

DESIGN, CONSTRUCTION AND CALIBRATION
OF A MODEL CHAMBER FOR EXPERIMENTATION WITH
THERMALLY STRATIFIED COOLING SYSTEMS

by

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TABLE OF CONTENTS

<u>Chapter</u>	<u>Page</u>
I. INTRODUCTION	1
II. PREVIOUS INVESTIGATIONS	4
III. DESCRIPTION OF THERMAL STRATIFICATION PROCESS . .	7
IV. PHYSICAL DESCRIPTION OF FACILITY	11
V. DESCRIPTION OF FACILITY	14
Features Inside the Model	14
Instrumentation	19
Data Reduction	25
VI. SYSTEM CALIBRATION	33
VII. CONCLUSIONS	39
LIST OF REFERENCES	40
Appendix A	43
Appendix B	47
Appendix C	52
Appendix D	61
Appendix E	75
ACKNOWLEDGEMENTS	
VITA	

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1.	Plant layout of thermally stratified cooling system . .	8
2.	Stratification process temperature pattern and load distribution	9
3.	Physical description of model chamber	12
4.	Inside view of the model	15
5.	Supply air duct perforations positions	16
6.	The position of perforations on return and ventilation air ducts	18
7.	Top view of the light bulbs positions and thermocouple posts (A-E)	20
8.	Roof heating panels, ventilation ducts, lights, and supply air ducts	21
9.	Positions of thermocouple posts, floor heating panels, and supply air ducts	22
10.	Side view of the chamber with instruments	24
11.	Temperature stratification profile for RUN 1	35
12.	Temperature stratification profile for RUN 2	36
13.	The description of an annubar	45
14.	The description of a heating panel	46
15.	Potentiometer calibration chart	53
16.	Air density chart	58
17.	Annubar calibration chart	60
18.	A front view of the preliminary model	76
19.	Isometric view of the small model chamber	77

LIST OF TABLES

<u>Table</u>		<u>Page</u>
1.	Calculations of uncertainties in RUN 1	37
2.	Calculations of uncertainties in RUN 2	38
3.	Barometric pressure corrections	57

**THIS BOOK
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CHAPTER I

INTRODUCTION

Concern with energy availability and with the cost of total volume air-conditioning in high ceiling industrial plants has generated an interest in thermal stratified cooling systems. This system, also known as "vertically-zoned," or "low level system," involves cooling only of the lower, occupied space while leaving the upper space uncooled. It provides satisfactory comfort for the workers and significant savings in installed and operating costs of the air-conditioning system. This system has not been extensively used, primarily because of complexity of design and lack of complete design information. However, successful systems do exist (5).

In the process of thermal stratification, the high temperature air surrounding the internal heat sources rises up gradually due to its bouyancy. This air accumulates near the ceiling, in the upper strata, preventing or reducing the transfer of roof heat load into the plant. This will result in a reduction in the plant cooling lead.

Interaction of load sources such as roof, lights, and internal loads will produce a thermal stratification temperature pattern which has relatively low, uniform temperature in the lower occupied level while the temperature in the upper level increases with height. Temperature rise of stratified air in the upper level has been observed to generally range from 1 to 2 °F per foot of height above the lower occupied level. The upper level can be ventilated to remove a considerable portion of the roof heat gain and sensible heat from lights (6). However, some designers

disagree with this technique and prefer to keep the upper level as motionless as possible (8).

The aim of this work was to design and build a structure to obtain data on thermal loads and temperature distributions as functions of geometric and thermal variables. Because the numerous variables (ceiling height, light height, supply air height, range of roof thermal load, floor and lights loads, supply flow rate, supply air temperature, ventilation flow rate, and return air height) are hard to control for system modification in an actual plant and also because full scale experiments are expensive in cost and time, scale model techniques were considered for this study. The model studies will provide experimental temperature distributions under the actual range of operating conditions which occur in industrial plants. These temperature distributions will be used for predicting thermal behavior in prototype or full scale enclosures.

The purpose of this thesis is to present a description of the design, construction, and calibration of a physical model of a high ceiling industrial plant to be used to provide design information about stratified cooling systems. The design information will include temperature distributions in the cooled zone and in the strata, and plant cooling loads as functions of ceiling height, light height, roof thermal load, light load, floor load, supply height, supply rates, supply air temperature, return air height and ventilation rates.

To obtain some design information prior to final design of the main model, a preliminary model 3' x 1' x 2' high was constructed and tested over a limited range of loads. A more detailed description of the preliminary model is presented in the Appendix.

The main chamber is 8' x 8' x 8' high, built of 2 x 4 stud construction with the stud space fully insulated. Electric heating panels were used to supply the thermal load at the roof and floor. The roof, lights, and supply air ducts could be adjusted to any level within the 8-foot high structure. Thermocouples were placed at the floor and every one foot above the floor and at the roof.

After design and construction were completed the model was tested to verify measurement techniques and to determine energy losses. In the following chapters, expected range of loads in real factories and simulated range of loads in model chamber are discussed. Also, chamber construction, instrumentation, and reduction of data are described. A discussion of thermal similarity of the model to a real industrial plant (prototype), which allows interpretation of results and extrapolation of model data to other systems, is presented in the Appendix.

CHAPTER II

PREVIOUS INVESTIGATIONS

During the recent decade the process of thermal stratification by natural convection heat transfer in lakes and liquid containers has been the subject of numerous analytical and experimental investigations. Winter and Schoenhals (1) analyzed and experimented with stratification of a contained fluid under transient natural convection heat transfer. Armaly and Lepper (2) presented an article concerning diurnal temperature distribution in a deep water impoundment exposed to heating and cooling loads at the surface. In their investigation, they used models to simulate the diurnal behavior and to demonstrate the stratification developments of the reservoir. Also Dake and Harleman (3) studied thermal stratification in lakes theoretically and, for verification of the theory, model techniques were used. They concluded that the development of the stratification phenomena may be simulated under laboratory conditions. The previously named references provided some information as background for the present study since use of a scale model was considered for development of understanding of the basic phenomena.

The idea of industrial building air-conditioning by the use of a thermally stratified cooling system is not new. There have been many attempts to cool the lower level of high ceiling factories with uniform worker population and successful systems with significant reductions in the cooling loads have been reported (4-11).

Beier (4) presented a theoretical analysis of the thermal stratification process. In his study, he considered lights as the only internal heat load sources which affect the stratification process. He also

concluded that the cooling load can be reduced by exhausting ventilation air through the roof. Beier and Gorton (5) continued this investigation and presented a load calculation method for design of air-conditioning system discussing the influence of the thermal loads, ventilation, and ceiling height on temperature stratification pattern.

Dralle (6) and Olivieri (7) reported of significant reduction in energy load through usage of thermal stratification and using of ventilation exhaust fans at the roof to remove a portion of the roof heat load and convection heat from lights. Dralle (6) also presented interesting case studies of some industrial plants and estimated that 70% of the roof heat gain and 50% of convected heat from lights would be removed by ventilation of the upper level, while Dean and Reese (8) stated that the upper level air should be kept motionless.

Allen (9) presented the design of an air-conditioning system for building modification of a major industrial plant to recognize the effect of stratification upon the cooling load.

In search of a new design concept for air-conditioning of industrial buildings to save energy, Holness (10) evaluated several different techniques. Thermal stratification with low level air supply providing comfortable conditions at the working zone was one of the systems to which he gave his major consideration. He described low level supply outlets, located 12 feet above the floor, which he believes can reduce the normal mixing action of air and, thus which can accentuate the stratification profile. In his analysis he used 3 cfm/ft^2 to be a conventional supply air flow rate.

The effect of supply and return outlet locations on stratification were investigated by Olivieri and Singh (11). For their study they

constructed a 1/3 scale model chamber and tested it at the ranges of 8.33 watts/ft² and 4.16 watts/ft² of roof load, 3.65 watts/ft² and 2.08 watts/ft² of floor load, and 8.2 watts/ft² and 4.1 watts/ft² of ceiling lights load. In their conclusions, they stated that most effective stratification is obtained by either using a supply-return combination diffuser or by introducing the supply air at the floor level and returning from the floor at a very low velocity.

Parczewski and Renzi (12), Katzoff (13), Vickers (14), and Thomas (15) discuss the thermal similarity criteria between a model and its prototype. Their studies were helpful in determination of the thermal similarity of the model chamber to a prototype enclosure.

Using some of the design parameters selected from the information obtained from these previous investigations, the model chamber was constructed.

CHAPTER III

DESCRIPTION OF THERMAL STRATIFICATION PROCESS

From the previous investigations of thermal stratification, a condensed description of the stratification process is presented here to provide some basic ideas about this phenomena as it occurs in high ceiling industrial plants.

In a stratified cooling system, the space is divided into two parts, a lower, air conditioned zone of practically uniform temperature and an upper zone of higher, varying temperatures. These two zones are pictured in Figure 1.

Figure 2 describes the distribution of load components and resulting stratified temperature pattern in an industrial plant. In the stratification process, the heat gained at the roof is radiated to the floor or is stored in the roof structure, and convection transfers the remaining part of the roof heat load into the air of the upper strata. The high temperature of the upper strata tends to block the convective roof heat load from entering the space.

The floor heat load partly convects to the air conditioned zone and the rest is stored in the floor or is conducted into the ground. The earth acts as a heat sink since far below ground level its temperature is uniform.

The lighting load divides into two portions; one portion radiates to the floor and the other portion is convected to the surrounding air. The air surrounding the lights becomes heated and rises to the upper strata due to the decrease in its density. The heated air moves up to the ceiling and then spreads out uniformly. Air circulation caused by the thermal plume above the lights causes mixing of the warm, upper air with the cool,

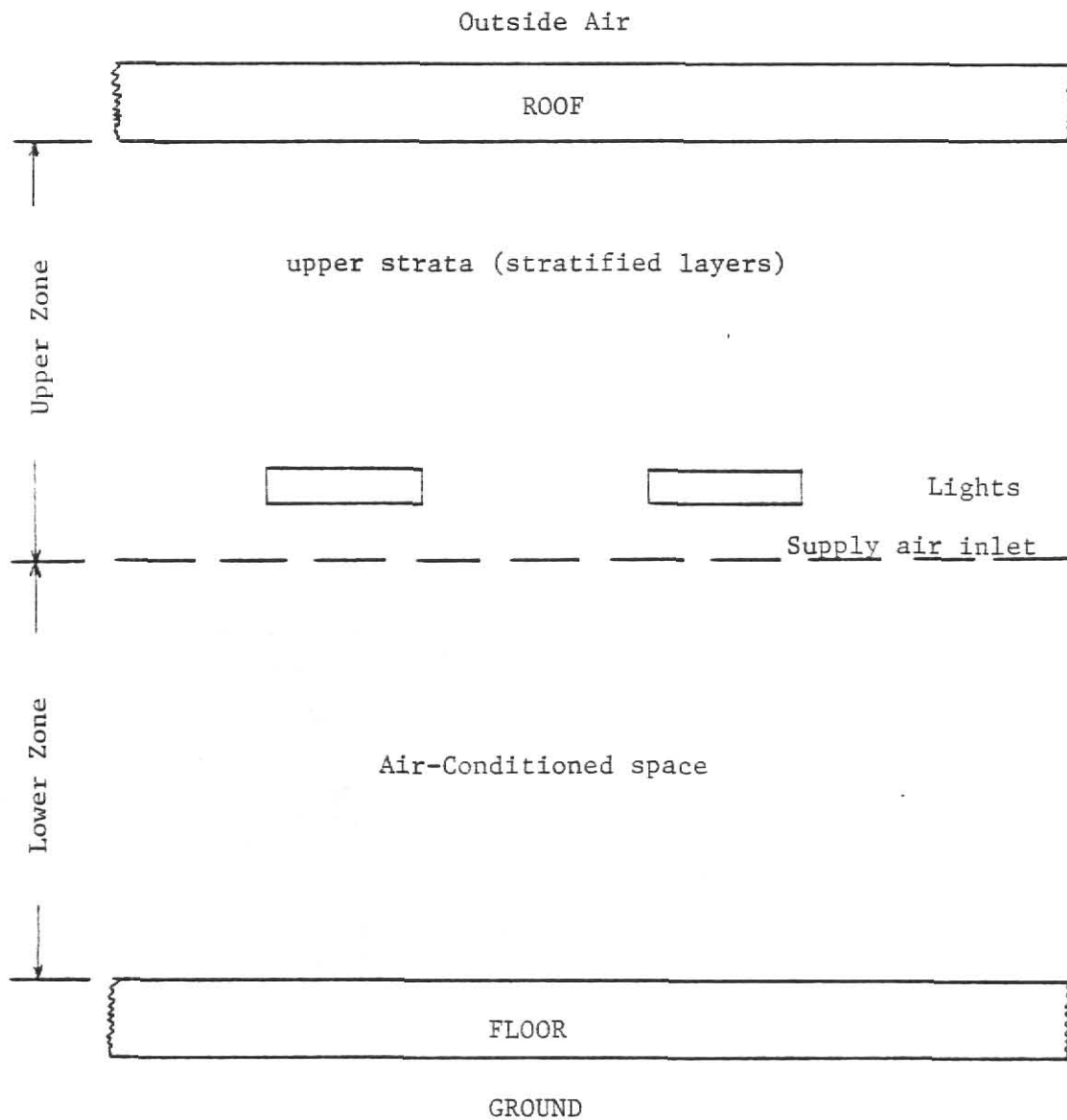


Figure 1. Plant layout of thermally stratified cooling system.

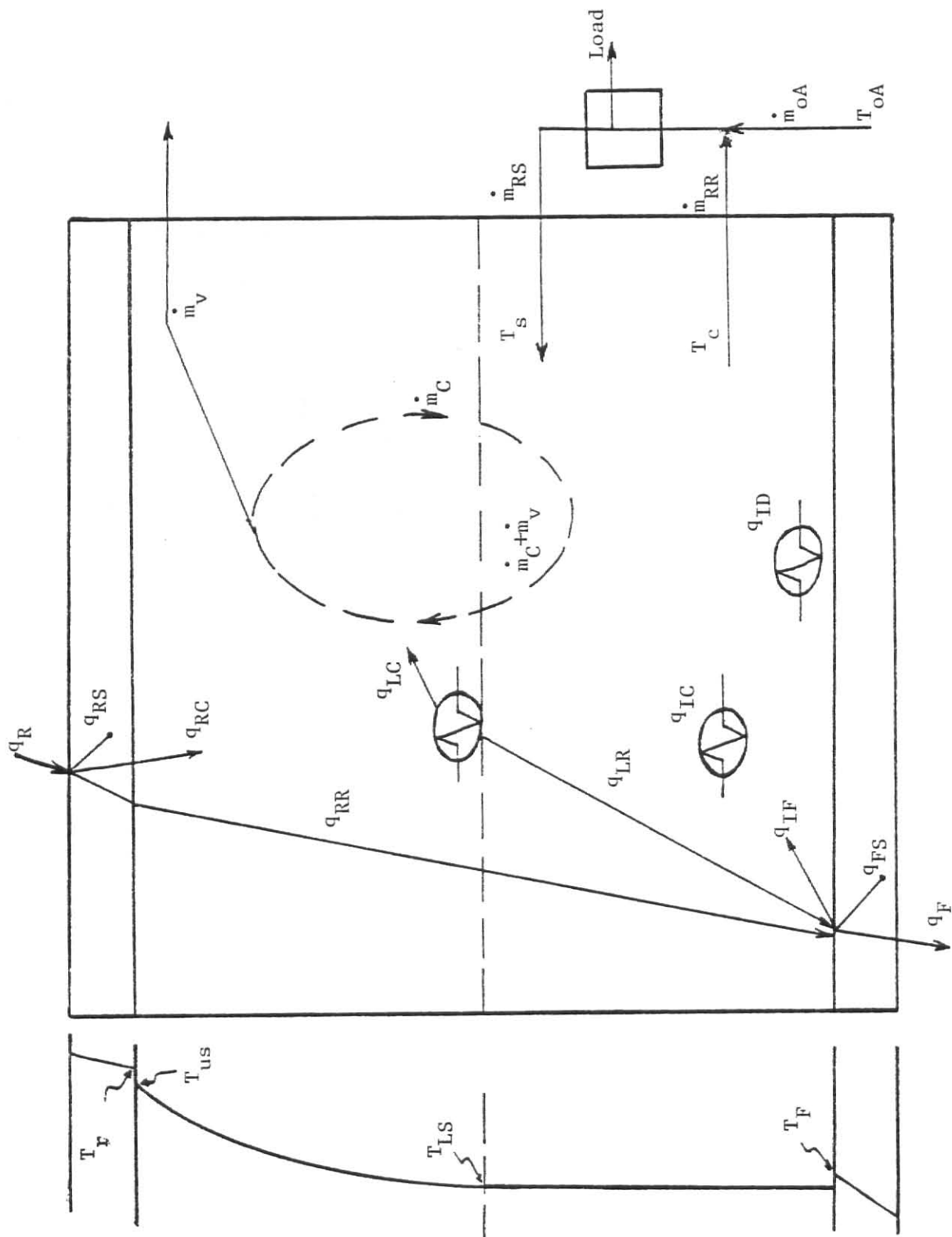


Figure 2. Stratification process temperature pattern and load distribution.

lower zone air. The ventilation of the upper level can reduce the intrusion of upper zone hot air into the lower conditioned level.

In the occupied zone, the hot air rising from the internal heat sources is neutralized by mixing with cool supply air.

