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Experimental Comparison of Three Heating Systems With Respect to Heat Distribution and Thermal Comfort

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ABSTRACT

Experimental Comparison of Three Heating Systems With Respect to Heat Distribution and Thermal Comfort

Martin Auger

Heating systems were compared based on experiments performed during two heating seasons. Measurements of air temperatures, surface temperatures, air velocity, relative humidity and comfort are presented. Results are interpreted in terms of heat distribution and thermal comfort produced by three types of systems: a zone convector and two central systems (forced air and hydronic baseboard) Recent studies on thermal comfort research are reviewed as well as recent experiments on comparison of heating systems The building and instrumentation used are briefly described. Sections cover air temperature distribution observed in space and time, temperature gradients, and estimation of comfort indices (Predicted Mean Vote, Predicted Percentage of Dissatisfied). In the end, suggestions and recommendations are made for improving the design and control of the three systems.

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Finally, I would like to dedicate this work to my parents, my family and Gisele.

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NOMENCLATURE

a	Coefficient of interpolation
b	Intercept
C	Convective heat exchange
E	Heat losses from sweat evaporation
Icl	Clothing insulation
K	Conduction heat exchange
LPPD	Lowest possible percentage of dissatisfied
M	Metabolism
m	Slope
PMV	Predicted mean vote
PPD	Predicted percentage of dissatisfied
R	Radiation heat exchange
RES	Heat exchange by respiration
Rh	Relative humidity
S	Heat storage or thermal load
t	Time
T	Temperature
Ta	Air temperature
Tg	Globe Temperature
T_{Otest}	Test outdoor temperature
T_{out}	Outdoor temperature
Tmr	Mean radiant temperature
Tw	Outdoor temperature from weather station
Va	Air velocity
W	Work
y	Height

CHAPTER 1

1. INTRODUCTION

The following study compares the indoor environment created by three different types of heating systems installed in a three storey apartment building located in downtown Montreal. Experiments were conducted during an entire winter season and consisted in the measurement of the physical parameters important for human thermal comfort namely, air temperature, surface temperatures, air velocity and relative humidity. The three systems were gas operated: a conventional forced air system, an hydronic system and a new type of modular system. The main objective was to compare the thermal distributions created by the systems in a medium sized bedroom of the building and to rate these distributions according to thermal comfort criteria.

The discussion is addressed mainly to persons interested in indoor environment, comfort studies, design and analysis of heating systems and building science in general.

1.1 General Problem Statement

In general, it is important for humans to use the best means available to at least survive and, far better, to live comfortably within a certain geographical area. In particular, for cold regions, this implies two main components which are an enclosure and/or a heating system. Eventually, there will exist the problem of finding what are the best type of enclosures and the best types of heating systems. Information on this subject can be obtained from previous theories or by conducting new experiments.

Looking at them closely, enclosures and heating systems become two intimately related research areas. The number of ways these areas can combine in the field is staggering. When the number of different buildings, heating systems, weather conditions and types of different people are taken together, it becomes difficult to answer questions such as: What is the best building and/or the best heating system? There is a need to select an efficient approach, be it empirical, theoretical or a combination of both to solve this problem.

This report is an effort to address this general problem from the experimental point of view. By describing and analyzing actual cases of systems working a building, it is hoped that interesting insights are possible to be discovered. These insights are necessary to improve the design and analysis of heating systems in general.

Within this experimental framework, two main steps were necessary to obtain results necesary to compare heating systems as they were installed in the building.

There was the collection of a large number of experimental data on the enclosure and the environment created by the heating systems in an apartment building. This first part meant recording air temperatures, air velocity, surface temperatures and relative humidity during five months of a winter season.

Second, the results were processed to establish heat distribution and comfort created by the systems. This meant studying the measured parameters to establish their time and space variations as well as their likely effects on human beings. It included understanding the physical meaning of the data as well as their implications on thermal comfort.

The general problem faced and treated by this report was to establish the behaviour of the systems, to describe how they functioned, to rate their performance and to test their behaviour and this, for real conditions of operation. The analysis is based on building environement studies, building science and recent studies on thermal comfort for human beings.

1.2 Chapters' Description

Chapters 2 and 3 respectively provide background information on thermal comfort research and test facilities. Chapters 4 to 7 contain the results obtained from the tests performed. Chapter 8 summarizes the results and ways to improve the heating systems are suggested.

Chapter two describes the results from the literature survey which preceded the actual test period. You will find information on the existing thermal comfort indices and more specifically on the predicted mean vote index (PMV) used for the evaluation of global comfort. Local discomfort and global comfort will then be discussed separately. The current standards on thermal comfort (ISO DIS 7730 and ASHRAE 55-81) will then be reviewed. Finally, a review of previous experiments on comparison of heating systems will be performed.

Chapter three describes the test facilities, instrumentation and the data acquisition system used. Information on the building's envelope properties is provided as a reference It is included to show the applicability of the results, and may be useful to compare the results with results from other existing facilities or to apply the results for design or simulation work.

Chapter four includes a description of the general test procedure used and the describes the distribution of outdoor temperatures obtained throughout the season for each system. As can be seen in Figure 1.1, analysis of the data was performed either for specific cases or for the whole season.

The case approach consists in studying only a few days in depth in order to draw certain conclusions on a particular point of interest.

The long period approach (whole season) consists in pooling together a large number of days in order to obtain general statistics and identify trends for the heating season. Note that the "seasonal" distribution is actually a subset of the winter season surveyed.

Chapter four also discusses four important aspects: the cycling patterns of each system, the room temperature distributions and fluctuations created in the rooms, and the systems' response to low initial room temperatures. Information is based mostly on air and surface temperature measurements. The general idea is to describe how each system cycles in time as a function of different outdoor conditions and to establish the influence of this cycling on room temperature distribution. Also, it is important to verify how temperatures were maintained with respect to thermostat settings and to find how fast the systems can warm up a cold room.

```
SEASONAL OUTDOOR
AND INDOOR TEMPERATURE
DISTRIBUTIONS
(subset of the actual
 winter season)
|
| chapter 4
|---------4 cases--------------------   |------   heater temperature cycling
|                                        |------   1 case-|----plane temperature
|                                        |                   distribution
|                                        |------   temperature fluctuations
| chapter 4
|---------whole --------------------    |------   average plane temperatures
|         season                        |------   temperature setting maintained
| chapter 4
|-------- 1 case -------------------    |------   transient warming
| chapter 5
|-------- 2 cases-------------------    |------   temperature profiles and
|                                        |         gradients
|                                        |------   1 case-|----relation between
|                                                             heater temperature
|                                                     and gradients
| chapter 5
|-------- 1 case -------------------    |------   temperature profiles
|                                                  with door opened
|                                                  (hydronic and forced air
|                                                  systems only)
| chapter 5
|---------whole -------------------     |---- -   temperature gradients
|         season                        |         distribution
|                                        |------   relation between gradients and
|                                                  outdoor temperatures
| chapter 7
\---------whole ------------------      |------   calculation of PMV, PPD, LPPD
          season                                  indices
```

Figure 1.1: Analysis Performed for Each System from Outdoor and Indoor Temperatures Data

Chapter five deals specifically with temperature profiles. They are discussed in terms of two specific cases (mild and cold outdoor temperatures) and trends in temperature gradients' magnitude are observed for the whole heating season (see Figure 1.1). A relation between heater temperature and gradients is also mentioned.

Chapter six concentrates on air velocity, mean radiant temperature and relative humidity results. This chapter is included separately since measurements of these parameters was not as extensive as for the other parameters but, establishing the relative importance of these parameters was necessary for comfort assessment.

Chapter seven, is devoted to thermal comfort assessment. This chapter contains a section on PMV index calculations and a section where measured parameters are compared with standards' requirements for local discomfort. For the PMV calculations, results from measurements using a comfort meter and the values of our measured parameters are compared. For the comparison of the results with the standards, a table summarizing the relative performance of the systems is shown.

As can be seen, the study first describes the physical behaviour of the systems (chapters 4 to 6) and later links this aspect with the thermal comfort aspect (chapter 7). Recommendations are being made in the end (chapter 8) to show how the systems could eventually be improved in their design when both of these aspects are related.

1.3 General Remarks

The particular limitations of field measurements should be taken into account when the results of this study are examined. As mentioned earlier, a large number of parameters could not be controlled or modified (e.g: weather variations, neighbouring buildings).

However, as far as possible, measurements were screened extensively beforehand to establish or limit the uncertainty created by these parameters on the results. Nevertheless, due to physical limitations, a certain level of uncertainty had to be accepted in the end in order to limit the scope of the research.

In this sense, the text contains either a verbal description of the numerical results obtained or interpretations of the phenomenon observed. Since interpretations are often subjective, it was tried as far as possible to make them correspond to the information available. In certain cases, however, due to the complexity of the field situation compared to the available information, certain assumptions were made based on personal experience and results from similar studies. Some results and their interpretation can still be debated until more tests are available.

CHAPTER 2

2. REVIEW ON THERMAL COMFORT RESEARCH AND COMPARATIVE STUDIES

This chapter describes the fundamental principles as well as the theoretical and experimental background on which the present study is based. The material presented was gathered for a literature review made at the beginning of the project and was used in the preparation of the experimental set-up to identify the type of experiments to be conducted.

The review is divided in three main sections. The first describes the selection of a thermal comfort index for global and local thermal discomfort. The second compares the current standards on thermal comfort. Finally, the third section reviews the experiments done up to now on thermal comfort for rooms equipped with different types of heating systems.

The original literature survey dealt with the same topics but with more detail and also covered other aspects such as energy conservation and the experimental procedures for evaluating thermal comfort. The summary presented here does not include as much detail and is a shortened version intended to cover only the most important aspects. Readers familiar with thermal comfort studies may prefer to go directly to the last section of this chapter.

2.1 Evaluation of Thermal Comfort

The study of the indoor environment in terms of thermal comfort can be divided in two groups with respect to the level at which the subject is dealt. First, for global comfort, an index can be used to describe globally or generally the thermal environment. The index provides a rating of the environment on a scale. This rating can be used to decide if the environment is acceptable or not. Any index is limited by the number of variables it incorporates, there is a possibility that other factors which are not taken into account will produce discomfort. Therefore, there is a second level of inquiry aside from global comfort which is local comfort or what is better known as local discomfort.

In the next two sections global comfort and local discomfort are treated separately. Indices available for global thermal comfort are described first. More emphasis is put on the Predicted Mean Vote (PMV) index and the Predicted Percentage of Dissatisfied index (PPD) selected for this study. The second section lists and describes important local thermal discomfort problems.

2.1.1 Global Thermal Comfort

Thermal comfort studies have been conducted since the beginning of the century. The efforts of many researchers were focused towards establishing an index or a scale to measure objectively and globally the subjective response of humans to their thermal environment At first, these studies concentrated on direct and empirical indices

Direct indices are derived from the behaviour of instruments which respond to the same thermal variables as humans. The readings from those instruments can be used as an index when these are correlated with the corresponding human response to the same variables Examples of direct indices are the dry and wet bulb temperatures [1]

In order to improve predictions, empirical indices were later developed. Empirical indices are obtained by systematically exposing subjects to different sets of thermal variables and noting their response [1].

Empirical indices include more variables than direct indices and the variables are selected to represent the different possible thermal environments likely to occur in the type of conditions for which the index is developed. Examples of empirical indices are the "effective temperature" developed by Houghten and Yaglou in 1923 and the "predicted four hour sweat" rate by MacArdle [2].

Empirical indices can be of great use to predict thermal sensations but they do not take into account the physiological mechanisms which produce these sensations. They require extensive experimental studies with subjects before results can be used.

More recently, rational models of human thermal response have been developed based on the heat transfer mechanisms of the human body with its surroundings. They allow to evaluate comfort without requiring subjects testing. Although other rational indices exist, in the following, the Pierce (Gagge) two-node model, the Kansas State University model (KSU) and the Fanger model are discussed briefly.

PMV and PPD Indices (Fanger Model)

The Fanger comfort model [3,4] is based on the "Comfort Equation" published in 1967 by Fanger. This equation is obtained by writing the heat balance over a human body taking into account the basic heat transfer modes of internal heat production (metabolism), work, heat losses by evaporation and respiration, conduction through clothing, radiation and convection. For the body to be neutral, the heat that is produced or gained must be equal to the losses with no heat stored. When this condition is not met, discomfort occurs. Negative heat losses indicate cold sensation.

The energy balance for a human being is written as follows at steady state:

$$S = M \pm W \pm R \pm C \pm K - E - RES$$

where S : Heat storage or load
M : Metabolism
W : External work
R : Radiation heat exchange
C : Convection heat exchange
K : Conduction heat exchange (through clothing)
E : Heat loss from sweat evaporation
RES : Heat exchange by respiration

There are separate equations for each term in the heat balance equation and each term can be calculated separately in terms of air temperature (Ta), mean radiant temperature (Tmr), air velocity (Va), vapour pressure or relative humidity (Pa or Rh), metabolic activity (M) and clothing insulation (Icl). Work (W) is a measure of the frictional losses due to body movements and can usually be taken as zero. Other variables are necessary to take into account the features particular to the human body (shape and clothed surface) and can be obtained directly from graphs.

In the heat balance equation, the expressions for surface temperature of the skin and the rate of body sweating had to be established. It was Fanger who quantified their linear relationship with metabolic activity using tests on individuals in tests chambers. This made possible the calculation of the heat balance equation using only six variables. When all six variables (Ta,Tr,Va,Pa,M,Icl) are known, they can be introduced in the comfort equation to calculate the magnitude of the thermal unbalance.

The magnitude of the thermal unbalance is of no practical use if it does not reflect the relative effect of a thermal unbalance on comfort which is a subjective evaluation In order to accomplish this, a series of tests on thermal preferences of subjects was made in controlled conditions.

The index thus derived is the predicted mean vote (PMV). It is expressed as follows [3]:

$$PMV=(.303e^{-0.036M}+.0276)*S$$

PMV gives value over the following thermal sensation scale:

PMV	Sensation
+3	hot
+2	warm
+1	slightly warm
0	neutral
-1	slightly cool
-2	cool
-3	cold

The PMV index provides a rating for thermal sensation but is not sufficient in itself to assess indoor environments. A given value of PMV does not explicitly indicate how most people will find indoor conditions except at the extremes values (cold or hot). The PMV index does not include the natural variations in thermal sensation found for different individuals within a group. Personal differences in thermal acceptance can be caused by factors such as age, sex, race or climatic adaptation.

Another index, the predicted percentage of dissatisfied (PPD), can be used to include these factors. The PPD uses as a criteria for comfort that the PMV should lie between +1 and -1 (slightly warm to slightly cool). Within these limits, serious discomfort does not occur.

The PPD is in fact a curve of the distribution of dissatisfied as a function of PMV. As can be seen in Table 2.1, even at PMV of zero, there will always be a 5% of dissatisfied. The best a heating system can do at any time is to satisfy 95% of the people present.

Table 2.1: PPD as a function of PMV

PMV	PPD Cold	PPD Warm	PPD Total
-2.0	76.4	0.0	76.4
-1.0	26.8	0.0	26.8
-0.5	9.9	0.4	10.0
0.0	2.5	2.5	5.0
0.5	0.4	9.8	10.2
1.0	0.0	26.4	26.4
2.0	0.0	75.7	75.7

For evaluating the overall thermal environment in a room, another index, the lowest possible percentage of dissatisfied (LPPD) index can be used [3]. The LPPD index is calculated using the average value of PMV calculated or measured at different points in a room. Then, the points PMV's are corrected by subtracting the room average PMV from their value and the corresponding PPD values at each points are then calculated. The average PPD for the room obtained from the corrected PMV values corresponds to the LPPD value for the room.

In the comfort equation, the six variables for comfort have a different importance in influencing comfort. For further discussion on the relative importance of each factor, refer to Fanger's Thermal Comfort [3].

Other Rational Indices

Let us mention here two other rational models which have been developed after Fanger's model. These are the Pierce (Gagge) Two Node Model and the Kansas State University Model (KSU). Other models exist that consider more variables and nodes to represent human heat exchange but the discussion of these other models is beyond the scope of the present study

The Pierce two node model [5] was proposed in 1970 by Winslow, Herrington and Gagge from the John B. Pierce Laboratory of New Haven. The model was developed to ameliorate the original effective temperature scale (ET) developed by Houghten and Yaglou in 1923 without repeating the long series of extensive test formerly done. The original ET overestimated the effect of humidity in cool and neutral conditions and underestimated its effect in warm conditions. Also, it failed to represent the effect of air velocity in hot-humid conditions.

The Pierce model represents the human body as consisting of two concentric compartments, the skin and the core. Each compartments having its own thermal characteristic. It incorporates the fact that skin temperature is a good indicator of thermal sensation and comfort in cold environments. At conditions where sweating occurs, skin wettedness is better for assessing discomfort.

The energy balance used by the Pierce model provides an expression to evaluate the core or skin temperature at any instant given initial conditions. The model also predicts the rate of weight loss and skin wettedness. The prediction of thermal sensation is done by converting the environment to a standard environment where a subject would have the same skin wettedness, skin temperature and heat transfer rate. The expression for core and skin temperature are given in integral form.

The KSU model (KSU) [5] was published in 1974 and improved in another version in 1977. The model is the same as the Pierce model except that the equations for sweat rate and blood flow are different. Also, it predicts the thermal sensation directly without the need to convert to a standard environment temperature. The thermal sensation equation is different whether the environment is cold (T_{SENS}-) or warm (T_{SENS}+).

The three rational models consider similar variables and equations in their prediction of the thermal behaviour of humans. For example, the Pierce model uses the same expressions as

Fanger for calculating respiratory losses. The expressions for radiative and convective losses are also the same except that they are combined to obtain the definition of the operative temperature

A comparison of the models made by Berglund [5] shows that the models agree in their prediction of thermal sensations when compared with Gagge and Rohles' subject studies However, the models agree better for conditions near thermal neutrality, and their predictions will diverge with respect to one another as the thermal conditions are more severe.

The field studies contrary to the chamber studies have shown that the neutral temperature is affected by both the general thermal experience of the subjects as well as by the outdoor temperature. The general thermal experience is related to the conditions in which the tests are performed, for example factory work, tropical climate, etc. Also, due to the different rating scale employed, results of field measurements have shown discrepancies between the predictions of the different models as mentioned by McIntyre [6]

Since we are interested in moderate thermal environments, the use of Fanger's model will not introduce significant errors in the assessment of thermal comfort Also, Fanger's model has gained more acceptance since it can be readily used by consulting tables while the other two require computers to calculate the equations which require numerical integration

There is no dispute however that rational indices have a greater accuracy than the direct and empirical indices for the amount of subjects testing they require Fanger's model has been incorporated into an equipment (PMV-meter) which measures the environmental parameters (Ta,Tr,Va) and gives the comfort temperature, PMV and PPD The measurement of comfort can be better assessed with a comfort meter since the human heat exchanges with the environment can be simulated directly.

The PMV, PPD and LPPD indices are therefore the necessary and sufficient tools to respectively predict people's thermal sensations, determine the predicted percentage of dissatisfied which is expected to result from these conditions and finally assess the thermal uniformity in rooms.

2.1.2 Local Thermal Discomfort

Local thermal discomfort is another important part of thermal comfort. It deals with specific comfort problems not necessarily incorporated in the comfort models and their related indices. The thermal neutrality on the PMV scale is not a complete guarantee of a persons comfort since the physical factors (Ta,Tmr,Va,Rh) can be combined in such a way that PMV is zero while in fact, the conditions for the subjects are uncomfortable due to local discomfort problems.

Most important problems of local discomfort include asymmetric thermal radiation, draught, vertical air temperature differences and warm or cold floors. This aspect of thermal comfort will be reviewed and most of the discussion presented here can be found in summary works by Olesen [7] and Fanger [8].

Asymmetric thermal radiation

Asymmetric thermal radiation is caused by surfaces which are either colder or warmer than the rest of an enclosure. It is usually caused by cold windows, badly insulated walls or heated ceilings or walls.

Asymmetric thermal radiation can be calculated using the difference between the radiant temperature on either side of a plane element [7]. Vertical asymmetry is calculated for a horizontal plane and horizontal asymmetry for a vertical plane element. The asymmetry is calculated using angle factors between planes or angle factors between human beings and planes.

Draught

Draught is an important cause of local thermal discomfort in ventilated spaces and especially during cooling periods [7]. Draught is defined as an undesirable local cooling of the human body. It occurs mostly in ventilated spaces with high air velocities or in spaces with high air infiltration.

Sensation to draught is dependant on air temperature, air velocity and its fluctuations. Studies have been done by Fanger and Pederson to establish the effect of constant and periodically fluctuating air movements on comfort [8]. These studies showed that constant air flows are more comfortable than fluctuating ones. However, the actual air fluctuations found in the field are stochastic in nature and not periodical as assumed in the study.

Vertical air temperature differences

The differences in air temperature between the floor and ceiling can be an important cause of local discomfort. Because of buoyancy, hot air tends to stay closer to the ceiling This effect is particularly undesirable since feet are the first to be sensitive to lower temperatures

The important levels for comfort with respect to vertical air temperature differences are the ankle level (0.1m) and the head level (1.1m) Vertical temperature differences between these levels can be up to 3°C for people performing sedentary work At this value there is a predicted 5% percentage of dissatisfied [7]

Warm or Cold Floors

Local discomfort can also be caused by too hot or too cold floors This is in fact a type of local discomfort more important when dealing with radiant heating systems Usually, for a relatively well controlled and insulated room, these problems will not arise

The above list of local discomfort factors now completes the list of important factors to be considered for a thermal comfort study and should be used with previous section on thermal comfort indices.

A few more remarks on some points which relate more specifically to field research should be made. Some of these aspects are discussed before finally ending the discussion on comfort indices and local discomfort.

There are a number of ways to establish thermal comfort and results are often dependant on the methods used. For example, the outdoor temperature experienced before a person enters a space can strongly affect the neutral temperature required for that person. This is partly due to the fact that there is a behaviourial adjustment made by the person to the prevailing conditions (e.g.: change of clothing, thermostat modification) These differences may have serious implications on the comfort criteria for design and control of HVAC systems.

Indices used to evaluate thermal comfort do not necessarily quantify the same aspects of a person's sensation. For example, using the method of direct determination according to Fanger shows no difference in preferred temperature with respect to age, race, sex, season or thermal experience. This is not the case for chamber studies [6]. The effect of the above mentioned criteria deserves more attention.

Additional studies remain to be done on the impact of thermal comfort on work performance and productivity [8]. The research in this area is not straightforward since the effect of the thermal environment is difficult to dissociate from other environmental factors (lighting, colours, indoor air quality). Also, the working environments can influence thermal sensations and this is why some differences are expected to occur between actual sensation of persons and the comfort levels predicted for them.

In the next section, a review is made of the current thermal comfort standards. In the analysis of the data, this section and the next will be used for comfort assessment.

2.2 Standards on Thermal Comfort

Because of the extensive studies done on thermal comfort, various associations have been able to set limits and guidelines for desirable indoor conditions. These rules and guidelines are intended for designers of HVAC systems and authorities to evaluate or predict thermal comfort in different situations.

The two most widely used and recognized standards on thermal comfort are the ISO DIS 7730 " Moderate Thermal Environment-Determination of the PMV and PPD Indices and Specifications of the Conditions for Thermal Comfort-1984" [9] and the ASHRAE 55-81 "Thermal Environment for Human Occupancy " [10] standard The requirements of each standard are listed in Table 2.2.

The ISO standard is based on the works of Fanger It uses the PMV and the PPD as a measure for design and analysis of global thermal conditions. Local discomfort is also treated for temperature asymmetry, temperature gradients, draught and cold floors The same points are also treated in the ASHRAE standard

The ASHRAE standard is based on the two node model of human thermoregulation developed by Gagge and the effective temperature It also includes the results of Fanger when the limits for operative temperature are set on the psychrometric chart

The desired values in terms of humidity range, air movements, temperature cycling and drifts are only implicit in ISO 7730 The correct PMV is assumed to be [illegible] for the comfort of those parameters. If the PMV is in the desired range when any of these variables is considered then the thermal climate is satisfactory.

The operative temperature ranges recommended by both standards are comparable. For the ASHRAE standard, it is between 19.5°C and 23.6°C for sedentary, winter conditions. For ISO it is between 20°C and 24°C for the same conditions.

The standards are often more conservative to insure that the thermal environments will be acceptable. Even though the maximum PPD is fixed for the ISO 7730, the combined effect of the local discomfort parameters guarantees only a minimum PPD of 20%.

Most of experimental setup was designed to accommodate the requirements of the standards, but some differences occurred since measurements were not limited only to the points recommended in the standards (occupied zone). In this way, both comfort and the physical behaviour of the systems could be considered in more detail.

Table 2.2: Summary of ASHRAE and ISO Specifications for Winter Conditions

Criteria	ISO [9]	ASHRAE [10]
Indoor Temperature	Such that PPD < 10% or -0.5 < PMV > 0.5 or 20.0 < To > 24.0	19.5 < To < 23.0, Tdp=16.7 20.2 < To < 24.6, Tdp=1.7 (comfort zone)
Relative Humidity	PPD < 10%	20% to 80% but within To comfort zone
Air Movement	average Va < 0.15 m/s	
Mean Radiant Temperature	PPD < 10%	Operative temperature kept in comfort zone
Temperature Cycling	Time weighted average of PMV and PPD is within limits for indoor temperature	For peak to peak variation > 1.1 °C, temperature change must be less than 2.2°C/hour
Temperature Ramps or Drifts	Same as temperature cycling	Rates must be less than 0.6°C/hour with To less than 0.6°C outside of comfort zone and for less than one hour
Vertical Temperature Gradients	Less than 3 °C between 0.1 m and 1.1 m levels	Less than 3 °C between 0.1 m and 1.7 levels
Radiant Asymmetry	Horizontal 10 °C maximum Vertical : 5 °C maximum	
Floor Temperature	from 19°C to 26°C up to 29°C for floor heating	from 18°C to 29°C

2.3 Previous Studies on Comparison of Heating Systems for Thermal Comfort

Previous studies which aim at comparing different systems for thermal comfort are different from this study for two reasons. The first reason is that most of them compared the systems in steady state conditions. Second, in most cases, the test facilities were "ideal" environments produced by environmental chambers and considered only one zone or room.

It is therefore possible to compare results with these studies in terms of the effect of transient conditions. The outdoor temperature varies over the normal range found in a heating season and there is an effect of all weather related parameters such as solar gains and infiltration rates. The effect of having adjacent zones is another of these factors which have a cyclical and repetitive nature on the enclosure.

Recent climatic chambers studies as well as field studies are listed in Table 2.3 and their results appear in Table 2.4. The measured parameters, the heating systems as well as the aim of the experiments and where the tests were performed are shown.

The first experiment was described by J. Hannay et al. [11] and was conducted in a climatic room at the University of Liege, Belgium. The aim of the experiment was to measure the conditions governing thermal comfort and relate them to energy consumption for different heating systems.

The second experiment was conducted by Olesen et. al in a climatic chamber at the Technical University of Denmark [12]. Again, the aim of the experiment was to compare thermal comfort and energy consumption in a room heated by different methods.

Table 2.3: Heater Comparison Studies

Study	Results
Hannay	-no major discomfort for systems (PPD<10%) -temperature distr. not influenced by infiltration -no radiation discomfort for well insulated surfaces -problems of compensation of window downdraft -classif. of systems not same for comfort and energy
Olesen	-all systems produced uniform environment (PPD<12%) -vertical temperature gradients less than standard -radiant temperature asymmetry less than 3.5 K -higher radiant asymmetry with higher infiltration -mean air velocities less than 0.2 m/s -subject response indicates comfort problems with radiator
Berglund	-thermal sensation of subjects similar for all the systems even though the systems produced different air and radiant temperatures thermal acceptability of 98% baseb.,88% forced air
Michaels	-forced air distribution is best with floor registers -problem of high outlet temperatures, low outlet velocity, high discharge levels (rad. & fan heaters) -isotherms plotted for the systems
Rohles	-different thermal sensation for similar temperature increases

Table 2.4: Results of Heater Comparison Studies

Author	Test Facilities	Aim	Systems	Measured Parameters
Hannay et al. (1978)	climatic chamber (4.8x3.5x2.7m)	thermal comf. and energy	panel radiator forced air	air/surf. temperatures resultant temperature air velocity, infiltration outdoor conditions
Olesen et al. (1980)	climatic chamber (4.8x2.7x2.4m)	thermal comfort	radiators,convectors ,forced air, heated floor and ceiling periphery heating	air and surface temperatures air velocity PMV, subject response infiltration outdoor conditions
Berglund et al. (1980)	climatic chamber (2.7x2.7x2.4m)	operative temperature control	heated floor and ceiling baseboards	air and surface temperatures operative temperature subject response
Michell and Biggs (1980)	actual dwellings	general temperature distribution	convectors forced air heated ceiling	air temperatures heater temperatures globe temperature outside temperatures
Rohles (1986)	climatic chamber (2.7x3.6x2.4)m	transient warming comfort	radiators forced air	room temperatures subject sensation

The experiment, performed by Berglund et al. [13], was different from the ones mentioned above in its preoccupation with the control system Its purpose was to show the effectiveness of using the operative temperature as a single input for to control the systems studied

The following experiment described was performed on heating systems installed in actual dwellings in Melbourne [14] The experiments concentrated mainly on the temperature distribution obtained from different heating systems as they were used by occupants For each system studied, temperature distributions were determined by recording temperatures through a vertical section in a room.

The last experiment, conducted in Japan by Rohles et al [15] used human subjects in measuring the comfort-producing effectiveness of space heaters during transient warming The tests were conducted on three different type of heaters, two of them produced natural convection and radiation while the third was with forced convection only For transient phenomena, the type of heating system was found to be an important factor and the satisfaction of people in transient warming is not dependant only on the rate of air temperature increase but also local discomfort considerations

All the studies described above have indicated that for local discomfort, the type of heating systems used is of significant importance On the whole, systems did not produce the same type of environment close to the main heat source For global or room comfort systems performance were similar

A few other important research aspects can be mentioned briefly before concluding The studies described above concentrated mainly on comparison of heating systems in controlled conditions and one zone only In addition to the factors considered in those studies some consideration must be given to the effect of inter zone convective heat exchange and direct solar radiation entering the building

For inter-zone heat transfer, the estimation of the adjacent room temperatures is necessary and depends on many factors. If the room is surrounded by interior zones only, it can be assumed that the temperatures everywhere will remain fairly constant. If some of the adjoining rooms are perimeter rooms, the problem is more complex. It is likely that important convective heat transfer will take place between different zones.

Inter-zone convective heat transfer research has been reviewed by Barakat [16]. The factors influencing this phenomena are the temperature difference between rooms and the pressurization caused by mechanical systems.

The influence of inter-zone convection is determinant in analyzing or predicting the thermal performance of buildings. The pure natural convection cases have been studied more extensively. Work in this area is still in progress to account for multi-zone configurations, special boundary conditions and combinations of mixed, natural and forced convection heat transfer [17].

Another important factor in determining the indoor conditions is solar radiations. Giaccone and Gianfranco have used a computer simulation to calculate thermal comfort maps to show the influence of the internal distribution of hourly radiation on thermal comfort [18]. Solar radiation can be considered as uniformly distributed on all room surfaces or, as is the case in nature, a light beam travelling within a room. The two methods will show great differences in their estimation of thermal comfort.

Inter-zone convective heat exchange and solar radiations are only some of the factors to be considered in a building and many more studies should be mentioned in the present context. For example, there is the combined effect of weather related parameters (humidity, wind velocity, solar radiations etc.), occupants' living habits (daily changes, activities performed etc.); seasonal changes in envelope properties (due to snow cover, ground freezing, thermal bridges), floor to floor conduction and convection; neighbouring buildings' influence and so on.

2.4 Summary of the Review

This review has provided necessary elements which can be used to analyze the problem of comparing three types of heating systems for thermal comfort and some examples of experiments done in the field.

The sections on human thermal comfort (2.1 and 2.2) will be used as a reference in chapter 7 to establish the performance of the systems for global comfort and local discomfort

The last section on previous experiments (2.3) will be used in chapters 4 to 6 to compare results with those of other studies Since few of the above studies had the exact same experimental conditions as this one, the results will be compared whenever possible

CHAPTER 3

3. TEST FACILITY AND INSTRUMENTATION

Although this section is mainly a description of the test facility and instrumentation, it includes some calculations and a certain amount of data analysis which was required for the determination of thermal loads and instrumentation accuracy. The aim is to describe where and by what means the data were collected and mainly to cover aspects which will be necessary in the following chapters.

There are five parts describing: the building's location and climate, the building's envelope properties (thermal loads and inter-zone heat transfer), the heating systems (specifications), the data acquisition system and finally, the instrumentation used for the tests.

3.1 Location and Climate

The three storey apartment building studied is located in the center east part of downtown Montreal (45°30'N, 73°35'W) on a north-south street in a residential block of similar buildings The buildings in the block are grouped in two rows separated by a small service alley (see Figure 3 1) Most buildings in this zone are similar to the one studied in size (total of approximately 300m^2 of habitable space), the number of floors and the presence of two side walls common with neighbouring buildings.

The climate surrounding the testing site is dominated by weather conditions characteristic of the eastern part of Canada and by the micro-climate due to the urban location of the building

The winter climate of the Montreal region is characterized by average daily temperatures of -5.7°C, -8.7°C and -7.5°C for the months of December, January and February respectively for data collected over a thirty year period at a downtown weather station [19] Extreme lows of 33°C and highs around 10°C were also observed for these months It is also not uncommon to find two consecutive days with outdoor temperatures oscillating near these extremes

The 97.5% design outdoor temperature recommended by ASHRAE is 23°C [1] and the number of degree days is 4471 [20] The concentration of buildings near the site will influence these general statistics and slightly warmer temperatures will be observed due to the presence of heat sources from neighbouring buildings

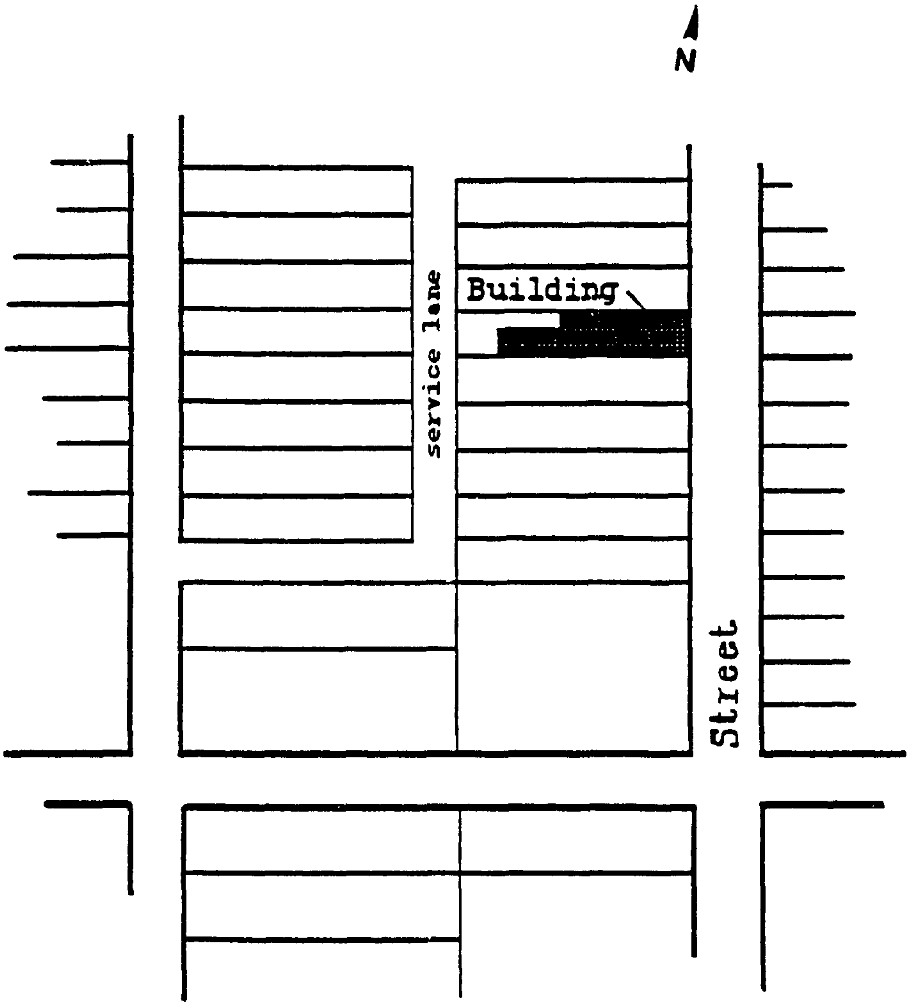


Figure 3.1: General Building Location (no scale)

3.2 Building Layout and Envelope Properties

The envelope of the building had been improved before the beginning of the test period since the original building dated back to the beginning of the century and needed some improvements. Most of the renovations consisted in improving the insulation levels in order to bring them to modern standards and some modifications were meant to improve the layout of the internal divisions.

The renovations were carried in a manner considered to be typical of average current renovation practices in the sense that the changes in the envelope properties did not aim at creating a model building with state of the art techniques

The elevation and floor plans appear in Figure 3 2 The building has a narrow facade (7 6m) wide, and in depth it occupies most of the 30m deep land with a 21m depth The first and second floors' plans are shown with the locations of the test rooms used for this study The second floor was similar to the third floor in terms of room locations The test rooms are referred here as "comfort rooms" Their exact dimensions are described later in the instrumentation section

Rooms are located mainly at the front and on the north side of the building with the living room and kitchen located in a common space in the center Most rooms were completely furnished The basement is a partially excavated unfinished space with height varying from 1 to 2 meters

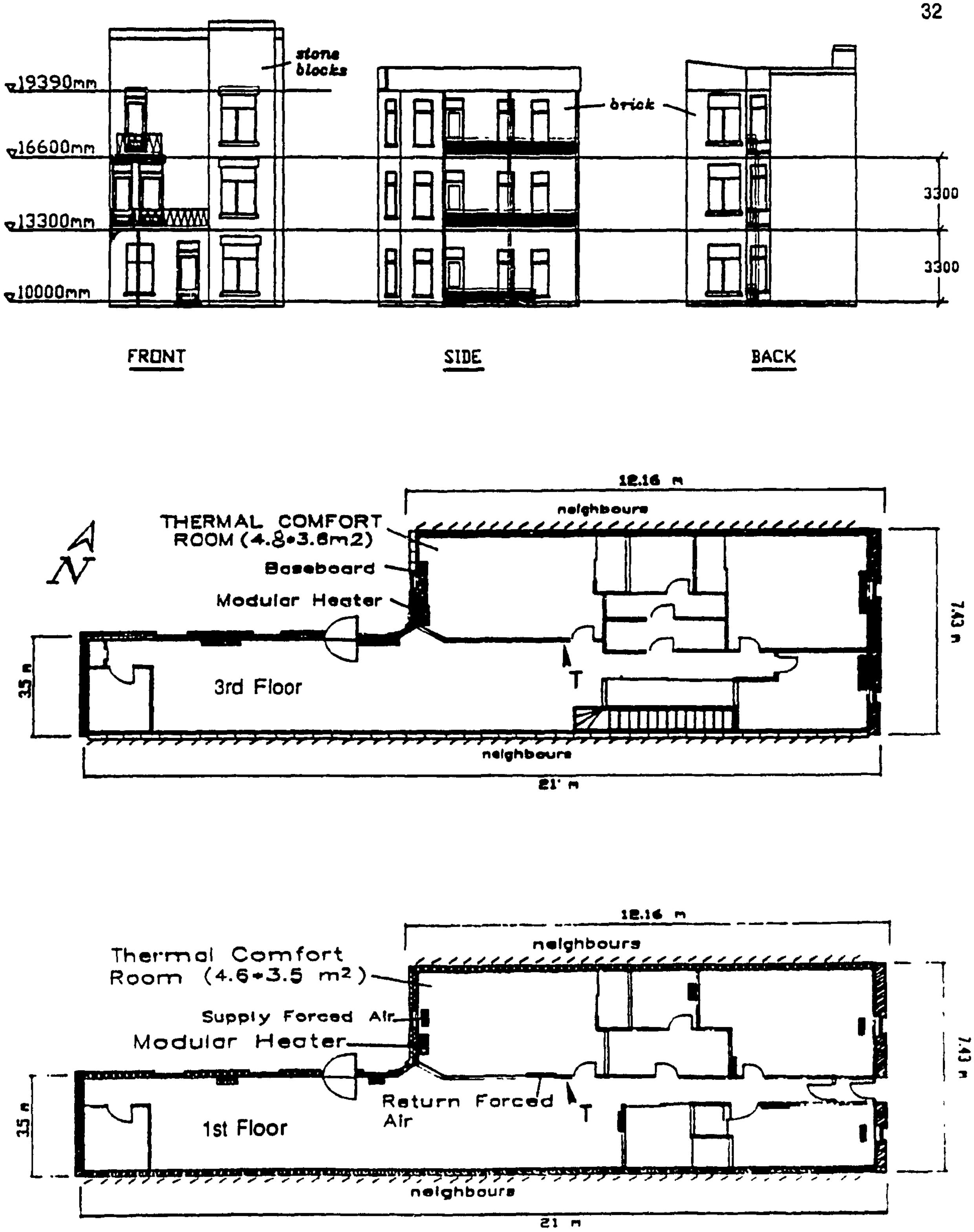


Figure 3.2: Building Elevation and Floor Plans, First and Third Floors

A general summary of R-values for the different envelope components surrounding the test rooms appears in Table 3.1. A percentage of error indicates and reflects the differences between the planned and built work, and the properties of old materials which remaining from the original construction. Refer to appendix A for a more detailed description of the surfaces and their U values calculations

The relative location and name of the envelope components surrounding the "thermal comfort rooms" are shown in Figure 3.3. The names for the surfaces will be used consistently in this study

The R-values of the envelope can be compared with some values suggested for cold climates (5000 to 6500 degree days) by the Canadian Mortgage and Housing Corporation [20]. Reference 32 suggests minimum RSI values between 2.8 and 3.0 for exterior walls and in our case, values between 2.2 and 2.8 were calculated. For roofs, minimum values between 4.6 and 5.0 are recommended and values between 4.8 and 5.8 were calculated. Therefore these two important exterior envelope components can be said to be at relatively average levels of insulation as mentioned earlier

Table 3.1 Summary of Surfaces U values

Surface	R value RSI	Estimated Error (%)
Exterior Wall	2.445	10
Interior Wall	0.403	8
Back Wall	0.403	8
Neighbour Wall	1.454	6
Floor (1st)	0.678	5
Floor (2 3)	1.675	25
Ceiling (3rd)	5.376	8
Window (glass	0.441	5

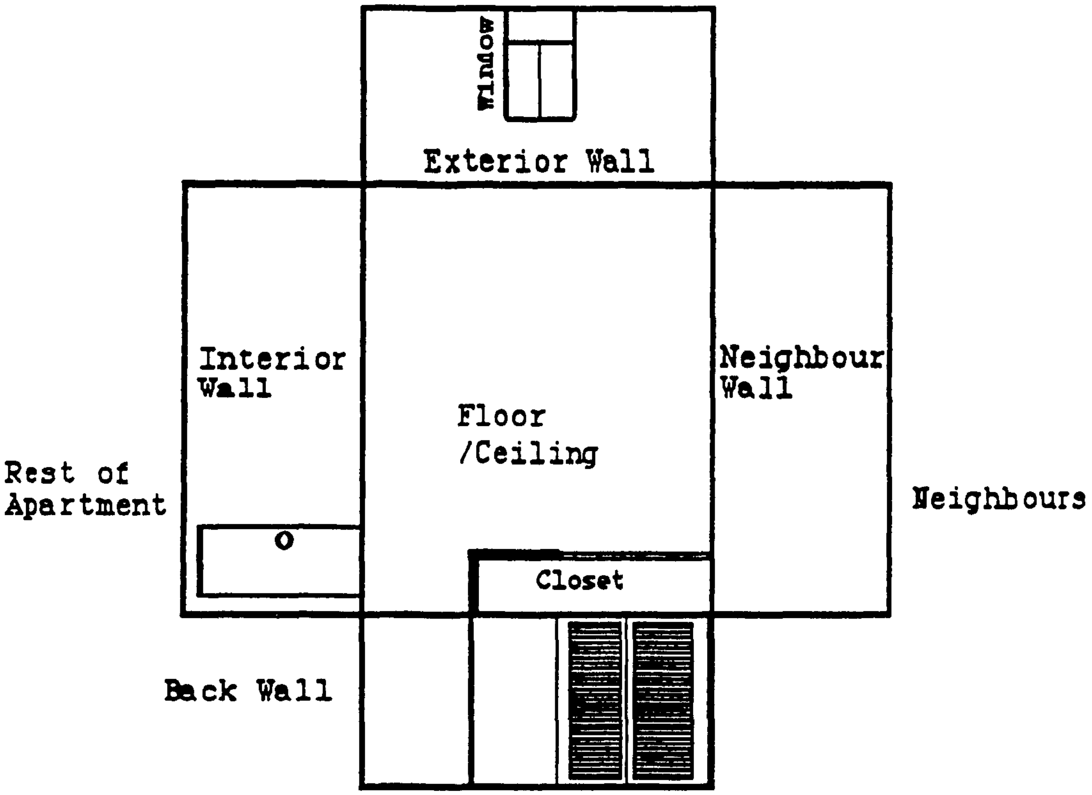


Figure 3.3: Reference Names for Thermal Comfort Rooms' Surfaces

Based on the envelope's properties, the thermal loads were calculated for the comfort rooms only. The loads were calculated to compare the first and third floor and to evaluate how the heating systems were sized with respect to demand

Results from thermal loads calculated for the two comfort rooms appear in Table 3 2 These calculations are not meant to be used as exact reference Room loads for the third floor room are approximately 40% higher than on the first floor Note that infiltration rates for each floor are based on average results from tests made by another research group using tracer gas method [21] The calculation at the design outdoor temperature on the first floor is different from the calculations at the other outdoor temperatures (table 3 2) since for design conditions a 2°C difference in temperature between room and basement was taken Standard assumptions have been made

Table 3 2 Conduction and Infiltration (sensible) Rooms Load Calculations

Outdoor Temperature (°C)	Room Load (Watts)	
	1st Floor	3rd Floor
21	0	0
11	190	310
1	380	630
-11	570	950
-19	760	1270
-29	950	1582
-23 (design)*	964	1424

* Includes infiltration (sensible) and conduction losses Infiltration calc based on air change per hour (0 8 ach)* for the first floor and 1 0 ach* third floor Indoor temperatures at 21°C Conduction through exterior surfaces only Design condition on first floor includes 2°C difference between room and basement temperatures

Note that room and floor losses are different being not calculated in the same way Results for floor loads calculated by another group involved in the project showed that the total design load on the first floor was 40% higher than on the third floor [22]. R-values obtained by the two groups were similar. However, the other group considered the basement and the first floor to be one zone when floor loads were calculated.

To calculate the floor loads on the first floor, the basement losses must be considered. They are not considered when room loads are calculated since the basement is maintained at a high temperature (close to 20°C) by the furnace of the forced air system.

Approximate calculations of the magnitude of heat transfer with adjacent zones by conduction was estimated and the results are shown in Table 3.3. The ratio of all the heat losses of a thermal comfort room to the adjacent zones over the total room losses of the room was calculated for each floor.

The percent of the load which could be attributed to the heat exchange by conduction between the comfort rooms and the other interior zones can be as high as 20%. This maximum percentage is less important for the third floor (10%) compared to the first floor (20%) and, at low outdoor temperatures the percentage of heat transferred between the rooms and the interior zones is smaller (4% to 9%).

If these calculations are used to represent the actual situation occurring in the building, then inter-zone heat transfer may be said to have a significant effect on rooms thermal balance. This influence is sufficient to bring room loads to similar values on both the first and third floor. The lower temperature of the basement can, for example, augment losses from the first floor room to values similar to those for the third floor room. In this case, the difference between loads on the two floors would be less than the 40% value mentioned above.

Table 3.3: Estimated Percent of Inter-zone Losses for 1°C Difference Between Room and Surrounding Internal Zones

Outdoor Temperature (°C)	(Interior Zone Losses)* / (Total Room Losses) %	
	1st Floor	3rd Floor
21	100	100
11	33	18
1	20	10
-11	14	7
-19	10	5
-29	9	4

* Interior zones around the rooms were assumed to be at 20°C commpared to room at 21°C Interior zones are remaining of the apartment second floor neighbours and the basement for the first floor room

For the case of an actual building establishing the complete heat balance is a very difficult task and in the above, simplified calculations were made for heat exchanges in order to obtain a relative value for the loads and the effect of inter zone conduction It was found that room loads for the third floor can be up to 40% higher, but with inter-zone conduction this difference can be smaller The calculations were necessary to later explain some of the differences that may occur between floors when systems are compared on different floors

3.3 Heating Systems

As mentioned earlier in terms of the mechanical components of the building there were three different types of heating systems studied forced air hydronic and modular systems In the following, the technical characteristics of each heating systems studied are described briefly and these specifications are related to the load calculations of the previous section. The thermostat's location for the forced air and the hydronic systems is discussed after the specifications for the three systems.

The systems were installed on different floors of the building as shown in Table 3.4. Note that two different heating systems could be run simultaneously on different floors.

Table 3.4: Floor Location of Heating Systems

Floor	System	Location
1	forced air	whole floor
1	modular 1st	thermal comfort room
2	modular 2nd	whole floor
3	hydronic	whole floor
3	modular 3rd	thermal comfort room

Forced Air

The first floor was equipped with a conventional forced air system with natural gas as the burning fuel. The brand name was the "Coleman Up-Flow (no. 2960-655) High Efficiency Condensing Furnace". The manufacturer's specifications are shown in Table 3.5.

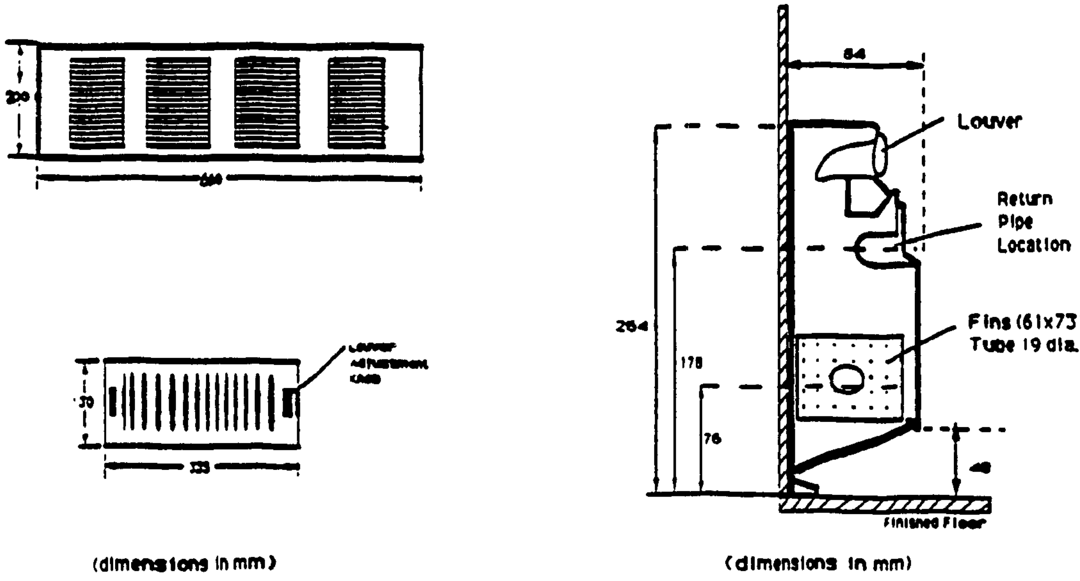
The furnace was installed in the crawl space of the basement. The distribution ducts passed under the first floor with six floor supply outlets having their surface parallel to the floor and five return grilles located at the bottom of the walls of the rooms. The dimensions of the supply and return grilles installed in the comfort room appear in Figure 3.4-a. It had its return air recirculated without outside air added.

Table 3.5: Specifications for Coleman Forced Air Furnace

Input	: 19045 W
Output	16848 W
Temperature rise	14-31°C
Design max. outlet temperature	66°C
Max external static pressure	0.12 KPa
Blower force	248 W
Thermostat	external

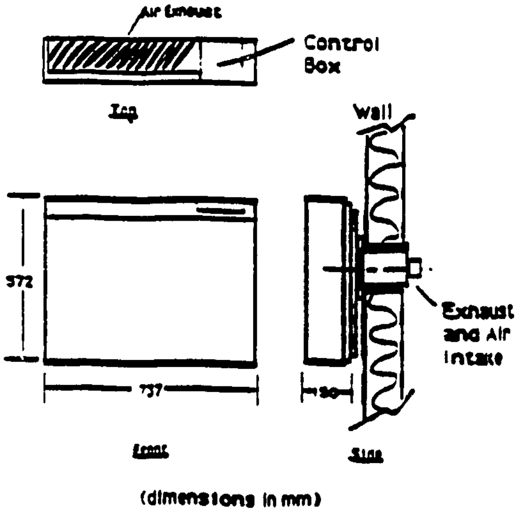
The forced air system was equipped with both a humidity control and electronic filter. The humidity in terms of percent was set on a dial located next to the thermostat. For the duration of the tests, the relative humidity dial was kept at 40%.

The supply grille of the forced air system could be controlled using the adjustment a control knob on the front of the grille. This knob was left in a full open position for the tests.



a) Forced Air

b) Hydronic



c) Modular

Figure 3.4: Heat Distribution Components of the Systems Studied

The maximum power output of the forced air system can be roughly estimated based on the manufacturer ratings. Considering the maximum total output of the forced air furnace to be divided equally between all the supply registers on the floor, an average maximum output of 2800W per register is obtained. Taking for example 15% duct losses, the output would be close to 2380W, and this is more than 40% required at the design load (964W at design outdoor temperature of -23°C).

Hydronic System

On the third floor a water boiler and a pump delivered hot water to baseboards in each zones using a two pipe reverse return system The boiler served for both heating and domestic hot water with control priority to heating

In the course of the project, two different boilers were installed with the rooms baseboards remaining the same The brand name of the systems were "Glow Core, Unicore Integrated Heating system (model UF 060)" and the "Celtic Hydro Therm (model 75FF) Wall Mount Boiler" The specifications for these two boilers are shown in Tables 3 6 and 3 7, the specifications for output compare with those of the forced air system The venting of the flues is made through an insulated pipe in the basement leading directly to the outside through the exterior wall at the back of the building

Table 3 6 Specifications for Glow-Core Unicore

Input	17 580 W
Output	15 646 W
Pump capacity	0 15 l/s
Ignition	piezzo electric/pilot

Table 3.7: Specifications for Celtic Hydro-Therm

Input	: 18 460 W
Output	: 15 880 W
Pump capacity	: 0.43 l/s
Ignition	: piezzo-electric/pilot

The baseboards used in the building were from the Mark-Hot company and were standard types of baseboards with finned tubes. Some were equipped with hydrostatic valves for individual room control. The valves could be used to control the amount of water going through the finned tube section. The heating for a room could be set individually using the valve's dial on the front of the baseboard which was scaled between 0 and 5. This valve was left at maximum dial position when the hydronic system was tested.

A cross section of the radiator installed in the comfort room is shown in Figure 3.4-b. Note that the return pipe for the hot water is located above the fins of the baseboard which is an unusual configuration. This means that the water cannot be completely prevented from entering the assembly and a certain amount of heat will always come from the radiator's return pipe even with the valves completely closed. Therefore, the baseboard of the system had to be insulated when the modular system was tested in the room.

The manufacturer's specifications for the baseboards were used to calculate output ratings for the baseboard installed in the comfort room and are listed in Table 3.8. Note that the calculations consider the heat exchange rated for air entering the lower part of the baseboard unit at 18°C. This is a value slightly lower than the value of 19°C measured in the field. Also, two values of flow rates are listed. The flow rates and water temperatures measured by the other team on heating efficiency were found to be at 0.1 l/s average (1.5 gpm) and 100°C maximum respectively.

Table 3.8: Specifications for Mark-Hot Radiator (comfort room 3rd floor)

	Output (W) (Entering Air at 18°C)	
	Water Temperature (°C)	
Flow (gpm)	75	100
0.06	700	1200
0.22	760	1300

From the output ratings, the baseboard in the comfort room is found to be slightly undersized with respect to the design load. The calculated load of 1424W is slightly higher than the maximum found in Table 3.8 for higher flow rates and water temperatures than those measured in the field.

Modular System

The modular systems use natural gas as the burning fuel They are called modular because each heating unit is installed in individual zones and the zones can therefore be controlled separately. Each system must be located against an exterior wall because of the direct vent to the outside (see Figure 3 4 c)

The selection of the temperature for a zone is done by adjusting manually a knob on the front of the system based on a numeric scale between 0 and 9 There are no temperature indicators in the space, the user's adjustment is relative to the feeling of warmth or cold Since the system has a temperature sensor, it is supposed to maintain the temperature at the time the adjustment is made

There were two types of modular systems installed in the building The system installed on the second floor was an European model (Auer MV 130) while in the rest of the building a

Canadian model called (Dettson CGM) was installed. The european system was designed to work as a radiator (high surface temperature) while the canadian model was designed as a convector (low surface temperature and increased heat exchange surface area). Only the Dettson model was tested in the course of the present study and its specifications appear in Table 3.9.

Table 3.9: Specifications for Dettson CGM

Input	: 3370 W
Output	: 2650 W
Regulation	: modulated for 0% or between 30% and 100%
Thermostat	: incorporated to system
Ignition	: piezzo-electric/pilot

From its specifications, the modular system is more than sufficiently powerful to satisfy the design load calculated for the first and third floor rooms. The 2650 W output is 2.7 and 1.9 times higher than the calculated room design loads for the first floor and third floor respectively.

Thermostat and Cycling Operation

The three systems varied with respect to their thermostat and their cycling operation. The thermostat control for the forced air and hydronic systems were wall mounted outside of the comfort rooms while the control for the modular system was on the unit. The cycling of both the forced air and the hydronic systems varied between on or off (no air flow or no water flow during off cycle period) while the modular system's output could be null or varied between 30% and 100% of the maximum.

The thermostat for both the forced air and the hydronic system was a Honeywell Chronotherm with programmable night set-back. The thermostats were installed in the living room area ("interior wall" of the thermal comfort rooms, see Figure 3.2) at a height of 1.5 meters

During the course of the study, these thermostats were found to be reliable based on their reaction to sudden changes in the set temperature When the temperature setting was varied, the thermostats responded correctly by turning the systems on or off with respect to the temperature read at the front of the thermostat (actual room temperature for the thermostat) Short delays lasting around 10 second preceded the actual turning on or off of the systems

The control knob for the modular system was found to be slightly difficult to use but after some experiments they were more reliable The problems occurred mainly at the beginning of the season when the outdoor temperatures varied widely and it was difficult to find the right knob adjustment Finally, the knob setting was selected to be around 8 or 80% for both floors and for this setting, room temperatures were maintained around 21°C

3.4 Data Acquisition System

The interesting aspect of the present study was the possibility to monitor a large number of data points over long periods of time To achieve this, an elaborate data acquisition system (DAS) consisting of different hardware and software components was developed and used

A room on the second floor was selected to become the data acquisition room to receive the input from the sensors' cables of the first and third floor (see Figure 3 5) of the building The DAS room contained three hardware components a programmable data acquisition system computer (brand name SAFE 8000) a personal computer and a communication modem

The microcomputer was used to program the DAS for preliminary calibration and to start and stop the data acquisition program

The SAFE 8000 was the main element in the system. All the inputs from the sensors were connected to it. Its program was capable of recording sensors' output at regular ana variable intervals.

The modem was responsible for periodically sending data for storage to a VAX 11/785 mainframe computer located at the CBS. The frequency of information transferred depended on the time rate at which information was accumulated. While the information was transferred (a process which may require up to 15 minutes), new data could still be gathered.

The information stored in a database could be accessed using either terminals or personal computers linked to the VAX. Due to the large amount of information collected (accumulated memory size of 86 megabytes for the whole project) and the specific aspect of data to be analyzed, most of the research was based on home-made softwares. These softwares developed by the author performed among other functions the instruments location listings, statistical analysis, graphical representations and batch processing. A brief description of some of these softwares appears in appendix B.

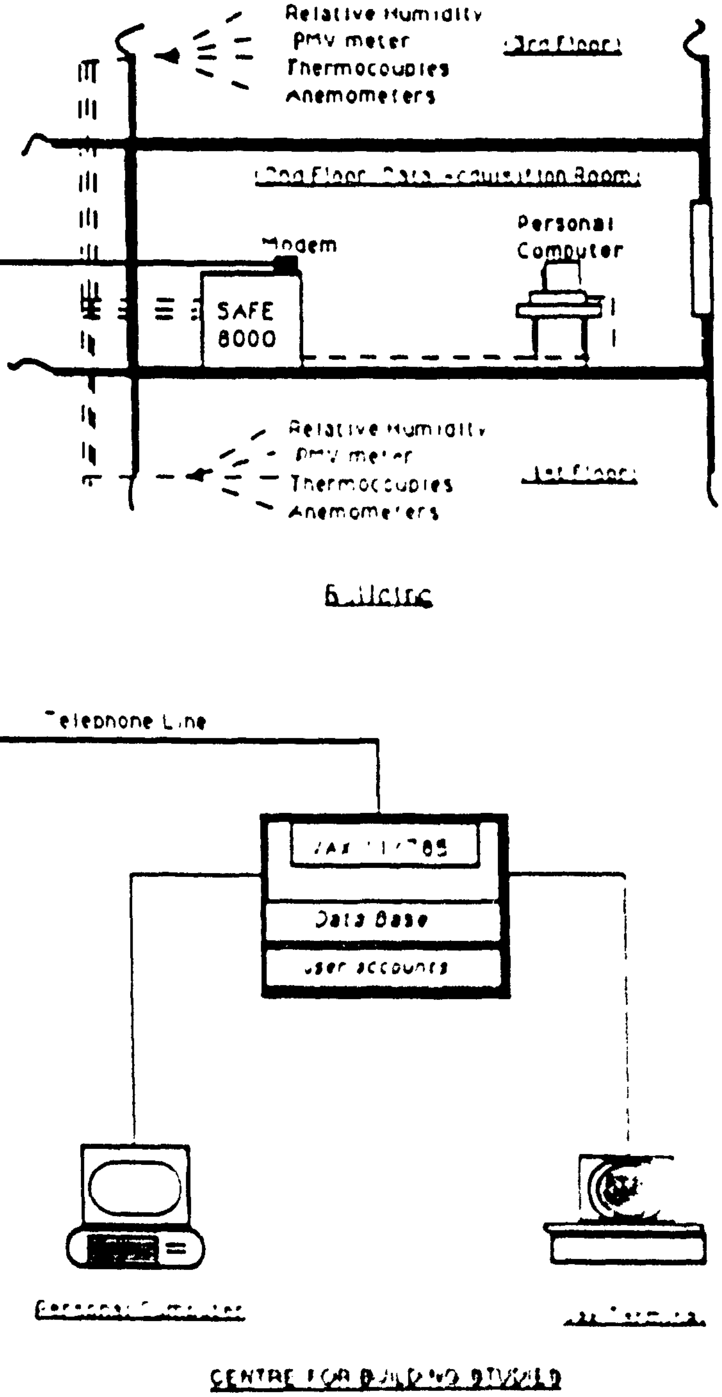


Figure 3.5 Data Acquisition System Components and Their Physical Locations

3.5 Instrumentation

The instruments used for the tests can be classified in two groups with respect to their sensors.

The first category is the instruments from which a direct output was measured. In this category are the thermocouples, the globe thermometers, the anemometers (air velocity) and hygrometers (relative humidity).

The second class of instruments is stand-alone equipments which are instruments equipped with a logic circuit able to processes the input value from sensors and output a numerical value for the measured parameters. In this category are a climate analyzer and a PMV meter.

3.5.1 Thermocouples

Around eighty thermocouples were installed in each of the thermal comfort rooms. The three main locations where thermocouples were installed were 1) the interior and exterior room surfaces, 2) a three dimensional grid in the center of the rooms and 3) in other zones in and out of the building. Their exact locations in the thermal comfort rooms are shown in Figures 3.6 to 3.9. The layout has been modified on certain occasions in order to vary the observation points from test to test or for special experiments.

