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Thermal comfort in office buildings

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THERMAL COMFORT IN OFFICE BUILDINGS

by
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CD for temperature, air distribution and concentration profiles of contaminants in several arrangements of air distribution in a typical office space. The

study included a long term monitoring of ambient air temperature and humidity, operating temperature, draft velocity, vertical temperature gradient and carbon dioxide concentration using standard three elevation stands as recommended by ASHRAE 55-04 and the additional carbon dioxide sensor. The preliminary results show that as far as the last two measured variables no perceived variation existed between the perception questionnaire responses and the guidelines of the ASHRAE standard 55-04. In the first four measured variables though there were some differences between the perception survey responses and the guidelines. Whereby the ASHRAE guidelines seem to be met by enlarge in the three buildings when averaged over time the survey responses did indicate responses of slight thermal discomfort. These responses ranged from slightly cool regarding ambient temperature, slightly dry for humidity, slightly drafty for air velocity to slightly cool for the operating temperature. The results of a 3D transient CFD simulations using STAR-CD are presented in this thesis. Three strategies of delivering supply air methods are investigated: 1) Ceiling air diffuser, 2) Under the floor air diffuser and,3) High wall air diffuser. Various profiles of velocity distributions/temperature and a simulated pollutant such as carbon dioxide were plotted as a function of time. A fixed amount of volumetric air flow was assumed for the supply air. It was found the there were significant variations in the temperature and velocity amongst the different supply air delivery strategies. An interesting time lag is noticed between when the air starts coming into the room and when the temperatures reach a minimum at a point in the

center of the room volume. The Ceiling Air Diffuser achieved better results as far as thermal stability and velocity uniformity was concerned uniformity in the room.

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CHAPTER 1

INTRODUCTION

The objectives of this thesis are:

1. Produce a written report on the literature and existing databases available for development of a protocol for measurement and verification of integrated building performance.

2. Develop a written protocol and supporting database structure for measurement and verification of integrated building performance ("the IBP Protocol") and selection criteria for participating buildings.

3. Conduct field application of the IBP protocol and collect and enter data in the database.

4. Conduct a computer simulated result to show the thermal comfort parameters fall within the specified values of ASHRAE 55-2004.

Four studies which have been done in the past have been chosen initially as they were related to the study being done here. These previous studies have been funded by ASHRAE that have dealt with similar studies they have been done at various places around the world.

The first report which stands as a herald to these reports was performed in the San Francisco titled "Field Measurement System for the Study of Thermal Comfort" authored by Dr.Charles.C.Benton et al. describes the instrumentation and protocol used for a filed study of environmental conditions and occupant comfort in 10 office buildings located in the San Francisco bay area. A total of 2342 visits to 304 participants were done during this survey. Standards referred were ASHRAE 55 81 and ISO 7730.

The field measurements included all variables specified in ASHRAE (1981) and ISO (1985) including repetition of measurements at 0.1m, 0.6mand 1.1m above the floor. For many workstation visits this repetition was excessive and average data measured at 0.6m would have been sufficient to characterize the local environment. The great majority of the analysis was based on average data for the last three minutes of each workstation visit.

As an addendum to this report, Dr.G.E. Schiller authored a paper titled "A Comparison of measured and Predicted Comfort in Office Buildings" whose aim was to determine the extent to which theoretical and laboratory based equations accurately predict workers thermal responses in existing office buildings. It also deals with comparison of both direct and indirect assessment of comfort in these buildings to examine the validity of associating thermal comfort with thermal neutrality.

This paper measured both measured and predicted responses in existing office buildings and examined the extent to which the workers sense of comfort was associated with thermal neutrality.

The range of thermal environments that existed in the office buildings during the measurements period was fairly narrow, although the thermal sensation votes covered the full range of the seven point ASHRAE thermal sensation scale.

Neutral temperatures in the buildings were lower than predicted from laboratory based comfort models. Optimum acceptability in the office environments was 12% dissatisfied compared to a predicted minimum of 5% based on experiments in laboratory conditions. This was explained by either the wider range of clothing worn by office workers at any given ET*(Effective Temperature).

The second and third reports were done in Australia in two different climatic conditions namely hot and arid conditions and hot and humid conditions in Kalgoorlie and Townsville respectively. They are titled "Field experiments on occupant comfort and office thermal environments in a hot-humid climate" and "Field study of occupant comfort and office thermal environments in a hot, arid climate" Authored by de Dear, Richard J. et al.

The former dealt with field investigation of indoor climates and occupant comfort in 12 air-conditioned office buildings in Townsville, located in Australia's tropical north. The project replicates an earlier ASHRAE investigation in San Francisco (RP-462). A total of 836 subjects provided 1,234 sets of questionnaire responses, each accompanied by a full set of physical indoor climatic measurements from laboratory-grade instrumentation.

The latter stressed on field study of occupant comfort and office thermal environment in 22 air-conditioned office buildings in Kalgoorlie-Boulder, Western Australia, a location characterized by a hot and arid climate. A total of 935 subjects provided 1,229 sets of data for winter and summer, each accompanied by a full set of indoor climatic measurements with laboratory-grade instrumentation. Standards referred were ASHRAE 55a 92 and ISO 7730.

Clothing insulation levels approximated to the standard ASHRAE 55 assumed summer value of 0.5 clo with a slightly more(0.1 clo) being worn in the dry season than in the wet .Chairs were estimated to add 0.15 clo to the clothing insulation of the office workers.

Thermal neutrality according to the ASHRAE seven point scale occurred at about 24.4C in both the seasons. Preferred temperature, defined as a minimum of subjects requesting temperature change, was one degree cooler than neutrality, 23.5C.

Little more than 50% of the indoor climatic observations fell within standard 55 summer comfort zone. Group mean thermal sensations showed a heightened sensitivity to temperature, changing approximately one unit on the ASHRAE seven point scale per 2C change in operative temperature.

The third report titled "Field study of occupant comfort and office thermal environments in a cold climate" authored by Donnini, Giovanna et.al was performed in Quebec Canada and it emphasized on office thermal environments in 12 mechanically ventilated office buildings in southern Quebec. A total of 877 subjects were surveyed during hot and cold months. Each interview provided a set of responses to a questionnaire and a set of physical indoor climatic measurements. Standards referred were ASHRAE 55 81 and ISO 7730.

Clothing insulation levels (0.7 clo in summer and 1.1 clo in winter) were slightly higher than those assumed in ASHRAE standard of 0.5 clo in summer and 0.9 clo in winter. This was due to the fact that clothing insulation effect of chairs added up to 0.15 clo in the summer and 0.19 clo during winter. Metabolic rates were estimated to be 1.21 met.

Thermal neutrality according to the ASHRAE seven point scale occurred at about 24.1C in the summer and at about 22.8C in winter. Preferred temperature, defined as a minimum number of subjects requesting a change in temperature was approximately one degree cooler than neutrality in both the seasons at 23C in the summer and 22C in winter. Only 63% of the indoor climatic observations fell within the ASHRAE standard 55 summer comfort zone; 27% of the observations fell within the winter comfort zone. Neither the ASHRAE standard 55 nor the ISO 7730 PPD index matched observations level of the thermal acceptability with useful accuracy.

There was little difference between the sexes in terms of thermal sensation, although there were significantly more frequent expressions of thermal dissatisfaction from the females in the sample despite their thermal environments being no different from the males.

The fourth report titled "A field study of the thermal environment in residential buildings in Harbin" authored by Wang, Zhao-Jun; et. al was performed in Northeastern China deals with the main findings of Project HIT.2000.25 supported by the Scientific Research Foundation of Harbin Institute of Technology, a field study of indoor climates and occupant comfort in 66 residential buildings in Harbin, located in northeastern China. One hundred and twenty sets of questionnaire responses were provided from 120 subjects for winter, each accompanied by a full set of physical indoor climatic measurements with an indoor climate analyzer and a thermal comfort meter, which met ASHRAE Standard 55-1992 and ISO Standard 7726 (1985) for accuracy, duration of sample, and response time. Standards referred were ASHRAE 55 81 and ISO 7726.

According to the ASHRAE 55-1992 and ISO 7730 comfort standards only 77.5 %of the thermal conditions fall within the comfort zone of 16.5C~22.5C;91.7% of the occupants considered their thermal environments to be acceptable due to adaptability measures taken by occupants.

Eighty percent of the occupants were comfortable with an operative temperature of 18C~25.5C which was higher than the winter comfort zone of 16.5C~22.5C.

Neutral operative temperature was 21.5C and preferred operative temperature was 21.9C. Around 81.1% of the occupants felt dry at relative humidity of 20%~30% while over 40% of the occupants felt dry from relative humidity 30%~55%.The lowest limit of relative humidity was recommended to be 30% instead of the 20% as suggested by the standards.

Only 47.6% of the people felt slightly cool at their feet when the vertical air temperature difference between the 1.1 and the 0.1 meters was over 3C and the operative temperature at 0.1 meter level was lower than 16C.So a conclusion that the vertical air temperature difference is not the main factor influencing human comfort, but the operative temperature at 0.1 meter level is the key was reached. Some additional references have been added at the end of this chapter.

Currently, buildings do not perform optimally or close to the predictions of many codes and forecasts largely because they are field assembled and there is no consistent procedure to identify deficiencies or to correct them. Solving this problem in new and existing buildings requires field performance evaluations using appropriate and agreed upon procedures. Many procedural elements to test the energy performance of energy-related components and systems of building and to assess their impacts on health and

safety already exist but not in a unified way. Some are ready to be integrated into a new process called non-residential commissioning. There are some parameters that play an important role in defining thermal parameters. Some of them are listed in the Appendix A.

The thermal comfort study requires several types of measurements according to the ASHRAE 55-99 standard and at different elevations. Previous studies have suggested also that in addition to the localized thermal comfort measurements at the different workstations there needs to be a "static measurement" of some of the variables at a representative location inside the space being monitored.

Calculating procedures for selected parameters related to thermal comfort are shown in Appendix A. All protocols are based on current international and national standards (ASHRAE 55-2004, ASHRAE 55a, ISO 7730 etc), proposed standards and recent literature. Measuring procedures for the environmental parameters - Thermal comfort see Appendix A.The sensors that are used in the measurement of the thermal comfort conditions take into account an arbitrary value which holds good for the Metabolic activities and the Clothing insulation levels which are important parameters in calculating the indices.

The significance of the three heights corresponds to an occupant either standing or sitting. These measurements are averaged out on a 3 minute basis which helps in creating a database for addressing all the thermal comfort indices on a general basis. From previous studies done by ASHRAE funded projects RP 884 and RP 921, as a protocol average values of the various parameters like the air velocity, temperature have been used in the final calculation of the Indices.

From previous studies the air temperature is averaged out from all the three heights similarly air velocity and the mean radiant temperature are also averaged on all the three heights. The relative humidity is measured at only one location. The formulae which have been presented in Appendix A for calculating the thermal indices are obtained by averaging the data on an hourly and daily basis. The weighted moving average was used for calculations. The Appendix gives us an estimate as to how the various thermal comfort parameters are calculated at various heights and how the measurement protocols are done.

Some of the important considerations that are taken into account for thermal comfort of an occupant on a general scale are as follows:

1. A healthy building would provide a range of effective temperatures (ET^*) for thermal comfort in the winter time between 20-23.5C and 60%RH and 2C dew point temperature for 80+% of occupants.
2. A healthy building would provide a range of effective temperatures (ET^*) for thermal comfort in the summer time between 23-26C and 60%RH and 2C dew point temperature for 80+% of occupants.
3. Vertical air temperature gradients should not be more that 3 C/m within the occupied zone for thermal comfort i.e. between 0.1-1.1m heights above the floor.
4. A healthy building should offer an environment where the percent feeling draft (PD) should be less than 15%.
5. For comfortable thermal conditions to exist floor temperatures should be between 18 and 29C.

6. A comfortable indoor air environment is achieved by providing at least 20CFM (cubic feet per minute) of outside air per person.

7. A comfortable indoor air environment is achieved by ensuring a CO_2 concentration inside the occupied space to be at most 700ppm (parts per million) over the outside air concentration provided the outside air environment not to exceed 500ppm.

Guideline for ASHRAE's Hypothesis

This section will deal with the hypotheses that have been posed and that will be tested and probed through the various online questionnaires and the engineering data which will be obtained in the field and is discussed later in this quarterly. It is the intent here that each hypothesis will be probed by one or more of the questions which will be highlighted in this discussion. Justifications to the hypotheses will be provided as well to provide a reasonable grounding to their basis. In some cases references will be provided to support the hypothesis and in others a collective sense of experience is used to come up with it.

The intent of the building selection criteria is to screen out "problem buildings" and hence the expectation is that most of the buildings that will be used for the study will be reasonably healthy. Percentages have been presented in some of the hypotheses statements that have been provided from the already available standards i.e. ASHRAE 55-2004 or ASHRAE 62-2001. It is expected that due to that expectation that the response of the dissatisfied respondents of the online questionnaires to show wider variations than what's mentioned in the standards. If the previous field studies are to be used as a metric, suggested ranges for thermal comfort provided by the standard 55 have not captured the expected percentages of satisfied/dissatisfied people in those studies (G.E. Schiller, 1988).

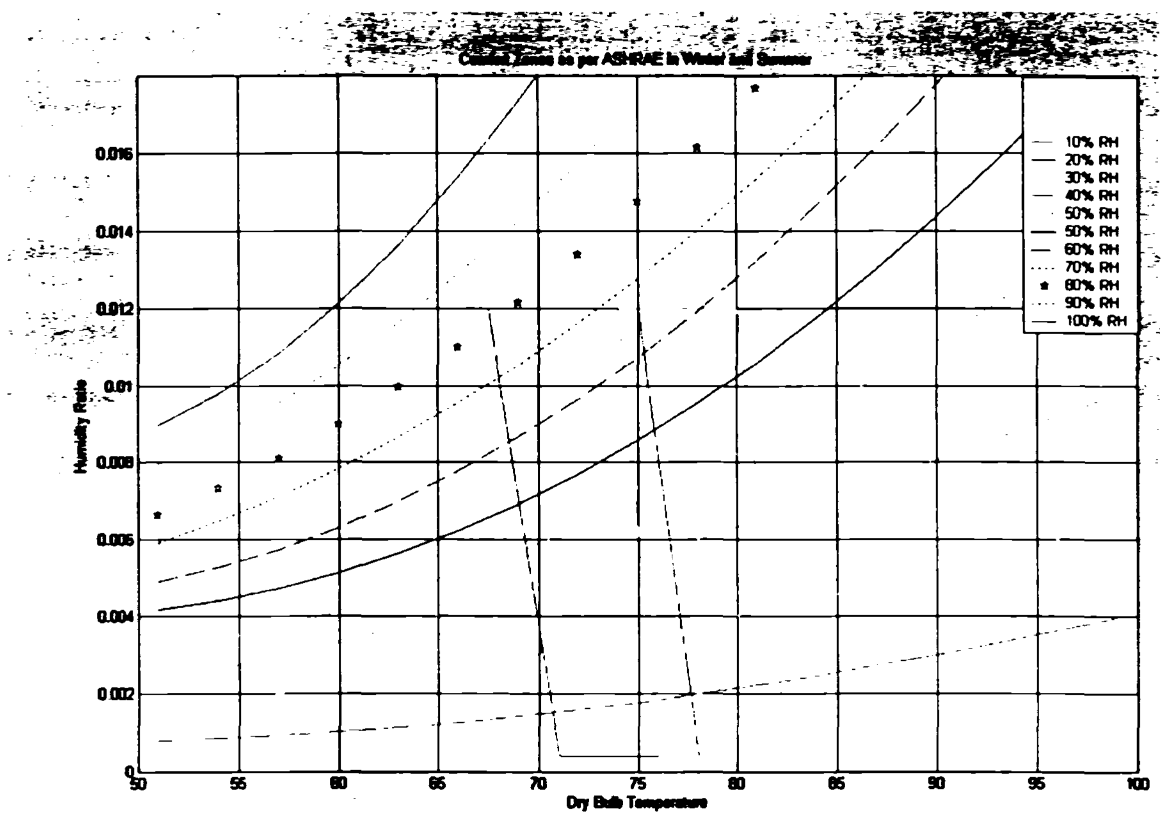

Figure 1.1 Thermal comfort zones for summer and winter according to ASHRAE 55-2004

Figure 1.1 shows a graphical representation on a psychometric chart which shows the range of high and low operative temperature limits for both summer and winter and the upper and lower limit bounds on relative humidity (RH). The summer and winter comfort zones which have been calculated by ASHRAE 55 2004 have been worked on in MATLAB and the comfort diagram is shown below. This is a representation of the thermal comfort in the summer and winter seasons.

The above graph has been generated using MATLAB. A second degree polynomial has been used to generate the curves of humidity. The two parallelograms represent the summer and winter comfort zone as specified by ASHRAE 55-2004.

Hypotheses and Their Justification

1. There is a significant statistical difference in thermal human perception for the occupants between the morning and afternoon (90% according to ASHRAE 55 1992).

Due to increased ambient temperature between afternoon and morning and sometimes expected increase in occupants as well, it is expected there maybe some differences between indoor ambient conditions in a building. A reference was found which mentioned from an international perspective the potential for the existence of that difference in thermal conditions (Ksenija Zaninović http://www.mif.uni-freiburg.de/isb/ws/papers/18_zaninovic.pdf)

2. 90% of the occupants will be comfortable thermally if the ET* values lie within the specified values of ASHRAE 55 1992 i.e. 23.0 C to 26.0 C in summer and 20.0 C to 23.5 C in winter (questions 1 and 2 for different seasons)

According to ASHRAE 55-92, 90% of occupants of a building would be thermally comfortable if the ET* lie between 20.0 and 23.5 in winter and between 23.0 and 26.0 in the summer.

3. 10% of the people take action when they are in a thermally unacceptable work space.

According to a paper from ASHRAE occupants would find a difference in either increasing or decreasing the temperature or humidity which would make their work area acceptable. R. Martin et al. (2002).

4. If the occupant's work area is unacceptable, 50% claim it affects their ability to work better. According to ACC (Accident Compensation Corporation) a government based organization, the thermal surroundings either enhances or hampers our ability to work.