CHAPTER IV

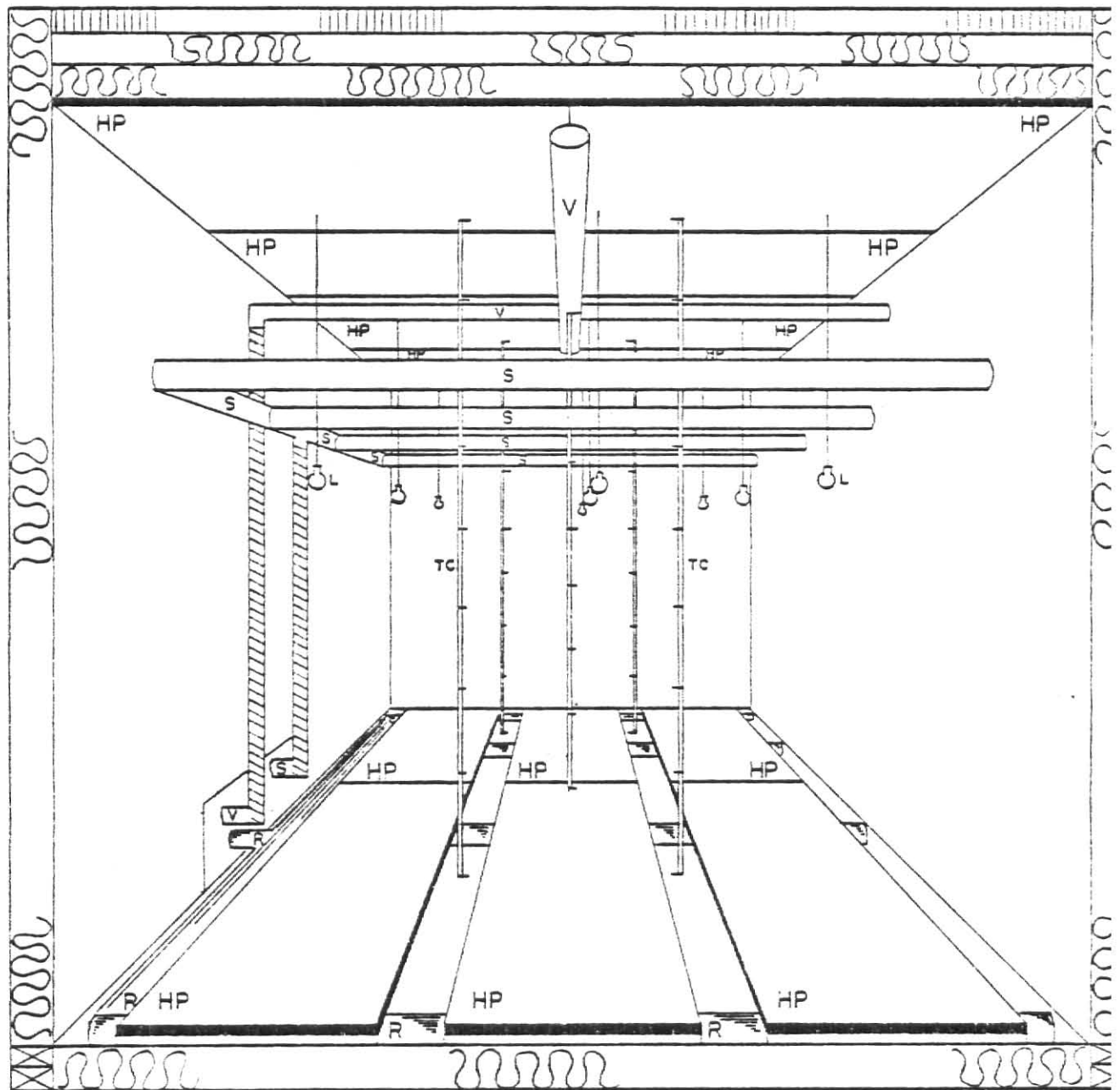
PHYSICAL DESCRIPTION OF FACILITY

To provide a facility for simulation of the thermal stratification process in a high ceiling industrial plant it was necessary to construct a scale model. The model is an 8' x 8' x 8' high chamber, built with nominal 2 x 4 wood studs at 16 in. o.c., with the stud space fully insulated. The insulation material used in construction of the chamber was 3" thick fiberglass with thermal resistance of 11. The wall insulation and studs were covered with a thick plastic sheet. Then 1/8 inch hardboard was used for the inside and outside surfaces of the walls. The wall thermal transmittance (U-value) was calculated to be 0.08 Btu/hr-ft²-F. A perspective view of the model chamber is shown in Figure 3.

The roof was constructed with eight 2' x 4' heating panels resting on angle supports. The panels were covered with two layers of 3-inch fiberglass insulation, one layer of plastic sheet and one layer of polystyrene insulation with thermal resistance of 16. The plastic sheet was taped to the surrounding walls to eliminate air leakage through the roof.

The chamber floor was built with the same type of construction materials used for the walls with the addition of 1/2-inch plywood on the floor surface. The floor heating panels were located 6 inches above the floor on small wooden studs.

In the middle of the front wall, a 27" x 78" access door with a glass window (20" x 20") was built. The door had the same thickness as the walls and was of similar construction materials.



HP _ HEATING PANEL

L _ LIGHT

TC _ THERMOCOUPLE

S _ SUPPLY

R _ RETURN

V _ VENT

Figure 3. Physical description of model chamber.

The construction materials of the model chamber were selected because of ease in fabrication, durability, low thermal conductivity, light weight, and availability.

CHAPTER V

DESCRIPTION OF FACILITY

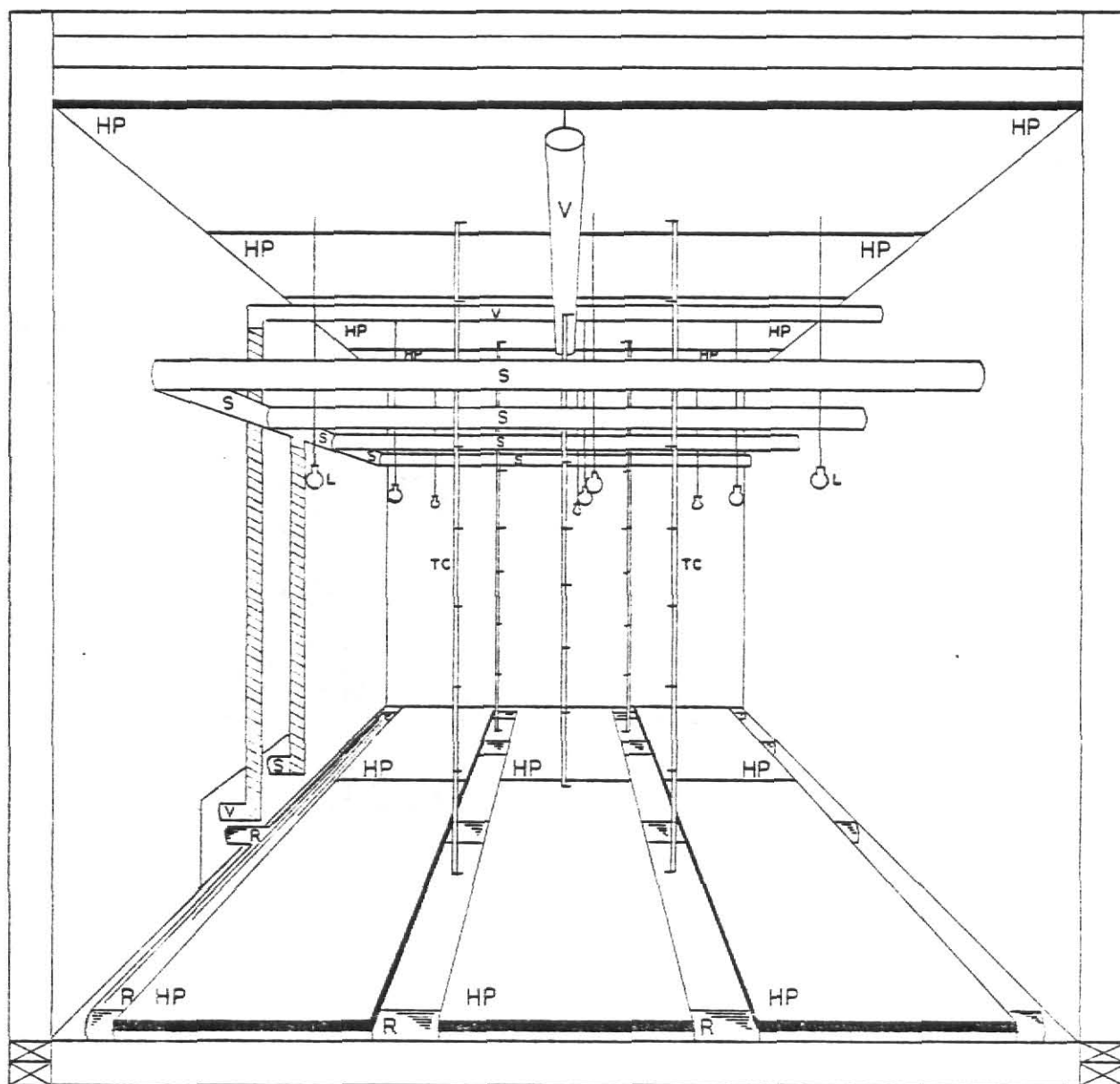
The description of the facility is discussed in three sections: the features inside the model, instrumentation, and data reduction. The major consideration throughout the construction of the model chamber was to obtain equal distribution of thermal loads and uniform air distribution.

FEATURES INSIDE THE MODEL

The main characteristics of the model were the features used inside the model such as: supply, return and ventilation exhaust air ducts, roof and floor heating panels, lights, and thermocouples. Figure 4 shows the inside view of the model, and the following is the description of these features.

Supply air ducts consisted of four 7' long, 4-inch diameter horizontally parallel commercial poly-vinyl chloride [(PVC) ASTM 2729 Grade C] white pipes, adjustable to any height within 7-feet. Perforations (9/32 in. diameter) through the duct were positioned at 4 inch increments along the axis. To obtain good air distribution two openings were located horizontally at 180° apart and four more below at the interval of 30° and 1" to the right of previous openings, as shown in Figure 5. Volume flow rate of supply air was selected to range up to 3.2 cfm/ft², and could be controlled by a gate valve, and measured with an annubar and two manometers.

Return air ducts were composed of four 7'-2" long, 3-inch diameter horizontally parallel ABS black schedule 40 pipes. The openings on the pipe were 3/8" diameter, located horizontally at 180° apart every four



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Figure 4. Inside view of the model.

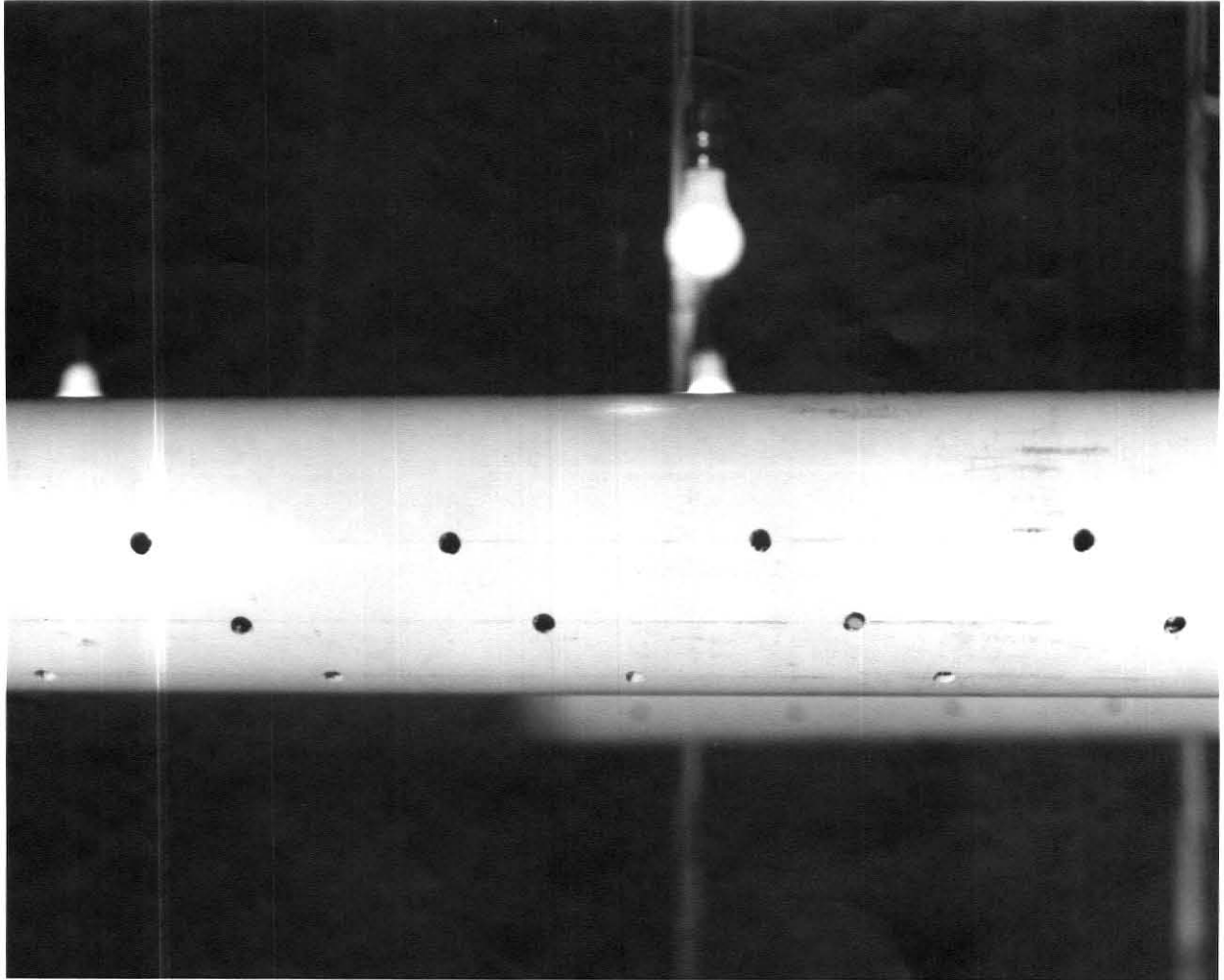


Figure 5. Supply air duct perforations positions.

inches along the axis. Figure 6a shows the perforations on the return air ducts. These ducts were positioned initially on the floor as in Figure 4, but they could be moved to any level within 7 feet. The return air flow rate was also controlled by a gate valve and measured with an annubar and two manometers.

Ventilation exhaust air ducts were made of two 7' long perpendicularly crossed horizontal PVC (ASTM 2729 Grade C) white pipes. The duct diameter is 4 inches with two 9/32" diameter perforations at 180° apart located horizontally at 4 inches of increments, as shown in Figure 6b. Flow rate of ventilated air was adjustable and could be controlled to be zero to 100% of supply air flow rate.

PVC and ABS pipes were selected for the means of supply, return, and ventilating air because the interior is smooth and the material is light and easy to work with. Also 4-inch white flexible pipes were used at the entrance of supply and ventilation air ducts into the chamber for easier raising or lowering the main horizontal supply and ventilation exhaust ducts.

The roof was built with eight 2' x 4' electric heating panels adjustable to any height within the maximum of seven feet from the floor. The panels were supported by four aluminum angles and a horizontal T-bar in the middle. The T-bar was hung by a steel wire which was tied to a 2' x 10' wood beam above the roof and supported by the two side walls. The aluminum angles were screwed to the chamber walls. These heating panels were used to simulate the roof thermal load in an industrial plant.

Six 2' x 4' electric heating panels were positioned near the floor to simulate the internal disperse loads. The panels used on the floor

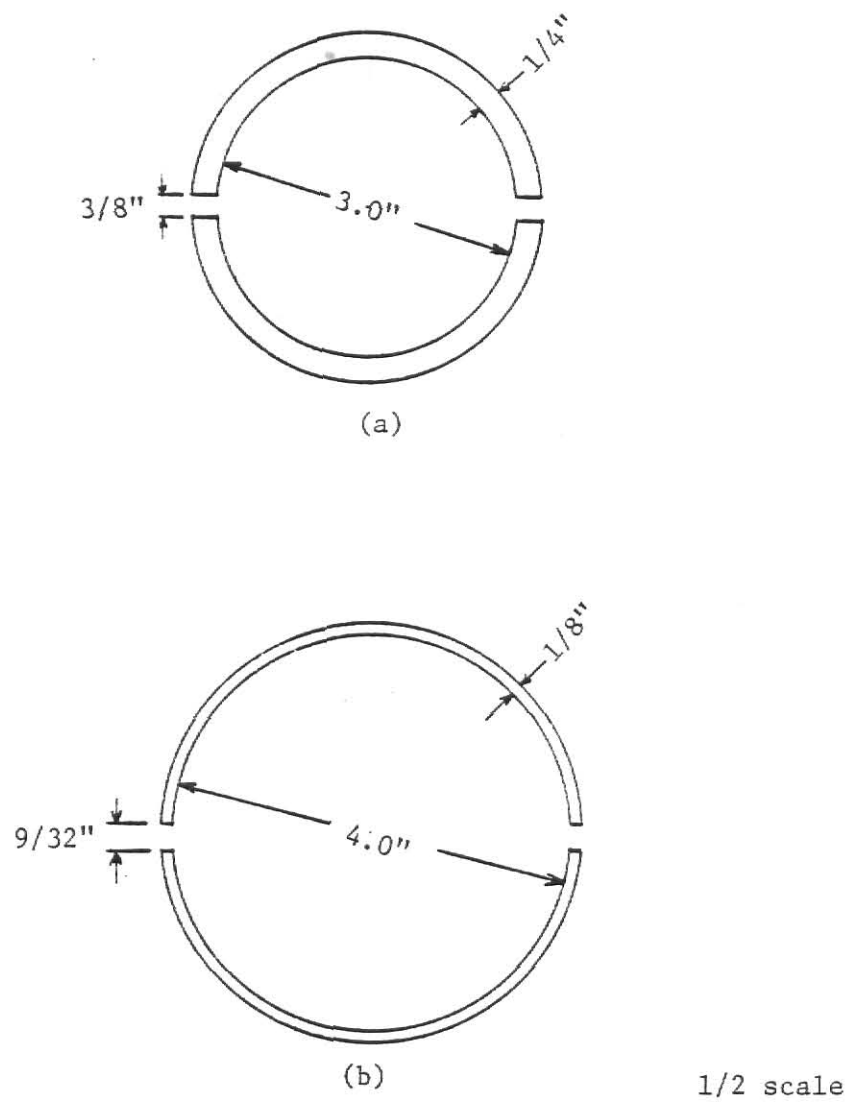


Figure 6. The position of perforations on return (a) and ventilation (b) air ducts.

were of the same type as the ones for the roof. The description of a heating panel is given in the appendix.