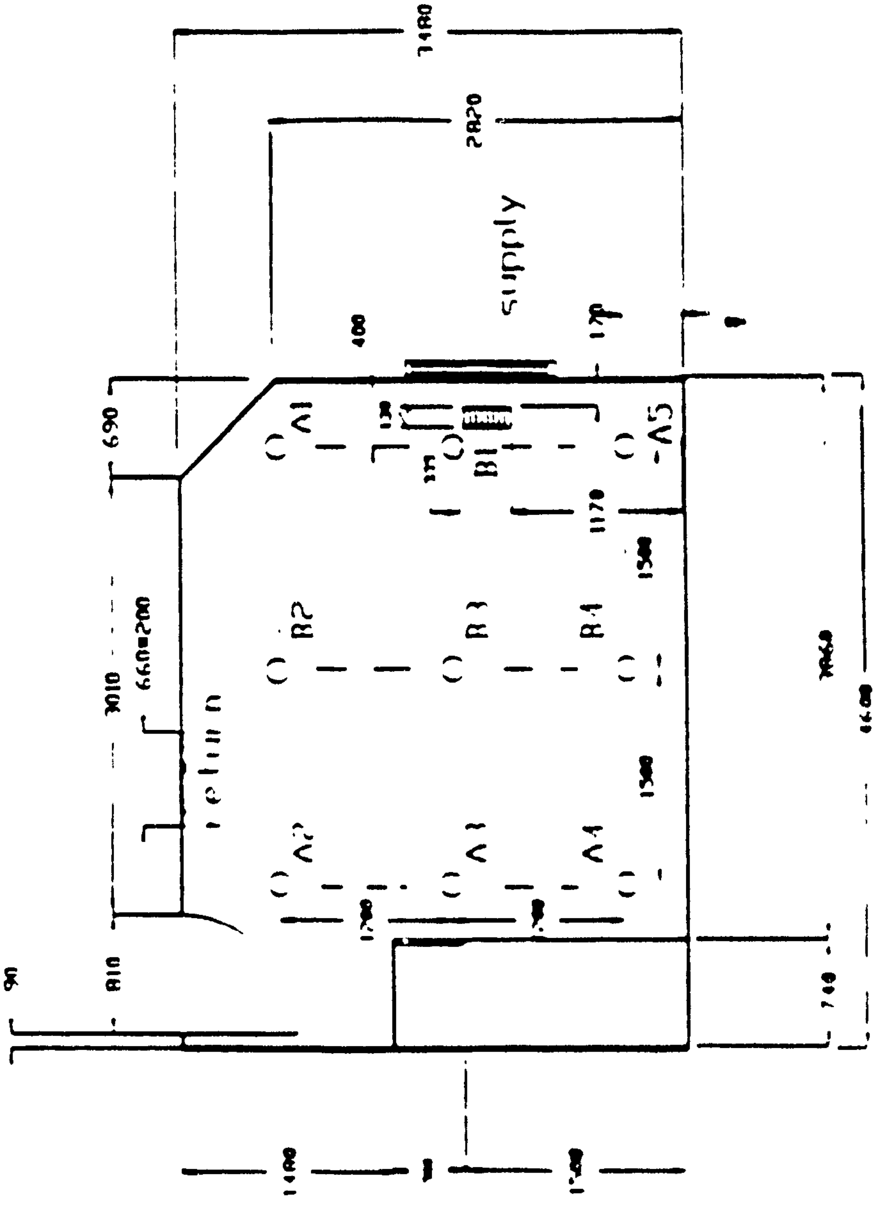


Figure 3.6 Plans of Thermal Comfort Room and Thermocouple Grid Location 1st Floor

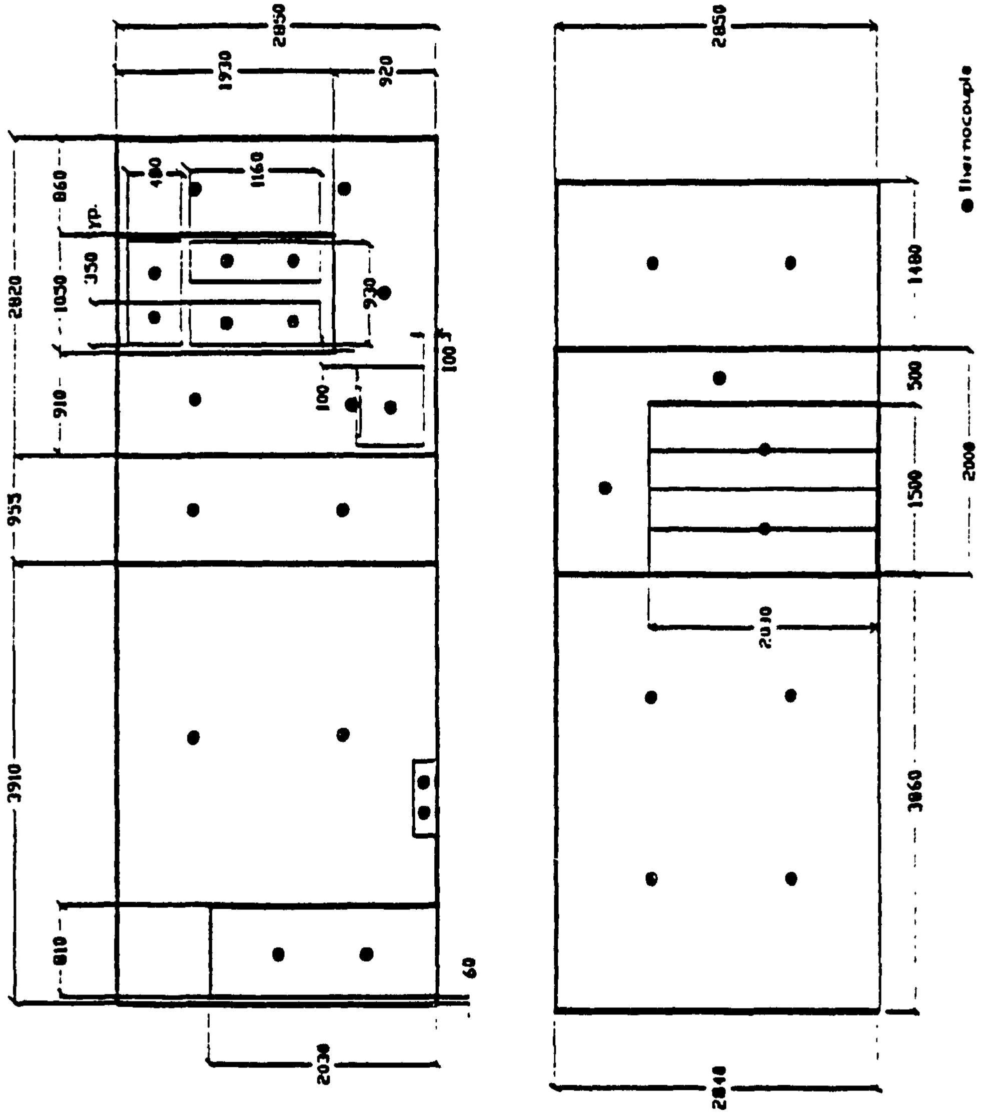


Figure 3.7: Thermocouple Locations on Surfaces, 1st Floor

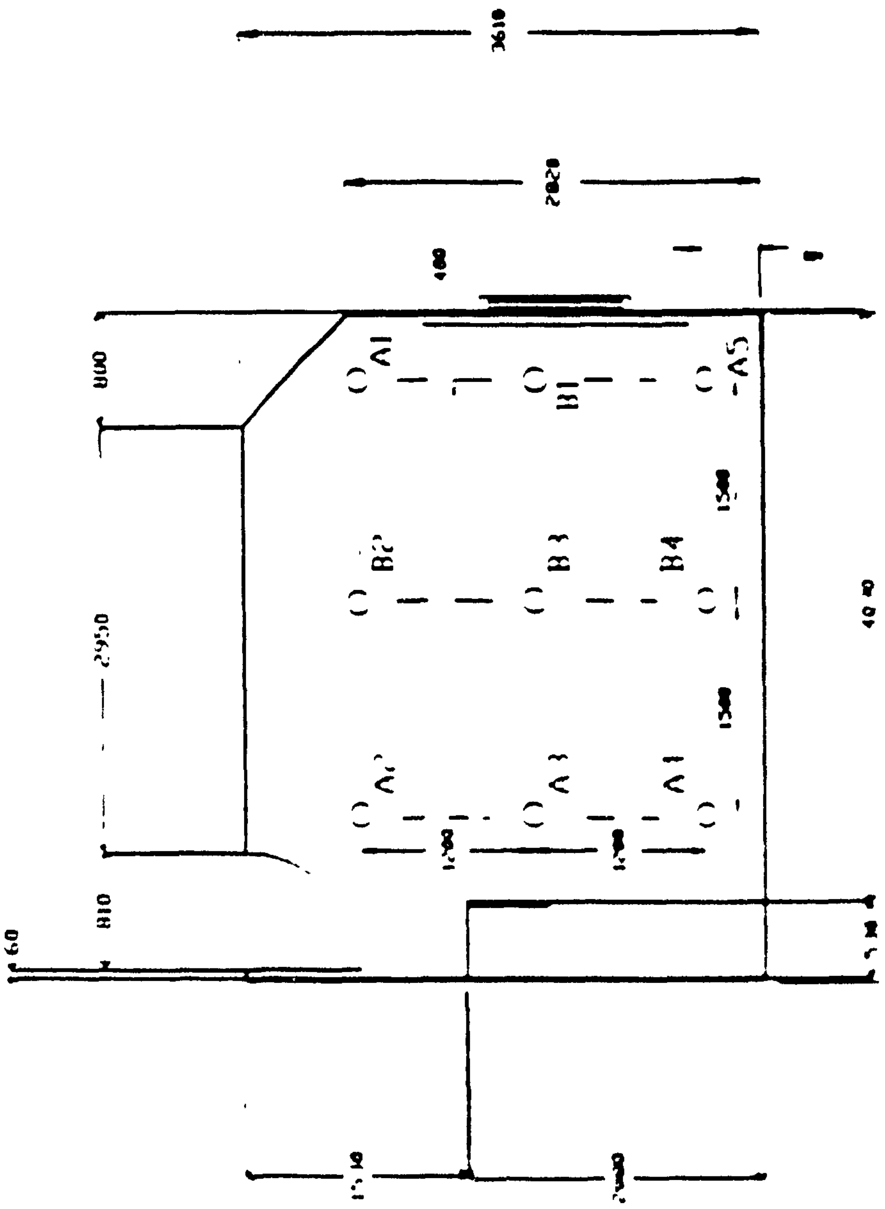


Figure 3.8 Plans of Thermal Comfort Room and Thermocouple Grid Location 3rd Floor

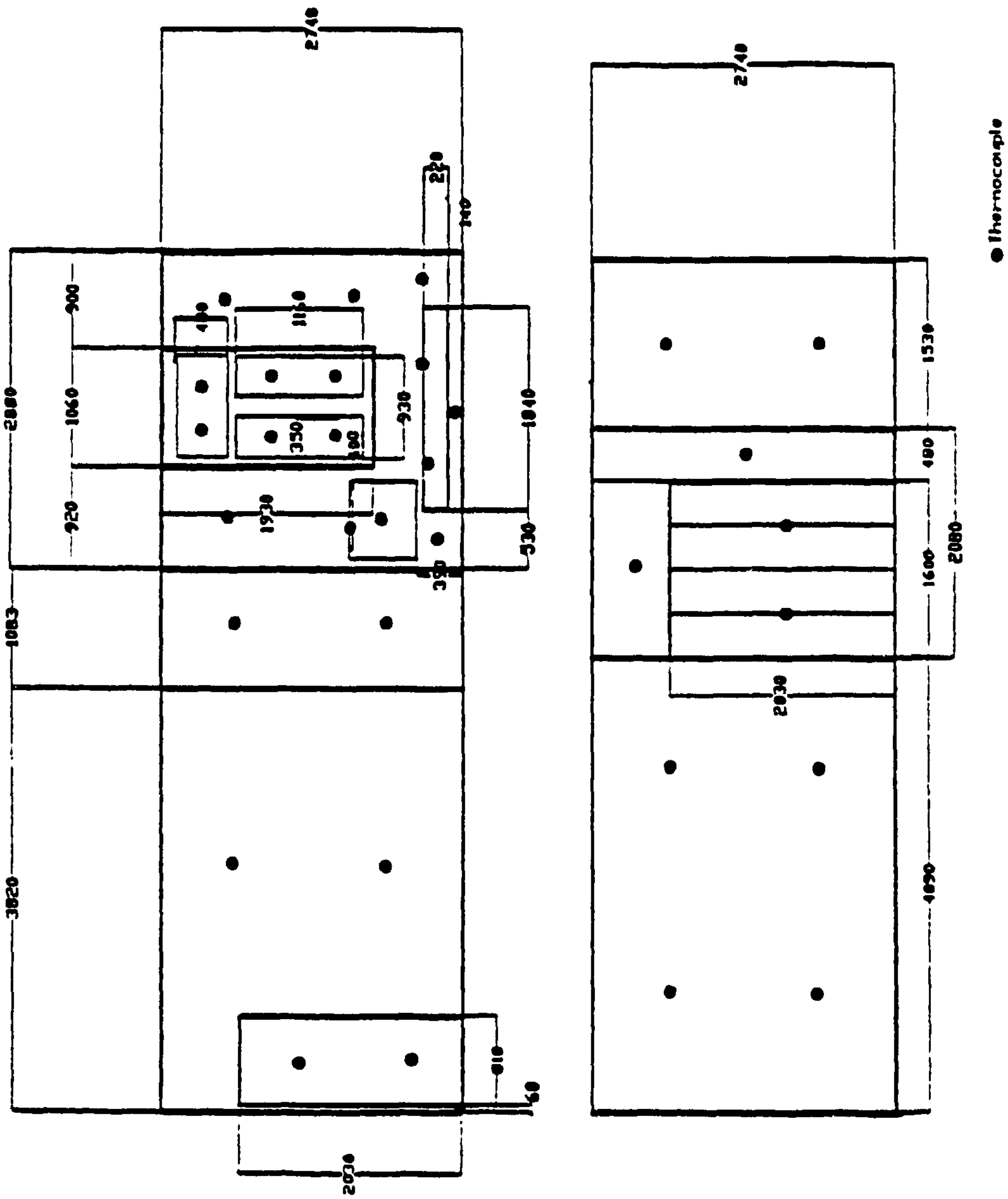


Figure 3.9: Thermocouple Locations on Surfaces, 3rd Floor

For the grid, two types of posts were installed (identified as type A and B). Type A had three thermocouples at heights of 0.8, 1.6 and 2.4 meters above the floor while type B had six thermocouples at heights of 0.1, 0.6, 0.8, 1.1, 1.6 and 2.4 meters. The 0.8, 1.6 and 2.4m levels were selected to divide the space between the floor and the ceiling in three equal portions while the 0.1, 0.6 and 1.1m levels were related to the specifications of the standards on thermal comfort. The grids were identical on both floors.

Locations of the thermocouples on surfaces had been selected to geometrically divide the surfaces into the largest possible number of similar surfaces. The thermocouples located on the surface of the heaters were located such that the surface and air temperatures of the systems could be measured.

Thermocouples were also located on the exterior surface of the window, on the exterior surface of the interior wall, and on the outside of the window. The thermocouple for outside temperatures were shielded with an aluminium cylinder to protect against direct solar radiations. Earlier measurements had shown the importance of shielding the exterior thermocouple since direct solar radiation could raise measured outdoor temperatures by up to 10°C compared to actual outdoor air temperatures. Also, surfaces' thermocouples were glued using a thermally conducting glue.

The thermocouples installed at the building were standard T type with manufacturer tolerance rated as ± 1°C. Experience with using aluminium cylinders to shield the tip of the thermocouples being unsuccessful, the thermocouples were left bare. Tests showed that the shielding methods available were not adequate thus this practice was abandoned.

All the thermocouples were read through a dedicated thermocouple channel input on the SAFE which was either multiplexed or simple. Around 80% of the thermocouples were going through a multiplexer (built from a series of electronic chips controlled by the SAFE), with one multiplexer per floor. In order to further ascertain the reliability of the thermocouples readings, two simple tests were made.

The first test was for linearity and consisted in using a thermocouple calibrator to input known values into the SAFE which could be compared with the SAFE's output readings. The response of the SAFE to fifteen input temperatures between 0 and 40.0°C was tested three times for one straight thermocouple input and four times for each of the two multiplexer's inputs.

The test for repeatability consisted in comparing the temperatures obtained from two thermocouples located side by side. This was done for more than 50 measurements. Repeatability is a measure of the performance an instrument to consistently provide the same output for the same measured input. The repeatability, R, was calculated using [23]:

$$Re = 0.707 * t * S_d \qquad (3.1)$$

where Re : repeatability of instrument

S_d : standard deviation of the difference between the measurements of two similar instruments measuring at the same time for n observations

T : value of t distribution for sample size n

The results for the linearity tests are listed in Table 3.10. The values for the slope and the intercept of the relationship between calibrator input and readings are shown for the three types of input channels. The results show that the input/output relationship is that of a straight line with a slope of 1 for all the cases. The major differences occur in the value of the intercept.

Table 3.10: Results for Thermocouples Linearity Test

Input Name	Response (°C/°C)	Intercept (°C)
Thermocouple	1.00	1.9
Multiplexer	0.97	3 5
Multiplexer	0.98	-0 20

Calibrator Accuracy = ± 0.25°C

For the different inputs, the results indicate that the zero values may differ by up to 3 7°C from one type of input to another. Although this difference is large, the test itself includes an error due to the calibrator (manufacturer rating of ± 0 25°C), its link with the different inputs tested and its stability in providing temperature outputs which was found to vary significantly in time

Based on this test, it was established that for a given input, the linear relationship guarantees a good response to temperature fluctuations, but the exact value of temperatures measured could vary by up to 4°C from input to input Therefore, the results which involved relative comparison of temperatures, were restricted within a single type of input (thermocouple or multiplexer) or they were corrected based on this test Even though the values for repeatability may vary within the season or for longer periods, comparison of readings made one year apart showed little drift in the measurements when tested with the calibrator

From the second test, the values for the repeatability of the thermocouples were found to be between 1 2% and 4% of reading values (0 8°C for worst case at 19°C) for seven tests with a total number of measurements ranging between 54 and 300 observations for each test The percentage is acceptable considering that repeatability was observed to change and was usually on the lower side of the range mentioned

In addition to testing the thermocouples, some rules were used to discard some of the thermocouples These rules are based on the type of damages commonly found for thermocouples (dry connection, soldering of the tips, surface thermocouple not in proper location) and their symptoms. During the course of the study, thermocouples readings characterized by the following patterns were discarded:

- Unusually high or low temperature compared to average room temperature distributions (e.g.: air temperature values at 40°C when average room is at 20°C).
- Cyclical response patterns unmatched with global room patterns (e.g.: temperature remains constant at a location while at another location 10 cm away variations of 4°C are observed).
- Discontinuous response patterns (e.g.:thermocouples cycles between 0°C and 20°C).

The thermocouples which were found to be acceptable based on the above criteria were used in the interpretation of data. The overall accuracy of the thermocouples can be stated to be near ± 1.5°C in the worst possible case based on the manufacturer specifications and the tests performed. For relative values, the linearity of the response of the thermocouples is found to be good with the condition that values from the same type inputs be compared (multiplexer with multiplexer, straight input with straight input).

3.5.2 Globe Thermometers

The globe thermometers were hollow, black painted copper spheres of 15cm in diameter with a thermocouple inserted in the center of the sphere. There were three globe thermometers in each room. Their locations were changed during the tests.

The mean radiant temperature is calculated from the temperature of the thermocouple inside the globe as follows:

$$Tmr = Tg + k\,Va\,(Tg-Ta)^{.5} \qquad (3.2)$$

where: Tmr = mean radiant temperature (°C)

Tg = globe temperature (C)

Ta = air temperature (C)

Va = air velocity (m/s)

$k = 2.2\,(0.15/D)^{0.4}$

D = globe diameter (m)

The accuracy of the mean radiant temperature calculated from the globe thermometers is dependant on the accuracy of the thermocouple installed in it and the estimation of the air velocity around it.

To calculate the globe thermometers accuracy, seperate terms must be considered. The thermocouples can at best detect differences of 0.5°C and in equation 3.2, the difference between Ta and Tg can therefore be inaccurate by 1°C due to this. Consider for example a case where the air velocity is 0.1m/s then, in equation 3.2, the error due to the difference between Ta and Tg is multiplied by $2.2*(0.1)^{0.5}$ which is 0.7°C. Then add another 0.5°C of inaccuracy due to Tg to obtain an overall accuracy of ± 1.2°C in mean radiant temperature as read from the globe thermometer and, so far, repeatability (eq. 3.1) is not considered At 20°C, the error due to repeatability is 0.8°C which brings the overall mean radiant temperature accuracy to ± 2°C.

3.5.3 Anemometers

The anemometers used for the experiments were temperature compensated omnidirectional probes. Their locations were varied within the thermal comfort rooms for the different test performed.

The specifications for the probes are shown in appendix C. Note that the accuracy of the probes is dependant on the range of utilization as well as the direction in which the probe is oriented. The calibration curves provided by the manufacturer were fitted to be used in the analysis of results, and the error of the fits was found to be less than 0.003m/s.

3.5.4 Humidity Measurements

Humidity measurements were made using a humidity meter with analog output and a wet and dry bulb mercury thermometer equipped with a small fan. The accuracy was estimated at ± 5% relative humidity for both.

3.5.5 PMV Meter

The PMV meter is an integrating instrument which is used to evaluate PMV based on a sensor designed to provide the same heat transfer properties as a human body with the surroundings. The specifications for this instrument and its probe are listed in appendix C.

In addition to the PMV value, the meter could also display the operative temperature, the comfort temperature; the equivalent temperature, the difference between the calculated comfort temperature and the ambient temperature; and the predicted percentage of dissatisfied (PPD). The comfort parameters for clothing insulation, activity level and vapour pressure were fixed.

3.5.6 Climate Analyzer

The climate analyzer is also an integrating instrument which can measure air and surface temperatures, plane radiant temperature, air velocity and relative humidity Each parameter has a specific probe. The specifications for the probes are shown in appendix C.

The climate analyzer is an accurate way of measuring comfort parameters, but for specific locations only, and it was used for comparing with measurements obtained with other methods

3.6 Summary

The description of weather conditions, building's surroundings and its plans plans can be used to compare the results from this study with results from other studies The main point is that the building is typical of similar buildings located in this area in terms of its dimensions, its general surroundings and the envelope's thermal properties Results can apply up to a certain extent to many other residential buildings of this type

Thermal load analysis showed the differences between losses on the first and third floor and the importance of inter-zone conduction Losses have been calculated to be higher for the third floor room although floor losses may differ depending on the temperature of the basement The results from one floor can be compared within reasonable limits for different outdoor temperatures, even though particular boundary conditions may differ

From the description of the heating systems it was possible to identify some of the differences which should be expected from their response to maintain temperatures in the room. The interesting aspect is that the three types of systems differ in terms of heat distribution, heat transfer modes and the temperature controls' location.

The section on the data acquisition system showed the different components assembled to perform data collection over a long period of time. Due to space limitations, the work on software development was left in appendix B. It accounted for a very important part of the work needed to obtain the data and process them, an integral part of the research aspect.

The last section on instrumentation showed the combination of instruments necessary to obtain meaningful results, to crosscheck results and to obtain accurate measurements.

With this section completed, chapter four follows with the results obtained in the conditions and with the equipment described in this section.

CHAPTER 4

4. OBSERVED OUTDOOR CONDITIONS, SYSTEMS CYCLING, GENERAL ROOM TEMPERATURE DISTRIBUTION AND RESPONSE TO LOW INITIAL ROOM TEMPERATURES

In this chapter, the main concern is in understanding and establishing the physical behaviour of the systems based on the measurements of temperatures in the thermal comfort rooms and for different outdoor temperature conditions The test procedure is described in the first two sections and the analysis is performed in the remaining three sections

In the first section, the general test procedure is described to show how the results from this chapter and the following have been obtained The rotation scheme used to test the systems during the season and the conditions at the time of the tests are reported

The second section consists in a brief description of the outdoor temperature distributions obtained from the data collected during the heating season as obtained from the test procedure listed in the first section All the cases later analyzed will be taken from this distribution

The next three sections describe results obtained for a number of cases and concentrate on a description of the temperature cycling of the systems, the temperature distributions obtained in the rooms (air and surface), and the system's response to low initial room temperatures

The results for systems' cycling behaviour are based on four different cases of outdoor temperatures selected for each system with the aim of using outdoor conditions typical of the range observed during the whole heating season. The results and outdoor conditions are described in detail for each case.

After having established the type of cycling produced by the systems, the third section will describe the temperature distribution created. This section will first describe average room temperature distributions using isotherms and the observed temperature fluctuations in time for the same cases as for cycling. Then, the room air temperatures maintained throughout the season by the systems are shown. The last part covers surface temperature distributions.

The last section of the chapter covers the more specific case of heating from low initial temperatures for two conditions. The first, is the decrease in temperatures observed when the systems are turned off. It is used to ascertain the effect of the enclosure on temperature distributions. The second aspect is the ability of the systems to respond to low initial temperatures to determine where and how fast heat is distributed by the systems when they start warming up the rooms.

4.1 General Test Procedure

The general test procedure was designed based on some major and minor constraints. The major constraints were to (1) account for the variability of the outdoor conditions, (2) obtain comparative results between systems, (3) test a number of different cases, (4) account for differences in room and floor combinations of heating systems. Minor constraints included (1) accounting for equipments failures or testing requirements, (2) optimizing use of the three or four days which were available for tests during the week and, (3) working within the limitations of the DAS capacity.

To work within these constraints, the following general procedure was selected

1- Define a standard condition for operation of the systems in the two comfort rooms

2- Measure and record parameters for the standard condition for most of the season and rotate times of operation of the systems each week. This is to obtain widespread and similar distributions of outdoor conditions for the systems

3- For instruments linked to the DAS (considering the limitations of the DAS), use a 15 minutes interval period for collection periods lasting more than two days, and 8 minutes for shorter periods

4- Measurements of relative humidity and air temperatures in the rooms are made at the beginning and end of test periods using the wet/dry bulb thermometer

5- For some specific experiments, the standard conditions are modified and changes are noted in the experimental log book

The standard conditions described in step 1 are as follows for both floors:

-Thermostat fixed at a constant setting (21°C) for the whole floor,

-Direct solar radiations blocked by reflective cardboard panel 5cm away from outside of the window and interior blinds partially opened,

-Door closed with air free to pass under ($\approx$240 cm^2 opening),

-No occupants, no furniture.

Following is a brief explanation of the purpose of these conditions , later referred to as the **"normal conditions"**, in this chapter and the next ones.

The thermostat setting of 21°C corresponds to air temperature expected to be selected by occupants for energy conservation and comfort. It is corresponds to a PMV of -0.10 (between neutral and slightly cool, PPD=5.2%) for typical indoor winter clothing, light indoor activity, air velocity less than 0.1m/s, Tmr=Ta and 50% relative humidity. This setting was maintained throughout the apartment and in the comfort rooms.

Solar radiation is eliminated in order to simplify the analysis and to concentrate on the heating effect of the systems only. Outside radiation gains are important for comfort but vary considerably in time during the season and therefore, the data available for comparing the systems would have had to be screened to precisely evaluate this factor. If direct radiations gain had been included, the outdoor conditions for each recording period would have had to be stated in terms of outdoor temperatures and outdoor radiation gains which would reduce the number of days when similar conditions would be available to compare the systems.

Preventing direct solar radiations using the reflective cardboard panel on the outside was done using a small space of approximately 5cm between the glass of the window. This allowed the natural convection process to take place. This way, the effect of a cold window surface could still be included as one of the important factors influencing the thermal performance of the systems.

Keeping the door closed was important when two systems were working on the same floor (forced air/modular or hydronic/modular cases running at the same time and) to limit the effect of the heat produced in the rest of the apartment by the system not tested. This had the disadvantage isolating the room for control (forced air and hydronic). For the forced air system, increases in the pressure of the apartment outside the comfort room was thus partially limited but not totally.

The forced air and hydronic systems were slightly disadvantaged by not having their control out of the comfort rooms, but the proximity of the thermostat was expected to suffice for the thermostat to respond to demands similar to those required by the room. In actual case, the thermostat used by building occupants would be located as was the case here and for these types of systems

Conditions with no occupants or furniture in the rooms had the advantage of simplifying heat distribution analysis. In any case, the distributions observed for an empty room can be extrapolated up to a certain extent using some judgement to cover occupied and furnished rooms The brief presence of researchers in the building at the time of the tests was necessary to control the tests and verify the conditions but did not influence the measurements

As seen here, the above "normal conditions" were defined and maintained in order to sufficiently restrict the number of variables involved in the problem to be able to compare the systems using the steps mentioned at the beginning The "normal conditions" were modified for certain cases and when changes were made, the nature of the changes was controlled and noted The next section discusses the outdoor temperature distributions observed when "normal conditions" were maintained for each system

4.2 Description of Outdoor Conditions Observed During the Heating Season

The three types of heating systems were studied for a complete heating season covering the months of November 1988 to March 1989. Since only two heating systems could be studied at the same time, the observed seasonal outdoor temperature distributions at the time of the tests differ from system to system and represent a part of the complete outdoor temperature distribution of the season. These distributions are shown here and their averages are compared with climate normals presented in chapter three.

A numerical summary of the outdoor temperatures which prevailed during the season at the time of the tests performed in normal conditions on each system appear in Table 4.1 and in Figures 4.1 and 4.2.

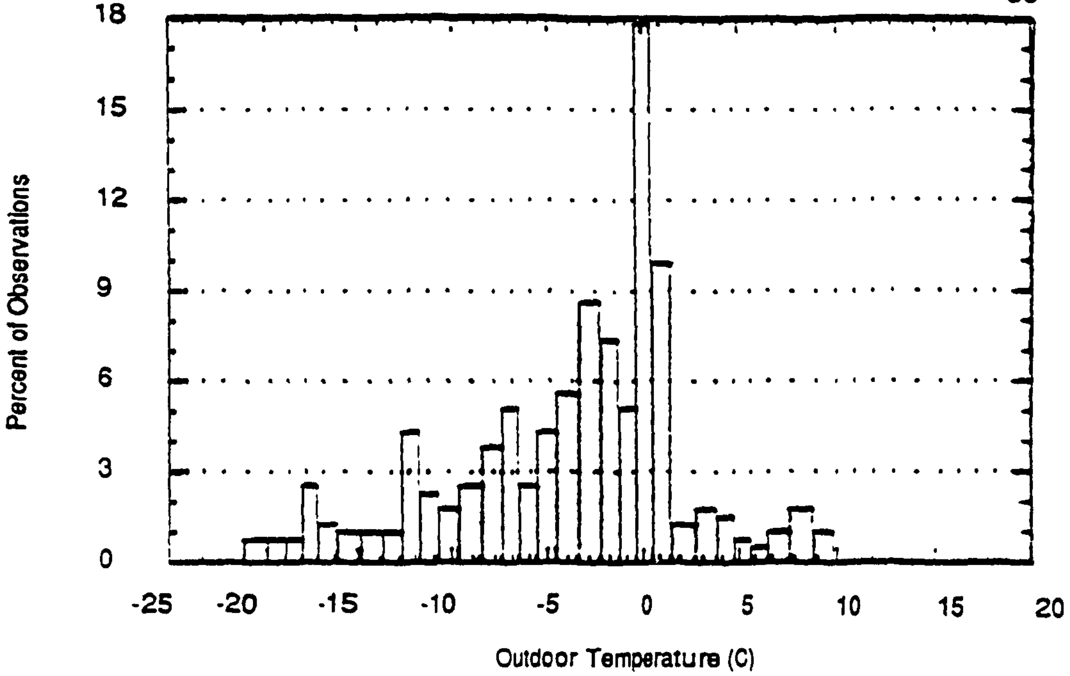
The outdoor temperatures data used are taken from the meteorological station at the Montreal international airport [24] in order to compare the averages with the climate normals listed in chapter three. Note however that outdoor temperatures measured at the house will be used later on, since the airport's values differed (by 2 to 5°C) with respect to values measured on the site.

The distribution in outdoor temperature obtained at the time of the tests for normal conditions are concentrated around 0°C for all systems but a higher percentage of the modular's observations for outdoor temperature were above 0°C (55%) than for the forced air and hydronic systems (23%).

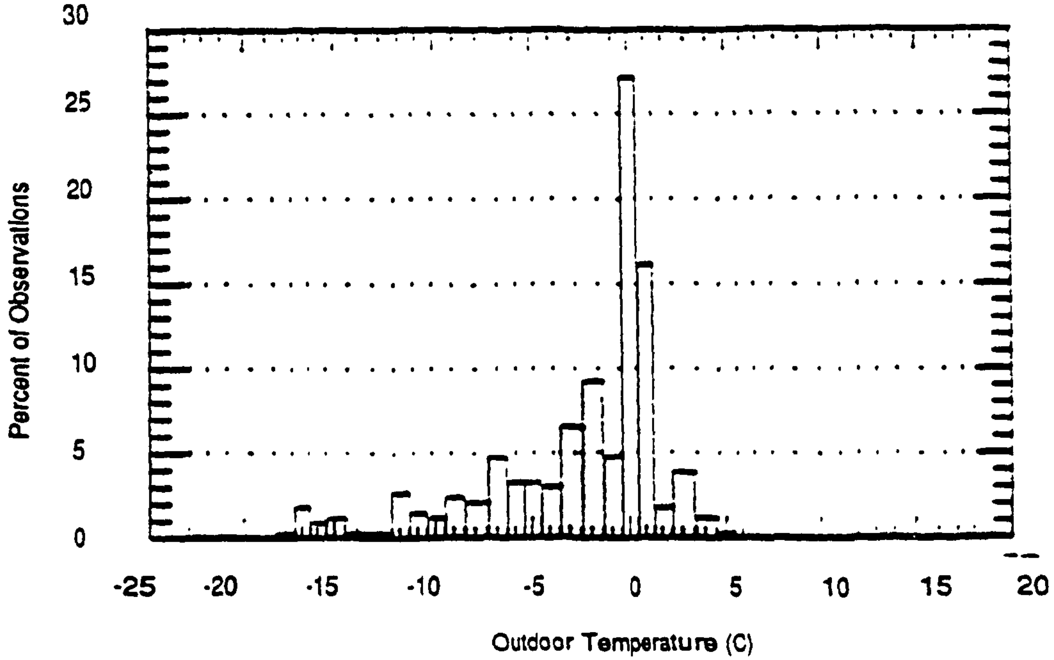
For analysis, the outdoor temperature distributions obtained for the hydronic and forced air systems are better as their distributions spread over a wider range with less days for which the same outdoor temperatures were repeated.

Table 4.1: Outdoor Conditions and Number of Observations at Normal Conditions of Operation (88'-89' season, Mtl. Int'l Airport Weather Data)

	FORCED AIR	HYDRONIC	MODULAR 1ST	MODULAR 3RD
MONITORING				
hours	393	335	286	342
days	16	14	12	14
T_{OUT} (°C)				
Maximum	9	5	17	17
Minimum	-21	-18	-12	-14
Average	- 4	- 3	2	1
Range	30	23	30	31
T_{OUT} DISTRIBUTION, (%) observed				
< -10°C	17	22	4	6
-10 to 0°C	60	55	42	39
> 0°C	23	23	54	55

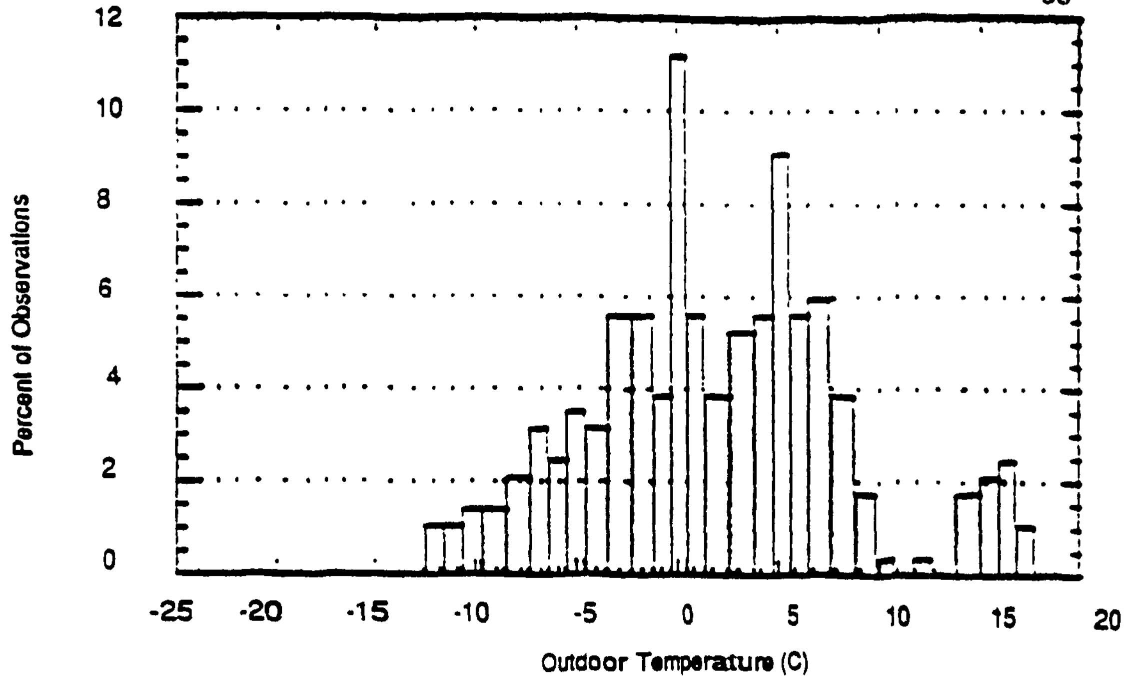


a) Forced air

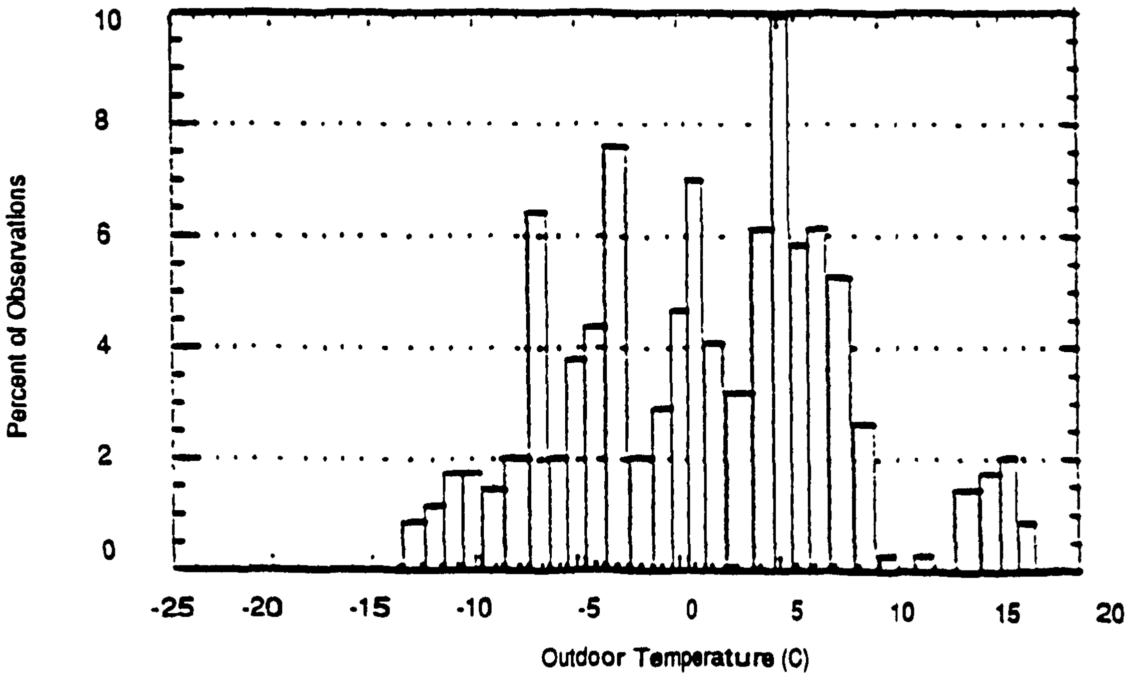


b) Hydronic

Figure 4.1: Outdoor Temperature Distributions Obtained During the Season, Forced Air and Hydronic Systems



a) Modular 1st



b) Modular 3rd

Figure 4.2: Outdoor Temperature Distributions Obtained During the Season, Modular Systems, First and Third Floor

If the average outdoor temperatures observed during normal conditions are compared with the averages mentioned in chapter three (section 3.1), it is found that the observed outdoor temperature values on site were warmer than the climate normals for the same months. Observed averages were above -3°C compared to the weather station's maximum average of -5.7°C for December, January and February combined and, therefore, the "seasonal" distribution of outdoor temperatures of the tests can be said to be a distribution warmer than what is expected to be found on the average for this region.

The outdoor test conditions presented in this section should be regarded as a reference for the following sections and chapters where they will be referred to as the "seasonal" distributions. A number of days and hours of information were not included in these statistics either because the conditions were not as defined for "normal operation" or because some of the days were used for checking instrumentation (calibration, repair, testing) or errors in the tests procedures were found.

4.3 Description of System's Cycling

It is essential to first concentrate the discussion of the results on the system's cycling behaviour to obtain the basic operating features of each system since they will be reflected later in the results for room air temperature distribution and comfort. Cycling behaviour includes (1) the system's outlet and surface temperature changes (range and rate of change), (2) the general cycling pattern and, (3) the variations in (1) and (2) with respect to different outdoor temperatures cases. These are the three points discussed below based for several days of observation for each systems, grouped into four different outdoor temperature cases.

4.3.1 Description of Cases Selected

From the distribution of outdoor temperatures listed in the preceding section, and from a review of the actual variations of outdoor temperatures observed for each day and each system, it was found that four cases could be used to generally analyze and describe the cycling of the systems. The four cases were:

Case 1: Average outdoor temperatures close to 0°C and remaining fairly constant.

Case 2: Average outdoor temperatures less than -5°C and remaining fairly constant.

Case 3: Outdoor temperatures rising from below 0°C to values near 0°C.

Case 4: Outdoor temperatures dropping from above 0°C to low values.

From these cases, the response of the systems to warm and cold outdoor temperatures (cases 1 and 2) and for cases of outdoor temperatures changing widely during a short period (cases 3 and 4) can be obtained. The first two cases are therefore for near steady state conditions while the other two are for transient conditions.

Intermediate cases were also observed during the season but this selection has the advantage to include outdoor conditions which were repeated for the four systems (possibility to compare) and basing the analysis on statistics for specific cases (instead of the whole heating season) provides more details.

4.3.2 Results Obtained for Each System

In the following, the results obtained for each system are shown separately and compared at the end. Before the results are discussed, the format of the information presented and some points related to the measurement procedure are mentioned.

The results obtained from the analysis related to the four systems and each cases is given in four statistical Tables (Tables 4.2 to 4.6) and 14 Figures (Figures 4.3 to 4.16). The statistics shown in the tables are related to the outdoor conditions at the time of the tests, the temperature of the air supplied by the heaters, and in addition to the average, maximum, minimum and ranges of these temperatures, the maximum rate of change in air temperatures delivered, the sampling time and the average number of cycles. The figures show the corresponding time variations in outdoor, supply and surface temperatures.

Note that for the hydronic and the modular system (3rd floor) only three cases out of four are shown. With the rotation in the operation of the systems, it was not possible to find days similar to the other two systems for the case of a major rise or decrease (cases 3 and 4) in outdoor temperature for these systems.

For the three types of systems, the "supply" and "surface" temperatures have a slightly different meaning:

Forced air :	Supply temperature measured at outlet of supply grille (thermocouple in the air approximately 1cm away from the grille). No heater surface temperature.

Hydronic	:	Supply temperature is the temperature of air measured between the top and the front covers of the baseboard. Surface temperature measured on the front cover.
Modular	:	Supply temperature is temperature of air measured right above the grille located on top of the unit. Surface temperature measured on front cover of the unit.

The rate of temperature change is defined by the ratio of the difference in temperature between two consecutive sampling points by the sampling period. Since the sampling period is different from case to case, (being either 8 minutes or 15 minutes), the rate of temperature change is a more relative and significant quantity than the absolute change in temperature. It incorporates both the change in temperature and the actual time interval used between readings. In this sense, the difference in sampling periods will have less influence on the results for rates of temperature changes.

The number of cycles per hour listed in the tables are the average number of cycles. These are calculated by dividing the number of on-off cycles for a given period by the total number of hours in the recording period. In some cases, when the maximum and minimum supply temperatures were not clearly defined in time, the number of cycles listed includes some of the intermediate values of supply temperatures observed. Also, the average number of cycles does include hours when data collection was interrupted.

With this information on the conditions of the tests and the format of the results, results obtained for the forced air, the hydronic and the modular systems are discussed. For the systems, the following parameters are discussed: (1) the system's outlet and surface temperature changes (range and rate of change), (2) the cycling pattern, and (3) the variations in (1) and (2) with respect to different outdoor temperatures.

Forced Air

Let us first describe the operation of the forced air system based Figures 4.3 to 4.6 which show the variations in outdoor, supply and return temperatures for the cases listed in Table 4.2.

i) Supply and Return Temperatures: For all cases, supply temperatures were between 20°C and 35°C (15°C range, 28°C average) with maximum temperature rise of 15°C between supply and return. The temperature rise is therefore on the lower end of the 14°C to 31°C rise range rated by the manufacturer (between inlet and outlet at the furnace).

Taking the maximum time rate of temperature change of the supply for the forced air system, it can be seen that the increases during the on-cycles are similar to the decreases during the off-cycle for most cases. For example, for cases 1, 2 and 3, the ratios of maximum increase to maximum decrease are equal to 1.1, 1.0 and 0.8 respectively. This indicates that in general the variations in supply temperatures are similar for on and off cycles.

ii) Cycling Pattern: The seesaw shape of the variations in supply temperatures indicates cycles with well defined maximum and minimum values of supply temperatures. The cycles in most cases are repeated consistently during the period and do not vary in shape from hour to hour.

iii) Case to Case Variations: For case 4, there were differences in the maximum rates of supply temperature change and the cycling patterns observed.

In cases 1 to 3 the maximum observed increase and decrease rates of supply temperatures were approximately the same but, for case 4 the maximum decreases was significantly higher (approximately 35%). The higher decrease in supply temperature during the off-cycle in this case has been influenced by the rapidly dropping outdoor temperatures (rate of 1°C/h) which cooled the room after hour 12 (Figure 4.6). For the period outdoor temperature ranges in the order of case 4 (T_{out} range of 17°C), the maximum supply temperature rates of change can therefore be expected to differ between on and off period of a cycle but are otherwise similar.

Table 4.2: Statistics for Forced Air System

	Case 1	Case 2	Case 3	Case 4
SAMPLING INTERVAL (min.)	15	8	15	8
T_{out} (°C)				
Average	- 0.7	- 6.2	- 2.3	4.1
Maximum	0.0	- 3.4	- 0.1	10.1
Minimum	- 2.3	- 9.0	- 6.3	- 7.0
Range	2.3	5.6	6.2	17.1
SUPPLY (°C)				
Average	28.6	28.2	28.4	25.5
Maximum	34.2	34.0	34.2	33.0
Minimum	24.0	24.0	24.4	20.0
Range	10.3	10.0	9.8	13.0
Maximum Rates (°C/n)				
Positive	30	60	57	26
Negative	-28	-62	-71	-40
CYCLES/HOUR (average)	1.2	2.1	2.2	0.7

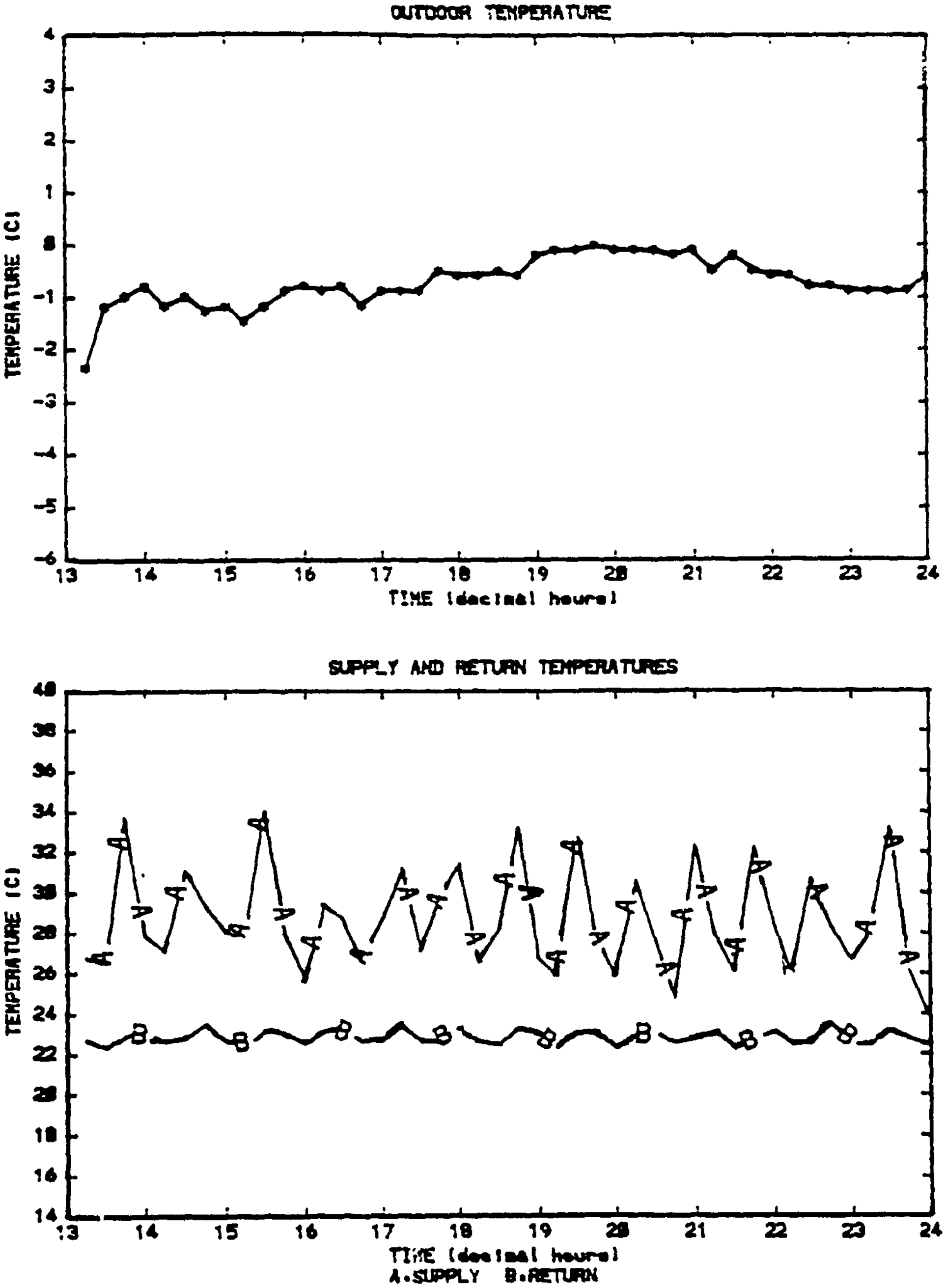


Figure 4.3: Case 1, Forced Air System, Outdoor and Supply Temperatures Variations

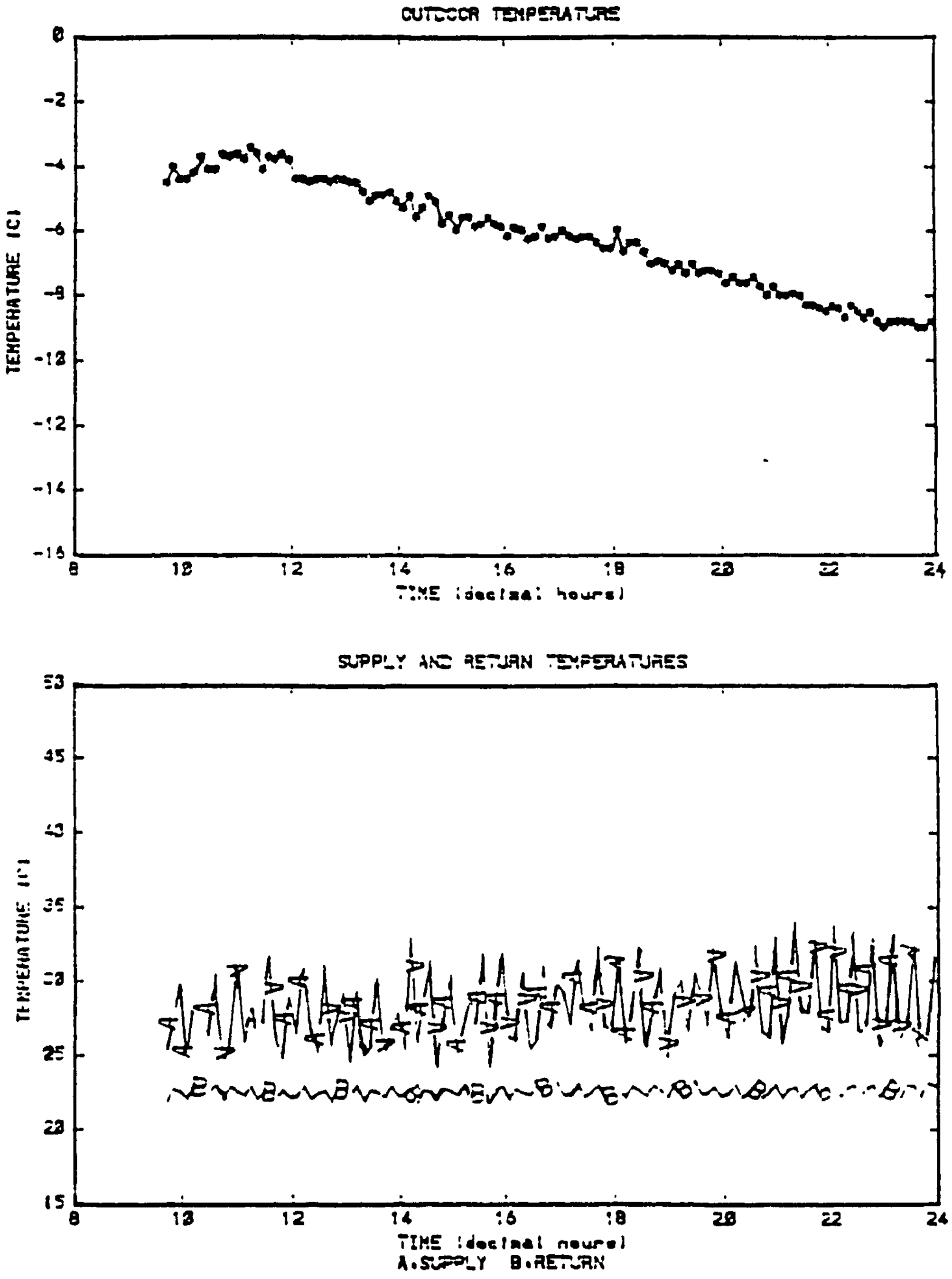


Figure 4.4: Case 2, Forced Air System, Outdoor and Supply Temperatures Variations

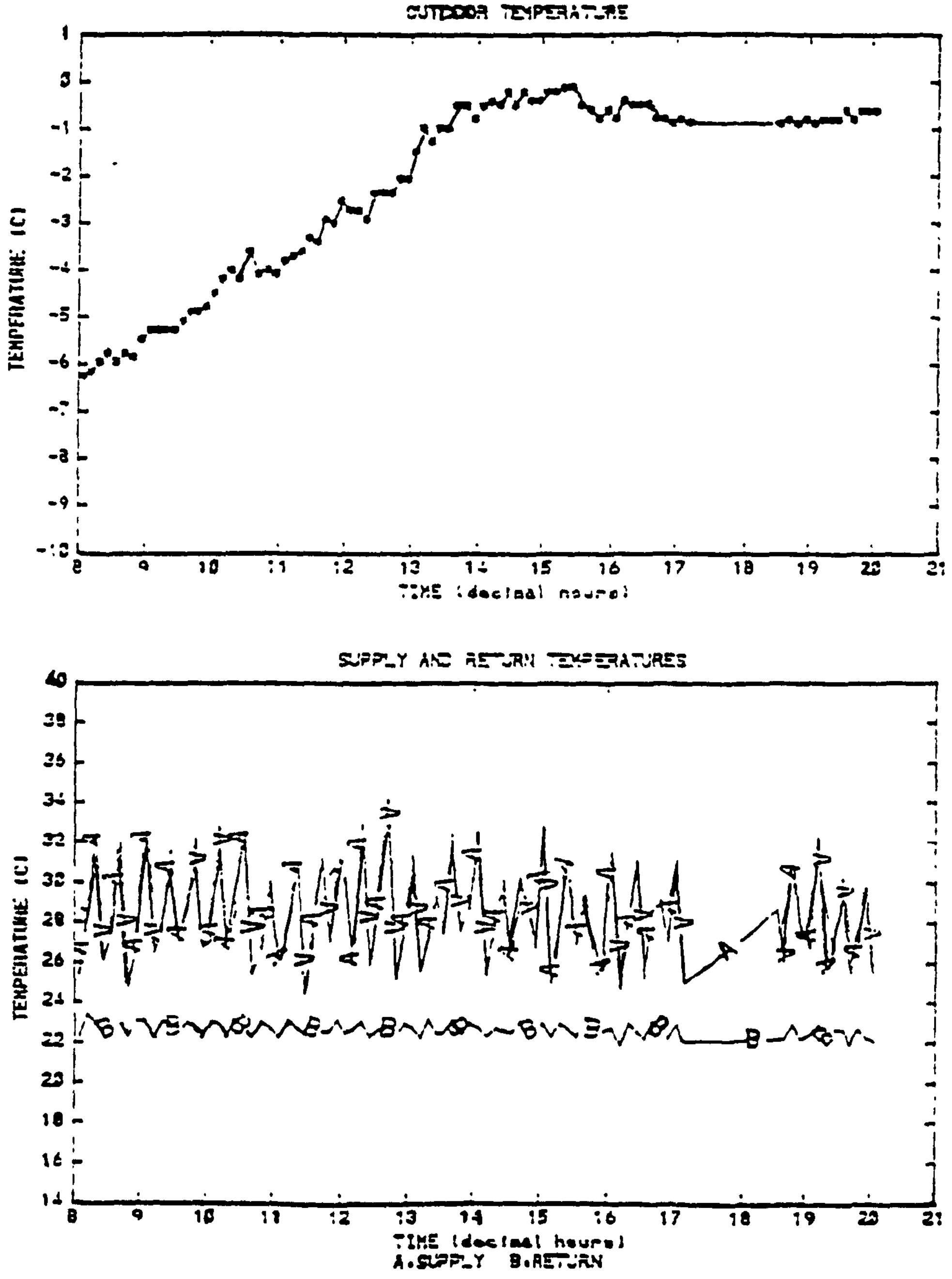


Figure 4.5: Case 3, Forced Air System, Outdoor and Supply Temperatures Variations

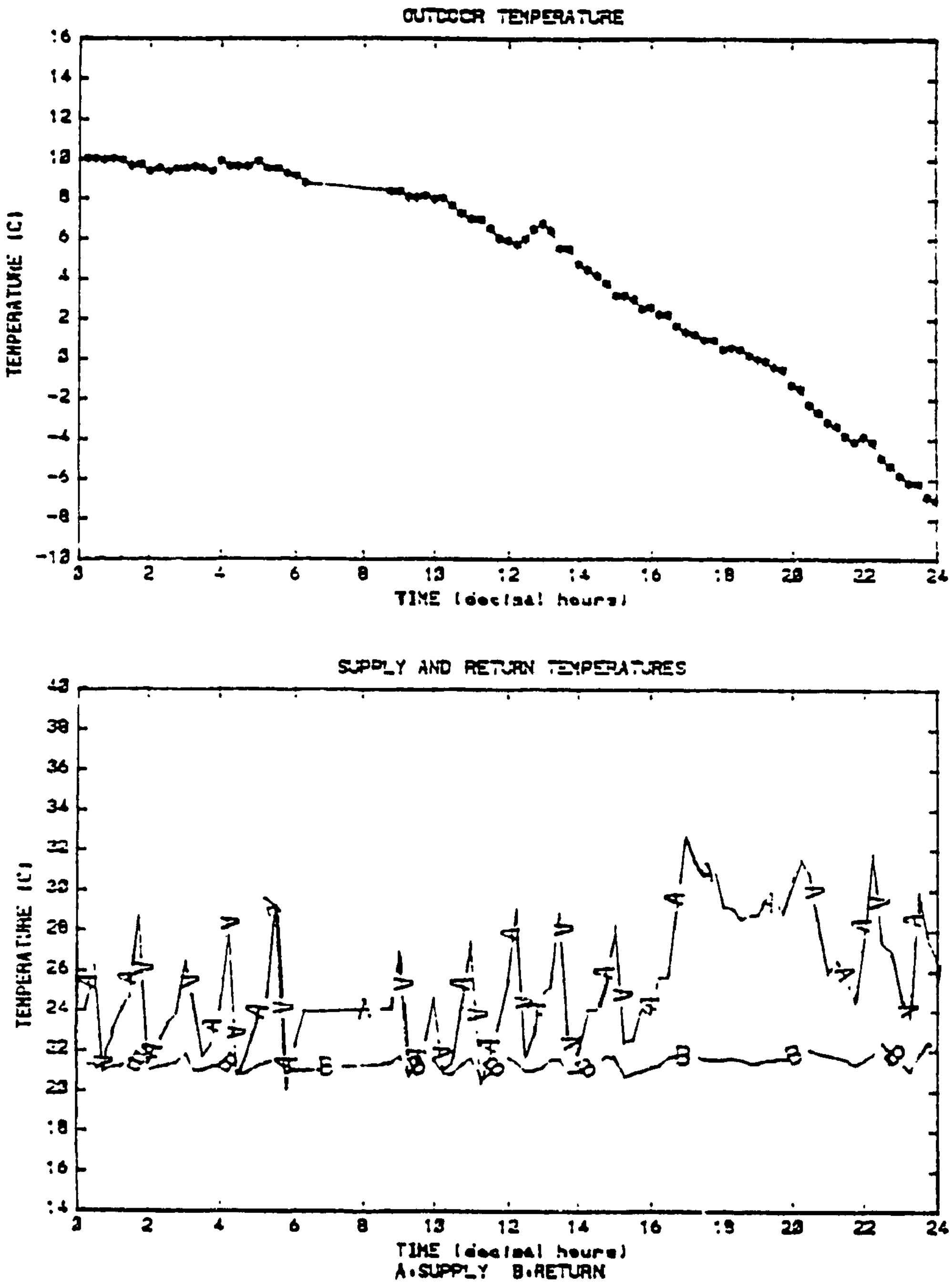


Figure 4.6: Case 4, Forced Air System, Outdoor and Supply Temperatures Variations

The average number of cycles varied from 0.7 to 2.2 for these different cases and as expected, the number of cycles increased as the average outdoor temperature decreased, but some changes in the cycling patterns were observed for case 4. In case 4, the cycling pattern changed when supply temperatures remained high for four hours (Figure 4.6 between hours 16 and 20). A definite change in the cycling pattern does not occur for the case of rising outdoor temperatures (case 3, Figure 4.5). However, for case 3, the range of variation in outdoor temperature is near 6°C compared to a 17°C range for case 4.

From the above points, the typical operation of the forced air system has been discussed in terms of temperature cycling. It was shown that except for outdoor temperature changes in the order of case 4, the operation of the system is fairly similar for different outdoor temperatures.

Hydronic System

The description of the operation of the hydronic system is based on Figures 4.7 to 4.9 which show the variations in outdoor, supply and return temperatures for the cases listed in Table 4.3. As noted previously outdoor temperature for case 4 were not available.

i) Supply and Surface Temperatures: For all cases, the hydronic system operated with air temperatures supplied between 26°C and 49°C, for a range of 23°C, and 30°C average. The surface temperature of the baseboard followed that of the air with a temperature on the average close to 25°C, less than the 30°C value for air, and varied over a 5°C range, compared to the 20°C value for the air.

The ratio of the maximum rates of increase to the maximum rate of decrease of air temperature supplied by the hydronic system is 1.1 for cases 2 and 3 and 0.5 for case 1. This shows that decrease and increase rates changed from case to case with lower rates of increase at the warmer outdoor temperatures. Rates of change (increases or decreases) were observed to vary between 34°C/h an 192°C/h.

ii) Cycling Pattern: The cycling pattern for the hydronic system did not repeat itself consistently from case to case. For case 1, the cycle remains constant in terms of the maximum and minimum temperatures of the cycle but the period changes. For cases 2 and 3, both the cycling period and the maximum and minimum temperatures of the supplied varied in time. The average number of cycles was observed to increase with decreasing temperatures, ranging between 0.6 and 2.9 cycles/hour.

iii) Case to Case Variations: Rates of supply temperature changes and cycling patterns were found to differ from case to case.

When cases 2 and 3 are compared with case 1, it can be asked which is the highest, the rate of increase or the rate of decrease of the baseboards' supplied temperatures? Cases 1 and 2 indicate that the baseboard's temperature should increase and decrease at approximately the same rate, while for case 1, the maximum decrease is two times higher than the maximum increase.

This result can be explained mostly by the difference between the number of cycles observed for the mild (case 1, Figure 4.7) and the lower outdoor temperature cases (cases 2 and 3, Figures 4.8 and 4.9). In case 1, when the thermostat requested heat from the boiler (beginning of the on-cycle), this water was allowed to cool during the off-cycle for at least twice as much time as for cases 2 and 3. Therefore, more time was required to reheat the water to satisfy the demand. For cases 2 and 3, the water temperature remained high throughout the recording period since the cycles were shorter and water in the system's pipes were allowed to cool down shorter periods before the on-cycle.