(http://www.acc.org.nz/injury-prevention/safe-at-work/worksafe/action/hazard-management/environment/thermal-comfort/)

5. The occupants of a building would be thermally comfortable if the vertical temperature gradient is less than or equal to 3.0 C. According to ASHRAE 55 92 a source of thermal discomfort can be the vertical temperature gradient between the head and feet.

6. There is a significant statistical difference in the human perception of relative humidity and occupant comfort between the morning and afternoon. A reasonable expectation for purposes of controlling building operations cost is that HVAC units will be turned off during periods of no occupancy. Buildings located in areas classified as climatically "humid areas" may endure humidity accumulation during the night time periods of non-operation of the HVAC system. The operational time needed for dehumidifying the building's cubic air mass during early morning HVAC start up may concurrently be expressed as a period of reduced occupant comfort. This is based on collective technical experience of the group.

7. 90% or more of the work place occupants will be comfortable if the humidity value lies within ASHRAE Standard 55, i.e. RH (Relative Humidity) 60% on the high side and an absolute humidity of 0.00436 kg of water vapor/kg of dry air) on the low side. Based on ASHRAE Standard 55, 90% or more of workplace occupants polled are expected to report that they are comfortable while working in a light-activity work mode if the relative humidity is controlled between 60% for a high value and 0.00436 kg of water/kg of dry air)

8. Ten percent of the occupants take action when they are in a muggy or dry environment. According to other online surveys used by previous ASHRAE research projects several

questions were posed to the occupants about how they would change their immediate thermal/humidity environment.

9. If work area is too muggy/dry, 10% or more of occupants will have difficult time concentrating on their works and affects the productivity. Work place occupants reporting that they are physically uncomfortable may suffer reduced work productivity in accordance with research project by ASHRAE RP 921, K.Cena and R.De Dear, 1998.

10. On the average, the risk of draft should be less than 15% at every point in the occupied zone. If the percentage feeling draft is lower than 15%, 80% or more of the occupants should feel comfortable in their work areas. The risk of draft, PD (equal to the percentage feeling draft), should be less than 15% at every point in the occupied zone. Air movement can improve thermal comfort in warm conditions, but it can also cause uncomfortable drafts feelings. The results show that the higher turbulent intensity can improve human thermal sensations and reduce the risk of feeling a draft in a warm isothermal environment. The results showed that the turbulence intensity had a significant impact on the sensation of draft. (X.Yizai et al 2000 Fanger. P.O et al 1989).

11. If the occupants feel draft in their work place, no more than 15% of entire occupants will feel uncomfortable and thus will take action to reduce the draft, otherwise, they will likely work less efficiently. 15% of the occupants will take action when they are uncomfortable due to drafts. (Fanger. P.O et al 1989).

12. 80% or more of the occupants in the work area shall feel not stuffy if the indoor to outdoor differential concentration of CO_2 is not greater than about 700 ppm. The specified ventilation rates and occupant densities for specified spaces listed in ASHRAE Standard 62 were selected to reflect the consensus that the provision of acceptable

outdoor air at these rates would achieve an acceptable level of indoor air quality by reasonably diluting human bio-effluents, particulate matter, odors and other contaminants common to those spaces. Most ventilation standards recommend that the Carbon Dioxide concentration should not exceed 500 ppm (difference). However, more recent studies have indicated that this limit is far too high for human comfort. At this concentration, the occupants can experience headaches and lethargy and 1000 ppm (inside) is now widely accepted as a limit for comfort. Ventilation is important because it affects indoor environmental conditions, including air pollutant concentrations that may modify the health of the occupants of a building, or their perceptions and comfort.(A.Hazim 1998, L.Zhongping et al 2003).

13. If the occupants feel the air in their work place is stuffy or stagnant, at most 20% of entire occupants will feel uncomfortable and thus will take action to reduce the feeling of stuffy or stagnant, otherwise, they will likely work less efficiently. Those work place occupants (20%) who are uncomfortable (stuffy/ stagnant) will be motivated by their personal discomfort to take action. Human occupants produce carbon dioxide, water vapor, and contaminants including particulate matter, biological aerosols, and volatile organic compounds. Where only dilution ventilation is used to control indoor air quality, an indoor to outdoor differential concentration not greater than about 700 ppm of Carbon Dioxide indicates comfort (odor) criteria related to human bioeffluents are likely to be satisfied. A key element in building design is ventilation. Ventilation is necessary for supporting life by maintaining acceptable levels of oxygen in the air, to prevent carbon dioxide from rising to unacceptably high concentrations and to remove odor, moisture and pollution produced internally.

14. If the occupants smell unpleasant odors in their work place, at most 20% of entire occupants will feel uncomfortable and thus will take action to reduce the concentration of odor, otherwise, they will likely work less efficiently. Those work place occupants (20%) who are uncomfortable (odors in work space) will take action.

15. Eighty percent of the occupants will show job satisfaction when the indoor air quality is acceptable. According to ASHRAE 62 with air adequately free of annoying contaminants 80% of people will be comfortable in the work area.

CHAPTER 2

INSTRUMENTATION

The section will describe in some detail the overall sensors and data acquisition system. The overall schematic arrangement of the Vivo cart is shown below.

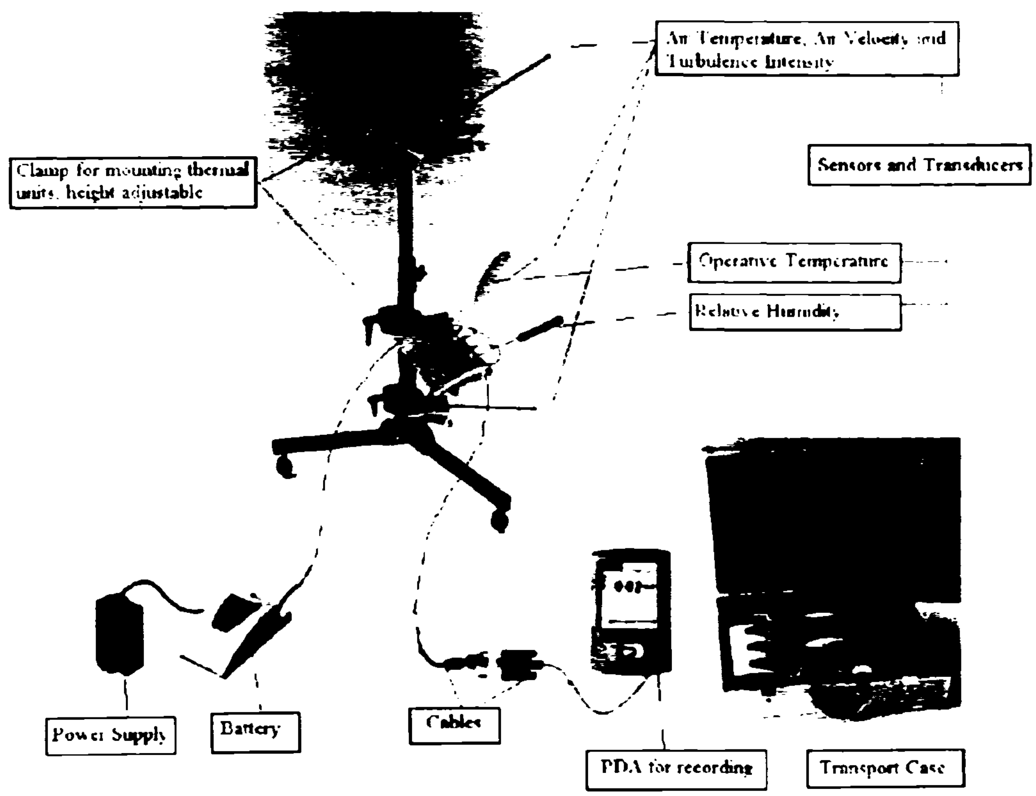

Figure 2.1 Schematic diagram of the VIVO sampling cart and related hardware

Five Vivo sensors will be installed on each of the carts that will be used in this study for indoor environmental quality. Six of the carts will have sensors at three heights 0.1, 0.6 and 1.1m. These carts will have their own battery powered supply for data acquisition and average and store data on the average every 3 minutes. A PDA as shown above can be used to examine the data being stored at various intervals in time to make sure of its reliability. At the end of each sampling day the data from all the sensors will be downloaded into a laptop for storage and for further processing. A diagram and a short description of each of the pieces of sensor hardware is described and shown below.

Figure 2.2 Vivo draught sensor (draft sensor)

The unit equipped with a tilt able arm for easy positioning of the transducer at pre-defined heights, measures low air velocity and air temperature, which are the necessary input parameters for calculating Draught Rate (DR). The low air velocity transducer has spherical thin-film sensors, which are omni directional, very accurate at low air velocities and so fast that they can measure air velocity fluctuations. The measurement range is 0.05 – 5 m/s and fluctuations in air velocity are measured up to 2 Hz.

Figure 2.3. Vivo temperature sensor

The unit measures the operative temperature at a single point. The transducer is an ellipsoidal device and the size is adjusted so that it integrates air and radiation temperature in the same way as a human being.

Figure 2.4. Vivo relative humidity sensor

The unit measures the humidity of the air. The moisture sensor is a capacitive type, which measures air humidity directly in per cent Relative Humidity (RH). The unit is also equipped with a sensor for measuring air temperature and a tilt able arm for positioning the transducer in pre-defined angels. The measurement range is 0-100% RH. The transducer is positioned vertically, horizontally, or at an angle of 30° from vertical to measure operative temperature for a standing, recumbent or sitting person respectively.

The operative temperature is what we refer to in everyday speech as room temperature. The unit is equipped with a tilt able arm for positioning the transducer at pre-defined angles. The settling time of the transducer is less than 5 minutes.

Palm hand terminal used for controlling the units and displaying the results. The Palm software can calculate all relevant index and key values. It is also possible to print out key values or transfer them to a PC. A decision on where the six stationary carts should be placed will be made on site and in conjunction with the other team partners i.e. sound and mold groups. This will ensure that the group has data in reasonably close proximity of each other to comply with the spirit of the aim of this thesis.

Figure 2.5. Hand held PDA for random checking

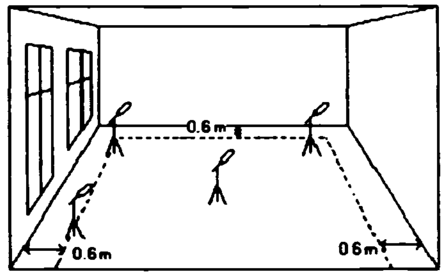

Figure 2.6 Sampling perimeter limits

The table below shows a summary of the accuracy of the sensor equipment versus what is suggested by ASHRAE.

Sensor Type	Units of measurement	Range of measurements	ASHRAE accuracy	Manufacturer accuracy
Air temperature	°C	0-45C	±0.2 °C	±0.5 °C
Air velocity	m/s	0.05-1 m/s	±0.05 m/s	±0.01 m/s
Operative Temperature Mean radiant temperature	°C	0-45C	±0.2 °C	±0.5 °C
Relative Humidity	%	0-100	-	-
Illuminance	Lux	0.01 to 299900	-	2%

Table 2.1 Accuracy of Vivo Sensors and ASHRAE's 55-92 Accuracy

It is to be mentioned here that due to the number and cost of the sensor equipment a balance had to be made between accuracy and cost of the sensors to achieve obtaining the require data. The number of raw experimental data and the calculated indices from the Vivo software at each averaged point in time are summarized in Table 2.1. Several of the indices mentioned in that table are indices already accepted in the research community and mentioned in ASHRAE standard 55-92. The measures are provided by Fanger 1970. The data will be obtained using the following protocol:

- The sensors will be checked before any data is taken by the use of the PDA. This will ensure that all the sensors are performing adequately.
- The stationary carts locations will be surveyed and decided on before occupants come to the building if possible.

Assembly Procedure for the Sensors

1. Press the two units together so the male and female connectors meet.
2. Turn the mechanical wheel until the two units are locked securely together.
3. Connect the units with the network cable. Note: The arrow on the cable must line up with the arrow on the connector.
4. When the cable has been plugged in, turn the cable's metal ring clockwise until the cable is locked in the connector.

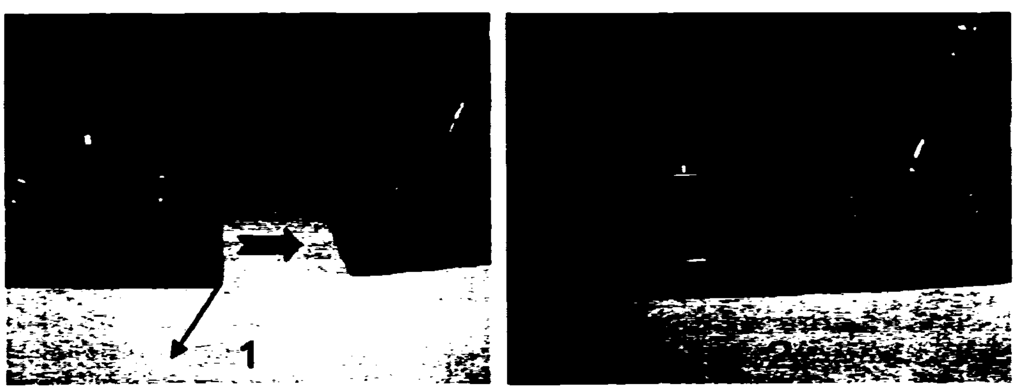

Figure 2.7 Connection assembly sequence

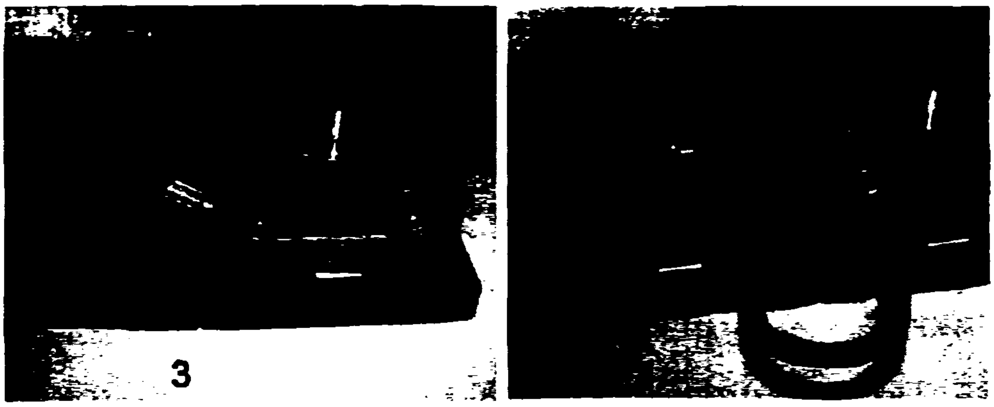

Figure 2.8 Connecting the pins of the sensors

Positioning the Mechanical Arm

None of the sensors should be touched as that will influence the precision. The mechanical arm can be adjusted by moving it up or down. The arm can be positioned from 0° to 90°, with fixed notches at 0°, 30° and 90°. Change the positions by using the arm alone (Figure 2.8), but the sensors should not be touched.

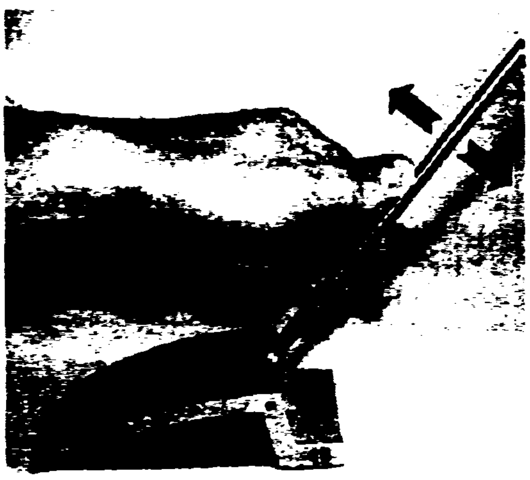

Figure 2.9 Positioning of the mechanical arm

The VIVO Draught instrument is different from the thermal sensors as it has a protective cover (cap) that has to be removed before the usage of the sensor. Lower the protective cap by turning the screw counter-clockwise to release it. When the cap has been lowered, tighten the screw again.

VIVO Stand

VIVO units are mounted on a stand using a clamp (Figure 2.10) that can grip rounded items with a diameter of 13 to 55 mm.). The units can be mounted on the clamp using the mounting bolt and a male connector (Figure 2.11).

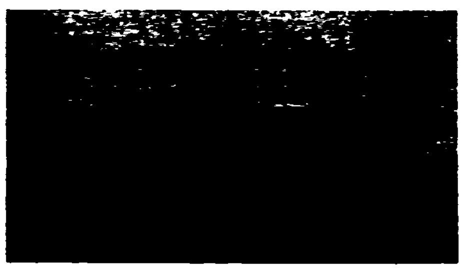

Figure 2.10 Clamping unit for VIVO stand

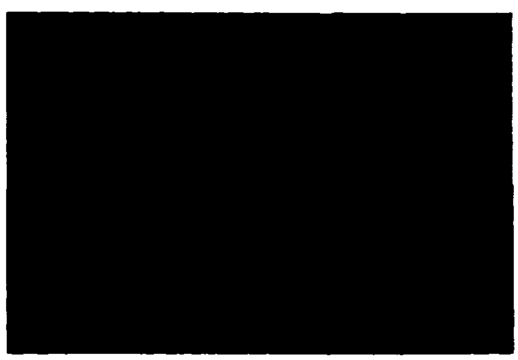

Figure 2.11 Mounting holes

Using different mounting holes makes it possible to mount the units at different angles. There are 2 places where the mounting bolt can be secured and 6 locations for the male connector (Figure 2.12).