Nine 40-watt incandescent light bulbs were located inside the chamber to provide equally distributed lighting. The light bulbs were set 32 inches apart, with the bulbs at the corners 16 inches away from the walls. The height of the bulbs was variable. A top view of the light bulbs positions is shown in Figure 7. The power input to the lights was selected to simulate the lighting load in a factory. Figure 8 pictures the roof heating panels along with ventilation ducts, lights, and supply air ducts in the chamber.

The electric power input to the roof and floor heating panels and light bulbs was designed to range from zero to 13.2, 8.7, and 4.2 watts/ft², respectively. These were controlled by separate watt-meters and variacs.

For purposes of measuring the temperature inside the chamber, copper-constantan thermocouples were positioned on five wooden posts forming a square with one post in the center. They were adjusted so that temperatures could be measured from floor to the roof in one foot increments. The thermocouple located at the floor was numbered 1 and the thermocouple near the roof, 8. For convenience in room air temperature measurements, the thermocouple posts were recognized by letters A through E, as shown in Figure 7. Figure 9 pictures the thermocouple posts with the heating panels and supply air ducts in the chamber.

INSTRUMENTATION

The equipment was selected for purposes of supplying cool air to the chamber, discharging warm air from the chamber and controlling the power

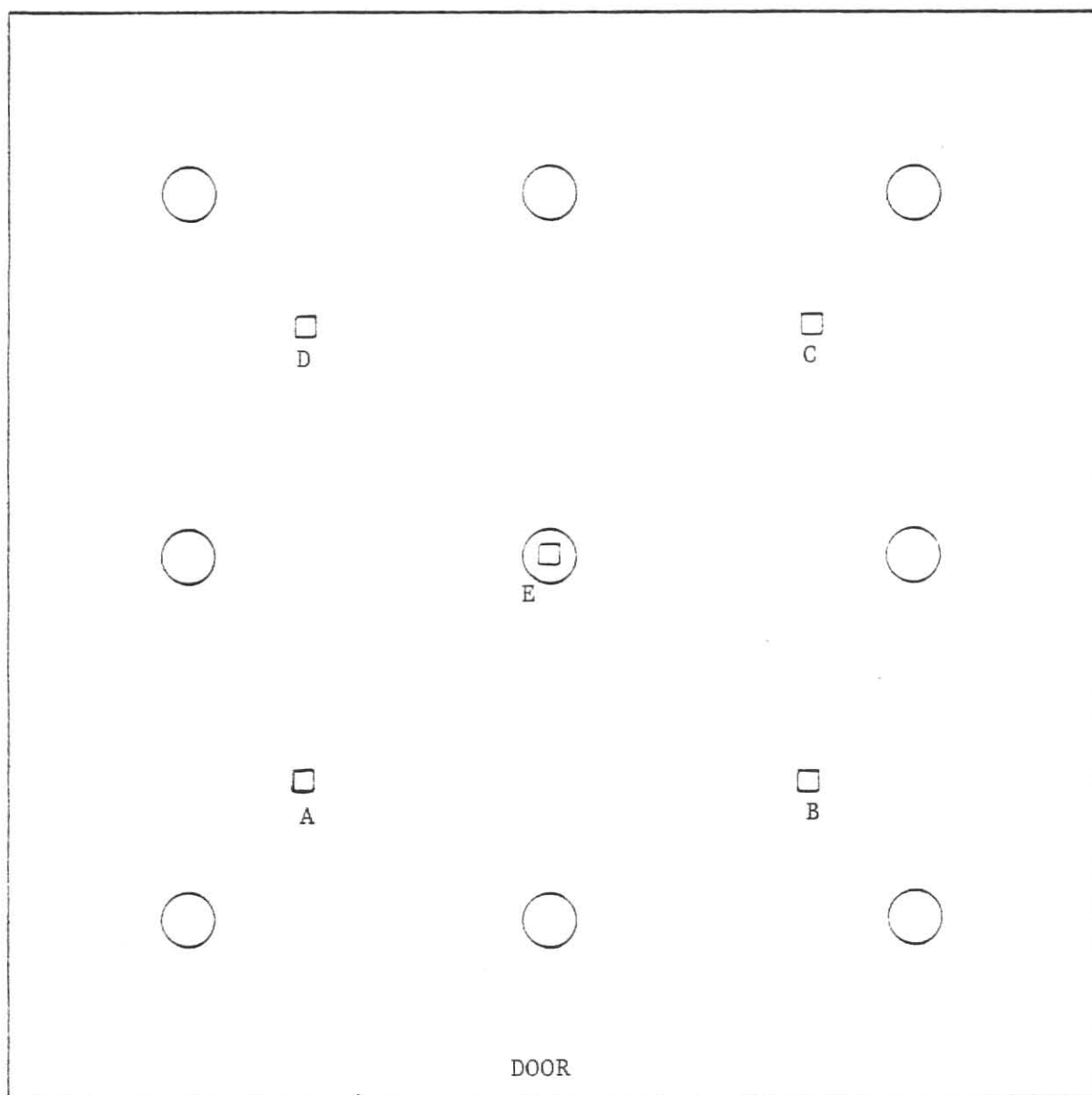


Figure 7. Top view of the light bulbs positions and thermocouple posts (A-E).

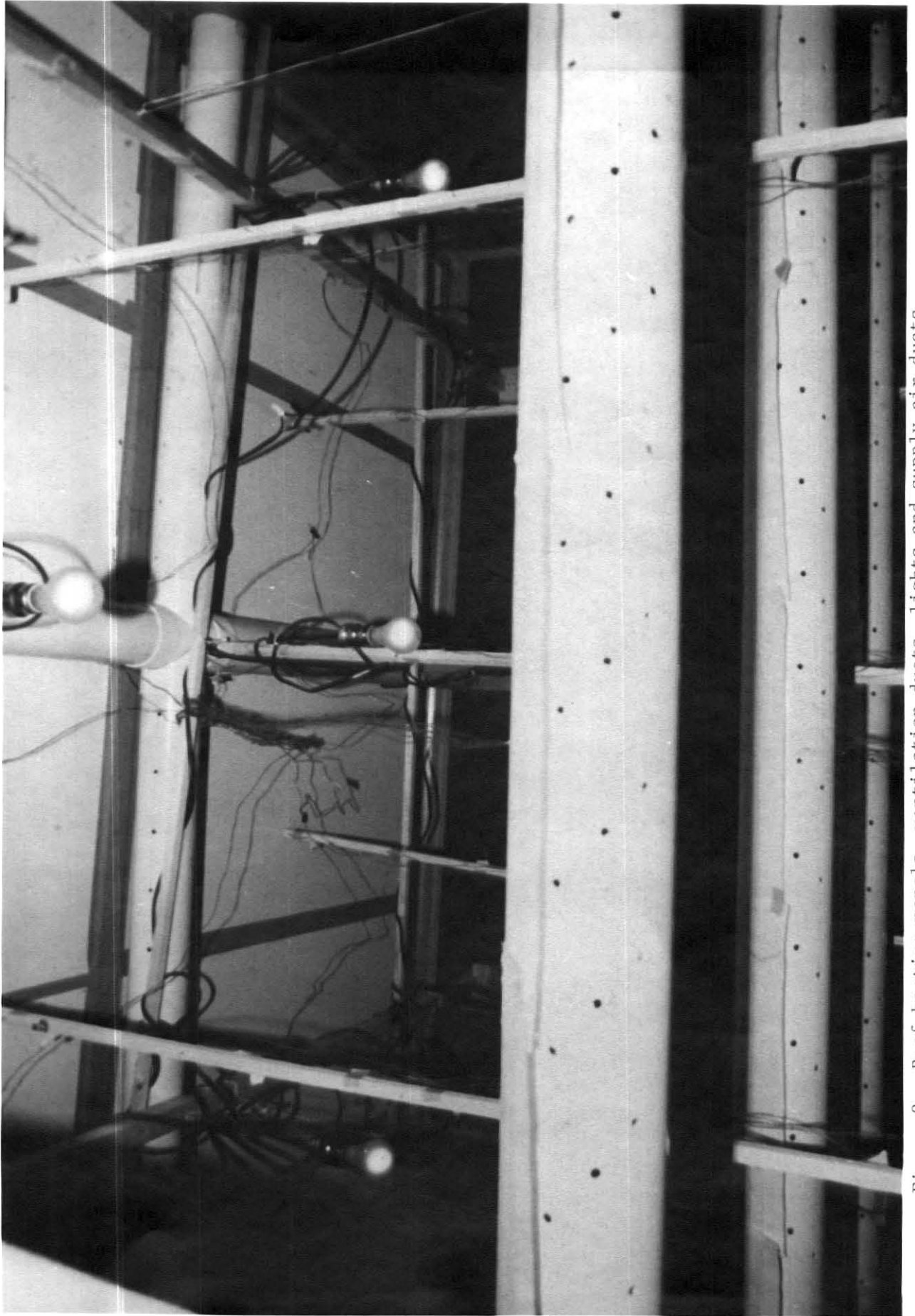


Figure 8. Roof heating panels, ventilation ducts, lights and supply air ducts.



Figure 9. Position of thermocouple posts, floor heating panels, and supply air ducts.

input to the load components. A system of temperature measurement versus height was included for use during testing of model stratification. The instruments and a side view of the chamber is shown in Figure 10. The specifications and description of the equipment are given in the Appendix.

Cool air was supplied to the chamber by a Whirlpool packaged air conditioning unit. With this unit the cool air temperature was within 60 to 65 °F. To test the chamber at lower supply air temperature, another back up unit air conditioner was used, which is not shown in Figure 10. The resulting supply air temperature was in the range of 55 to 60 °F. A blower was used to blow the cool air into the chamber. Two blowers of the same type were installed for purposes of drawing warm air from the chamber through return and ventilation exhaust ducts. Three 2 1/2" gate valves were connected to the blower piping to adjust the amount of air flowing through. Also, three flow element annubars with high and low pressure connections and two vertical manometers were used for means of measuring supply, vent, and return air volume flow rates. The two manometers were to indicate the static pressure (P_s) and pressure difference (ΔP) across each annubar. The method of calculation of actual flow rate under laboratory air conditions is described later in this chapter. The calibration of annubars is discussed in the appendix.

The power input to the roof, floor, and lights were varied by variable transformers (variacs) installed on the outside wall of the chamber. Wattmeters were wired in the circuit to indicate the amount of input loads.

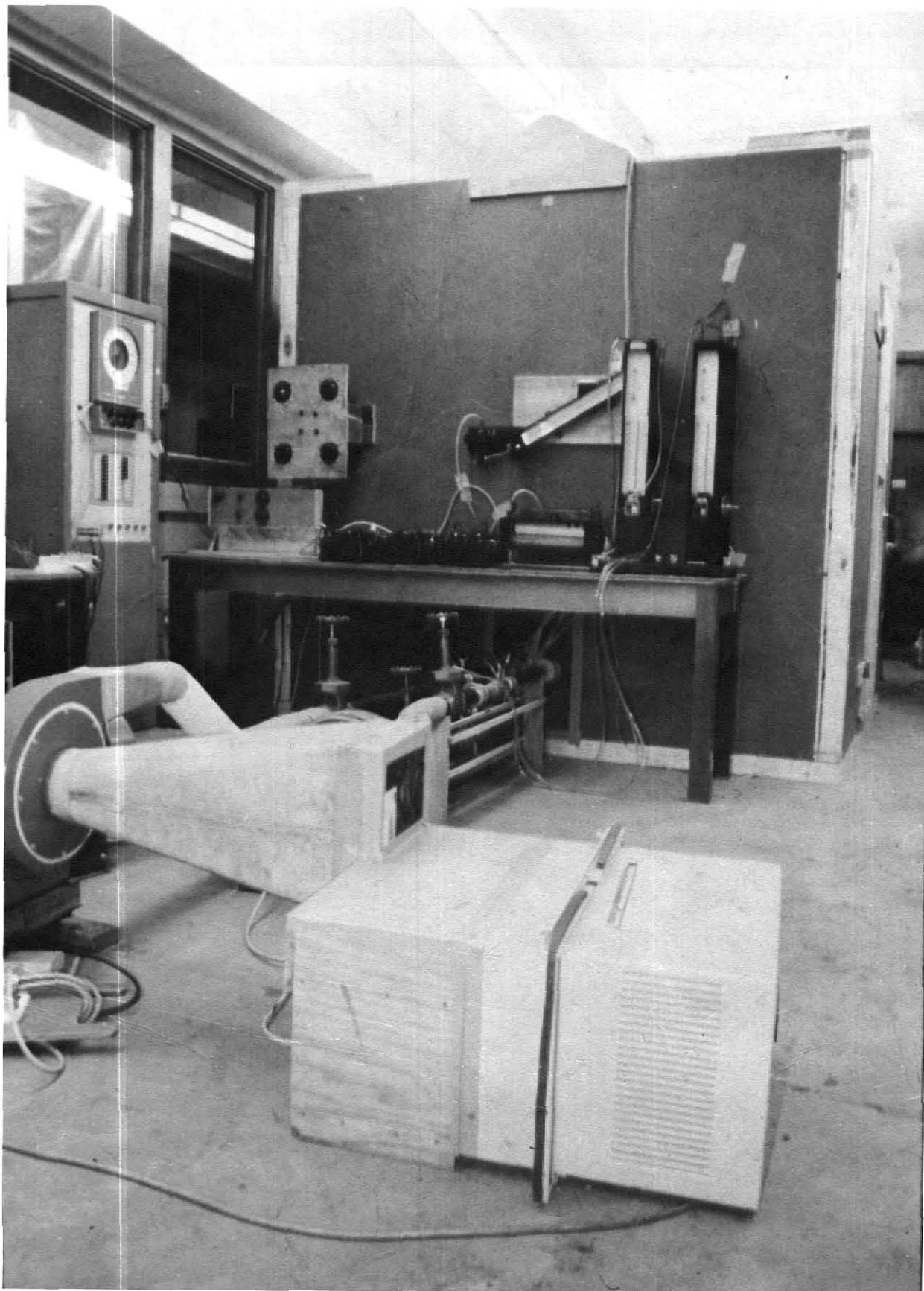


Figure 10. Side view of the chamber with instruments.

Temperature sensor control switches were set on a wooden board, and were recognized by letters A through G. Switches A to E were numbered from 1 to 8, corresponding to the air temperature at a certain level, since the thermocouple at the floor was known as 1 and the thermocouple near the roof as 8. Control switch F was to indicate the surface temperatures of fourteen roof and floor heating panels, and of supply air at the inlet, and return and ventilation air at the outlet. Switch G was used to select the sensors indicating the temperatures at: the chamber roof on the outside, the outside surface of north and east walls, an inside wall surface, and the chamber ambient temperature.

Measurement of thermocouples emf's was made using a Honeywell multichannel potentiometer using an ice bath as a reference junction. A sling psychrometer was used for measuring dry bulb and wet bulb temperatures of the supply, return, and ventilation air. A barometer was necessary for pressure measurement for calculation of flow rates.

DATA REDUCTION

Through the use of instruments described in the previous section temperatures at various points and surfaces of the chamber, and power input to the load components of the chamber could be measured. Also, the static pressure and pressure drop across the supply, return, and ventilation air annubars can be measured. In this section, the method of reduction of experimental data is presented and derivation of an expression for energy loss of the chamber is discussed.

For each of the experimental runs, roof surface temperature was determined by the average of temperatures sensed by thermocouples

positioned at center of each of the roof heating panels. The same method was applied for the floor heating panels.

The space air temperature was measured from floor to roof at one-foot intervals at five different positions inside the chamber. Average temperature of five thermocouples at the same level was taken as the temperature of air at that level. A thermocouple in the supply duct line at the entry to the chamber indicated the cool supply air temperature. The return and ventilation air thermocouples were located in the return and ventilation lines at the exit from the chamber.

Power input to the roof, floor, and light fixtures was indicated by the wattmeters, and was controlled using variacs.

The flow elements (annubars) in the supply, return, and ventilation air ducts were calibrated and curves relating pressure difference (ΔP) versus flow rate across each annubar at standard conditions of air (density = $.075 \text{ lb/ft}^3$) was established. The method of calculation of volume flow rates is given below.

1. Barometric pressure, dry bulb and wet bulb temperatures were measured and recorded.
2. The barometer reading was corrected for room temperature using Table 3 in the Appendix.
3. Total pressure was calculated by

$$P_T = P_{C.B.} + 0.0736 (P_s)$$

where

$P_{C.B.}$ = corrected barometric pressure

$.0736$ = ratio of density of water to density of mercury

P_s = measured static pressure at the annubar.

4. With total pressure, dry bulb temperature and the dry bulb-wet bulb temperature differential, air density was found from the Air Density Chart (Figure 16) in the Appendix.
5. With the measured pressure difference (ΔP) across the annubar, the volume flow rate at standard conditions of air was read from Figure 17 in the Appendix.
6. The actual volume flow rate at laboratory conditions was then calculated as:

$$Q_{\text{actual}} = Q_{\text{standard}} (.075/\rho_{\text{measured}})^{1/2}$$

where

Q 's are in cubic feet per minute and ρ_{measured} is the air density found from air density chart (Figure 16).