Table 4.3: Statistics for Hydronic System

	Case 1	Case 2	Case 3
SAMPLING INTERVAL (min.)	15	8	8
T_{out} (°C)			
Average	-1.7	-8.5	-4.5
Maximum	-0.8	-6.8	-0.1
Minimum	-3.4	-10.5	-7.6
Range	2.6	3.7	7.5
SUPPLY (°C) (air)			
Average	29.5	35.6	31.4
Maximum	35.5	48.4	40.3
Minimum	25.4	27.4	26.0
Range	10.1	21.0	14.3
Maximum Rates (°C/h)			
Positive	16	147	192
Negative	-34	-130	-173
SUPPLY (°C) (surface)			
Average	25.7	25.0	24.4
Maximum	28.2	27.5	26.2
Minimum	23.4	22.7	22.6
Range	4.8	4.8	3.6
CYCLES/HOUR (average)	0.6	2.9	2.1

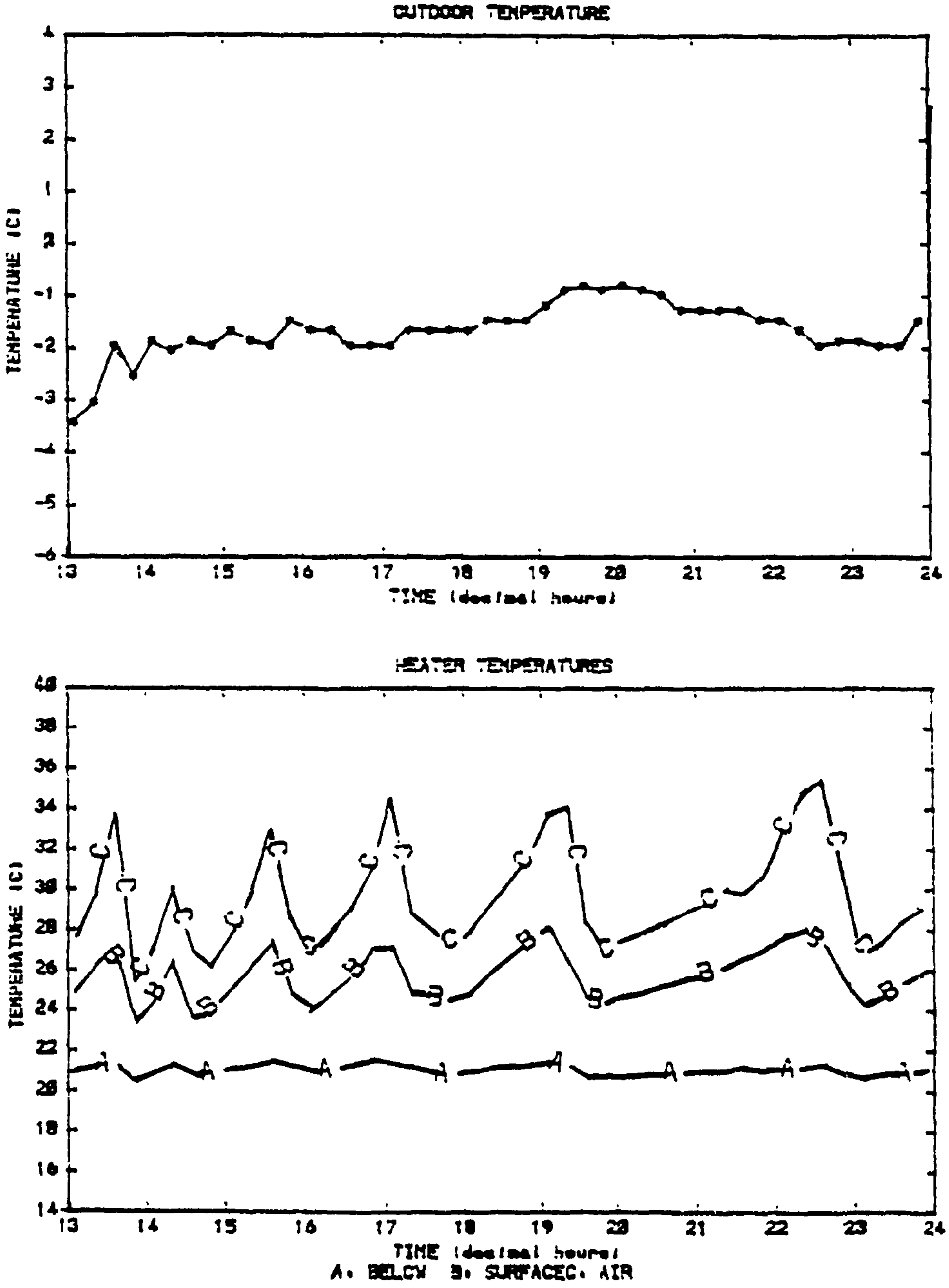


Figure 4.7: Case 1, Hydronic System, Outdoor and Supply Temperatures Variations

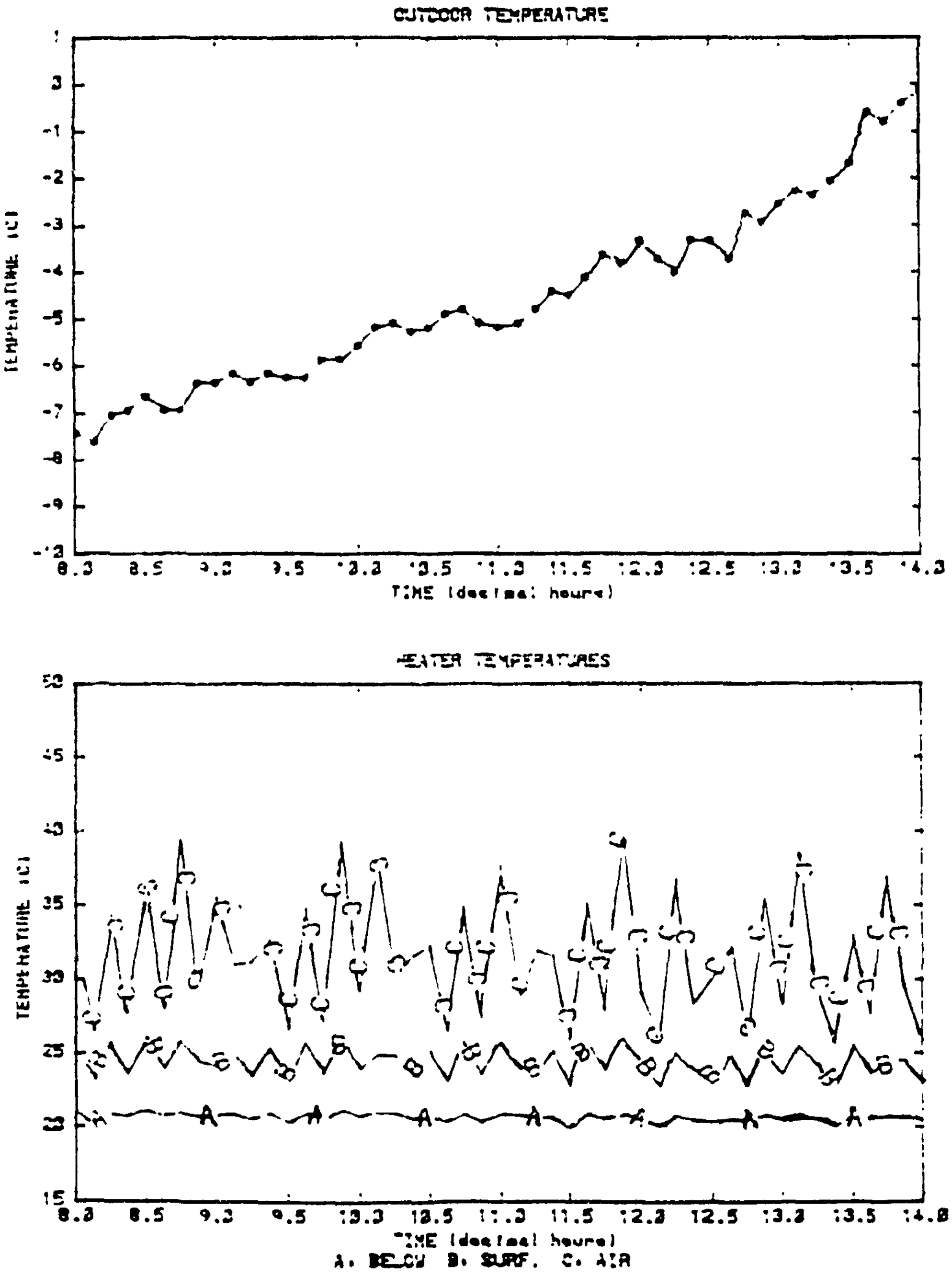


Figure 4.9: Case 3, Hydronic System, Outdoor and Supply Temperatures Variations

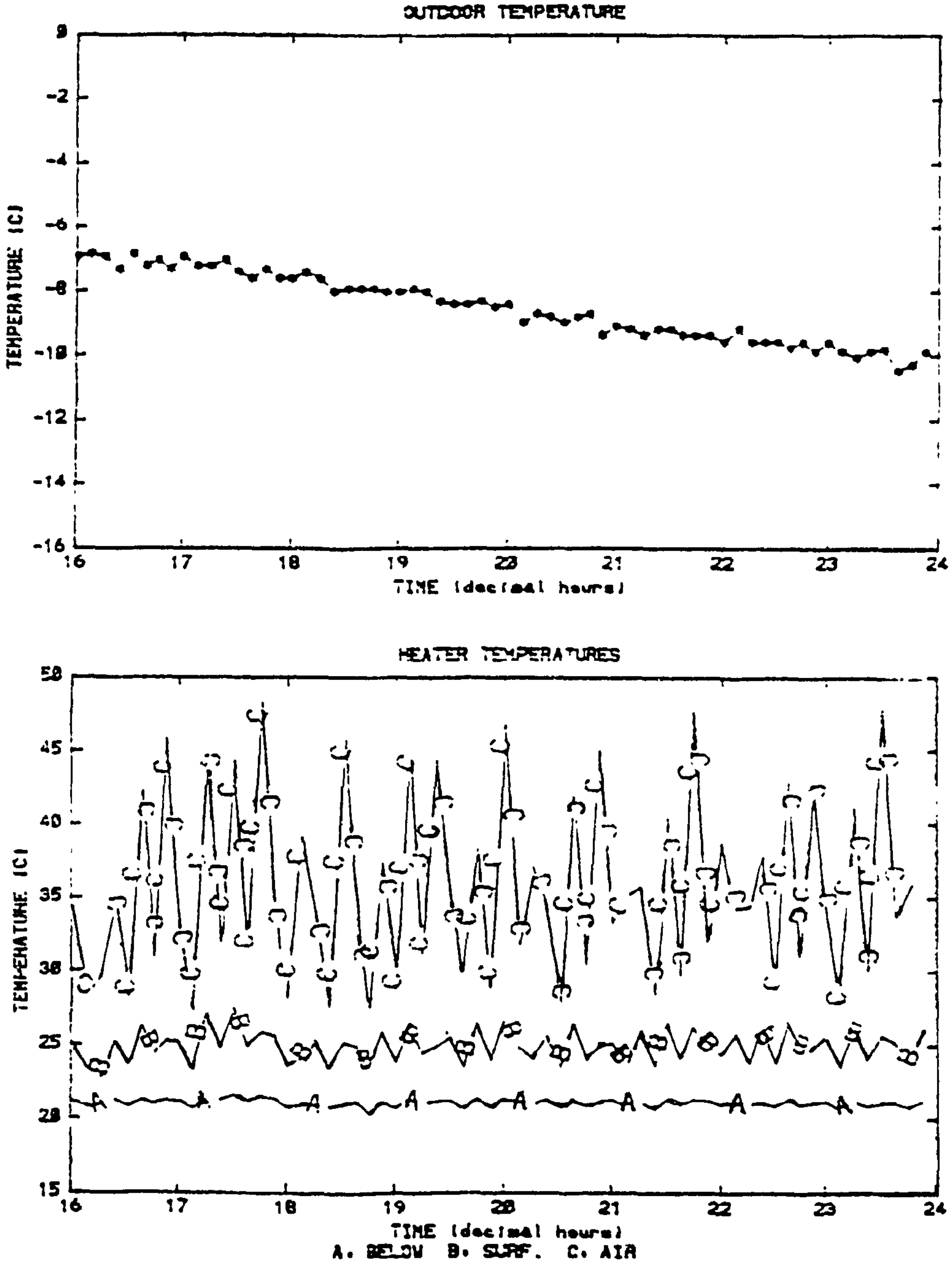


Figure 4.8: Case 2, Hydronic System, Outdoor and Supply Temperatures Variations

The different shapes of the cycles are also related to the cases observed just as the rates of supply temperatures were For cases 2 and 3, the temperature of the supply is behaving differently in the sense that the cycle pattern is not just the same cycle repeated a certain number of times. What is observed is peak and lowest cycle temperatures of supply being split between three different levels The levels can be considered to be at high, medium or low temperatures of operation

Case 2 illustrates this where, for example, the high, medium and low temperatures can be approximately assigned a value of 45°C, 35°C and 28°C respectively

A complete cycle can therefore be ascribed to the systems' supply temperature varying between either one of those values For example, a complete cycle can be between low and high, low and medium or medium and high temperatures etc Even if in general this plateau can be observed, the data show that the cycling does not follow exactly those rules and other levels that exist between the most frequent levels

From the discussion so far, the results for the hydronic system differ from those of the forced air both in terms of the supply temperature values and the variations from case to case The difference can be explained by the fact that hydronic system has a different ability to provide heat at different cycles duration and different outdoor temperatures

Modular System

The discussion of the cases for the modular system on the first floor is related to Table 4 4 and Figures 4.10 to 4.12 Note that data were not available for case 3 The results for the modular system on the third floor appear in Table 4 5 and in Figures 4 13 to 4 16 Results for both floors are discussed together since the systems behaved similarly

i) Supply and Surface Temperatures The modular system on both floors cycled with air temperatures between 36°C and 86°C for a 50°C range and on average near 50°C. The surface temperature at the front of the modular were on the average close to 25°C and varied over a range less than 7°C following air temperature variations.

The maximum rates of increase in supply temperatures were between 197°C/h and 260°C/h, and rates of decrease were lesser and between 144°C/h and 175°C/h. The ratio of rates of increase to decrease in supply air were between 1.3 and 1.4 and from the figures it can be seen that increases in heater's supply temperature were sharp while decreases were slow and followed a relation of the type $T_0{}^{*}e^{-t/t0}$, where T_0 would be the peak temperature, t the time and t0 a time constant.

<u>ii) Cycling Pattern:</u> The cycles remained fairly constant throughout a period and were characterized by a sharp temperature increase from minimum values to peak, and by slower drops following the end of the on period. The average number of cycles for the different cases were less than 1 cycle/h with a slightly larger number of cycles for lower outdoor temperatures. The shape of a cycle as well as the variations during a period are not significantly different from cases to cases.

<u>iii) Case to Case Variations:</u> From the data, only small differences were observed between the operation of the modular system between floors and between the operation of the system for the different cases.

The pattern for the cycling of the modular system on the third floor can be said to be similar to that of the modular system on the first floor with supply and surface temperatures having the same range and the same average values. The results for the third floor demonstrate that results are similar and thus the description of the operation of the systems is double checked

Table 4.4: Statistics for Modular System on 1st floor

	Case 1	Case 2	Case 4
SAMPLING INTERVAL (min.)	8	8	8
T_{out} (°C)			
Average	-2 2	-6.7	-3.2
Maximum	0 1	-3.7	-0 1
Minimum	-4 1	-7.7	-6 6
Range	4.2	4 0	6.5
SUPPLY (°C) (air)			
Average	47 4	50.2	48 6
Maximum	82 7	83 4	81 9
Minimum	35 9	36 2	36 3
Range	46 8	47 2	45 6
Maximum Rates (°C/h)			
Positive	225	233	226
Negative	-170	-177	-172
SUPPLY (°C) (surface)			
Average	24 2	24 7	24 5
Maximum	28 0	28 4	28 2
Minimum	22 5	22 5	22 6
Range	5 5	5 9	5 6
CYCLES/HOUR (average)	0 6	0 8	0 7

Table 4.5: Statistics for Modular System on 3rd floor

	Case 1	Case 2	Case 3	Case 4
SAMPLING INTERVAL (min.)	8	8	15	8
T_{out} (°C)				
Average	-3.1	-7.5	0.6	-4.3
Maximum	-0.2	-4.1	7.4	-1.0
Minimum	-5.2	-9.0	-5.5	-8.1
Range	5.0	4.9	12.9	7.1
SUPPLY (°C) (air)				
Average	44.3	47.4	39.9	46.1
Maximum	84.2	85.6	85.2	85.9
Minimum	39.4	38.9	35.8	39.2
Range	44.8	46.7	49.4	46.7
Maximum Rates (°C/h)				
Positive	260	241	197	251
Negative	-190	-187	-144	-191
SUPPLY (°C) (surface)				
Average	22.9	23.2	22.5	23.2
Maximum	25.6	25.6	25.4	25.7
Minimum	22.2	22.2	21.8	22.1
Range	3.4	3.4	3.6	3.6
CYCLES/HOUR (average)	0.4	0.6	0.4	0.5

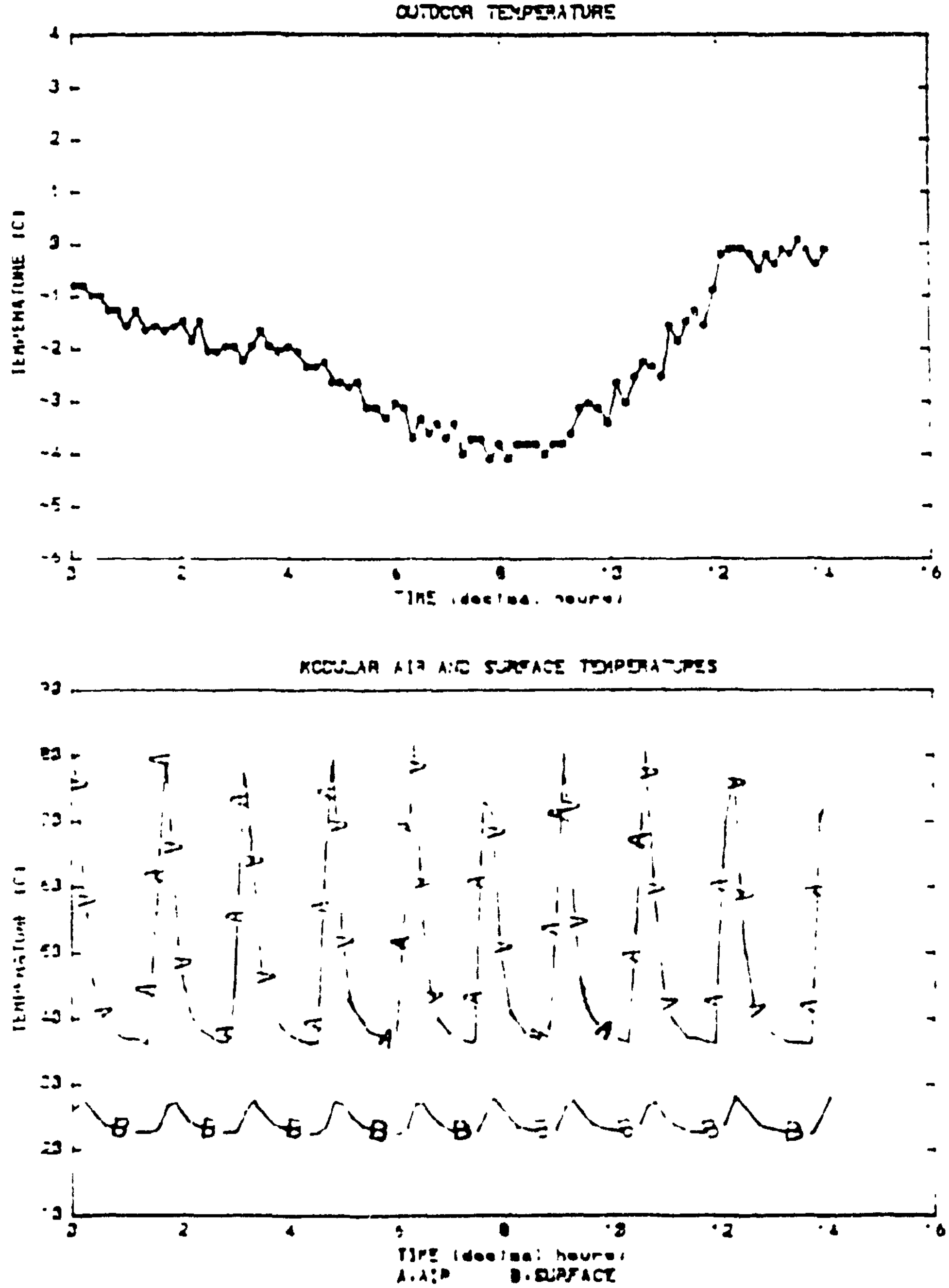


Figure 4.10 Case 1, Modular 1st Floor, Outdoor and Supply Temperatures Variations

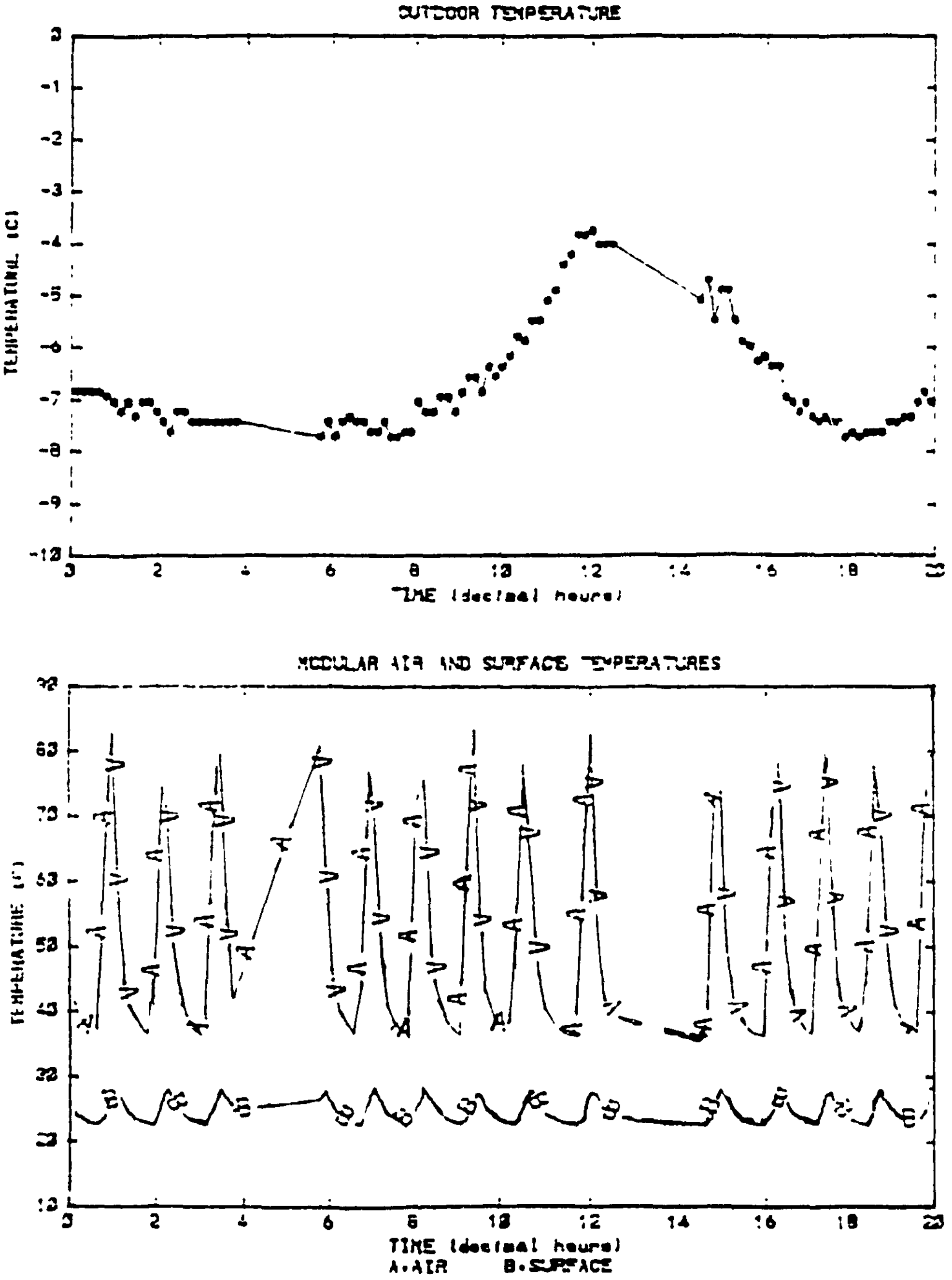


Figure 4.11: Case 2, Modular 1st Floor, Outdoor and Supply Temperatures Variations

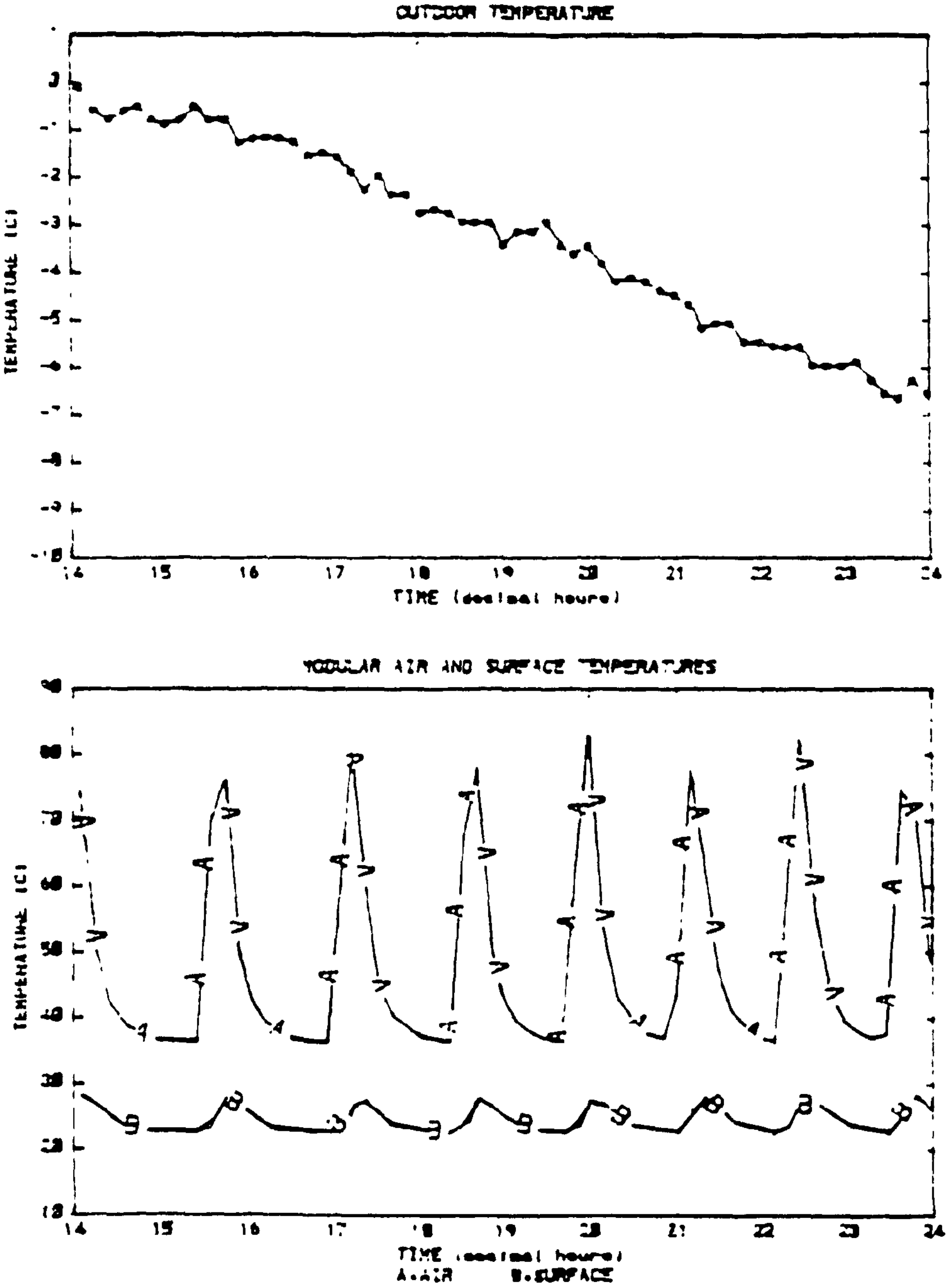


Figure 4.12: Case 4, Modular 1st Floor, Outdoor and Supply Temperatures Variations

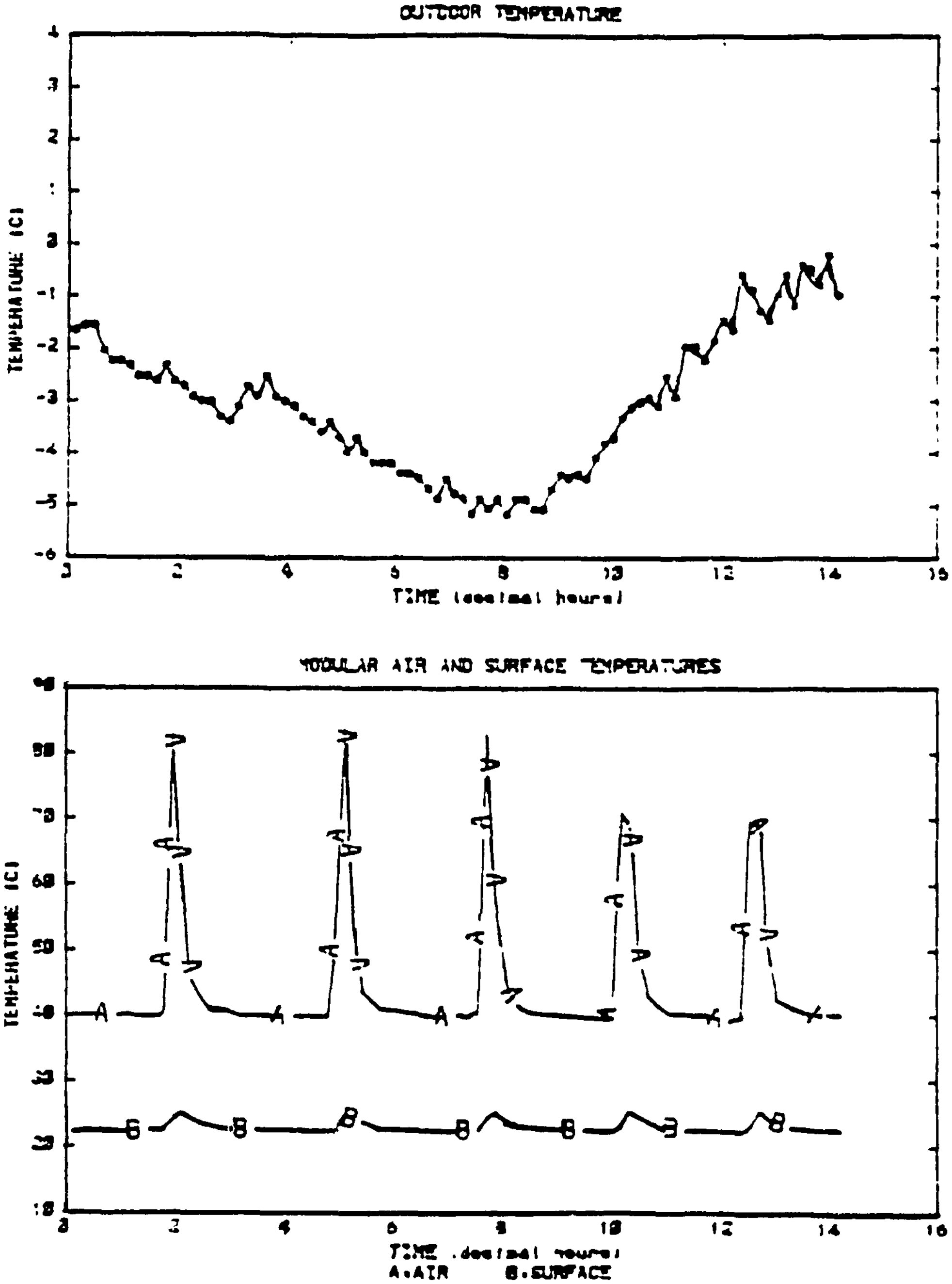


Figure 4.13: Case 1, Modular 3rd Floor, Outdoor and Supply Temperatures Variations

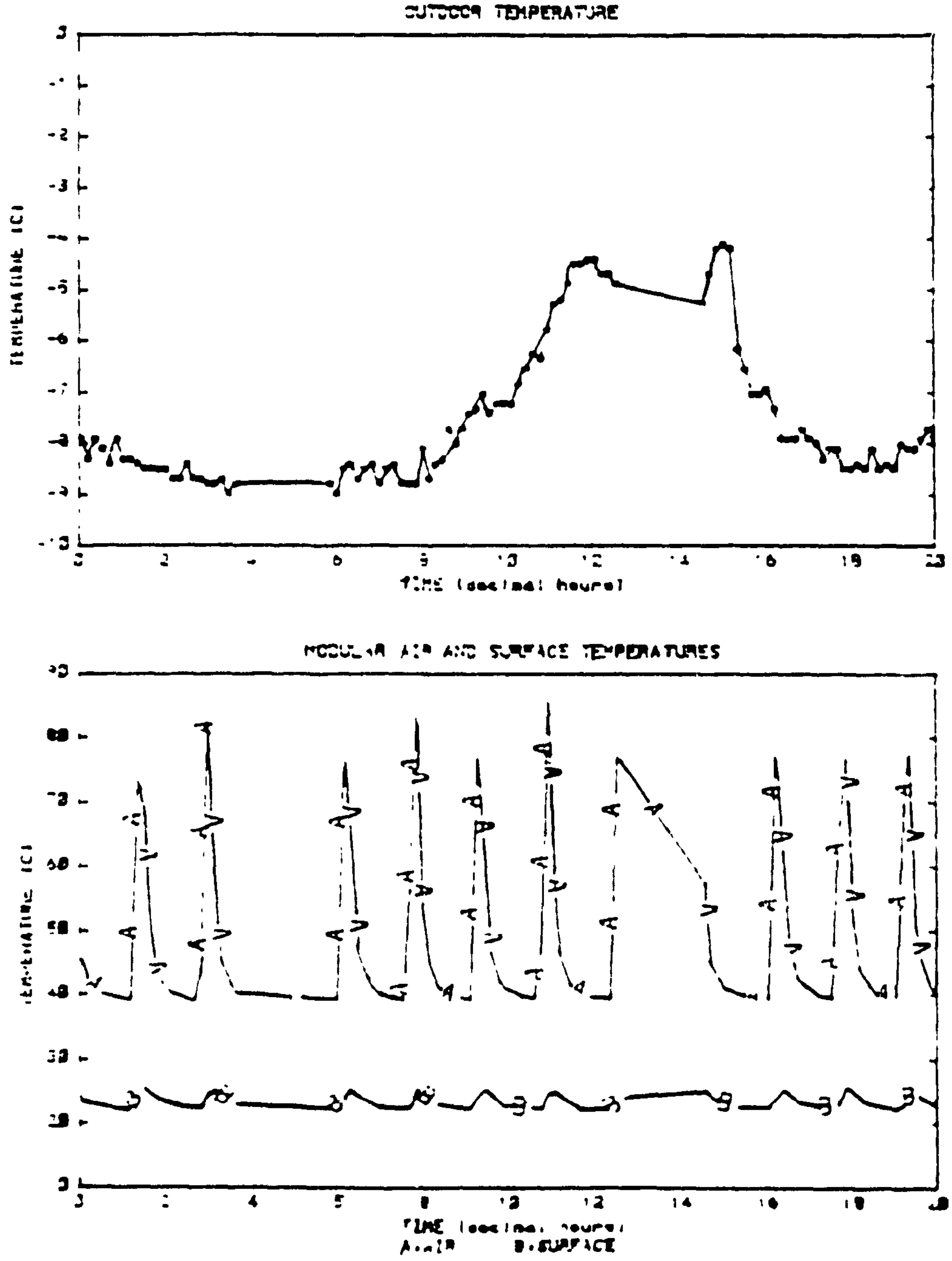


Figure 4.14 Case 2, Modular 3rd Floor, Outdoor and Supply Temperatures Variations

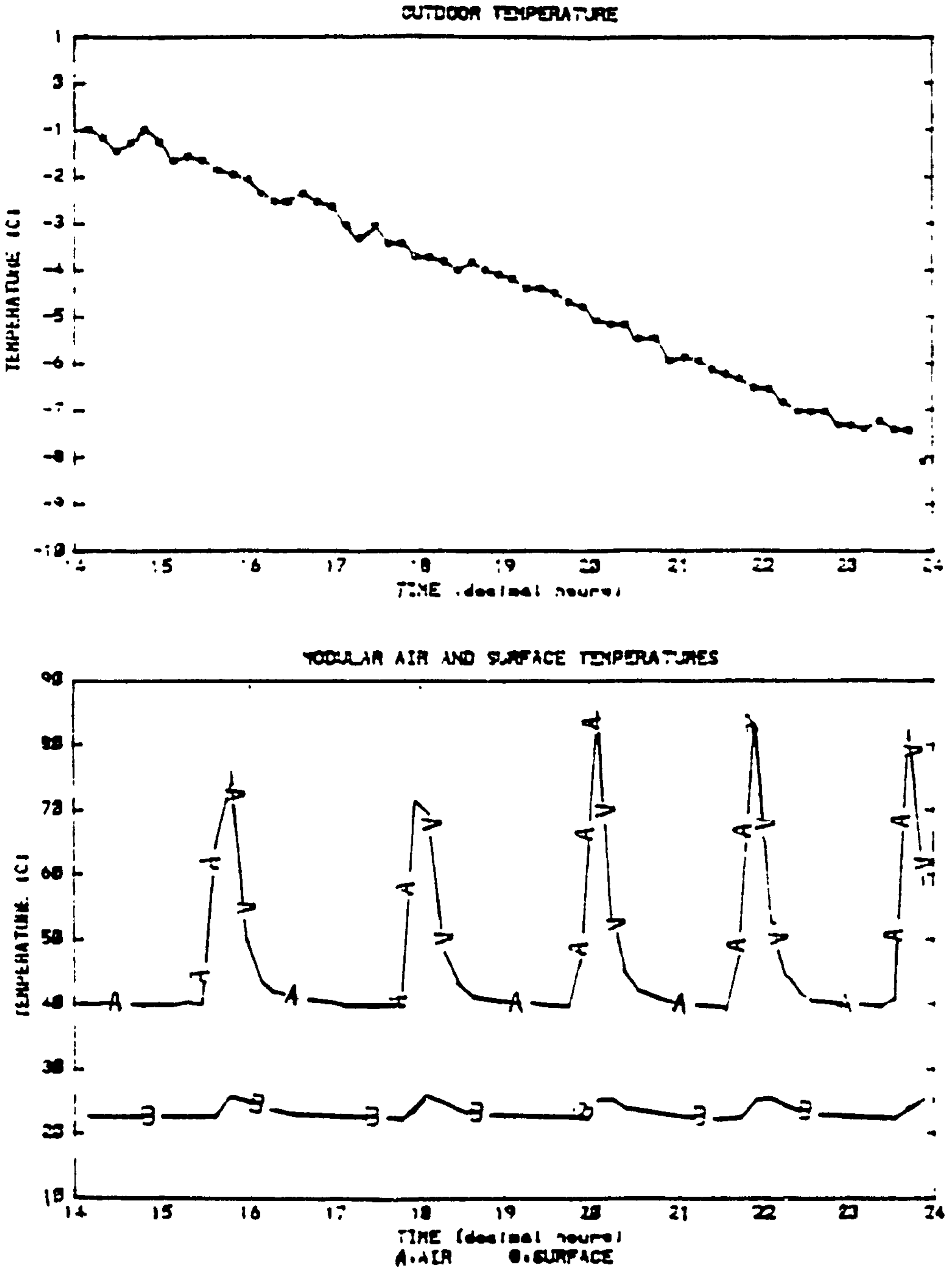


Figure 4.15: Case 3, Modular 3rd Floor, Outdoor and Supply Temperatures Variations

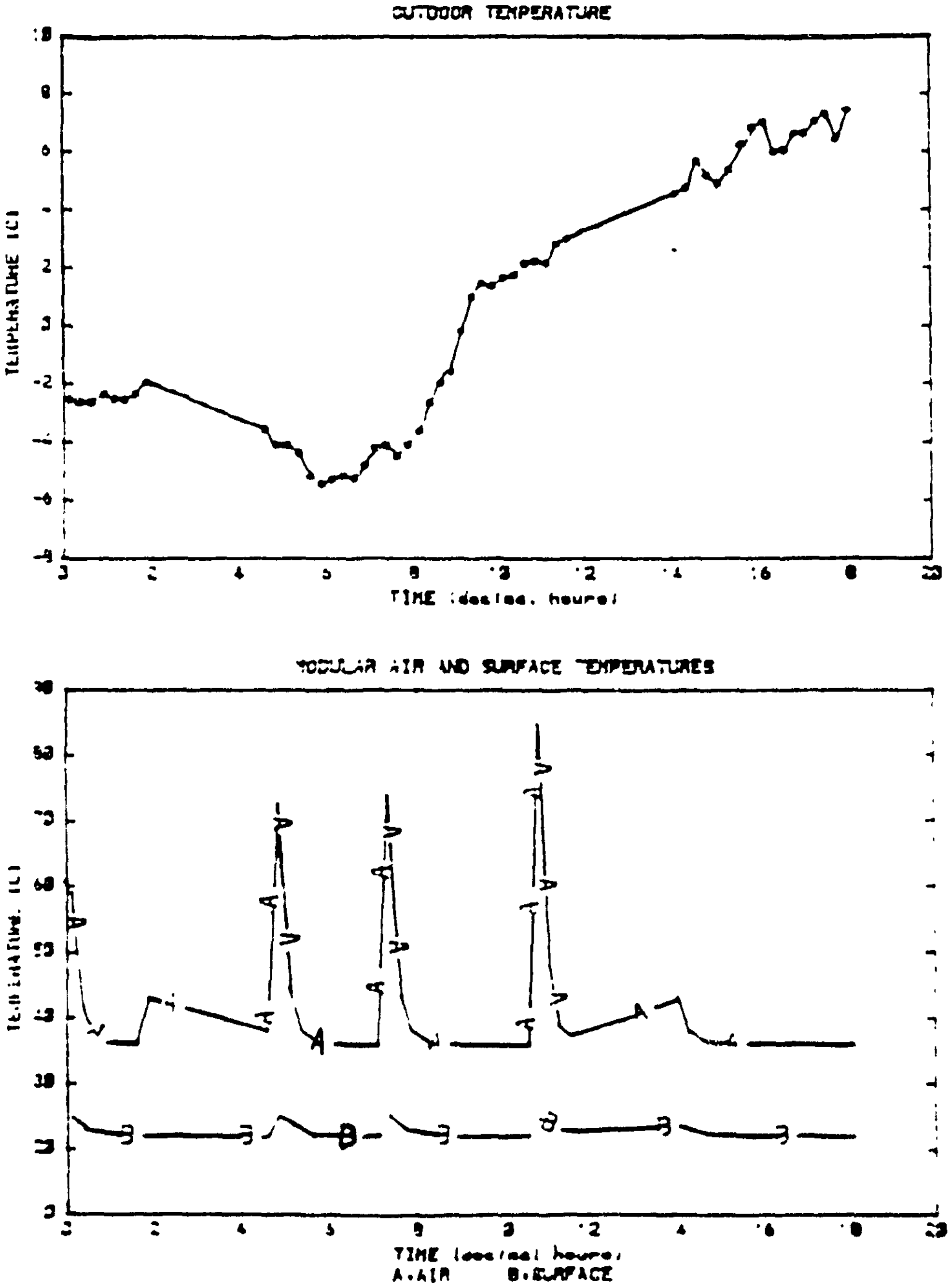


Figure 4.16 Case 4, Modular 3rd Floor, Outdoor and Supply Temperatures Variations

One difference, however, is the lesser number of cycles on the third floor than on the first floor. Note however that at the time of tests for the modular on the third floor, a certain quantity of heat came from the baseboard of the hydronic system. Even though the baseboard was insulated and the valves controlling the flow of water to the fins were closed, the return pipe located above the fins remained at a high temperature (refer to section 3.3 for baseboard description). This caused the heating load in the room to be reduced and consequently the number of cycles was less.

Another difference between the behaviour of the modular system on the third floor and the one on the first floor is the maximum temperature of the air supplied by the system. For the system on the first floor, the modular system operated between minimum and maximum temperatures which remained fairly constant throughout test periods. On the third floor, the maximum air supply temperature varied (peak of each cycle).

From Figure 4.13, the peak temperature was close to 80°C for the first three cycles, while it was around 70°C for the two last cycles. The same variation in peak air supply temperature occurred but was less marked for the modular system on the first floor.

The variation in peak air supply temperature for the modular system is typical of this system and relates to the information of the manufacturer on the ability of the system to modulate its output for 0% or between 30% and 100% of the maximum. In the cases presented, this system functioned more often between a high minimum temperature (~35°C) and a peak temperature (~85°C) without ever shutting itself off completely.

From the discussion, the modular system can be said to produce cycles which are consistently repeated for the different outdoor conditions observed and it operates at relatively high air temperatures of delivery. The system worked at high minimum temperatures with a number of cycles per hour less than one. The results were comparable for both floors.

4.3.3 Summary and Comparison of Systems' Cycling

So far, the systems' operation were discussed separately and in order to compare them. Parameters mentioned earlier have been grouped in Table 4.6. A few cycles for each system are shown on Figure 4.17 for comparison.

From the point of view of the range of discharge temperatures, each of the three systems occupy a category of its own where forced air, hydronic and modular systems can be said to respectively belong to low, medium and high discharge temperature categories. It is interesting to note also that the range of one category more or less ends where the next category begins.

For the hydronic and modular systems, the average surface temperature (important for the radiative component of heat transfer) are similar. This similarity exists for the range between which the surfaces' temperatures vary for both systems as well as the average surfaces temperatures. This result is surprising considering the much higher temperatures at which the modular systems operate. The cover on the front of the heat exchanger of the modular system is therefore effective in significantly reducing surface temperatures with its low conductivity plastic material and the use of reflective material on the inside.

Table 4 6: Comparison of Systems' Statistics

SYSTEM	Average Discharge Temperature (°C)	Discharge Temperature Range (°C)	Average Surface Temperature (°C)	Average Number of Cycles' Range
Forced Air	28	20 to 34	-----	0.7 to 2.2
Hydronic	30	26 to 50	25	0.6 to 2.9
Modular	50	45 to 80	24	0.4 to 0.8

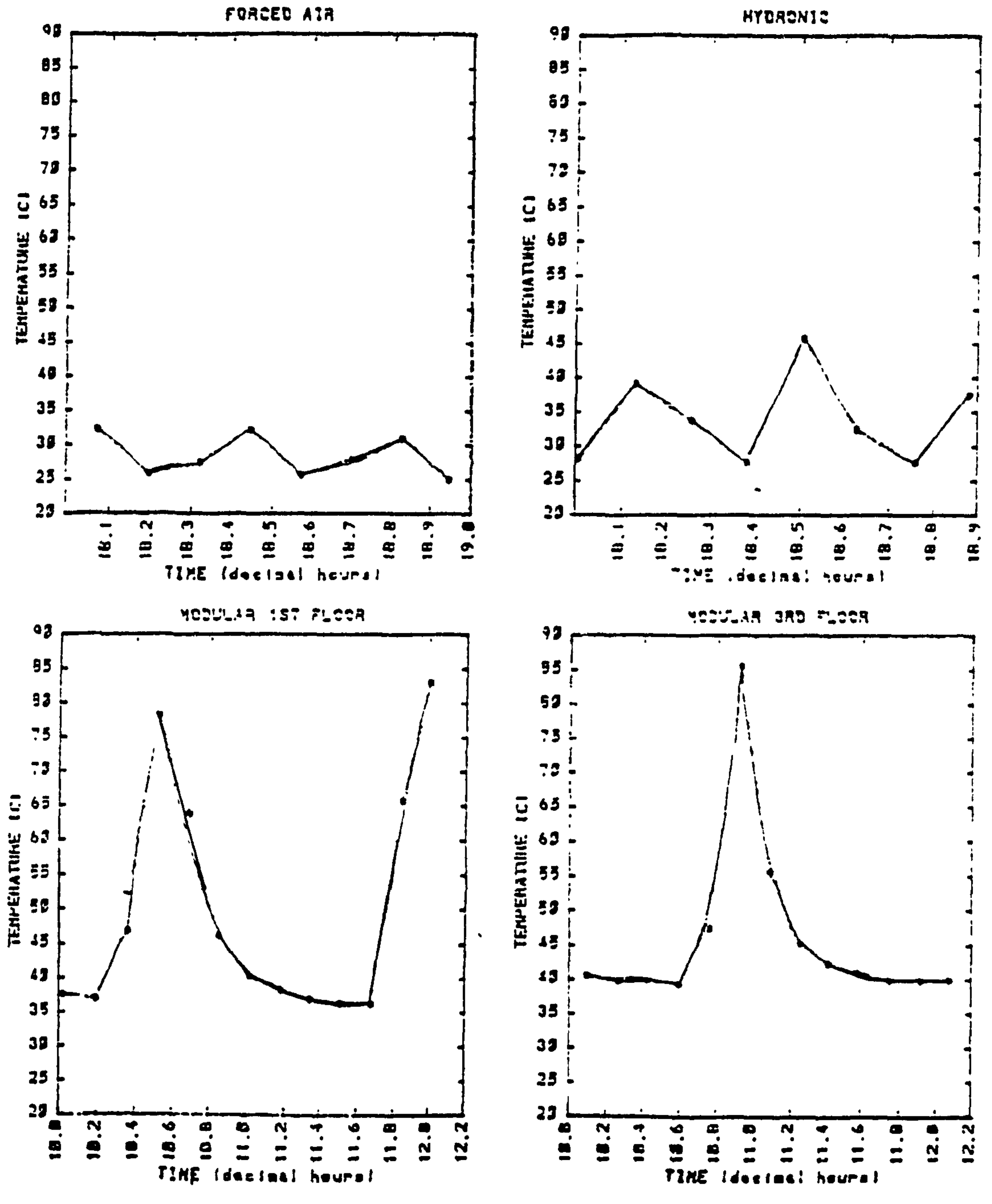


Figure 4.17: Comparison of Systems' Temperature Cycling Over a Short Time Period

In terms of cycling patterns, the modular and forced air systems provide cycling patterns which are repetitive for a given period with well defined maximum and minimum temperature supplied. The main difference is that the modular system is characterized by high rises followed by slow decreases in supply, while the forced air system produces sharp increases and sharp decreases in supply temperatures.

The hydronic system's supply temperature cycles were less well defined than for the other systems and had its cycling behaviour influenced by outdoor temperature conditions. For low outdoor temperatures, the supply temperature of this system cycled between high, low and medium values while for milder outdoor temperatures, the system cycled between high and low values only. This difference is due to the inertia involved in heating and circulating hot water in the system which is not as important for the forced air system and even less important for the modular system

The number of cycles per hour for the hydronic and the forced air systems are on the average higher than for the modular. This is due to the fact that a cycle for the modular system is not the same as a cycle for the other two systems and that the modular system has a high power output for the loads in the room The modular system is found to constantly deliver heat to the room and therefore does not need to undergo as many "cycles" as the other systems. It is surprising that for the case of rising outdoor temperatures (Figure 4.16, case 4), the minimum temperature of the modular system on the third floor did not drop after the outdoor temperature was well above 8°C. This suggests a possible problem of overheating due to inadequate control from the temperature sensor controlling the modular although we did not investigate here the conditions elsewhere in the room.

In the present section, the operation of the systems have been studied and classified for different outdoor temperatures values and it was shown that the three systems did not operate in the same fashion when cycling is concerned.

With the results from this section, the behaviour of the systems were separated from their effect on the environment. In the following sections, the conditions created in the environment by the different systems in terms of air temperature distributions and variations are discussed.

4.4 Temperature Distribution

In the preceding section, the operation of the systems were described in terms of their cycling over a certain period of time limiting ourselves to the mechanical components and neglecting their effect on air temperature distribution and variations in the rooms. In the present section, we will first describe the type of temperature distribution produced by the systems. Some comments are made on the limitations in the calculation procedure required. Then, the air temperature fluctuations and the surface temperature distributions will be discussed.

The analysis for plane temperature distribution will be performed for cases no.2 discussed in the preceding section. The temperature distribution produced will also be discussed for the whole heating season. Note that temperature profiles will be described in more details in chapter 5.

4.4.1 Description of Temperature Distribution Using Isotherms

In order to obtain a description of the temperature distribution in the rooms isotherms (or isothermal surfaces) can be approximated for certain planes formed by the intersection of three locations on the 3x3 grid of thermocouples (see figure 4.18). To describe the temperatures throughout the room, the isotherms will be shown for five different planes for each system: three horizontal (planes i,j and k) and two vertical (planes 2 and 5).

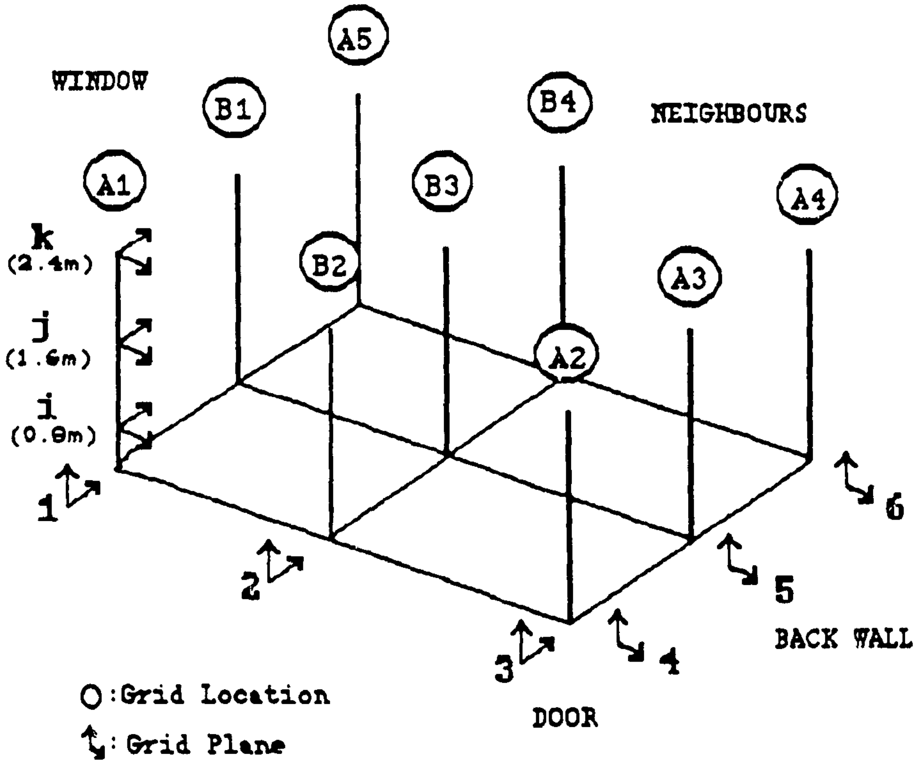


Figure 4.18: Reference Names for Grid Planes Labeling

The analysis of plane temperature distribution is somewhat limited by the experimental setup in terms of the location of the peripheral points of the thermocouple grid close to the walls of the room and, the distance (resolution) between grid points.

The peripheral grid locations (all locations except B3) were located close to the walls of the room (see Figures 3.6 and 3.8) where specific boundary conditions occur [25]. For example, the grid locations close to the neighbour wall (locations B4, A4 and A5) were located only 40cm away from this wall. Air currents flowing close to this wall are expected to be different from those found deeper into the room. This region can be considered as a transition zone between the conditions at the surface of the wall and in the rest of the room. Since all the peripheral grid locations are similarly located, using temperature measurements for these locations to interpolate the room temperature distribution limits the results obtained for room temperature distribution.

In addition to the use of the peripheral grid locations, temperature distributions must be interpolated between grid locations located 1.2m and 1.5m apart and with these two factors combined together, the accuracy of the interpolations is further reduced. It would have been necessary to use a certain number of additional temperature measurements close to the walls and in between grid locations to determine the limits of the transition zone.

Taking these limitations into consideration we can nevertheless obtain a certain representation of room temperature distributions using isothermal curves and isothermal surfaces which show the variations of temperature throughout the space studied. The isotherms were obtained using a graphics software [26] which performed the interpolation of the different isotherms given the data obtained for each case. The results from the isotherms represent some of the possible contour lines that can be obtained from the data. A more detailed approach to interpolation of isothermal curves was not required since only the relative differences between the systems was sought.

As mentioned earlier, the temperature distribution is expected to vary within a given time period and also during the whole heating season. It is not practical however, to plot isotherms for every single time a number of temperatures were measured and repeat this for all the days when observations were made. Only the average distribution for a selected day was investigated.

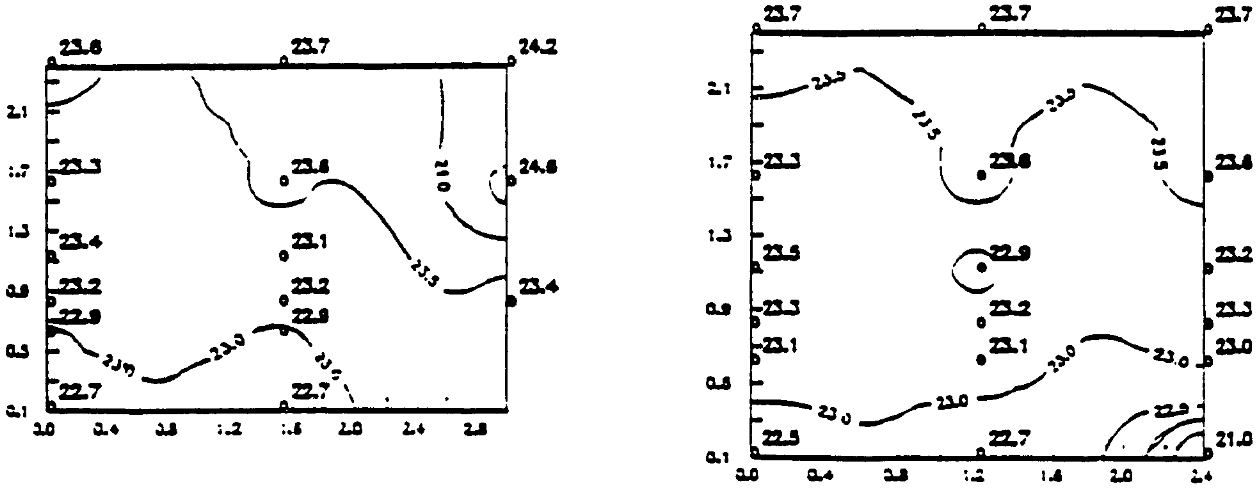
The day selected corresponds to case no.2 already described in the previous section. The data for the heaters' temperature as well as the outdoor temperature for this case were listed and plotted in the previous section. We therefore ask the reader to refer to Tables 4.2 to 4.5 for the forced air, the hydronic and the modular systems respectively. The heaters' temperatures variations and the outdoor temperatures appear in Figures 4.4, 4.8, 4.11 and 4.14 for the forced air, the hydronic and the modular systems respectively. The second case was selected since relatively cold outdoor temperatures prevailed for all systems, which also allows for comparison between the systems

Forced Air System

Isotherms and isothermal surfaces for the forced air system are displayed on Figure 4.19 and 4.21 for values of temperatures averaged over the collection period (case 2 of previous section). The measurement points used for the isotherms are shown on the graphs and the contour interval was selected to be 0.5°C. Isotherms for planes 5 (longitudinal) and 2 (cross section) are shown in Figure 4.19 a) and b) respectively and in Figure 4.21 a), isothermal surfaces are shown for planes i, j and k.

For the plane in the longitudinal direction of the room, (Figure 4.19 a) the forced air created a distribution with air temperatures generally increasing from the lower left corner (front of window. 22.7°C) to the upper right corner (back wall: 24.2°C). The difference between the coolest and warmest points is approximately 1.5°C which is relatively small

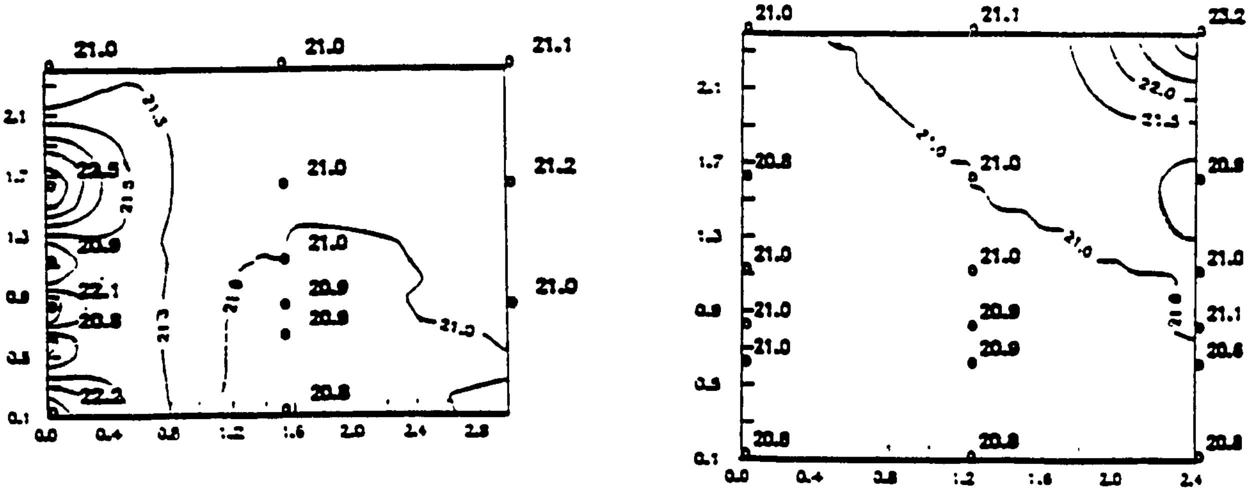
Figure 4.19-b indicates a fairly symmetric distribution in the cross section of the room. Decreases in temperature occur close to the lower side of the neighbour's wall (temperature of 21°C compared 23.5°C close to the ceiling).



a) plane 5, longitudinal

b) plane 2, Cross Section

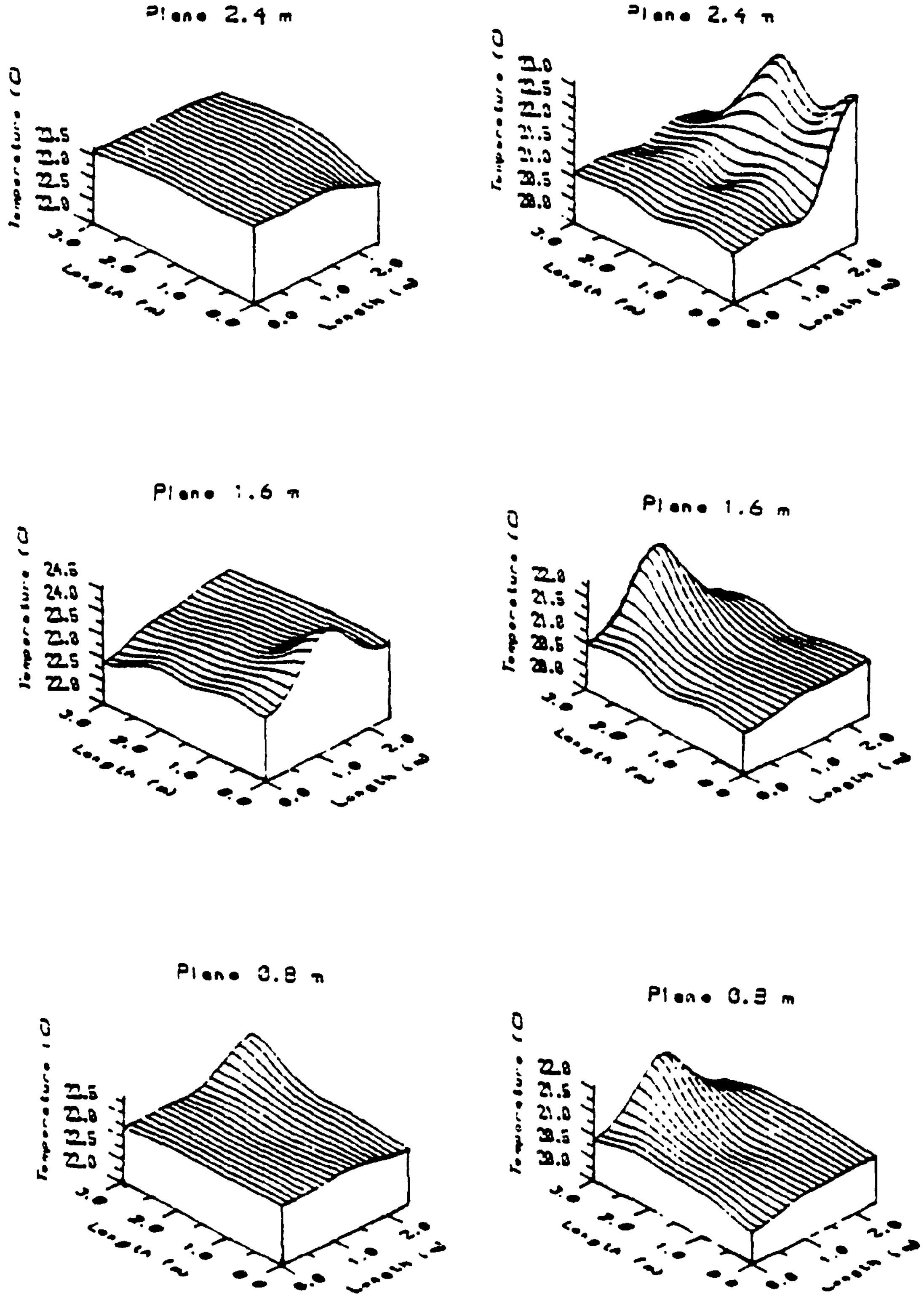
Figure 4.19: Examples of Isotherms, Forced Air (from data for case 2)



a) plane 5, longitudinal

b) plane 2, Cross Section

Figure 4.20: Examples of Isotherms, Hydronic System (from data for case 2)



a) Forced Air

b) Hydronic

Figure 4.21: Examples of Horizontal Plane Temperatures Distributions, Forced and Hydronic Systems (from data for case 2)

From Figure 4.21-a, we can see the distribution of temperatures on horizontal planes at 0.8, 1.6 and 2.4 m above the floor. According to this figure, the variations of temperature within a plane are found to be small. The location of the major variations are not the same for each plane but they all tend to appear at locations close to the neighbour wall and the back wall. The maximum range between maximum and minimum for a given plane is less than 1°C except for plane at 1.6 m where the range is higher (2°C) considering the location close to the back wall.

