRID SIZE IS 0.01
INCHES
C
      INTEGER JMAX
      REAL H
      DIMENSION T(60,60), TOLD(60,60)
      OPEN(UNIT=1,FILE='PC90.OUT')
C 2    WRITE(*,*)'ENTER MATERIAL THICKNESS
INCHES(0.125,0.25,0.375,0.5) '
C      READ(*,*) THICK
C      WRITE(*,*)'ENTER MATERIAL CONDUCTIVITY IN ENG. UNITS '
C      READ(*,*) AK
C
C      SET UP THE MAXIMUM NUMBER OF POINTS IN THE Y DIRECTION
DEPENDING
C      ON THE THICKNESS THAT IS SELECTED BY THE USER.
C
      2    THICK = 0.5
          AK = 5.6

          IF (THICK .EQ. 0.125) JMAX = 12
          IF (THICK .EQ. 0.25) JMAX = 25
          IF (THICK .EQ. 0.375) JMAX = 37
          IF (THICK .EQ. 0.5) JMAX = 50

C
C      THE MAXIMUM NUMBER OF POINTS IN THE X DIRECTION IS 50
C
      IMAX = 50
      WRITE(*,*)'ENTER WIRE TEMP IN DEGREES F '
      READ(*,*) TWIRE
C
C      ONE BOUNDARY CONDITION OF THIS PROBLEM IS A CONVECTIVE
HEAT
C      TRANSFER SURFACE.  THE USER WILL SELECT IF THE
CALCULATION
C      IS TO BE PERFORMED ON THE CEILING, FLOOR, OR WALL.
FROM THIS
```

```

C      A CONVECTIVE HEAT TRANSFER COEFFICIENT WILL BE
SELECTED BASED
C      ON PREVIOUS EXPERIMENTS.  THEY ARE:
C      H(CEILING) = 7.04 BTU/(ft^2-F)
C      H(WALL) = 1.76 BTU/(ft^2-F)
C      H(FLOOR) = 2.11 BTU/(ft^2-F)
C
WRITE(*,*) 'ENTER 0 FOR THE CEILNG,1 FOR THE WALL,2 FOR
THE FLOOR'
READ(*,*) L

IF(L .EQ. 0) H = 7.04
IF(L .EQ. 1) H = 1.76
IF(L .EQ. 2) H = 2.11

C      TINF IS THE AIR TEMPERATURE AT INFINITE DISTANCE FROM
THE
C      SURFACE, IT IS ASSUMED TO BE 75 DEGREES F.
C
TINF = 75

DELTX = 0.01
DEITY = 0.01

C      INITIALIZE THE GRID TO 75 F.

DO 3 I = 1,IMAX+1
DO 3 J = 1,JMAX
T(I,J) = 75.0
3 CONTINUE

DO 5 N = 1,5000

C
C      BOUNDARY CONDITIONS
C      THE BOTTOM AND TWO SIDES ARE CONSIDERED ADIABATIC
SURFACES

DO 10 I=1,IMAX+1
T(I,1) = T(I,3)
10 CONTINUE

DO 20 J=1,JMAX
T(1,J) = T(3,J)
T(IMAX+1,J) = T(IMAX-1,J)
20 CONTINUE

```

```

C          SET WIRE BOUNDARY CONDITIONS

          DO 30 I =2,6
          DO 30 J =2,6
            T(I,J) = TWIRE
30    CONTINUE

C          CALCULATE INTERIOR NODE POINTS TEMPERATURE
C          (REF. INCOPERIA AND DEWITT
C          "FUNDAMENTALS OF HEAT TRANSFER")
          DO 40 I = 7,IMAX
          DO 40 J = 2,JMAX-1
            T(I,J)=(T(I,J+1)+T(I,J-1)+T(I+1,J)+T(I-1,J))/4
40    CONTINUE
          DO 45 I = 2,6
          DO 45 J = 7,JMAX-1
            T(I,J)=(T(I,J+1)+T(I,J-1)+T(I+1,J)+T(I-1,J))/4
45    CONTINUE

C          CALCULATE TEMPERATURE ON SURFACE
C          (REF. INCOPERIA AND DEWITT
C          "FUNDAMENTALS OF HEAT TRANSFER")

          DO 50 I = 2,50
            T(I,JMAX)=(T(I-1,JMAX)+T(I+1,JMAX)+2*T(I,JMAX-
1)+2*H*DELTX
            $          *TINF/AK)/(2*(H*DELTX/AK + 2))
50    CONTINUE
          CHECK = 0.0
C          CHECK FOR CONVERGENCE OF SOLUTION
          DO 60 I = 2,IMAX
          DO 60 J = 2,JMAX
            IF (ABS(T(I,J)-TOLD(I,J)) .GT. CHECK) THEN
              CHECK = ABS(T(I,J) - TOLD(I,J))
            ENDIF
60    CONTINUE
          DO 65 I=2,IMAX
          DO 65 J=2,JMAX
            TOLD(I,J) = T(I,J)
65    CONTINUE
          IF (CHECK .LE. 0.01) THEN
            GOTO 500
          ENDIF
5    CONTINUE
C          WRITE THE SURFACE PANEL SURFACE TEMPERATURES

```

```
c 500 DO 400 J=1,JMAX
      500 DO 450 I=2,IMAX
          WRITE (1,900) T(I,JMAX)
450 CONTINUE
400 CONTINUE

      WRITE(1,*) 'NUMBER OF ITERATIONS',N
      WRITE(*,*) 'TYPE 1 TO RUN ANOTHER CASE'
      READ(*,*) M
      IF(M .EQ. 1) GOTO 2
900 FORMAT (50(F6.2,2x))

      STOP
      END
```

Appendix B: Calculation of Unit Resistance for the Floor, Ceiling and Walls of the Experimental Room

The thermal resistance is calculated by finding the conductance, UA, of the room element.

For the floor and ceiling, since they are identical constructions:

$$UA = \sum UA = U_{\text{stud}} A_{\text{stud}} + U_{\text{sty-ins}} A_{\text{sty-ins}}$$

Where: U = the overall heat transfer coefficient (Btu/(hr-ft²-°F))
A = the area (ft²)

The subscripts refer to the element of the floor or ceiling. Stud is the joist and sty-ins refers to the styrofoam and fiberglass insulation.

R-Values and Areas of Elements (Ref. 1989 ASHRAE Handbook of Fundamentals)

Element	Material R-Value (hr-ft ² -°F/Btu)	Element R-Value (hr-ft ² -°F/Btu)	U-Factor (Btu/hr-ft ² -°F)	Area (ft ²)
15" Thick Joist	1.0	15.0	0.067	0.734
0.75" Styrofoam	3.56	2.7		7.266
Fiberglass Insulation	19	57	0.017*	7.266

* Combined Styrofoam and Insulation

Multiplying the U-Factors by the areas to obtain the conductance, and summing the conductance results in a value of 0.171 Btu/(hr-°F)

The thermal resistance is found by inverting the conductance value.

$$1/0.171 = 5.84 \text{ hr-°F/Btu}$$

The unit resistance for the floor and ceiling is found by multiplying the thermal resistance by the unit area of a "honeycomb" cell, in this case, 8 ft².

$$5.84 * 8 = 46.7 \text{ hr - F - ft}^2 / \text{BTU}$$

The unit resistance for the walls is found the same way.

$$UA = \sum UA = U_{\text{stud-sty}}A_{\text{stud-sty}} + U_{\text{mason}}A_{\text{mason}} + U_{\text{ins-sty}}A_{\text{ins-sty}}$$

The subscripts stud - sty refers to the two 2x4 wall studs and the 11 inches of styrofoam between them, mason refers to the masonite and ins - sty refers to the fiberglass insulation and the styrofoam.

R-Values and Areas of Elements (Ref. 1989 ASHRAE Handbook of Fundamentals)

Element	Material R-Value (hr-ft ² -°F/Btu)	Element R-Value (hr-ft ² -°F/Btu)	U-Factor (Btu/hr-ft ² -°F)	Area (ft ²)
Wall Stud	1.0	7.0	0.022*	0.734
Styrofoam in Walls	3.57	39.25		0.734
Fiberglass Insulation	19	76	0.013**	7.03
0.75" Styrofoam	3.56	2.7		7.03
18" Thick Masonite	1.0	18	0.056	0.24

* Combined Wall Stud and Styrofoam

** Combined Insulation and Styrofoam

The conductance is calculated to be 0.12 Btu/(hr-F). Inverting this value gives a thermal resistance of 8.33 hr-F/Btu. Multiplying this value by the unit area of 8 ft² will give a unit resistance of 66.7 hr-F-ft²/Btu for the walls.

Appendix C: Control of Heated Panels and Temperature Recording Program

```
DECLARE SUB ALL.PANELS.OFF ()
DECLARE SUB ALL.PANELS.ON ()
DECLARE SUB CALC.PANEL.RESPONSE ()
DECLARE SUB COMPUTE.CONTROL.BYTES ()
DECLARE SUB COMPUTE.NEXT.SERIES ()
DECLARE SUB COPY.DATAFILE ()
DECLARE SUB DELETE.DATAFILE ()
DECLARE SUB DELETE.DELT.FILE ()
DECLARE SUB DISPLAY ()
DECLARE SUB ERROR.HANDLE (ERRCODE$, OTHERCODE$)
DECLARE SUB FIND.SURF.TCS ()
DECLARE SUB INIT.PORTS ()
DECLARE SUB INITIAL.COND ()
DECLARE SUB INPUT.CONFIG.FILE ()
DECLARE SUB MAKE.AUXDATA.FILE ()
DECLARE SUB OPEN.FLUKE.COM ()
DECLARE SUB PULSE.PANELS.OFF ()
DECLARE SUB PULSE.PANELS.ON ()
DECLARE SUB READ.AND.STORE.DATA ()
DECLARE SUB READ.WARMUP.DATA ()
DECLARE SUB SHUTDOWN ()
DECLARE SUB SIMPLE.SETPT.CTRL ()
DECLARE SUB TRIGGER.FLUKE.ONE ()
DECLARE SUB TRIGGER.FLUKE.TWO ()
DECLARE SUB UPDATE.ONOFF.ARRAYS ()
DECLARE SUB USER.INPUT ()
DECLARE SUB WARM.UP.LOOP ()
DECLARE SUB WARMUP.CONTROL.BYTES ()
```

```
'BUILDING HEAT TRANSFER LABORATORY, OKLAHOMA STATE UNIVERSITY
'INITIAL CONTROL PROGRAM. WRITTEN BY SCOTT SANDERS, BASED ON THE
'UIUC FACILITY CONTROL PROGRAM WRITTEN BY DAN FISHER. THIS PROGRAM
'WAS WRITTEN FOR SHAKEDOWN TESTS INVOLVING 12 PANELS ON THE WEST WALL.
'THE THERMOCOUPLES FOR THESE 12 PANELS ARE LOCATED ON CHANNELS 0 THRU 11
'AND AFTER MORE PANELS ARE ADDED, THEY SHOULD BE MOVED TO THEIR PROPER
'LOCATION. THE PANELS ARE PLUGGED INTO THE FIRST 12 SOCKETS FOR THE
'PANELS AND LATTER THEY SHOULD BE MOVED INTO THEIR PROPER LOCATION ON
'THE POWER BOX.
```

```
'DIMENSION THE PROGRAM ARRAYS
```

```
DIM SHARED TCDAT$(100), AVTEMP(100), TCADD$(100), TCDAT(100), TCINDAT(102)
DIM SHARED ONOFF$(64), CNTRL$(10), TEMP(21, 10), NEWDATA$(23)
DIM SHARED SURFTC(100), TCDAT$(100), TC$(100), RESIS(12)
DIM SHARED ONOFF1$(21), ONOFF2$(21), ONOFF3$(21), ONOFF4$(21), ONOFF5$(21)
DIM SHARED DELT1(21), DELT2(21), DELT3(21), DELT(21)
'DIM SHARED INCBAL1(500), INCBAL2(500)
```

```
'DEFINE VARIABLES AS GLOBAL
```

```
COMMON SHARED porta%, PORTB%, PORTC%, PANUM%, TCNUM%, SETPT, NEWS CRN
COMMON SHARED TITLE$, N%, OK, CALCLAG, SCANS.PER.SAVE, TIME, ERAS$
```

```

COMMON SHARED WARM.START, INTEMP, CEILING, SIDE, INLET.FLAG$, IFLAG, OUTTEMP
COMMON SHARED PM#, D1#, D2#, D3#, CAP1$, CAP2$, CAP3$, Q#, ACH#, HT#
COMMON SHARED DP#, SP#, A$, HOURS, MIN, SEC, HOUR$, MIN$, SEC$, INV
COMMON SHARED ICOUNT%, CP, DENS#, TOUTLET, TINLET, MDOT#, QAIR, QPANEL
COMMON SHARED HBALANCE, AVGBAL, RHOI#, RHOR#, DELTIME, RSTFLG%, HTN#
COMMON SHARED HTS#, HTN, HTS, INVN, INVS, AIRSIDE, PANSIDE, NOZTEMP

```