The energy balance of the system as a result of difference in temperature (ΔT) between the incoming air and leaving air flowing at standard conditions of air assuming no moisture content ($\Delta W = 0$) required that:

$$(\text{Energy input}) - (\text{Energy output}) = (\text{Energy loss})$$

or simply;

$$P_{\text{in}} - \dot{m} C_p (T_R - T_S) = P_{\text{loss}}$$

which also can be written as

$$P_{\text{in}} - \rho(60)(\text{CFM})C_p(T_R - T_S) = P_{\text{loss}} \quad (1)$$

Density of air (ideal gas) at standard conditions is defined as $\rho = \frac{P}{RT}$

where

$$P = 14.7 \text{ lb}_f/\text{in}^2$$

$$R = 53.3 \text{ ft. lb}_f/\text{lb}_m \text{ } ^\circ\text{R}$$

substituting for ρ into Eq.(1) results

$$P_{in} - \frac{P}{RT} (60)(\text{CFM}) C_p (T_R - T_s) = P_{loss}$$

$$P_{in} - \frac{(14.7 \text{ lb}_f/\text{in}^2)(144 \text{ in}^2/\text{ft}^2)}{(53.3 \text{ ft. lb}_f/16_m \text{ } ^\circ\text{R}) T_R} (60 \frac{\text{min}}{\text{hr}})$$

$$(.24 \text{ BTu/lb}_m \text{ } ^\circ\text{R})(\text{CFM}_R)(T_R - T_s) = P_{loss}$$

or

$$P_{in} - \frac{571.893}{T_R} (\text{CFM}_R)(T_R - T_s) = P_{loss}$$

The above equation simplifies to

$$P_{in} - 571.893 (\text{CFM}_R) [1 - \frac{T_s}{T_R}] = P_{loss} \quad (2)$$

where P_{in} = power input -- Btu/hr

CFM_R = return air volume flow rate -- ft^3/min

T_s = supply air temperature -- $^\circ\text{R}$

T_R = return air temperature -- $^\circ\text{R}$

P_{loss} = energy loss through chamber construction -- Btu/hr

Equation (2) implies that warm air is being discharged only through the return air ducts, and is equivalent to $P_{in} - 1.08 (CFM_R) (T_R - T_S) = P_{loss}$, where T_R and T_S are in degree Fahrenheit. In case of drawing warm air by two different flow rates (return air and ventilation air) Eq. (2) extends to:

$$P_{in} - 571.893(CFM_R) \left[1 - \frac{T_S}{T_R}\right] - 571.893(CFM_V) \left[1 - \frac{T_S}{T_V}\right] = P_{loss} \quad (3)$$

where

CFM_V = ventilation air flow rate -- ft^3/min

T_V = ventilation air temperature -- $^{\circ}R$

Equations (2) and (3) are used in determination of energy losses and calibration of the system discussed in the next chapter.

UNCERTAINTY IN MEASUREMENTS:

The uncertainty in measurements were considered to be in the thermocouples and potentiometers for temperature measurements, in the wattmeters for load input and in the micromanometers in measuring static pressure and pressure difference across the annubars.

For temperature measurements, the limit of error (λ) for copper-constantan thermocouple was $\pm 1^{\circ}C$ or $\pm 0.75\%$. For the portable potentiometer used in calibration of multichannel potentiometer the limit of error is $\pm (0.03\% \text{ of reading} + 3 \mu v)$; this will be neglected here because it is

small compared to other errors. The fluctuations in readings of the multichannel potentiometer were a maximum $\pm .01$ mV, which corresponds to ± 0.5 °F.

The maximum error in wattmeter indication due to underload standing was 0.2% of full scale reading; the maximum error due to change in ambient temperature was $\pm 0.01\%$ per degree centigrade.

The micromanometers used in calibration of annubars were also used in the experimental runs. The only uncertainty in the calibration process was the micromanometers resolution error, which, with the smallest scale division of .001" of water, would be .000707", which can be ignored. The fluctuations of the water surface in the micromanometer tube was within ± 0.1 " of water, which resulted in a maximum error of 5% in flow rate.

The resultant uncertainty in instruments measurements is considered to be 1% in single temperature measurement, and 0.2% of wattmeter span, not including the errors due to variations in line voltage or change in ambient temperature. Also the uncertainty in flow rate measurements was taken to be within 5%.

For the purpose of error analysis, the energy balance equation

$$P_{in} - P_{out} = P_{loss}$$

is broken into three parts, and each is discussed separately.

The error in power input indication was considered to be in wattmeter indications which is 0.2% of full scale reading.

The power output of the system could be expressed as

$$P_{out} = 1.08 (CFM_s)(T_R - T_s) \quad (a)$$

or

$$P_{out} = 571.893 (CFM_R) \left(1 - \frac{T_s}{T_R}\right) \quad (b)$$

when warm air is being exhausted through the return ducts only. The error analysis of Equations (a) and (b) results, respectively, in

$$\epsilon_{P_{out}} = \sqrt{S_{CFM_s}^2 \epsilon_{CFM_s}^2 + S_{T_R}^2 \epsilon_{T_R}^2 + S_{T_s}^2 \epsilon_{T_s}^2} \quad (a_1)$$

or

$$\epsilon_{P_{out}} = \sqrt{S_{CFM_R}^2 \epsilon_{CFM_R}^2 + S_{T_s}^2 \epsilon_{T_s}^2 + S_{T_R}^2 \epsilon_{T_R}^2} \quad (b_1)$$

when some of the warm air is discharged through the ventilation exhaust duct, from the previous section, the power output was written as

$$P_{out} = 571.893 (CFM_R) \left(1 - \frac{T_s}{T_R}\right) + 571.893 (CFM_v) \left(1 - \frac{T_s}{T_v}\right) \quad (c)$$

and the error analysis yields

$$\epsilon_{P_{out}} = \sqrt{S_{CFM_R}^2 \epsilon_{CFM_R}^2 + S_{T_s}^2 \epsilon_{T_s}^2 + S_{T_R}^2 \epsilon_{T_R}^2 + S_{CFM_v}^2 \epsilon_{CFM_v}^2 + S_{T_v}^2 \epsilon_{T_v}^2} \quad (c_1)$$

Power loss through chamber construction is described as

$$P_{loss} = \sum_{W,R,F} UA (\Delta T) \quad (d)$$

where uncertainty analysis yields

$$\epsilon_{p_{\text{loss}}} = \sqrt{\sum (S_{UA}^2 \epsilon_{UA}^2) + \sum (S_{\Delta T}^2 \epsilon_{\Delta T}^2)} \quad (d_1)$$

With these sets of uncertainty equations, the expected error in energy equation of the system can be found and compared with the balance resulting from the summation of the measured energy quantities. Theoretically, this balance should be zero. In practice, it will be non-zero due to measurement inaccuracies and the existence of unsteady state conditions. In the experimental runs, this quantity should be less than or equal to the total expected error.

CHAPTER VI

SYSTEM CALIBRATION

After completion of construction it was necessary to determine certain unknown parameters of the system before any meaningful data could be obtained. In the experimental runs, the unknown parameters were considered to be the thermal transmittance (U-values) of walls, roof, and floor. These values were calculated from thermal resistances (R) of the construction materials used in the structure to be .08, .04, and .1 Btu/hr.ft² °F for walls, roof, and floor, respectively.

With a set of three runs, the U-values were calculated using Equ. (2) from previous chapter (for air being exhausted through return only), which also could be written as

$$P_{in} - 1.08(CFM_S)(T_R - T_S) = U_w A_w (T_{aw} - T_{amb}) + U_R A_R (T_{ar} - T_{amb}) + U_F A_F (T_{af} - T_{amb}) \quad (2.a)$$

For the case of warm air being discharged through return and ventilation ducts, Equ. (3) (from previous chapter) is written as

$$P_{in} - 571.893(CFM_R)(1 - \frac{T_S}{T_R}) - 571.893(CFM_V)(1 - \frac{T_S}{T_V}) = U_w A_w (T_{aw} - T_{amb}) + U_R A_R (T_{ar} - T_{amb}) + U_F A_F (T_{af} - T_{amb}) \quad (3.a)$$

Using these equations, for a set of three runs, three equations with three unknowns (U_W , U_R , and U_F) were obtained. Solving the equations resulted in getting U -values different from the calculated values. The reason for the difference was considered to be the probable errors made in measurement of temperatures and flow rates. Also there occurred variations in power input to the load components due to fluctuations in line voltage and change in ambient temperature.

The temperature stratification profile versus height for two experimental runs with different heights of supply ducts and lights are shown in Figures 11 and 12. In these experiments the return air ducts were located on the floor and the ventilation exhaust ducts were positioned near the roof. The measured temperature patterns were similar to the ideal patterns discussed in Chapter III (Figure 2), with uniform temperature in the lower, air conditioned zone and varying high temperatures in the upper zone.

In Tables 1 and 2, the unaccounted loss (UL) is compared to total expected error in each experimental run. The unaccounted loss was determined by algebraic summation of P_{in} , P_{out} , and P_{loss} . The P_{loss} is the energy transmitted through chamber construction and was calculated using the right hand side of Equ. (2.a) with thermal transmittances (U -values) of wall, roof, and floor taken to be .08, .04, and .1. The data of each run is given in the figures.

In run 1, the unaccounted loss is larger than the expected error, but in run 2 it is smaller than the expected error. As mentioned in the previous chapter, the unaccounted loss should be less than or equal to the total expected error. For cases where this is not so, the reason is thought to be due to the unsteady state condition of system and to variations in power input caused by fluctuations in line voltage and change in ambient temperature.

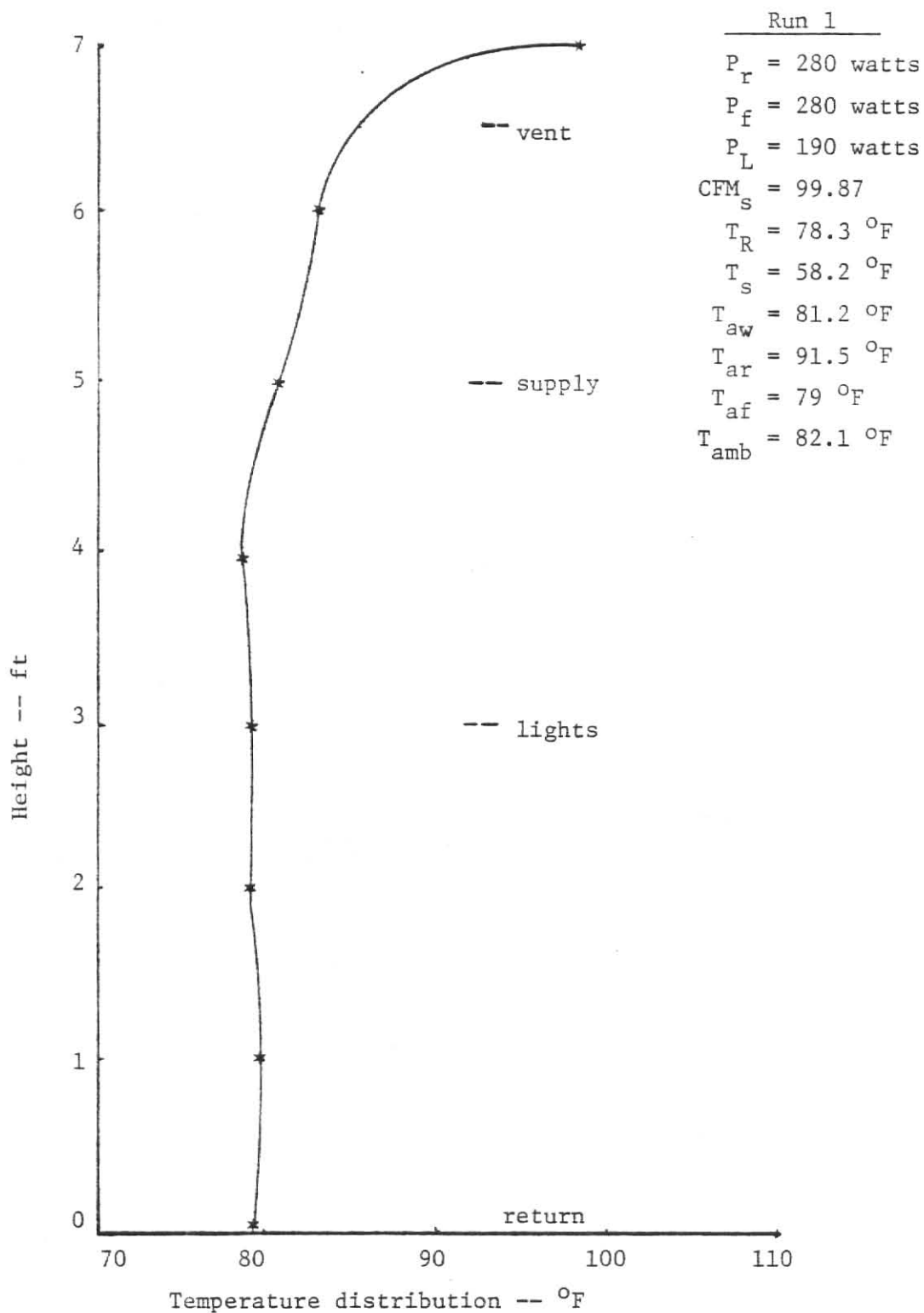


Figure 11. Temperature stratification profile for RUN 1.

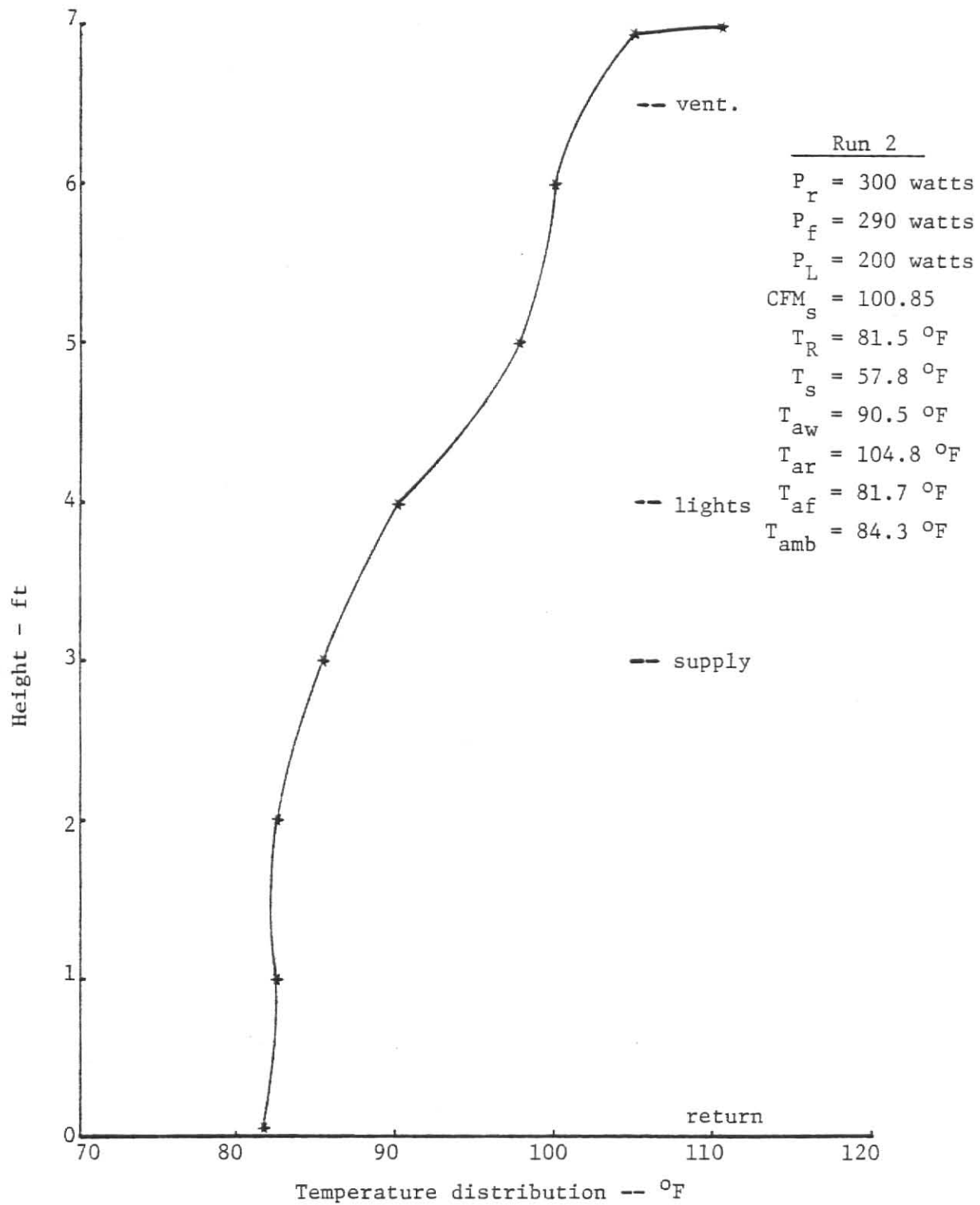


Figure 12. Temperature stratification profile for RUN 2.