If the planes shown in Figures 4.19 and 4.21 a) are considered together, the distribution observed for the vertical planes is reflected in the horizontal planes as well. The most significant planes are those of Figure 4.19 which show the increase in temperature from the front to the back and, the symmetry in temperatures with respect to either side of the room.

The isotherms of Figure 4.19 a) compare well with the results obtained by Michell and Biggs for a forced air system with floor diffuser [14], and for a larger number of points measured The same type of stratification in the longitudinal direction were observed, although the system they tested had a slightly higher supply temperature (40°C).

Hydronic System

The isotherms for the hydronic system were plotted in the same manner as for the forced air system and the graphs appear in Figure 4.20 and 4.21-b. Note that for plane Figure 4.20-a, the contour interval selected was smaller (0.2°C) to better show the thermal situation next to the heater.

From Figure 4.20-a, the air temperature distribution is characterized by two main zones if for the vertical plane perpendicular to the heater. The first zone is located near the heater and is characterized by large changes in temperatures with height. In this zone, there are three maximums around 22°C and two minimums around 21°C for differences of more than 1°C at several points. The second zone can be identified by a relatively uniform mass of air at constant temperature (21°C) from the center of the room to the back. In this zone, temperatures remain fairly constant throughout with almost no change in height or in other directions.

Note that the two zones of Figure 4.20-a are separated by a transition zone (contour at 21.3°C) which is located in the range between 0.5 m and 1.0 m away from the column nearest to the heater.

The variations in temperatures for the different horizontal planes (Figure 4.21-b) indicates that temperatures remain uniform within the room except at the location close to the heater and for the 2 4 m plane, a mass of warm air is located near the neighbour's wall. This mass of air is not present at the other levels and may be due to gains from the neighbours although there were no thermocouples on the wall at that location to confirm this The other levels reflect mostly the significant effect of the heater on the temperature distribution where the greatest variations occurs at the 1 6 m plane and are not apparent at the 2 4 m plane

Results published by Michell and Biggs [14] were similar to ours for the isotherms created by what they call a "hot water panel heater" and for a room approximately twice as long as ours The shape of their isotherms reflected the same distribution with two zones except that the variations observed above the heater were not as wide

Modular Systems

The isotherms for the modular systems on the first and the third floor were plotted in Figures 4 22 to 4.24 and the order of the graphs is the same as was used for the other systems

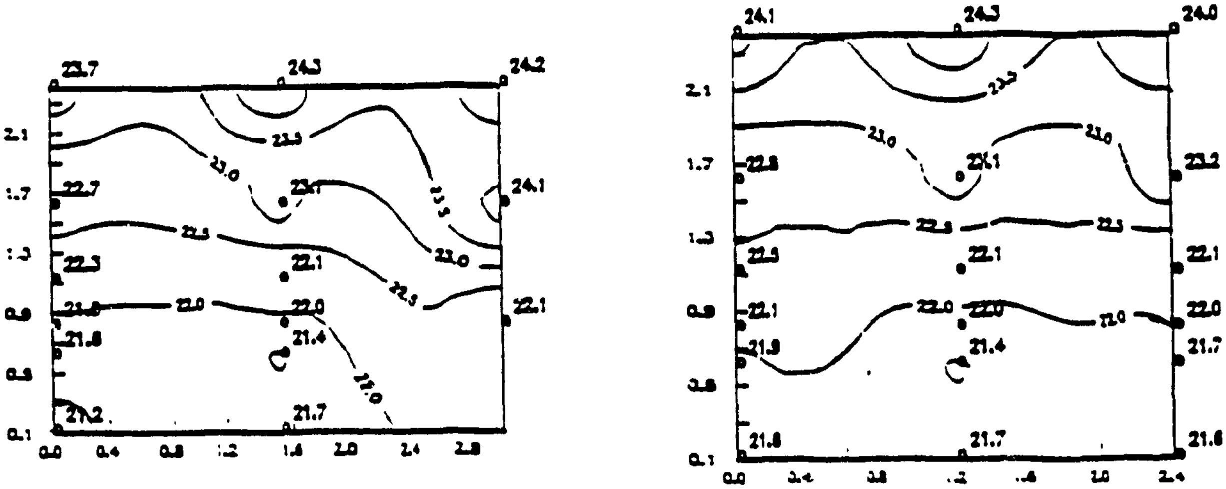
The temperature distribution for the modular systems is similar for both floors when Figures for the vertical planes (Figure 4 22 and 4 23) are compared Differences occur in the magnitude of the temperatures with temperatures on the third floor lower than on the first floor Also with only small differences, the temperature variations within the space occur (the contour lines), at the same locations. A major difference is seen on Figure 4.23-a for the location near the hydronic heater where, at the 0.8 m level, the effect of the heat provided by the return pipe of the hydronic radiator is increasing the temperature.

Figures 4.22 and 4.23, describe the average environment produced by the modular systems as being similar to that of the forced air system in terms of the distribution of temperatures along the vertical plane perpendicular to the window. The temperatures increase from the lower left portion of the graph (location close to the window) and increase progressively in the direction towards the upper right (ceiling at the back of the room). However, the increase in temperature from one point to the other is more important than for the forced air system.

For the temperature distribution at the vertical plane in cross section at the center of the room (Figures 4.22-b and 4.23-b), in both cases, the effect of the heater is to produce a fairly symmetric distribution with respect to the center of the room. This is slightly less evident for the modular system on the third floor. The symmetry would have been expected to be less than for the other two systems with the modular systems located on one side of the room.

Using the graphs of the temperature distribution over horizontal planes (Figure 4.24) the temperature distribution of the two modular systems with height can be compared.

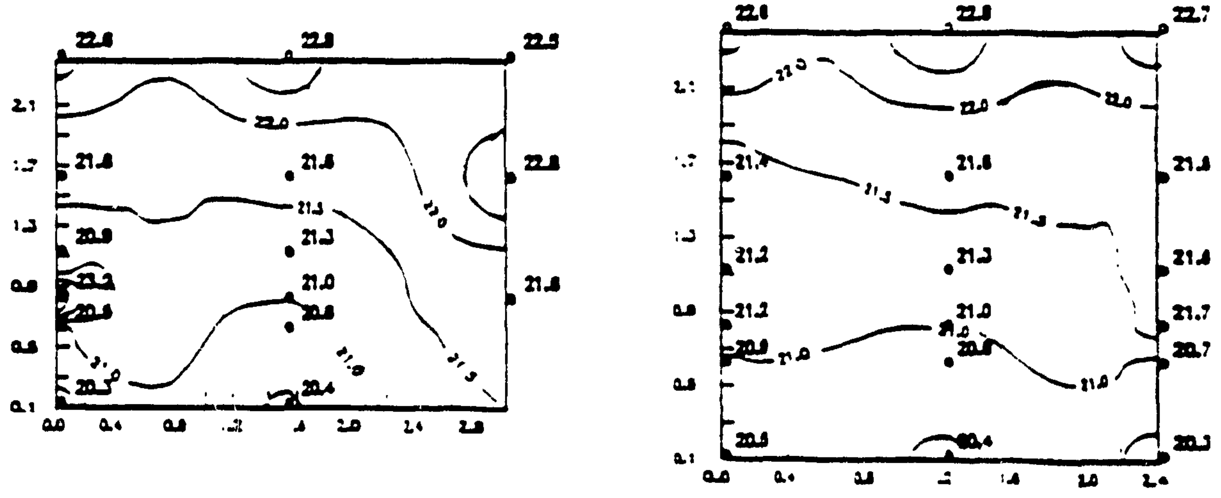
From Figure 4.24-b, the influence of the insulated baseboard of the hydronic system at the 0.8 m level where increases temperatures close to 2°C occur. This increase can be attributed to the combined effect of the hydronic and the modular system working at the same time since similar increases were not observed on the first floor (Figure 4.24-a).



a) plane 5, longitudinal

b) plane 2, Cross Section

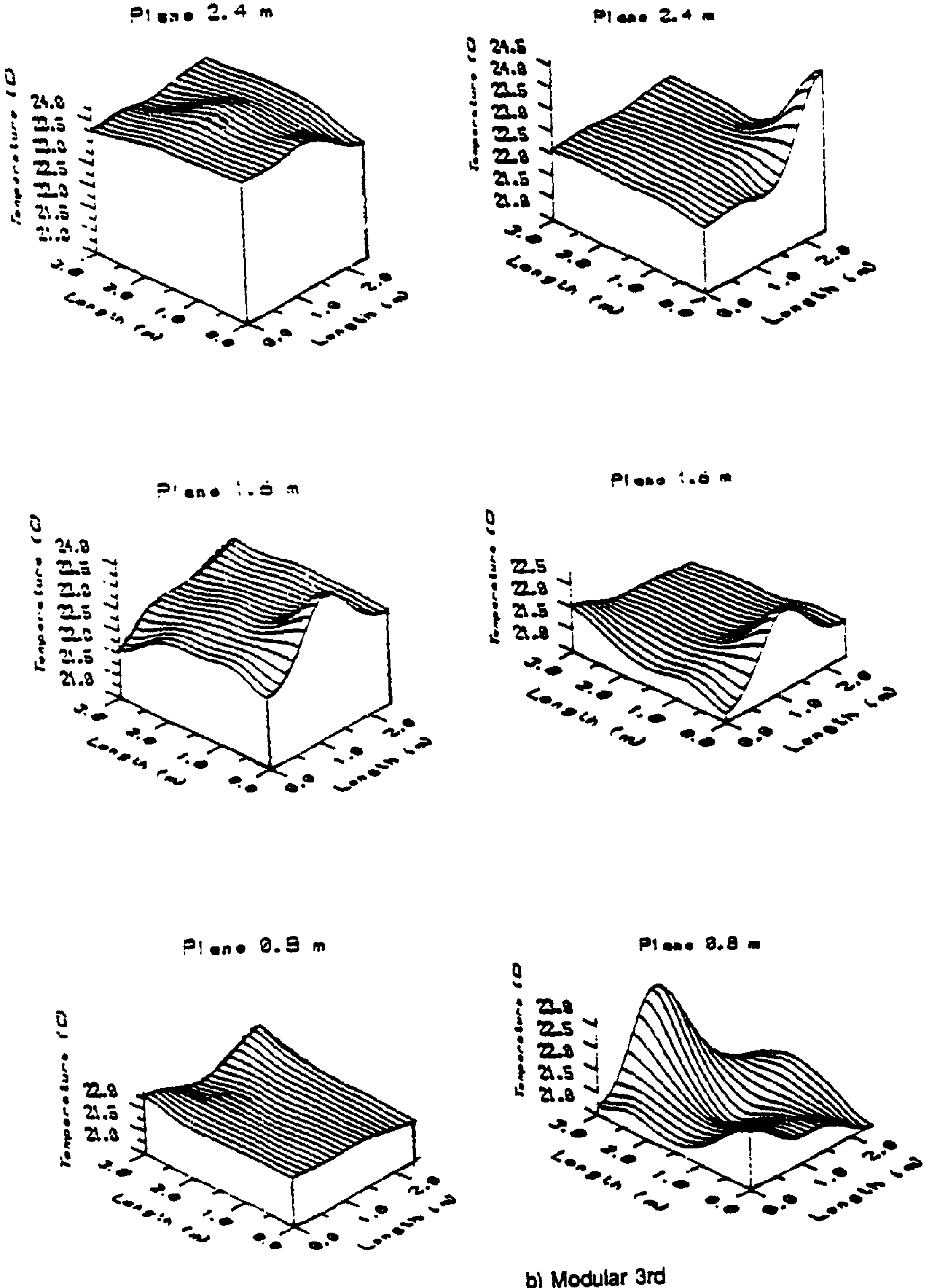
Figure 4.22: Examples of Isotherms, Modular 1st Floor (from data for case 2)



a) plane 5, longitudinal

b) plane 2, Cross Section

Figure 4.23: Examples of Isotherms, Modular 3rd Floor (from data for case 2)



a) Modular 1st

b) Modular 3rd

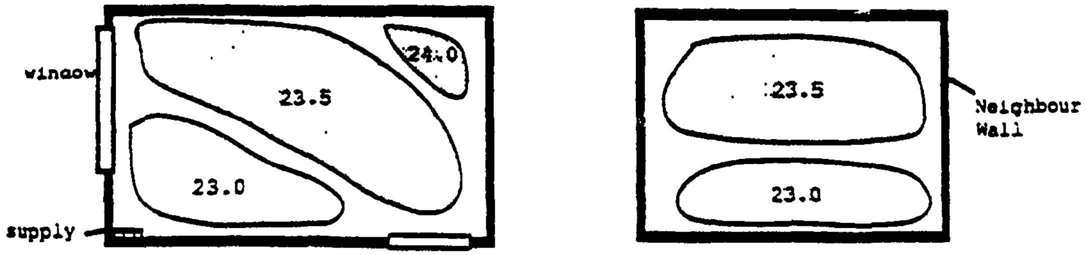
Figure 4.24: Examples of Horizontal Planes Temperatures Distribution, Modular Systems, 1st and 3rd Floor (from data for case 2)

For the other planes, the temperature within a plane remains fairly constant throughout. For the 1.6 m plane (Figure 4.24), the maximum temperature occurs at the back of the room (location A3) which was also the case for the hydronic system (Figure 4.21-b). For the 2.4 m plane, the modular system on the first floor shows no significant variation in temperatures for the 2.4 m plane (Figure 4.24-a) while on the third floor, a major increase in temperature occurs in the region close to the back wall (Figure 4.24-b). Note that for the hydronic system, such an increase near the back wall was also observed (Figure 4.21-b) and therefore, the increase in temperature in that region may not be specifically attributed to the effect of the system, but rather from the effect of a probable source of heat from the neighbour wall or the region of the adjacent bathroom.

Summary

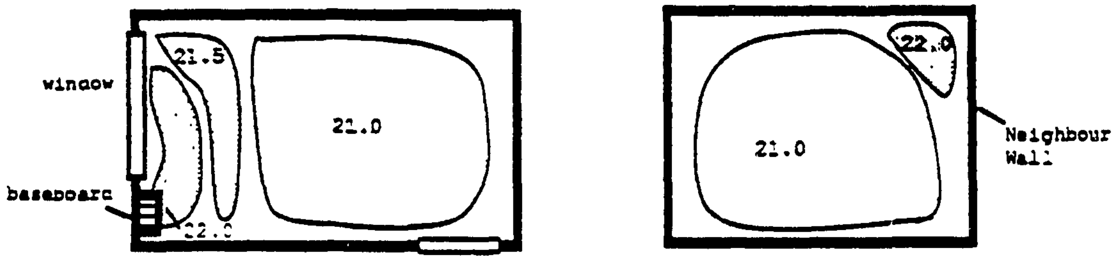
From the above discussion, it was possible to describe a general temperature distribution which can be attributed to each system. However, in order to better compare them, the graphs for planes 5 and 2 obtained earlier were simplified in Figure 4.25. This figure represents what we think is a reasonable version of the isothermal plots obtained earlier for each system and convert the temperature distributions in the rooms in terms of isothermal masses of air.

From Figure 4.25, the modular system provides a highly stratified temperature distribution within the room compared to the forced air and hydronic systems indicated by the larger number of masses of air. This is accompanied by regions of higher temperatures in a direction close to the ceiling and the deeper into the room. The difference in temperature along the plane parallel to the heater shows symmetric increases in temperatures with respect to either sides of the room.



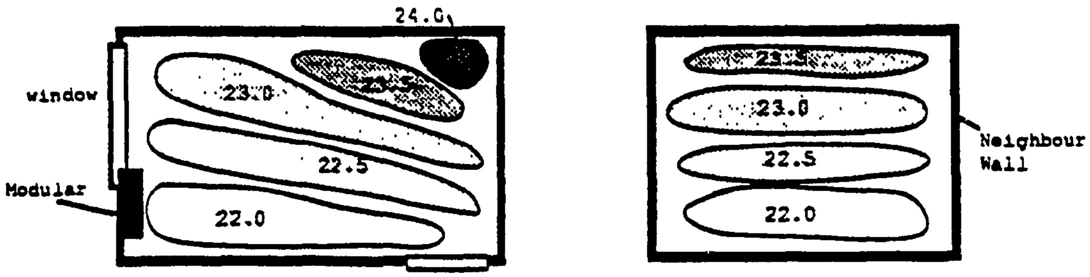
a) Forced Air, Longitudinal

b) Forced Air, Cross Section



c) Hydronic, Longitudinal

d) Hydronic, Cross Section



e) Modular, Longitudinal

f) Modular, Cross Section

Figure 4.25: Simplified Vertical Plane Temperature Distributions (°C), All Systems (case 2)

The forced air system also produces regions of temperatures increasing from the window to the back wall's ceiling (Figure 4.25-a), but the difference between the temperature of different regions is smaller than for the case of the modular system and the sizes of the masses of air with constant temperatures are larger. The temperature increase for the vertical plane parallel to the window wall (Figure 4.25-b) is symmetrical with respect to both sides of the room.

The hydronic system produces a temperatures distribution which is slightly different from the other systems. The temperatures in a plane perpendicular to the window wall (longitudinal section, Figure 4.25-c), the temperature distribution is characterized by three vertical masses of air. A warm mass occurs close to the radiator, then a slightly warmer mass of smaller size occupies the location deeper into the room and the rest of the room consists of a large mass of air with uniform temperature. For the plane parallel to the window wall and at the center of the room (cross section, Figure 4.25-d) a large mass of air occupies the whole plane with a smaller one on the upper right side (close to the neighbour wall).

In terms of air temperature distribution, the hydronic system provides the most uniform distribution both in the vertical and longitudinal directions neglecting the high temperature mass of air close to the heater. Nevertheless, since this mass of high temperature air occupies approximately 1.0 m of the room in the longitudinal direction, then this type of distribution can be classified as less desirable than for the forced air system.

Even though the forced air system produces slight gradients in the longitudinal direction, the variations in temperature from point to point are small. Because it does not produce a marked zone with high temperatures at the location close to the supply, the distribution it produces could be termed as being more acceptable than for the hydronic system.

The modular system produces a temperature distribution which is similar to the forced air system except that the differences in the temperature in both the longitudinal and vertical directions are higher. Accepting the hydronic system's concentrated mass of air at the radiator, then, the modular system ranks last in terms of uniformity. It is interesting to note that from the distribution of temperatures on the horizontal plans described before, no significant increase in temperature was found to occur near the supply of the modular. This indicates that most of the warm air produced by the modular system is moved away from the immediate surroundings of the heater.

So far, the distribution of temperature in the rooms which were described are only considering average distribution for a single period. It is not practical to cover all cases of observations which occurred during the heating season and present here the different cases. We must mention that the room air temperature distributions described here have been observed for other cases, but with specific variations occurring from case to case. The analysis of plane temperature distribution was somewhat limited by the location of the grids of thermocouples and other cases are not discussed in detail due to those limitations.

4.4.2 Air Temperature Fluctuations

In the previous section, the temperature distributions produced by the systems was static and considered only average temperature distributions. This section first describes the time fluctuations of temperatures for the cases described in section 4.1 to treat each different cases in detailed fashion and to deal with short period intervals. The second part covers fluctuations in room air temperatures during the whole heating season for each system.

These two sections combined with the previous one will allow to verify if the average temperature distributions are typical in general. Furthermore, more elements will be added to the description of the systems in terms of the air temperature fluctuations which characterize their distribution mechanisms.

4.4.2.1 Observed Short Period Temperature Variations

In this section, the air temperature variations in time for the cases mentioned at the beginning of section 4.2 are discussed. These cases are still used here since they cover a good range of possible outdoor conditions likely to occur throughout a normal heating season.

In terms of temperature fluctuations, important points are: (1) the variations of temperature between sampling time, (2) the rate at which these variations occur, (3) the range of temperature maintained for a measurement period and, (4) the difference between the values of (1),(2) and 3 at different height.

For the variation of temperature in time, the change in temperature between two data collection times can be used to estimate the change in temperature in time for a given system (ΔTa) for short intervals and the rate of temperature change ($\Delta Ta/\Delta t$). Negative and positive rates were observed to be of the same magnitude in the data, and only the absolute values of these variations are considered.

The range of air temperature for the whole period for the different cases can be calculated by taking the difference between maximum and minimum observed values to indicate how well the systems are able to maintain a constant temperature within the rooms. The actual value of the temperatures maintained (setting) is of lesser importance. The systems, the outdoor conditions and the enclosure properties are different in each case. The range is more important since it relates to relative variations instead of an absolute value.

One location, location B3 was selected as being representative of the center of the room and for this location two levels, 0.1 and 1.1 m, were selected. This location is only used as a reference. The air temperature variations will be discussed later for other locations.

In the following, the results from these calculations were made for all three types of systems and for all the cases listed in section 4.2. The results for the forced air, the hydronic and the modular systems discussed separately and then compared.

Forced Air System

The statistics for air temperature fluctuations at location B3 (0.1m and 1.1m heights) for the forced air system appear in Table 4.7.

i) Variations between sampling times: The maximum temperature change between two successive readings (ΔT) was always less than 1.0°C and on the average it was less than 0.4°C for all four cases.

ii) Rates of temperature Change: Rates of temperature changes (ΔT/Δt) were as high as 6.3°C/h (case 2) with average rates between 0.7°C/h and 3.0°C/h. The highest rates of temperature fluctuations were observed for cases 2 and 3 (near 3°C/h average, for 1.1m height) compared to the other cases (less than 1.1°C/h).

iii) Period ranges: The temperatures maintained during the period of the tests did not vary by more than 2°C for all the cases and, if we exclude case 4, then the temperatures range observed for the periods remained below 1.1°C, a relatively small value. The forced air system was less efficient in maintaining constant temperatures for case 4 where outdoor temperatures dropped at a high rate.

iv) Differences between 0.1 and 1.1 m level: For the four cases, the fluctuations observed were generally higher at the 1.1 m level than at the 0.1 m level and temperature maintained at the 0.1m level were on the average the same as for the 1.1m level.

Table 4.7: Temperature Fluctuations Statistics, Forced Air (Location B3)

Case	1		2		3		4	
Level (m)	0.1	1.1	0.1	1.1	0.1	1.1	0.1	1.1
ΔT max (°C)	0.4	0.6	0.5	0.8	0.5	1.0	0.4	0.7
ΔT/Δtmax (°C/h)	1.6	1.1	3.9	6.3	1.9	5.5	1.6	2.7
ΔT avg (°C)	0.3	0.3	0.4	0.4	0.2	0.4	0.2	0.3
ΔT/Δtavg (°C/h)	1.1	1.1	1.4	3.0	2.7	3.0	0.7	1.1
ΔT avg (°C)	23.1	23.3	22.7	23.0	22.7	23.0	21.8	21.9
T range (°C)	0.7	0.7	0.5	1.0	0.8	1.1	0.8	1.9

Hydronic System

The statistics for air temperature fluctuations at location B3 (0.1m and 1.1m heights) for the hydronic system appear in Table 4.8.

i) Variations between sampling times: The maximum temperature change between two successive readings was always less than 1.0°C with averages less than 0.5°C.

ii) Rates of temperature Change: The room temperature fluctuations observed for the three cases show that rates of temperature changes are less than 3.5°C/h with average rates between 0.7°C/h and 2.0°C/h.

iii) Period ranges: For all cases, the average temperatures at B3 were near 21°C and varied by less than 1.0°C.

iv) Differences between 0.1 and 1.1 m level: For the cases listed, rates of temperature change are equal or slightly higher at the 1.1 m level than at the 0.1 m level. Note that temperature range at the 1.1 m level were higher for the low outdoor temperature case (case 2) and indicates that as the outdoor temperatures became lower, the temperatures in the room varied over a wider range.

Table 4.8: Temperature Fluctuations Statistics, Hydronic (Location B3)

Case	1		2		3	
Level (m)	0.1	1.1	0.1	1.1	0.1	1.1
ΔT max (°C)	0.3	0.4	0.2	0.8	0.3	0.4
ΔT/Δtmax (°C/h)	1.2	1.6	1.6	1.9	2.3	3.1
ΔT avg (°C)	0.1	0.2	0.1	0.5	0.2	0.2
ΔT/Δtavg (°C/h)	0.7	0.8	1.0	2.0	1.4	1.8
Δ T avg (°C)	20.9	21.0	20.9	21.0	20.4	20.6
T range (°C)	0.5	0.4	0.3	0.9	0.6	0.5

Modular Systems

The statistics for air temperature fluctuations at location B3 (0.1m and 1.1m heights) for the modular system appear in Table 4.9 and 4.10 for the first and third floor respectively.

i) Variations between sampling times: The maximum temperature change between two successive readings was always less than 1.0°C with averages less than 0.5°C.

ii) Rates of temperature Change: For the modular system on the first floor, the room temperature fluctuations observed for the three cases mentioned earlier show that rates of temperature changes are less than 5°C/h with average rates between 1.0°C/h and 1.6°C/h. For the modular system on the third floor, results are similar with temperature changes less than 5°C/h with average rates between 1.0°C/h and 2.0°C/h.

iii) Period ranges: The air temperatures at B3 were maintained slightly lower on the third floor (~21°C) than for the first floor (~22°C) and, for all cases and both floors, the average temperatures varied by less than 1.2°C. The range of temperature variations remained more or less constant with differences less than 0.6°C from case to case.

Similar to the forced air system, the range of temperature for the periods was found to be higher for the case of the dropping outdoor air (case 4).

iv) Differences between 0.1 and 1.1 m level: For the cases listed, the average rates of temperature changes are equal or slightly higher at the 1.1 m level than at the 0.1 m level.

Table 4.9: Temperature Fluctuations Statistics, Modular 1st Floor (Location B3)

Case	1		2		4	
Level (m)	0.1	1.1	0.1	1.1	0.1	1.1
ΔT max (°C)	0.7	0.6	0.8	0.6	0.3	0.6
$\Delta T/\Delta t$max (°C/h)	1.3	1.6	4.7	3.6	1.1	3.6
ΔT avg (°C)	0.2	0.3	0.2	0.2	0.2	0.3
$\Delta T/\Delta t$ avg(°C/h)	1.3	1.5	1.4	1.5	1.0	1.6
ΔT avg (°C)	21.9	22.1	21.8	22.1	21.8	22.1
T range (°C)	0.9	0.9	0.9	0.8	0.6	1.0

Table 4.10: Temperature Fluctuations Statistics, Modular 3rd Floor (Location B3)

Case	1		2		3		4	
Level (m)	0.1	1.1	0.1	1.1	0.1	1.1	0.1	1.1
ΔT max (°C)	0.4	0.7	0.5	0.8	0.5	0.9	0.5	0.5
$\Delta T/\Delta t$max (°C/h)	2.4	4.1	3.0	4.7	1.6	3.5	3.0	3.0
ΔT avg (°C)	0.2	0.2	0.2	0.3	0.2	0.3	0.2	0.3
$\Delta T/\Delta t$avg (°C/h)	1.2	1.4	1.2	1.9	0.9	1.0	1.1	1.7
ΔT avg (°C)	20.7	21.3	20.4	21.2	20.2	20.8	20.6	21.4
T range (°C)	0.7	1.0	0.9	0.9	0.6	0.9	0.7	1.2

Summary

In order to compare the systems, statistics for room air fluctuations at location B3 are summarized in Table 4.11 where the temperature fluctuations cases 1 and 2 are listed. The maximum or worse values observed for those cases and for both the 0.1m and the 1.1m level are included in the table.

Table 4.11: Comparison of Temperature Fluctuations (case 1-2, location B3 0.1m and 1.1m levels, all statistics included)

System	Forced Air	Hydronic	Modular 1	Modular 3
ΔT max (°C)	0.8	0.8	0.8	0.8
ΔT/Δtmax (°C/h)	**6.3**	1.9	4.7	4.7
Δt avg (°C)	0.4	0.5	0.3	0.3
ΔT/Δt avg(°C/h)	**3.0**	2.0	1.5	1.9
T range (°C)	0.7	0.9	0.9	1.0

The maximum and average changes in temperature between two successive sampling times for the period considered showed that all the systems were similar with maximum changes of 0.8°C and average changes less than 0.5°C. This indicates that for all systems, if periods of 15 minutes or less are considered, the temperature is not expected to vary by more than 1°C on the average in cases similar to 1 and 2.

For the rates of temperature changes, the forced air system produces peak and average rates of temperatures changes higher than the other systems (ΔT/Δt max= 6.3°C/h).

The modular systems produced rates of temperature changes which were on the average similar to the hydronic system (close to 2°C/h) but the peak rates are more than two times those of the hydronic system.

In terms of range of temperature maintained at B3, all systems maintained a constant temperature which varied by at most 1°C during the whole collection period. In this sense, the control of the systems can be said to be accurate even considering the fact that the thermostats for the hydronic and the forced air systems are not located directly inside the room. These observations, however, concern only location B3.

A general observation can be made on the range of temperature fluctuations observed for the period. For all the systems, even though case 4 for the hydronic system was not available, the range is expected to be wider for outdoor temperatures dropping (case 4).

Also, the rates of temperature variations are more a particularity of the systems than of the outdoor conditions since the rates remained constant from test to test while outdoor temperatures varied. In this sense, the higher rates for the forced air and the modular systems are expected to be always higher than those of the hydronic system. This observation could, however, be better examined by using more cases than were presented here.

In order to further describe the variations of air temperatures within the room, points with variations significantly higher than at location B3 had to be mentioned here. These points for each system are listed in Table 4.12.

For the hydronic system, at the location close to the baseboard (location B1, 1.6m level) variations of up to 2.2°C between sampling times with a maximum rate of 18°C/h were observed. As can be seen, the fluctuations increase with height. It is therefore evident that the low fluctuations in air temperatures in the rest of the room have to be separated from those of the region close to the baseboard if this system is to be compared with the others.

Table 4.12: List of Points with High Temperature Fluctuations (cases 1-2).

SYSTEMS AND LOCATIONS	ΔT max. (°C)	ΔT/Δt max. (°C/h)
Hydronic		
B1 0.6 m	± 0.9	± 7.0
B1 0.8 m	+ 1.3	+ 10.0
B1 1.6 m	+ 2.2	+ 18.0
Modular 1st floor all 2.4 m points (B2 as an example)	+ 2.1	+ 12.0
Modular 3rd floor all 2.4 m points (B2 as an example)	+ 2.9	+ 18.0

For modular systems, on both floors, all the locations close to the ceiling were found to be the most sensitive to temperature fluctuations. Changes in temperatures of up to 3°C and rates of 18°C/h were observed at these locations and location B2 was used as a reference. The high fluctuations in air temperatures at the highest levels indicate that modular systems distributed most of the heat at those levels.

For the forced air system no particular points of high temperature changes and rates were observed and most points varied within the values mentioned for location B3.

From the above discussion, based on a small number of cases, the variations in air temperature observed were significantly different from system to system. The foreced air and modular systems had higher fluctuations in time than the hydronic system. For two systems, regions with higher variations were found (modular: ceiling location, hydronic: front of the baseboard). For all the systems, fairly constant temperatures were maintained at the center of the room. In the next section, temperatures maintained throughout the season for all the systems in the comfort rooms are discussed.

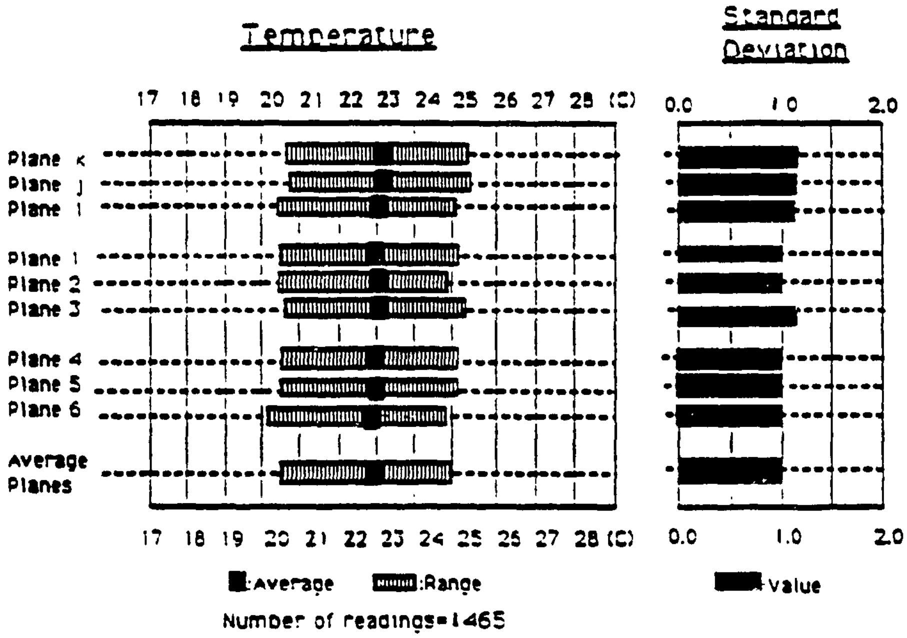
4.4.2.2 Seasonal Variations

In order to generalize some of the observations mentioned in the preceding sections, the average temperatures maintained in the rooms by the systems are investigated for the whole season of measurements. These data will be discussed for the different planes shown earlier in Figure 4.18 and in relation to the 21°C setting which was supposed to be maintained by all three systems.

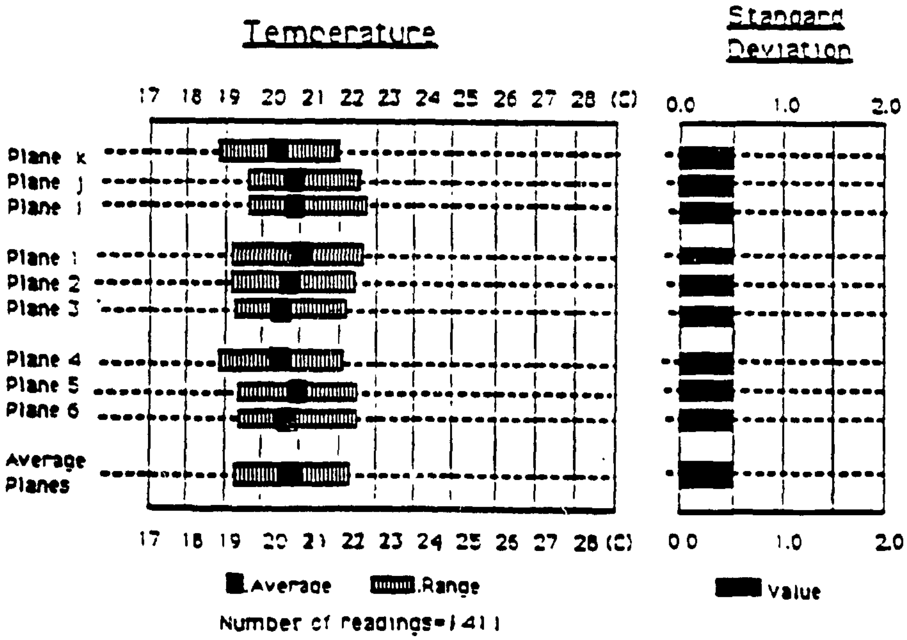
The results for the variation of temperature for the whole season, with normal operation conditions, are displayed in Figure 4.26 for the forced air and hydronic systems, and in Figure 4.27 for the two modular systems. The data represent the range, standard deviation and average air temperatures in planes sections.

From the average room air temperatures, the temperatures maintained by the hydronic and the modular system on the third floor were closer to 21°C than for the other two systems.

This difference in maintaining room temperatures can be attributed mainly to the combined effect of the different locations of the thermostat and the differences in losses between rooms on each floors. With the room isolated (door closed) and the thermostat outside not influenced by the same boundary conditions, the temperatures maintained in the rooms were different from the set point depending on their respective heat losses.

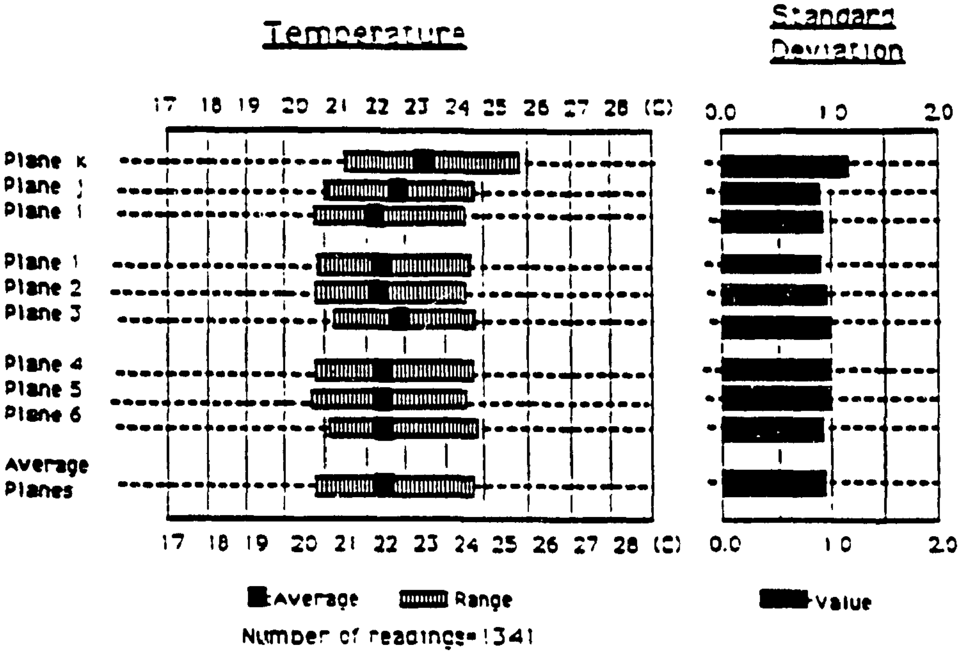


a) Forced air

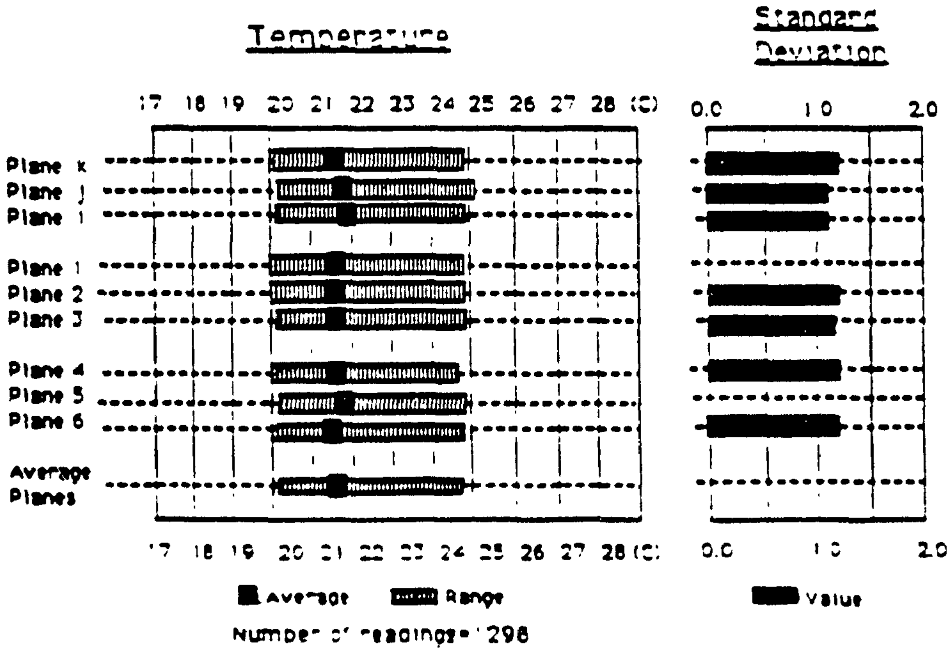


b) Hydronic

Figure 4.26: Planes' Seasonal Air Temperature Distribution, Forced Air and Hydronic Systems (see figure 4.18 for planes' definitions)



a) Modular 1st



b) Modular 3rd

Figure 4.27: Planes' Seasonal Air Temperature Distribution, Modular Systems, 1st and 3rd Floor (see figure 4.18 for planes' definitions)

Because of higher room losses on the third floor, the temperatures maintained in the rooms were not as high as those of the first floor. When the tests were performed, the thermostat on both floors were always showing a 21°C temperature and therefore this temperature (even though some errors can be attributed to the sensor of the thermostat) was maintained in the rest of the apartment which was also verified by temperature readings in the rest of the apartment. Since the room losses were high on the third floor, when the 21°C setting was achieved at the thermostat location, the room temperatures were slightly lower. On the first floor, the lower losses from the room compared to floor losses caused the temperature in the room to be slightly higher than for the rest of the apartment.

In terms of the planar distribution of temperatures, the range and standard deviation of the forced air and the modular systems are twice as high as those of the hydronic system. This indicates that on the whole the hydronic system was a relatively "quiet" system with small temperature variations in time throughout the season and reinforces the observations made in the preceding section to the effect that the temperatures vary less in the room with the hydronic system.

For the forced air system, temperatures in the room were maintained at an average near 22.5°C for the whole season with values ranging between 20.5°C and 25°C. From the seasonal distribution, it is difficult to discern differences either in the vertical or the horizontal planes distributions indicating that the air in the room is well mixed. The average value of the temperatures are centered between the maximum and minimum values which indicates that the distribution is not slanted either way.

The temperatures maintained by the hydronic system are the lowest maintained for all the systems with values near 20.5°C. We observe the standard deviation of the distribution is small, less than 0.5°C for more than 1000 readings and that the average is located symmetrical between the maximum and minimum values.

In terms of spatial distribution of the temperatures for the hydronic system, differences can be observed between the averages of the planes perpendicular to the wall (planes 4-5-6) and the planes parallel to the window wall. There is a decrease in temperature from plane 1 to plane 3, in other words, temperatures in planes parallel to the window wall are warmer close to the radiator and slowly decrease towards the interior of the room. The planes perpendicular to the window wall are generally warmer at the center than at the sides of the room. Although, differences are small, considering the large number of points used, a general trend is visible.

As previously noted, the average air temperatures maintained by the modular systems are lower on the third floor (near 22°C) than on the first floor (near 21°C) Nevertheless, the distributions shown in Figure 4.27 are comparable in terms of the magnitude of standard deviations calculated for the season (1.0°C for approximately 1300 observations).

In terms of the location of the average with respect to minimum and maximum values of temperatures, the average value for the modular system on the first floor is located nearer to the minimum value and is not symmetrical as is the case for the first floor The high values for the third floor were later found to be associated with the experiments performed early in the season, during the month of November when troubles of overheating were observed

These problems with control for the modular system were probably caused by the presence of the hydronic system functioning in the rest of the apartment In this sense, the data for the modular system on the third floor is biased for the range of planes temperatures The data for the first floor are more significative If these data are used, the average temperature varied by 4°C between minimum and maximum values

From above, it can be said that the trend of plane temperatures generally follows the observations made earlier on the general air distribution of the systems Also, in terms of the range of average temperatures maintained in the rooms, the forced air and the modular system on the first floor, allowed variations of up to 4°C compared to lower values of 3°C found for the hydronic system than for the other systems. Temperatures on the first floor were on the average maintained higher than on the third floor due to the differences in the building envelope characteristics between the floors.

For the modular system on the first floor, it is possible to observe a definite increase for the horizontal planes parallel to the floor reflecting the same results noted in section 4.3.1, and this time for the whole heating season. Therefore, the modular system can definitively characterized by major increases in temperatures in the vertical direction. For the planes parallel to the window wall, there is an increase for the plane temperatures at the back of the room, near the back wall. This was observed in section 4.3.1 as well. For the planes parallel to the side walls, there is not a definite increase in the temperatures observed as one goes from the interior wall to the neighbour wall.

From the above discussion, it can be seen that up to a great extent, the observations made for the cases discussed in the previous sections on air temperature distribution in planes and temperature fluctuations were also reflected in the general trends observed for each system for the entire season. In the next section, surfaces temperatures distributions for the particular cases used so far will be discussed.

4.4.3 Surface Temperatures

While mostly air temperatures were discussed in the preceding sections, surfaces temperatures were not treated. Surface temperatures, although an important aspect of the indoor environment, are treated here with less detail than for the air temperatures. Discussion is based on the cases mentioned in section 4.2 without extending the analysis to the whole heating season.

The average and range of the surface temperatures for the cases listed in the previous sections are summarized in Tables 4.13 to 4.16 for the forced air, the hydronic and the modular systems respectively. A graphical representation of these temperatures appears in Figure 4.28 for case 2.

Forced Air

For the forced air system (table 4.13), the variations in surfaces temperatures from case to case as well as the values of each surfaces temperatures are listed.

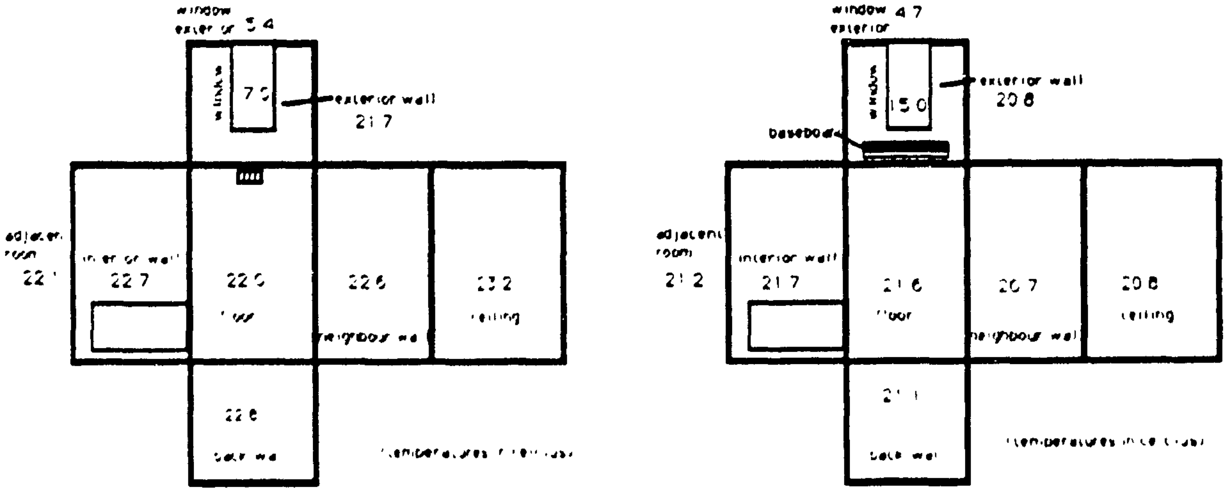
In terms of the variations from case to case, the surfaces temperatures remain approximately within the same relative order. The variations of the window as well as the exterior wall are the highest. They are directly exposed to the outdoor elements. Note that for all cases, all the interior surfaces temperatures vary by less than 1°C for a given period

In terms of the relative magnitude of the different surfaces temperatures for a specific case, the ceiling and interior walls have the highest values of all surfaces. The neighbour wall assumes values lower than the other walls. The difference between ceiling and floor temperatures is approximately 1°C for all cases. The difference between temperatures measured on the inside and the outside of the window range from 7°C to 10°C in different cases

The difference between the temperature measured on either sides of the interior wall show that the side of the wall on the exterior of the room was significantly lower than on the inside for the first case. Therefore, heat was transferred from the room to the rest of the apartment in that case.

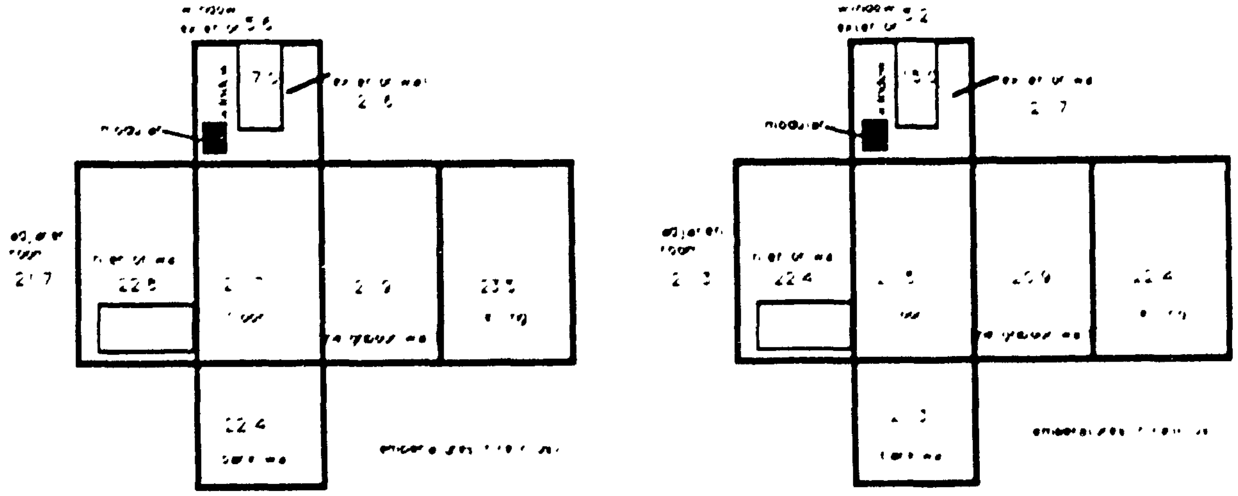
Table 4.13: Surfaces Temperatures Forced Air, Case 2

	Average and range (°C)							
Case	1		2		3		4	
INTERIOR WALL								
in	23.0	0.5	22 7	0.5	22 8	0.6	21.7	0.9
adjacent room	22.2	0.9	22 1	1 0	22 0	1.0	21.6	0.9
WINDOW WALL	21.9	0.6	21 7	0 7	21 6	0 7	20.6	1.4
WINDOW								
interior	18.0	0.8	17.0	1.3	17.6	1.5	18 0	2 5
exterior	8 5	1.6	5.4	3.4	7.9	5.6	11.3	10 1
NEIGH. WALL	23.1	0.2	22.6	0.5	22.6	0.8	21.7	0.8
BACK WALL	23 2	0 5	22.8	0.9	22.9	0 8	21.9	0.7
CEILING	23.5	0.6	23.2	0 7	23 2	0 8	22 0	0 9
FLOOR	22 3	0.3	22 0	0.5	22 1	0 6	21 1	0 8



a) Forced Air

b) Hydronic



c) Modular System 1st floor

d) Modular System 2nd floor

Figure 4.28 Averaged Surface Temperatures, All Systems (case 2)

Hydronic System

Data for surfaces temperatures for the hydronic system are listed in Table 4.14.

Table 4.14: Surfaces Temperatures Hydronic System, Case 2

	Average and ranges (°C)					
Case	1		2		3	
INTERIOR WALL						
in	21.6	0.5	21.7	0.5	21.3	0.5
adjacent room	21.4	0.9	21.2	0.8	21.0	0.5
WINDOW WALL	20.9	0.5	20.8	0.5	20.4	0.7
WINDOW						
interior	16.0	1.5	15.0	2.0	15.0	2.8
exterior	8.5	2.3	4.7	3.0	6.8	7.0
NEIGH. WALL	20.8	0.6	20.7	0.5	20.2	0.5
BACK WALL	21.1	0.6	21.1	0.4	20.8	0.5
CEILING	20.9	0.5	20.8	0.3	20.4	0.5
FLOOR	21.5	0.4	21.6	0.5	21.1	0.4

For the three cases listed for the hydronic system, the values of surfaces temperatures remained almost the same from case to case except for the surfaces exposed to the exterior. The surface temperatures of interior surfaces were all close to one another with 1 °C variations between them. Note that the ceiling and floor temperatures remained fairly close to each other, and that the neighbour wall had slightly lower values than the rest of the surfaces.

Modular Systems

Data for surfaces temperatures for the modular systems on both floors are listed in Tables 4.15 and 4.16.

For the modular systems on the first and third floor, again, most surfaces temperatures are similar in magnitude with interior surfaces generally warmer than the others. The noticeable difference is the larger difference between floor and ceiling temperatures for the modular system on the first floor with the ceiling being on the average 2°C higher than the floor. The difference for the modular system on the third floor is less (less than 1°C difference).

Table 4.15: Surfaces Temperatures Modular System 1st Floor, Case 2

	Average and Range (°C)					
Case	1		2		4	
INTERIOR WALL						
in	22.6	1.0	22.8	1.2	22.7	1.0
adjacent room	21.9	1 1	21.7	1.0	21.8	0.9
WINDOW WALL	21.7	1.0	21.6	1.4	21.7	1 0
WINDOW						
interior	17.0	1.5	17.0	1.5	17 0	1.6
exterior	8.5	3.0	5.6	3.2	7.3	3 9
NEIGH. WALL	22 0	0 5	21.9	0.5	22 0	0 5
BACK WALL	22 4	0.9	22 4	0 7	22 4	0 6
CEILING	23.3	1.5	23.5	1 6	23 4	1 6
FLOOR	21.7	0.5	21.7	0 7	21 9	0 6

Table 4.16: Surfaces Temperatures Modular System 3rd Floor, Case 2

	Average and Range (°C)							
Case	1		2		3		4	
INTERIOR WALL								
in	22.3	1.2	22.4	1.1	21.8	1.1	22.3	1.2
adjacent room	21.4	0.6	21.3	1.3	21.4	0.7	21.4	0.6
WINDOW WALL	21.7	1.1	21.7	1.0	20.9	1.1	21.7	0.3
WINDOW								
interior	15.5	2.0	15.0	3.0	16.0	2.6	15.0	2.5
exterior	7.5	4.0	5.2	5.9	9.8	6.7	6.6	5.4
NEIGH. WALL	21.0	0.6	20.9	0.7	20.4	0.6	21.0	0.5
BACK WALL	21.4	0.6	21.3	0.9	20.9	0.8	21.4	1.0
CEILING	22.2	2.2	22.4	2.0	21.4	2.0	22.3	2.2
FLOOR	21.7	0.5	21.5	0.5	21.2	0.5	21.7	0.7

For the modular system, the temperature of the location right above the discharge of the unit were found to be on the average 2°C to 5°C higher than the rest of the exterior wall temperatures due to the warm air leaving the unit. This is the only location where significant differences in surface temperature were observed and they are not shown on Figure 4.28 since the overall air temperature of the walls were listed in that table.

Summary

Considering the small differences between the distribution of surfaces temperatures for the cases studied, the typical surfaces temperatures for the different systems can be represented using case 2 as an average distribution (see Figure 4.28). As we can see, the magnitude of the temperatures are comparable for the same floor.

The first floor is characterized by generally higher values with lower floor temperatures, higher ceiling temperatures, high interior walls temperatures and low exterior wall and window temperatures independently of the systems.

The third floor is characterized by the influence of the systems with the modular system raising ceiling temperatures. The neighbour wall on the third floor is at a lower temperature, indicating that heat losses there are important.

From the above, we can characterized the relative influence of the systems on surface temperatures. The forced air system produces an environment with relatively high temperatures and with the high ceiling temperature. All other surface temperatures assume more or less the same temperature. The effect of the forced air system on the ceiling temperature can be said to be less than for the modular system on the same floor since the rest of the temperatures for the forced air system are in the same range as those of the modular system except for the ceiling.

The hydronic system also maintains surface temperatures with interior surfaces warmer and the neighbour wall temperatures relatively lower. The difference between floor and ceiling temperatures for the hydronic system is not as different as for the modular system on the same floor.

The modular systems, by comparing them with the other systems installed on the same floor and for similar conditions, have created ceiling temperatures significantly higher Most of the heat produced went towards the ceiling. The effect was less important for the third floor, where the ceiling (roof) losses are high.

The distribution of surfaces temperatures for the different systems can be related to the temperature distribution established in section 4 3 1 The effect of low temperatures at the neighbour wall are reflected in the distributions of room air temperatures for the hydronic and the modular system on the third floor in terms of disturbances along the side of that particular wall The higher temperatures of the interior wall have not been as significant as the effect of the warmer back wall for the cross sectional and longitudinal distributions respectively.

4.5 Systems' Transient Response to Low Initial Room Temperatures

Another tool serving the analysis of the temperature distribution created by the systems, as well as their intrinsic performance, is the response to low initial room temperatures. Temperature variations observed in cases when the temperatures in the room were allowed to drop significantly before the systems started to operate are presented.

The tests consisted in allowing temperatures to drop freely in the room, and on the whole floor of the building, with all the systems off. Then, after temperatures had reached a low value, the systems were started with a set point of 21°C selected for startup. Experiments of this sort were performed for the forced air, the hydronic and the modular system on the third floor.

The outdoor conditions at the time of the tests are shown in Figure 4.29. Note that the outdoor conditions for the forced air and the modular system were the same. These conditions correspond to very low outdoor temperatures compared to the rest of the season and can be considered as exceptional. The day selected for the hydronic system is not as cold as for the other two systems.

The temperature of the air leaving the systems are show in Figure 4.30 for all three systems. The air temperature supplied when the systems are turned on are similar in range to the temperatures described in section 4.1.

The forced air and the modular systems, were found to cycle at high rates after they were turned on. This is due mainly to the cold outdoor temperatures at the time of the test and the initial low temperatures in the building for these tests.

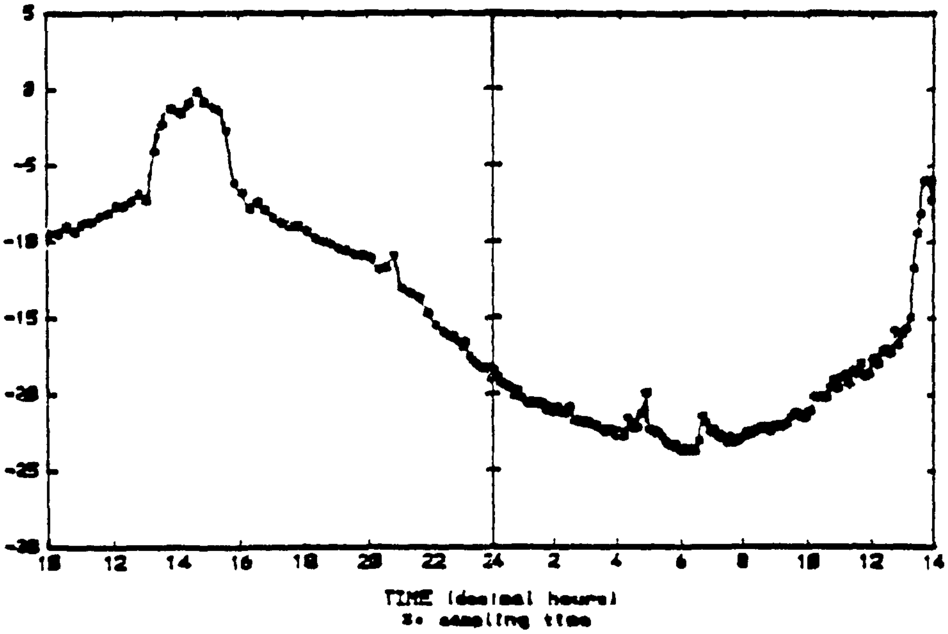
The hydronic system responded by progressively decreasing the delivered air temperatures. Note that for the hydronic system, part of the data for a similar test done on the previous day appear on the graph. Each cases are now discussed separately.

Forced Air System

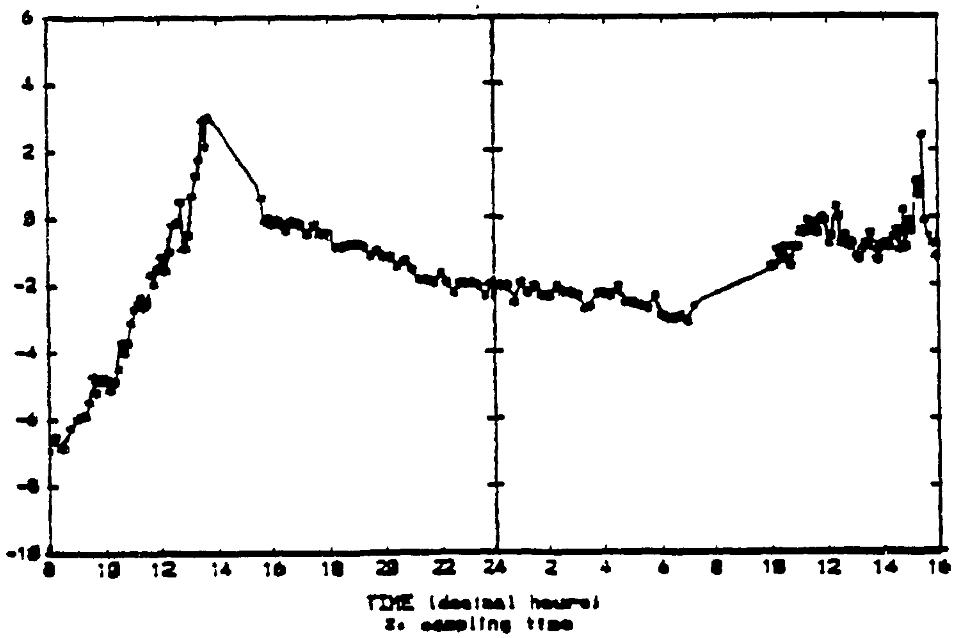
The air temperatures at the 0.8 m level before and after the forced air system was turned on appear in Figure 4.31. In Figure 4.31 a) temperatures at locations B1, B3 and A3 are shown for the longitudinal section of the room, and locations in the cross section of the room (locations B2, B3 and B4) are shown in Figure 4.31 b).

For the first part of the test, the decrease in temperatures can be observed. In the longitudinal section, locations closer to the window cool down faster than the locations deeper within the room. Variations of temperatures in cross section are similar at all points. The differences were small.

The air temperatures in the room for the forced air (Figure 4.31) were allowed to drop to 16°C before the systems was turned on. The observed effect in the room was a sudden raise temperatures at all locations. No significant differences were found between the rates of temperature increase either in the longitudinal or the cross-sectional directions.