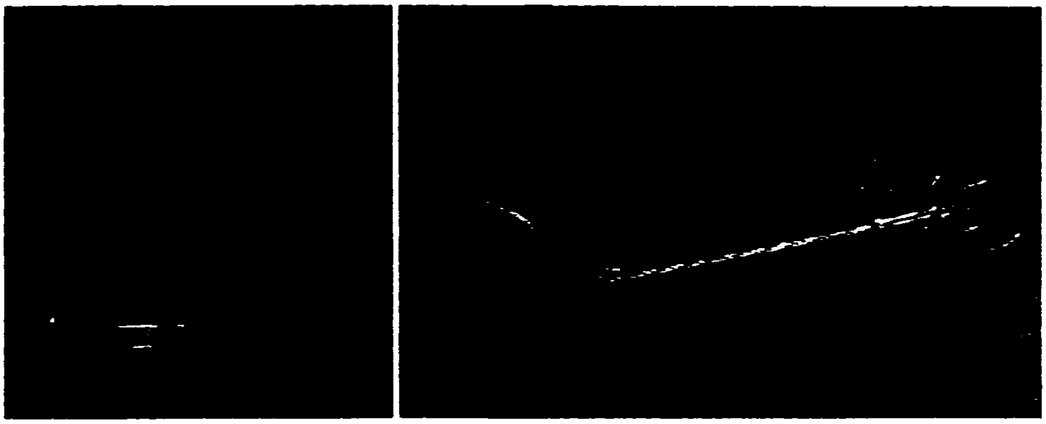

Figure 2.12 Location of mounting bolts

Power Supply for VIVO Thermal Units

Either a current feed or a VIVO battery can be used as the power supply. Connect the current feed to the 220 V or 110 V electricity mains.

Monitoring Data Using PDA

Before starting a measurement, check that all cables are connected and that the battery is fully charged. To start a measurement:

- Switch the PDA on and press on the Vivo icon.
- Check that it has the desired time set-up and press on the synchronize button

- Synchronization starts automatically and is finished when the program says that it is ready. Synchronization can be stopped by pressing the Cancel icon. If several units are not interconnected, you must synchronize them individually.
- The measurement starts automatically. The measurement finishes with an acoustic signal and "Status" is "Finished" message.
- When the measurement is finished, the units are resynchronized.

A Screen shot of the PDA is illustrated in Figure 2.13.

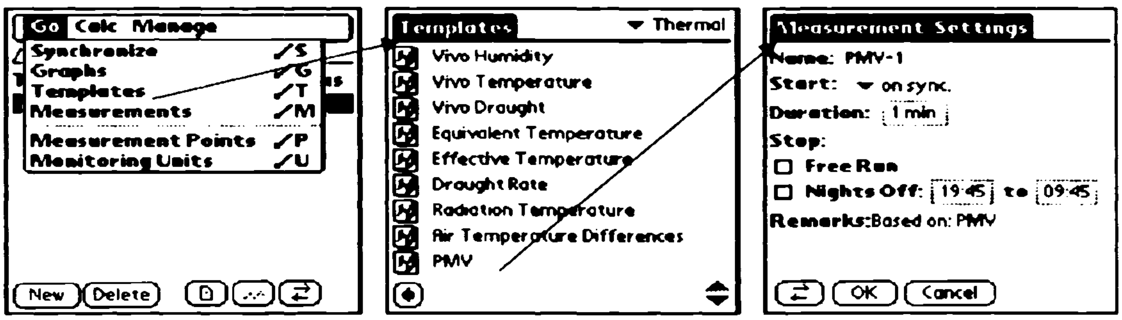

Figure 2.13 Screen shot of the PDA

Communicating with the VIVO Field Control (PDA)

To maintain accuracy, it is mandatory for the user to check that at regular intervals if the output of the sensors is within the range expected. A cable provided which acts as a coupler between the sensors and the PDA displays the results which can be periodically checked using the PDA as an interface (Figure 2.14).

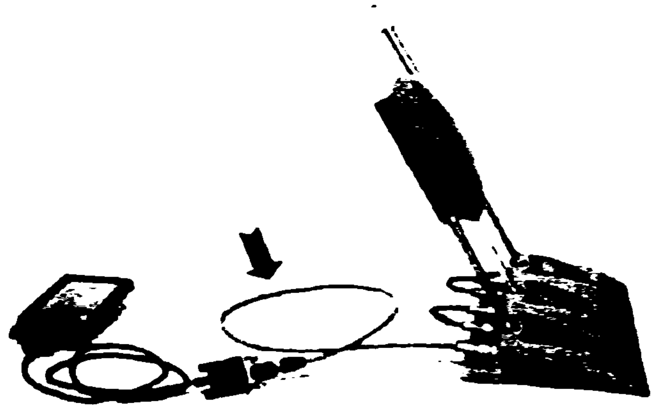

Figure 2.14 Assembly with connection to PDA

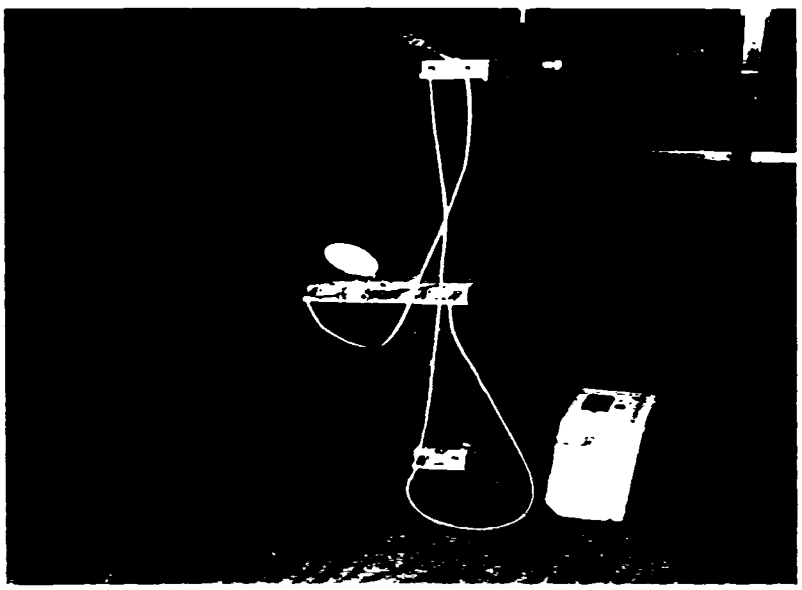

Figure 2.15 Final assembly of the stand with all the sensors

CHAPTER 3

DESCRIPTION OF PHYSICAL MODEL USING CFD SIMULATION

In order to study how temperature and velocity profiles vary through in a 3-D room, an actual room has been tried to be simulated using STAR CD as the simulation tool. The vector profiles of the temperature and velocity are studied under this case. Three cases of simulation for air flow have been attempted to be studied here. They are as follows:

1. Ceiling Diffuser
2. High wall ceiling Diffuser and
3. Under floor air distribution

Initial Parameters

A 3-D room with dimensions 11*8*11 feet is designed in STAR CD. As discussed above three cases will be taken into consideration in which the inlet and outlet would be at different locations as per the type of airflow distribution.

In a ceiling diffuser, both the inlet and the outlet would be on the ceiling of the room. This is the typical case in most of the office buildings. In the second case, which is High wall diffuser the outlet would be on the ceiling and the inlet would be on one of the walls. In the last case, under the floor air distribution, the inlet air would come from under the floor and the outlet would be on the ceiling. All the three

cases have the same area for the inlet and outlet. The initial boundary conditions are as follows:

The case under consideration is a k-e model with the default values of k and e with a high Reynolds number is taken into consideration. The flow is a turbulent one with a Reynolds number of approximately 1.50338* 10^6. The initial temperature of the room is taken at 302.5 Kelvin. The supply air temperature is taken at 287 K. A thermostat is setup on one of the walls which would be monitoring the temperature variations in the room. A fixed amount of volumetric air flow was assumed for the supply air (492 CFM). Also would be present on one of the walls, a constant cooling load representative of the cooling load for that kind of room. Once the temperature in the room goes beyond 300 K, the thermostat starts that means the flow through the inlet is initiated. Similarly if the temperature goes below 297.6 K, the inlet flow shuts off. The inlet and outlet are both 7"*8" for all the cases. The number of cells for each case was approximately 208000 cells. The simulation included a cell in the solution field to act as a thermostat where the supply air was introduced/shut off in the room when the air temperature at that cell reached 26.4Cor 23.6C respectively.

Simulation Steps

The cases under consideration are all transient that means the thermal parameters like temperature change with time. All the three cases have been done with 31000 time steps with an increment of about 0.01 minutes each which accounts for about 310 minutes of test time. Two different cells one at approximately the centre of the room and one at the inlet have been monitored continuously throughout the run to see if the inlet on and off conditions as used in the subroutine were working. The effects of gravitation and

buoyancy have been taken into consideration to simulate conditions similar to a normal 3-D room. The concentration of the contaminant carbon dioxide was initially setup at 1000 ppm (parts per million).Grid Independency was achieved from varying the number of cells in per foot from 4 to 6. The results between the same for 5 cells per foot and 6 cells per foot have been shown in the results. For the simulation to run faster, 5 processors were used instead of 1 which dramatically reduced the time taken from 24 hours to less than 6 hours. The two subroutines postdat.f and bcdefi.f were the control codes for the thermostat. Residual tolerances of the order 0.001 have been used for greater accuracy. The data was regularly stored at each increment of time. Two planes of cut AA and BB have been chosen for better analysis. The plane of cut has been shown in the figures in the next chapter.

CHAPTER 4

RESULTS AND DISCUSSIONS

Presented below are the results and discussions for both the field data and the computational data.

Temperature: Presented below is the figure that represents the summer and winter thermal comfort zones as specified by ASHRAE 55-2004.

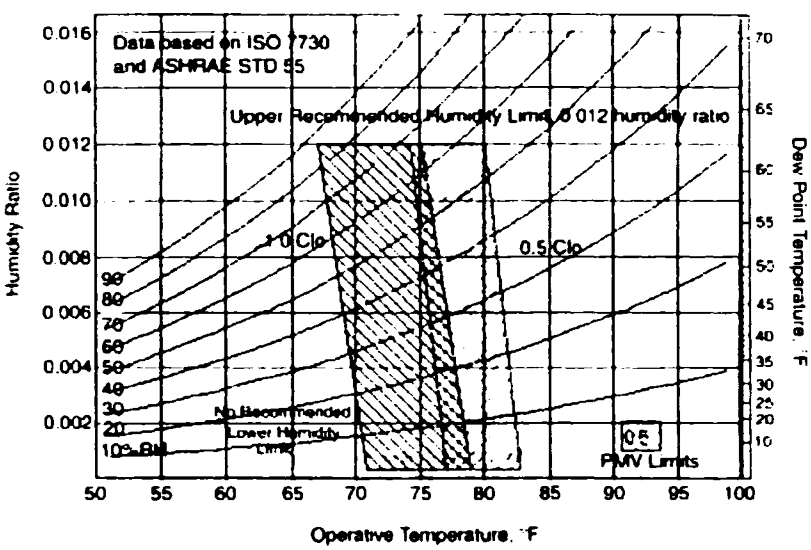

Figure 4.1 Thermal comfort zones in summer and winter

The temperature taken into consideration is the Dry Bulb Temperature (DBT). DBT that is a calculated parameter from the Vivo Sensors was always between the set values. Of the five buildings that have been studied four were in summer and one was in winter. 87% of the data that was analyzed in winter fell in the DBT in the specified range of ASHRAE values. Shown below are the plots for DBT Vs Time for all the five buildings. All the data has been taken from 10:00 AM- 5:00 PM. The data of the three days have been taken as a weighted moving average.

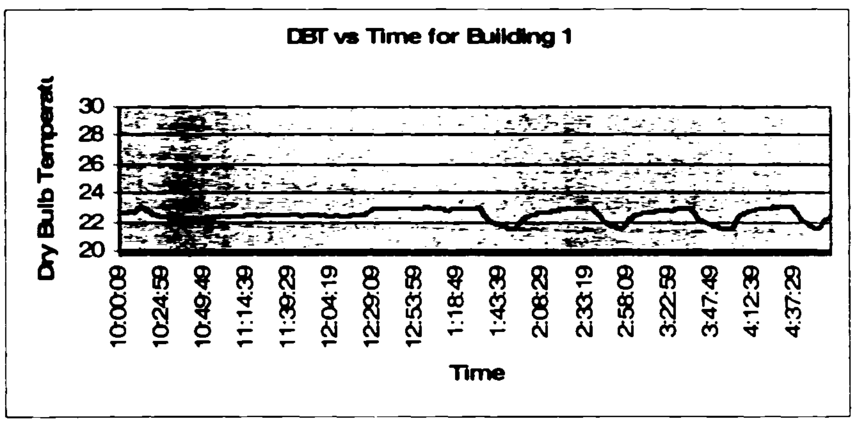

Figure 4.2 DBT vs. time for building 1

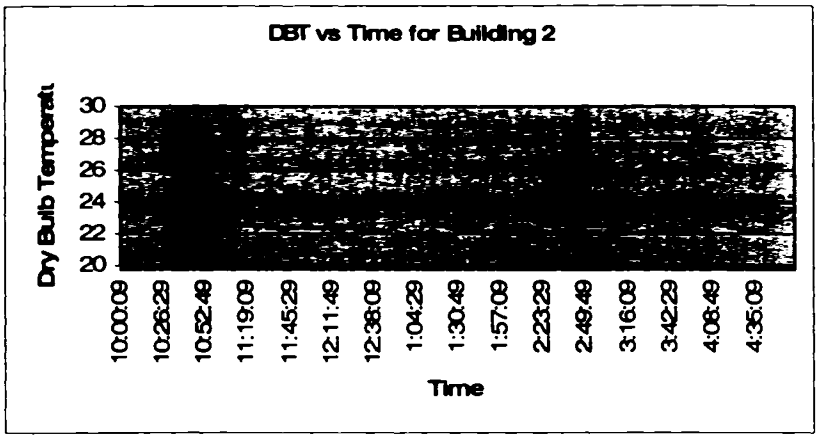

Figure 4.3 DBT vs. time for building 2

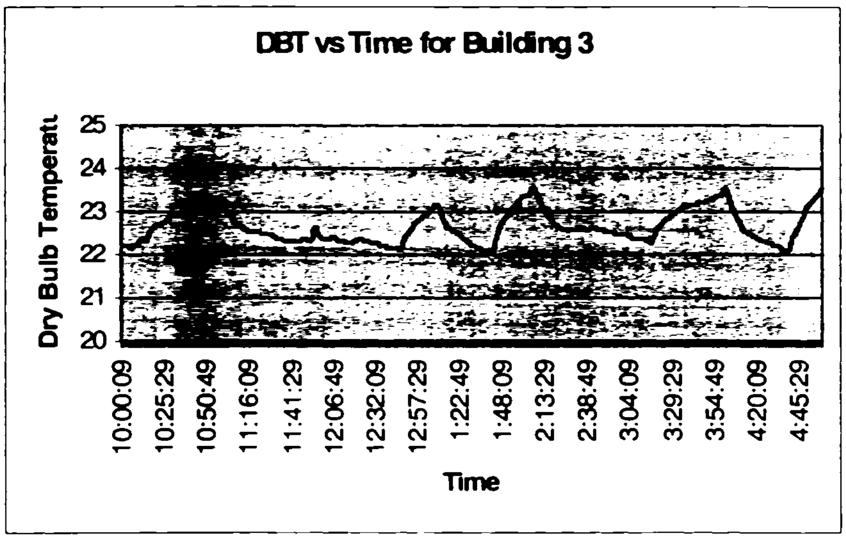

Figure 4.4 DBT Vs time for building 3

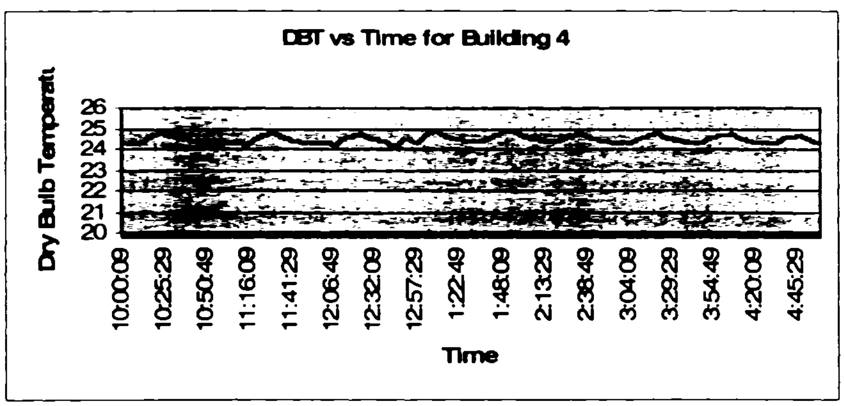

Figure 4.5 DBT Vs time for building 4

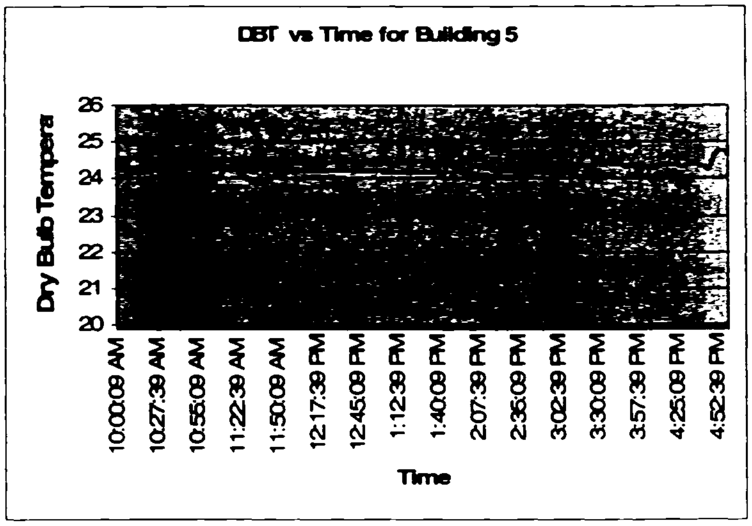

Figure 4.6 DBT Vs time for building 5

One more parameter that is important parameter that plays a pivotal role in thermal comfort is the vertical temperature difference. It is the numerical difference in temperature between the head and ankles of an occupant in a room. This has been calculated in the Vivo sensors as the difference between the 1.1 m and 0.1m. From ASHRAE 55-2004, the acceptable vale of this parameter is 3.0 C or 5.4 F. Presented below are the graphs for both the vertical temperature difference and another parameter that is a characteristic feature in thermal comfort studies.