```

CLS
INVN = 1
INVS = 1
ERA$ = " "
CALL ALL.PANELS.OFF
NEWSCRN = 1
RSTFLAG% = 0
CALL INITIAL.COND
CALL USER.INPUT
CALL INPUT.CONFIG.FILE
CALL INIT.PORTS
CALL OPEN.FLUKE.COM
CALL TRIGGER.FLUKE.ONE
OPEN "C:\DATAFILE\OUTPUT\DATAFILE.DAT" FOR OUTPUT AS #2
PRINT #2, TIME$, DATE$, " ", TITLE$
IF WARM.START = 0 THEN
    CALL WARM.UP.LOOP 'controls warm up period for the room
    CALL CALC.PANEL.RESPONSE 'calc. the panel on and off temp setpts

```

```

ELSE
    FOR I = 1 TO PANUM%
        ONOFF1%(I) = 2
    NEXT I

```

```

END IF

```

```

N% = 0

```

```

NEWSCRN = 1
OK = 1: TIME0 = TIMER
SCANS.PER.SAVE = 25
DELTIME = 0
WHILE OK
CALL READ.AND.STORE.DATA 'retrieves and stores fluke data
CALL FIND.SURF.TCS 'extracts surface temps from fluke data
CALL UPDATE.ONOFF.ARRAYS 'panel on and off setpts based on temp slopes
CALL COMPUTE.CONTROL.BYTES 'calcs control bytes and switches panels

```

```

KEY$ = INKEY$
IF KEY$ = " " THEN CALL ERROR.HANDLE("INTERRUPT", " ")

```

```

CALL DISPLAY

```

```

IF RSTFLG% = 1 THEN TIME0 = TIMER

```

```

DELTIME = TIMER - TIME0
HOURS = INT(DELTIME / 3600): MIN = INT(DELTIME / 60) - HOURS * 60
SEC = INT((DELTIME * 10) / 10 - HOURS * 3600 - MIN * 60)
Q = Q#: ACH = ACH#: HTN = HTN#: HTS = HTS#

```

```

HOUR$ = STR$(HOURS)
IF HOURS < 10 THEN HOUR$ = "0" + RIGHT$(HOUR$, 1) ELSE HOUR$ = RIGHT$(HOUR$,
LEN(HOUR$) - 1)
MIN$ = STR$(MIN)
IF MIN < 10 THEN MIN$ = "0" + RIGHT$(MIN$, 1) ELSE MIN$ = RIGHT$(MIN$, LEN(MIN$) - 1)
SEC$ = STR$(SEC)
IF SEC < 10 THEN SEC$ = "0" + RIGHT$(SEC$, 1) ELSE SEC$ = RIGHT$(SEC$, LEN(SEC$) - 1)

LOCATE 22, 1: PRINT "ELAPSED TIME "; HOUR$; ":"; MIN$; ":"; SEC$
LOCATE 22, 29: PRINT "AVG INLET TEMP = "; USING "##.#"; INTEMP
'LOCATE 22,48: PRINT"FLOW RATE = ";USING "####.##";Q;" CFM"
LOCATE 23, 1: PRINT "STATIC PRESSURE = "; USING "#.####"; SP#
LOCATE 23, 29: PRINT "NOZZLE TEMP ="; USING "##.#"; NOZTEMP

'LOCATE 23, 29: PRINT "DELTA PRESSURE = "; USING "#.####"; DP#
'LOCATE 23,51: PRINT" AIR CHANGES/HOUR = ";USING"##.##";ACH
LOCATE 23, 51: PRINT "AVG OUTLET TEMP = "; USING "##.#"; OUTTEMP
IF RSTFLG% = 1 THEN
    CALL DELETE.DATAFILE
    CALL TRIGGER.FLUKE.TWO

END IF
'IF N% = SCANS.PER.SAVE THEN
' CALL COPY.DATAFILE
' CALL TRIGGER.FLUKE.TWO
'END IF
N% = N% + 1
WEND

IF KEY$ <> "q" THEN PRINT "PROGRAM TIMED OUT. SHUTDOWN ROUTINE INITIATED"

CALL SHUTDOWN
END

*****
SUB ALL.PANELS.OFF
'TURNS ALL PANELS OFF REGARDLESS OF PREVIOUS CONDITION

OUT porta%, 0
OUT PORTB%, 0
OUT PORTC%, 0

END SUB

*****
SUB ALL.PANELS.ON
'TURNS ALL PANELS ON REGARDLESS OF PREVIOUS CONDITION
OUT porta%, 255
OUT PORTB%, 255
OUT PORTC%, 255

END SUB

*****
SUB CALC.PANEL.RESPONSE
'CALCULATES RESPONSE OF EACH PANEL TO A SERIES OF "ON OFF" PULSES

```

```

CLS
PRINT "CALCULATING CONTROL PARAMETERS"
CALL ALL.PANELS.OFF
SLEEP 15
CALL PULSE.PANELS.ON
SLEEP 17.5
CALL PULSE.PANELS.OFF

FOR I = 1 TO 9
FOR J = 1 TO PANUM%
TEMPDEL = TEMP(J, I + 1) - TEMP(J, 1)
IF TEMPDEL > DELT1(J) AND TEMPDEL < 5 THEN DELT1(J) = TEMPDEL
NEXT J
NEXT I
CLS

PRINT "DELT1 ARRAY COMPLETE"
PRINT "RE-ENTERING WARM-UP LOOP"
PRINT "PRESS ANY KEY TO BEGIN DELT2 ARRAY"
SLEEP 15

CLS
CALL WARM.UP.LOOP
CLS
PRINT "CALCULATING CONTROL PARAMETER DELT2"
CALL ALL.PANELS.ON
SLEEP 35
CALL PULSE.PANELS.OFF

FOR I = 1 TO 9
FOR J = 1 TO PANUM%
TEMPDEL = TEMP(J, I + 1) - TEMP(J, 1)
IF TEMPDEL > DELT2(J) AND TEMPDEL < 5 THEN DELT2(J) = TEMPDEL
NEXT J
NEXT I
CLS

PRINT "DELT2 ARRAY COMPLETE"
PRINT "RE-ENTERING WARM-UP LOOP"
PRINT "PRESS ANY KEY WHEN READY TO CALCULATE DELT3 ARRAY"
CLS
PRINT "CALCULATING CONTROL PARAMETER DELT3"
CALL ALL.PANELS.OFF
SLEEP 15
CALL PULSE.PANELS.ON
SLEEP 52.5
CALL PULSE.PANELS.OFF

PRINT "CONTROL PARAMETER CALCULATIONS COMPLETE"

FOR I = 1 TO 9
FOR J = 1 TO PANUM%
TEMPDEL = TEMP(J, I + 1) - TEMP(J, 1)
IF TEMPDEL > DELT3(J) AND TEMPDEL < 10 THEN DELT3(J) = TEMPDEL
NEXT J
NEXT I

```

```

OPEN "DELT.DAT" FOR OUTPUT AS #3
FOR I = 1 TO PANUM%
PRINT #3, DELT1(I)
PRINT #3, DELT2(I)
PRINT #3, DELT3(I)
ONOFF1%(I) = 2
NEXT I
CLOSE #3
CLS
PRINT "CONTROL PARAMETER CALCULATION COMPLETE"

```

```

END SUB

```

```

*****

```

```

SUB COMPUTE.CONTROL.BYTES
'SUBROUTINE TO CALCULATE CONTROL BYTE FOR EACH PORT
FOR I = 1 TO 7
CNTRL%(I) = 0
NEXT I

```

```

CNTRL%(1) = ONOFF%(1) * 128 + ONOFF%(2) * 64 + ONOFF%(3) * 32 + ONOFF%(4) * 16 +
ONOFF%(5) * 8 + ONOFF%(6) * 4 + ONOFF%(7) * 2 + ONOFF%(8)
CNTRL%(2) = ONOFF%(9) * 128 + ONOFF%(10) * 64 + ONOFF%(11) * 32 + ONOFF%(12) * 16
'OUT porta%, 255
'OUT portc%, 64
OUT porta%, CNTRL%(1)
OUT PORTC%, CNTRL%(2)

```

```

PRINT #2, ERA$
PRINT #2, "TIMER (IN SECONDS)"
PRINT #2, TIMER
PRINT #2, "CONTROL WORDS 1-2"
PRINT #2, CNTRL%(1), CNTRL%(2)
FOR I = 1 TO PANUM%
PRINT #2, ONOFF%(I); " ";
NEXT I
PRINT #2, ERA$

```

```

END SUB

```

```

*****

```

```

SUB COMPUTE.NEXT.SERIES
'DETERMINES WHICH ONOFF PATTERN SHOULD BE LOADED INTO THE ONOFF ARRAYS

```

```

CTRLTMP = SETPT - TCDAT(IFLAG)

```

```

IF CTRLTMP >= DELT3(IFLAG) / 2 THEN

```

```

ONOFF1%(IFLAG) = 1
ONOFF2%(IFLAG) = 1
ONOFF3%(IFLAG) = 1
ONOFF4%(IFLAG) = 0
ONOFF5%(IFLAG) = 2

```

```

ELSEIF CTRLTMP >= DELT2(IFLAG) / 2 THEN

```

```

ONOFF1%(IFLAG) = 1

```

```

ONOFF2%(IFLAG) = 1
ONOFF3%(IFLAG) = 0
ONOFF4%(IFLAG) = 2

ELSEIF CTRLTMP >= DELT1(IFLAG) / 2 THEN
  ONOFF1%(IFLAG) = 1
  ONOFF2%(IFLAG) = 0
  ONOFF3%(IFLAG) = 2

ELSE
  ONOFF1%(IFLAG) = 0
  ONOFF2%(IFLAG) = 2

END IF

END SUB

*****
SUB COPY.DATAFILE
CLOSE #2
N% = 0
NEWSCRN = 1
SHELL "FILCOPY"
OPEN "C:\DATAFILE\OUTPUT\DATAFILE.DAT" FOR OUTPUT AS #2

END SUB

*****
SUB DELETE.DATAFILE
CLOSE #2
N% = 0
NEWSCRN = 1
IF RSTFLG% = 1 THEN
  RSTFLG% = 0
  SHELL "DATFLDEL"

END IF
OPEN "C:\DATAFILE\OUTPUT\DATAFILE.DAT" FOR OUTPUT AS #2

END SUB

*****
SUB DELETE.DELT.FILE
SHELL "DELFILE"

END SUB

*****
SUB DISPLAY
DIM BORDER(6)
CLS
BORDER(1) = 1: BORDER(2) = 1: BORDER(3) = 9: BORDER(4) = 13: BORDER(5) = 17
BORDER(6) = 21

FOR I = 1 TO 6

```

```

LOCATE BORDER(I), 2
FOR J = 2 TO 79
PRINT "-";
NEXT J
NEXT I
BORDER(1) = 1: BORDER(2) = 20: BORDER(3) = 40: BORDER(4) = 60: BORDER(5) = 80
FOR I = 1 TO 5
FOR J = 2 TO 21
LOCATE J, BORDER(I)
PRINT "|"
NEXT J
NEXT I

```

```

LOCATE 2, 37: PRINT "WEST"
LOCATE 19, 8: PRINT SURFTC(3)
LOCATE 15, 8: PRINT SURFTC(2)
LOCATE 11, 8: PRINT SURFTC(1)
LOCATE 19, 28: PRINT SURFTC(6)
LOCATE 15, 28: PRINT SURFTC(5)
LOCATE 11, 28: PRINT SURFTC(4)
LOCATE 19, 48: PRINT SURFTC(9)
LOCATE 15, 48: PRINT SURFTC(8)
LOCATE 11, 48: PRINT SURFTC(7)
LOCATE 19, 68: PRINT SURFTC(12)
LOCATE 15, 68: PRINT SURFTC(11)
LOCATE 11, 68: PRINT SURFTC(10)

```

END SUB

```

SUB ERROR.HANDLE (ERRCODE$, OTHERCODE$)
ABORT% = 0
CALL ALL.PANELS.OFF
SELECT CASE ERRCODE$

    CASE IS = "INTERRUPT"
        PRINT #2, "PROGRAM INTERRUPTED;PREVIOUS CONTROL BYTES SET TO 0"
        BEEP: OK = 0
        INTTIME = TIMER
        LOCATE 23, 1: PRINT ERA$: LOCATE 23, 5
        PRINT "PROGRAM INTERRUPTED"
        CASE IS = "WARM UP INTERRUPT"
        BEEP: OK = 5
        INTTIME = TIMER
        LOCATE 23, 5
        PRINT "WARM UP INTERRUPTED"
        SLEEP 3: LOCATE 23, 1: PRINT ERA$: LOCATE 23, 5

END SELECT
IF OK = 5 THEN
    LOCATE 23, 1: PRINT ERA$: LOCATE 23, 5
    OK = 0
    WHILE OK <> 1 AND (TIMER - INTTIME < 20)
    KEY$ = INKEY$
    SELECT CASE KEY$

```