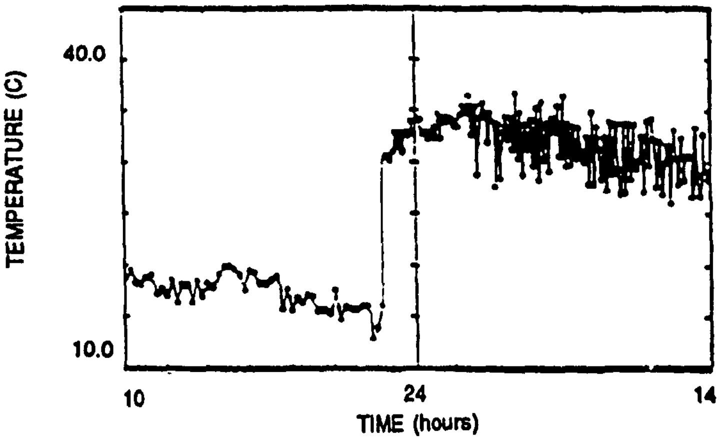


a) Outdoor Temperature, Forced Air and Modular Systems

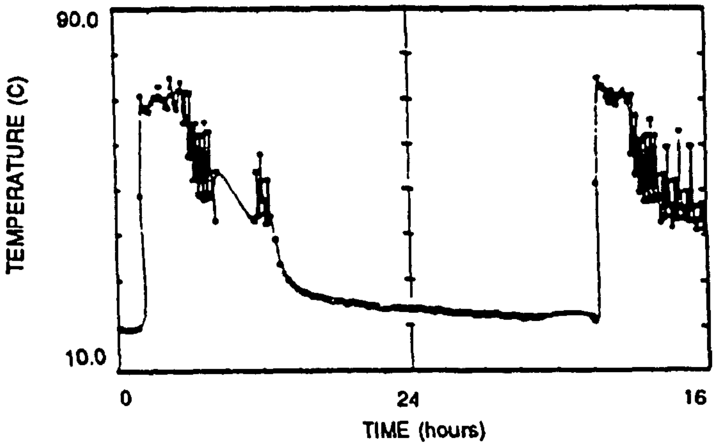


b) Outdoor Temperature, Hydronic System

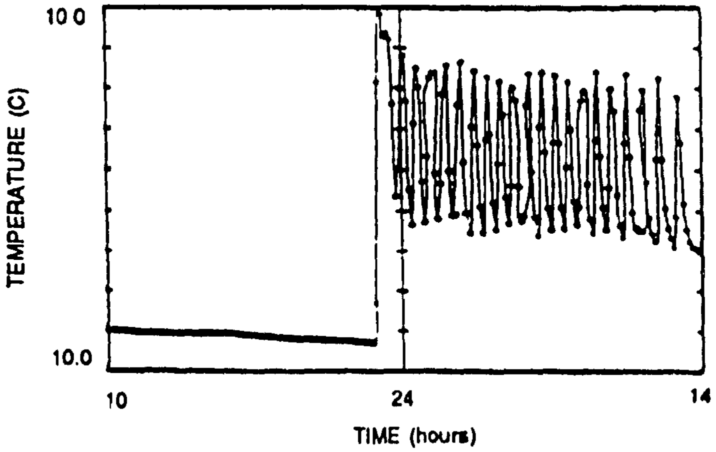
Figure 4.29: Outdoor Conditions During Transient Warming Tests, Forced Air, Hydronic and Modular Systems



a) Forced Air System

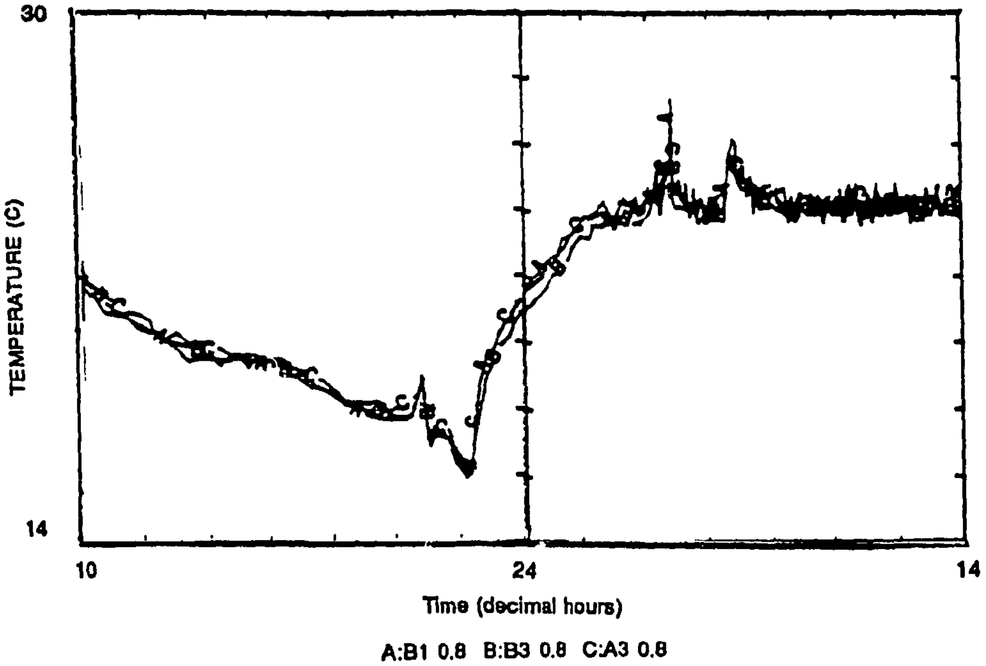


b) Hydronic System

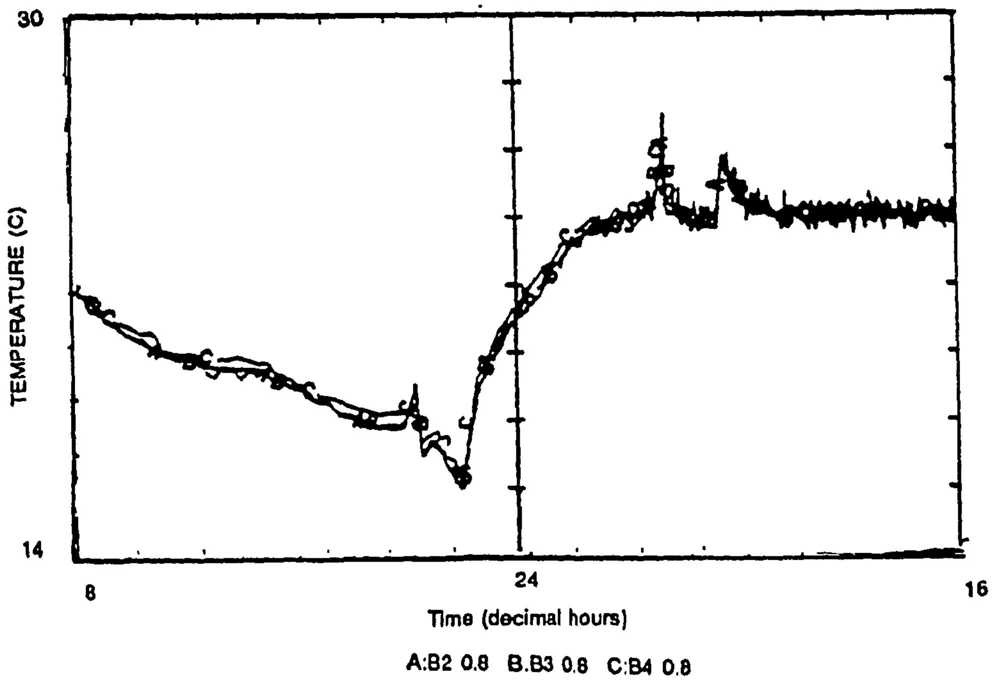


c) Modular System

Figure 4.30: Air Supplied by the Heaters During Transient Warming Tests



a) Plane 5, Longitudinal



b) Plane 2, Cross Section

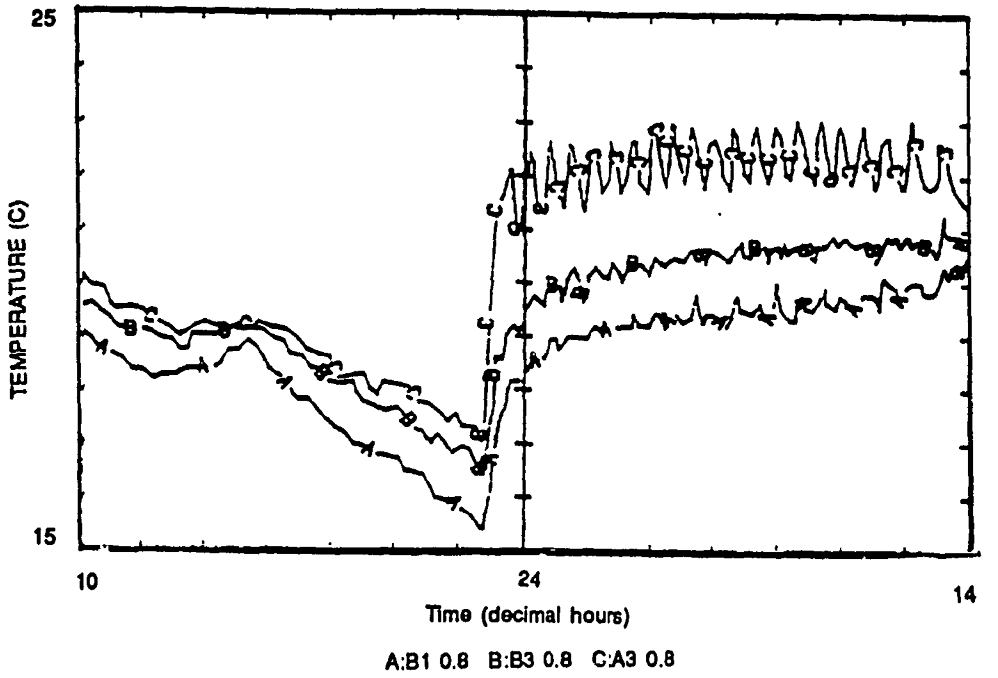
Figure 4.31: Room Air Temperature Variations in Time, Transient Warming Test, Forced air

The forced air system took approximately 90 minutes (22:19:00 to 23:48:00) to bring the air in the room from 16°C to 21°C (5°C difference) This corresponds to an average rate of +3.3°C/hour. Afterwards, temperatures kept rising until they reached 25°C. This indicates that the thermostat located outside the room still found the temperatures in the remaining of the apartment to be less than 21°C and requested more heat from the furnace at that time.

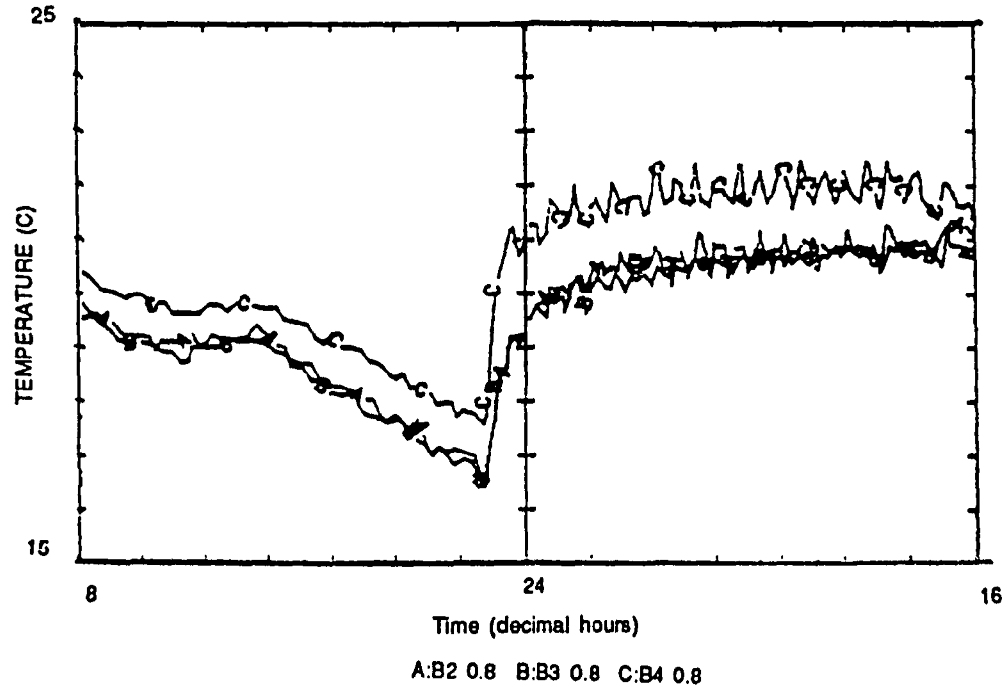
Modular System

The decrease in room temperatures at different locations for the case of the modular system show that not all points decreased in temperatures in the same fashion. As can be seen from Figure 4.32 a), the temperatures in the longitudinal section (before 22:00:00) points are warmer close to the interior of the room (location A3) and become progressively colder closer to the window. The difference is of approximately 1.0°C at the end of the cooling period. The variations in the cross section show that the location close to the neighbour wall was approximately 0.8°C higher than the other locations. Some heat was gained from the neighbour's wall at that location.

In terms of the time required to reach the desired setting, temperatures in the room took different times to reach 21°C for the 0.8 m height Temperatures at location B1, B2 and B3 increased at a rate of approximately 2°C/hour while location B4 (close to the neighbour wall) and location A3, back of the room increased at rates of 7°C/hour and 10°C/hour respectively Note that the rates for B1 location correspond to the temperatures maintained later by the system (near 20°C).



a) Plane 5, Longitudinal



b) Plane 2, Cross Section

Figure 4.32: Room Air Temperature Variations in Time, Transient Warming Test, Modular 3rd

Because the rates of increase for the locations B4 and A3 seem much higher than for the other locations, the levels other than 0.8 m were checked in the data and were found to also have high increases. The level of 0.1 m at location B4 was found to increase more in the range found for locations B1, B2 and B3 (2°C/h).

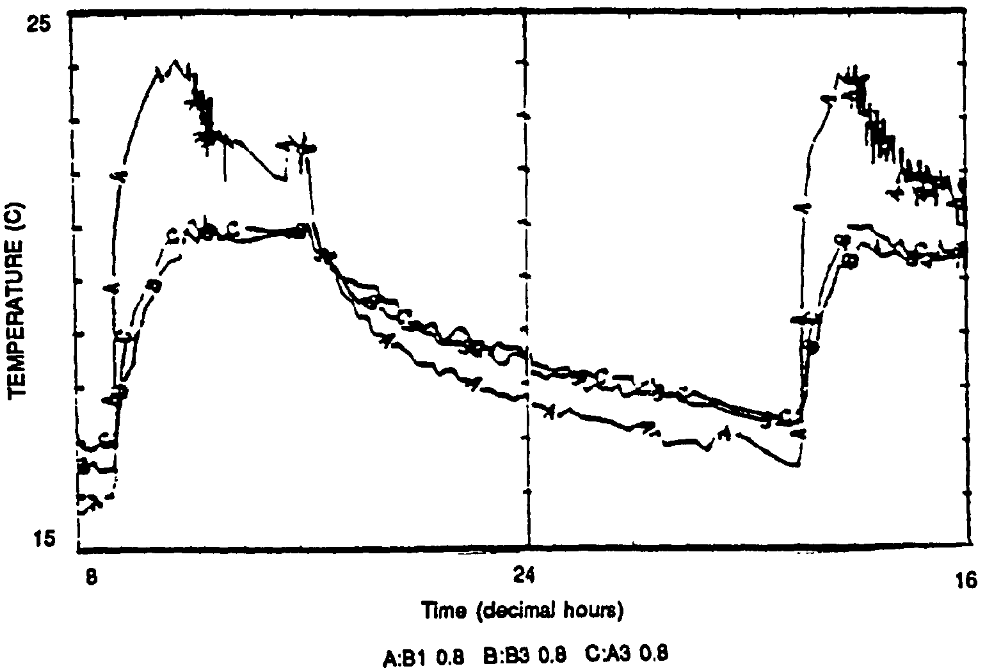
Hydronic System

For the hydronic system, the outdoor temperatures at the time of the test were warmer than for the other cases. The cool down periods shown for the hydronic system reveals the same characteristics as for the modular system. This is for the cool down of points either in the longitudinal or the cross section of the room at the 0.8 m level (see Figure 4.33 a and b).

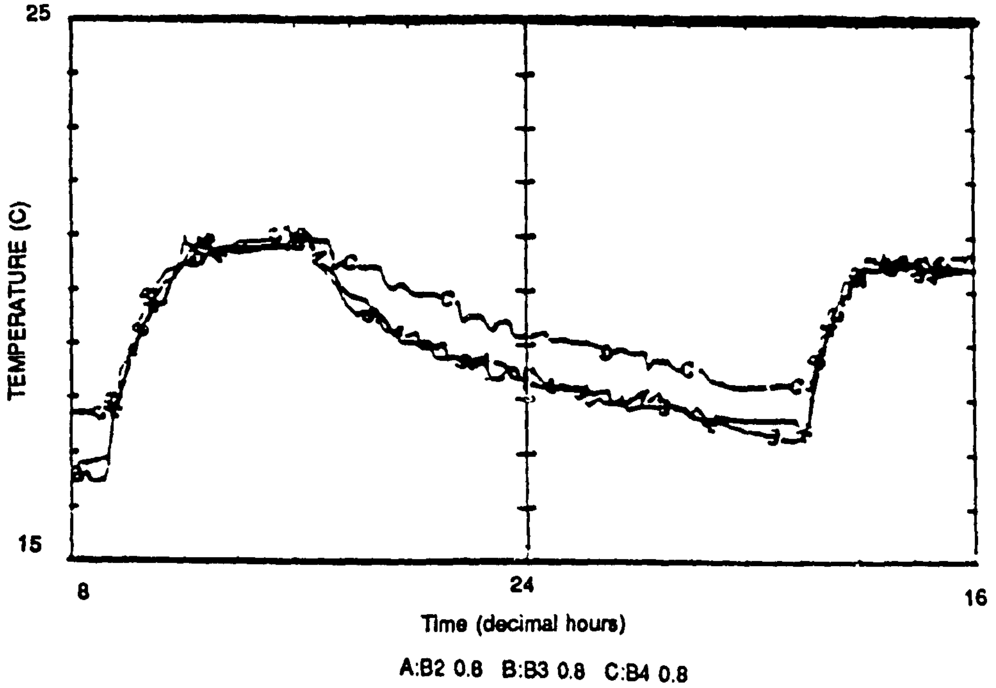
In the longitudinal section, the points cool down faster closer to the window than within the room. In the cross section, the effect of the heat gained from the neighbour wall still keeps locations in that region warmer (difference of approximately 1°C).

In terms of the warm up period, points away from the hydronic baseboard all increase in temperatures in the same fashion and at the same rate. For the location B1, level 0 8 m, the 21°C setting was reached in approximately 10 minutes for the two cases shown on the figure The corresponding average rate of increase at B1 was calculated to be near 20°C/h or higher

For the warm up at the other locations, the hydronic system raised temperature from 17°C to 21°C in approximately 142 minutes (1 7°C/h) for the two cases shown in Figure 4 33 The increase at the center of the room was slightly higher than for the back locations and similar in cross section



a) Plane 5, Longitudinal



b) Plane 2, Cross Section

Figure 4.33: Room Air Temperature Variations in Time, Transient Warming Test, Hydronic System

The transient performance of the systems for the conditions studied showed that the systems vary significantly in the way they distribute heat to attain the desired settings. It allowed to find the importance of the heat gained from the neighbour's wall on the third floor.

For the cool down stage, the tests showed the importance of the heat coming from the neighbour's wall into the room on the third floor. This indicates that the results in general for all the cases studied on that floor will be influenced by that condition. The warming of the room by the neighbours on the third floor is important for low temperature (modular case presented here) as well as warmer outdoor temperature conditions (hydronic case)

The modular system can increase the room temperature at a faster rates than the forced air system. The difference is that the modular system tended to increase temperatures nonuniformly. Although the effect of the enclosure is involved, the magnitude of the difference is significant. The forced air system created a more uniform environment with the air well mixed

Although the hydronic system case was different its warm up period is characterized by temperatures close to the baseboard increasing at a much faster rate than for other locations in the room

4.6 Summary

In the present chapter, different aspects of the system's operation were treated From the information obtained, a good picture of the intrinsic behaviour of each systems and their differences can now be obtained Points discussed have been summarized in Table 4 17

Table 4.17: Summary of Chapter's Observations for the Systems

CRITERIA	FORCED AIR	HYDRONIC	MODULAR
CYCLING			
supply temperature:			
-average (°C)	28	35	50
-range (°C)	10	20	45
-max. rate of change (°C/H)	60	192	250
-description	sharp breaks between on/off periods, seesaw shape same for all loads	slow rise and decrease for low loads and not well defined max. & min. for higher loads	constant minimum close to 22°C, on cycle rise to 80°C followed by exp. decrease
surface temperature:			
-average (°C)	no surf. involved	25	25
cycles per hour:			
-average's range	0.7 to 2.2	0.6 to 3.0	1 or less
-effect of low outdoor T	increase	increase	increase slightly
TEMPERATURE DISTRIBUTION			
horizontal:			
-location to location	well mixed	warm zone close to baseboard and uniform elsewhere	uniform
-increases with height (°C)	~1.5°C	~3°C close to baseb.	up to 3°C ~1°C elsewhere
vertical:			
-longitudinal	warmer from window to back and from floor to ceiling	warm close to heater, same elsewhere	warmer from window to back and floor to ceiling
-cross section	symmetric	symmetric	symmetric
-room	one zone	two zone: baseb. and rest of room	two zone: lower part and higher part of room
TEMPERATURE VARIATIONS (°C)			
center (0.1m and 1.1m):			
-max. between sampling	0.8	0.8	0.8
-avg. between sampling	0.5	0.5	0.3
-for a period	1.0	0.9	1.0
-for the season	4.5	3.5	4.0
rates, center:			
-avg. center (0.1m and 1.1m)	3.0	2.0	1.9
-difference 0.1 & 1.1m	none or 1.1 higher	none or 1.1 higher	none or 1.1 higher
-points with high variations other than center	none	front of baseb.	close to ceiling
SURFACE TEMPERATURES			
-ratio ceiling/floor	1.1	0.97	1.0 3rd floor 1.1 1st floor
TRANSIENT RESPONSE			
warming stage:			
-room distribution	uniform	faster close to baseboard	faster at back of room
-rate (°C/h)	4	2 to 20	2 to 10

CHAPTER 5

5. DETERMINATION OF TEMPERATURE PROFILES

As observed in the previous chapter, the heating systems created higher temperatures close to the ceiling. Although the relative importance of this phenomena was discussed, the actual increases of temperature with height were not precisely quantified. Therefore, in this chapter, the results for temperature distributions obtained in the previous chapter are further studied Points discussed include: finding the actual shape and magnitude of the vertical temperature profiles, and finding the changes in these gradients in time for short periods and for the whole heating season.

It is important to determine the temperature profiles produced by a heater for both energy conservation and comfort. High gradients means that heaters sends heat to the ceiling before the occupants can feel comfortable at their level The heater will finally heat both the top and the bottom of the room, and in the process more energy will be used. Gradients must be limited to reduce discomfort risks of the occupants (see chapter 2) and it is important to determine within what range they occur.

As a starting point, note that the temperature distribution in the environment is mainly dependant on the following parameters combined:

-Heating system behaviour and response
-Room response (room air transfer function)
-Controller or thermostat response (set point)
-Outdoor temperature
-Other factors

In a procedure, to correctly describe the temperature profiles produced by the systems, the above factors must be included. The problem can then be formulated by the following questions:

1) What is the shape of the temperature profile produced by a given system at different locations in the test room?

2) Can the gradients be related to the behaviour of the heating system in terms of the magnitude and range of the heater's temperature?

3) What is the effect of modifying the enclosure's configuration by opening the door on the temperature profiles in the room? This question incorporates the control aspect and the thermostat control response. With the room door opened, the thermostat is also influenced by the conditions in the test room near it.

4) For a given system, is the magnitude of the gradients observed the same, irrespective of the value of the outdoor temperature at the time of the test or are they related?

These questions cover only part of the problem of temperature gradients in the rooms.

Even though there are other considerations, the analysis is limited to these questions in the first place.

In the first section, the mathematical equations and the calculation procedure used later to determine the typical temperature profiles created by the different systems are described. Mathematical equations are based on first and third order least squares interpolation of temperatures with height. The procedure shows how the data recorded were used to find the values of maximum, minimum and averages gradients.

The second section gives the results obtained using the mathematical formulation to determine the **shape** of the temperature profiles created by the systems. The case of the coldest and of milder outdoor conditions observed during the season are used in order to describe differences between temperature profiles obtained at cold and warmer outdoor conditions. For the case of the coldest day, it will be determined if a simple relation between heater temperature and magnitude of the gradients observed can be established. In the last part, the effect of opening the door on the temperature profiles in the rooms is treated briefly.

In the next section, results for the observed seasonal variations of the gradients are discussed in order to obtain a more general idea concerning the temperature profiles for the entire heating season (as described in section 4.2). For each system, the section demonstrates, the alterations observed in temperature gradients as outdoor temperatures vary.

The analysis is structured so that the general problem of determining the temperature profiles starts from the particular (questions 1 to 3) to the general (question 4). The sections are structured such that these questions can be independently considered. In order to arrive at a more synthetic representation of the behaviour of each system, these questions will be treated in the end.

5.1 Mathematical Representation and General Procedure

Temperature gradients are obtained by calculating the difference between the temperature at a given level and the level below. If this difference is negative, it indicates a decrease in temperature with height while a positive difference indicates an increase. Since the gradients are expected to vary with height, a gradient between two points is not sufficient to describe completely the gradients at a certain location in the room. Also, since the temperature at every points on a vertical column cannot practically be measured. It is needed to use a function to interpolate between points.

In this study, the following linear and third order interpolations were used:

a) Linear Interpolation:

$$T(y)=my+b \quad (^\circ C) \tag{5.1}$$

where $T(y)$: temperature as a function of height y (°C).
y : height (m)
m : slope (°C/m)
b : intercept (°C)

b) Third Order Interpolation:

$$T(y)=a_0+a_1{*}y+a_2y^2+a_3{*}y^3 \tag{5.2}$$

where $T(y)$: temperature as a function of height y (°C).
y : height in (m)
a_n : fitting coefficients ($^\circ C/m^n$)

The coefficients in equations 5.1 and 5.2 can be found using least squares approximation using collected data at different heights in the room and at different locations. These calculations were performed using available computer routines incorporated into home made programs [26].

Using equation 5.1, linear equation is obtained which will be accurate if the temperatures in fact increase linearly with height. On the other hand, if the gradients are not linear, equation 5.2 will provide a better estimation of the actual gradients.

The choice of the third order estimation and not a higher order was based on anterior analysis of the data which showed that using higher order approximations does not significantly decrease errors found in the calculations and, third order estimation errors were on the average less than 0.5°C.

The experimental set-up is such that at some grid locations, only two or three points are available for temperature measurement. In these cases, the first order approximation is used, even though, the gradients may in fact not be linear. This is because the results obtained for a low number of points using least squares method with third order interpolation are not as meaningful.

To correctly represent the gradients for the case of transient conditions, the analysis must include considerations of the time dependencies (in addition to the space variations of temperatures). In fact, the temperature profile at a given location is changing constantly as the heating system cycles on and off.

To take into account the time varying aspect of the gradients over a test period, the statistical quantities of maximum, minimum and average values in units of °C/m can be used.

The problem is how to determine these statistical quantities. If equation 5.2 is used then, the maximum gradient at a certain time must be evaluated by solving the derivative of the equation and finding its zeroes. There may be more than one. Therefore, several maximums expressed as a function of height would be obtained.

On the other hand, equation 5.1 can provide a single value for the statistical quantities sought. These quantities will not represent the exact values of the temperatures, but an approximation describing the trend of the observations. Nevertheless, considering possible inaccuracies in the measurements, this method has the advantage of not being as sensitive to errors in the measurements.

Considering the above remarks, it was decided to follow the following procedure to evaluate the maximum, minimum and average temperature gradients (or temperature profiles) for a given test period:

1) For the test period the coefficients of equation 5.1, m and b, are evaluated for each grid point in the room at each sampling time during the test period (e.g.: record temperatures 50 times during a day and find m and b for each time).

2) The slopes (m) obtained from 1) are examined to find the value of the maximum and minimum slopes as well as their time of occurrence during the data collection period.

3) For the locations where there were more than one measurement point (B type locations) the coefficients in equation 5.2 (a_0,a_1,a_2,a_3) were evaluated for the time of occurrence of the maximum and minimum found in 2.

4) To obtain the average temperature profile, the data for the whole test period are used to find the coefficients of equation 5.1 only for type A locations or 5.1 and 5.2 for type B locations (e.g: for 50 measurements during the period at B3, the average temperature profile is based on the least squares approximation of 50 temperature readings at each of the heights measured at location B3).

The outcome of the above calculations is finally a first order approximation of the maximum and minimum slope of the temperature profile with height during a test period are obtained. For grid locations with four or more measurement points, a third order estimation of the temperatures for the maximum and minimum cases is calculated. Finally, the average distribution of the variation of temperature with height based on the observations made over a whole test period is obtained (which is linear or of order three). A test period is usually close to a day in duration and always more than five hours.

5.2 Temperature Profiles and Relation With Heater Temperature

In this section, data are analyzed for specific cases in order to establish the actual temperature profiles when the systems operate in normal conditions of operation. Normal operation is as defined in section 4.1 (door closed, window shaded on the exterior, thermostat fixed at a constant setting of 21°C, no occupants, no furniture).

The first section reports results of temperature profile measurements over specific periods for different ranges of outdoor temperatures (relatively cold and mild), shows the relation with heater supply temperature and, finally, the variations in profiles observed when the door is opened.

The results presented in this section are limited to a small number of cases to better describe the details related to temperature profiles evaluation. In the next section of the chapter (section 5.3), the results on seasonal variations will provide a more general overview.

5.2.1 Temperature Profile at Coldest Observed Outdoor Temperature

The temperature profiles created by a given system must be determined at first for a specific test period in order to first acquire a notion of their general shape and the range in which they can be found. It was decided to select the period for the coldest day during the heating season on which data were collected to obtain an analysis for the worst possible case. This was accomplished for each system while making sure that a sufficient data were available for the day selected.

After the description of the outdoor conditions selected for analysis and the conditions inside the rooms at the time of the tests, the results for the first order interpolation and the third order interpolation of temperatures as a function of height are described.

General Description of Outdoor Conditions

Figures 5.1 to 5.4 show data which characterize the days selected for the forced air, the hydronic and the modular systems on the first and third floor. Each figure show graphs of temperature data for the outdoor environment, the temperature of the air supplied by the heaters (as defined in section 4.2.2) and the floor and ceiling temperatures. The corresponding numerical values are summarized in Table 5.1.

The days selected for the hydronic and forced air system are similar with a day characterized by outdoor temperature raising from -12°C to -4°C until noon and remaining around -5°C until midnight.

The days selected for each modular systems are similar to each other but warmer on the average than for the other two systems. The day is characterized by outdoor temperature close to -7°C with a progressive raise and fall between 8:00 and 17:00, bringing temperatures up to -4°C for that period.

The average temperatures maintained on the third floor by the hydronic and the modular systems (20.1°C and 21°C respectively) were slightly lower than the temperatures maintained on the first floor by the forced air and the modular systems (23.2°C and 22°C respectively).

The air temperatures maintained at the center of the room (table 5.1), remained fairly constant during the day for all systems and varied over less than 1.3°C. The thermostat setting of 21°C and the average temperature maintained were the same for the modular on the third floor, less for the hydronic system and higher for the forced air and the modular system on the third floor.

The walls' surfaces temperatures statistics (table 5.1) show that for all systems, the exterior wall and the neighbour's wall were on the average the coldest. The interior and back walls were generally slightly warmer than the rest of the walls. For surfaces temperatures, an important note must be made concerning the data used for the modular system on the third floor. It was already mentioned in chapter 3 that with the presence of both the modular and hydronic system at the same time in the room, the baseboard of the hydronic system had to be insulated while the tests were conducted.

Nevertheless, because the return pipe of the baseboard was not passing under the floor but over the fins, a certain quantity of heat is expected to come from that location into the room even with insulation over the pipe. The temperature at the surface of the insulation was found to be between 22.6°C and 23.9°C for the whole period of the test which is higher than normal but not significantly.

Table 5.1: Surfaces and Air Temperatures Statistics

SURFACE	Forced 86 read.	Hydronic 86 read.	Mod. 1st 97 read.	Mod. 3rd 99 read.
HEATER				
Average	28.4	44.1	50.2	47.4
Maximum	33.5	53.4	83.4	85.6
Range	9.7	9.3	15.6	12.7
OUTDOOR				
Average	-6.1	-7.1	-6.7	-7.5
Maximum	-2.0	-3.0	-3.7	-4.1
Minimum	-11.1	-12.1	-7.7	-9.0
Range	9.2	9.2	4.0	4.9
WINDOW				
Average	16.1	15.2	15.8	15.4
Range	2.4	3.8	1.5	3.0
EXTERIOR WALL				
Average	21.6	19.5	20.4	20.5
Range	0.9	0.8	0.7	0.8
NEIGHBOUR WALL				
Average	22.4	19.1	22.0	20.7
Range	0.6	0.5	0.5	0.6
INTERIOR WALL				
Average	22.9	20.6	22.8	21.7
Range	0.6	0.8	1.2	0.7
BACK WALL				
Average	22.6	19.8	22.8	20.9
Range	0.3	0.9	1.3	0.9
FLOOR				
Average	22.1	20.3	21.7	21.5
Range	0.4	0.5	0.7	0.5
CEILING				
Average	23.1	20.0	23.6	22.3
Range	0.7	1.0	1.8	2.1
ROOM AIR CENTER 0.8 m				
Average	23.2	20.1	22.0	21.0
Maximum	24.0	20.4	22.6	21.3
Minimum	22.7	19.6	21.7	20.5
Range	1.3	0.8	0.9	0.7

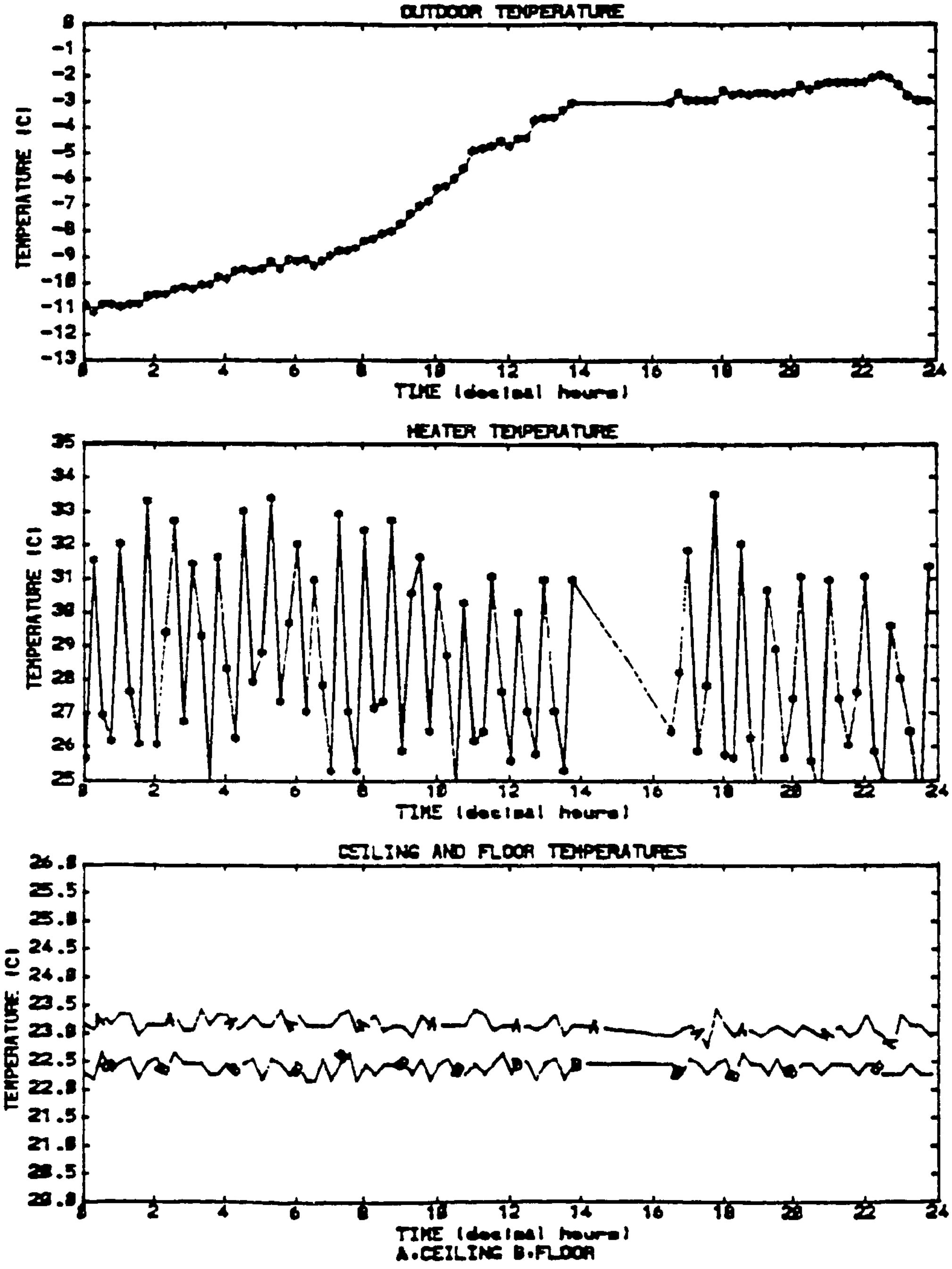


Figure 5.1: Outdoor, Heater and Surfaces Temperatures, Forced Air (day used below to determine temperature profiles at lowest observed T_{out})

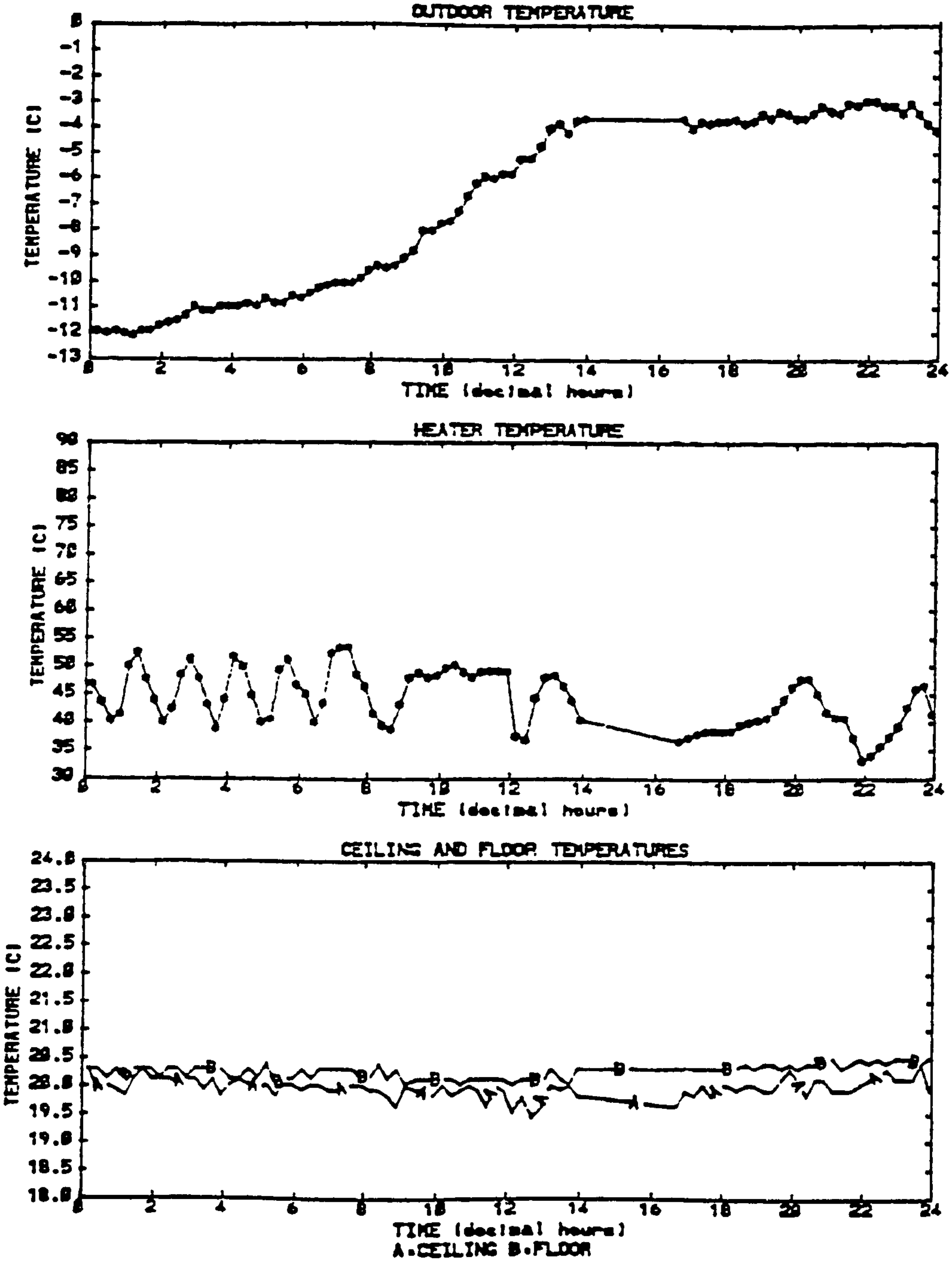


Figure 5.2: Outdoor, Heater and Surfaces Temperatures, Hydronic (day used below to determine temperature profiles at lowest observed T_{out})

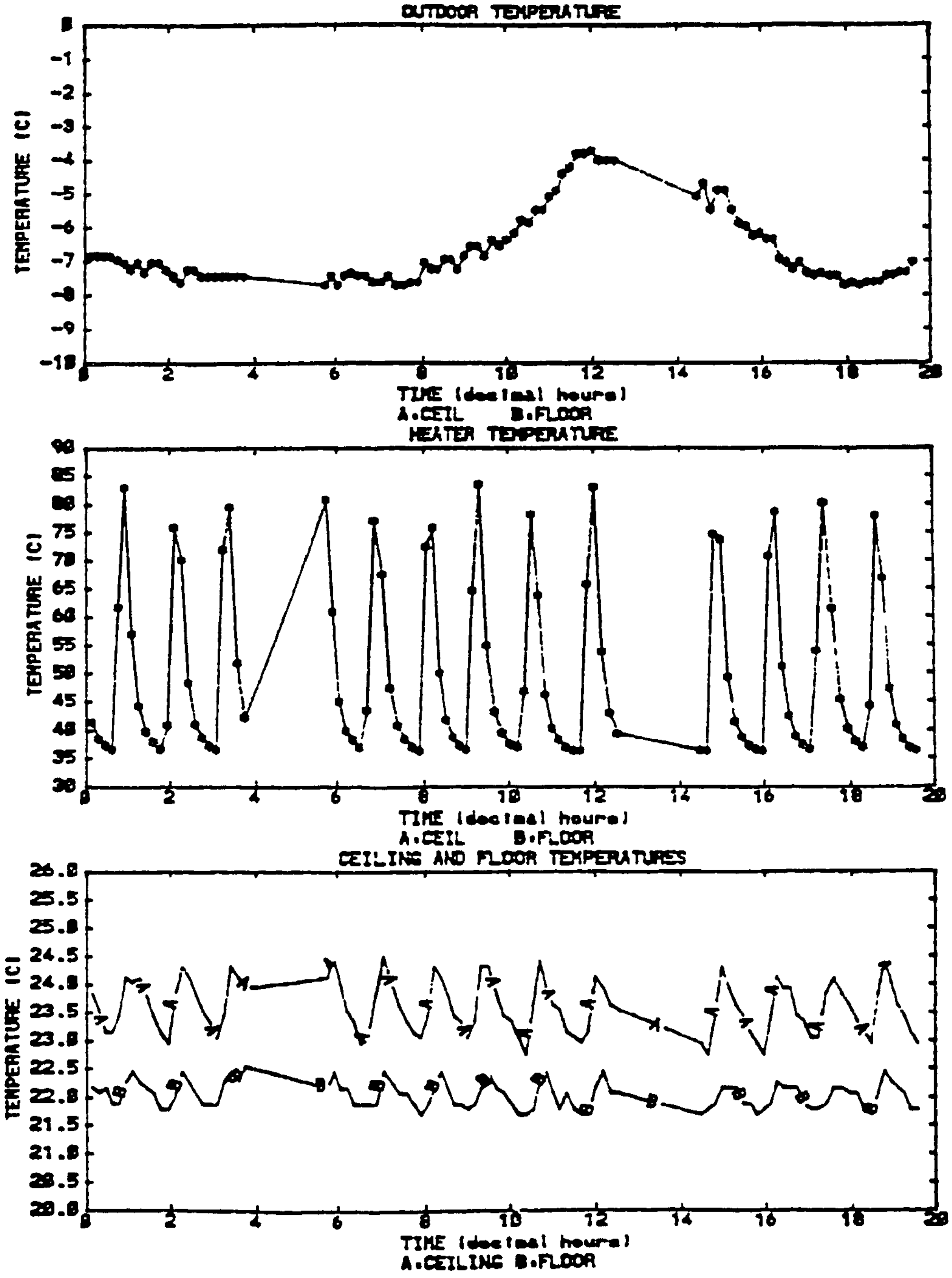


Figure 5.3: Outdoor, Heater and Surfaces Temperatures, Modular 1st Floor (day used below to determine temperature profiles at lowest observed T_{out})

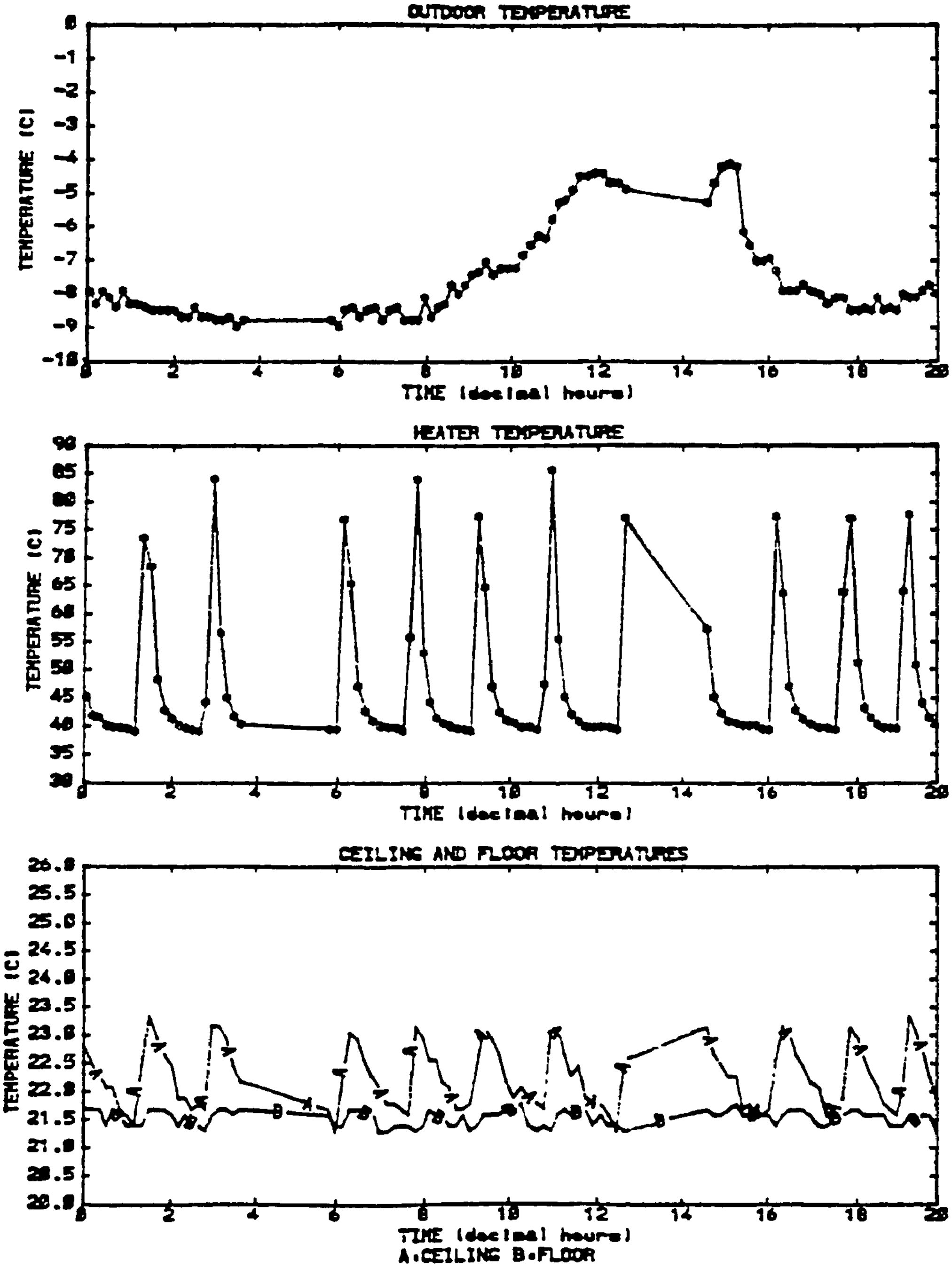


Figure 5.4: Outdoor, Heater and Surfaces Temperatures, Modular 3rd Floor (day used below to determine temperature profiles at lowest observed T_{out})

Note that the data collection period for the hydronic and forced air system is equally spaced at 15 minutes intervals while it is 10 minutes for the modular systems. Also, the data collection was interrupted at some points which is visible from the missing data points on the curves. The number of readings for each period is indicated in Table 5.1.

In terms of cycling of the systems, the two modular systems cycle on and off approximately every hour with average supply temperatures around 50°C and maximums of 86°C. The hydronic system completes on-off cycles approximately every two hours before 8:00 at temperatures between 40°C and 54°C and then cycles less often as the outdoor temperature rises. The forced air system cycles on and off approximately every hour at temperatures between 25°C and 34°C and the cycling pattern, unlike the hydronic system, does not change as the outdoor temperature rises.

In terms of floor and ceiling temperatures, the ceiling's temperature is higher than the floor for all systems except for the hydronic system for which these temperatures remain close to one another. The difference between floor and ceiling temperatures is following closely the cycling of the modular systems but for the forced air and hydronic, the ceiling and floor temperatures remain fairly constant throughout the day.

As can be seen, for the cases selected, the outdoor conditions were different for the systems (harsher for the forced air and hydronic systems) and general room temperatures conditions were within the range of those in chapter 4.

Results from First Order Approximation

The linear profiles obtained for maximum, minimum and average gradients obtained from equation 5.1 were plotted at each grid locations and appear in Figures 5.5 to 5.8 for the forced air, the hydronic and the modular systems on the 1st and 3rd floor respectively. The levels at which the temperatures were measured are listed in Table 5.2 and the results for the numerical values are listed in Table 5.3.

Table 5.2: Temperature Measurements Ranges

Grid Locations	1st Floor	3rd floor
	distances from floor (m)	
A1	0.8 - 1.6	0.8 - 2.4
A5	0.6 - 2.4	0.8 - 2.4
B2,B3,B4	0.1 - 2.4	0.1 - 2.4
A2	1.6 - 2.4	0.8 - 1.6
A3	---------	0.8 - 2.4
A4	0.8 - 2.4	0.8 - 1.6

For a better representation, the graphs are arranged such that each grid point is located in the same order as they are physically located in the rooms and note that:

-The curves show the increase in temperature with height from the lowest point measured and the value of the intercept (b) of the equation is not included in the graphs. The profiles therefore represent the relative variation of temperature with height.

-Locations where an insufficient number of thermocouples were available for calculation are not shown.

-The bottom scale for the temperature was usually kept between ± 2.5°C except where the data were outside of range.

-Although the curves shown extend up to 2.6m in height, the measurements were made up to 2.4m high and the ceiling height is of 2.8 meters. Values higher than the 2.4m height are extrapolations.

- The maximum error is the maximum error obtained for a particular temperature at a specific thermocouple location while the average error is the sum of the errors for all the thermocouples at a grid location divided by the number of thermocouples at that grid location.

The results indicate that the temperature gradients for the four heating systems are different based on first order approximation. The hydronic and forced air systems produce gradients which are less than one degree for all locations compared to the modular systems which can have gradients up to 3.8 degrees, but have maximums of 2 degrees on the average. The results summarized in Table 5.4 definitely show that the modular systems rank the highest for all categories.

The values for the forced air system as listed in Table 5.3 indicate that all locations have more or less the same gradients at all locations. The locations at the back of the room have minimums which are slightly negative. On the whole, the gradients are small compared to the other systems. Also, the variation in time, represented by the standard deviation is also small (maximum standard deviation of 0.2°C/m).

Table 5.3: Gradients' Numerical Values Calculated from Equation 4.1

Grid Locations	Forced	Hydronic	Mod. 1st	Mod. 3rd.
A1				
Maximum (°C/m)		0.6		-1.2
Minimum (°C/m)		-0.2		-1.7
Average (°C/m)		0.2		-1.5
St. Dev.(°C/m)		0.2		0.1
Range (°C/m)		0.8		0.5
B1				
Maximum (°C/m)	0.6	-0.3	1.8	1.5
Minimum (°C/m)	0.1	-0.8	0.6	0.1
Average (°C/m)	0.4	-0.5	1.1	0.6
St. Dev.(°C/m)	0.1	0.1	0.3	0.4
Range (°C/m)	0.5	0.5	2.4	1.4
A5				
Maximum (°C/m)	0.0	0.4	2.1	2.5
Minimum (°C/m)	-0.5	0.0	0.3	0.7
Average (°C/m)	-0.2	0.2	1.0	1.3
St. Dev.(°C/m)	0.1	0.1	0.5	0.5
Range (°C/m)	0.5	0.4	2.4	1.8
B2				
Maximum (°C/m)	0.9	0.4	2.0	1.9
Minimum (°C/m)	0.3	-0.1	0.6	0.3
Average (°C/m)	0.6	0.1	1.1	0.8
St. Dev.(°C/m)	0.1	0.1	0.4	0.4
Range (°C/m)	0.6	0.5	2.6	1.6
B3				
Maximum (°C/m)	0.9	0.5	2.1	2.1
Minimum (°C/m)	0.3	0.1	0.6	0.4
Average (°C/m)	0.6	0.3	1.1	1.1
St. Dev.(°C/m)	0.1	0.1	0.4	0.4
Range (°C/m)	0.6	0.4	2.7	1.7
B4				
Maximum (°C/m)	0.7	0.5	2.2	2.0
Minimum (°C/m)	0.1	0.1	0.8	0.5
Average (°C/m)	0.4	0.3	1.3	1.0
St. Dev.(°C/m)	0.1	0.1	0.4	0.4
Range (°C/m)	0.6	0.4	3.0	1.5

Table 5.3 (continued): Gradients' Numerical Values Calculated from Equation 4.1

Grid Locations	Forced	Hydronic	Mod. 1st	Mod. 3rd.
A2				
Maximum (°C/m)	0.9	0.6	3.8	-0.4
Minimum (°C/m)	-0.1	-0.1	1.0	-3.2
Average (°C/m)	0.4	0.3	2.0	-1.4
St. Dev.(°C/m)	0.2	0.2	0.8	0.7
Range (°C/m)	1.0	0.7	2.8	2.8
A3				
Maximum (°C/m)		0.5		1.40
Minimum (°C/m)		-0.2		0.18
Average (°C/m)		0.2		0.61
St. Dev.(°C/m)		0.2		0.30
Range (°C/m)		0.7		1.22
A4				
Maximum (°C/m)	0.4	0.9	2.0	2.20
Minimum (°C/m)	-0.4	-0.2	0.5	0.24
Average (°C/m)	-0.1	0.3	1.1	0.94
St. Dev.(°C/m)	0.2	0.2	0.4	0.43
Range (°C/m)	0.8	1.1	1.5	1.96

Table 5.4: Maximum Values of Interpolations

Maximum Values	Forced	Hydronic	Mod. 1st	Mod. 3rd	All Systems
Maximum (°C/m)	0.9	0.9	3.8	2.5	3.8
Minimum (°C/m)	-0.5	-0.8	0.3	-3.2	-3.2
Average (°C/m)	0.6	0.3	2.0	-1.5	2.0
St. Dev.(°C/m)	0.2	0.2	0.5	0.7	0.7
Range (°C/m)	1.0	1.1	2.8	2.8	2.8
Max. Err.(°C)	0.6	1.5	0.8	1.6	1.6
Avg. Err.(°C)	0.4	1.4	0.5	1.1	1.4

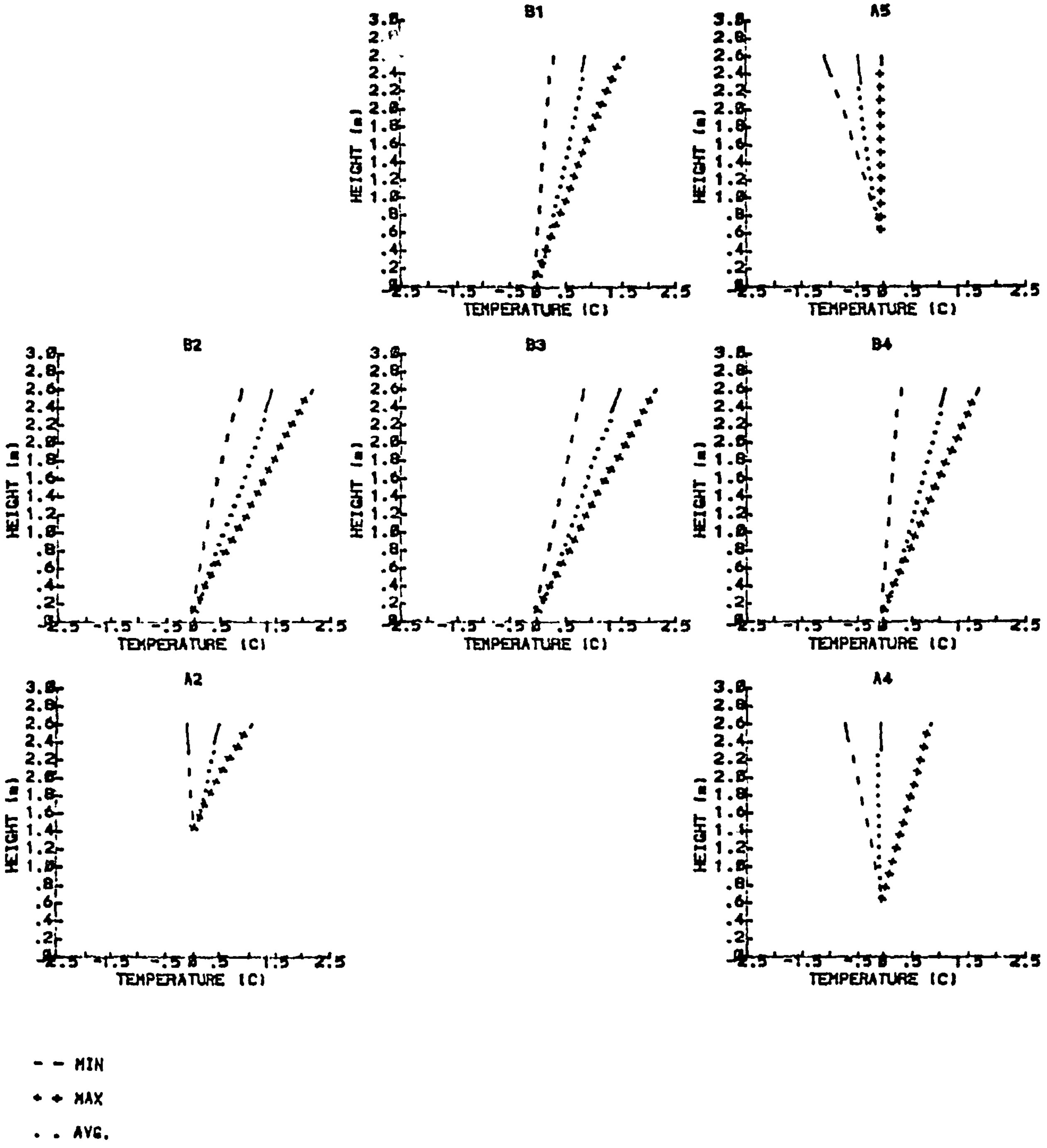


Figure 5.5: First Order Approximation of Temperature Profiles, Forced Air System (T_{out} average=-6.1°C, see fig. 5.1)

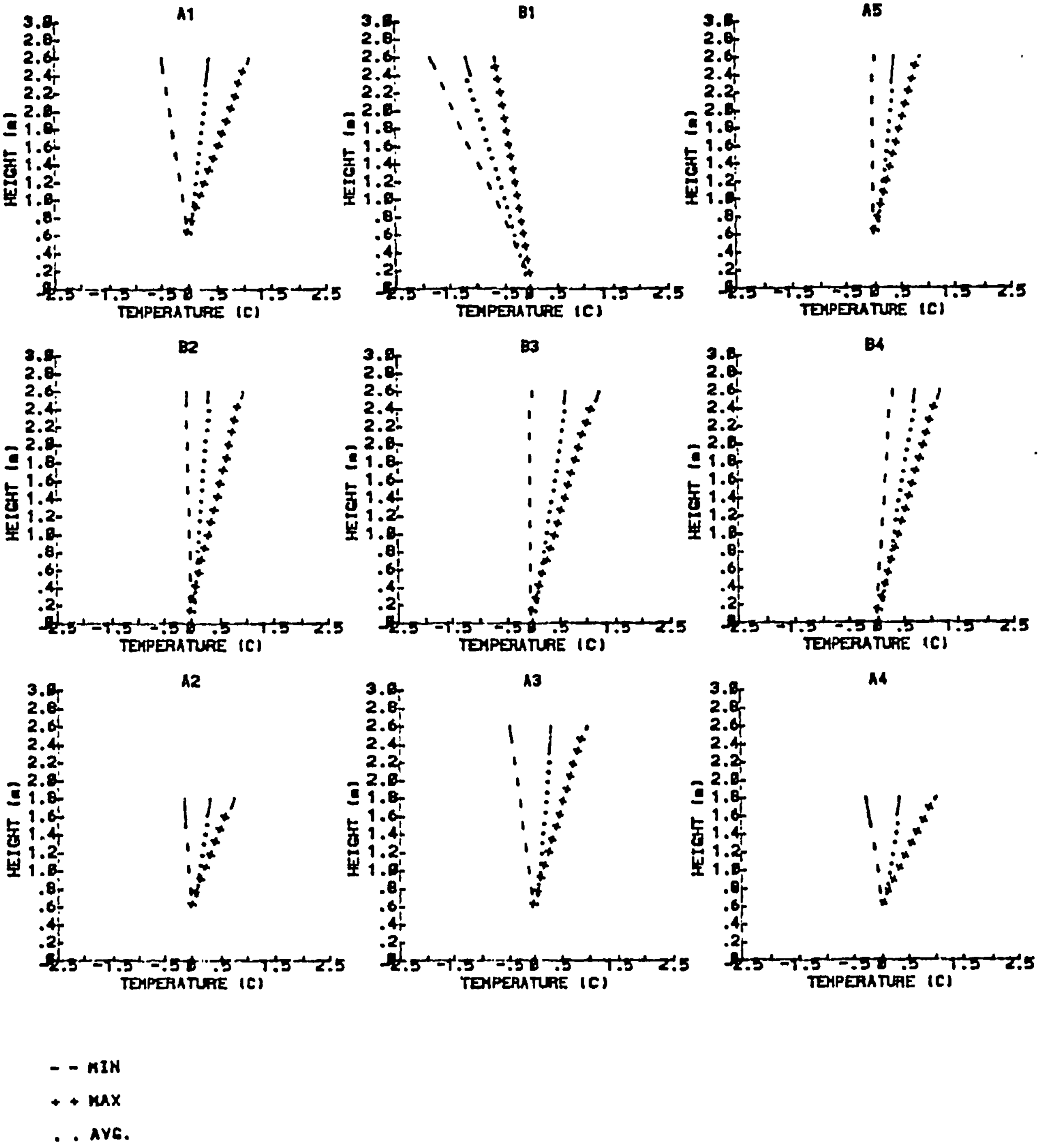


Figure 5.6: First Order Approximation of Temperature Profiles, Hydronic System (T_{out} average=-7.1°C, see fig. 5.2)

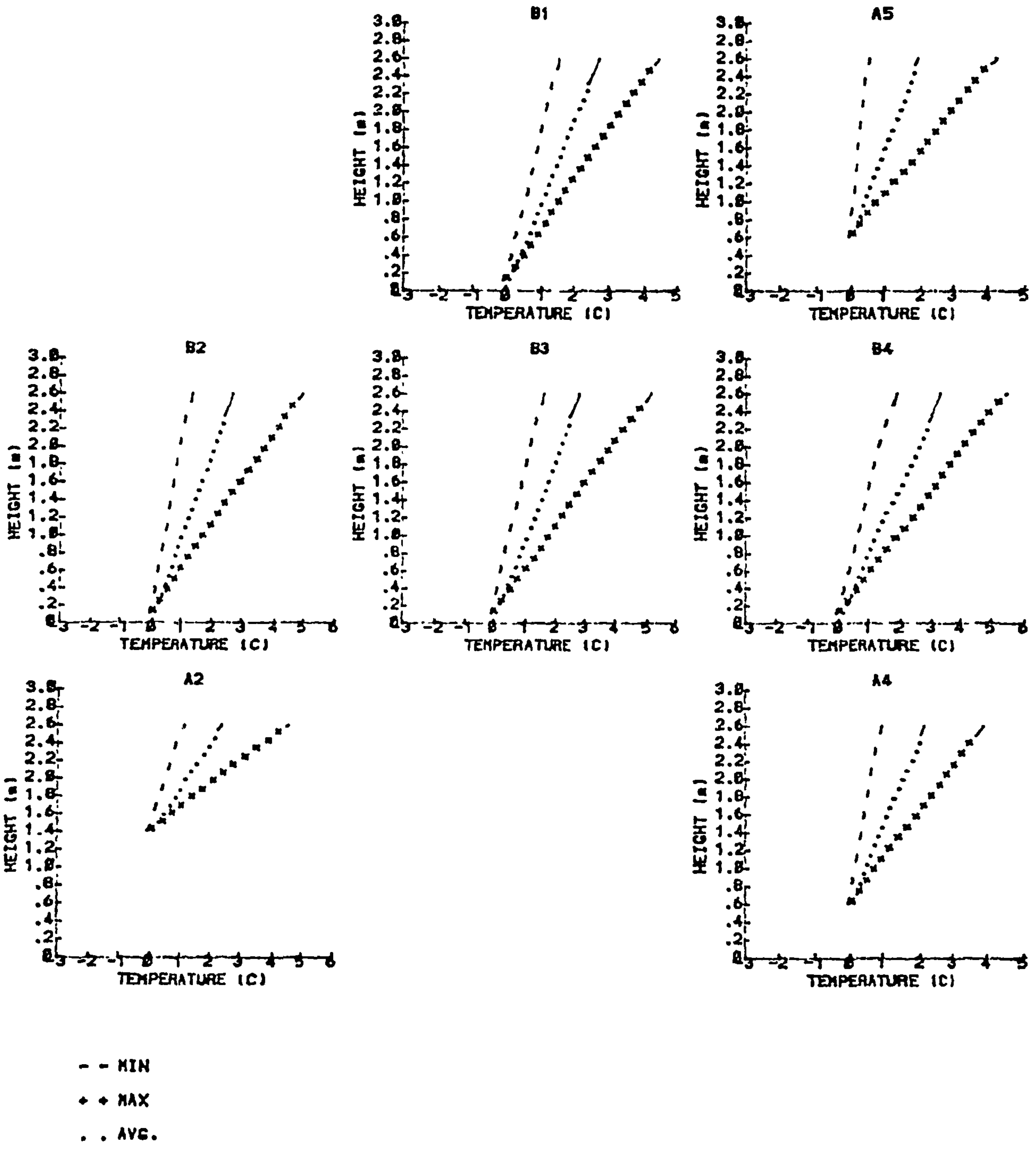


Figure 5.7: First Order Approximation of Temperature Profiles, Modular 1st Floor (T$_{out}$ average=-6.7°C, see fig. 5.3)

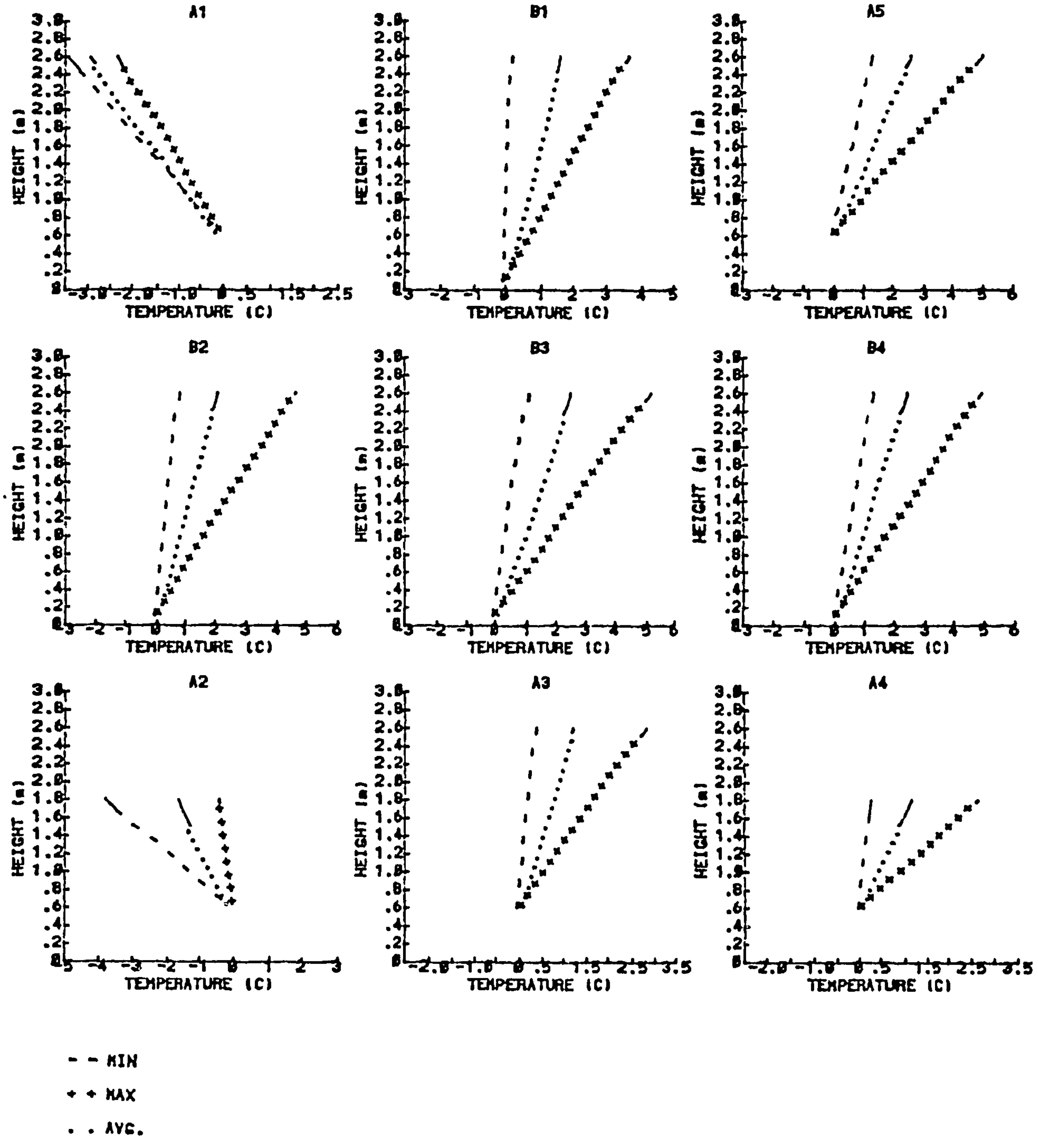


Figure 5.8: First Order Approximation of Temperature Profiles, Modular 3rd Floor (T_{out} average=-7.5°C, see fig. 5.4)

The results for the hydronic system are comparable to those of the forced air system. The gradients are uniform throughout the room except for location B1 where negative gradients occur. The standard deviation at all points is also small (less than 0.3°C/m) for all locations.