Table 1

Calculation of uncertainties
in RUN 1 (Figure 11)

$$P_{in} = 750 \text{ watts} = 2559.75 \text{ Btu/hr}$$

$$P_{out} = 1.08 (\text{CFM}_s) (\Delta T) = 2168 \text{ Btu/hr}$$

$$P_{loss} = \sum UA \Delta T = -12.1$$

$$\text{unaccounted loss (UL)} = P_{in} - (P_{out} + P_{loss}) = 403.66 \text{ Btu/hr}$$

$$\text{error in } P_{in} : 6 \text{ watts} = 20.5 \text{ Btu/hr} \approx 0.8\% \text{ of } P_{in}$$

$$\text{error in } P_{out} : \text{with } S_{\text{CFM}_s} = 1$$

$$S_{T_R} = \frac{T_R}{T_R - T_s} = \frac{78.3}{78.3 - 58.2} = 3.89$$

$$S_{T_s} = \frac{-T_s}{T_R - T_s} = \frac{-58.2}{78.3 - 58.2} = 2.89$$

$$\epsilon_{P_{out}} = \sqrt{1^2(5)^2 + (3.89)^2(1)^2 + (2.89)^2(1)^2} = 6.96\% \text{ of } P_{out}$$

$$\text{error in } P_{loss} : \text{taking } \epsilon_{UA} = 20\% , S_{UA} \text{ and } S_{\Delta T} = 1$$

$$\epsilon_{P_{loss}} = \sqrt{3(1)^2(20)^2 + 3(1)^2(1.4)^2} = 34.7\% \text{ of } P_{loss}$$

$$\text{Total expected error} \approx (.008)(2559.75) + (.0696)(2168) + (.347)(-12.1)$$

$$\approx 167.17 \text{ Btu/hr.}$$

unaccounted loss > total expected error

Table 2

Calculation of uncertainties
in RUN 2 (Figure 12)

$$P_{in} = 790 \text{ watts} = 2696.27 \text{ Btu/hr}$$

$$P_{out} = 1.08(CFM_s)\Delta T = 2581.36 \text{ Btu/hr}$$

$$P_{loss} = \int UA\Delta T = 146.94 \text{ Btu/hr}$$

$$\text{unaccounted loss (UL)} = P_{in} - (P_{out} + P_{loss}) = -32.03 \text{ Btu/hr}$$

$$\text{error in } P_{in} : 6 \text{ watts} = 20.5 \text{ Btu/hr} = .76\% \text{ of } P_{in}$$

$$\text{error in } P_{out} : \text{with } S_{CFM_s} = 1$$

$$S_{T_R} = \frac{T_R}{T_R - T_s} = \frac{81.5}{81.5 - 57.8} = 3.44$$

$$S_{T_s} = \frac{T_s}{T_R - T_s} = \frac{-57.8}{81.5 - 57.8} = -2.44$$

$$\epsilon_{P_{out}} = \sqrt{1^2(5)^2 + 1^2(3.44)^2 + 1^2(2.44)^2} = 6.54\% \text{ of } P_{out}$$

$$\text{error in } P_{loss} : \text{taking } \epsilon_{UA} = 20\% , S_{UA} \text{ and } S_{\Delta T} = 1$$

$$\epsilon_{P_{loss}} = \sqrt{3(1)^2(20)^2 + 3(1)^2(1.4)^2} = 34.7\% \text{ of } P_{loss}$$

$$\begin{aligned} \text{Total expected error} &\approx (.0076)(2696.27) + (.0654)(2581.36) + (.347)(146.94) \\ &\approx 240.30 \text{ Btu/hr} \end{aligned}$$

$$\text{unaccounted loss} < \text{total expected error}$$

CHAPTER VII

CONCLUSIONS

For experimentation with thermally stratified cooling systems in high ceiling industrial plants, a model chamber has been designed and built. It was constructed by conventional techniques with available and light weight materials. The necessary instrumentation was obtained, installed and calibrated with standard precision instruments using accepted procedures.

The model chamber was designed to simulate the thermal loads of roof, floor, and lights of an actual plant in the range of zero to 13.2, 8.7, and 4.2 watts/ft², respectively. The height of roof, lights, and supply, return and ventilation exhaust ducts were variable. Volume flow rate of supply air was selected to range up to 3.2 cfm/ft², with ventilation volume flow rate to be within zero to 100% of supply air flow rate.

Initial operation of the chamber for a set of test runs indicated that the chamber can be operated as a reliable and effective means of obtaining the data necessary for the stratification studies for which it was designed.

LIST OF REFERENCES

1. Winter, E. R. F. and Schoenhals, R. J., "Analytical and Experimental Studies of Thermal Stratification Phenomena," Final Report, Part V, Heat Transfer Laboratory, Purdue University, (To NASA), October, 1968.
2. Armaly, B. F. and Lepper, S. P., "Diurnal Stratification of Deep Water Impoundments," ASME Publication, 75-HT-35.
3. Dake, M. K. and Harleman, R. F., "Thermal Stratification in Lakes, Analytical and Laboratory Studies," Water Resources Research, Vol. 5, April 1969, pp. 484-495.
4. Beier, R. A., "Thermal Stratification in Factories With High Ceilings," Master's Thesis, Kansas State University, Manhattan, Kansas, 1976.
5. Beier, R. A. and Gorton, R. L., "Thermal Stratification in Factories - Cooling Loads and Temperature Profiles," ASHRAE Transactions, Vol. 84, Part 1, 1978.
6. Dralle, W. F., "Vertically Zoned Air Conditioning For High Ceiling Industrial Plants," Heating Piping and Air Conditioning, December, 1970. pp 69-72.
7. Olivieri, J. B., "A Consultant Looks at Complaints About Unitary A/C for Industrial Applications," Air Conditioning, Heating and Refrigeration News, February 4, 1974.
8. Dean, F. R. and Reese, J. A., "A New Approach to Factory Air Conditioning," Heating, Piping and Air Conditioning, June 1974, pp. 45-49.
9. Allen, C. J., "Stratification by Design," ASHRAE Transactions, Vol. 85, Part 1, 1979.
10. Holness, G. V. R., "Energy Efficient Industrial Building Design," Heating, Piping and Air Conditioning, February 1981.
11. Divieri, J. B. and T. Singh, "Effect of Supply and Return Outlet Locations on Stratification," ASHRAE Transactions, Vol. 85, Part 1, 1979.
12. Parczewski, K. I. and Renzi, P. N., "Scale Model Studies of Temperature Distributions in Internally Heated Enclosures," ASHRAE Transactions, No. 1854, July 1963.
13. Katzoff, S., "Similitude of Thermal Models," N.A.S.A. Technical Note 1631, April 1963.
14. Vickers, J. M. F., "Thermal Scale Modeling," Astronautics and Aeronautics, May 1965, pp. 34-39.

15. Thomas, Lindon, "Model Theory as Related to Convection Heat Transfer," unpublished, May 5, 1966.
16. Kreith, F., "Principles of Heat Transfer," Third Edition, International Textbook Company, Scranton, Pa. 1976.
17. McAdams, W. H., "Heat Transmission," McGraw-Hill Book Company, 1954.
18. Lionel Mark's Engineers Handbook, Fifth Edition, McGraw-Hill, 1951.
19. Air Engineering Journal, February 1964.
20. Sprague, C. H., and Nash, R. T., Introduction to Engineering Experimentation, 1972. Mechanical Engineering Department, Kansas State University, Manhattan, Kansas: Unpublished.

APPENDIX

APPENDIX A

INSTRUMENTS

Whirlpool air conditioner: 1/6 HP 208-230 volts
60 Hz 1050/950/800 RPM
EER = 8.7

Friedrich air conditioner: SPo6D10 6500 BTuh
cooling capacity 745 watts
115 volts, 6.5 Amp.
EER = 8.7

Dayton blowers: Motor capacity model C2548C2
220 volts, 5.6 Amp.
60 cycle
RPM 3450
dia. wheel 10 5/8"

Supply and vent. flow elements: Type 71 - Annubar, 2 1/2" with
(see Figure 13) #5 (valve) fittings or equivalent

Return flow element: Type 71 - Annubar, 1 1/2" with
(see Figure 13) #5 (valve) fittings or equivalent

Vertical Meriam manometers: Type: micro
Model: A-750A
Range: 10/20
Read to: .001" of water

Variable transformers (variatics): Type 501 STACO
(Floor and lights) Single phase, 120 volt input
Output: 0-120/140 volts
5 Amps
0.70 KVA

Variable transformer (variac) Type 1010 STACO
(Roof) Single phase, 120 volt input
Output: 0-120/140 volts
10 Amps
1.4 KVA

Weston wattmeter (roof) Model 310: Ranges: 62.5/125/250 volts with
125 volts, maximum current 10 Amps
Multiple scale reading 2

Weston wattmeter (floor) Model 310: Ranges: 50/100/200 volts with
100 volts, maximum current 10 Amps
Multiple scale reading 3

General Electric wattmeter (lights): Type P-3, single phase
Ranges: 100/200 volts, with
100 volts, maximum current 3 Amps
Multiple scale reading 3

Radiant heating panels (Singer) #3490 Model CP-5 Singer, 500 watts at
(See Figure 14) 120 volts, 1706 BTu/s
Dimensions: 23 3/4" x 47 3/4" x
1 1/4"

Copper-constantan insulated wire(Thermocouples) Pr-T-24

Honeywell multichannel potentiometer: Electronic 18
Range: -0.5 to + 5.5 mV
Smallest scale division 0.01 mV
Mech. Engg. Dept. Property No.
172704

Portable millivolt potentiometer: Leeds and Northrup Co., Cat. No. 8686

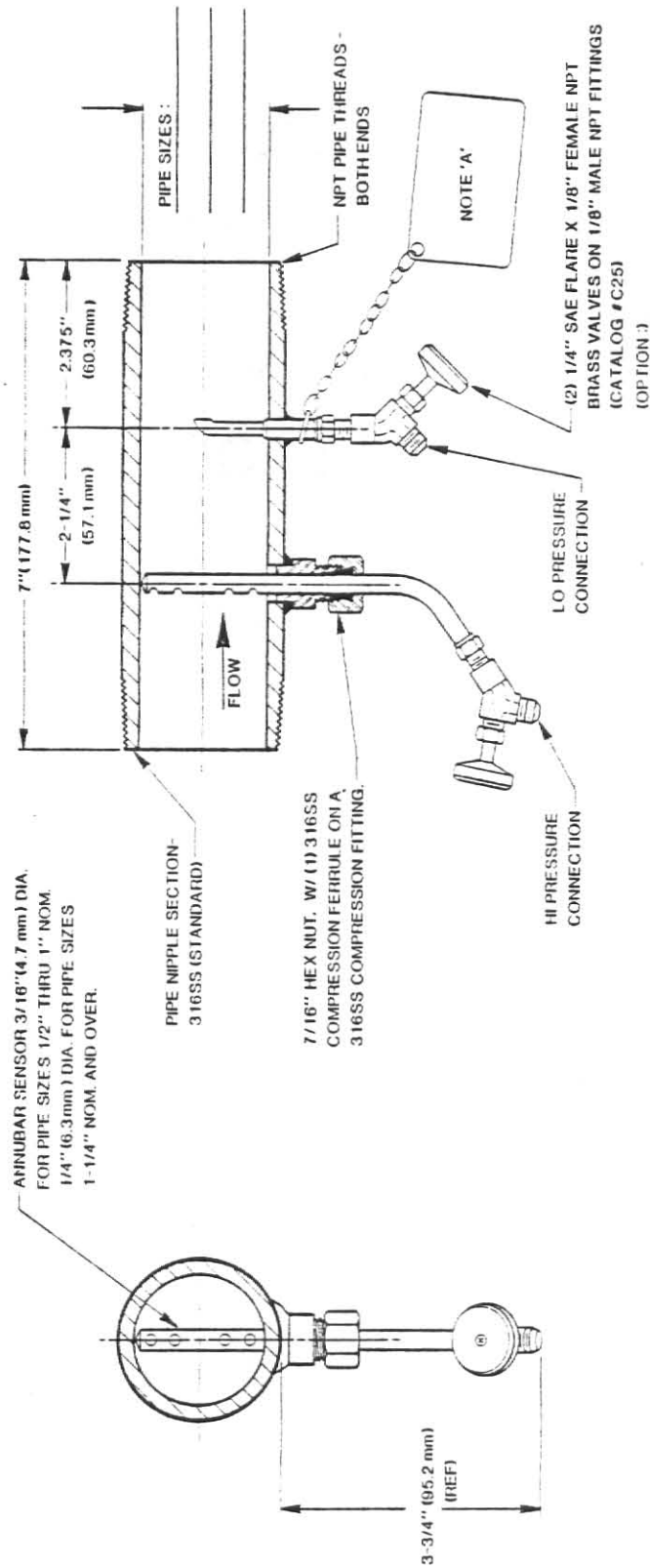
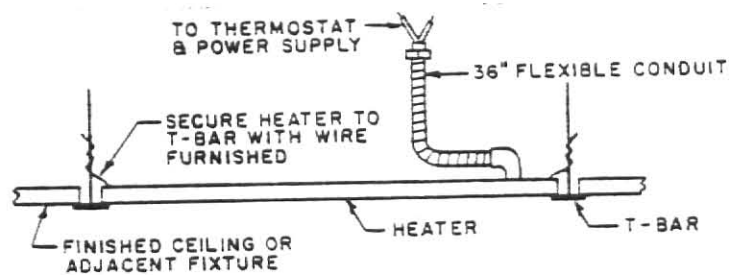
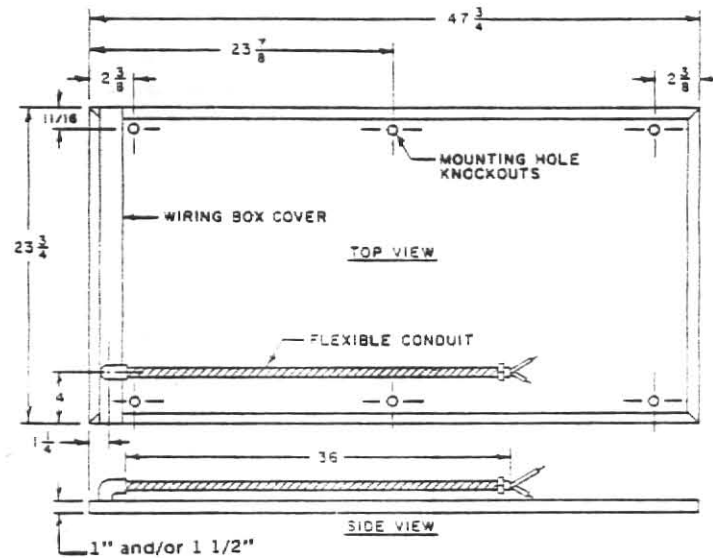
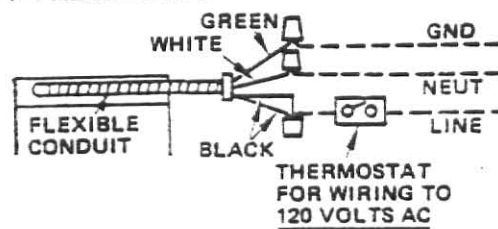


Figure 13. The description of an Annubar.



TO MOUNT HEATER IN SUSPENDED CEILING



WIRING DIAGRAM

Figure 14. The description of a heating panel.

APPENDIX B

LIST OF SYMBOLS

Symbol	Significance	Units
A_1	cross sectional area of the nozzle	ft^2
A_2	cross sectional area of the annubar	ft^2
C_p	constant pressure specific heat	$\text{Btu/lb}_m \text{ } ^\circ\text{R}$
CFM_R	return air volume flow rate	ft^3/min
CFM_S	supply air volume flow rate	ft^3/min
CFM_V	ventilation air volume flow rate	ft^3/min
g	gravitational acceleration	ft/sec^2
h_a	velocity head of air	feet of air
h_D	convection heat transfer coefficient, disperse heat load sources	$\text{Btu}/\text{hr}.\text{ft}^2\text{-F}$
h_F	convection heat transfer coefficient, floor	$\text{Btu}/\text{hr}.\text{ft}^2\text{-F}$
h_L	convection heat transfer coefficient, lights	$\text{Btu}/\text{hr}.\text{ft}^2\text{-F}$
h_R	convection heat transfer coefficient, roof	$\text{Btu}/\text{hr}.\text{ft}^2\text{-F}$
h_W	velocity head of air	ft. of water
h_{W1}	velocity head across nozzle	ft. of water, in. of water
h'_R	characteristic length of roof	ft
h'_{uz}	height of upper zone	ft
K	thermal conductivity	$\text{Btu}/\text{hr}.\text{ft} \text{ } ^\circ\text{F}$
L	length	ft
m_1	mass flow rate across the nozzle	lbm/sec
m_2	mass flow rate across the annubar	$\text{lbm}.\text{sec}$