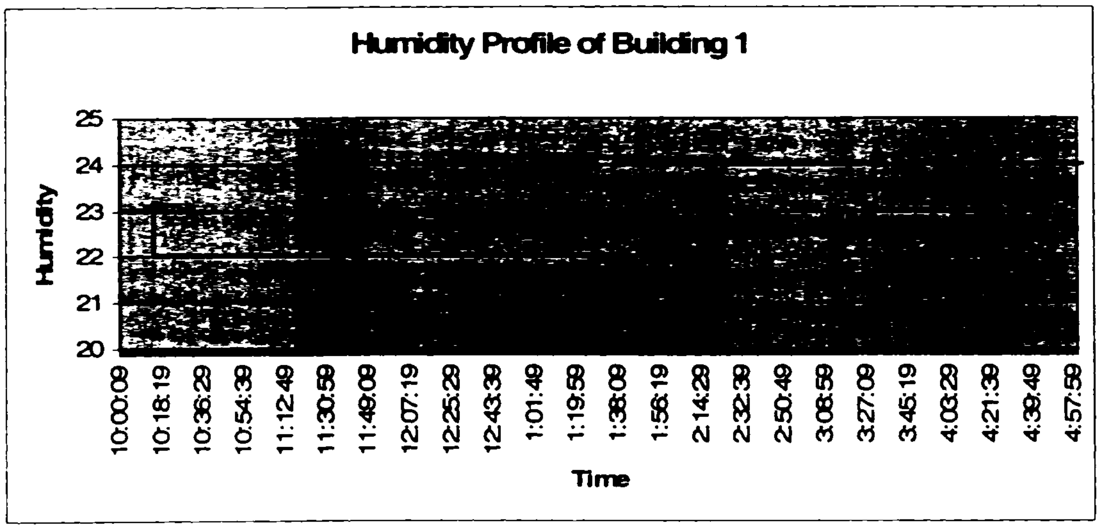

Figure 4.7 Humidity Vs time for building 1

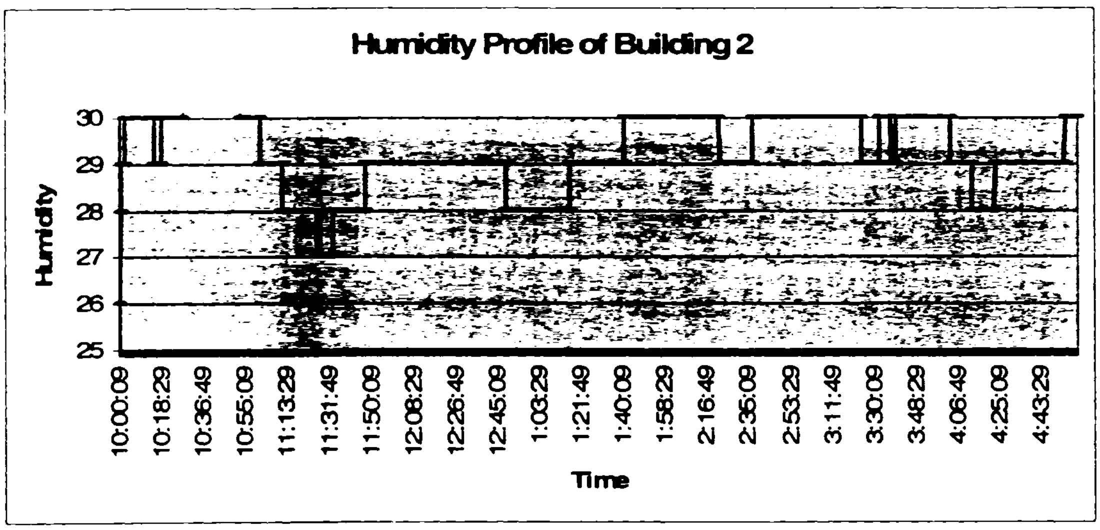

Figure 4.8 Humidity Vs time for building 2

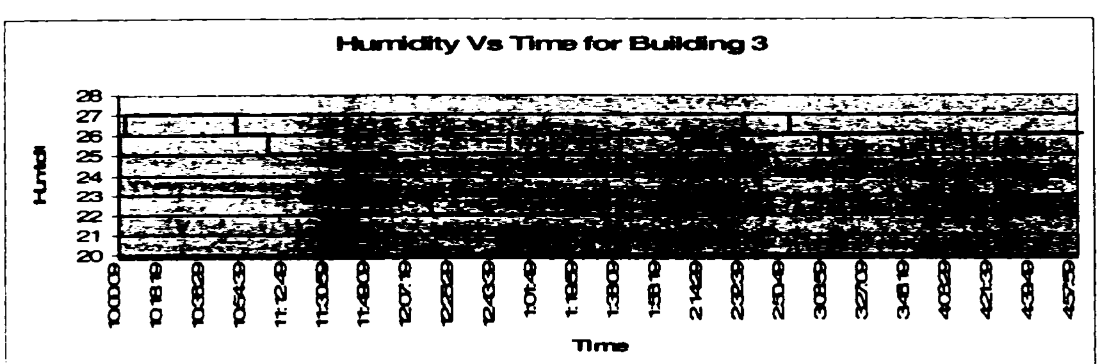

Figure 4.9 Humidity Vs time for building 3

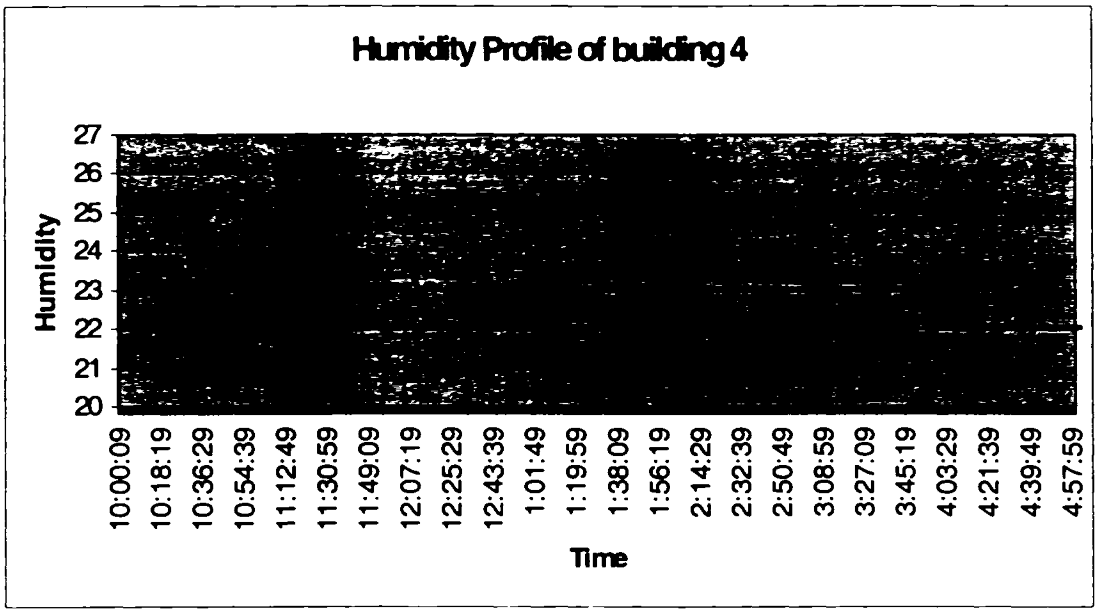

Figure 4.10 Humidity Vs time for building 4

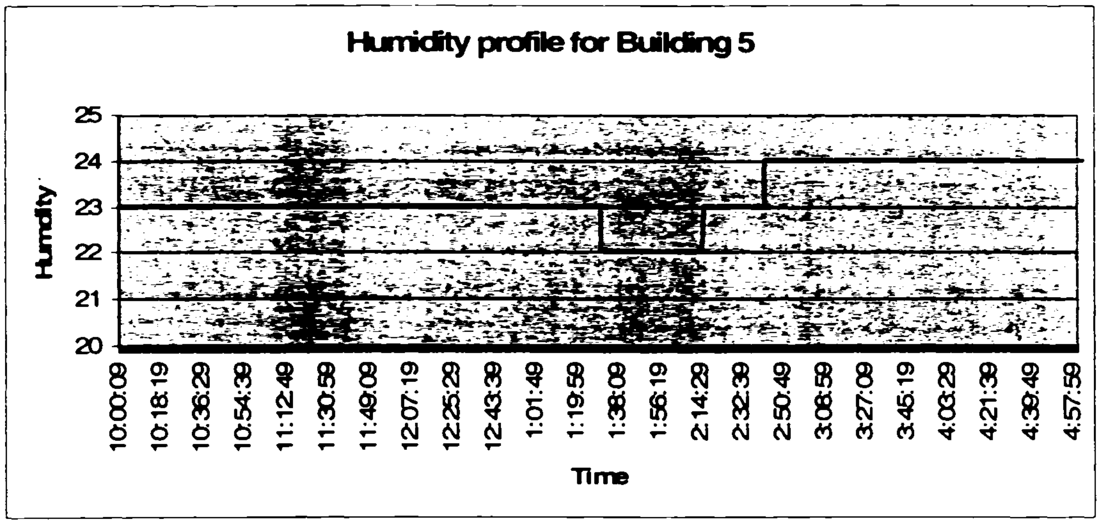

Figure 4.11 Humidity vs. time for building 5

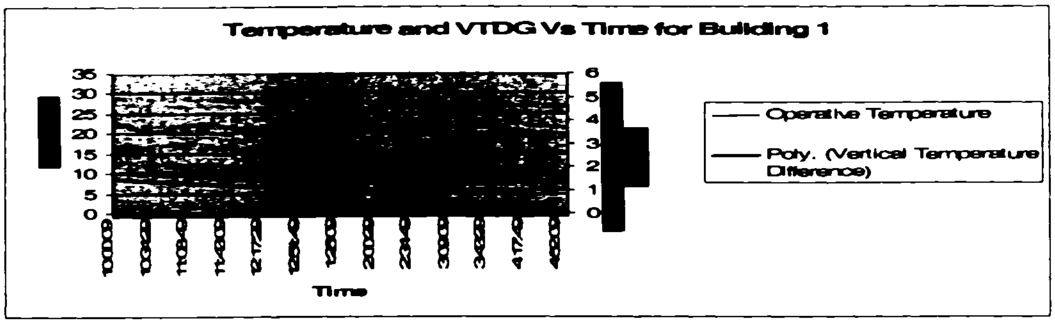

Figure 4.12 Temperature and vertical temperature difference gradient vs time for building 1

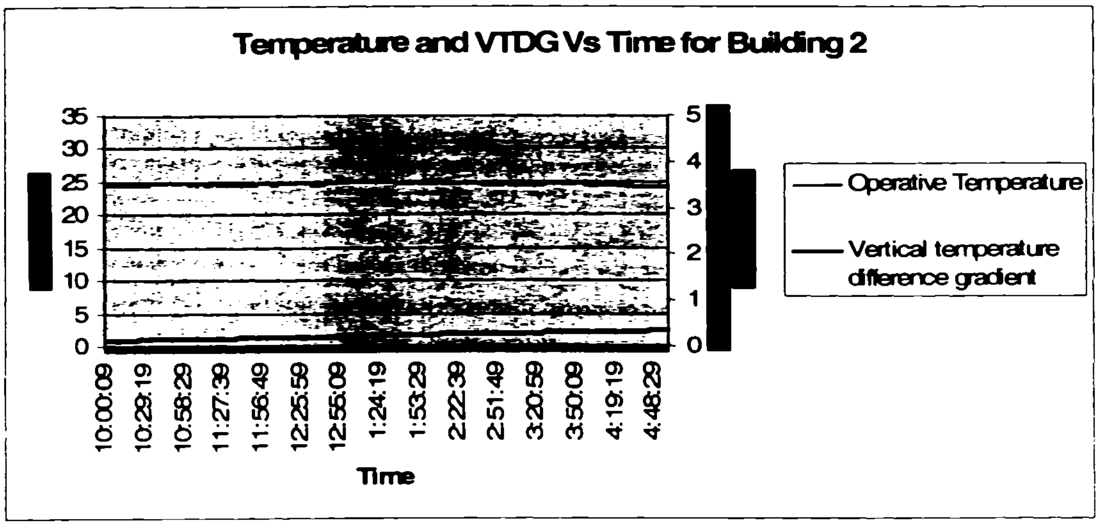

Figure 4.13 Temperature and vertical temperature difference gradient vs time for building 2

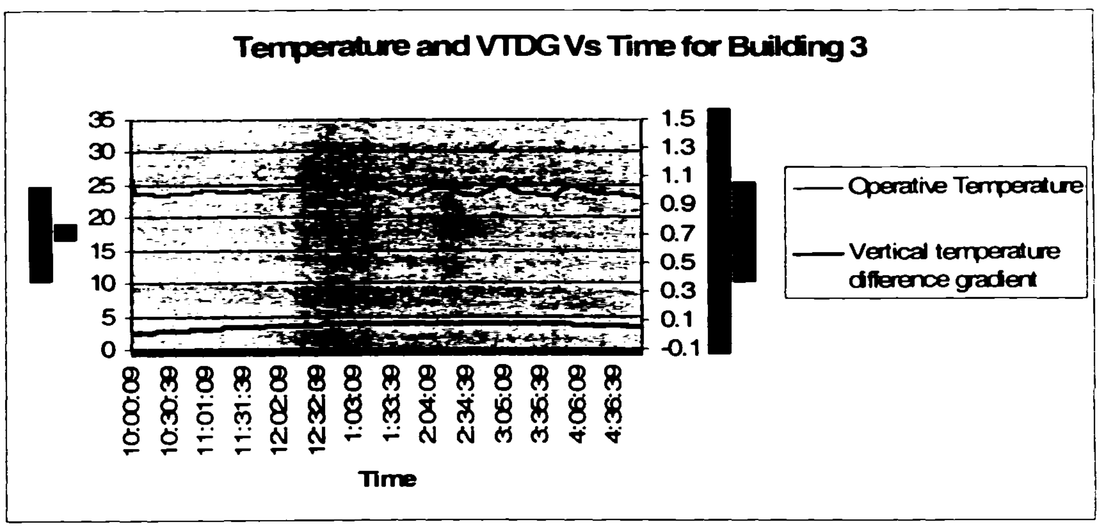

Figure 4.14 Temperature and vertical temperature difference gradient vs time for building 3

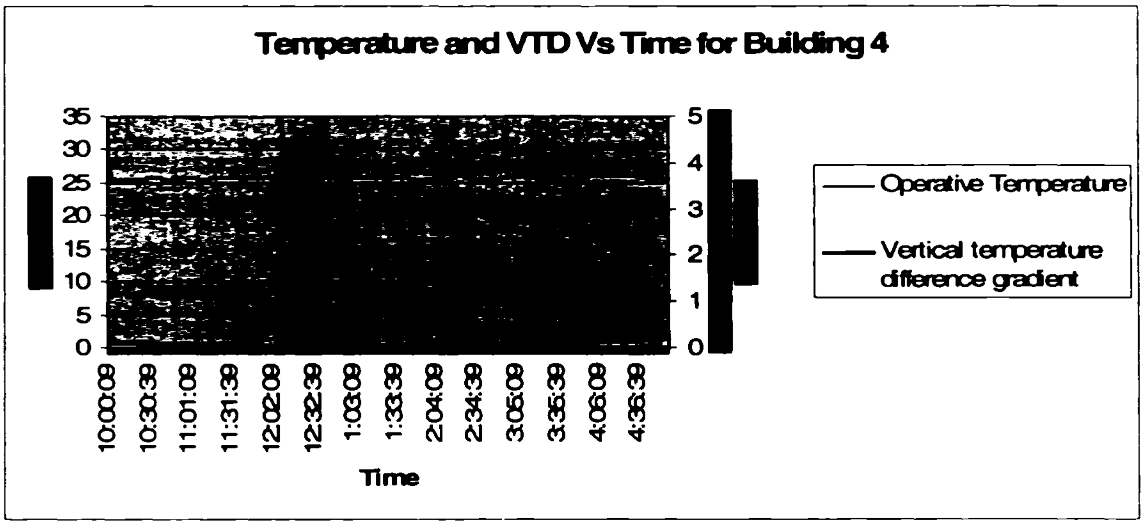

Figure 4.15 Temperature and vertical temperature difference gradient vs time for building 4

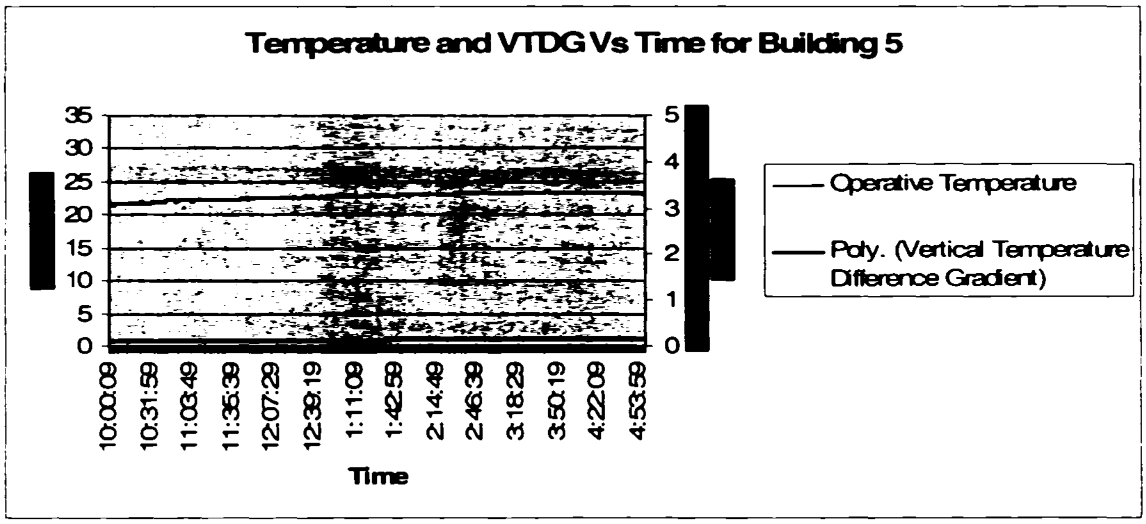

Figure 4.16 Temperature and vertical temperature difference gradient vs time for building 5

As we can see, all the vertical temperature difference falls in the zone which is less than 3.0 C/m as cited in ASHRAE 55-2004. Also specified in the figure shown were the acceptable ranges of Relative Humidity. It was observed that 84% of the data

fell in the region that was designated to be comfortable. All the data of PMV and PPD which are thermal indices fell within the range as suggested by the standard. Also within the range were the velocities that were flowing through the various rooms.