```

CASE IS = "Q", "q"
ABORT% = 1
CLS
PRINT "ABORTING PROGRAM...ARE YOU SURE?"
CASE IS = "C", "c"
OK = 1: NEWSCRN = 1
PRINT "RESUMING WARM UP OPERATION"
CASE IS = "Y", "y"
IF ABORT% = 1 THEN END
CASE IS = " "
CASE ELSE
LOCATE 23, 5: PRINT "ENTER A 'C' TO RESUME OR A 'Q' TO QUIT"
END SELECT
WEND

ELSE
LOCATE 23, 5: PRINT "PRESS 'C' TO CONTINUE, 'Q' TO QUIT OR 'R' TO RESUME"
OK = 0
WHILE OK <> 1 AND (TIMER - INTTIME < 60)
KEY$ = INKEY$
SELECT CASE KEY$
CASE IS = "Q", "q"
PRINT "NORMAL PROGRAM TERMINATION"
CALL SHUTDOWN

CASE IS = "C", "c"
OK = 1: NEWSCRN = 1
ICOUNT% = 1

PRINT "RESUMING NORMAL OPERATION"
SLEEP 3

CASE IS = "R", "r"
OK = 1: NEWSCRN = 1
ICOUNT% = 1
PRINT "RESETTING HEAT BALANCE AND DELETING DATFILE.DAT"
SLEEP 3
RSTFLG% = 1
CASE IS = " "
CASE ELSE
PRINT "ENTER A 'C' TO CONTINUE,A 'Q' TO QUIT OR A 'R' TO RESUME"
END SELECT
WEND
END IF

END SUB

*****
SUB FIND.SURF.TCS
'EXTRACT HEATER PANEL SURFACE THERMOCOUPLES

FOR I = 1 TO PANUM%

SURFTC(I) = TCDAT(I)

```

NEXT I

END SUB

SUB INIT.PORTS

INITIALIZING PIO-96 BOARD,PORT 1 OF 4 FOR CONTROL OF PANELS

ADDRSS1% = 768

porta% = ADDRSS1% + 4

PORTB% = ADDRSS1% + 5

PORTC% = ADDRSS1% + 6

CTRLWD1% = ADDRSS1% + 7

OUT CTRLWD1%, 128

END SUB

SUB INITIAL.COND

TCNUM% = 12

PANUM% = 12

CALCLAG = 3

SCNTIME = 17

END SUB

SUB INPUT.CONFIG.FILE

THIS SUBROUTINE RETRIEVES THE PANEL DATA

THE PANEL DATA FILE IS IN THE FOLLOWING FORMAT

COLUMNS	CONTENTS
1-3	PANEL NUMBER
5-7	PANEL NAME
9-11	FLUKE CHANNEL NUMBER FOR TC
13-16	RESISTANCE OF PANEL

OPEN "C:\DATAFILE\CNTLDAT\ PANEL.DAT" FOR INPUT AS #1

FOR I = 1 TO PANUM%

LINE INPUT #1, CODELINE\$

TCADD%(I) = VAL(MID\$(CODELINE\$, 9, 3))

RESIS(I) = VAL(MID\$(CODELINE\$, 13, 4))

NEXT I

END SUB

SUB MAKE.AUXDATA.FILE

CLS

IN% = 1

WHILE IN% = 1

LOCATE 10, 5: INPUT "ENTER TEST CODE (Emmddy#) "; TESTNO\$

LOCATE 1, 5: PRINT "TEST CODE "; TESTNO\$

LOCATE 10, 5: PRINT ERA\$: LOCATE 10, 5: INPUT "INPUT LINE VOLTAGE "; VOLTAC#

LOCATE 2, 5: PRINT "LINE VOLTAGE = "; VOLTAC#

```

LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5: INPUT "AVERAGE STATIC PRESSURE (in wg) ";
AVGSTAT#
LOCATE 3, 5: PRINT "AVERAGE STATIC PRESSURE = "; AVGSTAT#; " in wg"
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5: INPUT "AVERAGE DELTA PRESSURE (in wg) ";
AVGDELTA#
LOCATE 4, 5: PRINT "AVERAGE DELTA PRESSURE = "; AVGDELTA#; "in wg"
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5: INPUT "ENTER DRY BULB TEMPERATURE (F) ";
DBULB#
LOCATE 5, 5: PRINT "DRY BULB TEMPERATURE IS "; DBULB#
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5: INPUT "ENTER WET BULB TEMPERATURE (F) ";
WBULB#
LOCATE 6, 5: PRINT "WET BULB TEMPERATURE IS "; WBULB#
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5
INPUT "IS THE ABOVE INFORMATION CORRECT (Y OR N) "; INFO$
IF INFO$ = "Y" OR INFO$ = "y" THEN
    IN% = 0
    ELSE
    CLS
    END IF

WEND

OPEN "C:\DATAFILE\OUTPUT\AUXDATA.DAT" FOR OUTPUT AS #7

'THE FIRST THREE LINES ARE TEXT
PRINT #7, TESTNO$
PRINT #7, TITLE$
PRINT #7, "COMMENTS:"

'NEXT IS THE LINE VOLTAGE
PRINT #7, "LINE VOLTAGE "; VOLTAC#

'AVERAGE STATIC PRESSURE
PRINT #7, "STATIC PRESSURE "; AVGSTAT#

'AVERAGE DELTA PRESSURE
PRINT #7, "DELTA PRESSURE "; AVGDELTA#

'CONVERSION OF BAROMETRIC PRESSURE FROM in Hg TO MILLIBARS
PBMB# = PM# * 3.38 / .1

'OUTPUT BAROMETRIC PRESSURE
PRINT #7, "BAROMETRIC PRESSURE IN MILLIBARS "; USING "####.##"; PBMB#

'DRY BULB TEMP
PRINT #7, "DRY BULB TEMP "; DBULB#

'WET BULB TEMP
PRINT #7, "WET BULB TEMP "; WBULB#

'NEXT ARE THE NOZZLES THAT ARE USED
PRINT #7, "7 INCH NOZZLE IS "; CAP1$
PRINT #7, "3 INCH NOZZLE IS "; CAP2$
PRINT #7, "1.6 INCH NOZZLE IS "; CAP3$

CLOSE #7

```

CLS

END SUB

SUB OPEN.FLUKE.COM

'OPENS COMMUNICATION PORT BETWEEN PC AND FLUKE

CLOSE #1

OPEN "COM2:9600,N,8,1,CS,DS,CD" FOR RANDOM AS #1

PRINT #1, "RESET"

PRINT #1, "MODE = COMP" 'SETS FLUKE MODE TO COMPUTER

20 INPUT #1, A\$

IF A\$ <> "!" THEN GOTO 20

PRINT #1, "FORMAT = DECIMAL" 'SETS FLUKE TO DECIMAL FORMAT

30 INPUT #1, A\$

IF A\$ <> "!" THEN GOTO 30

PRINT #1, "TUNIT = FAHRENHEIT" 'SETS READINGS TO FAHRENHEIT

40 INPUT #1, A\$

IF A\$ <> "!" THEN GOTO 40

'NEXT THE CHANNELS OF THE FLUKE ARE DEFINED AS THERMOCOUPLES, TYPE T

PRINT #1, "DEF CHAN(0..99) = TC,TYPE = TNBS"

END SUB

SUB PULSE.PANELS.OFF

'TURNS OFF ALL PANELS FOR 9 TIMESTEPS

CALL ALL.PANELS.OFF

FOR I = 2 TO 10

CALL READ.WARMUP.DATA

CALL FIND.SURF.TCS

FOR J = 1 TO PANUM%

TEMP(J, I) = TCDAT(J)

NEXT J

NEXT I

END SUB

SUB PULSE.PANELS.ON

'PULSES ALL PANELS. CALLED FROM SLOPE CALCULATION SUBROUTINE AND REQUIRED

'IN ORDER TO DETERMINE RESPONSE OF EACH PANEL

CALL READ.WARMUP.DATA

CALL FIND.SURF.TCS

FOR J = 1 TO PANUM%

TEMP(J, 1) = TCDAT(J)

NEXT J

CALL ALL.PANELS.ON

END SUB

SUB READ.AND.STORE.DATA

CALL TRIGGER.FLUKE.TWO

FOR I = 1 TO 51

IF TCDAT(I) > (SETPT + 10) THEN

IF TCDAT(I) > (SETPT + 20) THEN BEEP

LOCATE 23, 10: PRINT "ALARM TEMP! TC "; I; " = "; TCDAT(I)

SLEEP 5: LOCATE 23, 1: PRINT ERA\$

HOUR\$ = STR\$(HOURS)

IF HOURS < 10 THEN HOUR\$ = "0" + RIGHT\$(HOUR\$, 1) ELSE HOUR\$ = RIGHT\$(HOUR\$,
LEN(HOUR\$) - 1)

MIN\$ = STR\$(MIN)

IF MIN < 10 THEN MIN\$ = "0" + RIGHT\$(MIN\$, 1) ELSE MIN\$ = RIGHT\$(MIN\$, LEN(MIN\$) - 1)

SEC\$ = STR\$(SEC)

IF SEC < 10 THEN SEC\$ = "0" + RIGHT\$(SEC\$, 1) ELSE SEC\$ = RIGHT\$(SEC\$, LEN(SEC\$) - 1)

LOCATE 22, 1: PRINT "ELAPSED TIME "; HOUR\$; ":"; MIN\$; ":"; SEC\$

LOCATE 22, 29: PRINT "AVG INLET TEMP = "; USING "##.#"; INTEMP

LOCATE 22, 48: PRINT "FLOW RATE = "; USING "####.##"; Q; " CFM"

LOCATE 23, 1: PRINT "STATIC PRESSURE = "; USING "#.####"; SP#

LOCATE 23, 29: PRINT "NOZZLE TEMP = "; USING "##.#"; NOZTEMP

LOCATE 23, 29: PRINT "DELTA PRESSURE = "; USING "#.####"; DP#

LOCATE 23, 51: PRINT "AIR CHANGES/HOUR = "; USING "##.##"; ACH

LOCATE 23, 51: PRINT "AVG OUTLET TEMP = "; USING "##.#"; OUTTEMP

IF N% <> SCANS.PER.SAVE THEN CALL TRIGGER.FLUKE.TWO

END IF

PRINT #2, USING "####.##"; TCDAT(I);

NEXT I

IF INLET.FLAG\$ = "SIDE" THEN

INTEMP = (TCDAT(51) + TCDAT(50) + TCDAT(49) + TCDAT(48)) / 4

OUTTEMP = (TCDAT(47) + TCDAT(46) + TCDAT(45) + TCDAT(44)) / 4

NOZTEMP = (TCDAT(43) + TCDAT(42) + TCDAT(41) + TCDAT(40)) / 4

END IF

END SUB

SUB READ.WARMUP.DATA

'READS THE FLUKE DATA, BUT DOES NOT STORE IT

CALL TRIGGER.FLUKE.TWO

FOR I = 1 TO 51

IF TCDAT(I) > (SETPT + 10) THEN

IF TCDAT(I) > (SETPT + 20) THEN BEEP

LOCATE 24, 1: PRINT "ALARM TEMP ! TC "; I; " = "; TCDAT(I)

NEWSCRN = 1

END IF

NEXT I

IF INLET.FLAG\$ = "SIDE" THEN

```
INTEMP = (TCDAT(51) + TCDAT(50) + TCDAT(49) + TCDAT(48)) / 4
OUTTEMP = (TCDAT(47) + TCDAT(46) + TCDAT(45) + TCDAT(44)) / 4
NOZTEMP = (TCDAT(43) + TCDAT(42) + TCDAT(41) + TCDAT(40)) / 4
```

```
END IF
```

```
END SUB
```

```
*****
```

```
SUB SHUTDOWN
CALL ALL.PANELS.OFF
CALL MAKE.AUXDATA.FILE
END
```

```
END SUB
```

```
*****
```

```
SUB SIMPLE.SETPT.CTRL
'TURNS PANEL OFF IF SURFACE TEMP > SET POINT
'TURNS PANEL ON IF SURFACE TEMP < SET POINT
FOR I = 1 TO PANUM%
  IF TCDAT(I) < SETPT THEN
    ONOFF%(I) = 1
  ELSE ONOFF%(I) = 0
  END IF

```

```
NEXT I
```

```
END SUB
```

```
*****
```

```
SUB TRIGGER.FLUKE.ONE
'THIS SUBROUTINE REQUESTS THE TEMPERATURES FROM THE FLUKE. THE DATA
'HAS TO BE READ AS A CHARACTER STRING, OR THE PROGRAM WILL NOT OPERATE
'AND WILL CRASH. THE VAL FUNCTION IS USED TO CHANGE THE CHARACTER STRING
'TO A VALUE AND SET THE VALUES IN THE ARRAY TCDAT
'THIS SUBROUTINE IS REQUIRED FOR PROPER DATA STORAGE, OR THE TEMPERATURES
'WILL BE OFF BY ONE PLACE IN THE ARRAY.
```