$\dot{m}_c$	circulation flow rate per unit floor area	lbm/hr ft ²
$\dot{m}_v$	ventilation exhaust flow rate per unit floor area	lbm/hr ft ²
$\dot{m}_{Rs}$	room supply air flow rate per unit floor area	lbm/hr ft ²
$\dot{m}_{RR}$	room return air flow rate per unit floor area	lbm/hr ft ²
$\dot{m}_{OA}$	outside air flow rate per unit floor area	lbm/hr ft ²
P	atmospheric pressure	lb _f /in ²
P _{CB}	corrected barometric pressure	inches of mercury
P _f	power input to the floor heating panels	watts
P _{in}	power input to the system	Btu/hr
P _L	power input to the light	watts
P _{loss}	energy loss through chamber surfaces	Btu/hr
P _r	power input to the roof heating panels	watts
P _s	measured static pressure of the annubar	inches of water
P _T	total pressure	inches of mercury
ΔP ₁	pressure difference across the nozzle	inches of water
ΔP ₂	pressure difference across the annubar	inches of water
ΔP _{2std}	pressure drop across annubar at standard conditions of air	inches of water
q	heat flow rate per unit floor area in the model	Btu/hr ft ²
q _c	convective energy rate per unit floor area of model	Btu/hr ft ²
q _F	heat loss by the floor per unit floor area of model	Btu/hr ft ²
q _{IC}	internal concentrated heat flow rate per unit area of model	Btu/hr ft ²
q _{ID}	internal disperse heat flow rate per unit floor area of model	Btu/hr ft ²

q_{IF}	floor internal heat flow rate per unit floor area of model	Btu/hr ft ²
q_K	conduction energy rate per unit area in the model	Btu/hr ft ²
q_{LC}	lights convective heat flow rate per unit floor area of the model	Btu/hr ft ²
q_{LR}	radiant heat flow rate from the lights to the floor surface per unit floor area in the model	Btu/hr ft ²
q_R	total roof heat flow rate per unit area in the model	Btu/hr ft ²
q_{RC}	roof convective heat flow rate per unit floor area in the model	Btu/hr ft ²
q_{RR}	radiant heat flow rate from the roof to the floor surface per unit floor area of the model	Btu/hr ft ²
q_T	total rate of heat flow per unit floor area in the model	Btu/hr ft ²
Q	heat flow rate per unit floor area in the prototype	Btu/hr ft ²
Q_C	convective energy rate per unit floor area of prototype	Btu/hr ft ²
Q_F	heat loss by the floor per unit floor area of prototype	Btu/hr ft ²
Q_{IC}	internal concentrated heat flow rate per unit floor area of prototype	Btu/hr ft ²
Q_{ID}	internal disperse heat flow rate per unit floor area of prototype	Btu/hr ft ²
Q_{IF}	floor internal heat flow rate per unit floor area of prototype	Btu/hr ft ²
Q_K	conduction energy rate per unit area in the prototype	Btu/hr ft ²
Q_{LC}	light convective heat flow rate per unit floor area of the prototype	Btu/hr ft ²
Q_{LR}	radiant heat flow rate from the lights to the floor surface per unit floor area in the prototype	Btu/hr ft ²

Q_R	total roof heat flow rate per unit area in the prototype	50 Btu/hr ft ²
Q_{RC}	roof convective heat flow rate per unit area in the prototype	Btu/hr ft ²
Q_{RR}	radiant heat flow rate from the roof to the floor surface per unit floor area of the prototype	Btu/hr ft ²
Q_T	total rate of heat flow per unit floor area in the prototype	Btu/hr ft ²
Q_{actual}	volumetric flow rate across an annubar at laboratory air conditions	ft ³ /min
Q_{standard}	volumetric flow rate across an annubar at standard air conditions	ft ³ /min
R	gas constant	ft lb _f /lb _m °R
S	sensitivity factor	dimensionless
S_c	specific heat scale factor	dimensionless
S_d	roof length scale factor	dimensionless
S_ρ	density scale factor	dimensionless
S_t	time scale factor	dimensionless
S_u	height of upper zone scale factor	dimensionless
$\bar{T}$	average air temperature	°F
T_{af}	inside air temperature near floor	°F
T_{ar}	inside air temperature near roof	°F
T_{aw}	inside air temperature near wall	°F
T_C	conditioned space air temperature	°F
T_D	disperse heat load sources temperature	°F
T_F	floor surface temperature	°F, °R
T_L	lights temperature	°F
T_{Ls}	lower strata air temperature	°F
T_R	return (outlet) air temperature	°F, °R

T_r	roof surface temperature	$^{\circ}\text{F}, ^{\circ}\text{R}$
T_s	supply air temperature	$^{\circ}\text{F}, ^{\circ}\text{R}$
T_{us}	upper strata air temperature	$^{\circ}\text{F}$
T_{uz}	upper zone air temperature	$^{\circ}\text{F}$
T_v	ventilation exhaust air temperature	$^{\circ}\text{F}$
V_1	velocity of air at the nozzle	ft/sec
V_2	velocity of air at the annubar	ft/sec
NU	Nusselt number	dimensionless
Gr	Grashof number	dimensionless
Pr	Prandtl number	dimensionless

Greek Symbols

ρ	density	lbm/ft^3
ρ_1	density of air at the nozzle	lbm/ft^3
ρ_2	density of air at the annubar	lbm/ft^3
ρ_w	density of water	lbm/ft^3
θ	time	sec
β	thermal expansion coefficient	$\frac{1}{^{\circ}\text{R}}$
μ	dynamic viscosity	$\text{lb}_f\text{sec/ft}^2$
σ	Stefan-Boltzmann constant	$\text{Btu/hr ft}^2 ^{\circ}\text{R}^4$
Δ	the difference between any two quantities	
ϵ	uncertainty	dimensionless

APPENDIX C

CALIBRATION OF INSTRUMENTATION

This section presents the procedures for calibration of the annubars (flow elements) and of the multichannel potentiometer.

The Honeywell multichannel potentiometer was calibrated using a portable precision potentiometer which has a high measuring accuracy. A chart was developed (shown in Figure 15) to present the measured millivolts versus the actual millivolt signal from the precision potentiometer. For the experimental runs, the millivolt indication of the multichannel potentiometer were corrected using this chart.

Calibration of the annubars was carried out using a standard 2" ASME nozzle. Micromanometers were used to measure the static pressure (P_s) and the pressure difference (ΔP) for the flow through the nozzle and each annubar.

In the calibration apparatus set up the nozzle was considered as station 1 and the annubar as station 2. For steady flow, the conservation of mass, or continuity equation at points 1 and 2 in the system may be written as

$$\dot{m}_1 = \dot{m}_2$$

or

$$\rho_1 V_1 A_1 = \rho_2 V_2 A_2$$

or

$$\rho_1 V_1 A_1 = \rho_2 Q_2$$

The flow rate across the annubar (Q_2) is then

$$Q_2 = \frac{\rho_1}{\rho_2} A_1 V_1$$

ILLEGIBLE DOCUMENT

**THE FOLLOWING
DOCUMENT(S) IS OF
POOR LEGIBILITY IN
THE ORIGINAL**

**THIS IS THE BEST
COPY AVAILABLE**

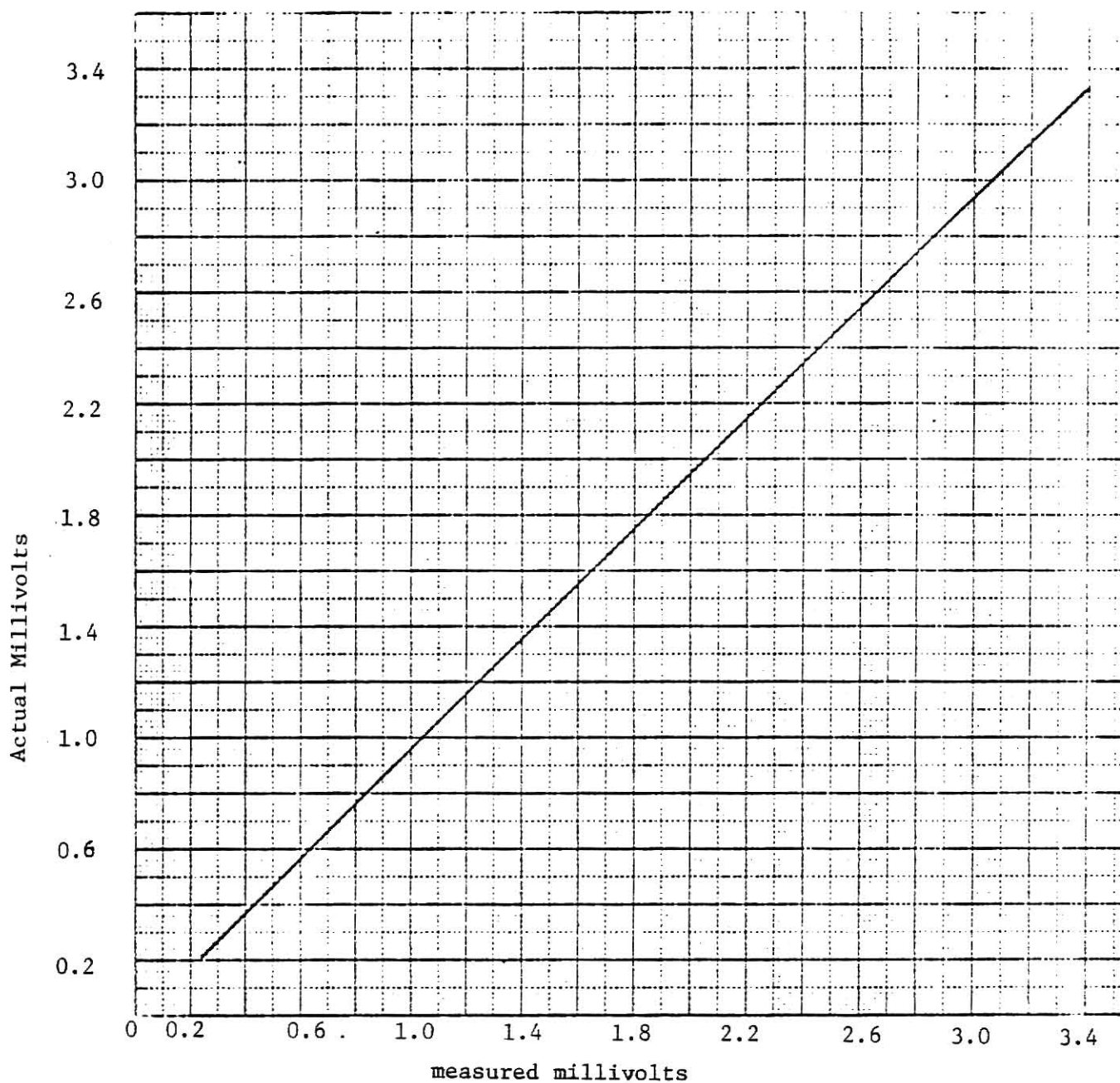


Figure 15. Potentiometer calibration chart.

The basic statement of this relationship between velocity and velocity pressure is:

$$V_1 = K \sqrt{2gh_{a1}}$$

where

V_1 = velocity of air at nozzle, ft/sec

g = gravitational acceleration, ft/sec²

h_a = head of air, feet

K = proportionality constant

Since air is being measured by a manometer reading inches of water, the head of air should be converted to head in inches of water by the equation

$$\rho_a h_a = \rho_w h_w$$

or

$$h_a = \frac{\rho_w}{\rho_a} h_w$$

substitute for h_a into the equation for V_1 and then substitute this into the expression for Q_2 :

$$Q_2 = \frac{\rho_1}{\rho_2} A_1 K \frac{\sqrt{2g\rho_w h_{w1}}}{\sqrt{\rho_a}}$$

but at the nozzle $\rho_1 = \rho_a$, therefore

$$Q_2 = A_1 K \frac{\sqrt{2g\rho_w} \sqrt{\rho_1 h_{w1}}}{\rho_2} \quad (C.1)$$

where

Q_2 = volume flow rate across annubar, ft^3/min

A_1 = cross sectional area of nozzle, (dia = 2")

K = proportionality constant, (= .99)

ρ_w = density of water, (= 62.4 lb_m/ft^3)

ρ_1 = density of air at the nozzle, lbm/ft^3

ρ_2 = density of air at the annubar, lbm/ft^3

h_{w1} = head across nozzle, feet of water

Substituting numerical values into Equation (1) simplifies it to

$$Q_2 = 23.677 \frac{\sqrt{\rho_1}}{\rho_2} (h_{w1})^{1/2} \quad (\text{C.2.})$$

where

h_{w1} is in inches of water.

Equation (2) was used to calculate flow rates to establish the relationship between flow rate through each annubar and pressure drop across them at standard conditions of air. The procedure used for collection of data for construction of the calibration chart is given below.

1. At a certain fan speed, the static pressure and the pressure drop across the annubar and the ASME nozzle were measured and recorded.
2. Laboratory barometric pressure and dry bulb and wet bulb temperatures were recorded.

3. The barometric pressure was corrected to room air temperature by adding or subtracting the values given in Table 3.
4. Total pressure was calculated for the nozzle and the annubar from

$$P_T = P_{c.b} + .0736 (P_s)$$

where

$P_{c.b}$ = corrected barometric pressure, inches of mercury

P_s = static pressure, inches of water

.0736 = ratio of density of water to density of mercury.

5. Using Air Density Chart (Figure 16), the density of air across the nozzle and annubar was determined.
6. Pressure drop across the annubar was corrected for standard air density ($.075 \text{ lbm/ft}^3$) using

$$\Delta P_{2_{\text{std}}} = \Delta P_2 (.075/\rho_2)$$

where

$\Delta P_{2_{\text{std}}}$ = annubar pressure drop at standard air conditions

ΔP_2 = measured pressure drop across annubar

ρ_2 = measured air density (read from air density chart).

7. Flow rate across the annubar Q_2 was calculated from Eq. (2)

$$Q_2 = 23.677 \frac{\sqrt{\rho_1}}{\rho_2} (\Delta P_1)^{1/2} \quad (\text{C.2})$$

8. The same procedure was followed at different fan speeds, and flow rate across the annubar (Q_2) and standardized pressure drop across the annubar ($\Delta P_{2_{\text{std}}}$) were obtained.

Barometer Corrections for Temperature

Temperature of mercury column, deg F	Observed reading of mercury column, in. of mercury						
	20	22	24	26	28	30	32
-20	+0.09	+0.10	+0.11	+0.11	+0.12	+0.13	+0.14
0	+0.05	+0.06	+0.06	+0.07	+0.07	+0.08	+0.08
20	+0.02	+0.02	+0.02	+0.02	+0.02	+0.02	+0.02
40	-0.02	-0.02	-0.02	-0.03	-0.03	-0.03	-0.03
60	-0.06	-0.06	-0.07	-0.07	-0.08	-0.08	-0.09
80	-0.09	-0.10	-0.11	-0.12	-0.13	-0.14	-0.15
100	-0.13	-0.14	-0.15	-0.17	-0.18	-0.19	-0.20

Table 3. Barometric Pressure Corrections [Ref. (18)]

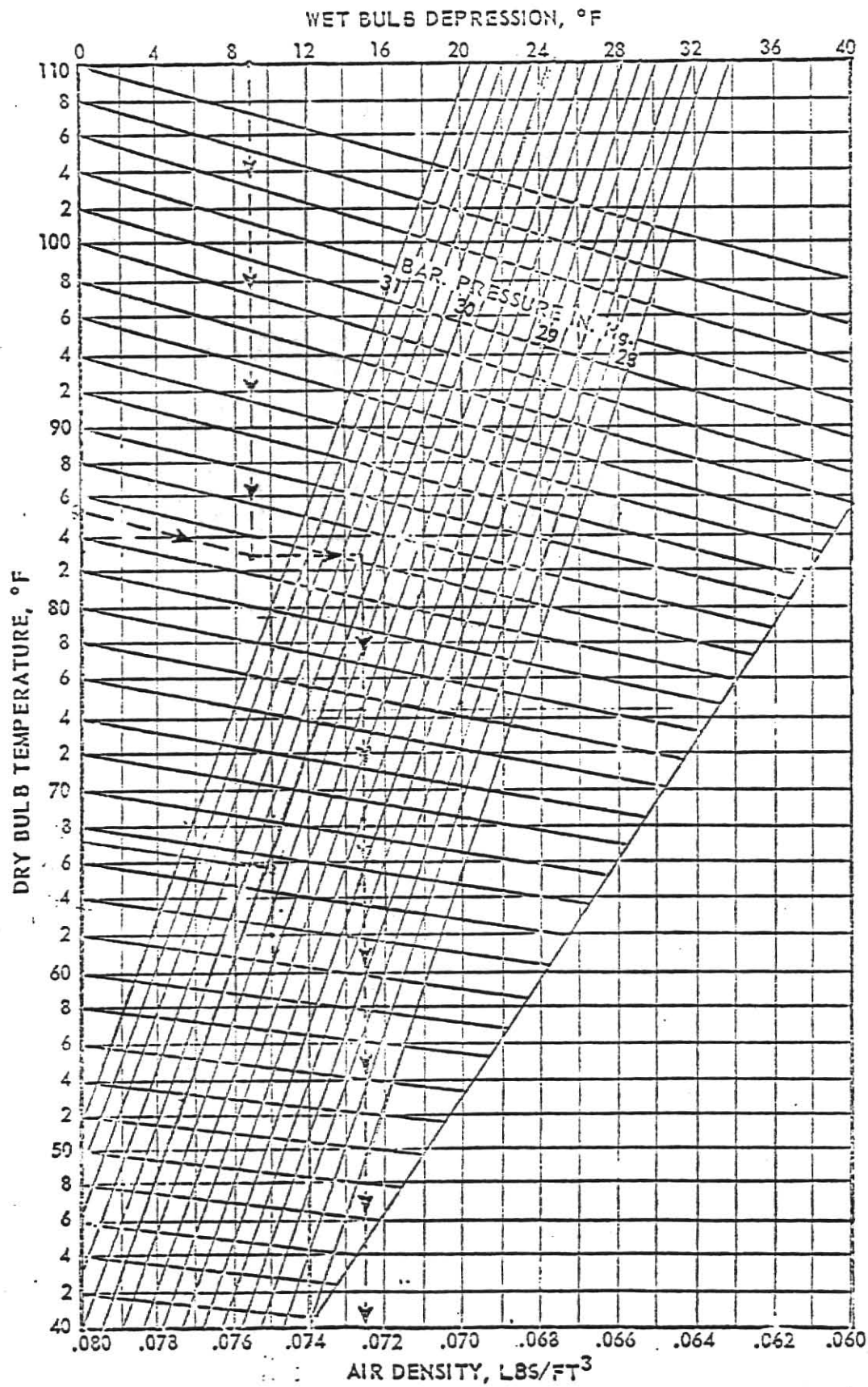


Figure 16. Air Density Chart [Ref. (19)].

9. The relationship between volumetric flow rate and pressure drop across annubars was plotted as ΔP versus Q_2 as shown in Figure 17.

As described in the data reduction section, this chart was used to indicate the volumetric flow rate of air across the annubar at standard conditions.