Building Number	Temperature in C	Humidity in % RH	Vertical Temperature Difference Gradient C/m	PMV and PPD
1 Winter	93%	91%	100%	100%
2 Summer	96%	94%	100%	100%
3 Summer	94%	100%	100%	100%
4 Summer	97%	100%	97%	92%
1 Summer	94%	100%	100%	92%

Table 4.1 Percentage of thermal parameters that fell within ASHRAE's comfort zone for all the buildings

Analysis of Building One

The building was visited once in winter and once in summer. Some of the important conclusions that have been observed are as follows. The results typically are based on the hypothesis mentioned in the aforesaid chapter. The number of people who were a part of the survey was twenty one. This happens to be an added advantage owing to the fact that the answers given by people in the questionnaire who were sitting to certain locations were related to the thermal stations that were close to them. The other buildings did not have this benefit because of two reasons namely a) The buildings were huge and b) The number of people who answered the questionnaire were very less. The hypotheses that were used in the previous chapter were tested in these conditions and their conclusions have been drawn. Since the number of people who took the questionnaire was not a major one, firm conclusions about the facts that were made could not be assumed on a general basis. The season in which the building was visited was in winter. Presented below are the hypotheses that were initially made up and the conclusions that were drawn from the same.

1. Hypothesis: Ninety percent or more of the occupants will be comfortable thermally if the Operating Temperature values are within 23.0° C to 26.0° C in summer and 20.0° C to 23.5° C in winter.

Findings: All the data that was obtained regarding temperature fell between 20.0° C to 23.5°C(the season in consideration being winter), which can be said that the occupants were thermally comfortable. The hypothesis holds good.

2. Hypothesis: Ten percent or less of building occupants will make adjustments to make their work area thermally more comfortable.

Findings: Thirty two percent of the building occupants are making adjustments/changes to their work area to make it thermally more comfortable. Occupants either modulate the thermostat or wear lighter or heavier clothing depending on the thermal conditions that prevail around them. The hypothesis that was initially set up does not hold good for this result.

3. Hypothesis: If the occupant's work area is thermally unacceptable, the occupants in that area will likely work less efficiently.

Findings: Forty-Seven percentage of the occupants felt that their efficiency was affected adversely at some workdays if the work area was unacceptable thermally. The hypothesis that was initially set up does not hold good for this result.

4. Hypothesis: Occupants would find a difference in either increasing or decreasing the temperature, which would make their work area acceptable.

Findings: The hypothesis does not hold good.

5. Hypothesis: Twenty percent or less of occupants will make adjustments when they are in a dry environment.

Findings: Sixteen percent of the occupants made adjustments when they were in a dry environment. The hypothesis that was initially set up holds good for this result.

6. Hypothesis: Twenty percent or less of occupants will make adjustments when they are in a muggy environment.

Findings: Eleven percent of the occupants made adjustments when they were in a muggy environment. The hypothesis that was initially set up holds good for this result.

7. Hypothesis: On the average, less than 15% of building occupants should feel a draft at every point in the occupied zone. If the percentage of occupants feeling draft is lower than 15%, 80% or more of the occupants should feel comfortable in their work areas.

Findings: Fourteen percent of the building occupants felt the presence of draft which is lesser than the normal fifteen percent. So 80% or more of the occupants should feel comfortable in their work areas. The hypothesis that was initially set up holds good for this result.

8. Hypothesis: There is a significant statistical difference in thermal perception for the occupants between the morning and afternoon.

Findings: The variation in temperature from morning to afternoon in the building was less than 1.5 ^{0}C. Also the occupants were not affected with the change in temperature. There was no significant statistical difference in thermal perception for the occupants between the morning and afternoon. The hypothesis does not hold good.

9. Hypothesis: The occupants of a building would be thermally comfortable if the vertical temperature gradient is less than or equal to 3.0° C

Findings: The vertical temperature gradient was not greater than 3.0° C at any point of time. The occupants of the building on a whole were thermally comfortable. The hypothesis holds good.

10. Hypothesis: If the occupant's work area is thermally unacceptable, the occupants in that area will likely work less efficiently.

Findings: About 5 % of the occupants (1 out of 21) felt that their productivity was affected when the work area around them was thermally unacceptable. Not many occupants in the building felt that the area around them was thermally unacceptable.

11. Hypothesis: There is a significant statistical difference in relative humidity and occupant comfort perception between the morning and afternoon.

Findings: The relative humidity remained almost at a constant level all through the day. There was not a significant change in the morning and afternoon. The hypothesis does not hold good.

12. Hypothesis: Twenty percent or less of the occupants make adjustments when they are in a muggy environment.

Findings: About 19% of the occupants (4 out of 21) made adjustments when they felt that they were in a muggy environment. The hypothesis holds good.

13. Hypothesis: Occupants will make adjustments to reduce the presence of a draft in their work area.

Findings: About 14 % (3 out of 21) of the occupants made an adjustment to reduce the presence of draft in their work area.

14. Hypothesis: If occupants smell unpleasant odors in their work area, at most 20% of them will be uncomfortable and make adjustments to reduce the concentration of odor. Findings: About 5 % of the occupants (I out of 21) made adjustments to reduce the concentration of odor when they were in an uncomfortable situation as a result of odors. The hypothesis holds good.

Computational Results

Grid Independency

The definition of the problem has already been defined in the previous chapter. Grid Independency is the process of determining if or not the size of the grid has an influence on the results that occur in the room. In other words it is the process of trying to achieve stability criterion for parameters like temperature and velocity which are being monitored here. The grid independency was initially tested with 4 cells per foot then increased to 5 and finally stability was achieved at 6 cells per foot. The temperature and Velocity plot for cells with 5 and 6 cells have been shown below for all the cases.

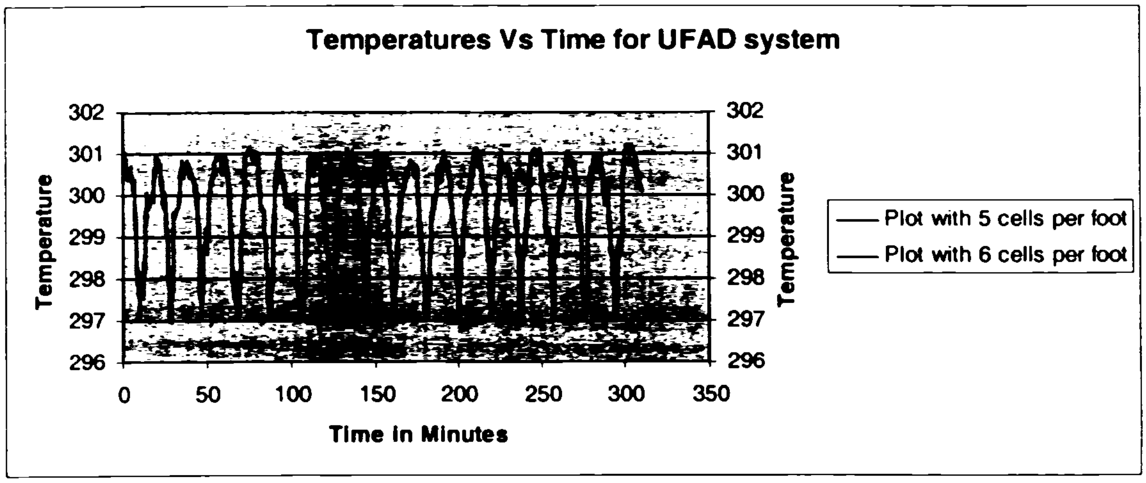

Figure 4.16 Grid independency for UFAD system of five vs. Six cells per foot

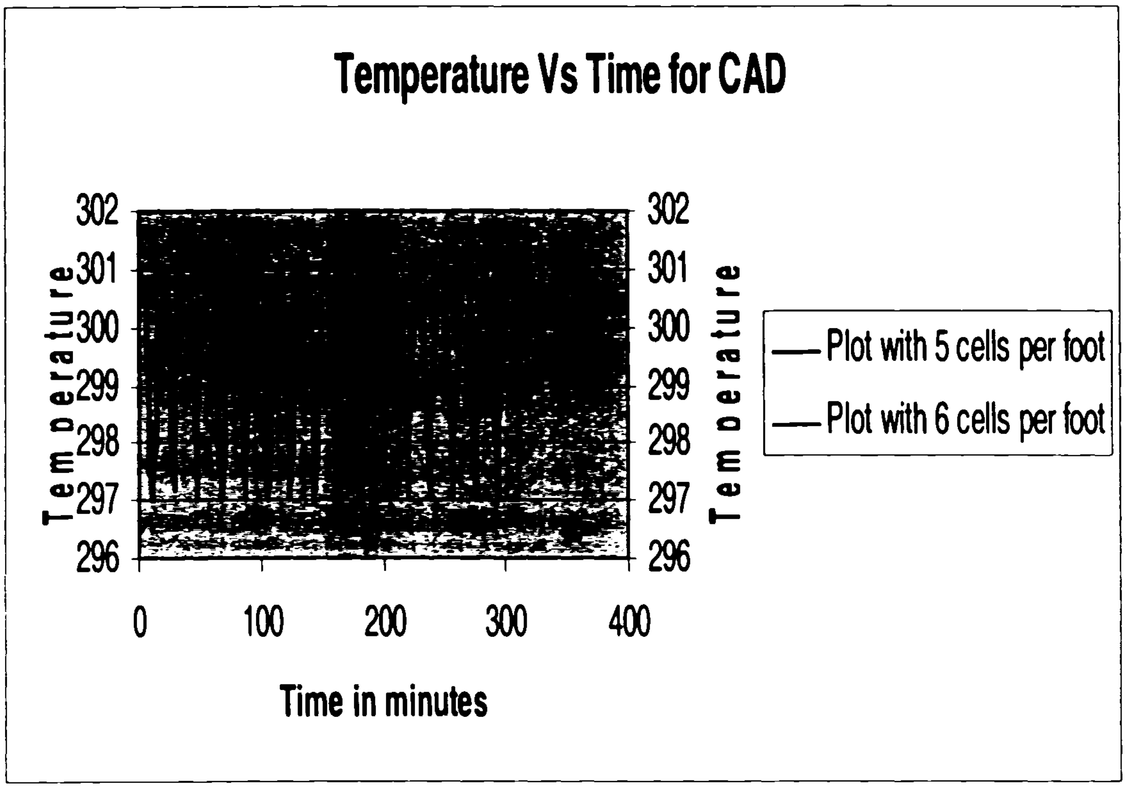

Figure 4.17 Grid independency For CAD System for five Vs six cells per foot

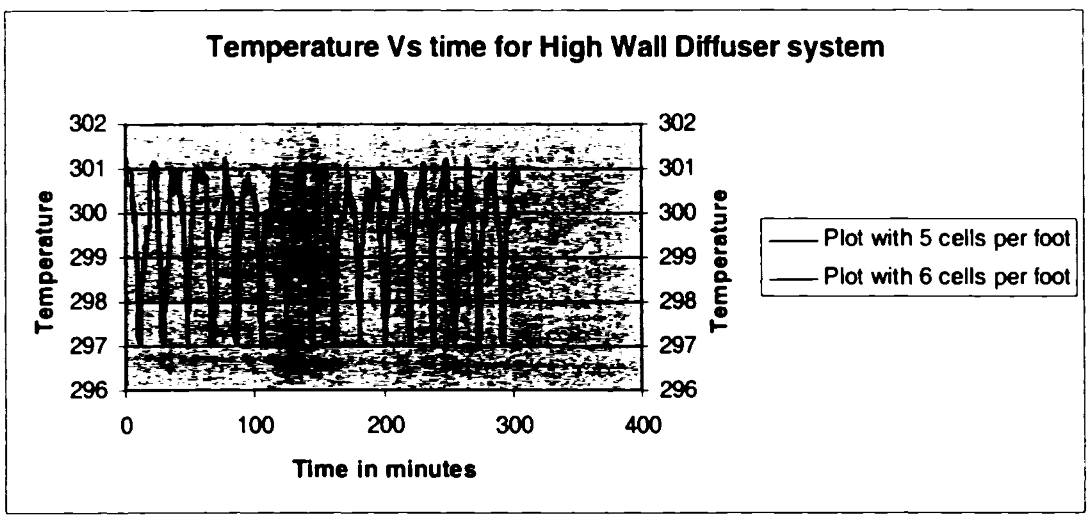

Figure 4.18 Grid independency for high wall system for five vs. six cells per foot