The modular systems create the highest gradients in the rooms on both floors. Comparing Figures 5.7 and 5.8 and also the values in Table 5.3, the results are similar for the 1st and the third floor. The exception is location A2 where high gradients occurred on both floors (with different signs). The maximum standard deviation (0.7°C/m) of the gradients during the recording period for the modular systems is approximately two times higher than for the other two systems indicating that the profiles vary more in time.

The above results are for a linear interpolation of the temperature profiles and are included here to describe the temperature profiles for a large number of locations in the room. Note that when the coefficients of equation 5.1 were obtained, the error in calculating the temperature at the measured heights was calculated. As shown in Table 5.4, the maximum and average errors can be as high as 1.5°C. This means that the profiles included so far describe only the general trend in the data.

Results from Third Order Approximation

In order to obtain a better representation of the gradients the temperature profiles over the measurement periods described above were found using equation 5.2. The results of the third order interpolation are shown on Figures 5.9 to 5.12 for the forced air, the hydronic and the modular systems on the first and third floors respectively. The numerical results are listed in the same order in Tables 5.5 to 5.8. The tables include the errors found in the approximations.

Forced Air System

From Figure 5.9 and Table 5.5, the following observations for the forced air system can be made:

-There are two types of shape for the temperature profiles. The first one is an S-shape increase of the temperatures (locations B1 and B2), and the second one is a near linear shape (locations B3 and B4).

-The greatest increase in temperature occurs between 0.1m and 1.0m.

-The average and minimum gradients for all locations except B4 are close to 0°C/m.

-The maximum error for the interpolation over the entire period (table 5.5 c) is less than one degree for all grid locations which indicates that the average temperature profile provides a good description of the temperature profiles.

-The maximum error in the interpolation is less than 0.6°C for maximum and minimum with an average error less than 0.5°C (table 5.5 a and b). This indicates that equation 5.2 is sufficiently accurate.

-The temperature profiles are distributed over a wider range for locations closer to the neighbour's wall (locations B3 and B4).

-The difference in profile at location B4 cannot be explained by the temperature of the neighbour wall next to this grid location since the temperature of that wall is approximately the same as for the other walls (see Table 5.1).

Table 5.5: Results for 3rd Order Interpolation, Forced Air

Table 5.5 a): Results for Maximum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	22.01	22.15	22.31	22.3
a_1	2.15	3.18	2.05	0.43
a_2	-1.44	-1.98	-0.84	0.48
a_3	0.36	0.46	0.16	-0.16
Max. Error (°C)	0.3	0.1	0.6	0.2
Avg. Error (°C)	0.1	0.1	0.3	0.1

Table 5.5 b): Minimum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	22.54	22.10	22.69	24.65
a_1	0.82	1.86	-0.11	-3.44
a_2	-0.34	-0.75	0.26	2.62
a_3	0.03	0.06	-0.03	-0.58
Max. Error (°C)	0.3	0.1	0.3	0.2
Avg. Error (°C)	0.1	0.0	0.1	0.1

Table 5.5 c): Average Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	22.46	22.08	22.50	23.47
a_1	1.49	2.84	0.36	-1.42
a_2	-0.94	-1.93	0.35	1.57
a_3	0.21	0.44	-0.11	-0.38
Max. Error (°C)	0.9	0.8	1.1	0.9
Avg. Error (°C)	0.3	0.3	0.3	0.3
r^2	0.5	0.7	0.6	0.4

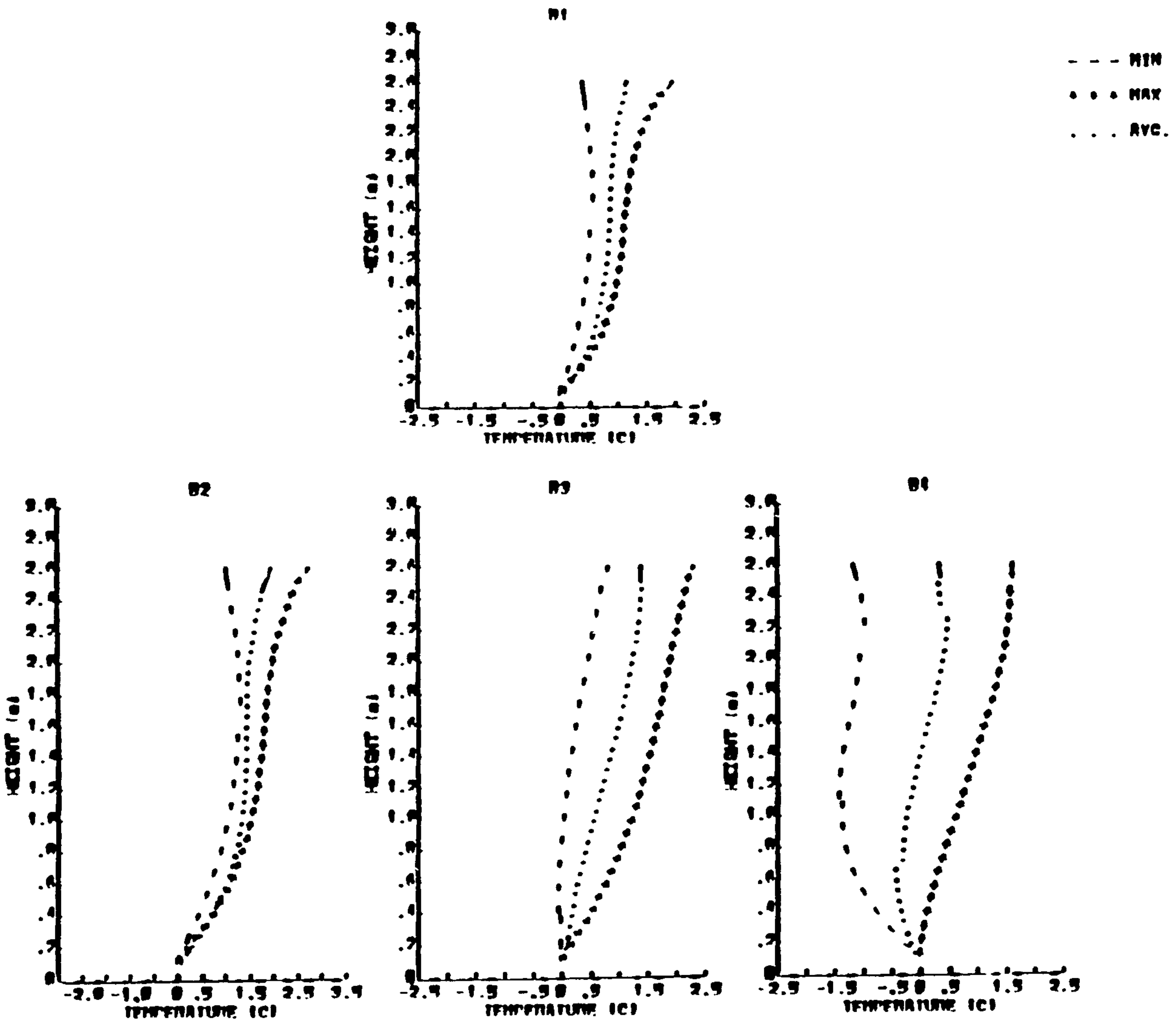


Figure 5.9: Third Order Approximation of Temperature Profiles, Forced Air System (T_{out} average=-6.1°C, see fig. 5.1)

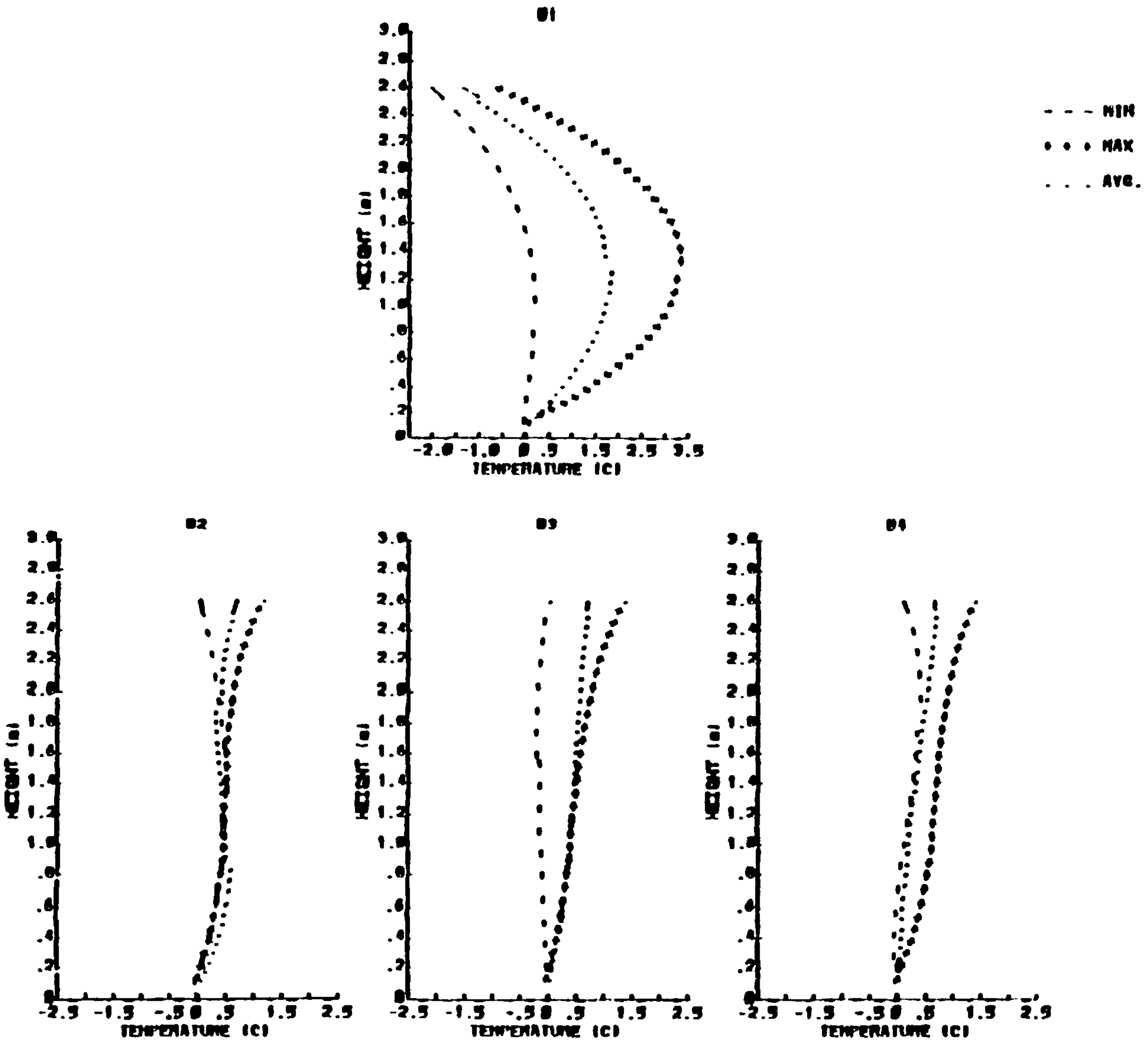


Figure 5.10: Third Order Approximation of Temperature Profiles, Hydronic System
(T_{out} average=-7.1°C, see fig. 5.2)

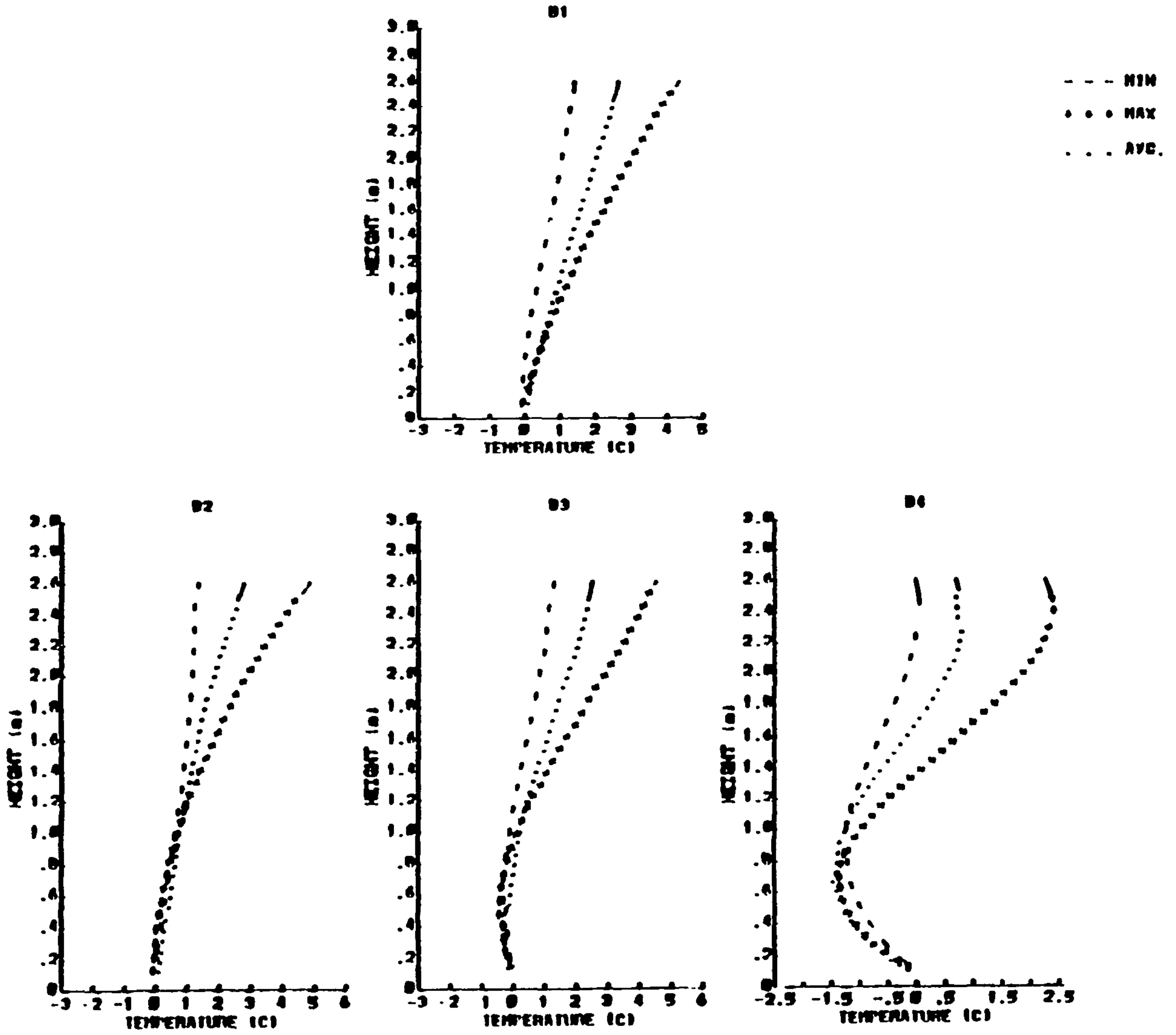


Figure 5.11: Third Order Approximation of Temperature Profiles, Modular 1st Floor (T_{out} average=-6.7°C, see fig. 5.3)

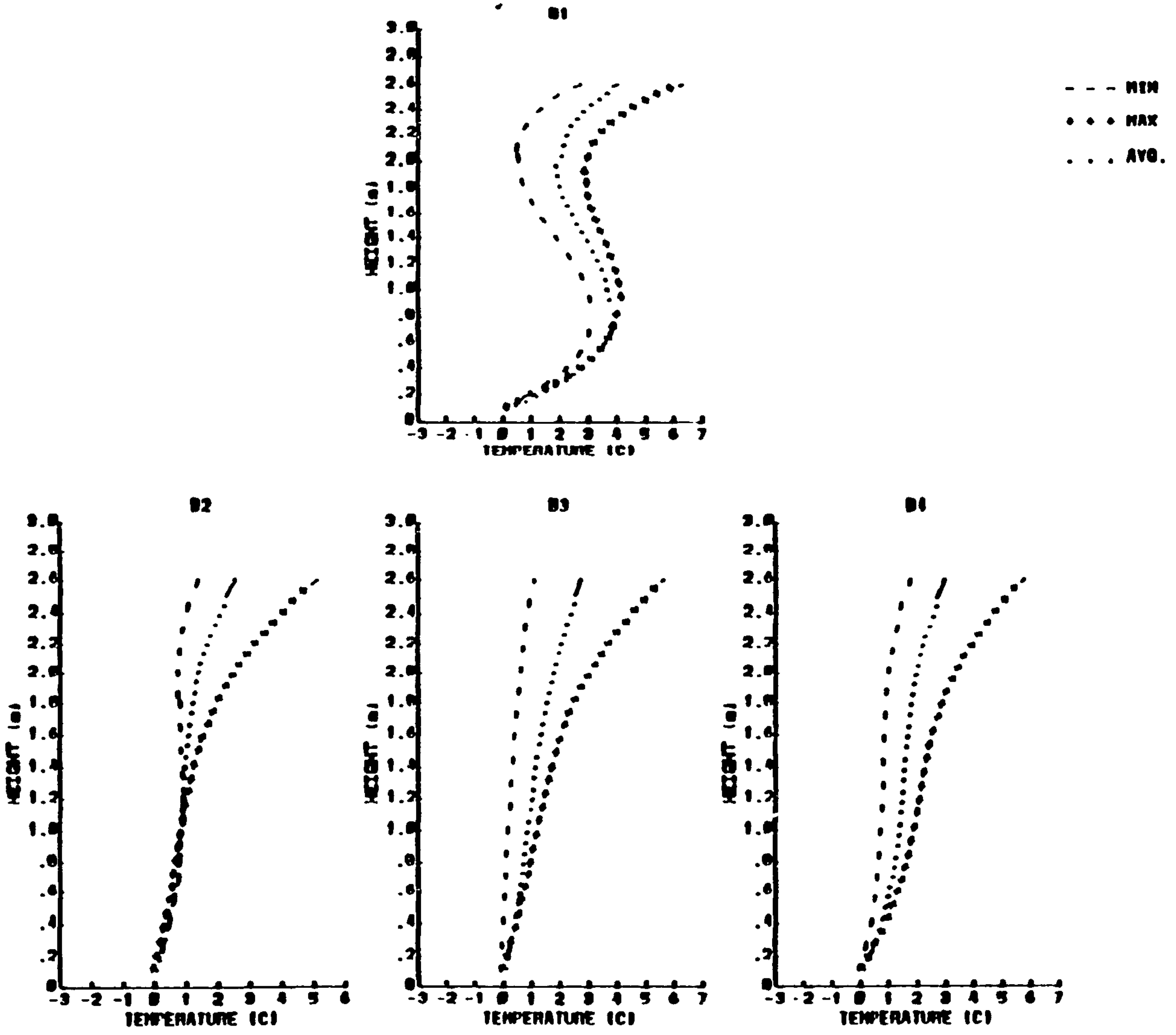


Figure 5.12: Third Order Approximation of Temperature Profiles, Modular 3rd Floor (T_{out} average=-7.5°C, see fig. 5.4)

Hydronic System

From Figure 5.10 and Table 5.6, the following observations for the hydronic system can be made:

-The temperature profile created by the hydronic system is characterized by high variations of temperatures at the location close to the baseboard (location B1) but relatively small gradients elsewhere (table 5.6).

-At the location closest to the baseboard (location B1), temperatures increase steadily between the 0 m and 1.5 m levels to decrease afterwards. This increase can be as high as 2.5°C and is on the average close to 1.5°C.

-The temperatures in the center of the room are close to vertical straight lines with a small increase in the last portion above the 2.0 m level. This distribution is repeated across the room.

-The maximum error for the interpolation over the entire period (table 5.6 c) is less than 1.5 degree for all grid locations which indicates that the average temperature profile provides a good description of the temperature profiles. This is also reflected in the value of r^2 which indicate that the average accounts for more than 40% and up to 87% of the observations.

-The maximum error in the interpolation is less than 0.3°C for maximum and minimum (Tables 5.6 a and b) calculated by using equation 5.2 with average errors less than 0.1°C. This indicates that equation 5.2 is sufficiently accurate.

Table 5.6 : Result for 3rd Order Interpolation, Hydronic

Table 5.6 a): Results for Maximum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	20.86	19.43	19.42	19.10
a_1	5.55	1.17	0.87	1.29
a_2	-2.28	-0.88	-0.56	-0.82
a_3	-0.02	0.24	0.18	0.21
Max. Error (°C)	0.0	0.1	0.2	0.1
Min. Error (°C)	0.0	0.0	0.1	0.1

Table 5.6 b): Minimum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	21.83	19.43	19.81	19.92
a_1	0.46	1.04	-0.11	-0.17
a_2	-0.06	-0.56	-0.07	0.58
a_3	-0.18	0.06	0.05	-0.20
Max. Error (°C)	0.0	0.1	0.2	0.1
Min. Error (°C)	0.0	0.0	0.1	0.0

Table 5.6 c): Results for Average Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	21.06	19.45	19.66	19.65
a_1	2.76	1.61	0.65	0.06
a_2	-0.76	-1.39	-0.34	0.17
a_3	-0.22	0.34	0.08	-0.03
Max. Error (°C)	1.2	0.7	0.5	0.7
Min. Error (°C)	0.3	0.2	0.2	0.2
r^2	0.9	0.4	0.5	0.6

-The temperature profiles are distributed over a wider range for locations closer to the neighbour's wall (location B3 and B4).

-Even with the wall close to location B4 (neighbour wall) being on the average at a temperature less than the other walls, the effect the temperature profile at that location is not different from the other locations.

Modular System, 1st Floor

From Figure 5.11 and Table 5.7, the following observations on the modular system for the first floor can be made:

-The temperature profiles created by the modular system on the first floor show that most of the temperature increases occur between the 1.2 m and 2.4 m levels in a close to linear fashion. The different profiles in the room are similar except for location B4.

-Negative gradients occur only at the location closest to the neighbour's wall (location B4) and between the 0.1 m and 0.8 m level. Elsewhere, the temperature profiles are similar.

-The average and minimum increases in temperatures are close to zero for all locations except B4.

-The different temperature profile at location B4 could be explained by the lower temperature of the neighbour wall, and the back wall (table 5.1) relative to the other walls' temperatures.

-The maximum errors for the interpolation over the entire period (table 5.7 c) are between 1.4 and 2.0 degree for all grid locations which indicates that the average temperature profile does not provide a good description of the temperature profiles. The values of r^2 indicate that the average accounts for more than 75% of the observations. The higher values of r^2 indicate in this case that the increase in temperature with height is more important.

-The maximum error in the interpolation is less than 0.3°C for maximum and minimum (Tables 5.7 a and b) with average errors less than 0.2°C. This indicates that equation 5.2 is sufficiently accurate.

Modular System, 3rd Floor

As noted previously, due to the presence of a small amount of heat coming from the return pipe of the hydronic system, the results for the third floor are discussed separately and compared here with the results obtained on the first floor.

From Figure 5.12 and Table 5.8, the following observations for the modular system on the third floor can be made.

-The temperature profiles created by the modular system on the third floor are similar to those observed on the first floor Most of the temperature increases occur between the 1.2 m and 2 4 m levels in a close to linear fashion. The different profiles in the room are similar except for locations B1 and B4

-The effect of the heat delivered by the return pipe of the hydronic system is to modify the gradients between the 0 1 m and the 1.2 m level at location B1 (see location B1, Figure 5.12). The shape instead of being linear is curved between those levels This indicates that with the temperatures under the window increased by approximately 3°C at constant value the effect is felt directly on the temperature profile at location B1 By comparing with the results on the first floor, the temperature profiles at locations B2 and B3 are the same

-The average and minimum increases in temperatures are close to 1°C for all locations except B1

-The maximum errors for the interpolation over the entire period (table 5 8 c) are over 2.0°C higher for all grid locations which indicates that the average temperature profile does not provide a good description of the temperature profiles The values of r^2 indicate that the averages account for between 65 and 80% of the observations which indicates in this case that the increase in temperature with height is significant.

-The maximum error in the interpolation is less than 0.4°C for maximum and minimum (Tables 5.8 a and b) and with errors on the average less than 0.3°C. This indicates that equation 5.2 is sufficiently accurate for describing the profiles.

Table 5.7: Results for 3rd Order Interpolation, Modular 1st

Table 5.7 a): Minimum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	21.06	21.66	21.87	23.02
a_1	0.93	0.19	-1.95	-5.34
a_2	0.48	0.77	2.94	6.03
a_3	-0.06	-0.03	-0.58	-1.41
Max. Error (°C)	0.2	0.0	0.2	0.0
Avg. Error (°C)	0.1	0.0	0.1	0.0

Table 5.7 b): Minimum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	21.34	21.32	21.78	22.91
a_1	0.04	0.84	-0.99	-3.70
a_2	0.57	-0.02	1.32	3.28
a_3	-0.14	-0.04	-0.29	-0.72
Max. Error (°C)	0.2	0.1	0.1	0.1
Avg. Error (°C)	0.1	0.0	0.1	0.1

Table 5.7 c): Results for Average Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	21.14	21.51	21.85	23.23
a_1	0.76	0.86	-0.98	-4.85
a_2	0.23	-0.17	1.57	4.74
a_3	-0.04	0.11	-0.31	-1.08
Max. Error (°C)	1.5	2.1	2.0	1.4
Avg. Error (°C)	0.3	0.3	0.4	0.3
r^2	0.80	0.76	0.8	0.80

Table 5.8: Result for 3rd Order Interpolation, Modular 3rd Floor

Table 5.8 a): Results for Maximum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	19.20	20.14	19.83	19.59
a_1	12.17	1.60	2.15	3.88
a_2	-11.04	-1.15	-1.29	-2.61
a_3	2.87	0.53	0.53	0.79
Max Error (°C)	0.0	0.1	0.1	0.5
Avg Error (°C)	0.0	0.1	0.0	0.2

Table 5.8 b): Minimum Gradients, Coefficients and Errors

	B1	B2	B3	B4
a_0	19.84	20.19	20.66	20.29
a_1	10.68	2.45	0.23	1.87
a_2	-10.67	-2.05	0.06	-1.38
a_3	2.73	0.52	0.01	0.36
Max Error (°C)	0.00	0.1	0.1	0.3
Avg Error (°C)	0.00	0.0	0.1	0.1

Table 5.8 c) Results for Average Gradients, Coefficients and Errors

	A1	B1	B2	B3	B4
a_0	23.05	19.26	20.34	20.25	20.03
a_1	-1.47	11.76	1.78	1.24	2.72
a_2	0.00	-10.69	-1.42	-0.59	-1.84
a_3	0.00	2.65	0.44	0.21	0.49
Max Error (°C)	2.1	2.1	2.3	2.1	2.2
Avg Error (°C)	0.9	0.4	0.3	0.3	0.4
r^2	0.50	0.81	0.69	0.76	0.68

5.2.2 Temperature Profile at Milder Outdoor Temperatures

In the previous sections, the shape of the temperature profiles for the four heating systems for the coldest day for which data were available were described. Since the day selected was one of the coldest, it is important to note that the temperature gradients may be less important on other occasions. The results above apply for the worst case observed, and included here are profiles obtained for milder outdoor conditions.

Figures 5.13 to 5.16 show the temperature profiles obtained for the forced air, hydronic and modular systems respectively for outdoor conditions listed in Table 5.9.

Forced Air

For the forced air system, comparison of Figure 5.13 with Figure 5.9 indicates that the temperature profile for the milder temperatures is similar to the profile at colder temperature. The difference is that the gradients are less pronounced in the case of milder temperatures Also, the range of the gradients at location B4 is not as wide as for the coldest case even though location B4 still has greater gradients than for the other locations.

Table 5.9: Outdoor Conditions for Temperature Profiles at Milder Outdoor Temperatures.

Heating System	Outdoor Temperature
Forced Air and Hydronic (figure 5.13-5.14)	constant at -2°C
Modular 1st (figure 5.15)	between 0°C and -3°C
Modular 3rd (figure 5.16)	between 0°C and -4°C

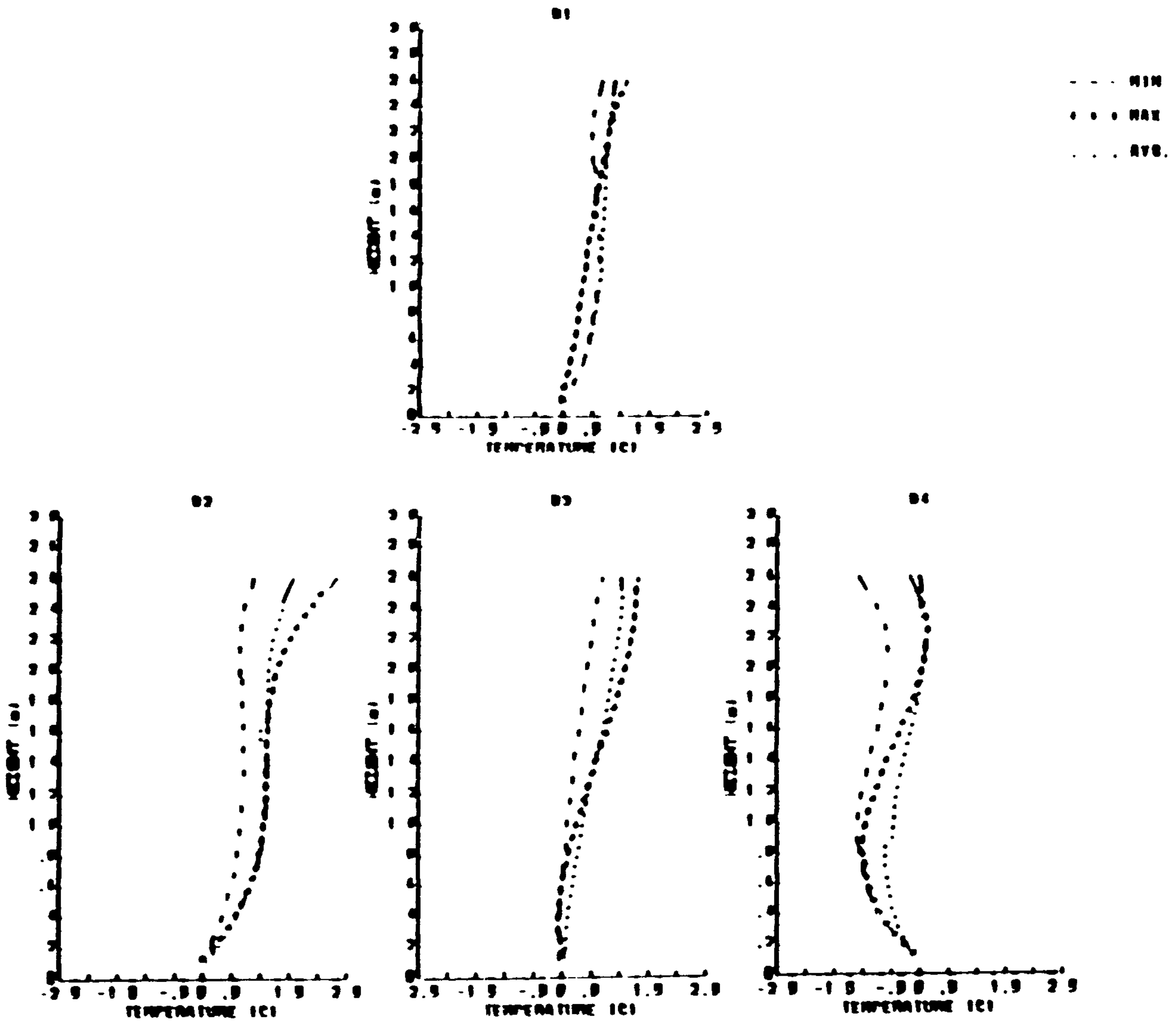


Figure 5.13: Temperature Profiles, Mild Outdoor Conditions, Forced Air (T_{out} average=-2.0°C)

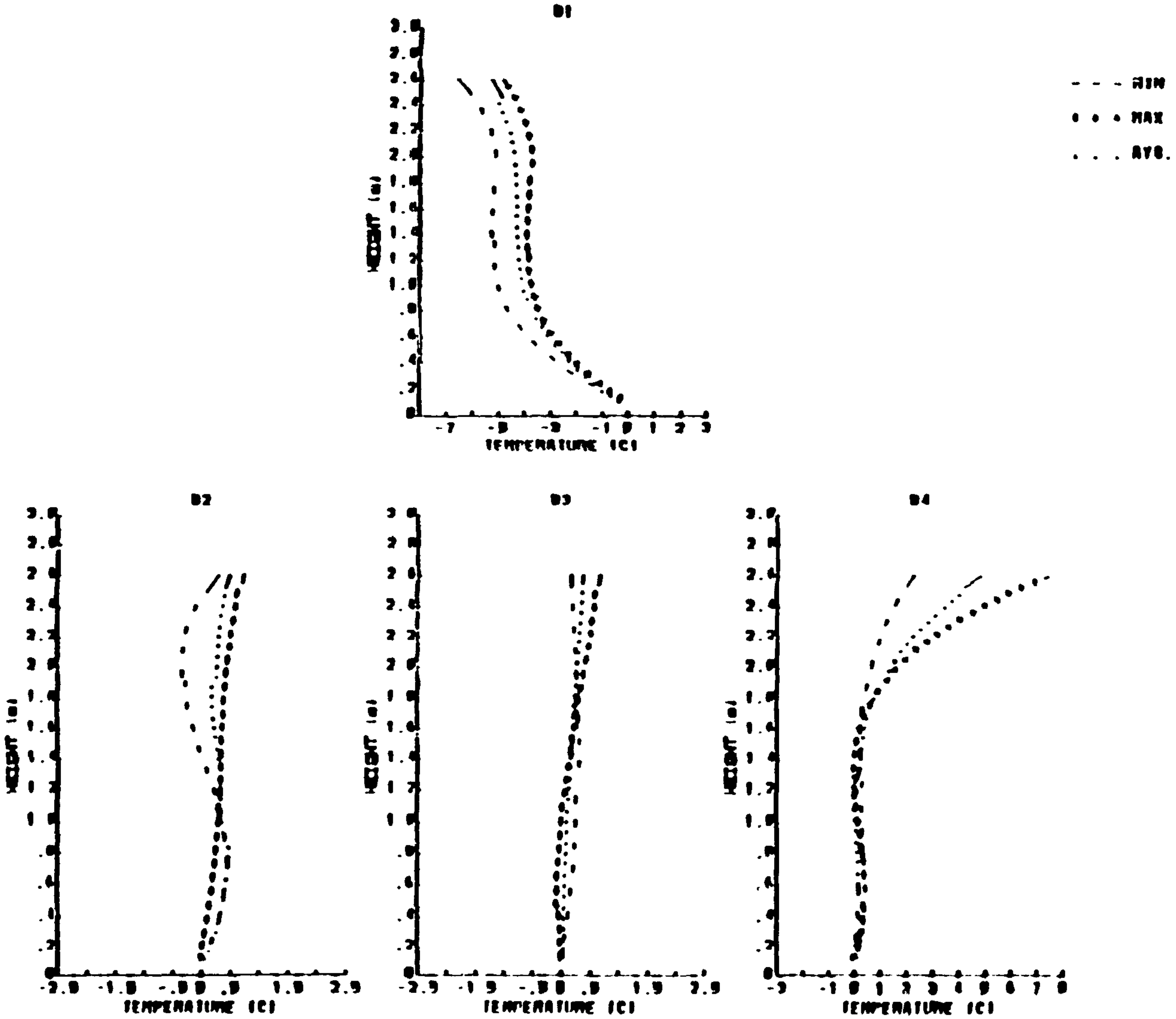


Figure 5.14: Temperature Profiles, Mild Outdoor Conditions, Hydronic (T_{out} average=-2.0°C)

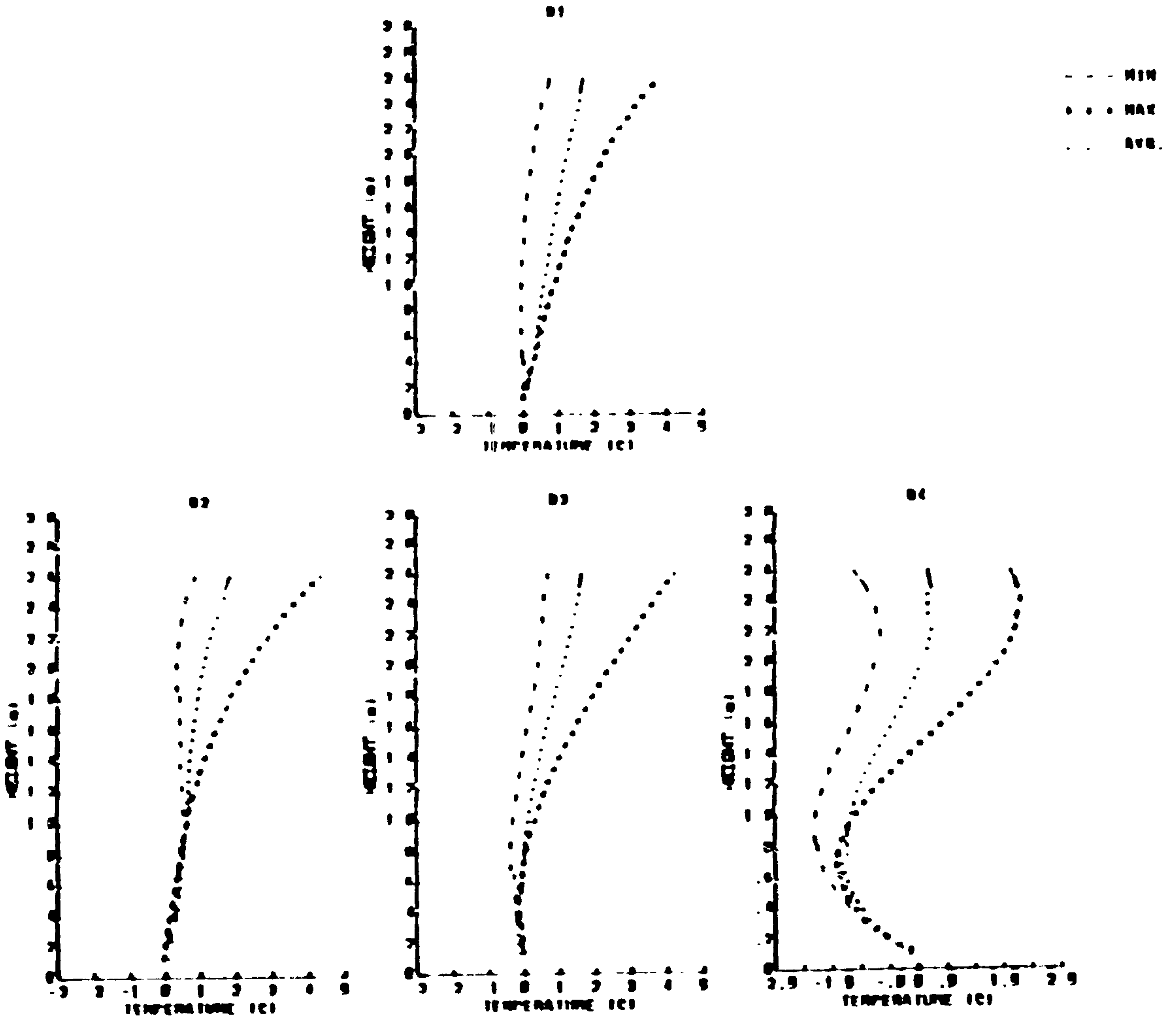


Figure 5.15: Temperature Profiles, Mild Outdoor Conditions, Modular 1st Floor (T_{oa} average=-2.0°C)

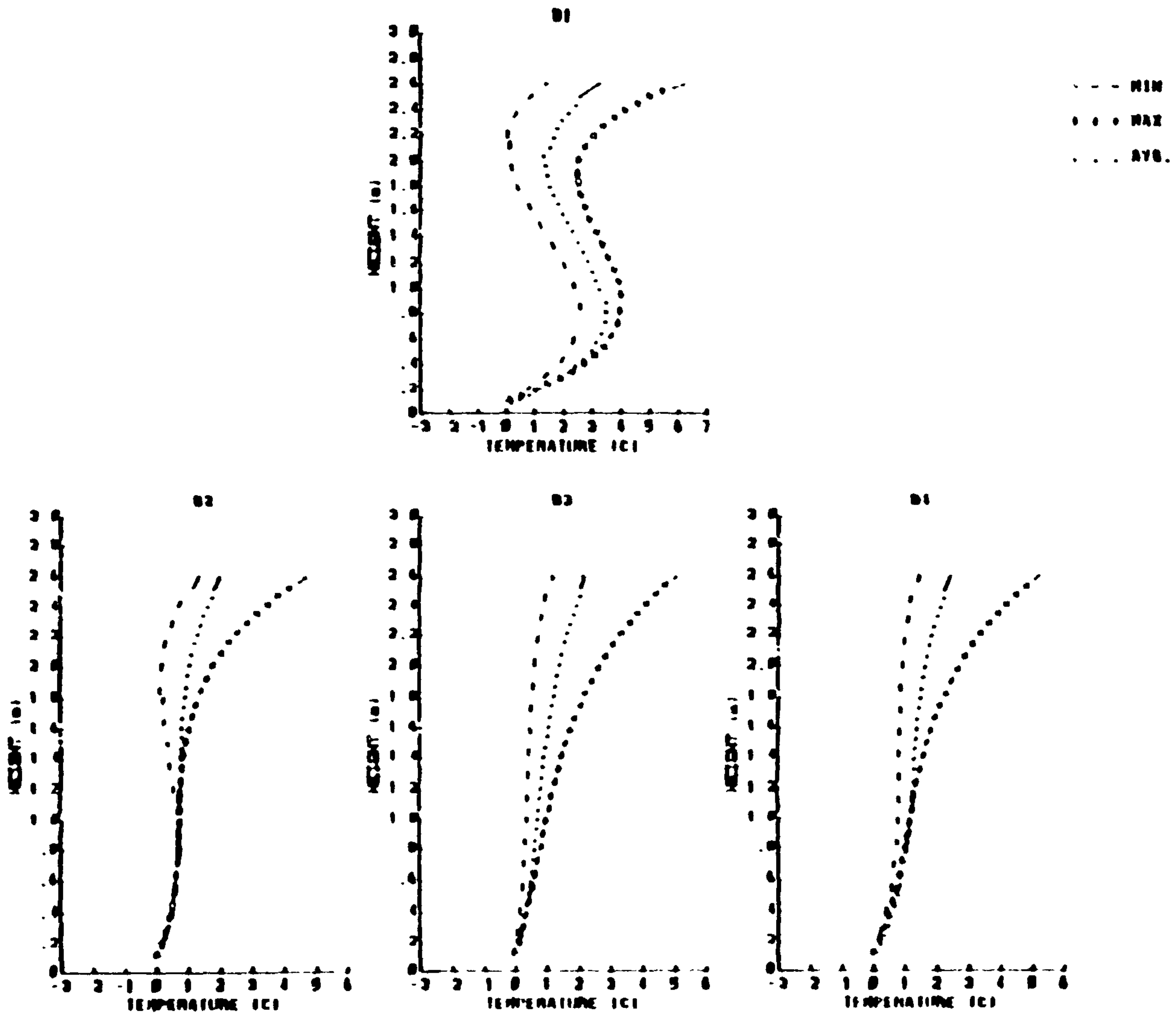


Figure 5.16: Temperature Profiles, Mild Outdoor Conditions, Modular 3rd Floor (T_{out} average=-3.0°C)

Hydronic System

For the hydronic system, the gradients at locations B2 and B3 are similar for the milder outdoor temperature (Figure 5 14) and for the lower outdoor temperature (Figure 5.10) For locations B4 and B1, differences exist

At first glance, the profile at location B1, when compared with the other locations, seems to indicate that temperatures there are much colder there than in the rest of the room In fact, since the temperature profiles shown are only relative to the 0 1 m level, the profile shows that temperatures in the lower level are much higher at that level (by approximately 5°C)

There is a significant difference in the temperature profiles at location B1 when the hydronic system operates at lower outdoor temperatures The baseboard produces a highly concentrated mass of warm air close to the floor caused by the lower temperatures at which the baseboard operates (see section 4 3 2 under "Hydronic system" , item iii "case to case variations") Therefore there is an increase of temperature between the 0 1 and 0 6 m level (Figure 5 14) instead of the increase between 0 1 m and 2 4 m levels (Figure 5 10) The major increase occurs lower and is by almost 2°C more important

From the results the shape of the temperature profiles observed for the hydronic system was found to change depending on the outdoor temperature at which the system operates

Modular Systems

For the modular systems, comparison of Figure 5 11 with Figure 5 15, and Figure 5 12 with Figure 5 16, shows that the gradients are not significantly smaller at milder temperatures than at the colder temperature In fact, the shape of the temperature profile in both cases is almost identical

From the results of the two cases observed, the same temperature profiles are expected at different outdoor conditions of operation for the modular system.

5.2.3 Relation Between Gradients and Heaters' Supply Temperature

From section 5.2.1, it was seen that for the modular systems, the errors for the average temperature profiles are higher than for the other systems. This indicates that the profiles vary more in time and can not be as well represented by an average. Therefore, it can be asked if these variations occur at times when the systems are at the peak temperature in their cycles For the cases described in section 5.2.1, it was decided to verify a relationship between the magnitude of the gradients' slope as (equation 5.1) and the temperature of the heater could be found

At first, a graph of the gradients versus heater temperature at B types of locations was plotted in Figures 5.17 to 5.20 for the forced air, the hydronic and the modular systems (1st and 3rd floors) respectively.

The graphs show that for the forced air and the hydronic systems (Figures 5 17 and 5 18) there is no apparent relationship between heater temperature and the magnitude of the gradients The points are equally distributed on either sides of a line Also, since the gradients are almost constant, a relationship is not possible

The case of the modular systems is different. The shape of the curves for both floors (Figures 5.19 and 5.20) indicate that there is a relationship between gradients' magnitude and heater temperature. Even though, the relationship does not account for all the points, there is a trend. More important, the trend is the same for both floors.

The curves for the modular systems shows that gradients increase sharply for heater temperatures up to a approximately 50°C. Passed 50°C, the gradients are stable and start increasing again beyond 70°C.

Results using a third order approximation for the relationship between gradient magnitude and heater temperature showed that such type of approximation could account for between 65% and 80% of the experimental results obtained for the modular systems. These results are not presented here since more research and some modifications in the experimental procedure would be required to validate this type of relationship

The observed effect of the heater temperature variations on the changes in temperature profiles can be illustrated by showing the gradients as they change in time for each of the systems Figure 5 21 shows the outdoor temperatures and the heater temperatures for the four heating systems The starting times were determined so that the periods include at least one cycle for each system The outdoor temperatures were selected to be more or less constant and close to each other for the different systems

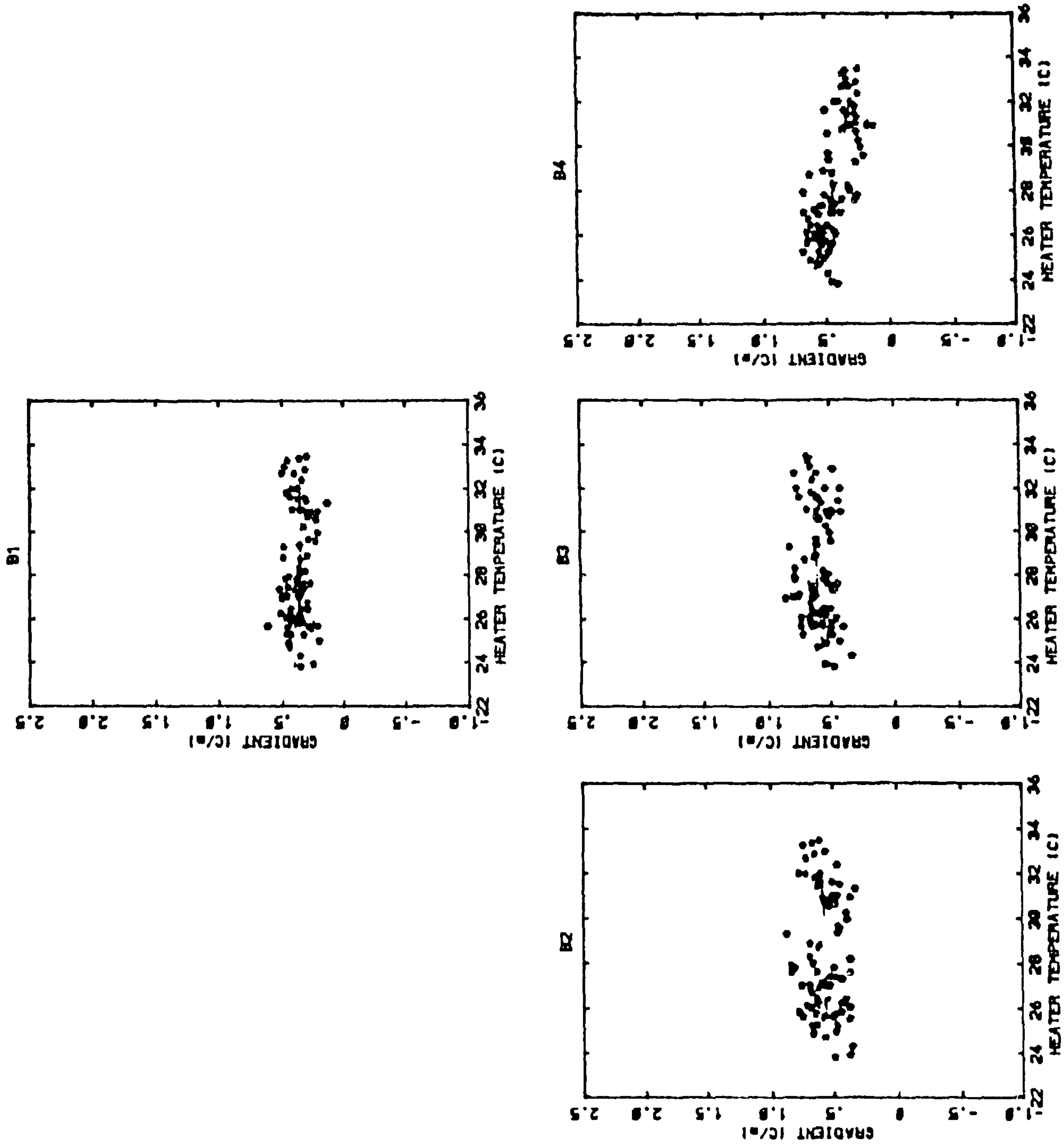


Figure 5.17: Gradients Versus Heater Temperature, Forced Air (see figure 5.1)

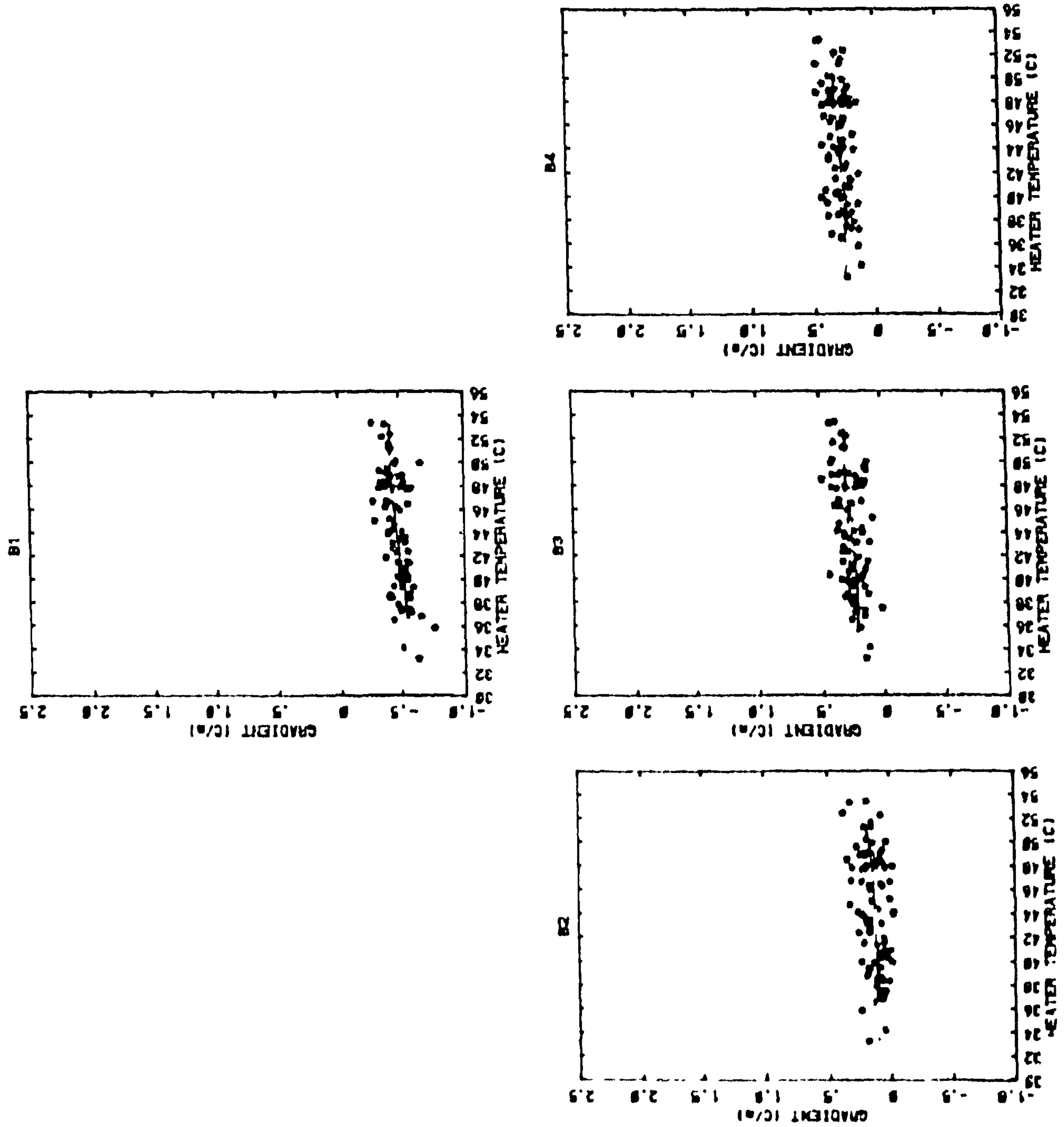


Figure 5.18 Gradients Versus Heater Temperature, Hydronic (see figure 5.2)

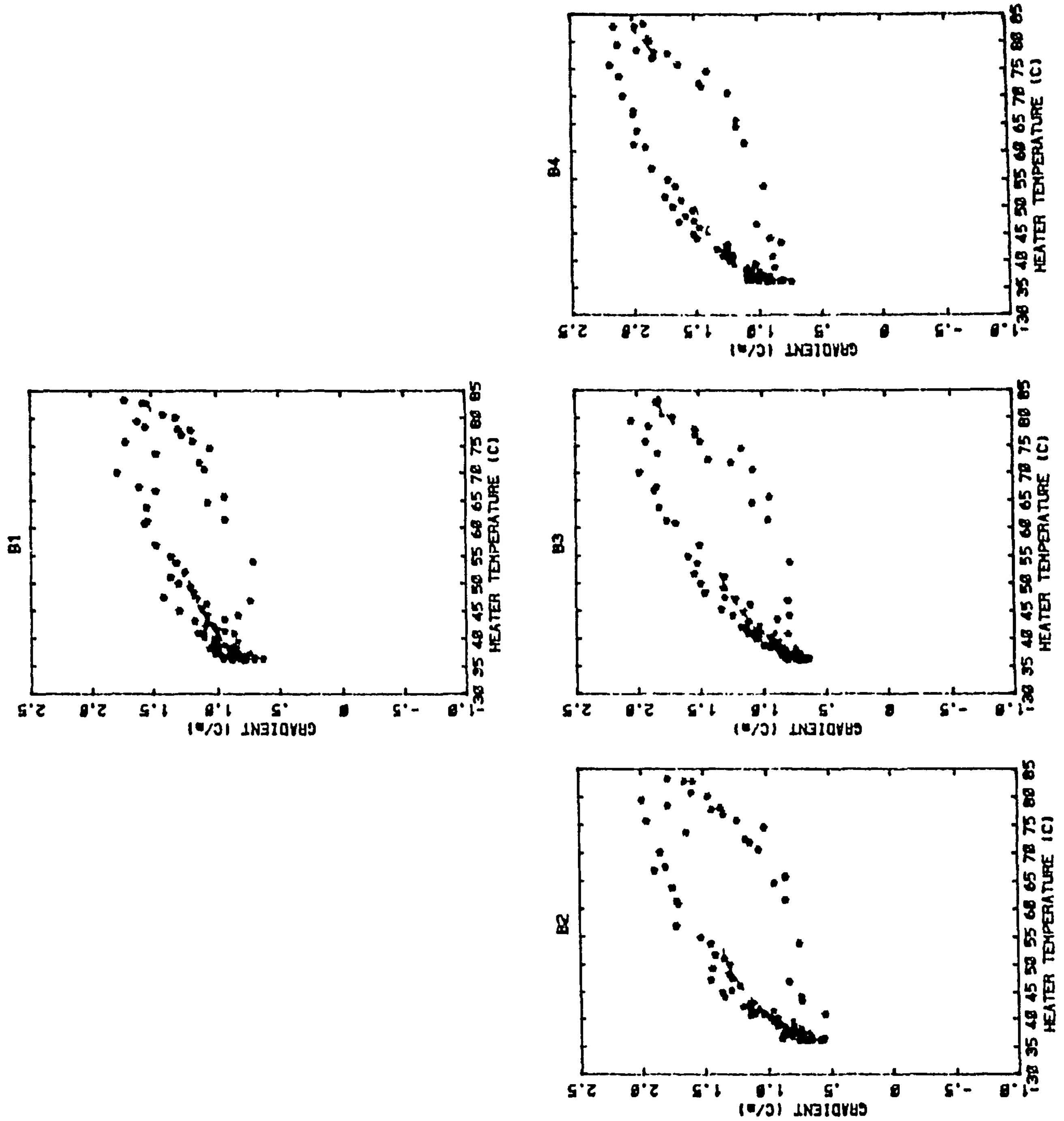


Figure 5.19: Gradients Versus Heater Temperature, Modular 1st (see figure 5.3)

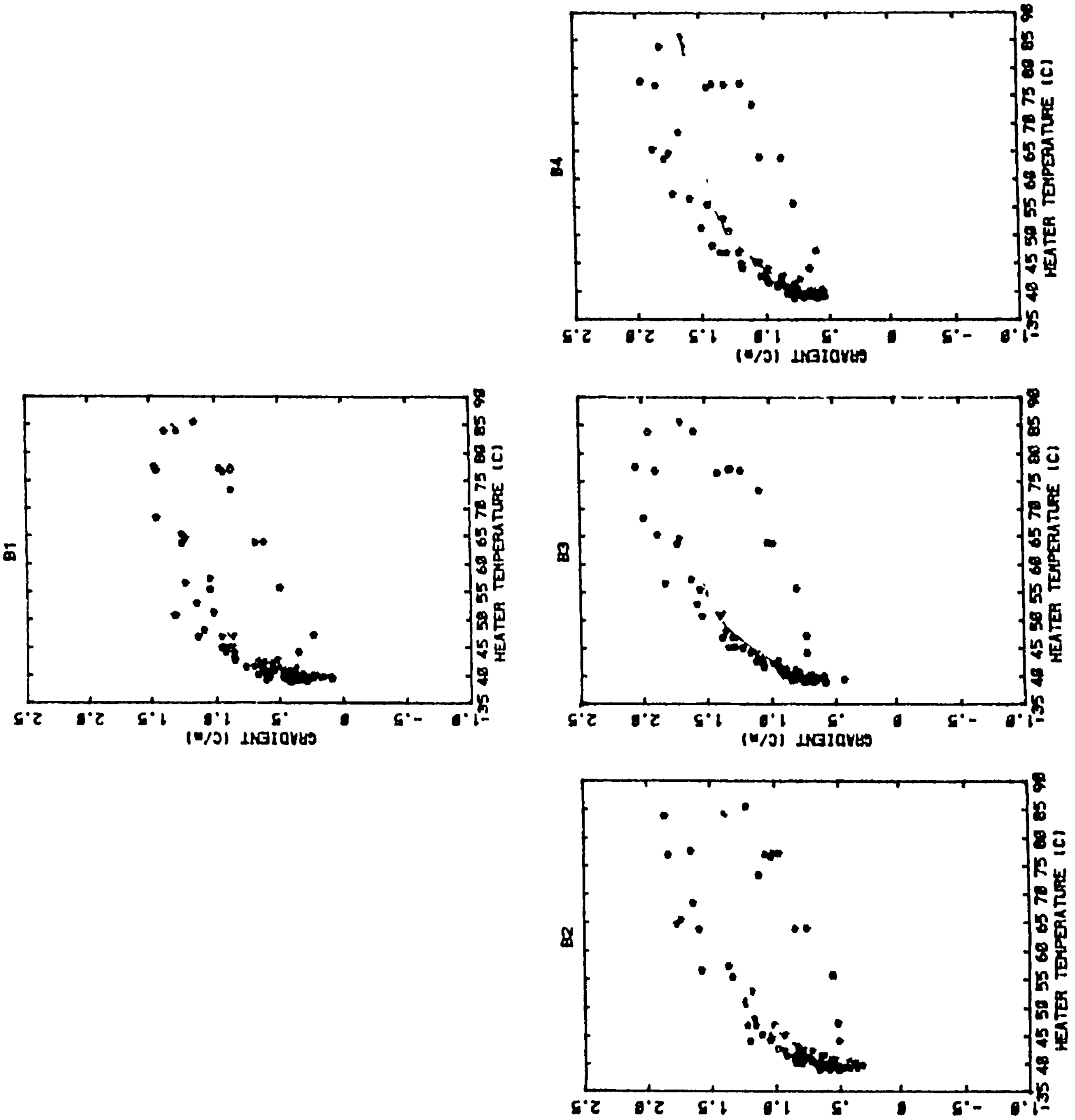


Figure 5.20: Gradients Versus Heater Temperature, Modular 3rd (see figure 5.4)

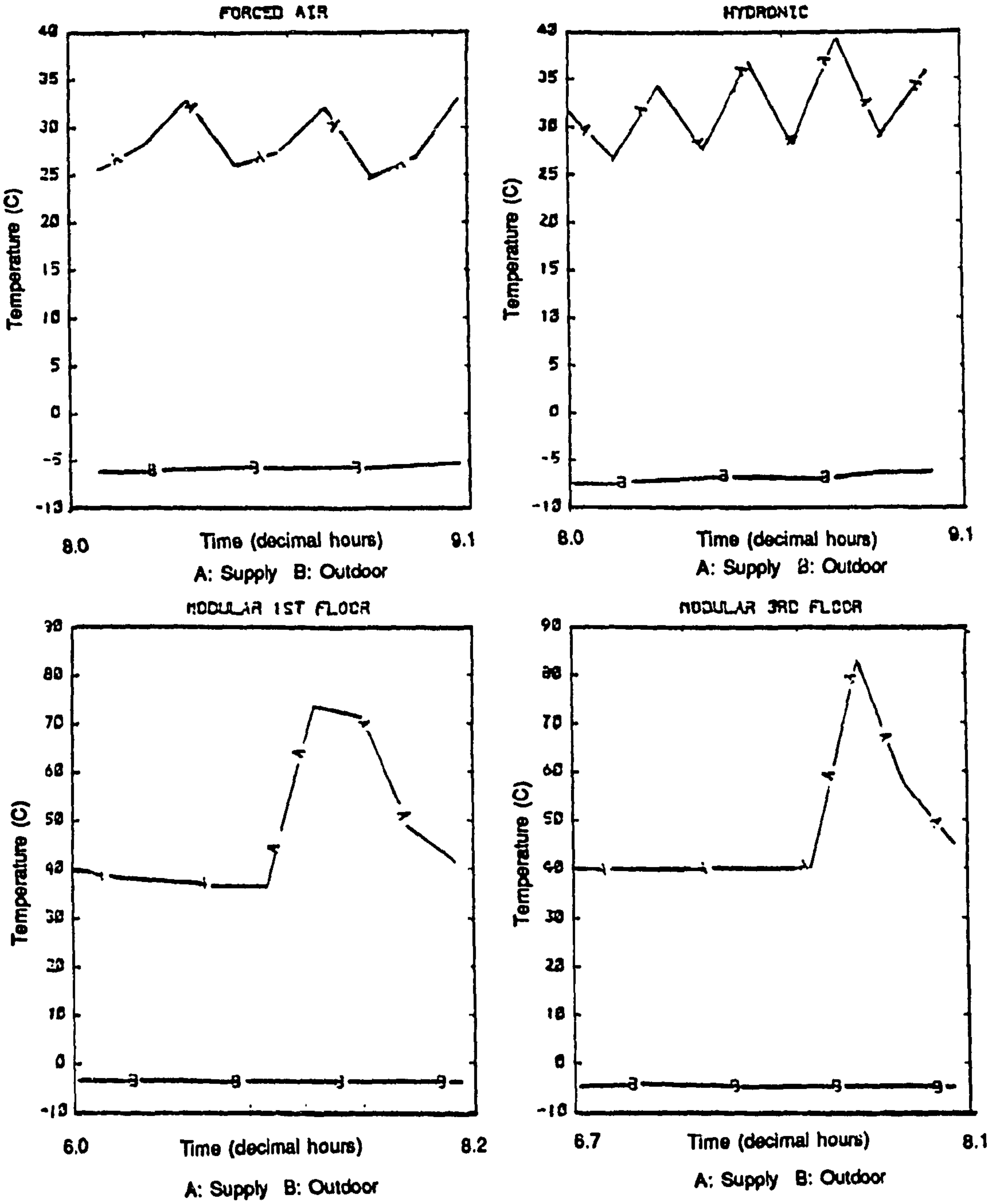


Figure 5.21: Outdoor and Heater Temperatures for Short Period Observation of Temperature Profiles

Figure 5.22 to 5.25 show the temperature profiles in time for the same periods as in Figure 5.21 for the forced air, the hydronic and the modular systems respectively at location B3, center of the room. Profiles were calculated using equation 5.2.