```
PRINT #1, "SEND CHAN(0..11,20..46,80..91)"
```

```
FOR I = 1 TO 53
  INPUT #1, TC$(I)
  TCINDAT(I) = VAL(TC$(I))

```

```
NEXT I
```

```
END SUB
```

```
*****
```

```
SUB TRIGGER.FLUKE.TWO
'THIS SUBROUTINE REQUESTS THE TEMPERATURES FROM THE FLUKE. THE DATA
'HAS TO BE READ AS A CHARACTER STRING, OR THE PROGRAM WILL NOT OPERATE
'AND WILL CRASH. THE VAL FUNCTION IS USED TO CHANGE THE CHARACTER STRING
'TO A VALUE AND SET THE VALUES IN THE ARRAY TCDAT
PRINT #1, "SEND CHAN(0..11,20..46,80..91)"
```

```

FOR I = 1 TO 51
  INPUT #1, TC$(I)
  TCDAT(I) = VAL(TC$(I))
NEXT I

```

```

END SUB

```

```

*****

```

```

SUB UPDATE.ONOFF.ARRAYS
'LOADS NEW ONOFF PATTERNS FOR EACH PANEL
'IF N% = SCAN.PER.SAVE - 1 THEN
' FOR I = 1 TO PANUM%
'   ONOFF1%(I) = 0
'   ONOFF2%(I) = 2
'   NEXT I

```

```

'ELSE

```

```

FOR I = 1 TO PANUM%
IFLAG = I
IF ONOFF1%(I) = 2 THEN CALL COMPUTE.NEXT.SERIES
  ONOFF%(I) = ONOFF1%(I)
  ONOFF1%(I) = ONOFF2%(I)
  ONOFF2%(I) = ONOFF3%(I)
  ONOFF3%(I) = ONOFF4%(I)
  ONOFF4%(I) = ONOFF5%(I)

```

```

NEXT I
END IF
END SUB

```

```

*****

```

```

SUB USER.INPUT
'ACCEPTS USER INPUT FROM KEYBOARD. A WARM RESTART WILL RESULT IN THE
'CONTROL PARAMETERS CALCULATED IN CALC.PANEL.RESPONSE FROM THE PREVIOUS
'EXPERIMENT TO BE USED AND NEW PARAMETERS WILL NOT BE CALCULATED.

```

```

CLS
LOCATE 10, 5: INPUT "DATA TITLE "; TITLES$
LOCATE 1, 5: PRINT "DATA TITLE IS "; TITLES$
LOCATE 10, 5: PRINT ERA$
LOCATE 10, 5: INPUT "HOT WALL SURFACE TEMPERATURE SETPOINT (F) "; SETPT
LOCATE 2, 5: PRINT "TEMPERATURE SET POINT IS "; SETPT
LOCATE 10, 5: PRINT ERA$
LOCATE 10, 5: PRINT "USE OLD CONTROL PARAMETER FILE? (Y OR N) "
OK = 0
WHILE OK <> 1
  KEY$ = INKEY$
  SELECT CASE KEY$
  CASE IS = "Y", "y"
    OK = 1
    WARM.START = 1
    LOCATE 6, 5: PRINT "NEW CONTROL PARAMETERS WILL NOT BE CALCULATED"
  CASE IS = "N", "n"

```

```

OK = 1
WARM.START = 0
LOCATE 6, 5: PRINT "NEW CONTROL PARAMETERS WILL BE CALCULATED"

CASE IS = " "
CASE ELSE
LOCATE 20, 5: PRINT "ENTER A 'Y' OR A 'N' "
SLEEP 1
LOCATE 20, 5: PRINT ERA$
LOCATE 10, 5
END SELECT

WEND

IF WARM.START = 1 THEN
OPEN "DELT.DAT" FOR INPUT AS #1
FOR I = 1 TO PANUM%
  INPUT #1, DELT1(I)
  INPUT #1, DELT2(I)
  INPUT #1, DELT3(I)
NEXT I

END IF

CLOSE #1
LOCATE 10, 5
PRINT "SIDE OR CEILING AIR INLET? (ENTER 'S' OR 'C')"
  

OK = 0
WHILE OK <> 1
KEY$ = INKEY$
SELECT CASE KEY$
CASE IS = "S", "s"
OK = 1
INLET.FLAG$ = "SIDE"
LOCATE 7, 5: PRINT "SIDE INLET"

CASE IS = "C", "c"
OK = 1
INLET.FLAG$ = "CEILING"
LOCATE 7, 5: PRINT "CEILING INLET"
CASE IS = " "
CASE ELSE
LOCATE 20, 5: PRINT "ENTER 'S' OR 'C' "
SLEEP 1
LOCATE 20, 5: PRINT ERA$
LOCATE 10, 5

END SELECT
WEND
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5
INPUT "ENTER BAROMETRIC PRESSURE (in inches Hg) "; PM#
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5
INPUT "ENTER IF 7 INCH NOZZLE IS CAPPED OR UNCAPPED ('C' OR 'U') "; CAP1$
LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5
INPUT "ENTER IF 3 INCH NOZZLE IS CAPPED OR UNCAPPED ('C' OR 'U') "; CAP2$

```

```

LOCATE 10, 5: PRINT ERA$: LOCATE 10, 5
INPUT "ENTER IF 1.6 INCH NOZZLE IS CAPPED OR UNCAPPED ('C' OR 'U')"; CAP3$
IF CAP1$ = "U" OR CAP1$ = "u" THEN D1# = 7
IF CAP2$ = "U" OR CAP2$ = "u" THEN D2# = 3
IF CAP3$ = "U" OR CAP3$ = "u" THEN D3# = 1.6

```

```

END SUB

```

```

*****

```

```

SUB WARM.UP.LOOP
'THIS ROUTINE CONTROLS THE WARM UP PERIOD

```

```

OK = 1: TIME0 = TIMER

```

```

WHILE OK
CALL READ.WARMUP.DATA
CALL FIND.SURF.TCS
CALL SIMPLE.SETPT.CTRL
CALL WARMUP.CONTROL.BYTES

```

```

KEY$ = INKEY$
CALL DISPLAY
DELTIME = TIMER - TIME0

```

```

HOURS = INT(DELTIME / 3600): MIN = INT(DELTIME / 60) - HOURS * 60
SEC = INT((DELTIME * 10) / 10 - HOURS * 3600 - MIN * 60)
Q = Q#: ACH = ACH#: HTN = HTN#: HTS = HTS#

```

```

HOUR$ = STR$(HOURS)
IF HOURS < 10 THEN HOUR$ = "0" + RIGHT$(HOUR$, 1) ELSE HOUR$ = RIGHT$(HOUR$,
LEN(HOUR$) - 1)
MIN$ = STR$(MIN)
IF MIN < 10 THEN MIN$ = "0" + RIGHT$(MIN$, 1) ELSE MIN$ = RIGHT$(MIN$, LEN(MIN$) - 1)
SEC$ = STR$(SEC)
IF SEC < 10 THEN SEC$ = "0" + RIGHT$(SEC$, 1) ELSE SEC$ = RIGHT$(SEC$, LEN(SEC$) - 1)

```

```

LOCATE 22, 1: PRINT "ELAPSED TIME "; HOUR$; ":"; MIN$; ":"; SEC$
LOCATE 22, 29: PRINT "AVG INLET TEMP = "; USING "###.#"; INTEMP
'LOCATE 22,48: PRINT"FLOW RATE = ";USING "####.##";Q;" CFM"
LOCATE 23, 1: PRINT "STATIC PRESSURE = "; USING "#.####"; SP#
LOCATE 23, 29: PRINT "NOZZLE TEMP = "; USING "###.#"; NOZTEMP
LOCATE 23, 29: PRINT "DELTA PRESSURE = "; USING "#.####"; DP#
'LOCATE 23,51: PRINT" AIR CHANGES/HOUR = ";USING"###.##";ACH
LOCATE 23, 51: PRINT "AVG OUTLET TEMP = "; USING "###.#"; OUTTEMP
IF KEY$ <> " " THEN CALL ERROR.HANDLE("WARM-UP INTERRUPT", " ")
WEND

```

```

END SUB

```

```

*****

```

```

SUB WARMUP.CONTROL.BYTES
'SUBROUTINE TO CALCULATE CONTROL BYTE FOR EACH PORT. FOLLOWING PROTOCOL
'WAS USED TO DESIGNATE THE CONTROL BIT FOR EACH PANEL

```

```

'ONOFF(1) HIGH BIT,PORT A
'ONOFF(9) HIGH BIT,PORT B

```

```
'ONOFF(17) HIGH BIT, PORT C
```

```
FOR I = 1 TO 7  
CNTRL%(I) = 0  
NEXT I
```

```
CNTRL%(1) = ONOFF%(1) * 128 + ONOFF%(2) * 64 + ONOFF%(3) * 32 + ONOFF%(4) * 16 +  
ONOFF%(5) * 8 + ONOFF%(6) * 4 + ONOFF%(7) * 2 + ONOFF%(8)  
CNTRL%(2) = ONOFF%(9) * 128 + ONOFF%(10) * 64 + ONOFF%(11) * 32 + ONOFF%(12) * 16
```

```
'SWITCH PANELS ON OR OFF
```

```
'OUT porta%, 255  
'OUT portc%, 64
```

```
OUT porta%, CNTRL%(1)  
OUT PORTC%, CNTRL%(2)  
'OUT PORT1C%, CNTRL%(3)
```

```
END SUB
```

Appendix D: Interchange Area Factors Calculation Program for Radiation Analysis

```
program jsintfac  
implicit real(a-z)  
integer i,j,k,nv1,nv2,n,npans,pannum,l,surfnum,m,npm,npmx2,  
& nss  
integer*2 list  
parameter(npm=18,npmx2=36)  
dimension v1(3,10),v2(3,10),test1(7),test2(7),pannum(60)  
dimension coord(18,4,3),af(18,18),area(18),viewf(18,18)  
dimension surfnum(18)  
dimension tmtrx(npm,npmx2),a(npm),emit(npm),e(npm,npm)  
c this program uses George Walton's VIEWPP subroutine  
c program furnished by Spitler, modifications for shakedown tests and conversion to MS Fortran  
c by Sanders, Spring 1995.  
data nv1,nv2/4,4/  
data viewf/324*0./  
data emit/12*0.9,0.25,0.25,0.25,0.85,0.25,0.25/  
data nss/18/  
c using data statement to provide surfnum is an expedient for  
c the problem at hand. Ideally, surface numbers should be  
c in the coordinates file.  
data surfnum/13*1,1*2,1*3,1*4,1*5,1*6/  
  
open(8,file='debug.out',status = 'unknown')  
open(6,file='intfac.out',status = 'unknown')  
open(9,file='intfac.dat',status = 'unknown')  
C INITIALIZATION FROM OTHER WALTON SUBROUTINES  
  
DIV=10.0  
TEST1(1) = 0.30*DIV*256.
```

```

TEST1(2) = 0.30*DIV*16.
TEST1(3) = 0.30*DIV*4.
TEST1(4) = 0.30*DIV/4.
TEST1(5) = 0.30*DIV/16.
TEST1(6) = 0.30*DIV/144.
TEST1(7) = 0.30*DIV/1024.
DO 20 N=1,7
  TEST2(N) = 0.6*TEST1(N)
20 CONTINUE

```

C INPUT SECTION

```

open(7,file='coord.dat',status='unknown')
read(7,*)npanels
do 100 i=1,npanels
  read(7,*)pannum(i),((coord(i,j,k),k=1,3),j=1,4)
  if (pannum(i) .ne. i) then
    write(*,*)i('i,') not equal to pannum(i)('pannum(i),')
  end if
* convert from inches to feet
  do 90 j=1,4
    do 90 k=1,3
90    coord(i,j,k)=coord(i,j,k)/12
100  continue

```

C PANEL AREA CALCULATION SECTION

```

do 200 i=1,npanels
  x1=coord(i,1,1)
  y1=coord(i,1,2)
  z1=coord(i,1,3)
  x2=coord(i,2,1)
  y2=coord(i,2,2)
  z2=coord(i,2,3)
  x3=coord(i,3,1)
  y3=coord(i,3,2)
  z3=coord(i,3,3)
  x4=coord(i,4,1)
  y4=coord(i,4,2)
  z4=coord(i,4,3)
  l12=((x2-x1)*(x2-x1)+(y2-y1)*(y2-y1)+(z2-z1)*(z2-z1))**0.5
  l23=((x3-x2)*(x3-x2)+(y3-y2)*(y3-y2)+(z3-z2)*(z3-z2))**0.5
  l34=((x4-x3)*(x4-x3)+(y4-y3)*(y4-y3)+(z4-z3)*(z4-z3))**0.5
  l41=((x4-x1)*(x4-x1)+(y4-y1)*(y4-y1)+(z4-z1)*(z4-z1))**0.5
c  Jeff's approximately rectangular area calculation
  area(i)=(l12+l34)*(l23+l41)/4.
  if (i .le. npm)a(i)=area(i)
  write(8,*)i,area(i)
200  continue

```

C AREA*VIEW FACTOR CALCULATION

```

do 400 i=1,npanels
  do 400 j=1,npanels
C    write(*,*)i,j
    if(i .eq. j) then

```

```

        af(i,j)=0.
        viewf(i,j)=0.
        else if(surfnum(i) .eq. surfnum(j)) then
        af(i,j)=0.
        viewf(i,j)=0.
        else
        a1=area(i)
        a2=area(j)
        do 300 k=1,4
            do 300 l=1,3
                v1(l,k)=coord(i,k,l)
                v2(l,k)=coord(j,k,l)
300             continue
            call VIEWPP(nv1,v1,a1,nv2,v2,a2,af(i,j),TEST1,LIST)
            viewf(i,j)=af(i,j)/area(i)
            end if
            write(*,*)'View factor from ',i,' to ',j,' is ',viewf(i,j)
400         continue
C     ACCOUNT FOR PANEL WITHIN PANEL (CEILING INLET)
C     The ceiling inlet is panel 56 and is completely surrounded by
C     panel 40. Artificial panels 57,58,59 are actually part of panel(40)
C     Therefore the areas are added to panel(40)'s areas and new view factors
C     are calculated.

c     area(40)=area(40)+area(57)+area(58)+area(59)
c     a(40)=area(40)
c     npanels=npanels-3
c     do 450 i=1,npanels
c         viewf(i,40)=viewf(i,40)+viewf(i,57)+viewf(i,58)+viewf(i,59)
c         af(i,40)=viewf(i,40)*area(i)
c         viewf(40,i)=viewf(i,40)*area(i)/area(40)
c         af(40,i)=viewf(40,i)*area(40)
c450     continue

C     CHECK SECTION
do 500 i=1,npanels
    write(8,*)i,area(i)
    viewsum=0.
    do 490 j=1,npanels
490        viewsum=viewsum+viewf(i,j)
        write(*,*)'The sum of the view f. from panel ',i,' = ',viewsum
500    continue
C     INTERCHANGE FACTOR SECTION
do 600 i=1,npm
    do 600 j=1,npm
600        tmtx(i,j)=af(i,j)

c     emissivities of inlet and outlet are 1
c     emit(54)=0.999
c     emit(55)=0.999

m=0
call FAC(tmtx,a,emit,e,nss,m,npm,npmx2,,true.)