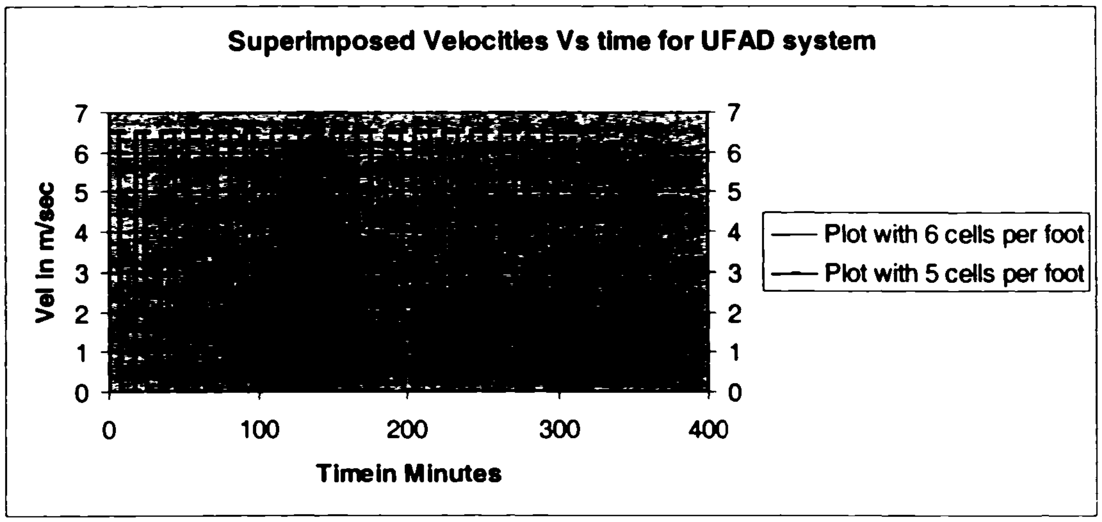

Figure 4.19 Velocity plot for 5 and 6 cells per plot for UFAD system

Case when the inlet is ooen for the maximum time at 157 minute

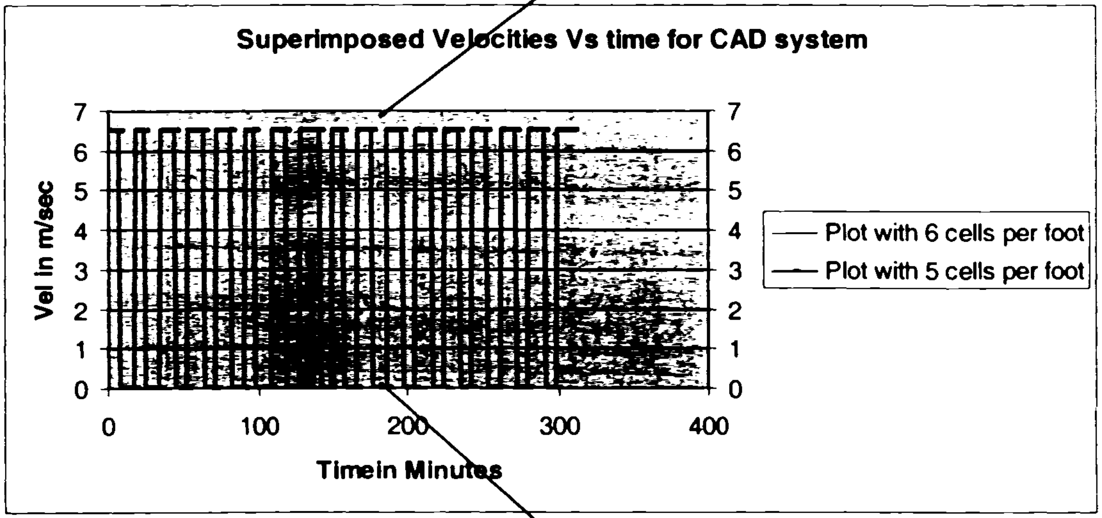

Figure 4.20 Velocity plot for 5 Vs 6 cells per foot for CAD system

Case when inlet is closed for maximum time at 171 minute

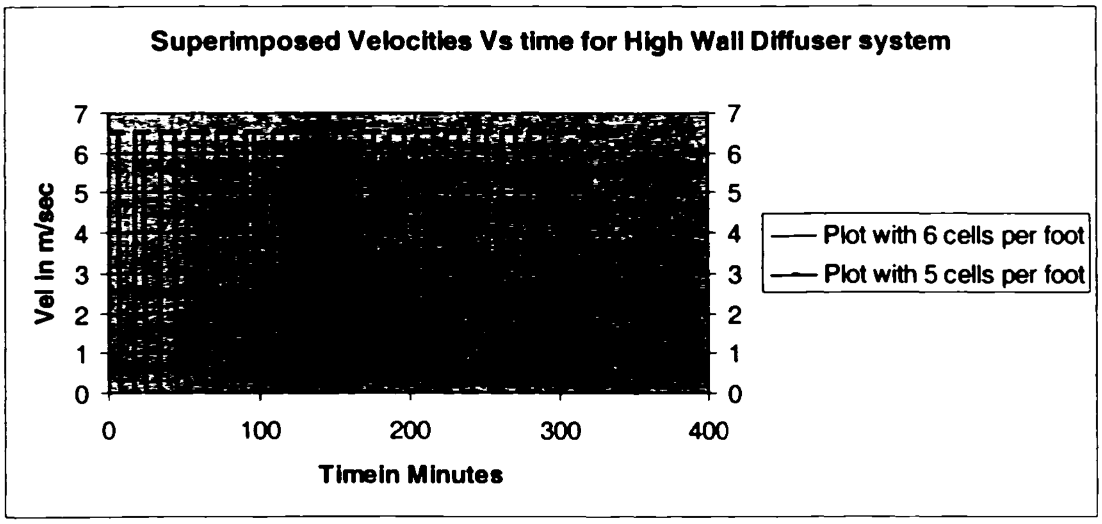

Figure 4.21 Velocity plot for 5 Vs 6 cells per foot for CAD system

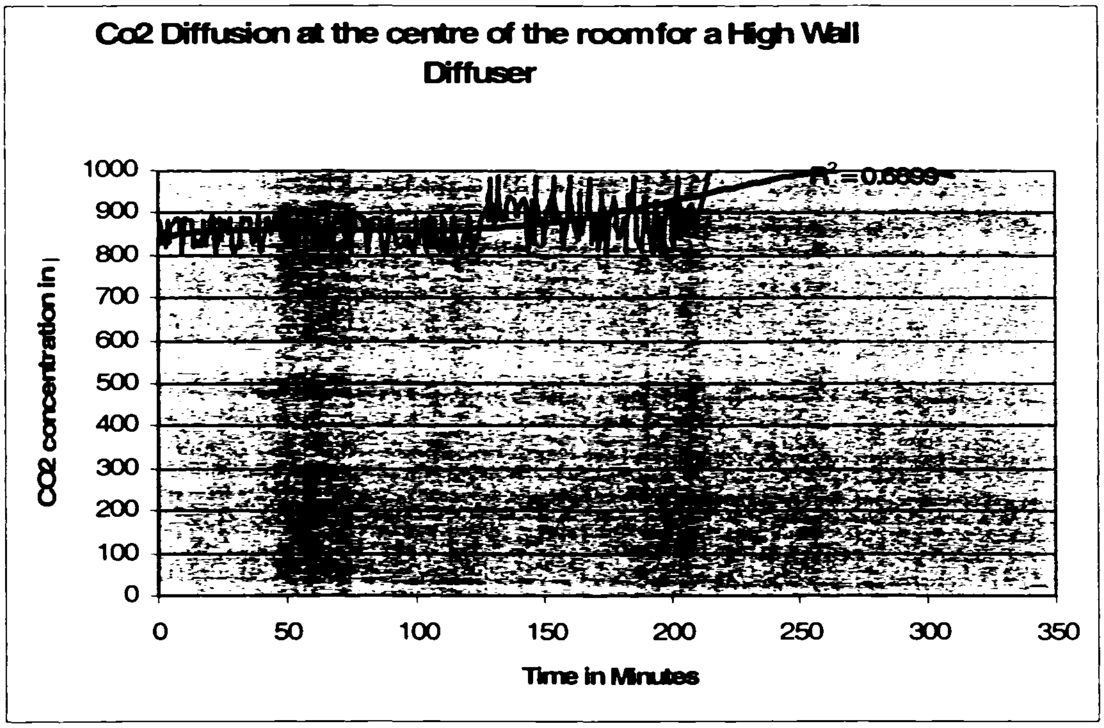

Figure 4.22 Dissipation of carbon dioxide in high wall diffuser

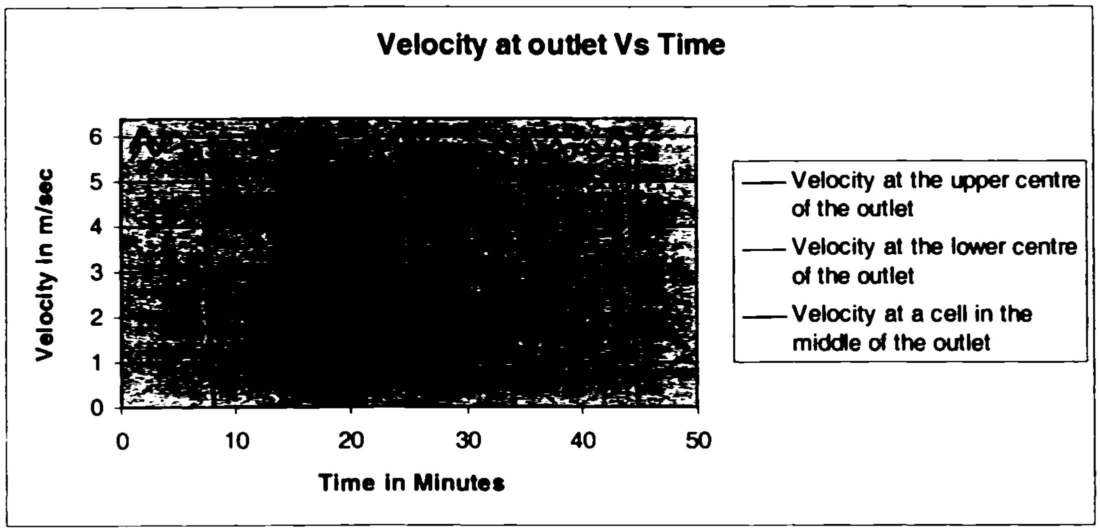

Figure 4.23 Velocities at the three points on the outlet to show mass conservation is achieved

As we have seen earlier, the High wall diffuser showed faster reaction in attaining thermal and velocity stability. The remaining two cases showed stability on a slower pace (they attained the stability criterion with the 6 cells per foot while High wall diffuser attained it at 5 cells per foot). The interesting lag and change in phase could be attributed to the fact that the room cannot come to a sudden and steep change in temperature as the inlet air was cutoff but gradually decreased over a period of time. That was clearly observed in all the three cases.

Presented below are the plots for the same: Two cases of once when the inlet was open for the longest time and the inlet closed for the longest time have been presented here. Let us first take the case when the Inlet is open for the maximum time i.e. the velocity of air is a finite value. The time involved in the two cases have been taken from immediate cycles. The case when the inlet was open for the maximum amount of time was at the 157th minute while the time when it was closed for the maximum amount was at the 166th minute. Each on off cycle was repetitive for about three times an hour.

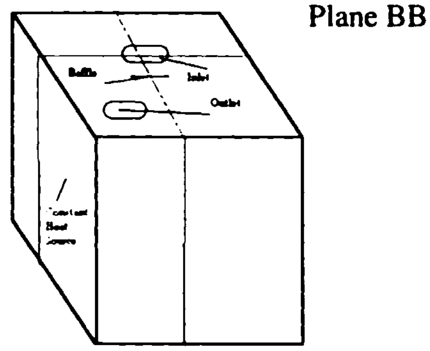

Figure 4.23 Cut AA and BB for the CAD

Ceiling Air Diffuser

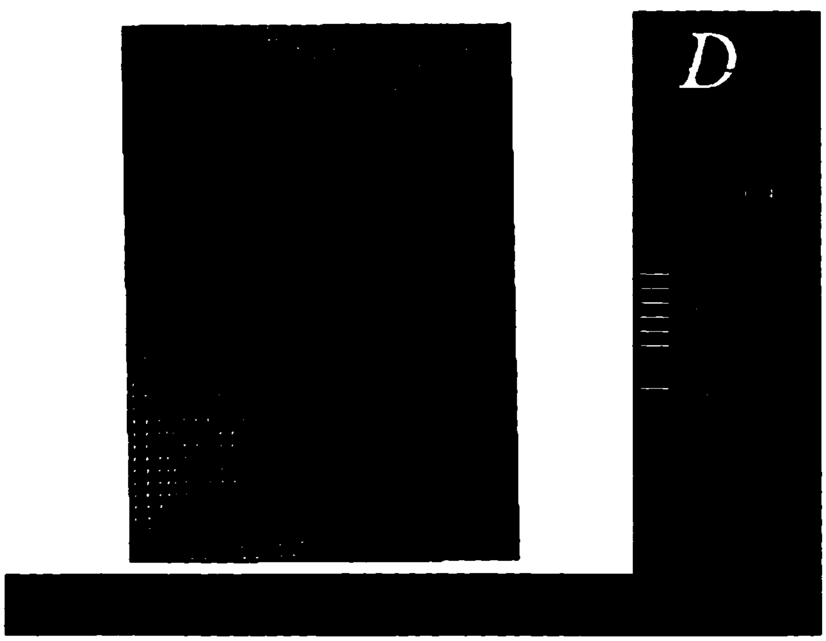

Figure 4.24.Velocity distribution for ceiling diffuser when inlet is open for maximum time

From the figure we observe that the temperature stratification is from 287.4° K to 304.8° K. The reason why the temperature stratification was from decreasing to increasing could be attributed to the fact that the floor of the room was adiabatic which is not allowing any heat to transfer through it making the temperature at a higher point at the bottom level.

The generation of heat pocket at the lower end of the room is because of the fact that the air that is getting distributed from inlet

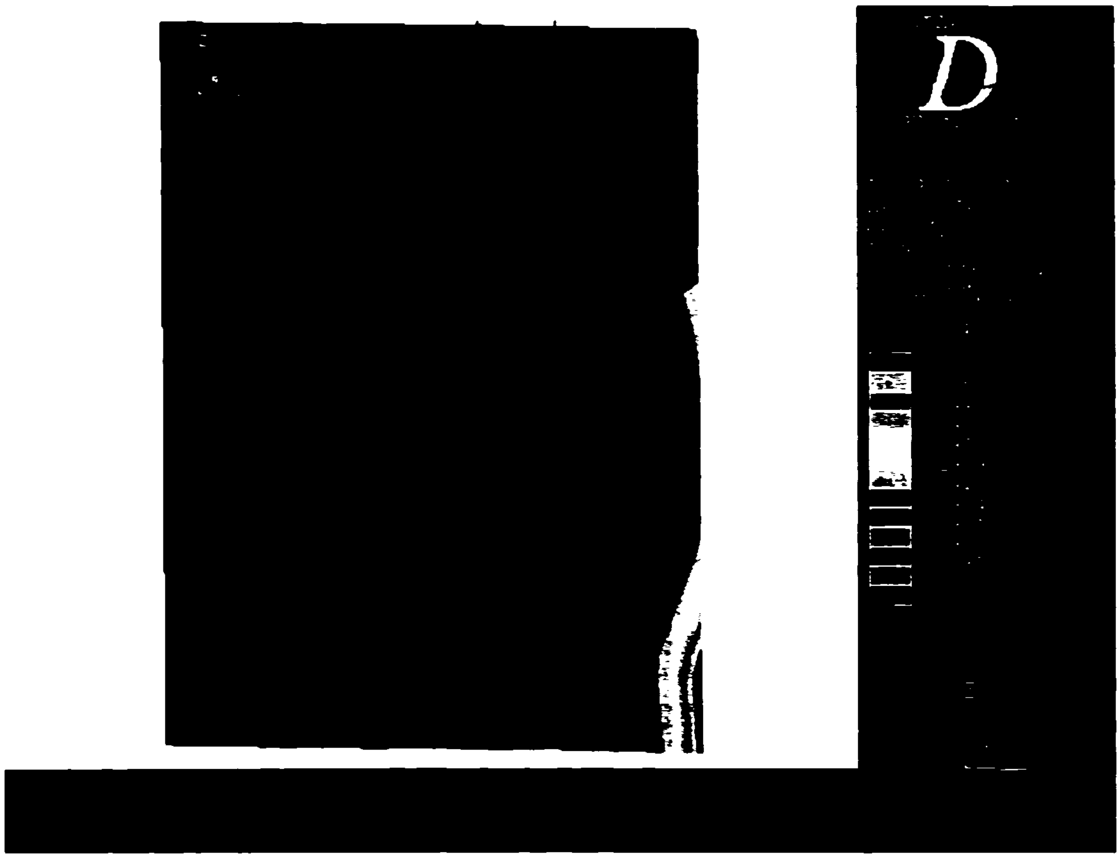

Figure 4.26. Temperature distribution for ceiling diffuser when inlet is open for maximum time

The inlet velocity is immediately obstructed by the baffle that causes the air to diffuse and circulate in the room in a manner similar to blades on a diffuser. The large magnitudes of velocity that are observed are found only at the inlet and the regions considered to be living space are more or less in the comfort zone as specified by the standards i.e. in the range of 50 to 150 fpm .The re-circulated air gets to the ceiling and circulates for some more time till the velocity comes to a point where it cannot circulate

by itself and is ejected from the outlet. Recirculation zones are observed in the centre of the room. Induced circulation is observed right under the baffle.

Distribution of contaminant CO_2

The distribution of carbon dioxide in the room is presented here. Initially a concentration of 1000 parts per million was introduced through the supply air duct to get into the room along with air and we can see the gradual decrease in the concentration of the same. According to ASHRAE 2004, the occupants of a room would be thermally comfortable if the difference in Carbon dioxide levels is less than 700 parts per million. The distribution of the contaminant and its diffusion is as anticipated.

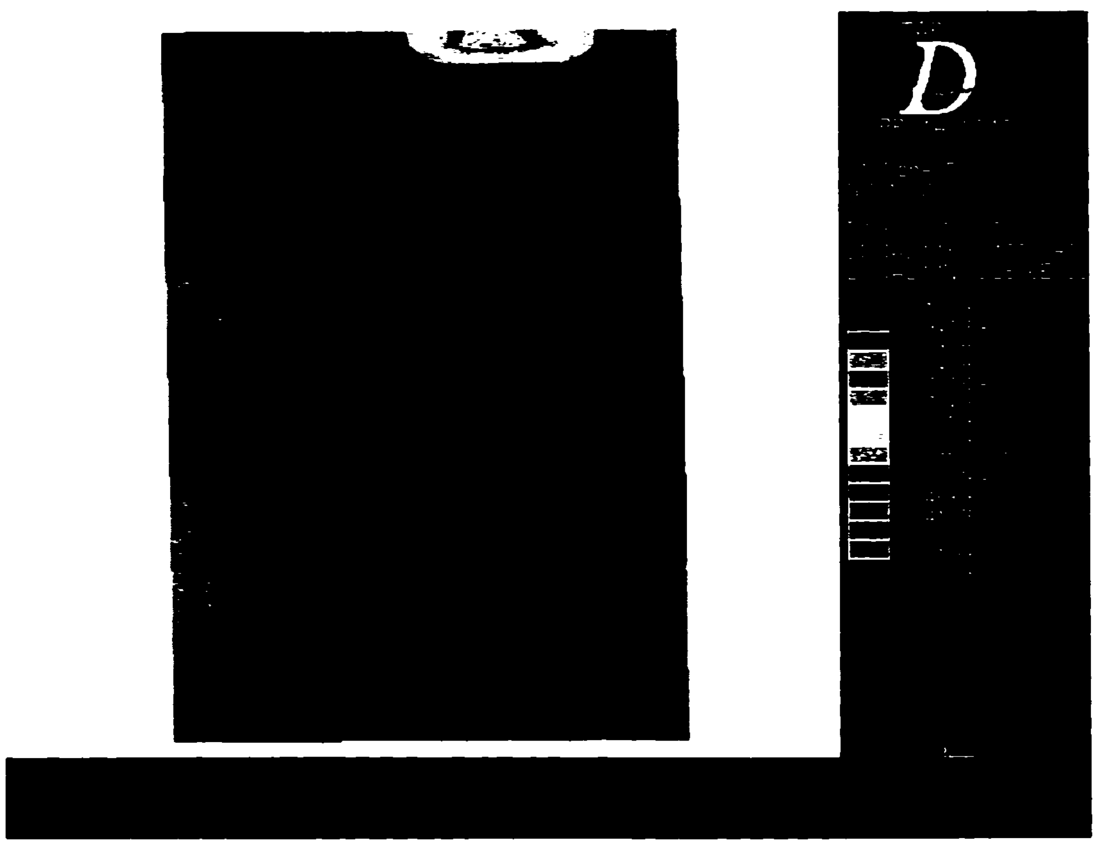

Figure 4.27. Carbon dioxide distribution for ceiling air diffuser when the inlet is open for maximum time

Let us now see the cases when the Inlet was closed for the maximum amount of time. In this case, the velocity at inlet is zero.

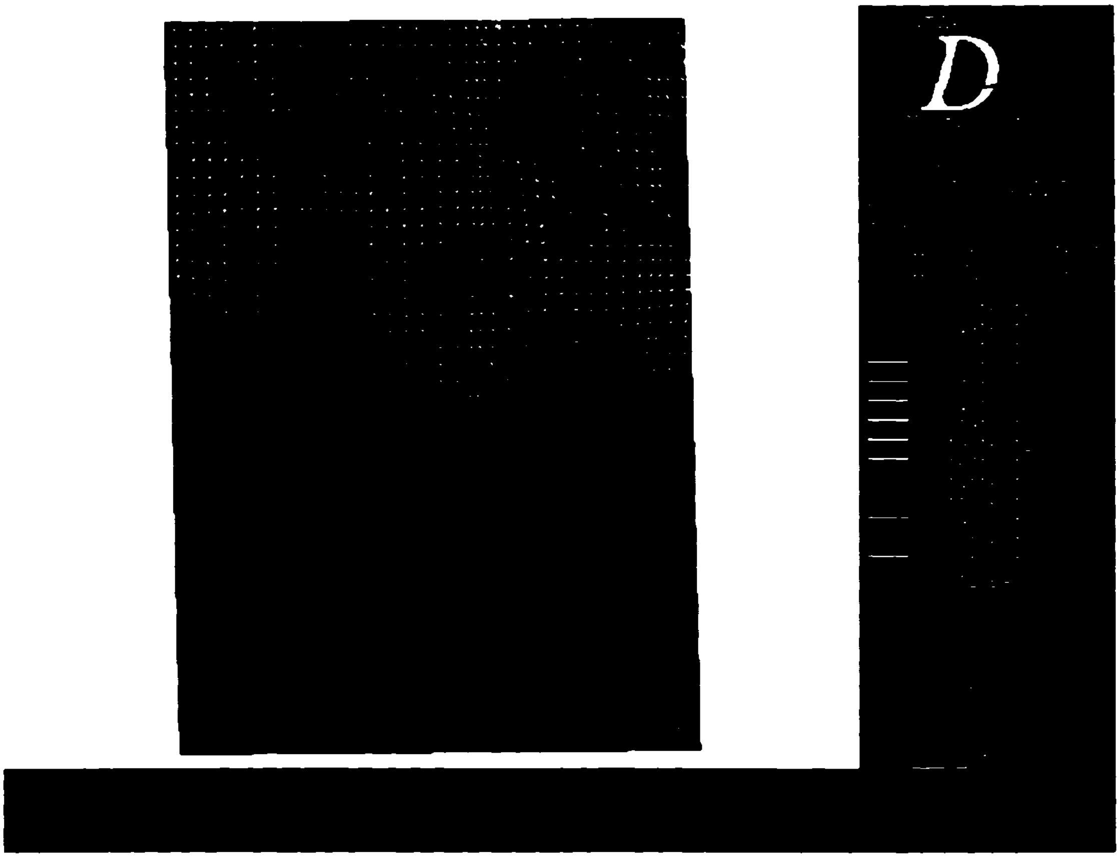

Figure 4.28 Velocity distributions for ceiling diffuser when inlet is closed for maximum time

We see that in this case, since the inlet is closed for the maximum extent, there is a gradual rise in temperature until the boundary condition subroutines are satisfied and then the inlet opens again. This holds true for the velocity and carbon dioxide profiles too. In the case of velocity plot, we see that the velocity reaches to an optimum value before the inlet opens again. The value of the velocity reaches an almost zero value that shows that the thermostat is functioning as it is desired and shuts off at regular intervals. Shown below are the profiles for the velocity and carbon dioxide. The lowest temperature found

when the inlet is closed for the maximum amount of time is higher than the case when the inlet is open for obvious reasons.