First, Figures 5.22 and 5.23 show that for the forced air and the hydronic systems, the temperature profiles are not significantly influenced by the heater's temperature in time for a cycle. As the systems cycles on and off in time, no major disturbance occurs in the profiles.

On the other hand, for the modular systems (Figures 5.24 and 5.25), the increase in the modular's temperature is immediately followed by a change of temperature profile above the 1.2 meter level. Below that level, temperatures remain fairly constant.

The fact that a relationship exists for the modular systems indicates that the temperature of the heater is important for gradients and this within a specific range. The gradients observed for the hydronic and forced air systems did not present a particular relationship and also these systems had lower supply temperatures than the modular systems.

The magnitude of the gradients for the modular systems increased the most for supply temperatures between 35°C and 50°C. This indicates that this range, for the conditions studied, is the most likely to produce high gradients The supply temperatures for the other systems did not go higher than 35°C and also no significant gradients were observed.

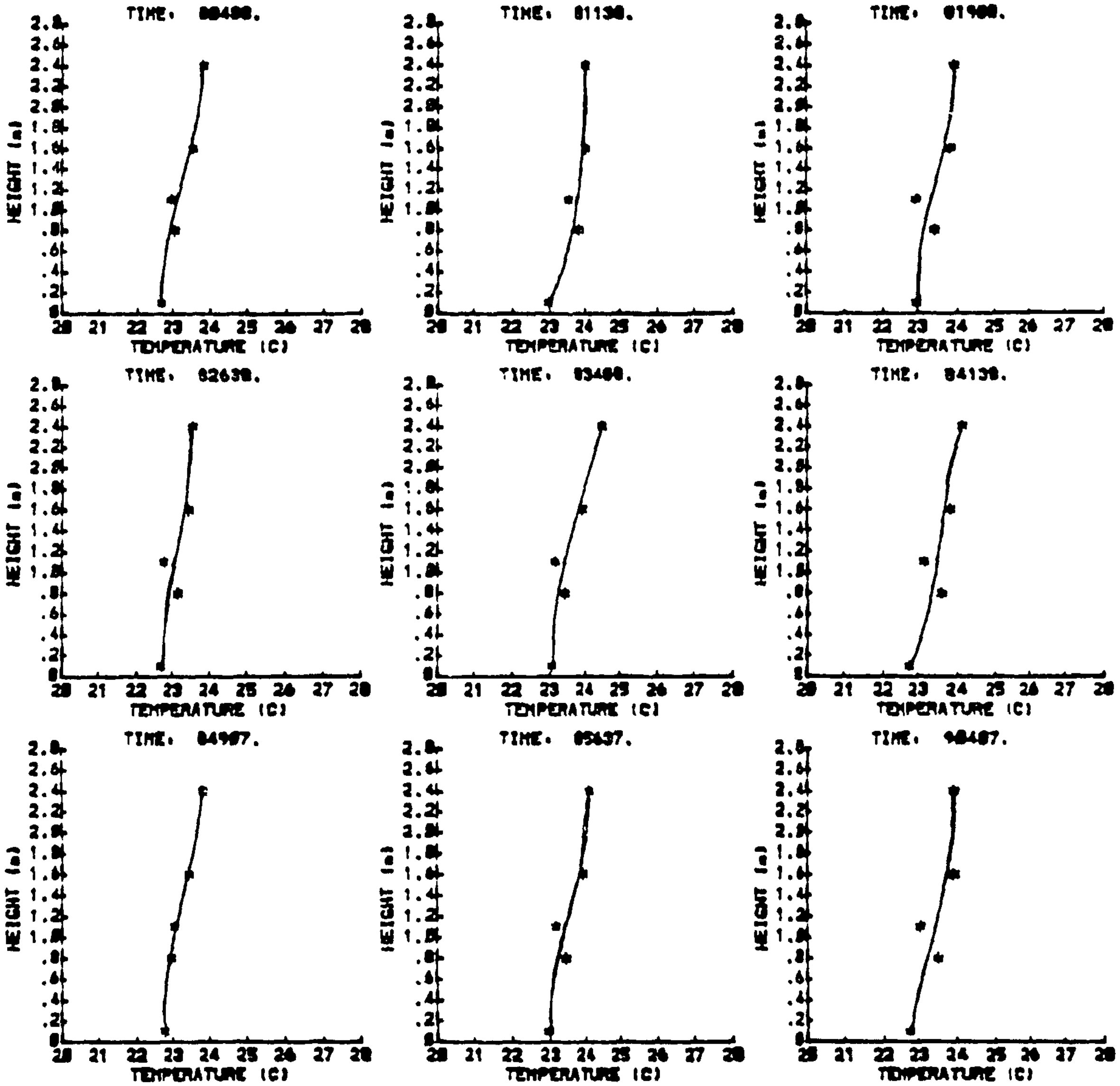


Figure 5.22: Temperature Profiles Corresponding to Period Shown in Figure 5.21, Forced Air

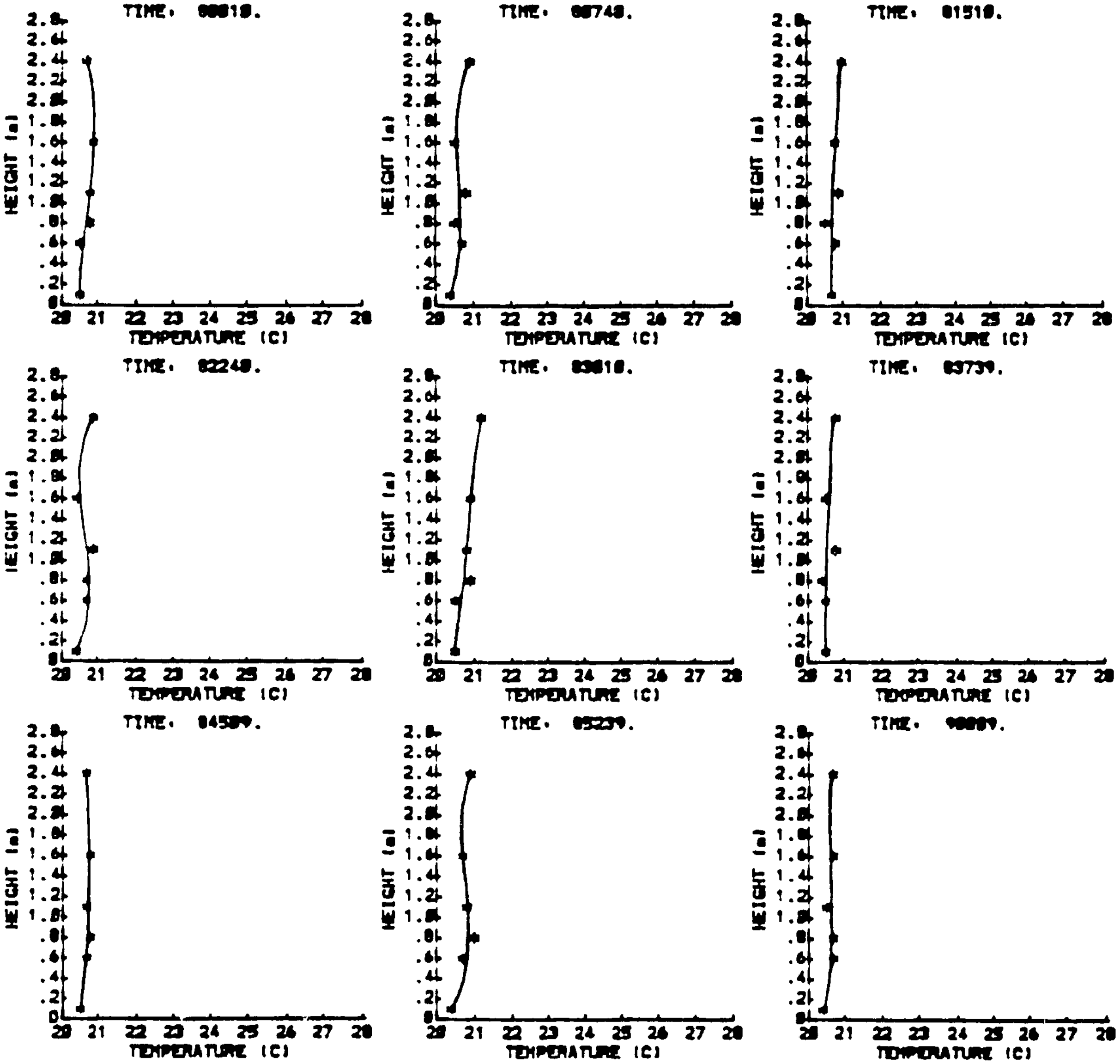


Figure 5.23: Temperature Profiles Corresponding to Period Shown in Figure 5.21, Hydronic

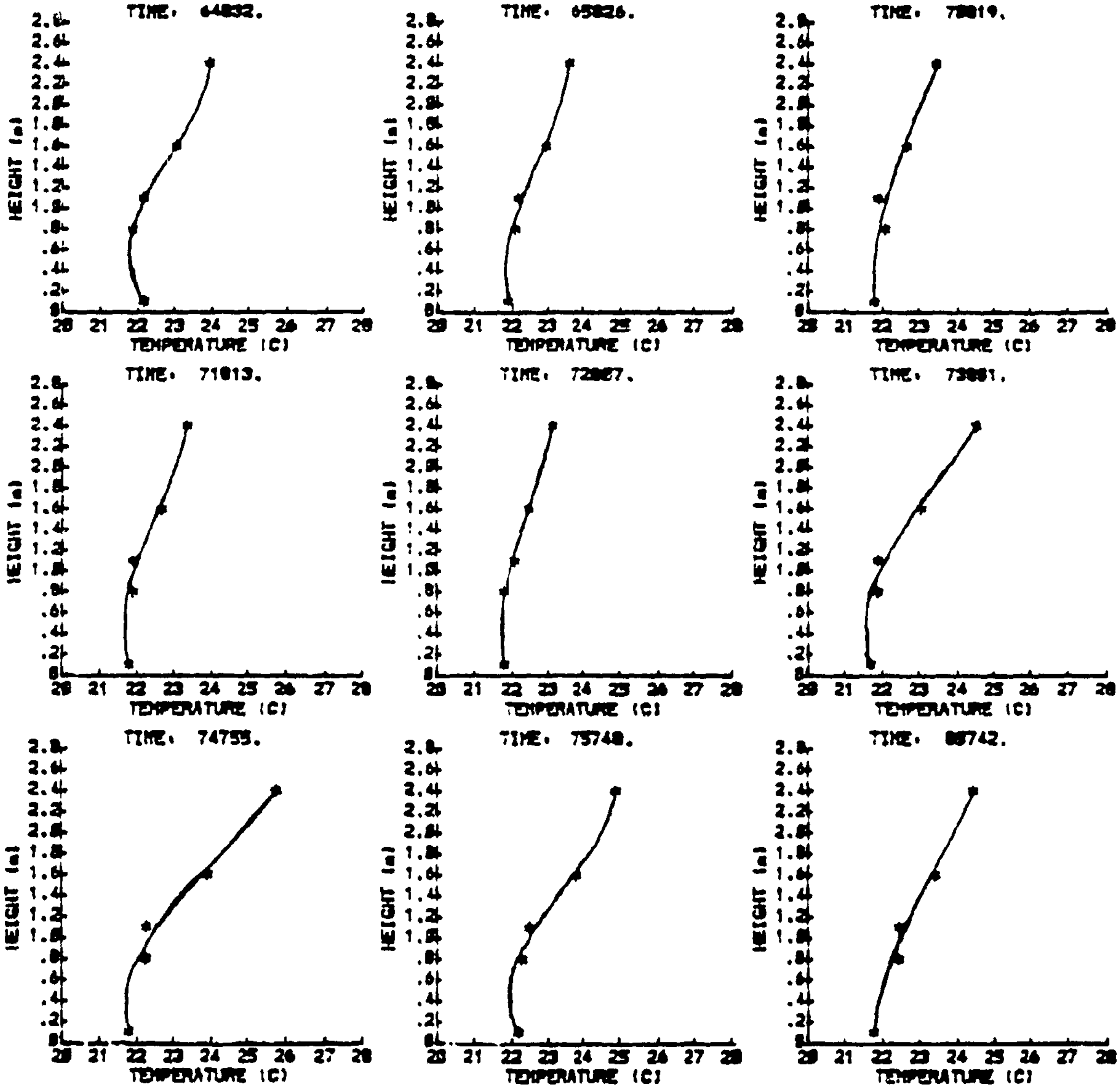


Figure 5.24: Temperature Profiles Corresponding to Period Shown in Figure 5.21, Modular 1st

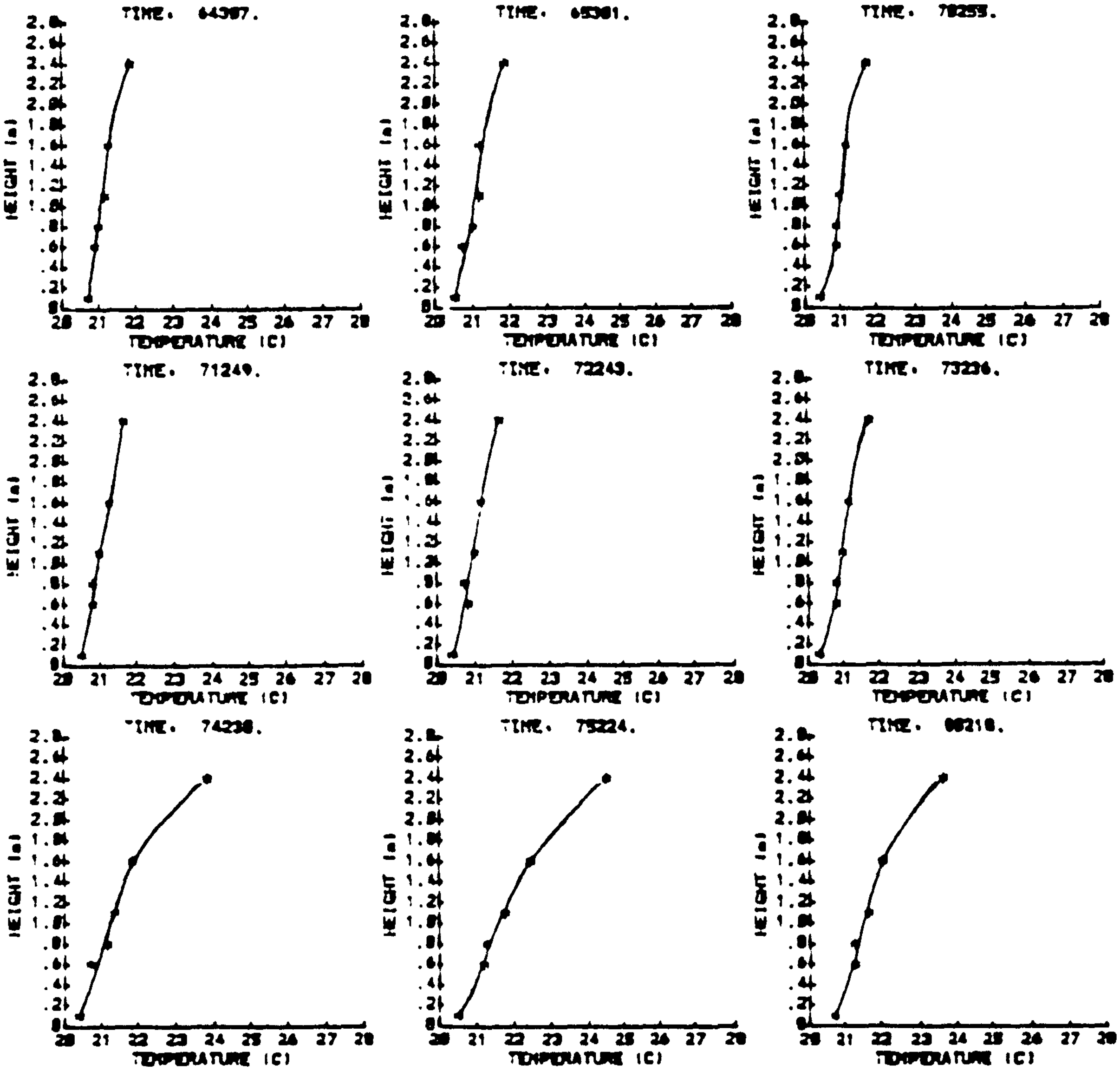


Figure 5.25: Temperature Profiles Corresponding to Period Shown in figure 5.21, Modular 3rd

5.2.4 Effect of Door Opening

During the heating season, the door of the test room was left opened during a few days to observe the effect on the temperature profiles. This was done only for the forced air and the hydronic systems. The profiles obtained are shown on Figures 5.26 and 5.27. The days on which these conditions were maintained had outdoor temperature profiles similar to those of Figures 5.1 and 5.2 as discussed in section 5.2.1 for the case of the coldest outdoor temperature.

Effect on Temperature Profiles

For the forced air system, the temperature profiles of Figures 5.26 and 5.9 can be compared to see the effect of opening the door. As can be seen, the shape of the profiles for both cases are not significantly different. The only notable difference is that location B4 does not have negative gradients when the door is opened, and, therefore, temperature profiles are uniform throughout the room. This is probably due to the change in pressures caused by opening the door. The comfort room pressure was being brought closer to the pressure in the rest of the apartment

For the hydronic system, the temperature profile at locations B2 and B3 for the case when the door is opened (Figure 5.27) are more or less the same as for the case with the door closed (Figure 5.10). The main difference is for location B1 where the increase of temperature caused by the baseboard is shifted upwards. Also, location B4 presents the same high increases above the 2 meter height as for the case at milder temperatures described in the preceding section.

From these two cases, there appears to be no definite or specific modification of the shape of the temperature profiles which could be attributed to door opening. For the forced air system, a more uniform distribution occurs, but the distribution for the hydronic system was already fairly uniform and similar changes cannot be compared.

Effect on Room Temperatures

For the hydronic system, the effect of keeping the door opened has been to maintain the air temperature closer to the 21°C set point. In Table 5.1, the temperature at the center of the room (location B3, 0.8 m height) was at an average of 20.1°C (close to seasonal averages mentioned in chapter 4) while, for the case with the door opened, the average temperature at the same location was 21.5°C. Also, for that location, the range of temperature change was higher for the case when the door was opened (around 1.5°C) compared to the door closed case (less than 1°C).

For the forced air system, the air temperature at the center of the room and its range of variation were not significantly different between the closed or opened door case.

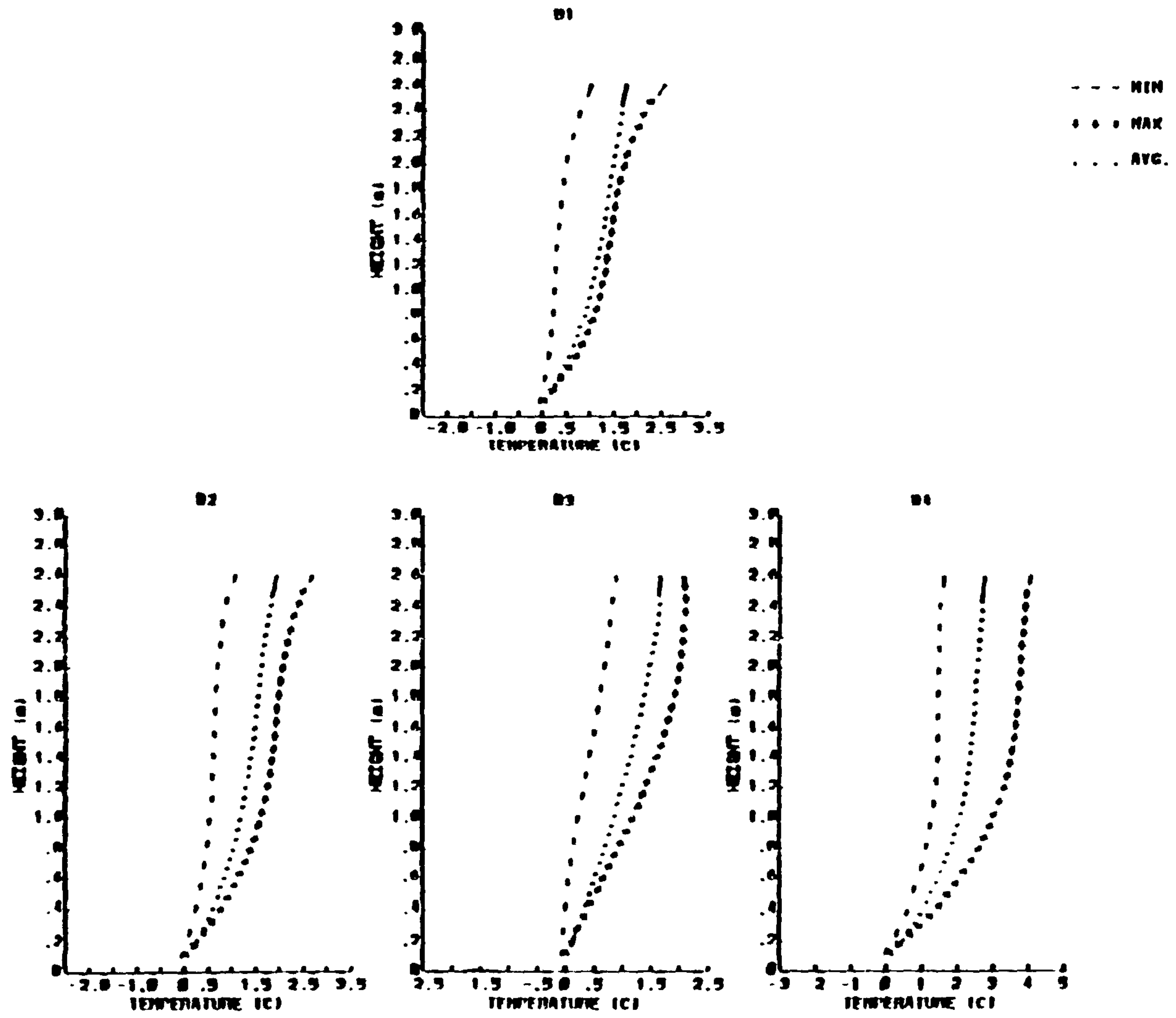


Figure 5.26: Temperature Profiles Obtained for Forced Air With Door Opened

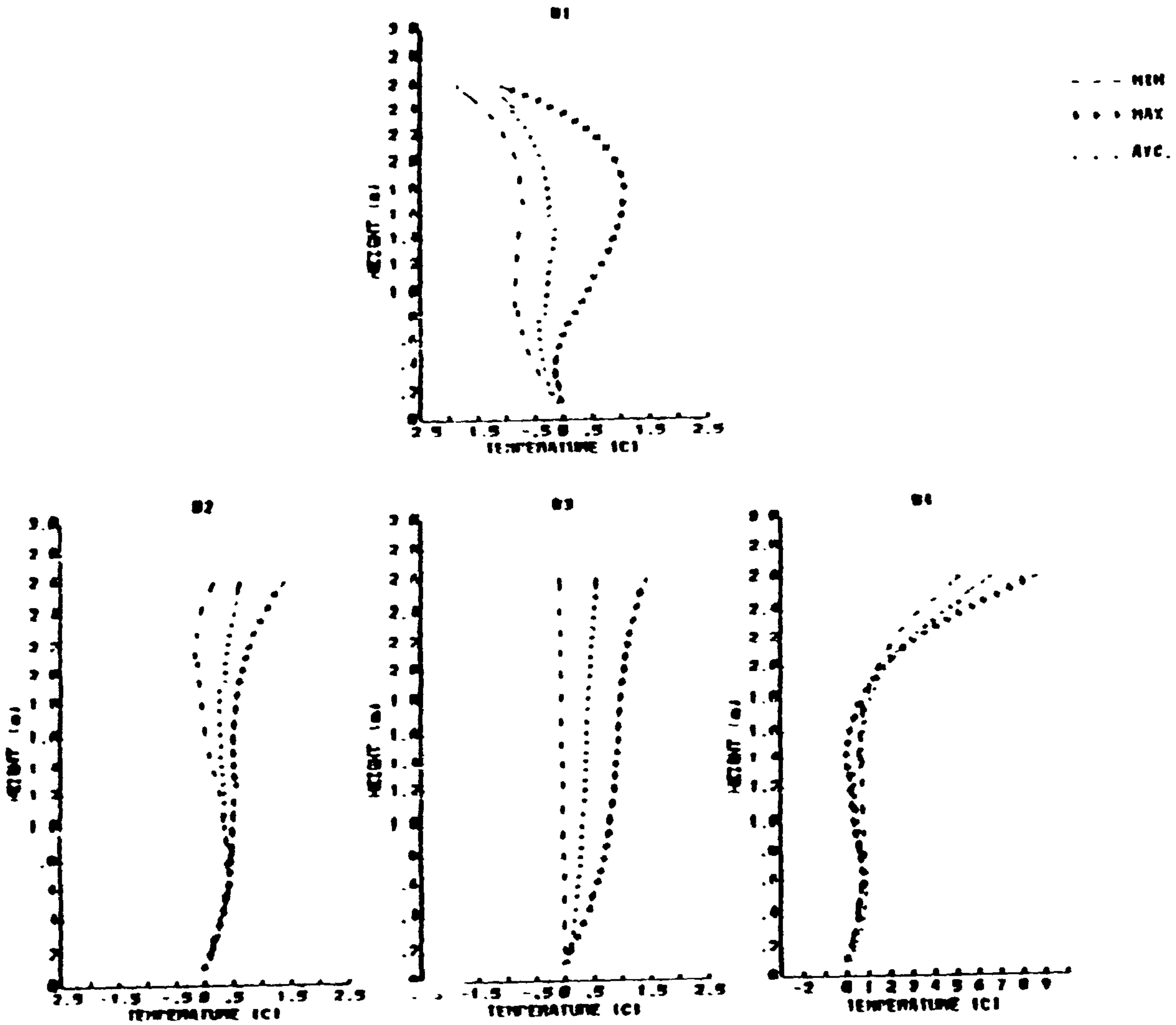


Figure 5.27: Temperature Profiles Obtained for Hydronic System With Door Opened

Summary

Results seem to indicate that there is an effect on temperature profiles caused by opening the door but since the data collected cover only a short period, the analysis cannot satisfactorily confirm the exact effect.

Nevertheless, the results indicate that for the hydronic system, opening the door will balance the conditions in the room by keeping the temperatures closer to the set point but the air temperatures will vary more. Since the forced air system mechanically delivers the air in the room, the effect of opening the door will not be sensed much in the average room air temperature and its variations.

5.3 Seasonal Variations of Gradients

The seasonal variations of outdoor temperatures and the corresponding indoor temperatures in the test rooms were measured for a large number of days with the systems in normal operation conditions. The gradients at various times during the season can be also be calculated.

For a given system, it can be verified if the magnitude of the gradients observed in the environment are the same irrespective of the outdoor temperature. In other words, can the system maintain the environment constant in the rooms throughout the season or is it better or not in achieving that function as the heating load increases?

The outdoor temperature at the site was measured throughout the season. However, due to changes in the thermocouples' locations and some readjustments, the outdoor temperature measurements are not available for every single day a measurement was made.

The outdoor temperature at the site can be approximated using weather data available from the closest weather station (Dorval Airport). Naturally, there is a discrepancy between the site's weather data and those at the meteorological station. It can assumed that the general trend in the values of the two sets of data are corresponding to each other within sufficient limits.

Two sets of weather data were compared for a day in December and it was found that the site's outdoor temperature is on the average warmer than the airport's data and that they changed less rapidly. This is easily explained by the fact that the weather station at the airport is located in an open area whereas the testing site is surrounded by other buildings which shield the wind. Also, the thermocouples for outdoor temperature are located on the other side of the windows, in a zone surrounded by walls. Nevertheless, as an approximation, the outdoor temperatures recorded at the airport are used.

Since the outdoor temperature at the weather station is measured once every hour and the measurements at the site are more frequent, the outdoor temperature from the weather station must be weighted so that they will correspond to the data collected at the site.

To find the outdoor temperature during a test (T_{Otest}), the following outdoor temperature definition was:

$$T_{Otest} = (\Sigma \, nobs(i) * T_w(i))/n \quad i=1,2,3 \,..n \tag{5.4}$$

where: T_{Otest} = Average outdoor temperature at the time of the test (°C)

$T_w(i)$ = Outdoor temperature at hour i read at the meteorological station (°C)

nobs(i) = number of readings during hour i

n = total number of readings during the recording period

i = index for hour i during the day

As defined by equation 5.4, the values of T_{Otest} can be calculated for each test period and provide an estimate of the outdoor temperature at the time of the test. This definition is used later after gradients for each test period, for all the systems, are calculated.

The maximum, average and minimum gradients were calculated using equation 5.1 for each days when were data collected for the four systems. Results are shown in Figures 5.28 to 5.30 for the forced air, the hydronic system, the modular system on the first floor and the modular system on the third floor respectively. The heights between which the gradients were evaluated in the measurements are as described in Table 5.2.

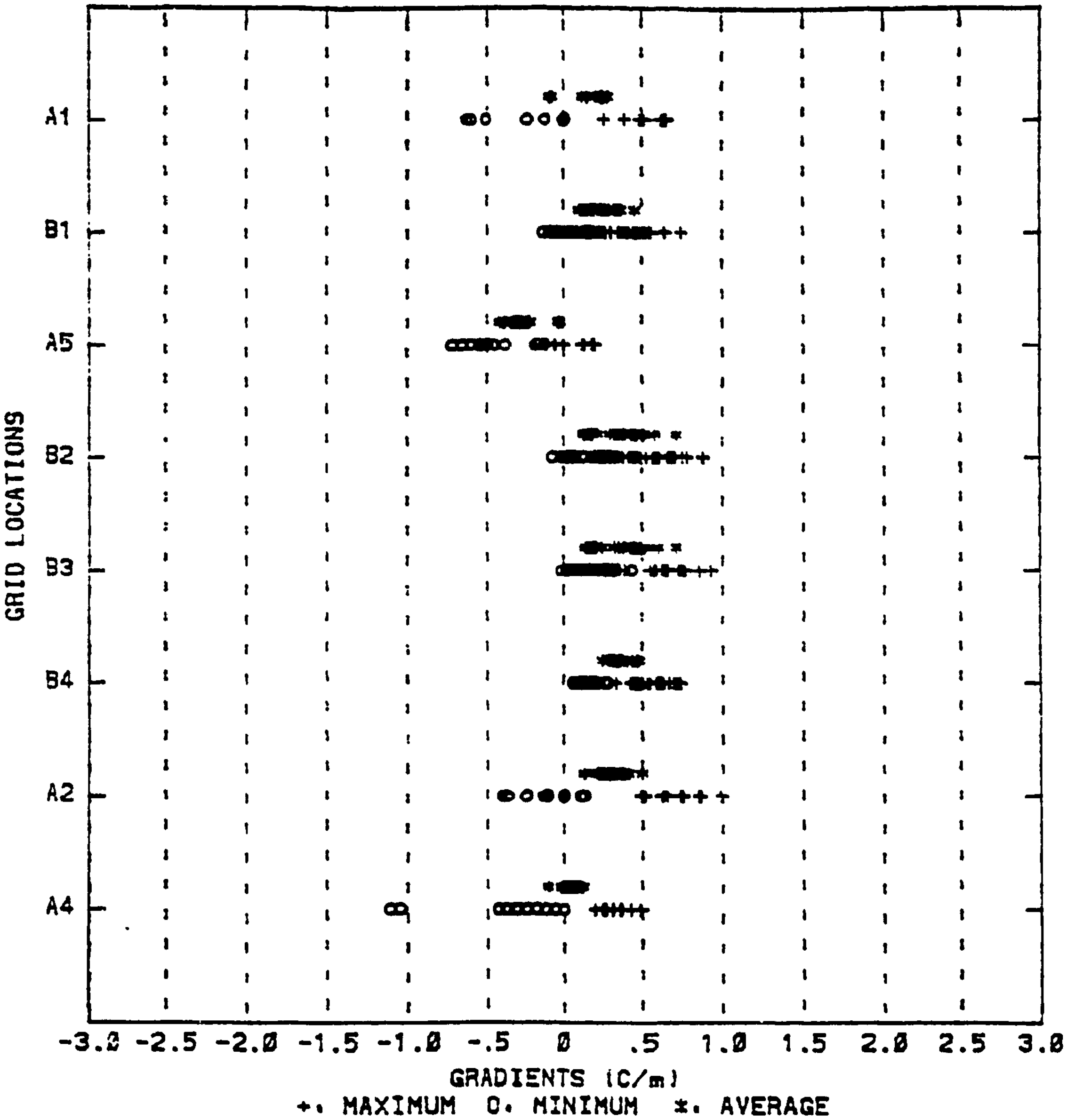


Figure 5.28: Seasonal Distribution of Gradients (linear), Forced Air System

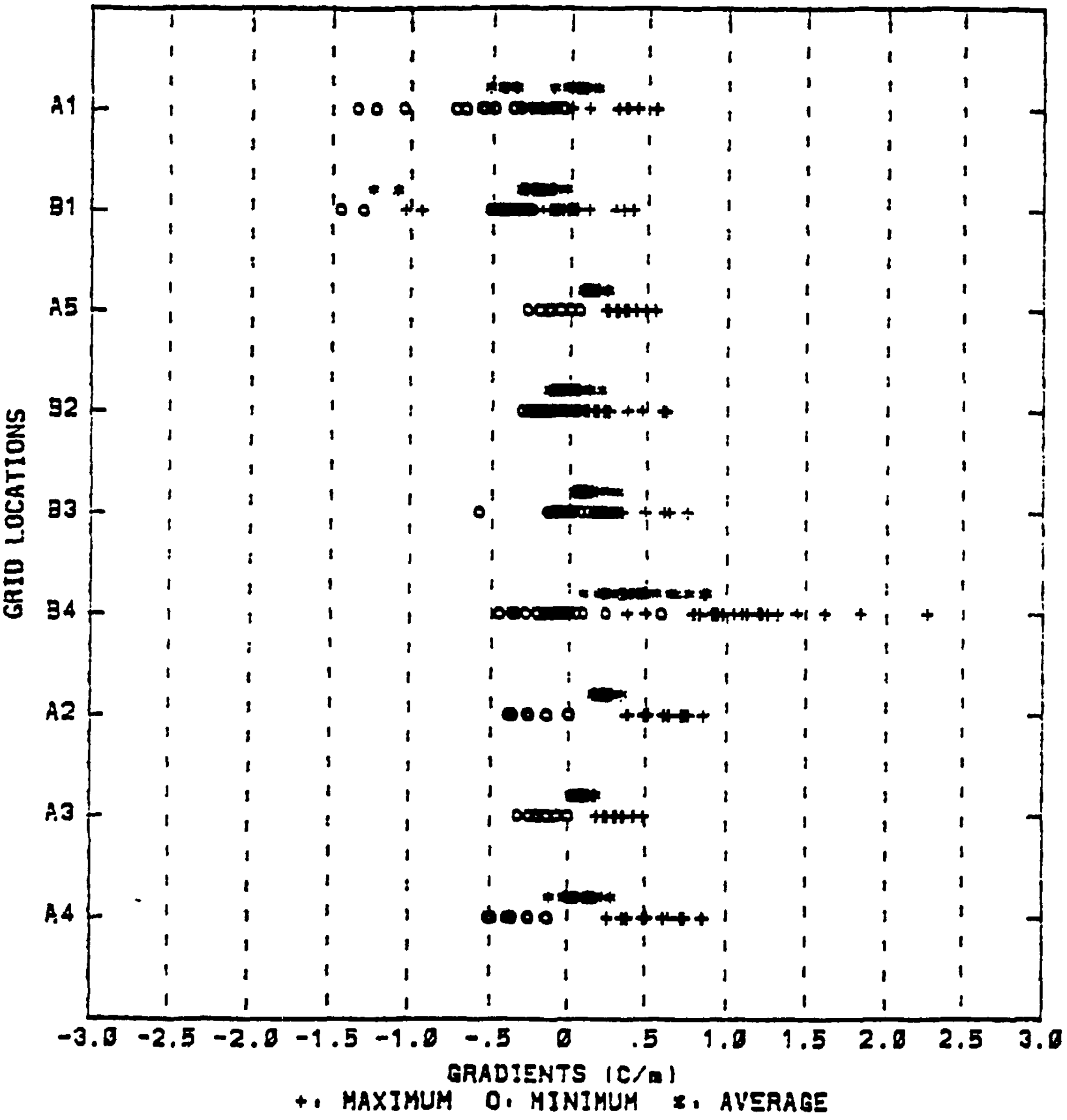


Figure 5.29: Seasonal Distribution of Gradients (linear), Hydronic System

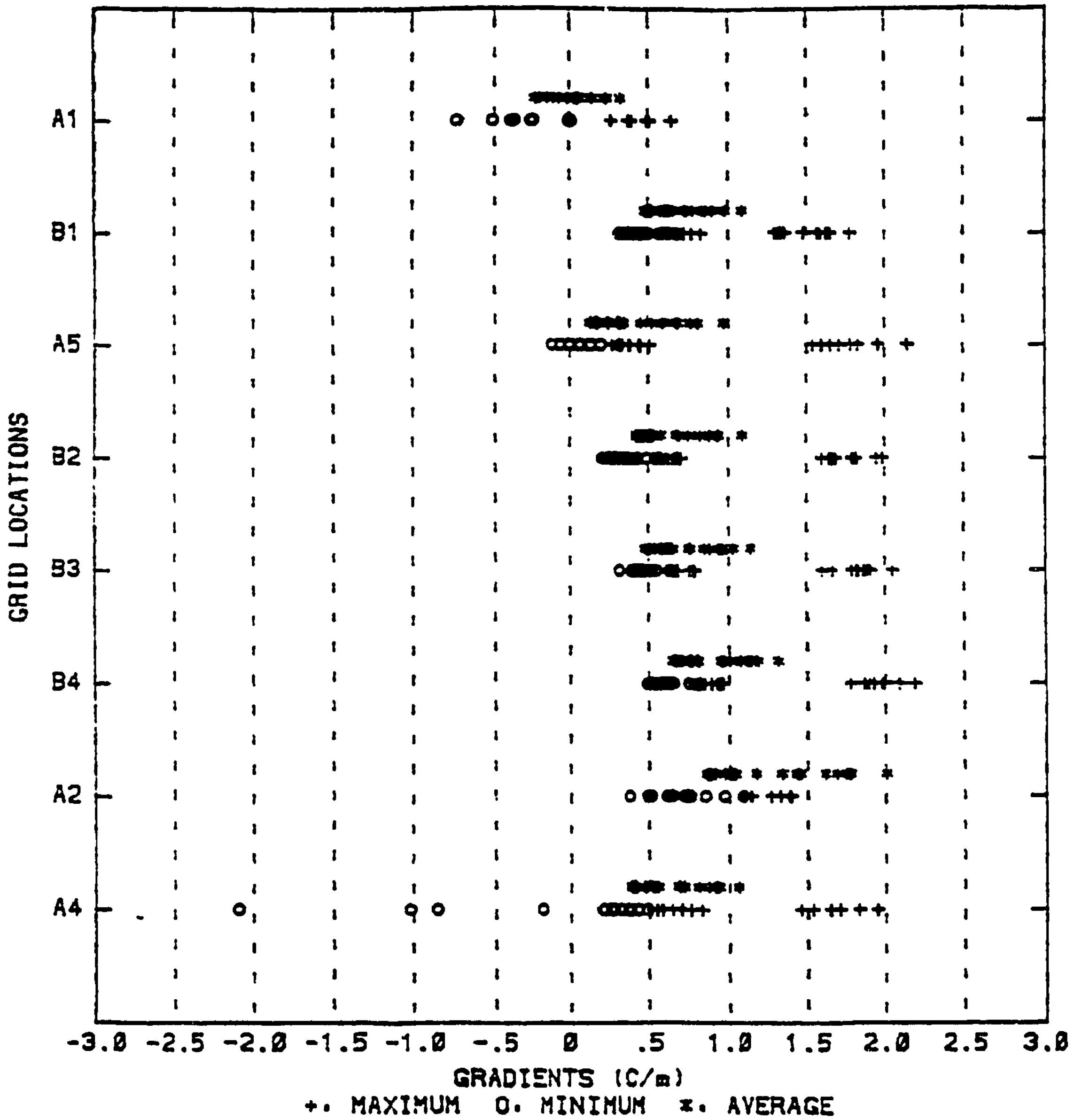


Figure 5.30: Seasonal Distribution of Gradients (linear), Modular 1st Floor

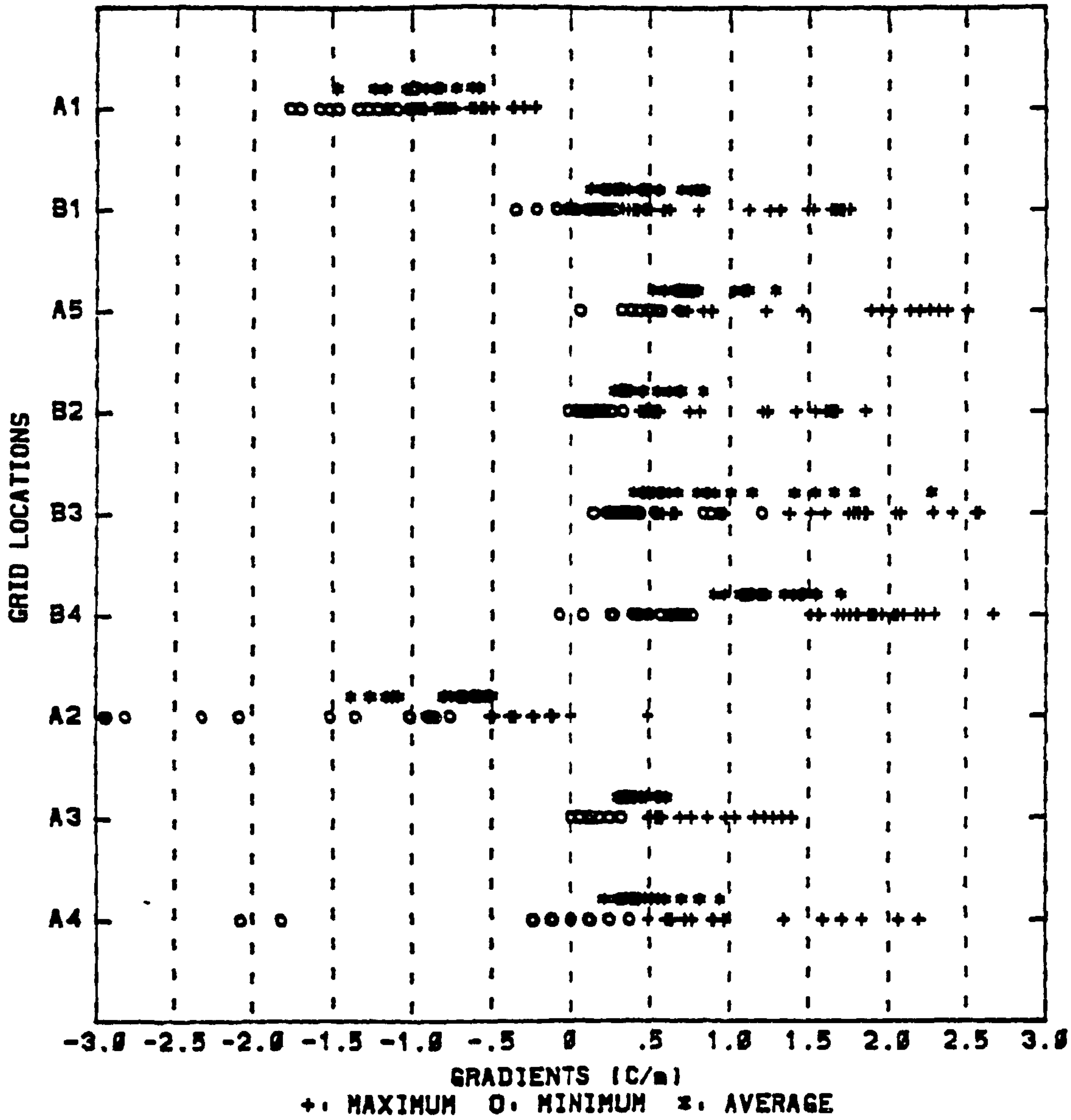


Figure 5.31: Seasonal Distribution of Gradients (linear), Modular 3rd Floor

Forced Air System

For the forced air system (Figure 5.26), it can be seen that the values of the average gradients observed are close to zero for most grid locations. There is a general tendency towards slightly positive values except for locations on the side of the neighbour's wall (A5). In order to describe the results from the graphs the half range value can be defined. The positive half range is the range of values limited by the maximum value and zero and, the negative half range is limited by minimum values and zero.

From the above definition, the positive and negative half range for the forced air system are less than one degree for all locations.

Therefore, the gradients observed for the forced air system throughout the season are small considering the accuracy of the thermocouples which allows at most to detect differences of half a degree. Note that for these results (and it is valid for the rest of the section) the gradients are assumed to be linear and in the case that they are not, the gradients may be greater than described here.

Hydronic System

If we consider the hydronic system (figure 5.28), it can be seen that the values of the average gradients observed are close to zero for most grid locations. The positive and negative half ranges of the gradients are less than one degree for all locations except A1, B1 and B4.

In general, the gradients observed for the hydronic system throughout the season are small except for location B4 where linear gradients higher than 1°C/m have been observed.

Modular Systems

The results for the modular systems on both floor are discussed together in order to compare the results.

For the results for the modular system on the first floor (figure 5.30), it can be seen that the values of the average gradients are concentrated between 0.0°C/m and 1.5°C/m for all grid locations except location A1 (near 0.0°C/m) and location A2 (up to 2.0°C/m). Note that from table 1, temperature measurements for A1 and A2 are available only over a small range which explains these differences.

Results on the third floor were similar to those of the first floor. On the third floor (figure 5.31), it can be seen that the values of the average gradients are also concentrated between 0.0°C/m and 1.5°C/m for all grid locations except locations A1 and A2 (-0.5°C/m to -1.5°C/m).

The positive half range of the gradients is close to 2.0°C/m for all locations except at location A1 where it is less than 1.0°C/m. The negative half range of the gradients is non existent for all locations except at locations A1 (-1.0°C/m) and A4 (-2.0°C/m).

On the third floor, the positive half range of the gradients is slightlty higher (close to 2.5°C/m) for all locations except at locations A1 and A2 where it is non existent. The negative half range of the gradients non existent for all locations except at locations A1, A2 and A4.

From the above, it can be seen that both modular systems produce significantly higher gradients than the other two systems. Also, the modular systems have a greater difference between their maxima and minima or in the range between which the gradients are found.

The distribution of gradients for the modular systems are similar for both floors except for location A2 which is not evaluated between the same heights on the two floors (see table 5.2).

The difference observed for location A1 between first and third floor can be explained by the different location of the systems with respect to location A1 on these floors. On the third floor, the system was located facing location A1 since it had to be installed further away from the window due to space limitations imposed by the presence of the hydronic baseboard. Therefore, the negative gradients indicate the presence of a warm zone in front of the system (at ~0.8m height) with lower temperatures at the higher level (~2.4m height).

5.4 Relation Between Gradients and Outdoor Temperatures

The gradients in Figures 5.28 to 5.31 show the general distribution of the gradients during the heating season. They can be used to determine the overall limits between which the gradients of a particular system are expected to be found. Now, the same data are used along with T_{Otest} defined earlier to establish if the values of gradients can be related to the outdoor temperature. Figure 5.32 shows the average gradients at location B1 as a function of the outdoor temperature (T_{Otest}) for all four systems. Location B1 was selected since it is closer to the window and the systems.

For the forced air and the modular systems, the distribution of the points reveal a straight line with a negative slope. The magnitude of the slope is higher for the modular systems (close to 1°C/m per °C of T_{Otest}) than for the forced air system (0.5°C/m per °C of T_{Otest})

The data for the hydronic system do not follow any specific type of relationship. The gradients are clustered around 0°C/m with only two values lying outside of the range for high outdoor temperatures.

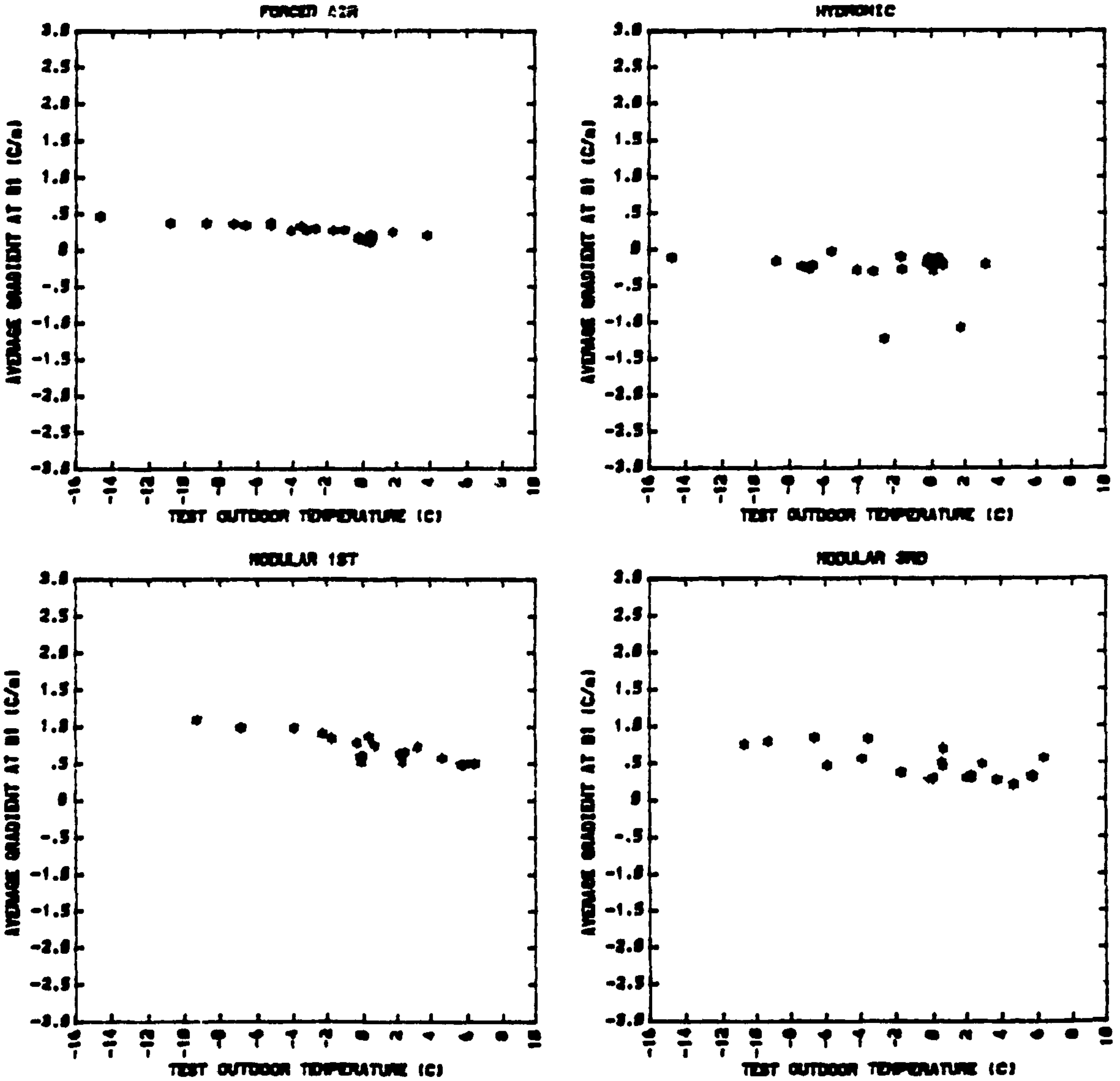


Figure 5.32: Average Gradients at Location B1 as a Function of the T_{otest}, All Systems

The same type of graphs were also plotted for the minimum and maximum values of the gradients as a function of T_{Otest} and are shown in Figures 5.33 and 5.34. The graph of the minimum values shows the same type of pattern as the average values. The graph for the maximum values is similar for the forced air system but the values for the other systems are not clearly distributed along a straight line.

From the above observations, it can be seen that except for the hydronic system, the average and minimum values of the gradients observed follow outdoor temperatures in a linear relationship for location B1. The slope of this relationship is negative which indicates that as the outdoor temperature decreases, the average and minimum values of the gradients tend to increase. The magnitude of the slope indicates the sensitivity of this dependence. For the two modular systems, the slope is greater than for the forced air system, indicating a greater sensitivity.

For the hydronic system, the relationship for average and minimum gradients at location B1 is not as well defined. The reason is that the baseboard faces directly that location and is the major factor acting at that location. Therefore, its effect is dominant. It is also true that the supply of the forced air is located in front of grid location B1 but,
since there is a relationship between gradients there and outdoor temperature, the effect of the supply is not a dominant factor- the heat is not directly distributed there.

The same graphs were repeated for location B3 at the center of the room. The graphs of the maximum, minimum and average gradients at location B3 as a function of T_{Otest} are shown on Figures 5.35 to 5.37.

For the forced air and hydronic system, the graphs indicate that the maximum, minimum and average gradients observed at location B3 for the season are more or less constant. From the graphs, the slope of the line appears to be slightly higher for the forced air system. Taking into account the accuracy of the measurements and the fact that the systems are not located on the same floors, this difference is not significant.