C     OUTPUT SECTION
close(8,status='keep')

```

```

open(8,file='viewfs.out',status='unknown')
write(8,*)npanels
do 1000 i=1,npanels
1000 write(8,*)area(i)
do 1100 i=1,npanels
do 1100 j=i,npanels
1100 write(8,*)i,j,viewf(i,j)

```

```

STOP
END

```

BLOCK DATA GLDATA

```

C
C DESCRIPTION *****
C
C Data statements for the Gauss-Legendre integration coefficients.
C
C DECLARATIONS *****
C
C PARAMETER (MAXGLO=10)
C PARAMETER (MAXGLC=MAXGLO*(MAXGLO+1)/2)
C
C COMMON /GLCOEF/ GINDEX, GX, GW
C
C INTEGER*2 GINDEX(MAXGLO)
C REAL GX(MAXGLC), GW(MAXGLC)
C
C CODE *****
C
DATA (GINDEX(I),I=1,10) / 0, 1, 3, 6, 10, 15, 21, 28, 36, 45 /
DATA GX(1), GW(1) / 0.500000000, 1.000000000 /
DATA (GX(I),I=2,3) / 0.211324865, 0.788675135 /
DATA (GW(I),I=2,3) / 0.500000000, 0.500000000 /
DATA (GX(I),I=4,6) / 0.112701665, 0.500000000, 0.887298335 /
DATA (GW(I),I=4,6) / 0.277777778, 0.444444444, 0.277777778 /
DATA (GX(I),I=7,10) / 0.069431844, 0.330009478, 0.669990522,
- 0.930568156 /
DATA (GW(I),I=7,10) / 0.173927423, 0.326072577, 0.326072577,
- 0.173927423 /
DATA (GX(I),I=11,15) / 0.046910077, 0.230765345, 0.500000000,
- 0.769234655, 0.953089923 /
DATA (GW(I),I=11,15) / 0.118463443, 0.239314335, 0.284444444,
- 0.239314335, 0.118463443 /
DATA (GX(I),I=16,21) / 0.033765243, 0.169395307, 0.380690407,
- 0.619309593, 0.830604693, 0.966234757 /
DATA (GW(I),I=16,21) / 0.085662246, 0.180380787, 0.233956967,
- 0.233956967, 0.180380787, 0.085662246 /
DATA (GX(I),I=22,28) / 0.025446044, 0.129234407, 0.297077424,
- 0.500000000, 0.702922576, 0.870765593,
- 0.974553956 /
DATA (GW(I),I=22,28) / 0.064742483, 0.139852696, 0.190915025,
- 0.208979592, 0.190915025, 0.139852696,
- 0.064742483 /
DATA (GX(I),I=29,36) / 0.019855072, 0.101666761, 0.237233795,

```

```

-          0.408282679, 0.591717321, 0.762766205,
-          0.898333239, 0.980144928 /
DATA (GW(I),I=29,36) / 0.050614268, 0.111190517, 0.156853323,
-          0.181341892, 0.181341892, 0.156853323,
-          0.111190517, 0.050614268 /
DATA (GX(I),I=37,45) / 0.015919880, 0.081984446, 0.193314284,
-          0.337873288, 0.500000000, 0.662126712,
-          0.806685716, 0.918015554, 0.984080120 /
DATA (GW(I),I=37,45) / 0.040637194, 0.090324080, 0.130305348,
-          0.156173539, 0.165119678, 0.156173539,
-          0.130305348, 0.090324080, 0.040637194 /
DATA (GX(I),I=46,55) / 0.013046736, 0.067468317, 0.160295216,
-          0.283302303, 0.425562831, 0.574437169,
-          0.716697697, 0.839704784, 0.932531683,
-          0.986953264 /
DATA (GW(I),I=46,55) / 0.033335672, 0.074725675, 0.109543181,
-          0.134633360, 0.147762112, 0.147762112,
-          0.134633360, 0.109543181, 0.074725675,
-          0.033335672 /
END

```

SUBROUTINE VIEWPP(NV1,V1,A1,NV2,V2,A2,AF,TEST,LIST)

```

C
C DESCRIPTION *****
C
C Compute the radiation interchange area between polygons 1 and 2
C by the line integral or Mitalas-Stephenson (M-S) integral method.
C This routine selects the number of edge divisions based upon
C some ad-hoc length and distance relationships.
C The M-S method becomes less accurate than LI at relative
C separation distances greater than about 2.
C SUM and SUMT must be high precision to reduce round-off error.
C
C DECLARATIONS *****
C
C Input:
C NV1 - number of vertices/edges of polygon #1.
C V1 - vertices of polygon #1 (includes room for dupl. vertex).
C A1 - (approximate) area of polygon #1.
C NV2 - number of vertices/edges of polygon #2.
C V2 - vertices of polygon #2 (includes room for dupl. vertex).
C A2 - (approximate) area of polygon #2.
C TEST - relative separation values for number of edge divisions.
C Output:
C AF - direct interchange area (area * view factor).
C Local:
C DL1 - direction numbers and edge lengths of polygon #1:
C DL1(1,I) = X direction number of edge I,
C DL1(2,I) = Y direction number of edge I,
C DL1(3,I) = Z direction number of edge I,
C DL1(4,I) = distance from vertex I to vertex I+1.
C DL2 - direction numbers and edge lengths of polygon #2.
C RMIN - length of the shorter edge (I or J).

```

```

C   RMAX - length of the longer edge (I or J).
C   SEP2 - square of relative separation dist. (between midpoints)
C   UMIN2 - square of minimum (perpendicular) distance between edges.
C   PARLEL - true if edges I and J are parallel.
C   R2 - square of length of long edge.
C   U - distance between parallel edges.
C
C   PARAMETER (MAXGLO=10)
C   PARAMETER (MAXGLC=MAXGLO*(MAXGLO+1)/2)
C
C   COMMON /GLCOEF/ GINDEX, GX, GW
C
C   INTEGER*2 GINDEX(MAXGLO)
C   REAL GX(MAXGLC), GW(MAXGLC)
C
C   integer NV1,NV2
C   INTEGER*2 LIST, I, J, K, M, N, LMIN, ND
C   LOGICAL PARLEL
C   DOUBLE PRECISION SUM, SUMT
C   REAL V1(3,10), V2(3,10), A1, A2, AF, TEST(7)
C   REAL DL1(4,12), DL2(4,12), DV, X, Y, Z
C   REAL SEP2, RMAX, RMIN, TEST1, TEST2, UMIN2, A, B, C
C   REAL EPS0, EPS1, H, DOT, U, OMEGA, R2, S2, T2
C   REAL SD, SE, SX, SY, SZ, TD, TE, TX, TY, TZ
C
C   CODE *****
C
C   DATA EPS0, EPS1, H / 0.00001, 0.99999, 0.5 /
C   FACTOR = 1.0
C
C           Compute direction numbers and length for
C           all edges of polygon #1.
C   V1(1,NV1+1) = V1(1,1)
C   V1(2,NV1+1) = V1(2,1)
C   V1(3,NV1+1) = V1(3,1)
C   DO 10 N=1,NV1
C       DL1(1,N) = V1(1,N+1)-V1(1,N)
C       DL1(2,N) = V1(2,N+1)-V1(2,N)
C       DL1(3,N) = V1(3,N+1)-V1(3,N)
C       DL1(4,N) = SQRT(DL1(1,N)**2 + DL1(2,N)**2 + DL1(3,N)**2)
C   10 CONTINUE
C
C           Compute direction numbers and length for
C           all edges of polygon #2.
C   V2(1,NV2+1) = V2(1,1)
C   V2(2,NV2+1) = V2(2,1)
C   V2(3,NV2+1) = V2(3,1)
C   DO 12 N=1,NV2
C       DL2(1,N) = V2(1,N+1)-V2(1,N)
C       DL2(2,N) = V2(2,N+1)-V2(2,N)
C       DL2(3,N) = V2(3,N+1)-V2(3,N)
C       DL2(4,N) = SQRT(DL2(1,N)**2 + DL2(2,N)**2 + DL2(3,N)**2)
C   12 CONTINUE
C
C           Process all edges to compute AF.
C   20 SUMT = 0.0
C   DO 100 I=1,NV1
C   DO 100 J=1,NV2
C
C           Compute/test dot product of unit vectors.

```

```

DOT = (DL1(1,I)*DL2(1,J)+DL1(2,I)*DL2(2,J)+DL1(3,I)*DL2(3,J))
- / (DL1(4,I)*DL2(4,J))
IF(ABS(DOT) .LE. EPS0) GO TO 100
SUM = 0.0
C          Determine integration method.
IF(DL1(4,I).GT.DL2(4,J)) THEN
  LMIN = 2
  RMIN = DL2(4,J)
  RMAX = DL1(4,I)
ELSE
  LMIN = 1
  RMIN = DL1(4,I)
  RMAX = DL2(4,J)
END IF
A = V1(1,I)+V1(1,I+1)-V2(1,J)-V2(1,J+1)
B = V1(2,I)+V1(2,I+1)-V2(2,J)-V2(2,J+1)
C = V1(3,I)+V1(3,I+1)-V2(3,J)-V2(3,J+1)
SEP2 = 0.25*(A*A+B*B+C*C)
R2 = RMAX*RMAX
TEST1 = FACTOR*SEP2/R2
IF(TEST1.GT.TEST(2)) GO TO 80
C          M-S method for relatively close edges.
C          Compute minimum distance between lines
IF(ABS(DOT).GT.EPS1) THEN
C          for parallel lines,
  PARLEL = .TRUE.
  A = V2(1,J)-V1(1,I)
  B = V2(2,J)-V1(2,I)
  C = V2(3,J)-V1(3,I)
  UMIN2 = ((DL2(3,J)*B-DL2(2,J)*C)**2
- + (DL2(1,J)*C-DL2(3,J)*A)**2
- + (DL2(2,J)*A-DL2(1,J)*B)**2) / DL2(4,J)**2
  IF(DOT.LT.0.0 .AND. UMIN2.LT.EPS0*EPS0) GO TO 70
  U = SQRT(UMIN2)
ELSE
C          or for skew lines.
  PARLEL = .FALSE.
  A = DL1(2,I)*DL2(3,J)-DL1(3,I)*DL2(2,J)
  B = DL1(3,I)*DL2(1,J)-DL1(1,I)*DL2(3,J)
  C = DL1(1,I)*DL2(2,J)-DL1(2,I)*DL2(1,J)
  UMIN2 = ((A*(V2(1,1)-V1(1,1))
- + B*(V2(2,1)-V1(2,1))
- + C*(V2(3,1)-V1(3,1)))**2
- / (A*A+B*B+C*C)
  END IF
C          Determine number of edge divisions.
TEST2 = FACTOR*UMIN2*A1*A2/(RMAX*RMIN*RMIN)**2
IF(TEST2.LT.TEST(3)) GO TO 30
ND = 2
GO TO 38
30 IF(TEST2.LT.TEST(4)) GO TO 31
ND = 3
GO TO 38
31 IF(TEST2.LT.TEST(5)) GO TO 32
ND = 4
GO TO 38

```