Figure 4.29 Temperature plot for ceiling diffuser when inlet is closed for maximum time

High Wall diffuser

Figure 4.31 Velocity distributions for high wall diffuser when the inlet is open for the maximum time

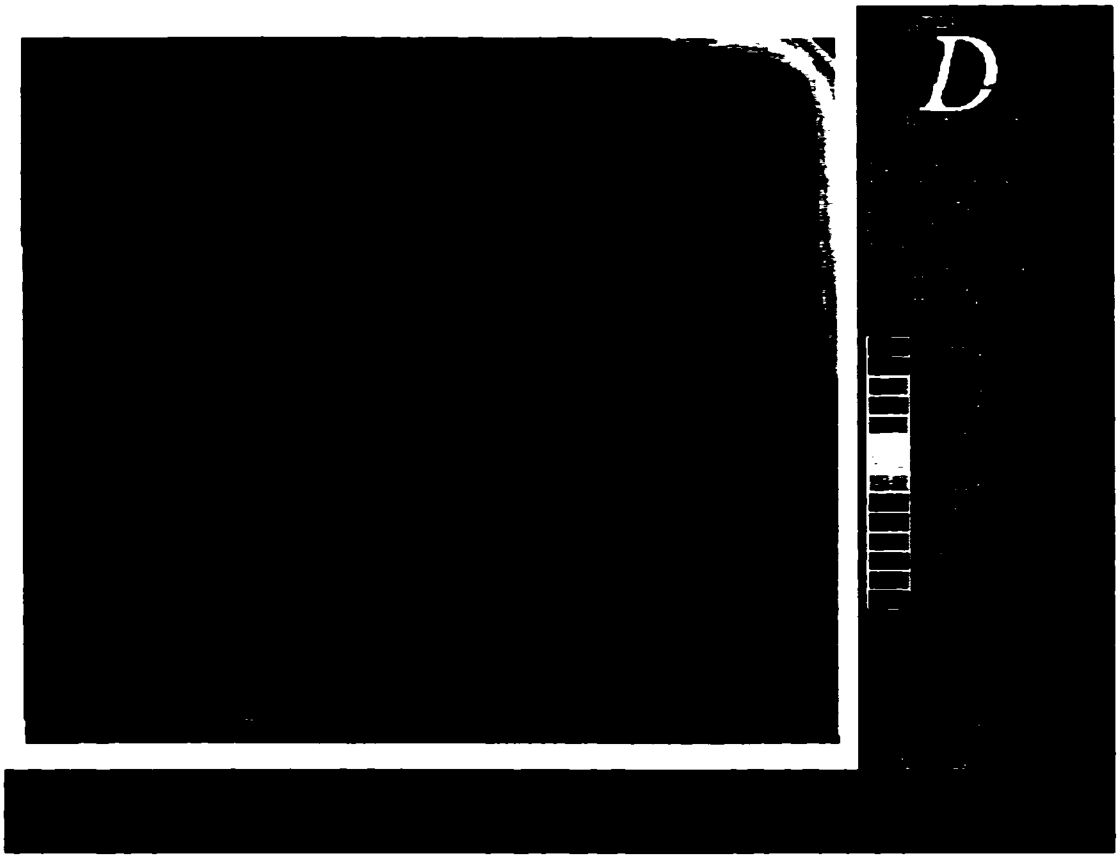

Figure 4.32 Temperature distributions for high wall diffuser when the inlet is open for the maximum time

Again the velocities initially are in a greater range, but once they hit the baffle they come down to values that are normally found in a room. The value of temperature magnitude found in this case is slightly higher than in the case of a CAD owing to the fact that in a CAD the distribution of air is more uniform and since cold air is heavier than hot air, the baffle acts as a good deflecting medium that makes uniform distribution of the air (cold) throughout the room than in the case of a high wall diffuser.

Let us now consider the case when the inlet is closed or the velocity is zero.

Figure 4.34 Velocity distributions for high wall diffuser when the inlet is closed for the maximum time

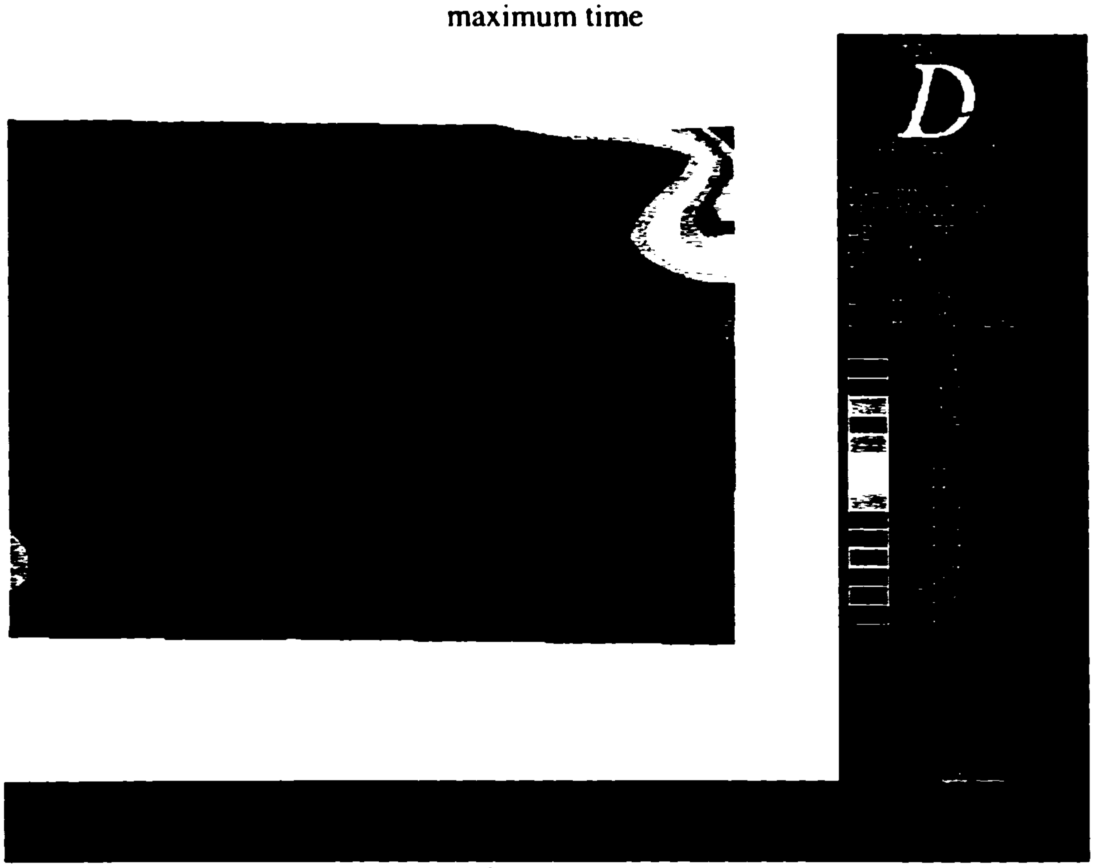

Figure 4.35 Temperature distributions for high wall diffuser when the inlet is closed for the maximum time

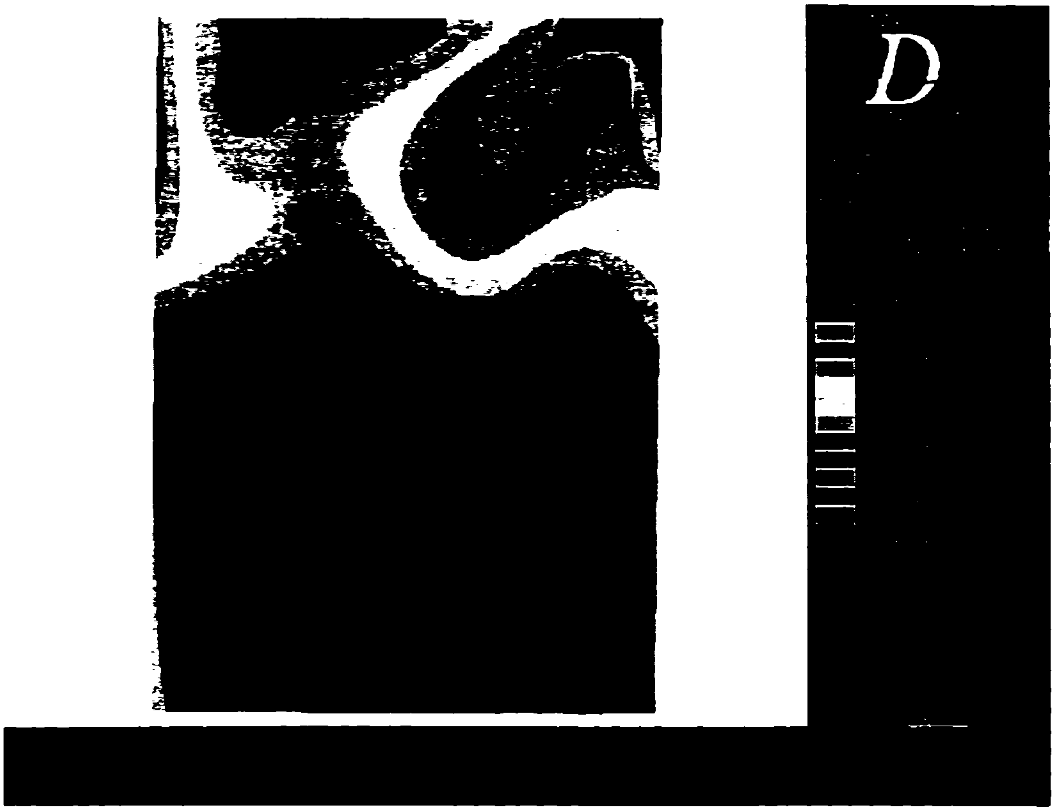

Figure 4.36 Carbon dioxide distributions for high wall diffuser when the inlet is closed for the maximum time

Under floor air Distribution

The last case under consideration is the under floor air distribution where the air enters the system through the floor. Let us consider the case when there is a finite velocity for the air that is coming into the room. In this case, we see that the cold air which has to come under the floor is heavier and again comes down. So we see that greater amount of room has temperatures higher than the initial temperature that is entering the room. The air that gets heated up escaped through the outlet. The velocity profiles that are seen are similar as in the other two cases.

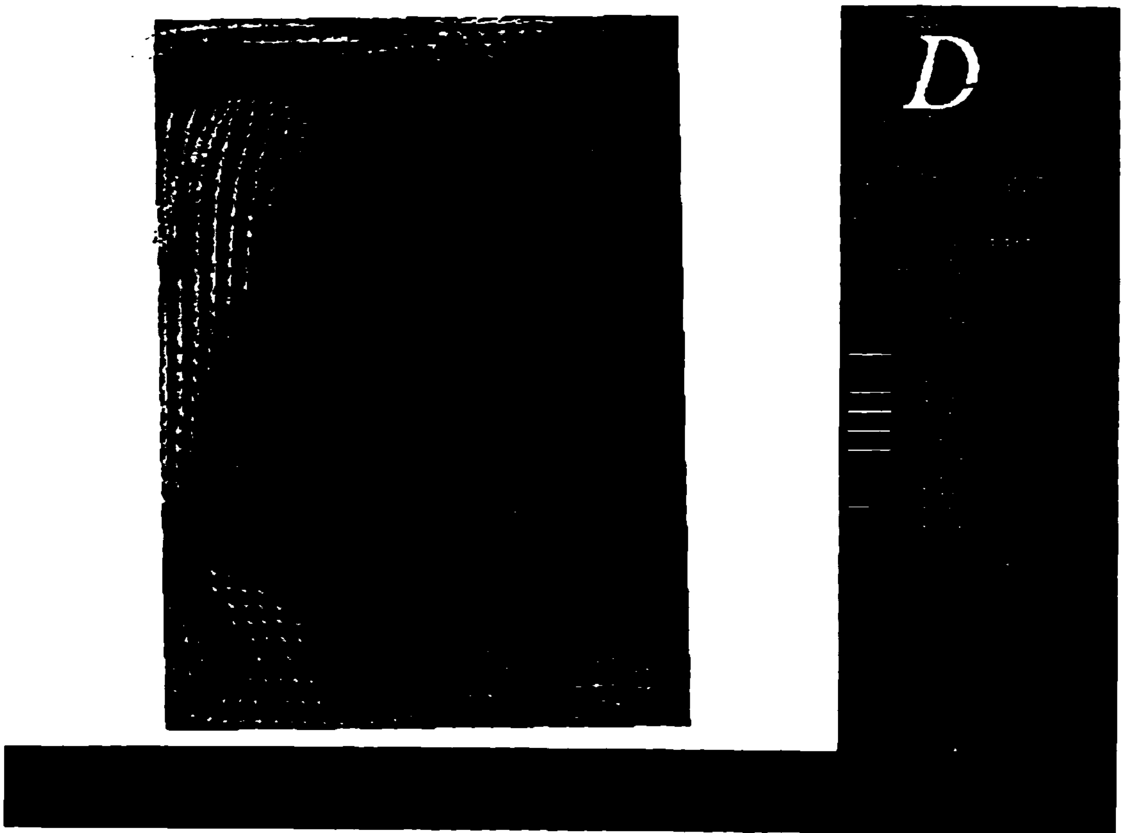

Figure 4.37 Velocity distributions for ufad when the inlet is open for the maximum time

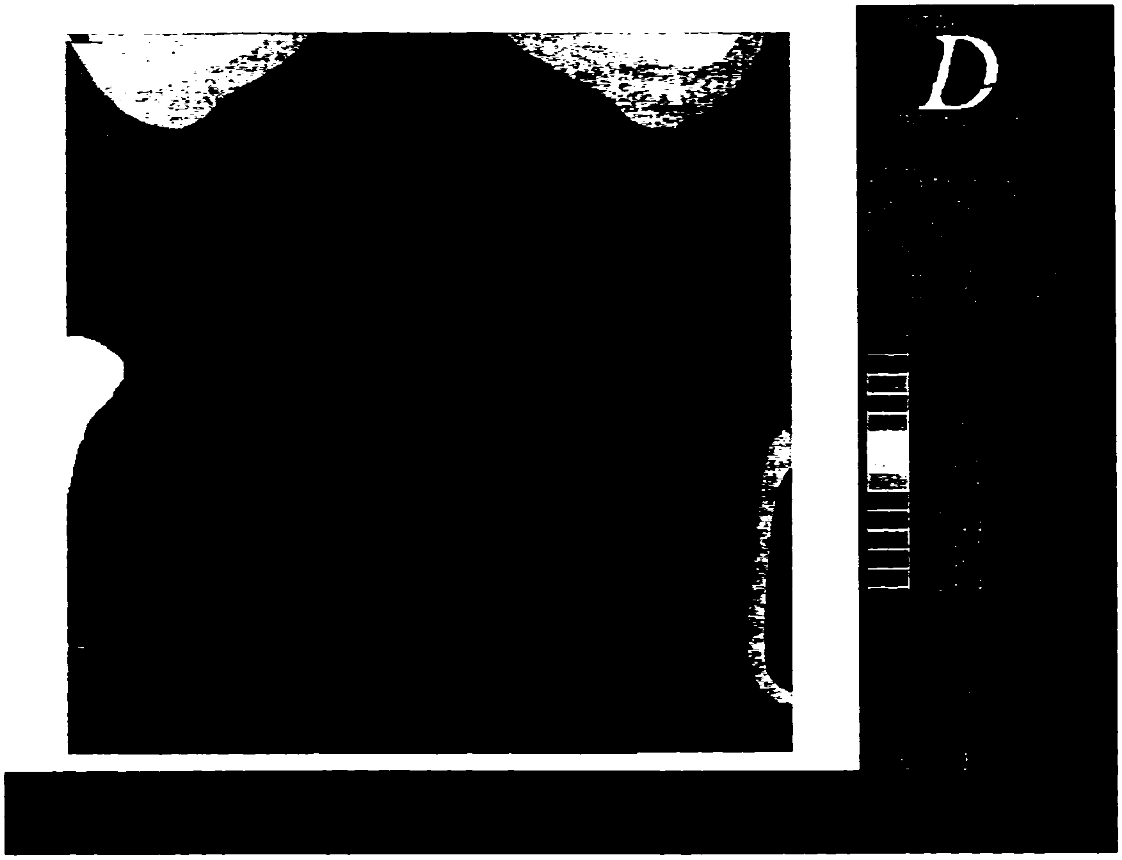

Figure 4.38 Temperature distributions for ufad when the inlet is open for the maximum time

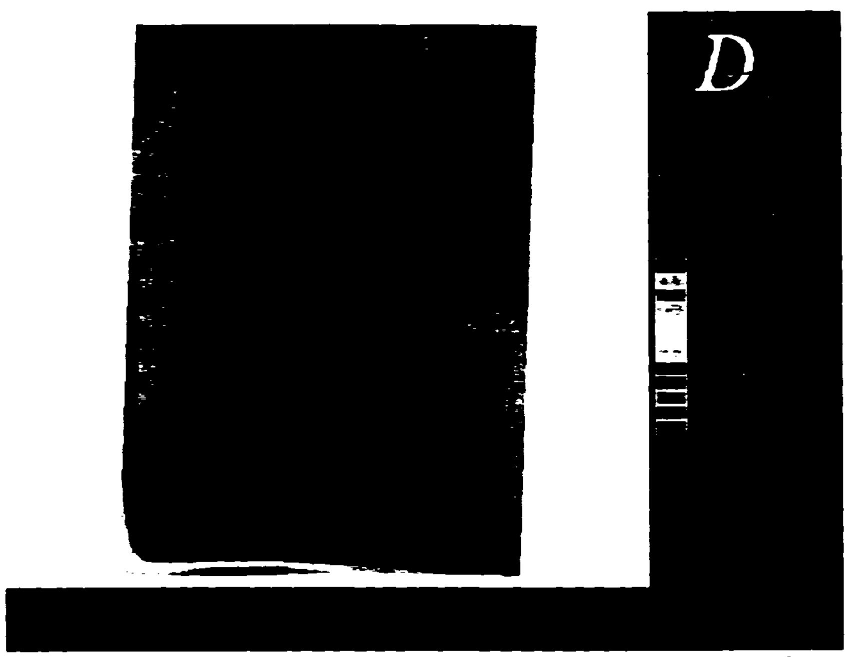

Figure 4.39 Carbon dioxide distributions for ufad when the inlet is open for the maximum time