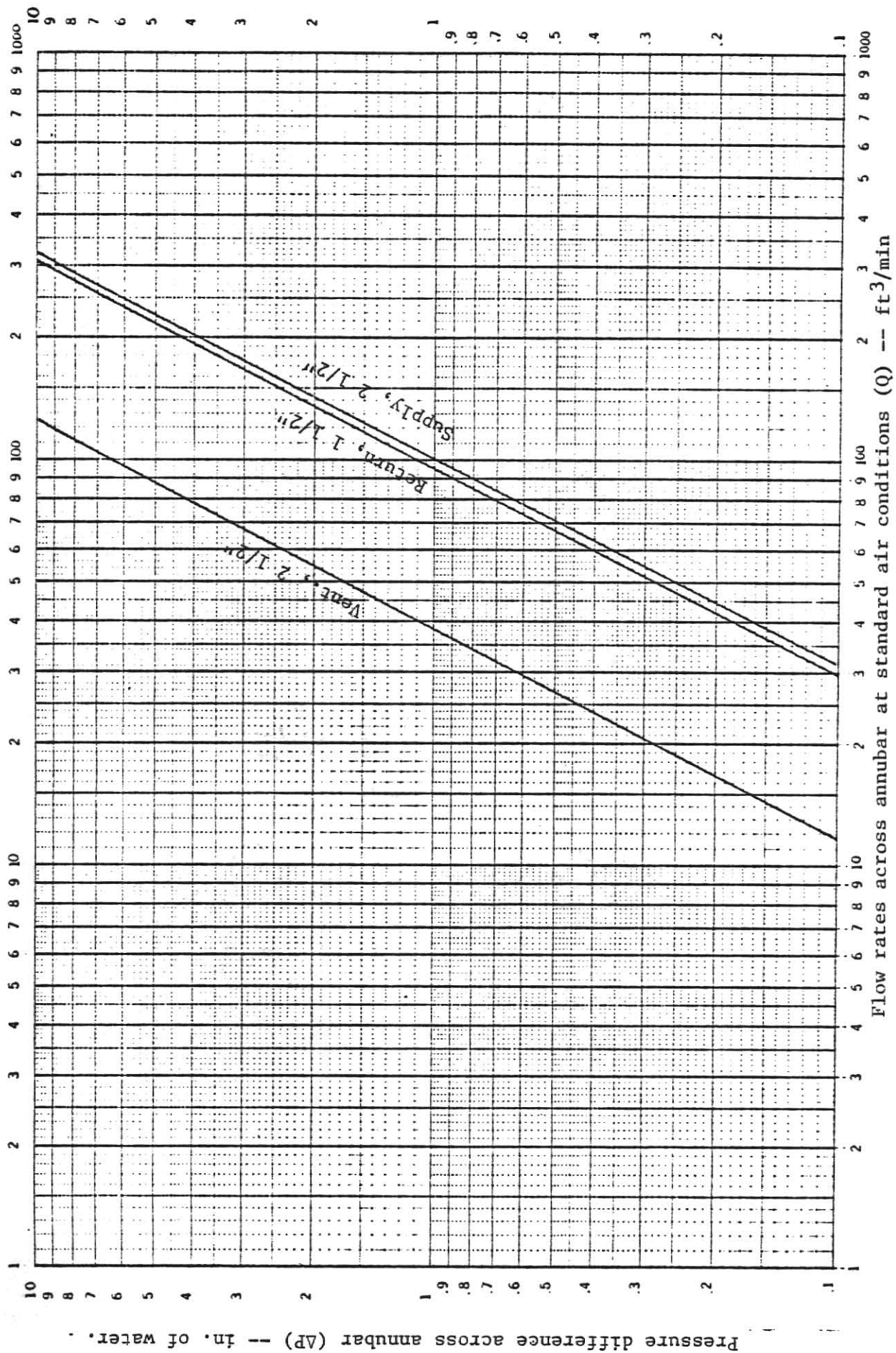


Figure 17. Annubars calibration chart.

APPENDIX D

THERMAL MODELING

A scale model study of the thermal stratification process with simulation of heat loads and air quantities was undertaken to predict the temperatures that would exist in a full scale high ceiling industrial plant. For that purpose an 8' x 8' x 8' chamber was built and pertinent thermal characteristics were examined over a range of variables. In this section the requirements for thermal similarity between model and prototype is discussed by developing the energy balance equations on each individual heat source (or zones, as shown in Figure 2) in the model and prototype.

Thermal similitude is defined as the state for which the ratio between the temperature difference between any two points in the model to the temperature difference between homologous points in the prototype is constant (12). A more restrictive definition would be that air temperature and all surface temperatures in the prototype be identical to the temperatures at the corresponding points in the model. Parczewski and Renzi (12) state that thermal similarity will be achieved if convection, conduction, and radiation heat fluxes (heat/unit area) are identical in model and prototype, which requires

$$\frac{q_c}{Q_c} = \frac{q_R}{Q_R} = \frac{q_k}{Q_k} = \frac{q_T}{Q_T} = 1.$$

This definition of thermal similarity, in addition to equality of mass flow rates per unit area in the model and prototype, are being considered throughout this discussion. In developing the appropriate modeling relationships, the subscript m is used for model quantities and P for prototype quantities.

LOWER ZONE

Referring to Figure 2 in Chapter III and assuming steady state conditions in the lower zone, the energy balance in the air-conditioned space for the model and prototype can be written as

$$\begin{aligned} \text{Model} \quad q_{IC} + q_{ID} + q_{IF} - (\dot{m}_c + \dot{m}_v) C_p T_c + \dot{m}_c C_p T_{Ls} + \dot{m}_{Rs} C_p T_s \\ - \dot{m}_{RR} C_p T_c = 0 \end{aligned}$$

$$\begin{aligned} \text{Prototype} \quad Q_{IC} + Q_{ID} + Q_{IF} - (\dot{m}_c + \dot{m}_v) C_p T_c + \dot{m}_c C_p T_{Ls} + \dot{m}_{Rs} C_p T_s \\ - \dot{m}_{RR} C_p T_c = 0 \end{aligned}$$

Equality of mass flow rates per unit area of floor of the model and prototype are assumed. Also, from the definition of thermal similarity, identical temperatures at corresponding points were assumed in the model and prototype. These assumptions required that in the above equations;

$$\dot{m}_{cm} = \dot{m}_{cp}, \quad \dot{m}_{vm} = \dot{m}_{vp}, \quad \dot{m}_{Rsm} = \dot{m}_{Rs}, \quad \dot{m}_{RRm} = \dot{m}_{RRp}$$

$$T_{cm} = T_{cp}, \quad T_{sm} = T_{sp}, \quad T_{Lsm} = T_{Lsp}$$

The internal concentrated load was taken to be zero in the model study

$$(q_{ICm} = Q_{ICp} = 0).$$

From the above energy balance equations, thermal similitude between model and prototype in the conditioned zone occurs if

$$q_{ID})_m = Q_{ID})_p \quad (D.1)$$

and

$$q_{IF})_m = Q_{IF})_p \quad (D.2)$$

Equation (1) deals with internal disperse load in the lower zone and can be written

$$h_{Dm} (T_D - T_c)_m = h_{Dp} (T_D - T_c)_p \quad (D.1.a.)$$

or

$$\frac{h_{Dm}}{h_{Dp}} = \frac{(T_D - T_c)_p}{(T_D - T_c)_m} \quad (D.1.b.)$$

If T_D is equal in model and prototype ($T_{Dm} = T_{Dp}$), Eq. (D.1.b.) reduces to

$$\frac{h_{Dm}}{h_{Dp}} = 1 \quad \text{or} \quad h_{Dm} = h_{Dp}$$

Following the same procedure for Eq. (D.2.) it can be written

$$h_{Fm} (T_F - T_c)_m = h_{Fp} (T_F - T_c)_p \quad (D.2.a.)$$

or

$$\frac{h_{Fm}}{h_{Fp}} = \frac{(T_F - T_c)_p}{(T_F - T_c)_m} \quad (D.2.b.)$$

If $T_{F_m} = T_{F_p}$ and $T_{C_m} = T_{C_p}$, then $h_{F_m} = h_{F_p}$.

Equality of the convection coefficients (h_D) and floor convection coefficients (h_F) in the model and prototype requires that they be independent of length of the heated surface. For free convection heat transfer on heated plates facing upward in the turbulent range [$2 \times 10^7 < Gr Pr < 3 \times 10^{10}$] McAdams (12) recommends the equation

$$Nu = \frac{hL}{K} = 0.14 \left[\frac{L^3 \rho^2 g \beta \Delta t}{\mu^2} \left(\frac{C_p \mu}{K} \right) \right]^{1/3}$$

For air at atmospheric pressure and at room temperature this equation simplifies to

$$h = 0.22 \Delta t^{1/3}$$

For the model the product of Grashof number and Prandtl number is in the turbulent range. A flow of the same type (turbulent) is assumed to exist in the lower zone of the prototype. As shown in the above equation, in turbulent free convection, the geometrical factor (L) cancels out, and hence the convection coefficient (h) is independent of the length (L) of the heated surface.

UPPER ZONE

Due to change in temperature of upper strata, the conditions of air in the upper zone are assumed to be unsteady and vary with time. The energy balance equation for the upper zone in the model and prototype can

then be written as

$$\begin{aligned} \text{Model } q_{RC} + q_{LC} + (\dot{m}_v + \dot{m}_c) C_p T_c - \dot{m}_c C_p T_{Ls} - \dot{m}_v C_p T_{Us} \\ = \rho C_p h'_{uz} \left. \frac{d \bar{T}_{uz}}{d\theta} \right|_m \end{aligned}$$

$$\begin{aligned} \text{Prototype } Q_{RC} + Q_{LC} + (\dot{m}_v + \dot{m}_c) C_p T_c - \dot{m}_c C_p T_{Ls} - \dot{m}_v C_p T_{Us} \\ = \rho C_p h'_{uz} \left. \frac{d \bar{T}_{uz}}{d\theta} \right|_p \end{aligned}$$

From previous assumptions made between the model and prototype:

- a) similar mass flow rates per unit area of floor
- b) similar heat fluxes (Btu/hr-ft²)

Thermal similitude applies if

$$q_{RC)m} = Q_{RC)p \quad (D.3.)$$

$$q_{LC)m} = Q_{LC)p \quad (D.4.)$$

and

$$\rho C_p h'_{uz} \left. \frac{d \bar{T}_{uz}}{d\theta} \right|_m = \rho C_p h'_{uz} \left. \frac{d \bar{T}_{uz}}{d\theta} \right|_p \quad (D.5.)$$

Equations (3) and (4) can be written as:

$$h_{R_m} (T_r - T_{uz})_m = h_{R_p} (T_r - T_{uz})_p \quad (D.3.a.)$$

$$h_{L_m} (T_L - T_{uz})_m = h_{L_p} (T_L - T_{uz})_p \quad (D.4.a.)$$

Identical temperatures of roof (T_r), upper zone (T_{uz}) and lights (T_L) simplifies equations (D.3.a.) and (D.4.a.) to

$$h_{R_m} = h_{R_p} \quad (D.3.b.)$$

and

$$h_{L_m} = h_{L_p} \quad (D.4.b.)$$

The above conditions assure equality of heat transfer by convection of the roof and lights in the model and prototype.

In Equation (5) let $\rho_m = \rho_p$ and $C_{p_m} = C_{p_p}$ then

$$h'_{uz_m} \frac{d \bar{T}_{uzm}}{d\theta_m} = h'_{uz_p} \frac{d \bar{T}_{uzp}}{d\theta_p} \quad (D.5.a.)$$

Now let $\frac{h'_{uzm}}{h'_{uzp}} = S_u$ height of upper zone scale factor, then substitute into Eq. (D.5.a.)

$$S_u \frac{d \bar{T}_{uzm}}{d\theta_m} = \frac{d \bar{T}_{uzp}}{d\theta_p} \quad (D.5.b.)$$

also let $\frac{d\theta_m}{d\theta_p} = S_t$ time scale factor, and substitute again into Eq.

(D.5.b.)

$$S_u \frac{d \bar{T}_{uzm}}{S_t d\theta_p} = \frac{d \bar{T}_{uzp}}{d\theta_p}$$

or

$$S_u d \bar{T}_{uzm} = S_t d \bar{T}_{uzp}$$

If $S_u = S_t$, then $d \bar{T}_{uzm} = d \bar{T}_{uzp}$

Example: Assume for a 1/8 scale model $S_u = 1/8$ and $S_t = 1/8$ then

$d\theta_m = 1/8 d\theta_p$ which means one hour model operation is the same as eight hours for the prototype.

ROOF

In an actual plant, due to diurnal variations in solar radiation the heat storage inside the roof structure is unsteady and a function of time. Therefore for the model and prototype energy balance equations for the roof can be expressed as

$$\text{Model} \quad q_R - q_{RR} - q_{RC} = \rho_R C_R h'_R \left. \frac{d \bar{T}_r}{d\theta} \right)_m$$

$$\text{Prototype} \quad Q_R - Q_{RR} - R_{RC} = \rho_R C_R h'_R \left. \frac{d \bar{T}_r}{d\theta} \right)_p$$

Based on assumptions made earlier, the requirements for thermal similarity are

$$q_R)_m = Q_R)_p \quad (D.6.)$$

$$q_{RR})_m = Q_{RR})_p \quad (D.7.)$$

$$q_{RC})_m = q_{RC})_p \quad (D.8.)$$

$$\rho_R C_R h'_R \left. \frac{d \bar{T}_r}{d\theta} \right)_m = \rho_R C_R h'_R \left. \frac{d \bar{T}_r}{d\theta} \right)_p \quad (D.9.)$$

In Eq. (6), q_R is the total power input to the roof per unit area of the model, which was considered to be in the range of roof heat load of an industrial plant. Equation (7) deals with radiation heat transfer of the roof and it is assumed that the roof radiates heat only to the floor. Therefore Eq. (7) can be written

$$\frac{q_{RR})_m}{q_{RR})_p} = \frac{F_{RFm} \sigma (T_r^4 - T_F^4)_m}{F_{RFp} \sigma (T_r^4 - T_F^4)_p} \quad (D.7.a.)$$

which reduces to

$$\frac{F_{RFm}}{F_{RFp}} = \frac{(T_r^4 - T_F^4)_p}{(T_r^4 - T_F^4)_m} \quad (D.7.b.)$$

where F_{RF} is the geometric shape factor. If the prototype is of the same geometry as the model then $F_{RFm} = F_{RFp}$. Therefore from Eq. (D.7.b.) there should exist the relation

$$(T_r^4 - T_F^4)_m = (T_r^4 - T_F^4)_p \quad (D.7.c.)$$

In the model, the roof and floor loads were chosen to simulate the roof and floor loads in a prototype enclosure and to reach the same corresponding

temperatures. Thus Eq. (D.7.c.) requires equality of temperatures of the roof or floor in the model and prototype,

$$T_{r_m} = T_{r_p} \quad \text{and} \quad T_{F_m} = T_{F_p}$$

The relation between shape factors and temperatures inside the enclosure are discussed more, later in this section.

Equation (D.8.) is the equality of roof heat fluxes convected to the upper zone in the model and prototype, which can be written

$$h_{R_m} (T_r - T_{uz})_m = h_{R_p} (T_r - T_{uz})_p \quad (D.8.a.)$$

Near the roof, the process of natural convection heat transfer is considered laminar since warm air in the upper strata is motionless or is being exhausted through ventilation ducts with a small percentage of supply air flow rate. So for heated plates facing downward in laminar range ($3 \times 10^5 < Gr \, Pr < 3 \times 10^{10}$) the Nusselt number is

$$NU = \frac{h_R L}{K} = 0.27 \left[\frac{L^3 \rho^2 g \beta \Delta T}{\mu^2} \left(\frac{C_p \mu}{K} \right) \right]^{0.25}$$

which for air at atmospheric pressure simplifies to

$$h_R = 0.12 \left(\frac{\Delta T}{L} \right)^{0.25}$$

This equation implies that the convection coefficient of the roof of the model and prototype is a weak function of the characteristic length (L) of the roof.

Recalling Eq. (D.9)

$$\left. \rho_R C_R h'_R \frac{d \bar{T}_r}{d\theta} \right)_m = \left. \rho_R C_R h'_R \frac{d \bar{T}_r}{d\theta} \right)_p \quad (D.9.)$$

Let

$$\frac{\rho_{Rm}}{\rho_{Rp}} = S_\rho = \text{density scale factor}$$

$$\frac{C_{Rm}}{C_{Rp}} = S_c = \text{specific heat scale factor}$$

$$\frac{h'_{Rm}}{h'_{Rp}} = S_d = \text{roof length scale factor}$$

and

$$\frac{d\theta_m}{d\theta_p} = S_t = \text{time scale factor}$$

Substitute these scale factors into Eq. (D.9.)

$$S_\rho S_c S_d \frac{d \bar{T}_{rm}}{S_t d\theta_p} = \frac{d \bar{T}_{rp}}{d\theta_p} \quad (D.9.a.)$$

or

$$\frac{d \bar{T}_{rm}}{d \bar{T}_{rp}} = \frac{S_t}{S_\rho S_c S_d} \quad (D.9.b.)$$

From the above equation, it can be said that $\rho C_R h'_R$ for the model and prototype can be chosen to give the relation $S_\rho S_c S_d = S_t$ which implies

$$d \bar{T}_{rm} = d \bar{T}_{rp} .$$

FLOOR

The energy balance equation on the floor surface is described as

$$\text{Model } q_{RR} + q_{LR} - q_{IF} - q_F = \rho_F C_F h'_F \left. \frac{d \bar{T}_F}{d\theta} \right)_m$$

$$\text{Prototype } Q_{RR} + Q_{LR} - Q_{IF} - Q_F = \rho_F C_F h'_F \left. \frac{d \bar{T}_F}{d\theta} \right)_p$$

In the model the floor is insulated and heat loss by the floor (q_{F_m}) is considered negligible, and for the same reason the floor heat loss in prototype is also neglected ($Q_{F_p} = 0$). The temperature of the floor was assumed to remain constant. Therefore $\frac{d \bar{T}_F}{d\theta} = 0$. Thus from above equations, thermal similarity of model and prototype requires

$$q_{RR)m} = Q_{RR)p} \quad (D.10.)$$

$$q_{LR)m} = Q_{LR)p} \quad (D.11.)$$

and

$$q_{IF)m} = Q_{IF)p} \quad (D.12.)$$

Equation (D.12) is similar to Eq. (D.2.) discussed in the lower zone energy balance section.