```

32 IF(TEST2.LT.TEST(6)) GO TO 33
   ND = 5
   GO TO 38
33 IF(TEST2.LT.TEST(7)) GO TO 34
   ND = 6
   GO TO 38
34 ND = 7
38 CONTINUE
   K = GINDEX(ND)
   IF(LMIN.EQ.2) GO TO 50
C           Edge I shorter than edge J; subdivide I to
C           perform the M-S integration.
   DO 40 M=1,ND
C           Compute integration points.
   X = V1(1,I)+GX(M+K)*DL1(1,I)
   Y = V1(2,I)+GX(M+K)*DL1(2,I)
   Z = V1(3,I)+GX(M+K)*DL1(3,I)
   DV = GW(M+K)*DL1(4,I)
C           Compute D*R*T*cos(PHI)*log(T).
   TX = V2(1,J+1)-X
   TY = V2(2,J+1)-Y
   TZ = V2(3,J+1)-Z
   T2 = TX*TX+TY*TY+TZ*TZ
   SUM = SUM+H*DV*(TX*DL2(1,J)+TY*DL2(2,J)+TZ*DL2(3,J))*LOG(T2)
C           Compute D*R*S*cos(THETA)*log(S).
   SX = X-V2(1,J)
   SY = Y-V2(2,J)
   SZ = Z-V2(3,J)
   S2 = SX*SX+SY*SY+SZ*SZ
   SUM = SUM+H*DV*(SX*DL2(1,J)+SY*DL2(2,J)+SZ*DL2(3,J))*LOG(S2)
C           Compute D*R*U*OMEGA.
   IF(PARLEL) THEN
     OMEGA = H*(S2+T2-R2)/SQRT(S2*T2)
     IF(ABS(OMEGA).GE.1.0) THEN
       IF(ABS(OMEGA).GT.1.002)
-       write(*,*)'ACOS argument too large (VIEWPP)'
       OMEGA = 0.0
     ELSE
       OMEGA = ACOS(OMEGA)
     END IF
     SUM = SUM+DV*RMAX*U*OMEGA
   ELSE
     SXR2 = (SY*DL2(3,J)-DL2(2,J)*SZ)**2
-     +(SZ*DL2(1,J)-DL2(3,J)*SX)**2
-     +(SX*DL2(2,J)-DL2(1,J)*SY)**2
     OMEGA = SQRT(SXR2/(S2*T2))
     IF(OMEGA.GE.1.0) THEN
       IF(OMEGA.GT.1.002)
-       write(*,*)'ASIN argument too large (VIEWPP)'
       OMEGA = 1.5707963
     ELSE
       OMEGA = ASIN(OMEGA)
     END IF
     IF(R2.GT.S2+T2) OMEGA = 3.141592654-OMEGA
     SUM = SUM+DV*SQRT(SXR2)*OMEGA
   ENDIF

```

```

40  CONTINUE
    SUM = SUM/RMAX-RMAX*RMIN
    SUMT = SUMT+DOT*SUM
    GO TO 100
C
    Edge J shorter than edge I; subdivide J to
C
    perform the M-S integration.
50  CONTINUE
    DO 60 M=1,ND
C
    Compute integration points.
    X = V2(1,J)+GX(M+K)*DL2(1,J)
    Y = V2(2,J)+GX(M+K)*DL2(2,J)
    Z = V2(3,J)+GX(M+K)*DL2(3,J)
    DV = GW(M+K)*DL2(4,J)
C
    Compute D*R*T*cos(PHI)*log(T).
    TX = V1(1,I+1)-X
    TY = V1(2,I+1)-Y
    TZ = V1(3,I+1)-Z
    T2 = TX*TX+TY*TY+TZ*TZ
    SUM = SUM+H*DV*(TX*DL1(1,I)+TY*DL1(2,I)+TZ*DL1(3,I))*LOG(T2)
C
    Compute D*R*S*cos(THETA)*log(S).
    SX = X-V1(1,I)
    SY = Y-V1(2,I)
    SZ = Z-V1(3,I)
    S2 = SX*SX+SY*SY+SZ*SZ
    SUM = SUM+H*DV*(SX*DL1(1,I)+SY*DL1(2,I)+SZ*DL1(3,I))*LOG(S2)
C
    Compute D*R*U*OMEGA.
    IF(PARLEL) THEN
        OMEGA = H*(S2+T2-R2)/SQRT(S2*T2)
        IF(ABS(OMEGA).GE.1.0) THEN
            IF(ABS(OMEGA).GT.1.002)
-           write(*,*)'ACOS argument too large (VIEWPP)'
            OMEGA = 0.0
        ELSE
            OMEGA = ACOS(OMEGA)
        END IF
        SUM = SUM+DV*RMAX*U*OMEGA
    ELSE
        SXR2 = (SY*DL1(3,I)-DL1(2,I)*SZ)**2
-       +(SZ*DL1(1,I)-DL1(3,I)*SX)**2
-       +(SX*DL1(2,I)-DL1(1,I)*SY)**2
        OMEGA = SQRT(SXR2/(S2*T2))
        IF(OMEGA.GE.1.0) THEN
            IF(OMEGA.GT.1.002)
-           write(*,*)'ASIN argument too large (VIEWPP)'
            OMEGA = 1.5707963
        ELSE
            OMEGA = ASIN(OMEGA)
        END IF
        IF(R2.GT.S2+T2) OMEGA = 3.141592654-OMEGA
        SUM = SUM+DV*SQRT(SXR2)*OMEGA
    ENDIF
60  CONTINUE
    SUM = SUM/RMAX-RMAX*RMIN
    SUMT = SUMT+DOT*SUM
    GO TO 100
C
    Use an exact integrtrion for colinear edges.

```

```

70 CONTINUE
   SE = (V1(1,I+1)-V2(1,J))**2 + (V1(2,I+1)-V2(2,J))**2
   -   + (V1(3,I+1)-V2(3,J))**2
   TE = (V1(1,I)-V2(1,J+1))**2 + (V1(2,I)-V2(2,J+1))**2
   -   + (V1(3,I)-V2(3,J+1))**2
   IF(SE .LT. EPS0 .AND. TE .LT. EPS0) THEN
C       Identical edges.
       SUM = H*(R2*LOG(R2)-R2)-R2
   ELSE
C       Non-identical edges.
       IF(SE .GT. EPS0) SUM = SE-SE*LOG(SE)
       IF(TE .GT. EPS0) SUM = SUM+TE-TE*LOG(TE)
       SD = (V1(1,I)-V2(1,J))**2 + (V1(2,I)-V2(2,J))**2
   -     + (V1(3,I)-V2(3,J))**2
       IF(SD .GT. EPS0) SUM = SUM+SD*LOG(SD)-SD
       TD = (V1(1,I+1)-V2(1,J+1))**2 + (V1(2,I+1)-V2(2,J+1))**2
   -     + (V1(3,I+1)-V2(3,J+1))**2
       IF(TD .GT. EPS0) SUM = SUM+TD*LOG(TD)-TD
       SUM = 0.25*SUM-RMAX*RMIN
   END IF
   SUMT = SUMT+DOT*SUM
   GO TO 100
C       LI method for relatively distant edges.
80 CONTINUE
C       Determine number of edge divisions.
   IF(TEST1.GT.TEST(1)) THEN
       ND = 1
   ELSE
       ND = 2
   END IF
   K = GINDEX(ND)
   DO 90 N=1,ND
C       Compute integration points.
       X = V1(1,I)+GX(N+K)*DL1(1,I)
       Y = V1(2,I)+GX(N+K)*DL1(2,I)
       Z = V1(3,I)+GX(N+K)*DL1(3,I)
       DV = GW(N+K)*DL1(4,I)
       DO 90 M=1,ND
C       Compute separation and integrate.
           R2 = (V2(1,J)+GX(M+K)*DL2(1,J)-X)**2
   -         + (V2(2,J)+GX(M+K)*DL2(2,J)-Y)**2
   -         + (V2(3,J)+GX(M+K)*DL2(3,J)-Z)**2
           SUM = SUM+H*DV*GW(M+K)*DL2(4,J)*LOG(R2)
90 CONTINUE
       SUMT = SUMT+DOT*SUM
100 CONTINUE
C       Divide by 2*pi.
   AF = 0.159154943*SUMT
   IF(AF.GT.0.0) GO TO 999
   IF(AF.GT.-.00005*MIN(A1,A2)) GO TO 110
   FACTOR = 0.25*FACTOR
   IF(FACTOR.GT.0.0008) GO TO 20
   write(*,*)'Negative view factor'
110 AF = MAX(AF,0.0)
999 RETURN
   END

```

```

SUBROUTINE FAC(TMTRX,A,EMIT,E,N,M,NPM,NPMX2,PRINT)
DIMENSION TMTRX(NPM,NPMX2),A(NPM),EMIT(NPM),E(NPM,NPM)
logical print
C *
C * THIS SUBROUTINE CALCULATES THE RADIATION INTERCHANGE
C * AREAS BETWEEN SURFACES IN AN ENCLOSURE.
C * RADIATION INTERCHANGE AREAS ARE HOTTEL'S SS
C * PRODUCTS. THEY ARE USED AS FOLLOWS:
C * Q FROM SURFACE I TO SURFACE J IN AN ENCLOSURE
C * IS GIVEN BY  $SS(I,J)*(E(I)-E(J))$  WHERE
C * E IS THE EMISSIVE POWER OF THE SURFACE  $SIGMA*T**4$ 
C *
C * THE ARRAY TMTRX SHOULD CONTAIN THE DIRECT EXCHANGE
C * AREAS WHEN THE SUBROUTINE IS CALLED. THAT IS,
C *  $TMTRX(I,J)$  SHOULD BE EQUAL TO  $A(I)*F(I,J)$ . ONLY THE
C * UPPER TRIANGLE NEEDS TO BE FILLED.  $F(I,J)$  REFERS TO
C * THE NORMAL DIRECT VIEW FACTOR IN RADIATION HEAT
C * EXCHANGE.  $A(I)$  IS THE AREA OF SURFACE I.
C *  $EMIT(I)$  IS THE EMITTANCE OF SURFACE I, BOTH THE A
C * AND THE EMIT ARRAYS SHOULD BE COMPLETED BEFORE THE
C * SUBROUTINE IS CALLED.
C * A REFRACTORY SURFACE IS ONE WHICH IS RADIATIVELY
C * ADIABATIC.
C *
C * IF REFRACTORY SURFACES ARE PRESENT, THEY MUST BE AT
C * THE BOTTOM OF THE ARRAY, THAT IS  $A(N+1)$  THROUGH
C *  $A(N+M)$ .
C *
C *
C * N = NUMBER OF SOURCE-SINK SURFACES
C * M = NUMBER OF REFRACTORY SURFACES
C *  $NPM=N+M$ , AND IT MUST BE LESS THAN 50
C *  $NPMX2= 2$  TIMES  $N+M$ 
C *
C * ZERO THE EXCITATION VECTOR SPACE
  NPMP1=NPM+1
  DO 1 I=1,NPM
    DO 1 J=NPMP1,NPMX2
1    TMTRX(I,J)=0.0
C * FILL IN LOWER TRIANGLE
C *
  DO 2 I= 1, NPM
    DO 2 J = I, NPM
2    TMTRX(J,I) = TMTRX(I,J)
C *
C *
C * THE EMMITANCE OF REFRACTORY SURFACES IS ZERO
C *
C * SUBTRACT THE "A OVER RO" TERM FROM THE ELEMENTS OF
C * THE MAIN DIAGONAL AND SET UP THE EXCITATION
C * VECTORS, PUT THEM IN TMTRX AFTER COLUMN NPM.
  DO 3 I=1, NPM
    IPN = I +NPM
    TMTRX(I,I) = TMTRX(I,I) - A(I)/(1.0- EMIT(I))
3    TMTRX(I,IPN) = -A(I)* EMIT(I) / (1.0 -EMIT(I))
C *

```

```

C *
C *
C *   SOLVE THE SYSTEM OF EQUATIONS
C *
      CALL SIMEQ(TMTRX,E,NPM,NPMX2,NPM,N)
C *
C *   ANSWERS ARE RETURNED IN THE E-MATRIX, THE RESPONSE
C *   VECTORS APPEAR IN THE ROWS OF THE MATRIX
C *   THERE WILL BE N RESPONSE VECTORS
C *
C *   CALCULATE TOTAL INTERCHANGE AREAS AND PUT THEM IN
C *   THE TRANSFER MATRIX
C *
      DO 4 I = 1,N
      DO 4 J = 1,N
      DELTA = 0.0
      IF (I .EQ. J) DELTA = 1.0
4   TMTRX (I,J)= A (I)*EMIT(I)/(1.0-EMIT(I))*(E(J,I)-
& DELTA*EMIT(J))
* **
* ** CALCULATE SCRIPT F'S, PUT THEM IN ARRAY E.
* **
      DO 5 I = 1,N
      DO 5 J = 1,N
5   E(I,J)=TMTRX(I,J)/A(I)
C *
C *   PRINT OUT THE INTERCHANGE AREAS
C *
      WRITE (6,200)
200  FORMAT (1H1)
      WRITE (6,201) N,M
201  FORMAT (20X,' TOTAL INTERCHANGE AREA CALCULATION '
& // 25X,'NUMBER OF SOURCE-SINKS:', I4 , //
& 25X,' NUMBER OF REFRACTORY SURFACES :', I4 ///)
      WRITE (6,202) (( I, J, TMTRX (I,J), J =
&I,N),I=1,N)
      WRITE(9,205)((I,J,TMTRX(I,J),J=I,N),I=1,N)
202  FORMAT ( 25X, 'SURFACES', 7X,'INTERCHANGE AREAS'
& /(27X,I2,'-',I2,12X, E12.4 ))
C *
C *   PRINT OUT INTERCHANGE FACTORS (SCRIPT F)
C *
      WRITE (6,203)
203  FORMAT(//// 20X,'INTERCHANGE FACTORS (SCRIPT F)')
      WRITE(6,204)((I,J,E(I,J),J=I,N),I=1,N)
204  FORMAT (25X, ' SURFACES', 7X, ' SCRIPT F ' /(27X,
& I2,'-', I2, 7X, F8.4 ))
205  format(i3,2x,i3,2x,e14.6)
      RETURN
      END