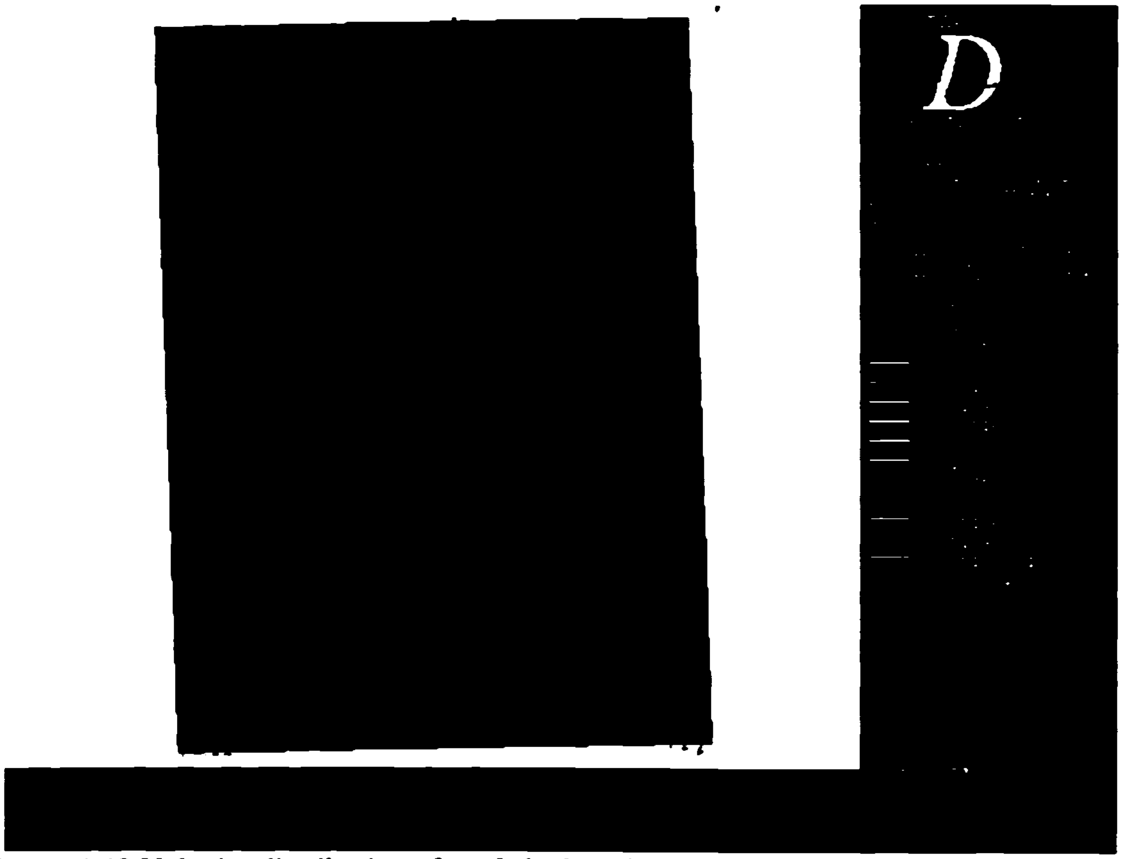

Figure 4.40 Velocity distributions for ufad when the inlet is closed for the maximum time

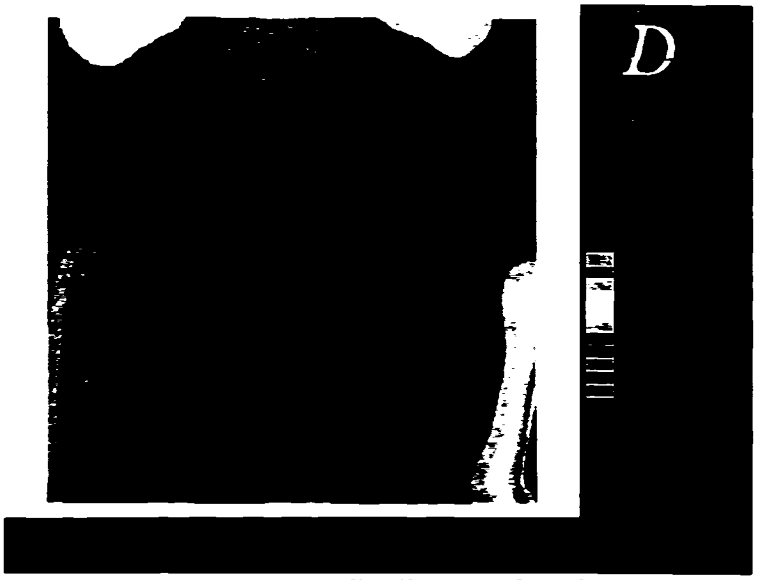

Figure 4.41 Temperature distributions for ufad when the inlet is closed for the maximum time

Figure 4.42 Carbon dioxide distributions for ufad when the inlet is closed for the maximum time

CHAPTER 5

CONCLUSIONS AND FUTURE WORK

Some of the important conclusions that can be drawn from the above work are categorized as follows:

Detailed Analysis Using Multivariate Tests it is found that, at 95% confidence level, the building has significant influence to all the measured parameters, and the day and time (morning or afternoon) have no significant influence to all the measured parameters but the day to the relative humidity. Of the five buildings that were analyzed, three buildings were within the comfort zone according to ASHRAE 55-2004 (thermally) and in one of the buildings, 21% of the occupants felt it was slightly cool and the remaining occupants felt that it was thermally comfortable. Four buildings were in the comfort zone as far as the humidity was considered and in one of the building, 12% of the occupants felt that it was slightly dry. 5 % of the occupants felt that the building was slightly drafty. None of the occupants felt that there was any inconvenience due to vertical temperature difference. Of the five buildings three were found to be fresh all the time while one of them was fresh most of the time and another one fresh some of the time. Most (92%) of the temperature fell within the specified range of ASHRAE comfort zone for both summer and winter conditions. The relative humidity data was well within the specified range of ASHRAE 55-2004 for thermal comfort. All the Vertical temperature difference fell within the range as specified by the ASHRAE for the comfort of the occupant

The PMV and PPD which are important indices in calculating thermal comfort parameters showed consistency in their results. Since the data obtained was over a small period of time and small data set (i.e. number of buildings were 5), general agreement cannot be made on the validity of the standard ASHRAE 55-04. An error percentage based on standard deviations for the questionnaire has been calculated and it was found that the percentage of error in the occupant perception survey was of the order approximately 17% in building 1.

Results of CFD simulation

The grid independency showed that the chosen size of the cells did not have a significant effect over the variables solved by CFD code as shown in the previous graphical results. The on-off of the thermostat was taking place as expected by the subroutine to take care of . The Ceiling Air Diffuser showed faster tendency to reach stability both temperature wise and velocity wise than the remaining two configurations. The temperature, velocity and carbon dioxide distribution was found to be the best suited for the "living conditions" which are between 0 ft and 7 feet. The values of the same are given below

The following results were obtained by visual inspection, dominant areas

- CAD showed velocities, temperature and distribution of carbon dioxide which were lesser than the remaining two

The values for the same are shown below

Velocity – around 1.015 m/sec

Temperature – 288K

Distribution of contaminant was seen to be the best in a CAD than in the other two cases and the average value was found to be around 325 ppm.

- UFAD

Velocity - around 1.520 m/sec

Temperature – 291 K

Carbon dioxide distribution – around 475 ppm

-High Wall Diffuser

Velocity -- around 1.63 m/sec

Temperature ---289 Kelvin

Carbon dioxide concentration- around 450 ppm

The velocity plots showed a regular on off situation at every 14 minutes on an average which seemed like a realistic value that happens in a room normally. The carbon dioxide which was used as a contaminant diffused uniformly throughout the room in all the three cases and the concentration of the contaminant was well defined in the time region that was specified. We notice that when the inlet is closed for the maximum time, the concentration of the contaminant reduces from the maximum value of 1000 ppm. We also see that when the inlet is closed for a certain period of time, stagnation points are created. The variation of carbon dioxide diffusion is at a lesser gradient when the inlet is closed for obvious reasons. The velocity magnitudes at outlet are within the region of the velocity region at inlet, which leads to the conclusion that mass conservation is

maintained. Also we see that the momentum equations are converged. Also it was noticed that the region in which human occupancy is found i.e. from 0 feet to 8 feet, the temperature, velocity and the distribution of the contaminant were within limits. As shown above, the values of velocity, temperature and diffusivity of the contaminant were well observed in the case of a CAD. Thus it can be concluded that a CAD is better than the remaining two types of flow situations under the same set of flow conditions.

The initial study has been done on five office spaces and the remaining five office buildings are being monitored in the next couple of months. In the following two years, academic institutions and health care facilities are being taken into consideration. Also expected is the fact that there will be a greater statistical data that may help one reach to a better conclusion that may potentially challenge the ASHRAE 55-2004 as far as the results are considered for suggesting some changes in the standard. The hypotheses that are formulated will be tested and will be checked if they are correct or not. A more realistic source of contaminant is to be introduced at more regular intervals that would make the flow pattern more realistic. Similarly a more realistic cooling load that is generated is to be used for better results.

APPENDIX A

THEORETICAL MODELS

PMV-PPD

PMV represents the 'predicted mean vote' (on the thermal sensation scale) of a large population of people exposed to a certain environment. PMV is derived from the physics of heat transfer combined with an empirical fit to sensation. PMV establishes a thermal strain based on steady-state heat transfer between the body and the environment and assigns a comfort vote to that amount of strain. PPD is the predicted percent of dissatisfied people at each PMV. As PMV changes away from zero in either the positive or negative direction, PPD increases.

The PMV equation for thermal comfort is a steady-state model. It is an empirical equation for predicting the mean vote on an ordinal category rating scale of thermal comfort of a population of people. The equation uses a steady-state heat balance for the human body and postulates a link between the deviation from the minimum load on heat balance effector mechanisms, e.g. sweating, vaso-constriction, vaso-dilation, and thermal comfort vote. The greater the load, the more the comfort vote deviates from zero. The partial derivative of the load function is estimated by exposing enough people to enough different conditions to fit a curve. PMV (Predicted Mean Vote), as the integrated partial derivative is now known, is arguably the most widely used thermal comfort index today.

The ISO (International Standards organization) Standard 7730 (ISO 1984), "Moderate Thermal Environments -- Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort," uses limits on PMV as an explicit definition of the comfort zone.

The PMV equation only applies to humans exposed for a long period to constant conditions at a constant metabolic rate. Conservation of energy leads to the heat balance equation:

$H - Ed - Esw - Ere - L = R + C$

Where:

H = internal heat production

Ed = heat loss due to water vapor diffusion through the skin

Esw = heat loss due to sweating

Ere = latent heat loss due to respiration

L = dry respiration heat loss

R = heat loss by radiation from the surface of the clothed body

C = heat loss by convection from the surface of the clothed body

The equation is expanded by substituting each component with a function derivable from basic physics. All of the functions have measurable values with exception of clothing surface temperature and the convective heat transfer coefficient which are functions of each other. To solve the equation, an initial value of clothing temperature is estimated, the convective heat transfer coefficient computed, a new clothing temperature calculated etc., by iteration until both are known to a satisfactory degree.

Now let us assume the body is not in balance and write the heat equation as:

L = H-Ed-Esw-Ere-L-R-C,

where L is the thermal load on the body.

Define thermal strain or sensation, Y, as some unknown function of L and metabolic rate. Holding all variables constant except air temperature and metabolic rate, we use mean votes from climate chamber experiments to write Y as function of air temperature for several activity levels. Then substituting L for air temperature, determined from the heat balance equation above, evaluate the partial derivative of Y with respect to L at Y=0 and plot the points versus metabolic rate. An exponential curve is fit to the points and integrated with respect to L. L is simply renamed "PMV" and we have (in simplified form):

PMV = exp [met]*L.

Where:

L=F (Pa, Ta, Tmrt, Tcl)

PMV is "scaled" to predict thermal sensation votes on a seven point scale (hot, warm, slightly warm, neutral, slightly cool, cool, cold) by virtue of the fact that for each physical condition, Y is the mean vote of all subjects exposed to that condition. The major limitation of the PMV model is the explicit constraint of skin temperature and evaporative heat loss to values for comfort and "neutral" sensation at a given activity level.

ET*-DISC

ET*- DISC also uses a heat balance model to predict thermal comfort, but the model evolves with time rather than being steady-state like PMV. ET* stands for New Effective Temperature where "effective temperature" is a temperature index that accounts for radiative and latent heat transfers. ET* can be calculated using the '2-Node' model. The 2-node model determines the heat flow between the environment, skin and core body areas on a minute by minute basis. Starting from an initial condition at time=0, the model iterates until equilibrium has been reached (60 minutes is a typical time). The final mean skin temperature and skin wetted ness are then associated with an effective temperature. DISC predicts thermal discomfort using skin temperature and skin wetted ness. The 2-node model was introduced in 1970 specifically to formulate a new effective temperature scale. The purpose was to determine particular combinations of physical conditions producing equal physiological strain. Backed by extensive data from climate chamber experiments, it was determined that while skin temperature is a good indicator of thermal comfort sensation in cold environments, skin wetted ness is a better indicator in warm environments where sweating occurs because skin temperature changes are small by comparison. The model represents the human body as two concentric cylinders, a core cylinder and a thin skin cylinder surrounding it. Clothing and sweat are assumed to be evenly distributed over the skin surface. At time "zero", the cylinder is exposed to a uniform environment, and the model produces a minute-by-minute simulation of the human thermoregulatory system. After the user-specified time period is reached, the final surface temperature and surface skin wetted ness of the cylinder are used to calculate ET*, SET*, and other indices. ET* is the temperature of an environment at 50% relative

humidity in which a person experiences the same amount heat loss as in the actual environment.

SET*

SET* numerically represents the thermal strain experienced by the cylinder relative to a "standard" person in a "standard" environment. SET* has the advantage of allowing thermal comparisons between environments at any combination of the physical input variables, but the disadvantage of also requiring "standard" people.

Based on a laboratory study with a large number of subjects, empirical functions between two comfort indices, and skin temperature and skin wetted ness, were developed. These functions (both linear) are used in the 2-Node model to produce predicted values of the votes of populations exposed to the same conditions as the cylinder.

TSENS, DISC

TSENS, the first index, represents the model's prediction of a vote on the seven point thermal sensation scale. DISC, the second index, predicts a vote on a scale of thermal discomfort:

DISC:

Intolerable

Very uncomfortable

Uncomfortable

Slightly uncomfortable

Comfortable

The 2-Node model has undergone many iterations and refinements. In the most recent iteration, a new temperature index, PMV*, that incorporates skin wetted ness into the PMV equation using SET* or ET* to characterize the environment.

Empirical Models

Apart from the thermal comfort models described above, there are many more theoretical models, both deterministic and empirical. Some empirical models with application to building design and/or environmental engineering are outlined below.

PD

PD or "predicted percent dissatisfied due to draft" is a fit to data of persons expressing thermal discomfort due to drafts. The inputs to PD are air temperature, air velocity, and turbulence intensity. PS is a fit to data of comfortable persons choosing air velocity levels. The inputs to PS are operative temperature and air velocity. TS is a fit to data of thermal sensation as a linear function of air temperature and partial vapor pressure.

A 'draft' is unwanted local cooling. The draft risk (or PD) equation is:

$PD=3.413(34-Ta)\ (v-0.05)^{0.622}+0.369vTu\ (34-Ta)\ (v-0.05)^{0.622}$

Tu is the turbulence intensity expressed as a percent. 0 represents laminar flow and 100% means that the standard deviation of the air velocity over a certain period is of the same order of magnitude as the mean air velocity. v is the air velocity (in meters per second)

and Ta is the air temperature in degrees Celsius. The PD equation arises from two studies in which 100 people were exposed to various combinations of air temperature, air velocity, and turbulence intensity. For each combination of conditions, the people were asked if they felt a draft. PD represents the percent of subjects who voted that they felt a draft for the selected conditions.

PS

The PS equation predicts the air velocity that will be chosen by a person exposed to a certain air temperature when the person has control of the air velocity source. The PS equation is

PS=1.13SQRT (Top)-0.24Top+2.7SQRT (v)-0.99v

Top is operative temperature (in degrees Celsius) and v is the air velocity in meters per second. The PS equation arises from a study in which 50 people were asked to adjust an air velocity source as they pleased when exposed to a specific air temperature. PS represents the cumulative percent of people choosing a particular air velocity at the specific temperatures tested in this experiment

TS

TS is an equation that predicts thermal sensation vote using a linear function of air temperature and partial vapor pressure. The TS equation is:

TS=0.245Ta+0.248p-6.475

Ta is the air temperature in degrees Celsius and p is the partial vapor pressure in kilo-Pascal. The TS equation arises from a study similar to the PMV-PPD study described above.

Adaptive Models

Adaptive models include in some way the variations in outdoor climate for determining thermal preferences indoors.

Auliciems

An adaptive model developed by Auliciems fits sensation data based on field investigations of thermal comfort in Australia spanning several climates. Auliciems equation is:

$$Tn=9.22+0.48Ta+0.14Tmmo$$

Humphreys

Humphreys equation is a fit to considerable data for climate-controlled and non-climate controlled buildings:

$$Tn=23.9+\frac{0.295(Tmmo-22)}{e^{-\left[\frac{(Tmmo-22)}{24SQRT(2)}\right]^{2}}}$$

For both the Auliciems and Humphreys models, Tn is the neutral temperature, Ta is the air temperature, and Tmmo is the mean monthly outdoor temperature.

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