Equation (D.10.) expands to

$$F_{RF_m} \sigma (T_r^4 - T_F^4)_m = F_{RF_p} \sigma (T_r^4 - T_F^4)_p \quad (D.10.a.)$$

or

$$\frac{F_{RF\ p}}{F_{RF\ m}} = \frac{(T_r^4 - T_F^4)_m}{(T_r^4 - T_F^4)_p}$$

Also Eq. (D.11.) concerning thermal heat transfer from the lights to the floor simplifies to

$$\frac{F_{LF\ p}}{F_{LF\ m}} = \frac{(T_L^4 - T_F^4)_m}{(T_L^4 - T_F^4)_p} \quad (D.11.a.)$$

If the model and prototype have similar geometry ($F_{RF\ m} = F_{RF\ p}$), and similar type lighting system is used in the prototype ($F_{LF\ m} = F_{LF\ p}$), then Equations (D.10.b) and (D.11.a) reduce to

$$(T_r^4 - T_F^4)_m = (T_r^4 - T_F^4)_p \quad (D.10.c.)$$

and

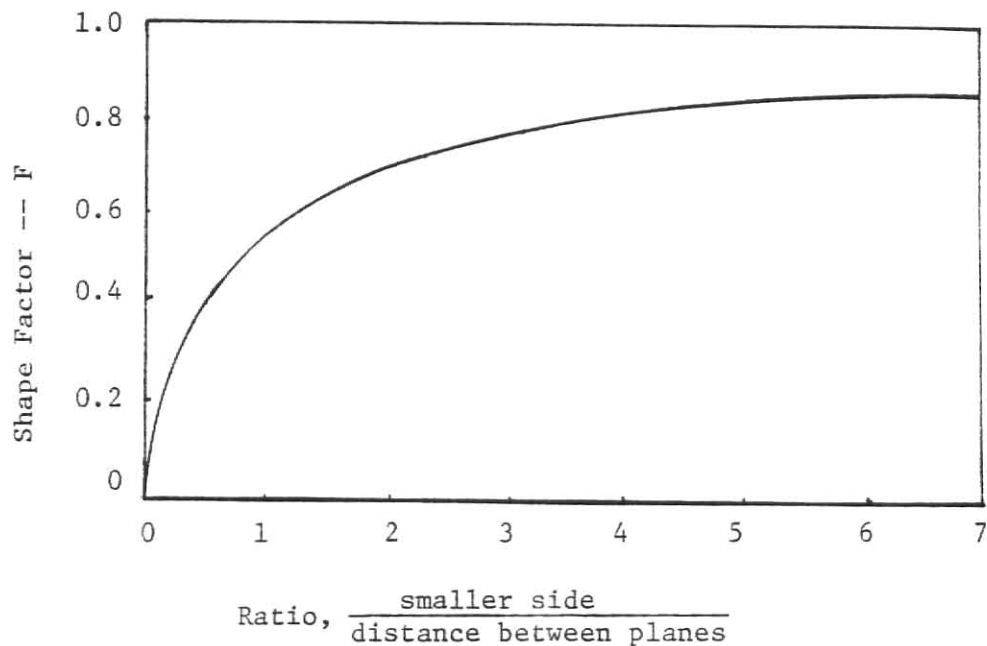
$$(T_L^4 - T_F^4)_m = (T_L^4 - T_F^4)_p \quad (D.11.b.)$$

The discussion carried out for Eq. (D.7.) also applies to equations (D.10.c) and (D.11.b).

DISCUSSION OF SHAPE FACTORS

The fraction of diffusely distributed radiation leaving one surface and reaching another surface is called the radiation shape factor (F). For this model and prototype shape factors are found using the ratio of smaller side of the plane (roof) to the distance between the planes

(height). For planes connected by non-conducting but reradiating walls, Kreith (16) presents the figure shown below.



For thermal similarity, the shape factors were related by the equation

$$\frac{F_p}{F_m} = \frac{(T_r^4 - T_F^4)_m}{(T_r^4 - T_F^4)_p}$$

From the previous figure, at model ratio of $\frac{8}{7}$ and prototype ratio of $\frac{8}{8}$ it can be written

$$\frac{.55}{.57} = .965 = \frac{(T_r^4 - T_F^4)_m}{(T_r^4 - T_F^4)_p}$$

which shows the radiation heat transfer in the prototype would be 3.5% greater than that in the model. Estimating the roof radiant load to be 30% of its total load, then $3.5\% (.3) = .0105 = 1.05\%$ is the resulting error in total heat flux. But for a model ratio of $\frac{8}{2}$ and prototype ratio of $\frac{8}{8}$, there exists

$$\frac{.55}{.81} = .679 = \frac{(T_r^4 - T_F^4)_m}{(T_r^4 - T_F^4)_p}$$

Then the error would be $1 - .679 = .321 = 32.1\%$ or
 $.321 (.3) = .0963 = 9.63\%$ of the total heat flux.

In conclusion, the identical heat transfer per unit area assumption in the model and prototype is valid for all energy exchanges except radiant, for which requires equality of shape factors or similar geometries. Equal heat fluxes and equal flow rates per unit area also cause corresponding surface temperatures in the model and prototype to be equal, with small error.

APPENDIX E

PRELIMINARY MODEL STUDY

To obtain some design information prior to final design of the main model (8' x 8' x 8'), a preliminary model 3' x 1' x 2' high was constructed and tested over a limited range of loads. The preliminary model was used to establish the existence of stratification under simulation of heat loads and air quantities to be used in the experiments with the larger model. Figures 18 and 19 show views of the preliminary model.

The walls of the small model were constructed with 3/4" hardboard except the front wall which consisted of a 3' x 2' (1/8" thick) glass sliding into the 1/8" slots located on the edges of the side walls. The ventilation exhaust air ducts were positioned inside the model chamber through an 8-inch vertical slot located in the middle of the right side wall. The duct was 35" long (7/8" o.d.) across the length of the chamber with 1/8" diameter perforations located horizontally at 1" increments. The ventilation duct was connected to a valve and a U-tube manometer was used for measurement of the amount of discharged air. Copper tubes were used as ducts in the small model.

The back wall with inside dimensions of 3' x 2' had six vertical 12-inch slots at 5 inches apart. The slots were 1-inch wide and were cut from 6" above the floor level to 6" below the roof level. The supply ducts were placed through these slots into the chamber and their heights were adjustable over a 12 inch range. The ducts were 12 inches in length with an outside diameter of 3/4". For air distribution, 1/8" diameter perforations were positioned 2 inches apart horizontally on the ducts

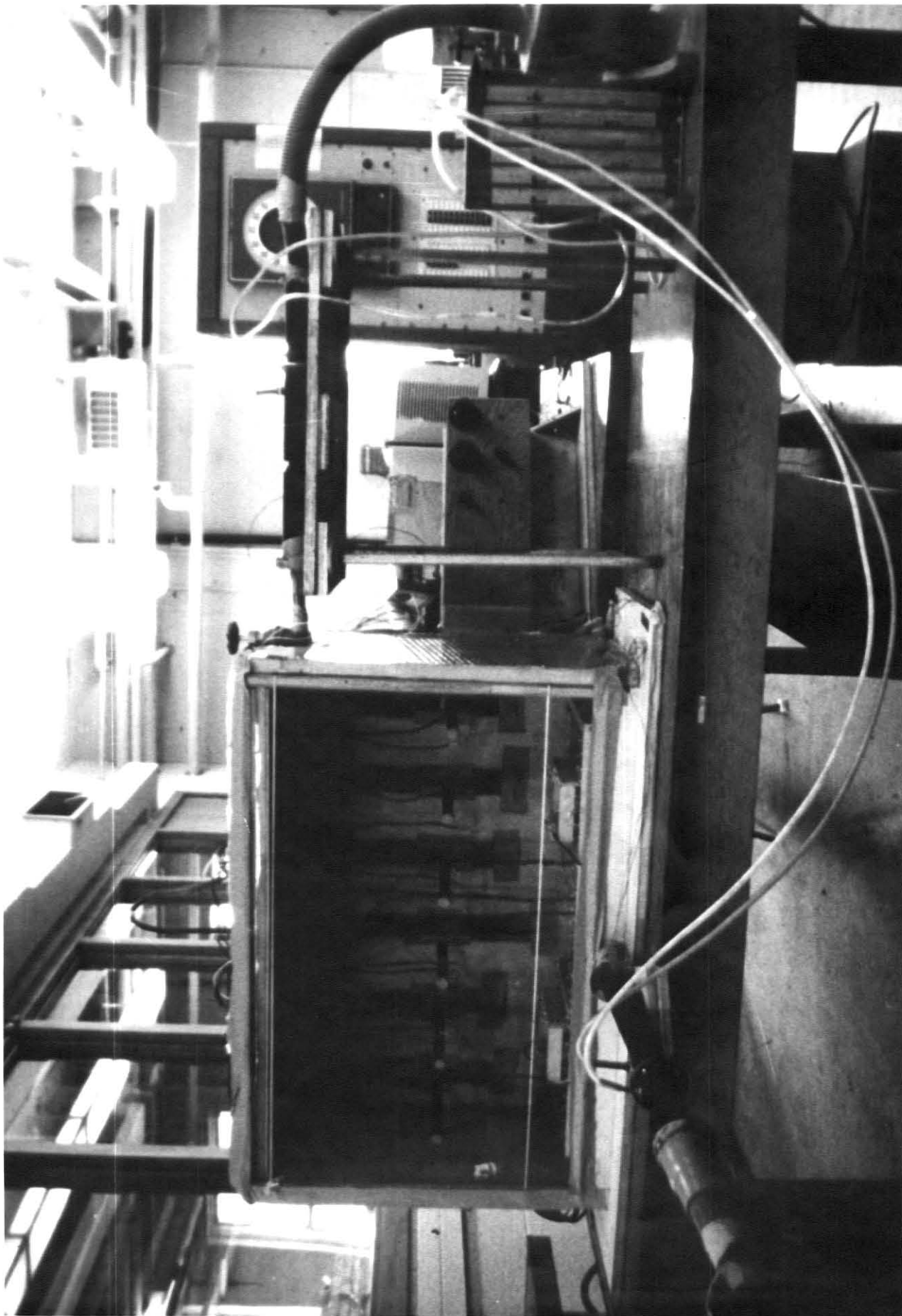


Figure 18. A front view of the preliminary model.

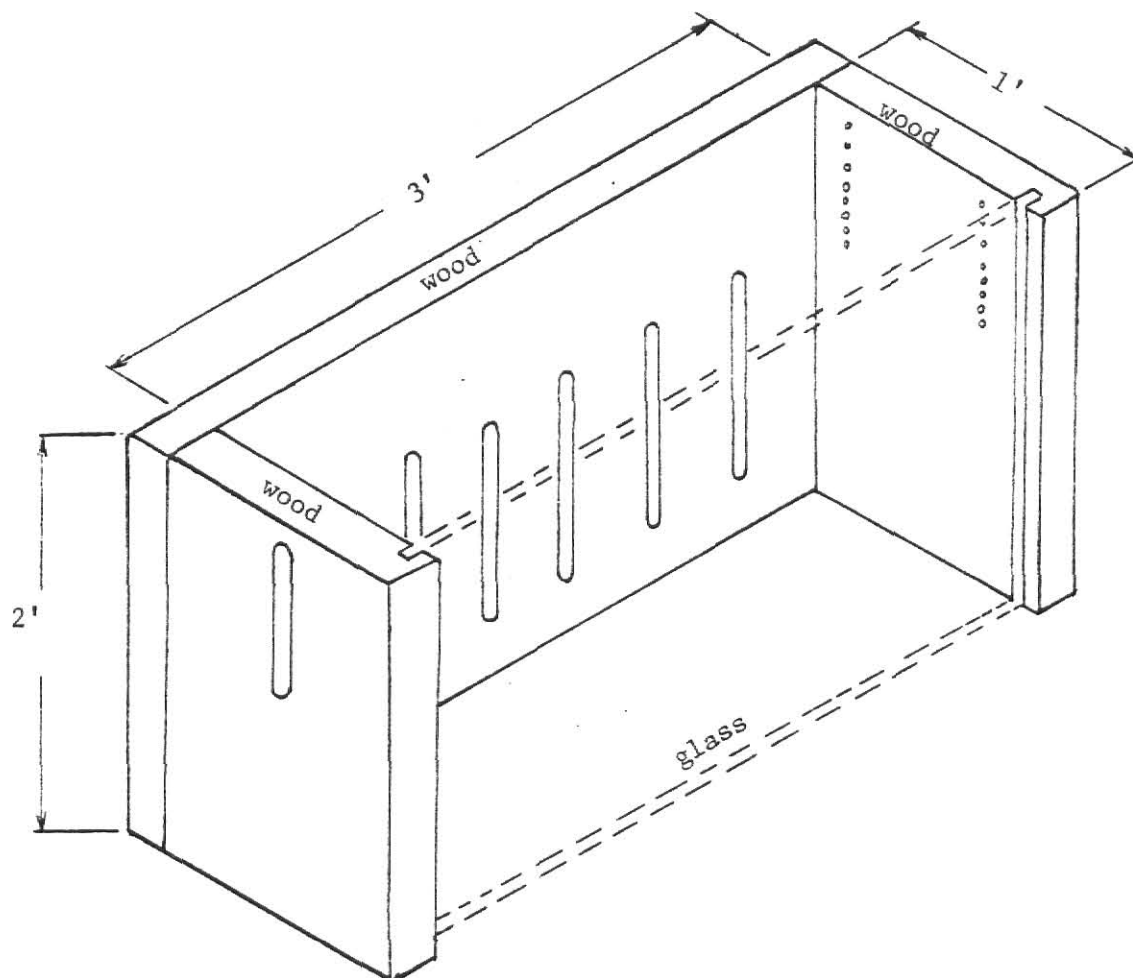


Figure 19. Isometric view of the small model chamber.

with two more perforations located 30° from previous openings on both sides of the ducts.

The floor of the small chamber was built of $1/8$ " perforated plywood located 2" above the bottom surface of the chamber. Six $7/8$ " diameter return air ducts were placed under the perforated board. Return air through the board and into the ducts. These six ducts were connected to one $1\ 1/2$ " duct coming out of the chamber through the front. The perforations on return ducts were arranged the same as the ones on the ventilation ducts.

The roof of the preliminary model was made of $3/4$ " hardboard with an $1/8$ " aluminum plate underneath. The roof was supported by four small angles placed into the side walls. The height of the roof was adjustable (at 1" increments) over a 9 inch range. Twenty seven openings ($5/16$ " in diameter) were located through the roof at 4" increments for placing thermocouple posts through and into the chamber. Also, there were sixteen ($1/8$ " diameter) perforations for positioning electric wires inside the chamber for purposes of simulating the lighting load.

Nichrome resistance wire was used to produce the simulated roof, floor, and lights loads. The resistance of the three load components, roof, floor, and lights were measured to be 16, 18.5, and 5.45 ohms. The roof resistors were placed between the aluminum plate and the hardboard. The height of the lights resistors were adjustable, and the floor resistors were located near the floor of the chamber. The power input to these resistors was controlled and measured by variacs and wattmeters.

Twelve thermocouples were positioned at 1" increments on five posts inside the chamber. The posts could be moved through the openings in the roof to measure the temperatures at any level and any point inside the chamber. The temperatures of supply, return ventilation air, and roof

surface were indicated by different thermocouples. The Honeywell multichannel potentiometer was used to indicate the millivolts sensed by thermocouples.

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VITA

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DESIGN, CONSTRUCTION AND CALIBRATION
OF A MODEL CHAMBER FOR EXPERIMENTATION WITH
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ABSTRACT

Concern with energy availability and with cost of total volume air-conditioning in high ceiling industrial plants has generated an interest in thermal stratified cooling systems. This system is also known as "vertically-zoned," or "low level system," and involves cooling only of the lower, occupied space while leaving the upper space uncooled. Thus, the space is divided into two parts, a lower air conditioned zone of practically uniform temperature and an upper zone of higher, varying temperature.

This thesis describes the design, construction, and calibration of physical model of a high ceiling industrial plant to be used to provide design information about stratified cooling systems. The model is 8' x 8' x 8' high, built of 2 x 4 stud construction with the stud space fully insulated. Electric heating panels were used to simulate the thermal loads at the roof and floor. Nine 40-watt incandescent light bulbs were positioned inside the chamber to simulate the lighting load. The power input to the roof, floor, and lights were selected in the range of zero to 13.2, 8.7, and 4.2 watts/ft², respectively. The supply air flow rate can be varied up to 3.2 cfm/ft², with ventilation flow rate from zero to 100% of the supply flow rate. The height of roof, lights, supply ducts, ventilation ducts, and return ducts are adjustable to any level within the 8-foot high structure. Thermocouples were positioned on five wooden posts for measurement of the air temperature from floor to roof in one-foot increments. A thermocouple in the supply duct line at the entry to the chamber indicates the cool supply air temperature. The return and

ventilation air thermocouples were located in the return and ventilation lines at the exit from the chamber.

The chamber has been tested and was found to operate satisfactorily over the range of variables for which it was designed. Initial test results indicate that it is a reliable system, suitable for use in the stratification studies for which it was built.