      SUBROUTINE SIMEQ(A,S,NMAX1,NMAX2,IL,LV)
C *   A = NAME OF COEFFICIENT MATRIX
C *   NMAX1 = MAXIMUM FIRST DIMENSION OF A
C *   IL = ACTUAL NUMBER OF EQUATIONS TO BE SOLVED

```

```

C *   LV = ACTUAL NUMBER OF CONSTANT VECTORS
C *   NMAX1 = MAXIMUM SECOND DIMENSION OF A
C *   SOLUTION VECTORS ARE THE ROWS OF ARRAY S(LV,IL)
C *   UPON RETURN
      DIMENSION A(NMAX1,NMAX2)
      DIMENSION S( NMAX1,NMAX1)
      JL = IL + LV
150  IH = 0
300  IH = IH + 1
      DEN = A(IH,IH)
      IF (DEN) 301,302,301
302  WRITE(6,116) IH
      RETURN
301  DO 160 J = IH,JL
160  A(IH,J) = A(IH,J) / DEN
      IT = IH + 1
      DO 200 I = IT, IL
      IF(A(I,IH)) 240,200,240
240  DEN = A(I,IH)
      DO 245 J = IH,JL
245  A(I,J) = (A(I,J)/DEN) - A(IH,J)
200  CONTINUE
      IF ( IH - IL+ 1) 300,400,400
400  DO 401 K = 1, LV
      KA = IL + K
401  S(K,IL) = A(IL,KA) / A(IL,IL)
      IX = IL -1
      DO 900 KA = 1,LV
      K = IL +KA
      DO 500 I = 1,IX
      IZ = IL - I
      SUM = A(IZ,K)
      JX = IL +1 -I
      DO 450 J = JX, IL
450  SUM = SUM - (A(IZ,J)* S(KA,J))
500  S(KA,IZ)= SUM
900  CONTINUE
116  FORMAT ( ' MAIN DIAGONAL OF ROW ', I2 , ' IS
      & ZERO' )
      RETURN
      END

```

```

REAL FUNCTION SS1(L1,S1,L2,S2)
REAL L1,L2
SS1=0.5*(SQRT((L1+S1)**2 +S2**2)
& +SQRT((L2+S2)**2 +S1**2)
& -SQRT(S1**2 +S2**2)
& -SQRT((L1+S1)**2 +(L2+S2)**2))
RETURN
END

```

```

REAL FUNCTION SS2(L1,S1,L2,S2,S3)
REAL L1,L2
SS2=0.5*(SQRT((L2+S2-S1)**2 +S3**2)
& +SQRT((L1+S1-S2)**2 +S3**2)

```

```
& -SQRT((S2-S1)**2 +S3**2)
& -SQRT((L2+S2-L1-S1)**2 +S3**2))
RETURN
END
```

Appendix E: Data Reduction and Analysis Program

```
IMPLICIT REAL(A-Z)
REAL NWALLTMP(900)
INTEGER I,J,K,L,N,PANNUM,ADPANTC(12)
INTEGER ONOFF(900,12),CNTRLWD(900,2)
DIMENSION TEMP(900,52),TIMER(900),TINTEMP(900)
DIMENSION NOZTEMP(900),OUTTEMP(900),SS(18,18),AREA(18)
DIMENSION RESIST(12),PANBTUH(900,12),QCOND(900,12)
DIMENSION CEILTEMP(900),FLORTMP(900),QRAD(900,18)
DIMENSION SWALLTMP(900),EWALLTMP(900),QRADFLUX(900,18)
DIMENSION BBEMISS(18),PSUMPWR(12),QCONDSUM(12),QRADSUM(12)
DIMENSION TEMPSUM(12),PAVGPWR(12),QCONDAVG(12),QRADAVG(12)
DIMENSION TEMPAVG(12),QCONVAVG(12),HAVG(12)
CHARACTER*80 TITLE,LINE1,LINE2,LINE3
CHARACTER*4 PLABEL(12)
CHARACTER*3 CHK1,CHK2
c   open experimental data and read it. Change file name when analyzing other data
c   don't forget to change the auxillary data name down below also.

OPEN (7,FILE='e040595b.DAT',STATUS='UNKNOWN')

READ(7,1000) TITLE

1000 FORMAT(A80)
1010 FORMAT(A3)
c   use the following format statement for e032595*.dat data
c1020 FORMAT(51F6.2)
DO 50 I=1,900
c   change * to 1020 when reading e032595*.dat data
READ(7,*) (TEMP(I,J),J=1,51)
c   WRITE(*,*) I

C   CALCULATE AIR STREAM AVERAGE TEMPERATURES

OUTTEMP(I) = (TEMP(I,44)+TEMP(I,45)+TEMP(I,46)+TEMP(I,47))/4
TINTEMP(I) = (TEMP(I,48)+TEMP(I,49)+TEMP(I,50)+TEMP(I,51))/4
NOZTEMP(I) = (TEMP(I,40)+TEMP(I,41)+TEMP(I,42)+TEMP(I,43))/4

IF (I .EQ. 900) THEN
WRITE(*,*) OUTTEMP(I),NOZTEMP(I),TINTEMP(I)
write(*,*) temp(i,48),temp(i,49),temp(i,50),temp(i,51)
END IF
READ(7,1010,END = 400) CHK1
IF (CHK1 .NE. 'TIM') THEN
WRITE(*,*) 'ERROR IN DATA, EXPECTED TIMER, FOUND ',CHK1
END IF
READ(7,*,END = 400) TIMER(I)
IF (I .EQ. 1) THEN
STARTTIM = TIMER(I)
END IF
LASTTIM = TIMER(I)
TIMER(I) = TIMER(I)-STARTTIM
READ(7,1010,END = 400) CHK2
```

```

IF (CHK2 .NE. 'CON') THEN
  WRITE(*,*) 'ERROR IN DATA, EXPECTED CONTROL, FOUND ',CHK2
END IF
READ(7,*,END = 400) (CNTRLWD(I,J),J=1,2)

READ(7,*,END = 400) (ONOFF(I,J),J=1,12)
50 CONTINUE
close(7)
C  CALCULATE AVERAGE TEMPERATURE FOR FIVE OF THE REMAINING
C  UNHEATED SURFACES
DO 38 I=1,900
C  CEILING AVERAGE TEMPERATURE
  CEILTEMP(I) = (TEMP(I,13)+TEMP(I,14)+TEMP(I,15)+TEMP(I,16)
&              +TEMP(I,17)+TEMP(I,18))/6
C  FLOOR AVERAGE TEMPERATURE
  FLORTMP(I) = (TEMP(I,19)+TEMP(I,20)+TEMP(I,21)+TEMP(I,22)
&              +TEMP(I,23)+TEMP(I,24))/6
C  SOUTH WALL AVERAGE TEMPERATURE
  SWALLTMP(I) = (TEMP(I,25)+TEMP(I,26)+TEMP(I,27)+TEMP(I,28)
&              +TEMP(I,29)+TEMP(I,30))/6
C  NORTH WALL AVERAGE TEMPERATURE
  NWALLTMP(I) = (TEMP(I,31)+TEMP(I,32)+TEMP(I,33)+TEMP(I,34)
&              +TEMP(I,35)+TEMP(I,36))/6
C  EAST WALL AVERAGE TEMPERATURE
  EWALLTMP(I) = (TEMP(I,37)+TEMP(I,38)+TEMP(I,39))/3
38 CONTINUE
C  IT IS NECESSARY TO ESTIMATE THE TEMPERATURE OF SURFACE 13
C  THIS SURFACE IS LOCATED ABOVE THE PANELS AND THE TEMPERATURE IS
C  ESTIMATED USING THE FOLLOWING EQUATION:
C  (T(I,28)+T(I,31))/2. THIS IS A CRUDE ESTIMATE AND ESTIMATES THAT
C  AN ISOTHERM RUNS AT A LEVEL 8 FEET ABOVE THE FLOOR.
DO 39 K=52,13,-1
DO 39 I = 1,900
  TEMP(I,K)=TEMP(I,K-1)
  IF (K .EQ. 13) THEN
    TEMP(I,K)=(TEMP(I,29)+TEMP(I,32))/2
  ENDIF
39 CONTINUE
C  SET THE SURFACES TO VALUES

DO 53 I=1,900
  TEMP(I,14) = CEILTEMP(I)
  TEMP(I,15) = EWALLTMP(I)
  TEMP(I,16) = FLORTMP(I)
  TEMP(I,17) = SWALLTMP(I)
  TEMP(I,18) = NWALLTMP(I)

53 CONTINUE
write(*,*)'temperature averaging completed'

c  open aux. data (manually measured and entered data) and read

open(UNIT = 8,file = 'a040595b.dat', status='unknown')
READ(8,1101) LINE1
READ(8,1101) LINE2
READ(8,1101) LINE3

```

```

READ(8,1105)ACVOLTS
WRITE(*,*) ACVOLTS
READ(8,1106)STATICP
WRITE(*,*) STATICP
READ(8,1107)DELTAP
WRITE(*,*)DELTAP
READ(8,1108)PBAROM
WRITE(*,*)PBAROM
READ(8,1109)DRYBULB
WRITE(*,*)DRYBULB
READ(8,1110)WETBULB
WRITE(*,*)WETBULB
READ(8,1102)ND7
READ(8,1102)ND3
READ(8,1103)ND1
C   IF NOZZLE IS OPEN, THEN SET DIAMETER INTO PROGRAM IN INCHES
IF(ND7 .EQ. 'u'.OR. ND7 .EQ. 'U') THEN
  ANOZD7 = 7.0
ENDIF
IF(ND3 .EQ. 'u'.OR. ND7 .EQ. 'U') THEN
  ANOZD3 = 3.0
ENDIF
IF(ND1 .EQ. 'u'.OR. ND7 .EQ. 'U') THEN
  ANOZD1 = 1.6
ENDIF
1101 FORMAT(A80)
1102 FORMAT(17X,A1)
1103 FORMAT(19X,A1)
1105 FORMAT(13X,F6.1)
1106 FORMAT(15X,F6.2)
1107 FORMAT(14X,F6.2)
1108 FORMAT(33X,F6.1)
1109 FORMAT(14X,F1.0)
1110 FORMAT(14X,F1.0)
WRITE(*,*) ANOZD7,ANOZD3,ANOZD1
close(8)
C   READ IN PANEL AREAS (FT FT.) FOR THE RADIATION CALCULATION
C   READ IN INTERCHANGE AREAS (FT^2)
OPEN(UNIT=15,FILE='AREAS.DAT',STATUS='UNKNOWN')
OPEN(UNIT=16,FILE='INTFAC.DAT',STATUS='UNKNOWN')
READ(15,*) N

DO 100 I=1,N
  READ(15,*)AREA(I)
100 CONTINUE

C   READ IN INTERCHANGE AREAS NOW
DO 250 I=1,N
  DO 250 J=I,N
    READ(16,205)K,L,SS(I,J)
250 CONTINUE
205 FORMAT(I3,2X,I3,2X,E14.6)
C   FILL IN MATRIX BELOW DIAGONAL
DO 300 I=2,N
  DO 300 J=1,I-1
300 SS(I,J)=SS(J,I)

```

```

close(15)
close(16)
C READ IN PANEL DATA (PANEL NUMBER,TC ADDRESS,RESISTANCE)

OPEN (6,FILE='PANEL.DAT',STATUS='UNKNOWN')
DO 325 I=1,12
  READ(6,1300) PANNUM,PLABEL(I),ADPANTC(I),RESIST(I)
325 CONTINUE
  close(6)
1300 FORMAT(I3,1X,A4,2X,I2,1X,F4.1)

C CALCULATE THE POWER INPUT OF EACH PANEL (V^2/PANEL RESIS)

do 352 i=1,900
  do 352 j=1,12
    if(onoff(i,j) .eq. 1) then
      panbtuh(i,j) = acvolts**2/resist(j)*3.41
    elseif(onoff(i,j) .eq. 0) then
      panbtuh(i,j) = 0.0
    endif
352 continue
  open(13,file='power.dat',status='unknown')

  do 353 i=1,900
    write(13,1700)(panbtuh(i,j),j=1,12)
1700 format(12(2x,f6.2))
353 continue
  close(13)
  write(*,*)'panel power calculation complete'
C ESTIMATE THE CONDUCTION HEAT TRANSFER FROM THE BACK OF THE
C PANEL FOR EACH TIME STEP. THE TEMPERATURE OF THE BACK OF THE
C WALL IS ESTIMATED TO BE 72F

DO 375 I=1,900
  DO 375 J = 1,12
    QCOND(I,J) = (TEMP(I,J)-72)/8.33
C 8.33 IS THE THERMAL RESISTANCE OF THE WALL, SEE APPENDIX
C OF THESIS FOR CALCULATION OF THERMAL RESISTANCE.

375 CONTINUE

  open(20,file='condflux.dat',status='unknown')
  do 377 i=1,900
    write(20,1800) (qcond(i,j),j=1,12)
1800 format(12(2x,f5.2))
377 continue
  write(*,*)'conduction calculation completed'
  close(20)
C CALCULATE THE RADIATIVE FLUX FOR EACH PANEL
C FOR EACH TIME STEP
DO 385 L=1,900
  DO 385 I=1,18
    QRAD(L,I) = 0
  DO 385 J = 1,18
C WRITE(*,*) L,I,J

```

```

BBEMISS(I) = 0.1714E-8*(TEMP(L,I)+460)**4
BBEMISS(J) = 0.1714E-8*(TEMP(L,J)+460)**4
QRAD(L,I) = QRAD(L,I)+SS(I,J)*(BBEMISS(I)-BBEMISS(J))

```

```
385 CONTINUE
```

```
DO 387 I = 1,900
```

```
DO 387 J = 1,12
```

```
QRADFLUX(I,J) = QRAD(I,J)/AREA(J)
```

```
387 CONTINUE
```

```
OPEN(2,FILE='RADFLUX.DAT',STATUS='UNKNOWN')
```

```
DO 388 I = 1,900
```

```
WRITE(2,1999) (QRADFLUX(I,J),J=1,12)
```

```
388 CONTINUE
```

```
1999 FORMAT(F8.2,12(3X,F10.2))
```

```
close(2)
```

```
write(*,*)'radiation flux calculation completed'
```

```
c compute the convective heat transfer component
```

```
C OUTPUT CONVECTIVE HEAT TRANSFER
```

```
c OPEN(4,FILE='TEMPERAT.DAT',STATUS='UNKNOWN')
```

```
c WRITE(4,*) 'PANEL TEMPERATURES AND AIR TEMPERATURES'
```

```
c DO 395 I = 1,900
```

```
c WRITE(4,2001) NOZTEMP(I),OUTTEMP(I),TINTEMP(I),(TEMP(I,J),J=1,12)
```

```
c 395 CONTINUE
```

```
WRITE(*,*)'COMPUTATIONS COMPLETE!'
```

```
c close(4)
```

```
2001 FORMAT(3F6.2,12(2X,F6.2))
```

```
C AVERAGE HEAT FLUXES AND TEMPERATURES IN ANTICIPATION OF
```

```
C CALCULATING CONVECTIVE HEAT TRANSFER AND FILM COEFFICIENT
```

```
DO 700 J=1,12
```

```
DO 700 I=540,900
```

```
PSUMPWR(J)=PSUMPWR(J)+PANBTUH(I,J)
```

```
QCONDSUM(J)=QCONDSUM(J)+QCOND(I,J)
```

```
QRADSUM(J) = QRADSUM(J)+QRADFLUX(I,J)
```

```
TEMPSUM(J) = TEMPSUM(J)+TEMP(I,J)
```

```
700 CONTINUE
```

```
DO 810 J = 1,12
```

```
PAVGPWR(J) = PSUMPWR(J)/360
```

```
QCONDAVG(J) = QCONDSUM(J)/360
```

```
QRADAVG(J) = QRADSUM(J)/360
```

```
TEMPAVG(J) = TEMPSUM(J)/360
```

```
810 CONTINUE
```

```
DO 815 I=540,900
```

```
NOZSUM = NOZSUM+NOZTEMP(I)
```

```
OUTSUM = OUTSUM+OUTTEMP(I)
```

```
TINSUM = TINSUM+TINTEMP(I)
```

```
815 CONTINUE
```

```
NOZAVG = NOZSUM/360
```

```

TINAVG = TINSUM/360
OUTAVG = OUTSUM/360

DO 820 J=1,12
  QCONVAVG(J) = (PAVGPWR(J)-QCONDAVG(J)-QRADAVG(J))/AREA(J)
  HAVG(J) = QCONVAVG(J)/(TEMPAVG(J)-OUTAVG)
820 CONTINUE
  DO 830 J=1,12
    TOTHT=TOTHT+HAVG(J)
830 CONTINUE
  TOTHAVG=TOTHT/12
C  CALCULATE AIR FLOW RATE SUPPLIED TO THE ROOM
  R = 53.35
  TW=TINAVG
  PB=PBAROM/68.95*2.036
  PE=.000296*TW**2-.0159*TW+.41
  PP=PE-PB*(OUTAVG-TW)/2700
  RHOR=70.73*(PB-.378*PP)/(R*(OUTAVG+459.7))
  RHOI=RHOR*((OUTAVG+459.7)/(TINAVG+459.7))*((STATICP+13.63*
&  PB)/(13.63*PB))
  A1=3.14159*ANOZD7**2/(4*144)
  A2=3.14159*ANOZD3**2/(4*144)
  A3=3.14159*ANOZD1**2/(4*144)
  RE1=1363000*ANOZD7/144*(DELTAP*RHOI)**.5
  RE2=1363000*ANOZD3/144*(DELTAP*RHOI)**.5
  RE3=1363000*ANOZD1/144*(DELTAP*RHOI)**.5
  IF (RE1 .EQ. 0) THEN
    RE1=1E-6
  ENDIF
  IF (RE2 .EQ. 0) THEN
    RE2=1E-6
  ENDIF
  IF (RE3 .EQ. 0) THEN
    RE3=1E-6
  ENDIF

  C1=.9986-6.688/RE1**.5+131.5/RE1
  C2=.9986-6.688/RE2**.5+131.5/RE2
  C3=.9986-6.688/RE3**.5+131.5/RE3
  ALPHA=1-(5.2*DELTAP)/(RHOI*R*(TINAVG+459.7))
  Y=1-.548*(1-ALPHA)
  Q=1096*Y*(DELTAP/RHOI)**.5*(C1*A1+C2*A2+C3*A3)
  ACH=Q/(12*16*10)*60
  MDOT=Q*RHOI*60
  VELIN=Q/4
  ENERGY=MDOT*0.24*(OUTAVG-TINAVG)
  OPEN(22,FILE='RESULTS.OUT',STATUS='UNKNOWN')
  WRITE(22,*)'COMPUTED AVERAGES'
  WRITE(22,4000)
4000 FORMAT(25X,'PANEL NUMBER')
  WRITE(22,4001) (J,J=1,12)
4001 FORMAT(12(6X,I2))
  WRITE(22,4002)(TEMPAVG(J),J=1,12)
4002 FORMAT(12F7.2)
  WRITE(22,4002) (PAVGPWR(J),J=1,12)
  WRITE(22,4002) (QRADAVG(J),J=1,12)

```

```
WRITE(22,4002) (QCONDAVG(J),J=1,12)
WRITE(22,4002) (QCONVAVG(J),J=1,12)
WRITE(22,4002) (HAVG(J),J=1,12)
WRITE(22,4003) NOZAVG,TINAVG,OUTAVG,TOTHAVG,Q,ACH,MDOT,VELIN,
& ENERGY
WRITE(*,*) Q,ACH,MDOT,VELIN
4003 FORMAT('NOZZLE AVERAGE TEMPERATURE: ',F7.2,/,
& 'ROOM INLET AVERAGE TEMPERATURE: ',F7.2,/,
& 'ROOM OUTLET AVERAGE TEMPERATURE: ',F7.2,/,
& 'WALL AVERAGE FILM COEFFICIENT: ',F4.2,/,
& 'VOLUMETRIC FLOW RATE: ',F7.4,' CFM',/,
& 'AIR CHANGES PER HOUR: ',F7.4,/,
& 'MASS FLOW RATE (LBm/HR): ',F9.4,/,
& 'VELOCITY OF INLET (FT/MIN): ',F8.4,/,
& 'ROOM ENERGY BALANCE (BTU/HR): ',F9.3)
400 END
```

Appendix F: 15 ACH Experimental Averages

Panel No.	Panel Temp (F)	Panel Power (Btu/hr)	Radiative Flux (Btu/hr)	Conductive Flux (Btu/hr)	Convective Flux (Btu/hr-ft ²)	h (Btu/hr-ft ² -F)
1	95.86	470.48	17.86	2.84	56.22	1.85
2	105.21	473.73	28.21	3.96	55.20	1.39
3	105.19	460.08	28.55	3.96	53.45	1.35
4	104.87	477.76	27.79	3.92	55.76	1.42
5	105.19	447.64	28.41	3.96	51.91	1.31
6	105.04	397.04	28.56	3.94	45.57	1.15
7	103.34	476.53	26.13	3.74	55.83	1.48
8	105.16	439.18	28.39	3.96	50.85	1.28
9	105.17	470.12	28.72	3.96	54.68	1.38
10	104.42	481.48	27.18	3.87	56.30	1.45
11	105.21	389.52	28.27	3.96	44.66	1.13
12	103.00	464.59	26.18	3.70	54.34	1.45

Power Input to Room (Btu/hr):		5448.15
Air Heat Gain (Btu/hr):	5003.62	
Air Temp at Nozzle Bank (F):		50.35
Air Temp at Room Inlet (F):	56.29	
Air Temp at Room Outlet (F):		65.55
Film Coefficient for Wall (Btu/hr-ft ² -F):	1.39	
Volumetric Flow Rate (cfm)	488.29	
Velocity at Inlet (fpm):	122.07	
Mass Flow Rate (lb _m /hr):	2250.81	
Air Changes per Hour:	15.26	

Appendix G: 20 ACH Experimental Averages

Panel No.	Panel Temp (F)	Panel Power (Btu/hr)	Radiative Flux (Btu/hr)	Conductive Flux (Btu/hr)	Convective Flux (Btu/hr-ft ²)	h (Btu/hr-ft ² -F)
1	94.46	470.48	18.26	2.67	56.19	1.86
2	104.39	480.86	29.15	3.86	55.98	1.40
3	104.43	483.26	29.51	3.87	56.24	1.40
4	104.22	479.53	28.98	3.84	55.84	1.40
5	104.58	481.39	29.59	3.89	55.99	1.39
6	104.81	448.33	30.11	3.91	51.79	1.28
7	102.42	476.53	27.02	3.63	55.74	1.46
8	105.07	466.07	30.16	3.95	54.00	1.32
9	105.22	466.57	30.59	3.96	54.00	1.32
10	102.31	481.48	26.79	3.61	56.38	1.48
11	105.09	435.77	30.02	3.95	50.23	1.23
12	101.88	464.59	26.79	3.56	54.28	1.44

Power Input to Room (Btu/hr): 5634.86
 Air Heat Gain (Btu/hr): 6111.42
 Air Temp at Nozzle Bank (F): 50.75
 Air Temp at Room Inlet (F): 55.80
 Air Temp at Room Outlet (F): 64.30
 Film Coefficient for Wall (Btu/hr-ft²-F): 1.41
 Volumetric Flow Rate (cfm) 649.92
 Velocity at Inlet (fpm): 162.48
 Mass Flow Rate (lb_m/hr): 2993.88
 Air Changes per Hour: 20.31

Appendix H: 25 ACH Experimental Averages

Panel No.	Panel Temp (F)	Panel Power (Btu/hr)	Radiative Flux (Btu/hr)	Conductive Flux (Btu/hr)	Convective Flux (Btu/hr-ft ²)	h (Btu/hr-ft ² -F)
1	95.34	480.70	18.04	2.78	57.48	2.02
2	105.09	480.38	28.72	3.95	55.96	1.46
3	104.83	481.00	28.68	3.92	56.05	1.48
4	104.96	450.16	28.62	3.93	52.20	1.37
5	105.00	453.73	28.81	3.94	52.62	1.38
6	105.06	417.50	29.09	3.94	48.06	1.26
7	104.86	485.07	28.53	3.92	56.58	1.49
8	104.85	434.06	28.66	3.92	50.19	1.32
9	105.01	445.88	29.06	3.94	51.61	1.35
10	102.99	493.75	26.35	3.70	57.96	1.61
11	104.86	419.79	28.54	3.92	48.42	1.27
12	102.86	476.43	26.56	3.68	55.77	1.55

Power Input to Room (Btu/hr):		5518.46
Air Heat Gain (Btu/hr):	5527.50	
Air Temp at Nozzle Bank (F):		56.71
Air Temp at Room Inlet (F):	60.29	
Air Temp at Room Outlet (F):		66.88
Film Coefficient for Wall (Btu/hr-ft ² -F):	1.46	
Volumetric Flow Rate (cfm)	762.20	
Velocity at Inlet (fpm):	190.50	
Mass Flow Rate (lb _m /hr):	3497.00	
Air Changes per Hour:	24.20	

VITA

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Master of Science

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