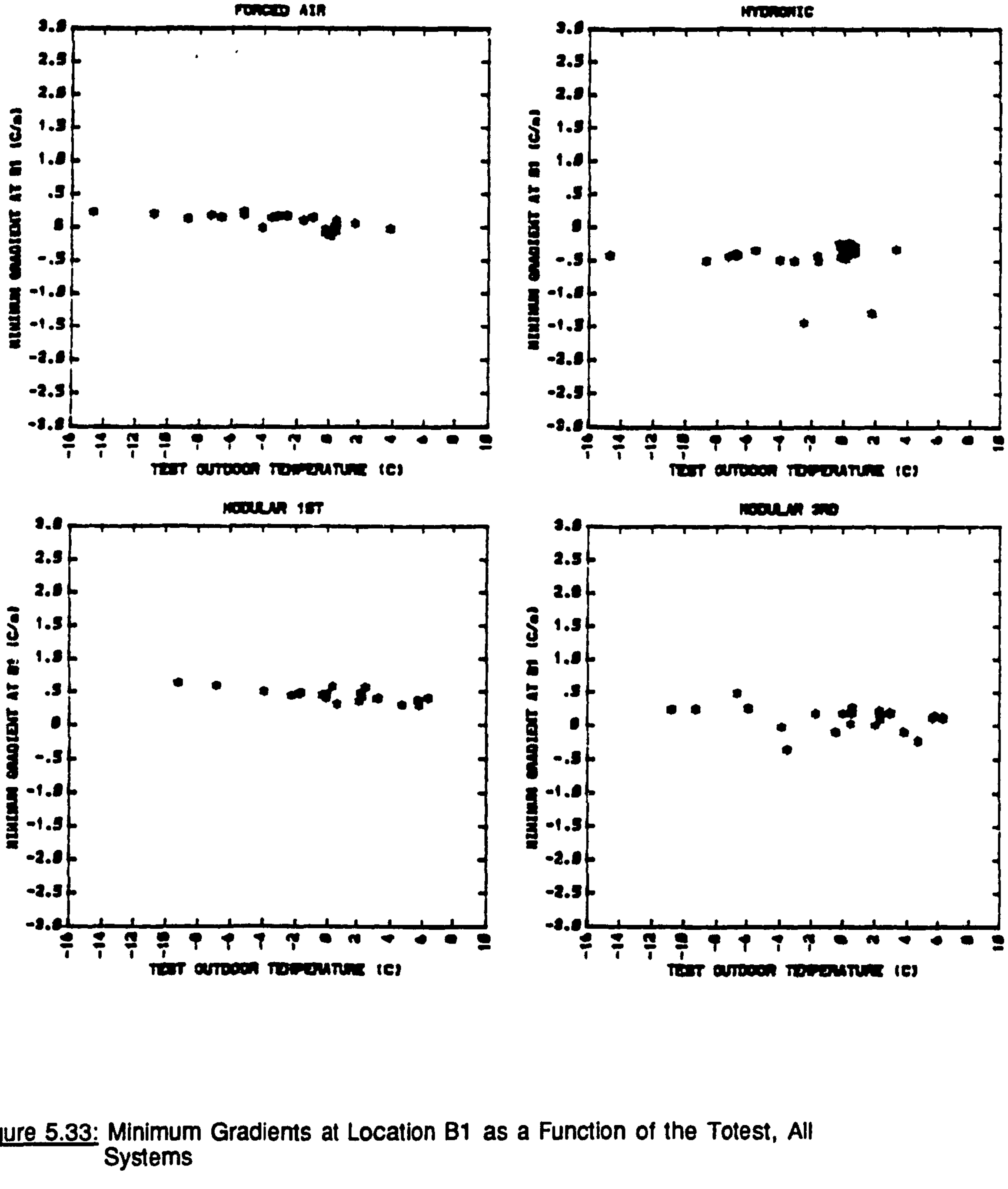


Figure 5.33: Minimum Gradients at Location B1 as a Function of the Totest, All Systems

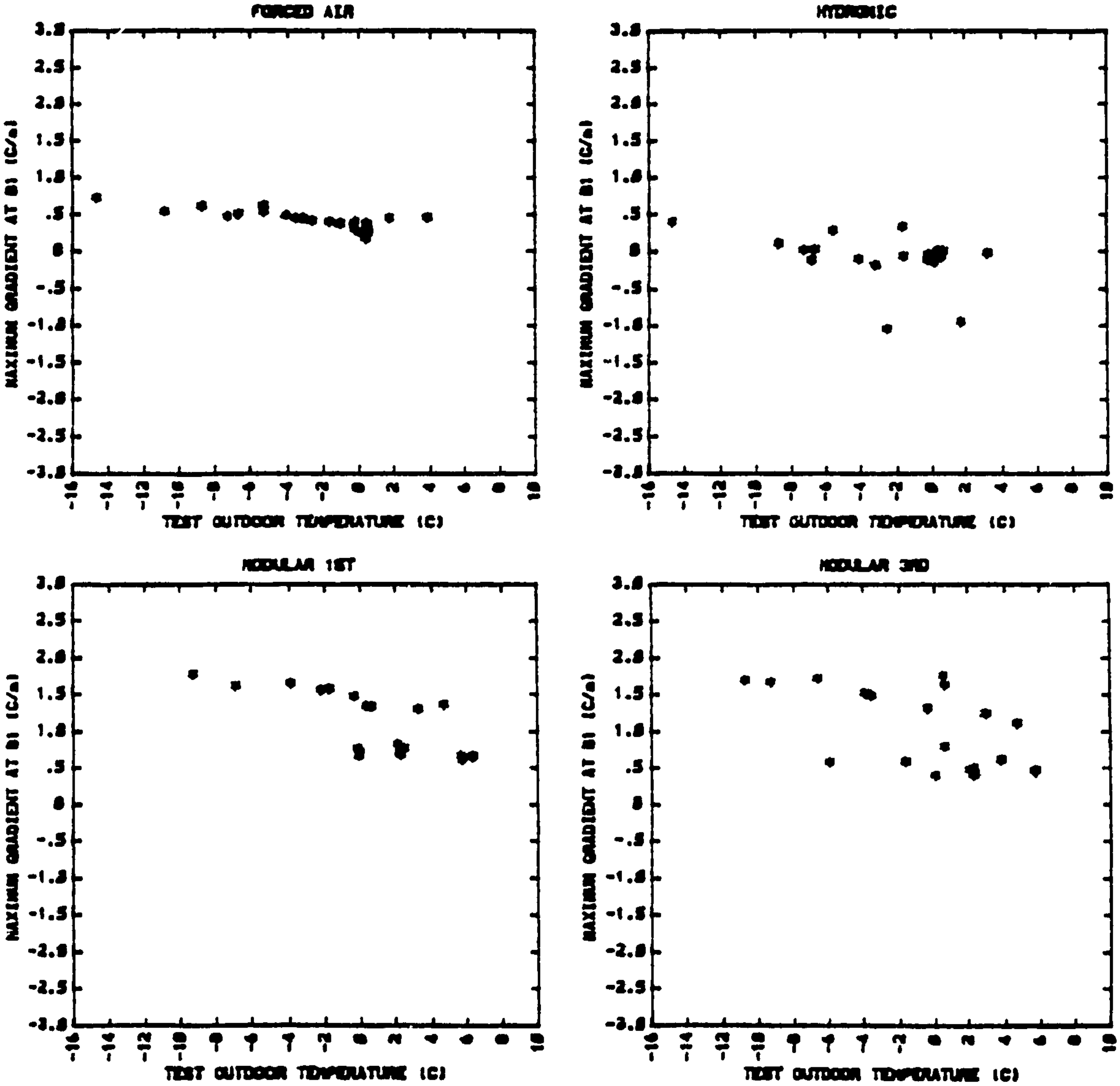


Figure 5.34: Maximum Gradients at Location B1 as a Function of the T_{otest}, All Systems

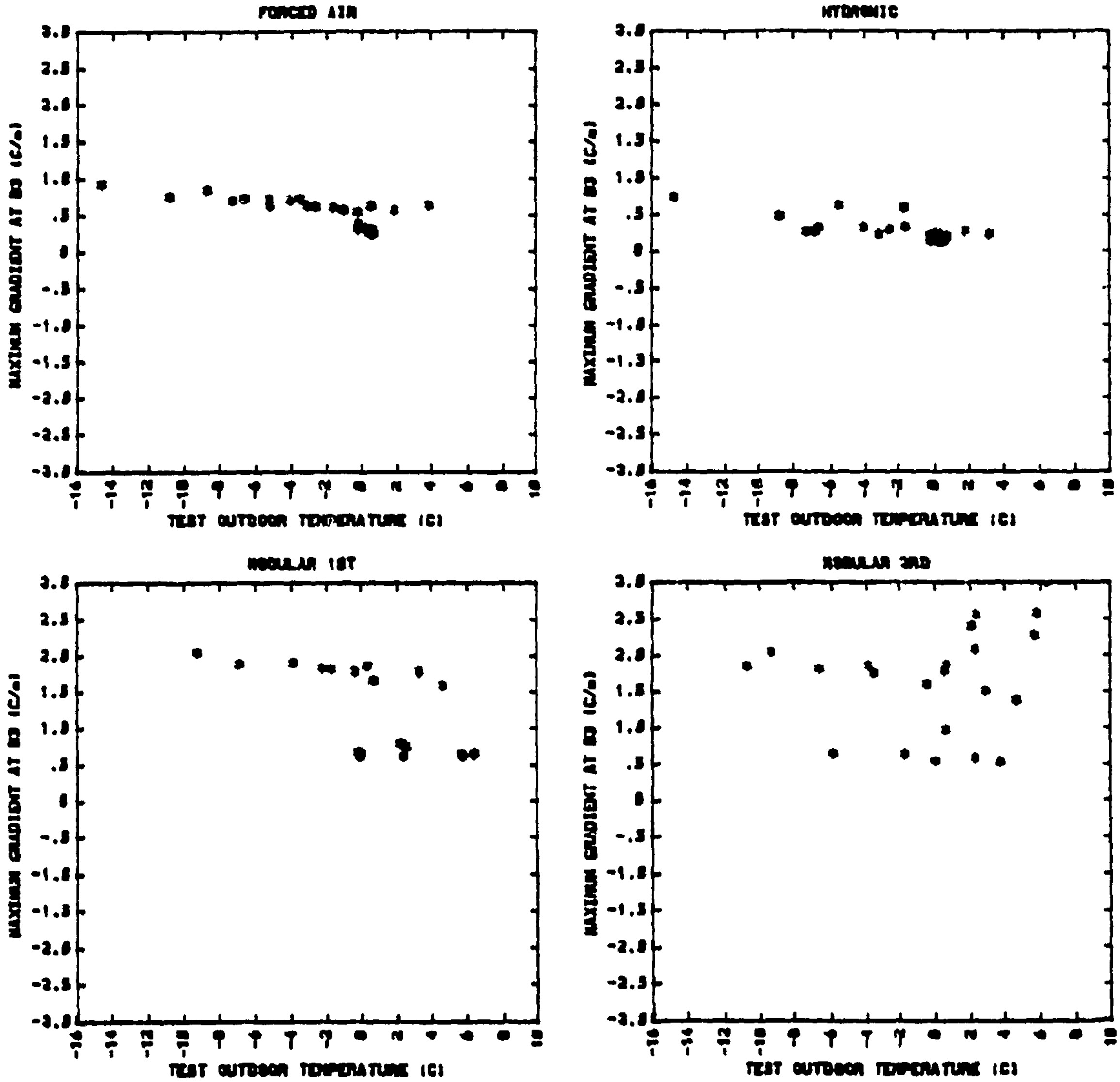


Figure 5.35: Maximum Gradients at Location B3 as a Function of the T_{otest}, All Systems

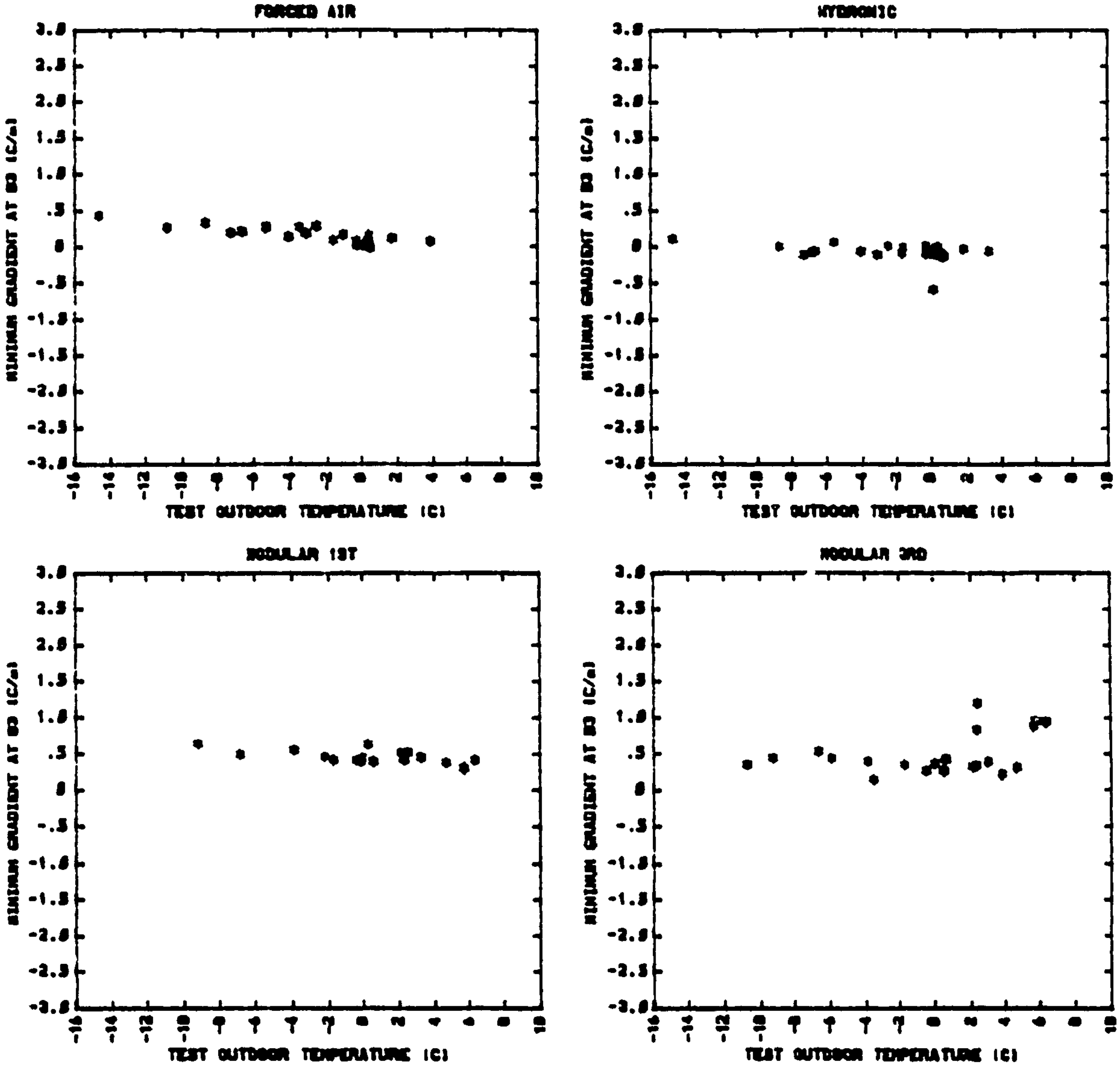


Figure 5.36: Minimum Gradients at Location B3 as a Function of the T_{otest}, All Systems

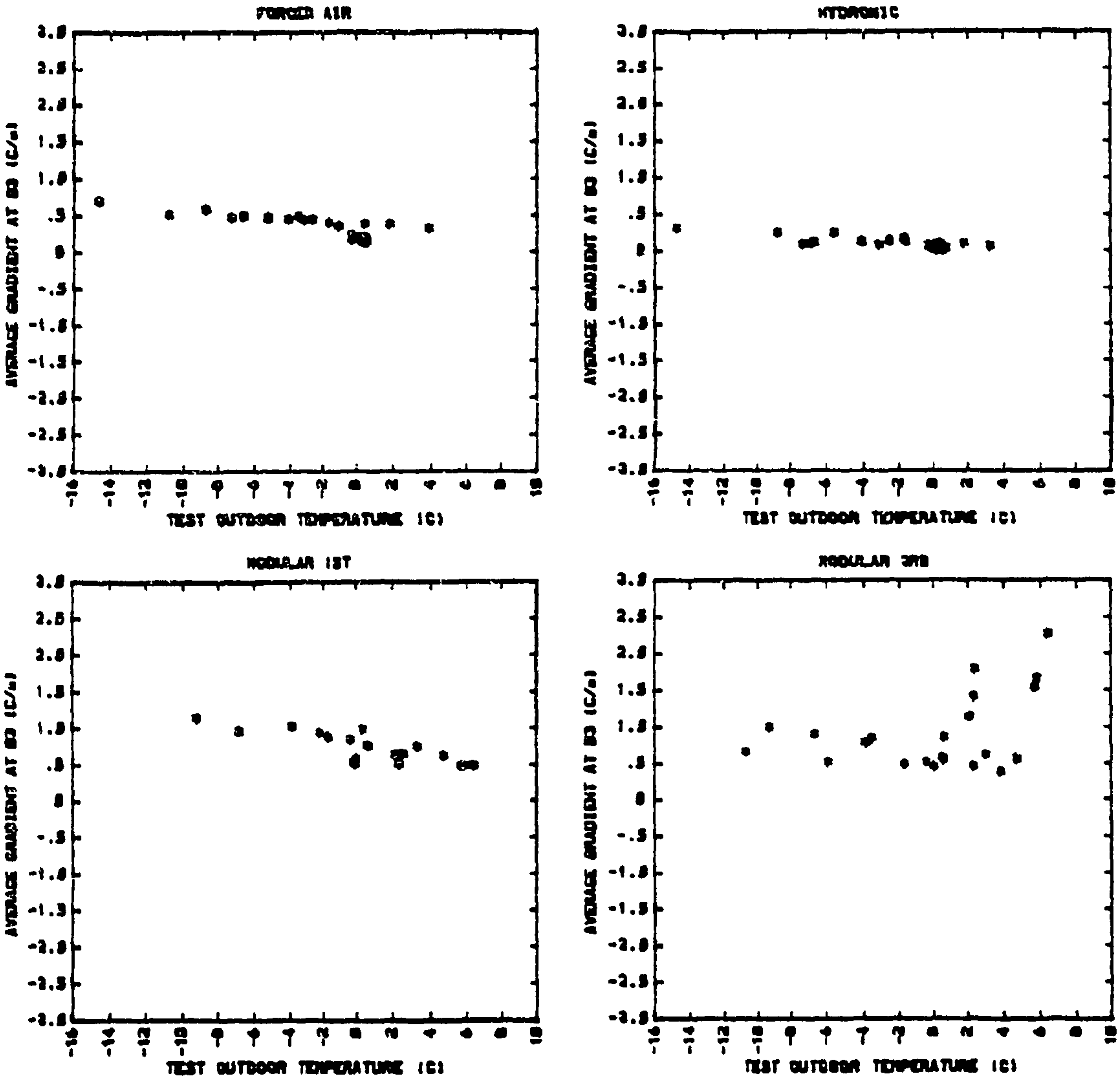


Figure 5.37: Average Gradients at Location B3 as a Function of the T_{otest}, All Systems

For the modular systems, the relationship between gradients at location B3 and test outdoor temperature is not as well defined. For the modular system on the first floor, the relationship is clearly defined for the average and minimum gradients as was the case for location B1. For both modular systems however, the relationship is not observed outdoor temperature above -2°C. Therefore, the gradients at temperatures higher than -2°C cannot be entered in the same category as the gradients for temperatures higher than -2°C.

The linear type of relationship observed for location B1 for the modular systems is found for the range of outdoor temperatures above -2°C, This relationship is again linear with negative slope. From the graphs of the average gradients, the slope for the first floor is situated between 0.5°C/m and 1.5°C/m for the first floor while values for the third floor are between 0.5°C/m and 1.0°C/m. Although this difference is small, the average gradients on the third floor could be said to be less sensitive to outdoor temperatures than on the first floor.

From the above discussion, it can be concluded that the gradients observed at different locations in the test rooms either vary linearly with outdoor temperatures or remain fairly constant (based on linear interpolations). The variations are less important for the forced air and hydronic systems than for the modular systems. The relationship is not as well marked for temperatures higher than -2°C for the modular systems when data were collected at the beginning of the season with control settings for the modular system still not yet fully stable.

5.5 Summary

In this section, the observations made in the previous sections are summarize. The questions which were asked in the first section are reviewed Results are listed in a tabular form for easy comparison between the systems and to discuss the results of this section in relation to the previous section's results.

The first question asked at the beginning of the chapter was: What is the shape of the temperature profiles produced by the different heating systems? This question was treated in section 5.2.1 for the coldest outdoor temperature observed and in section 5.2.2 for milder temperatures.

For the forced air system it was found that the temperature profiles at the coldest and milder temperatures were similar in shape with gradients slightly less severe at the milder temperatures. Also, the profiles were similar at different locations in the room indicating that the distribution of heat in the room was uniform at all ranges of outdoor temperature conditions.

For the hydronic system, it was found that to qualify differences between the temperature profiles at milder and colder outdoor temperatures, the observations must be divided in two: profiles at the location facing the heater and the profiles in the rest of the room. The profiles in the center of the room remained the same independently of the outdoor conditions at which the system functioned.

On the other hand, when colder and milder conditions temperatures profiles were compared for the hydronic system, the height of discharge of warm air from the baseboard was lower at milder temperatures. Not only was the height of discharge lowered, but also gradients in the region close to the floor were much higher because of the concentration of heat. This indicates that the way the baseboard distributes the heat in the room is influenced by the temperature at which it operates.

For the modular systems, it was found that the temperature profiles were similar for the two floors except for two locations: facing the system and to the window for the third floor (influenced by some heat coming from the baseboard of the hydronic system). Even with this disturbance, the profiles created in the rest of the room were not affected.

The second question mentioned at the beginning of the chapter on whether the magnitude of the gradients were related to the temperature of the heaters, was dealt within section 5.2.3. It was found that for the forced air and the hydronic systems, the magnitude of the gradients could not be directly related to the heater's temperature.

For the modular systems on the other hand, it was found that the gradients can be related to the heater temperature for the higher levels. This was demonstrated by an example showing profiles in time. The relation between heater temperature and gradients in the room enabled to determine that the high temperatures at which the modular systems were operating, and the range over which these vary, are partly responsible for the high gradients observed.

The third question mentioned at the beginning of the chapter was as to whether opening the door of the enclosure would modify the temperature profiles in the rooms. From data available for the forced air and the hydronic system, no major modification in the temperature profiles were found which could be attributed to opening the door based on the small number of cases studied More data are needed than what was available to study correctly this modification

On the other hand, opening the door has had the effect of bringing the temperatures in the room closer to the set point than when the door was closed for the hydronic system This indicates that this system works at temperatures closer to the set point if the door is opened Another effect, which is negative, is that the temperatures will fluctuate more For the forced air system, these two observations do not hold and the temperatures maintained in the room as well as their variations are relatively unchanged by opening the door

The last question at the beginning of the chapter was whether the magnitude of the gradients varies as the outdoor temperature changes

For all systems, it was found that up to a certain extent, some relationship existed between outdoor temperature and average gradients in the room The relation was marked for the modular system for which outdoor temperature decreases corresponded to gradients increases. Also, the relationship is less reliable for outdoor temperatures above 0°C for the modular systems. The relationship however does not hold for the hydronic system at the location close to the window since that location is strongly influenced by the baseboard.

The slope of the increase in average gradients with decreasing outdoor temperatures indicates how sensitive the changes were for a particular system. It was found that the modular systems are more sensitive to variations in outdoor temperatures. Needless to say, the relation singles out the outdoor temperature as a factor for gradients' magnitude. However, the cycling of the systems being increased as the outdoor temperature decreases which increases gradients.

The variations in the magnitude of the gradients was observed based on linear interpolation only. Differences were small even though a relationship could be found with outdoor temperature. In this sense, the variations observed and the relationship established must be taken with care since the linear interpolation creates errors and this even more for the modular systems. The range of variations being small, it is difficult to obtain a high resolution when the results are analyzed. This is further worsened if considering the accuracy of the thermocouples.

Taking the magnitude of the gradients as a ranking factor for comparison indicates that the forced air system performed the best. If it is accepted that only the location closer to the heater be with high gradients, then the hydronic system is second. The modular systems will rank last because of the gradients they produce are twice as high than for the other two systems.

In terms of the uniformity of the temperature profiles, all systems, except the hydronic system, produced fairly uniform temperatures throughout the room. The hydronic system created great difference between the location close to the heater and the rest of the room.

In order to simplify comparison, Table 5.10 lists some criteria that can be used to compare the systems' performance.

Table 5.10: Comparison of Temperature Profiles for the Different systems based on Different Criteria.

CRITERIA	Forced Air	Hydronic	Modular
Temperature Profiles	Figure 5.13	Figure 5.14	fig. 5.15-5.16
Difference between mild and cold outdoor temperatures profiles	smaller gradients and same profiles	different at heater location, smaller gradients elsewhere	no significant differences
Range of highest temperature change	0.1 m to 1.0 m	above 2.0 m (0.1 to 0.6 m front of heater)	1.2 m to ceiling
Relation with outdoor temperature	yes (small)	yes (small) except close to heater	yes (larger)
Maximum Average gradient (linear)	0.8°C/m	0.9°C/m	2 0°C/m

The following remarks concerning the accuracy of the results and the effect of comparing systems for different locations and different floors can be made

-Thermocouples and their locations were the same on a given floor When the temperature profiles were estimated, if errors occurred for one system, they occurred for the other system on that floor also

-Even though the temperature profiles for the different systems were not all evaluated for the same days, the days selected were chosen so that differences in outdoor conditions were as small as possible.

-Results for the modular systems indicated that the difference between the profiles for the first and third floors were similar , and therefore, comparison between the forced air and the hydronic system should also be valid.

-The use of first order interpolation is accurate for the case when gradients actually follow a linear relationship. For the case that they do not, the linear relationship only provides an average. This is why negative gradients (such as was found for the hydronic system at the location facing the baseboard) obtained with linear approximation do not necessarily exclude the possibility of positive gradients at a certain location.

In this chapter, the results of the preceding chapter have been observed here from the temperature profile point of view:

-The forced air system was found to produce even room temperature distributions and regular cycling behaviour and also produced similar gradients for different locations in the room and a constant behaviour during the season.

-The hydronic was observed to create two zones (location near heater and rest of the room) with cycle operations dependant on outdoor conditions. Similarly, two zones of gradients were observed and the temperature profile at the front of the baseboard changed when outdoor conditions were more or less severe.

-The modular systems were found to produce highly stratified plane temperature distributions and had sharply defined cycles. It was found that gradients were high and the sharp cycles combined with the high heat output capacity created profiles which also varied sharply in time.

-The higher gradients found when outdoor temperatures were lower are due to an ineffective compensation of the cold downdraft at the window by the modular system High surface temperatures on the wall above the heater indicated that most of the heat delivered by the system went along the wall, directly to the ceiling. At warmer outdoor temperatures, the downdraft did not contribute much to increasing gradients compared to the more important influence observed at lower outdoor temperatures Therefore, the combined effect of the system distribution along the wall and the downdraft increased temperature gradients.

Up top now, with this chapter and the previous one, most of the information readily available on temperature measurements was used. So far, the differences between the operation of the systems have been noted based on specific cases or for long periods of measurements The next section will discuss other parameters which were also measured before the results of this section are finally used for thermal comfort assessment of the systems

CHAPTER 6

6. RESULTS ON AIR VELOCITY, RADIANT TEMPERATURE AND RELATIVE HUMIDITY

Air velocity, mean radiant temperature, and relative humidity measurements complete the list of the physical parameters necessary to evaluate comfort for comparing the three heating systems. The analysis will focus on the range in which these parameters observed in some typical cases. More attention will be paid to air velocity and mean radiant temperature since these parameters are more important than relative humidity for comfort.

Air velocity results were selected to show typical variations in time for different locations in the rooms and to establish the average and the range between which they occur for the different systems. Measurements locations included grid points in the occupied space of the rooms.

Mean radiant temperature measurements from globe thermometers were selected to evaluate differences between air and mean radiant temperatures in the rooms. Mainly, the concern was to find if the low window temperatures were well compensated by the heaters, and if other surfaces in the enclosure would be involved in creating radiant discomfort.

Relative humidity measurements show the differences observed between floors and, since comfort is not very sensitive to this parameter, the discussion is limited to average values.

6.1 Air Velocities

The magnitude of air velocities for the different systems will be discussed for different locations in the rooms and for different days. Before results are presented for the systems, the conditions of the tests and the instrumentation used are described.

Test Procedure

Air velocity measurements require sophisticated instruments and special considerations for two main reasons. From results of other studies, the air velocities in the rooms were expected to be low (between 0 and 0.25m/s) and stochastic. Therefore the data collection had to be performed using anemometers working in that low range and using an averaging method

Two types of omnidirectional transducers were used: the climate analyzer anemometer probe and four individual probes connected to the acquisition system (see chapter 3). The location of the devices could be varied to obtain measurements at different locations and heights as long as they were away from a major heat source. Physical restrictions and calibration integrity limited use of the probes to temperatures near 21°C.

The climate analyzer's anemometer was used extensively in the first year of the project to provide measurements of air velocities on the third floor. It recorded over 24 hour periods using values averaged over 5 seconds with a ±0 05m/s accuracy

The individual probes located on the first floor had similar accuracy (±0 03m/s) but measurements had to be averaged by the DAS The DAS was programmed to record velocities in groups of 5 measurements made at 10 seconds intervals every 15 minutes (or more frequently in some cases)

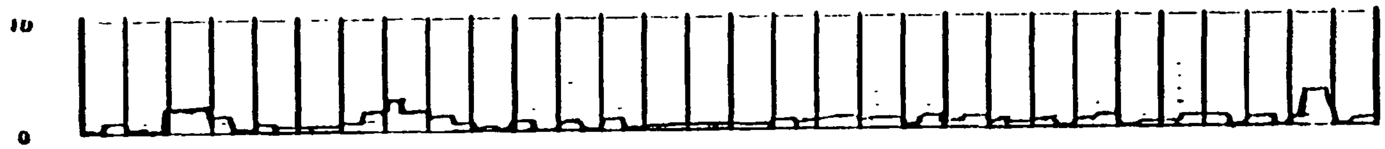
Climate analyzer and anemometers results were compared several times and concordance was found between the results of the two measurements. This indicated that the calibration of the probes was correct and values obtained from either instruments were valid.

The amount of information finally available for the air velocities measured for the forced air and the modular system was more important than for the hydronic system. The reason is that there were up to four anemometers in the comfort room of the first floor while data for the hydronic system relied solely on climate analyzer measurements. The results obtained using these instruments to measure air velocities during periods of normal conditions of operation (as defined in chapter 4) of the heating systems are listed below.

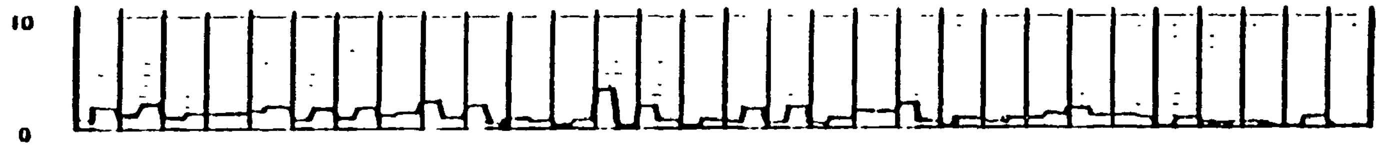
Hydronic System

The results of air velocity measurements during four days when the hydronic system was in operation are shown in Figure 6.1 with climate analyzer readings for 0.6 m height at the center of the room (location B3). The outdoor temperature of the four cases presented were below -10°C. Note that for the case at outdoor temperature of -15°C, the system was turned off for the period between the hours 11 and 19.

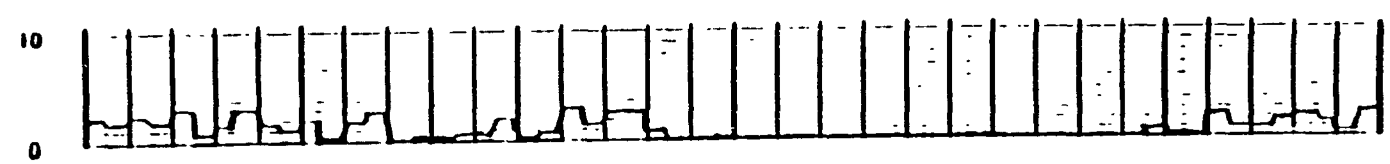
With the system in normal operation (door closed thermostat set at 21°C), the maximum value of air velocity at the center of the room was found to be 0.05m/s. Considering the accuracy of 0.05m/s, a typical value for air velocity could be between 0.0 and 0.1m/s. If we assume no error, then, the air velocity observed for the hydronic ranged between 0.0 and 0.05m's with average near 0.02m/s.



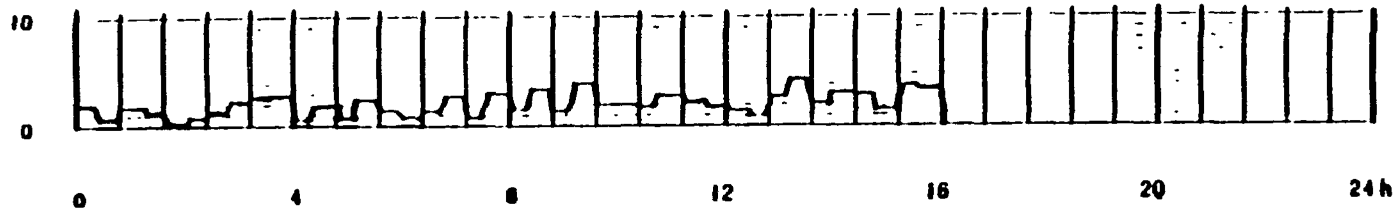
a) Average T_{out}= -12°C



b) Average T_{out}= -14°C



c) Average T_{out}= -15°C



d) Average T_{out}= -24°C

<u>Figure 6.1:</u> Air Velocity, Hydronic System (Climate Analyzer Readings)

The air velocities tended to be higher at lower outdoor temperature when the system cycled more often. The air velocities for the first three cases (Figures 6.1 a,b and c) with average outdoor temperatures near -14°C are on the average lower than for the last case with average outdoor temperature near -24°C (high cycling).

The direct influence of the system on measured air velocities can be observed from the case when the system was off for a certain period of time. Note that in Figure 6.1 c), when the system was not functioning between the 11th and the 19th hour of measurements, the air velocity measured in the room remained null. When temperature measurements were compared in time with air velocity measurements (not shown), the increases in air velocity corresponded exactly to the beginning of the on cycle of the system and followed cycles related to the system's cycling.

From the above results, the cycling of the hydronic system produced air velocities between 0 and 0.1m/s. The higher values of this range occurred when the system cycled on. With the system off, negligible air movements were observed, even at outdoor conditions near -14°C.

Air velocities closer to the heater should be higher than the 0.1m/s maximum observed at the center and, for other locations inside the room air velocities are expected to be lower away from the heater. Although, this was not measured, it is a reasonable assumption with the air velocity found to follow the system's cycling.

Forced Air and Modular Systems

The results for the modular and the forced air systems appear in three tables and eight figures displayed on the following pages. The two tables contain respectively air temperatures information (table 6.1) and air velocity measurements (table 6.2) for four cases while Figures 6.2 to 6.9 show the time variations corresponding to the results of the tables.

Table 6.1: System/Weather Data Air Velocity Measurements

	FORCED AIR		MODULAR	
	F1	F2	M1	M2
CASE	average (°C)			
Outdoor	-1	-7	0	-4
Heater Surface	29	28	45	48

Forced Air

Interpretation of the magnitude, room variations and case to case variations indicate that the measured air velocities for the forced air system are typical of this system. The basis for this interpretation is detailed here and the analysis performed will be repeated for the modular system below.

The magnitude of velocities observed in all cases ranged between 0.00 and 0.2m/s with averages less than 0.05m/s. The measured velocities varied considerably in time between these two extremes with cycles similar to the temperature of the supply for the peak values and randomly for the lower values.

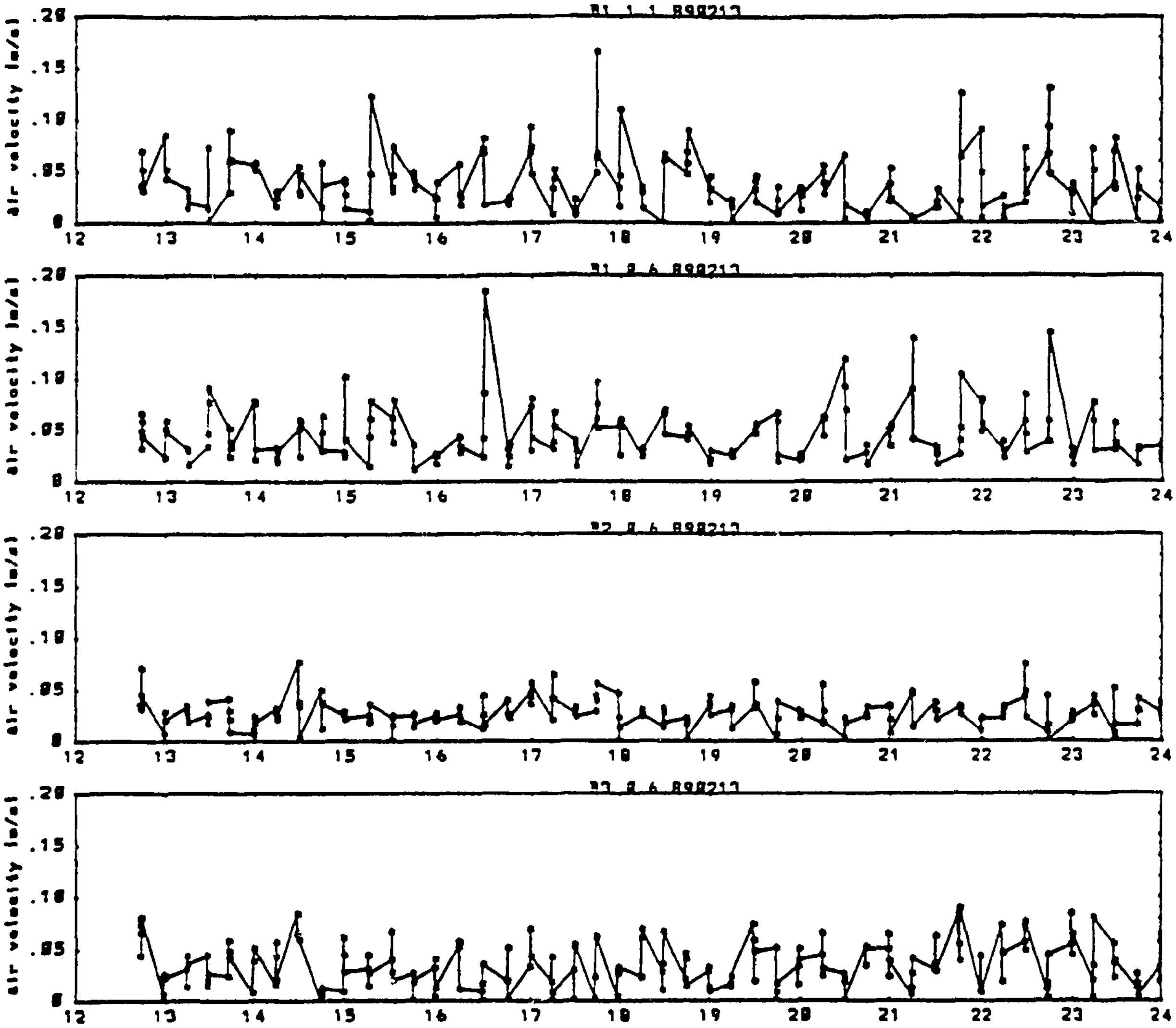


Figure 6.2: Forced Air, Air velocity Case F1 (see outdoor conditions, figure 6.3)

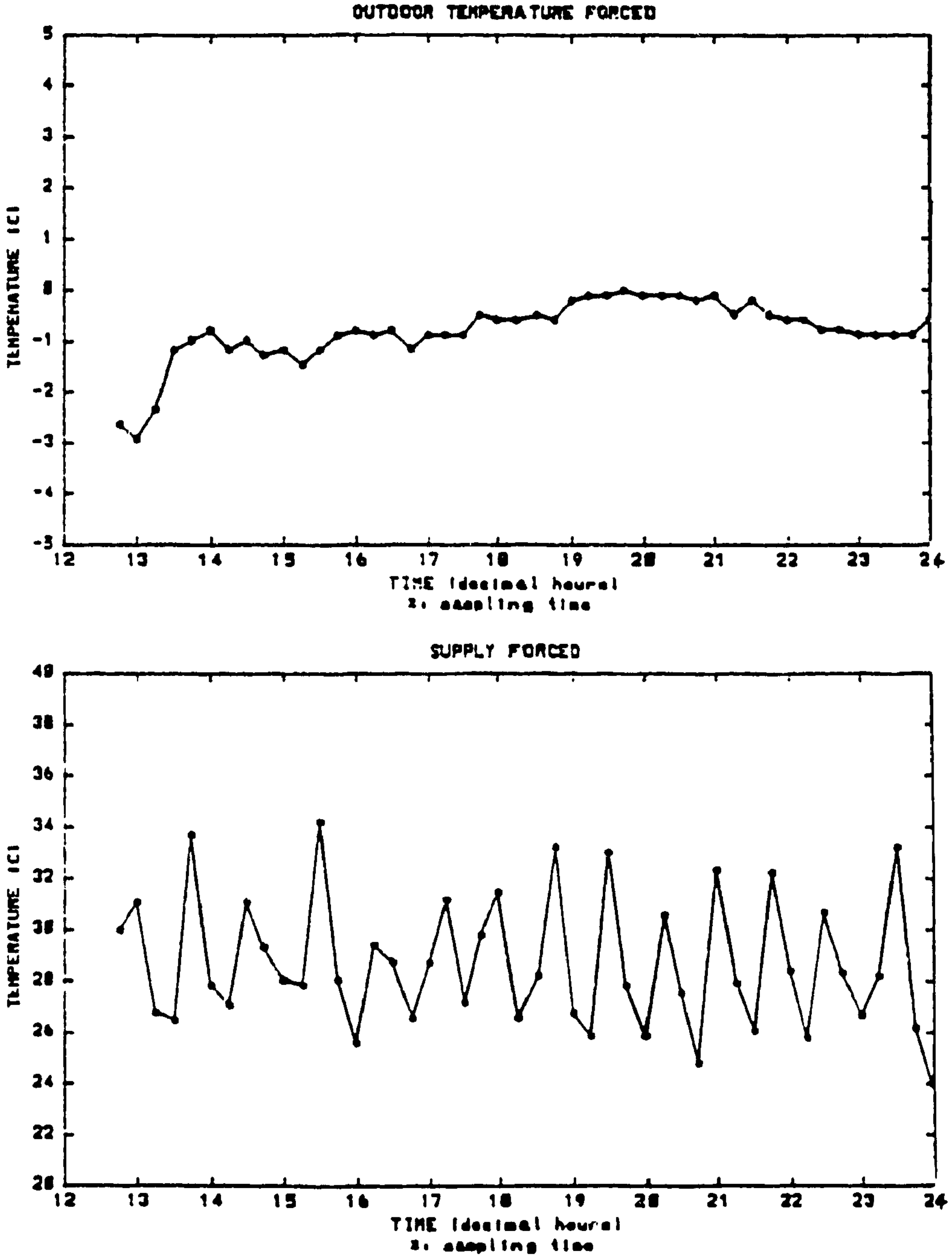


Figure 6.3: Forced Air, Outdoor and Supply Temperatures for Case F1

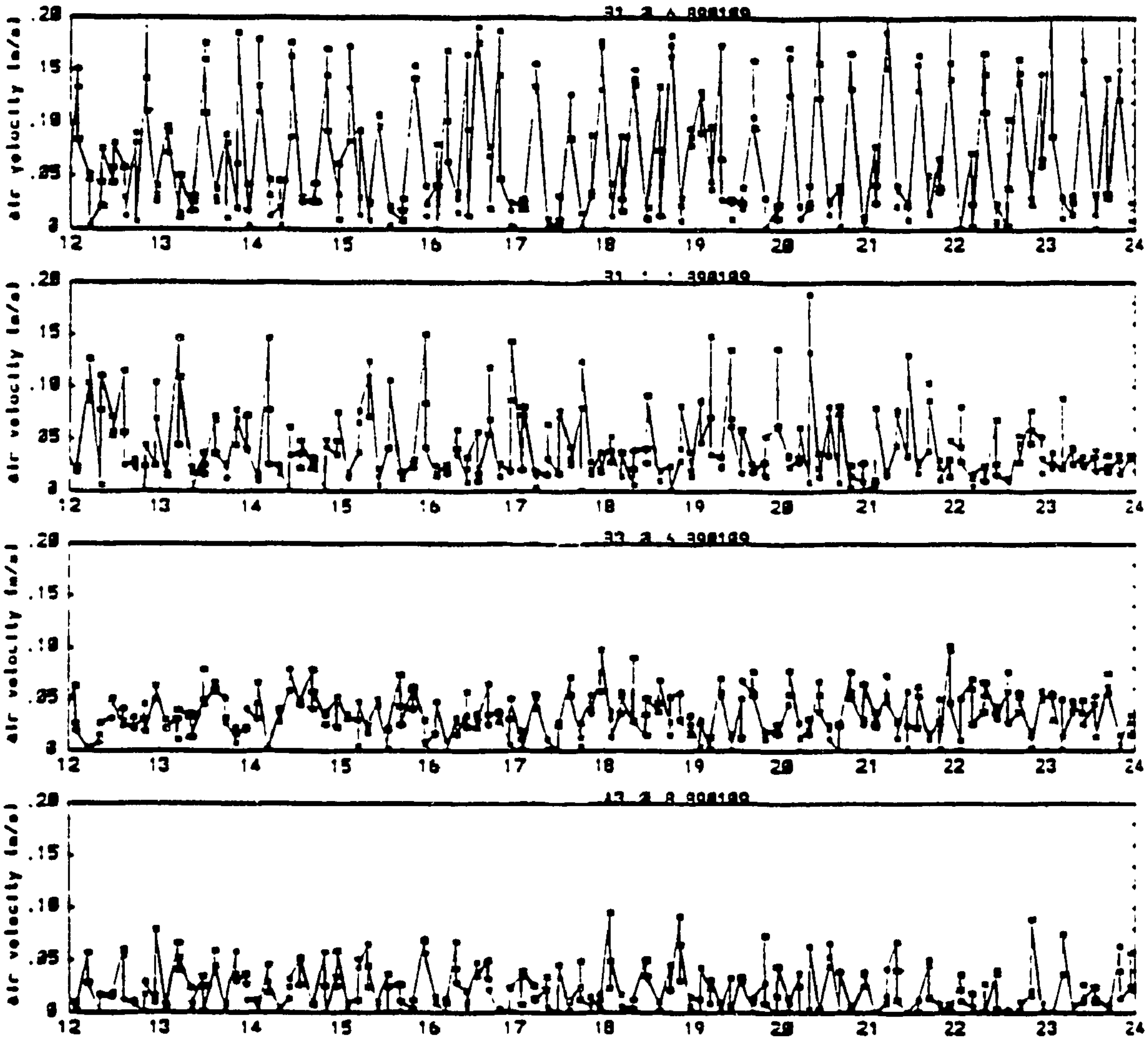
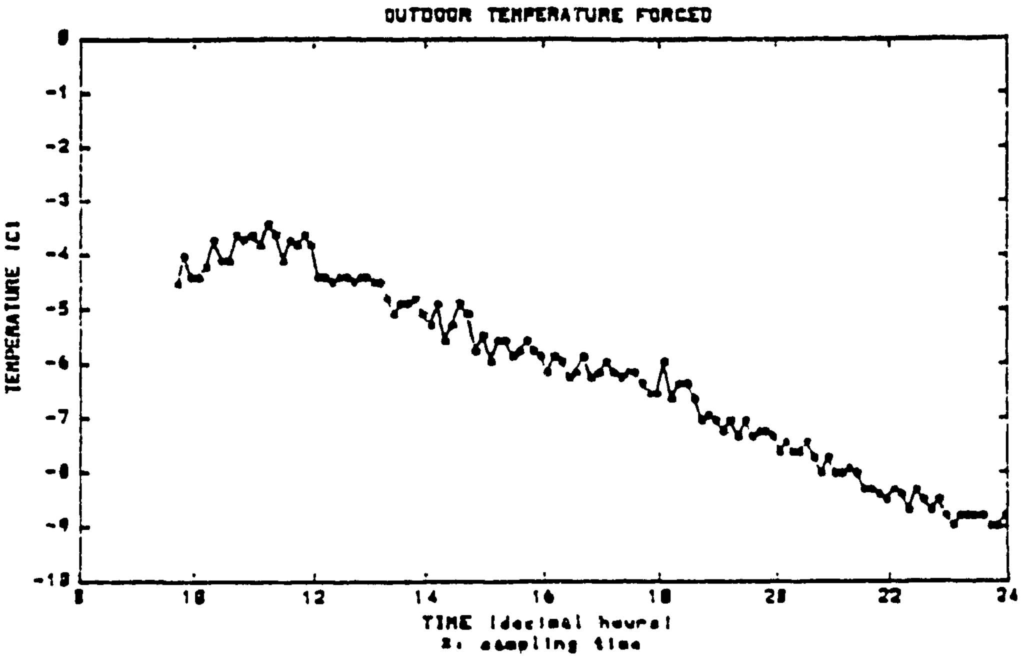


Figure 6.4: Forced Air, Air velocity Case F2 (see outdoor conditions, figure 6.5)



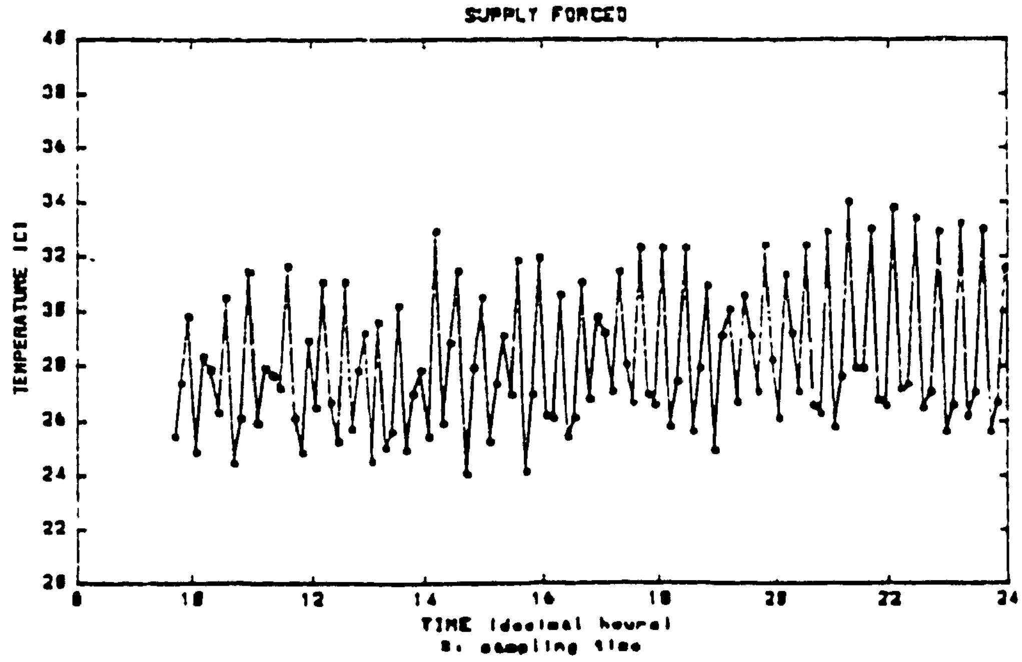


Figure 6 5: Forced Air, Outdoor and Supply Temperatures for Case F2

Table 6.2: Air Velocity Measurements, Forced Air

	Average			Maximum	
Cases	F1	F2		F1	F2
Location			m/s		
B1 1.1	0.04	0.04		0.17	0.19
B1 0.6	0.04	0.07		0.18	0.23
B2 0.6	0.03	-----		0.08	-----
B3 0.6	0.03	0.04		0.09	0.10
A3 0.8	-----	0.02		-----	0.10

With measurements covering different locations in the room, comparison of air velocity obtained for three locations is possible: close to the supply (B1), center (B3) and back of the room (A3). This comparison shows air velocities near the supply to be twice higher in terms of the peak values observed but, average values are not significantly different throughout the room (see Table 6.2).

Results for different heights for one grid location can also be compared. If air velocities at B1 are compared between the 0.6m and the 1.1m levels, for case F1, the difference in average and maximum values are not significant. For case F2, air velocities observed at 0.6m level were slightly higher (avg: 0.07m/s, max: 0.23m/s) than at the 1.1m level (avg:0.04m/s, max: 0.19m/s). Since the system's supply is the major source of air movements, this results indicates that the main forced air's supply level is closer to the 0.6m level than to the 1.1m level in this case and for this location.

In terms of case to case comparison, the magnitude of air velocity increased with the number of cycles of operation (case F2 compared to case F1) As the system cycled more often, the maximum and average air velocities tended to increase. The increase for location B1 0 6m level was more pronounced than for the B1 1.1m level, suggesting that the discharge height is closer to the 0.6m level than the 1.1m level.

The discharge height close to the 0 6m level was also observed in the temperature profiles of the preceding chapter for location B1. In this section, air velocities were found to be higher close to the 0.6m height and less at the 1 1m height and correspondingly, temperature profiles were also observed to vary the most between the 0 1m and the 1.1m heights Therefore, 0 6m height is the best approximation value for the height of discharge based on the measurement points available at location B1.

The forced air system produced air velocity fluctuations higher than the hydronic system Air velocities in the rooms were found to be null on the third floor when the hydronic system was not cycling Therefore, similarly the air velocity on the first floor should be null when the forced air is off since the rooms are similar and no other cause of air movement is important enough (e g.:infiltration, window downdraft) In other words, the air velocities observed were created by the operation of forced air system only and were typical of this system

Modular System

The results for the modular system, for the same room and the same instruments, can be summarized by simply stating that air movements are close to null in terms of averages (maximum of 0 03m/s) and maximums (maximum of 0 04m/s) and these are significantly low values relative to the available accuracy of 0 03m/s

Table 6.3: Air Velocity Measurements, Modular 1st Floor

	Average		Maximum	
Cases	F1	F2	F1	F2
Location	m/s			
B1 1.1	0.01	0.01	0.02	0.02
B1 0.6	0.01	0.01	0.03	0.04
B3 0.6	0.02	0.02	0.03	0.03
A3 0.8	0.00	0.00	0.01	0.01

Assuming perfect measurements then the spatial variations reveal a trend similar to the forced air system. The velocities observed closer to the window (locations B1 and B3, 0.04m/s maximum) are higher than for the back of the room (0.01m/s maximum). For the forced air system, however, this difference could be directly attributed the supply. In this case, the low values indicate no direct effect of the system on air movements.

In terms of difference between cases, the maximum and minimum air velocities were similar for high and low operation cycling. Case M1 compared to case M2 is not significantly different in terms of outdoor temperature but, this was also true for the forced air system and yet, differences were measured.

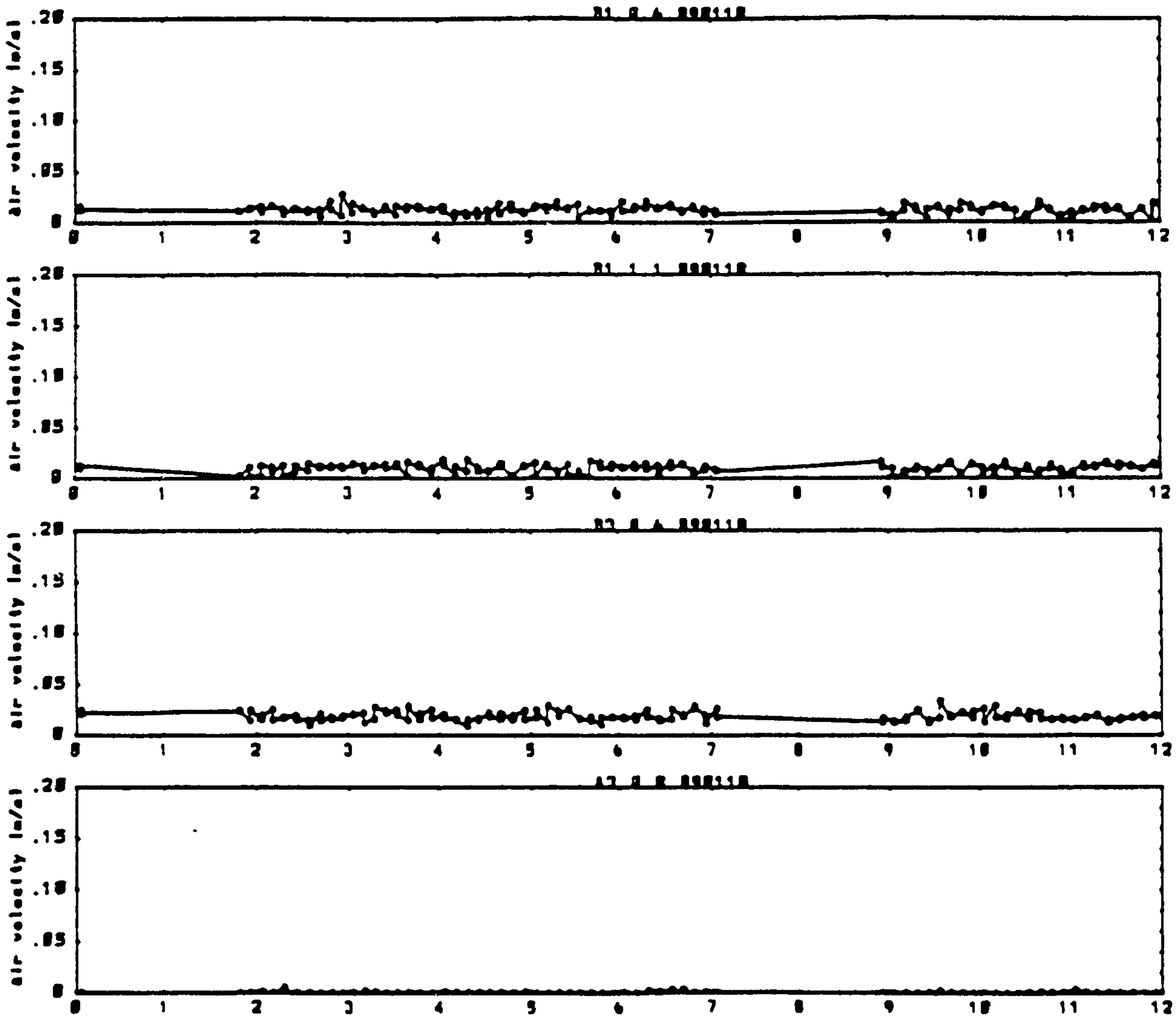


Figure 6.6. Modular 1st Floor, Air velocity, Case M1 (outdoor conditions, figure 6 7)

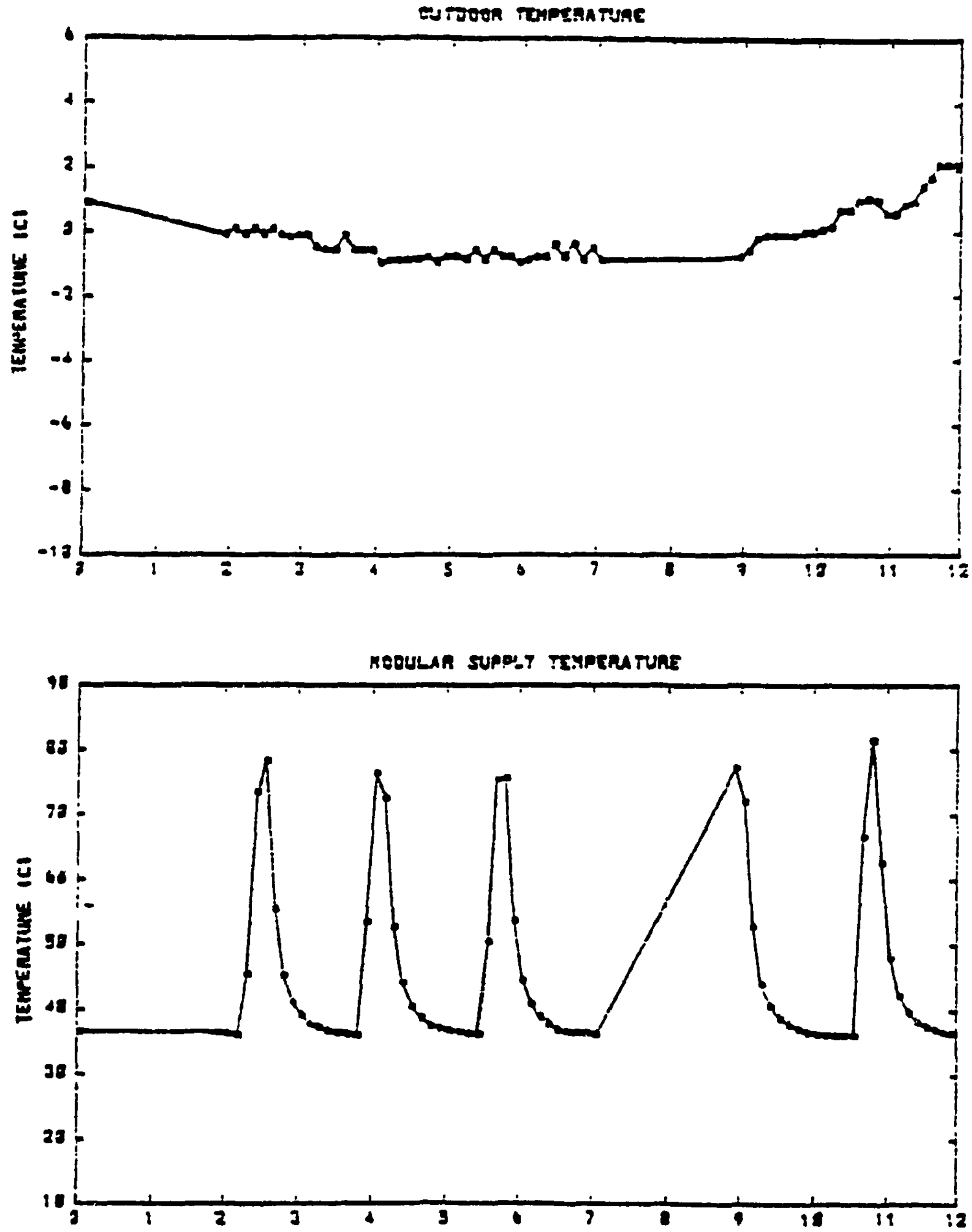


Figure 6.7: Modular 1st Floor, Outdoor and Supply Temperatures for Case M1

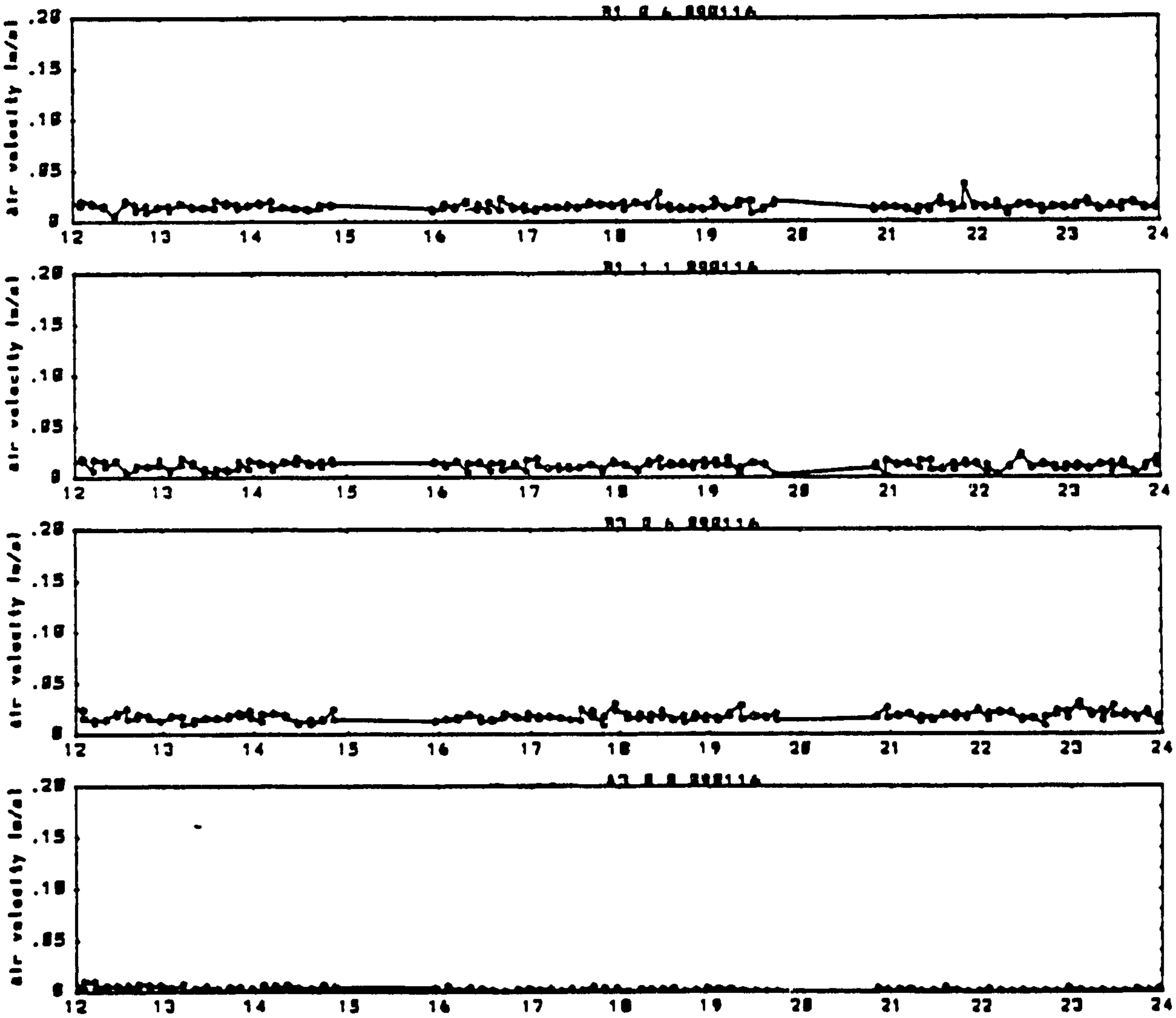


Figure 6.8: Modular 1st Floor, Air velocity, Case M2 (outdoor conditions, figure 6.9)

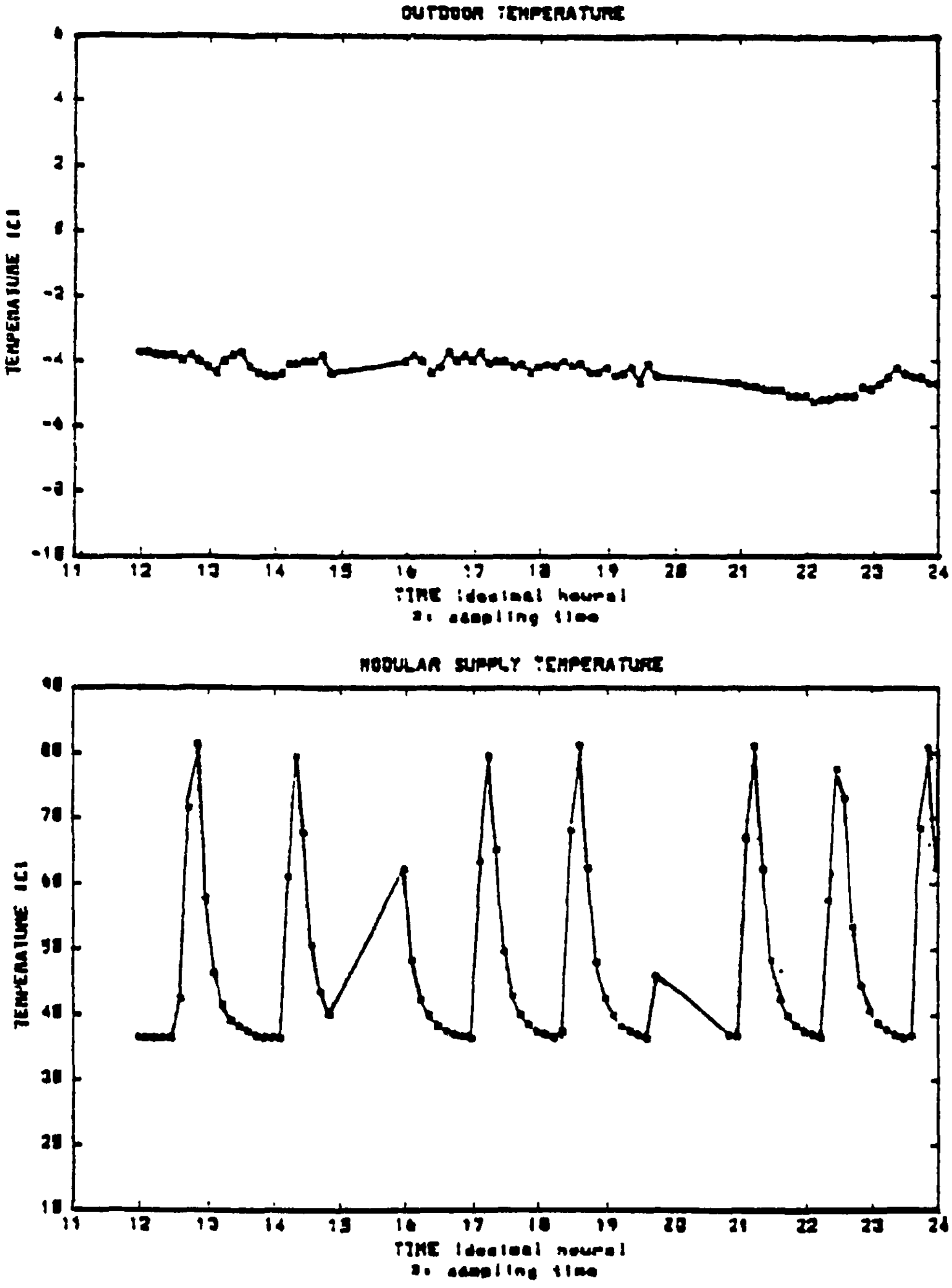


Figure 6.9: Modular 1st Floor, Outdoor and Supply Temperatures for Case M2

The air velocity observed for the modular system were small at all locations in the room (between the 0.6m and 1.1m levels). Higher values can be assumed to occur at higher levels based on temperature profiles. In chapter 5, temperature profiles were found to vary more above the 1.2m level for the modular system indicating that warm air was distributed to these levels. Therefore, it can be assumed that most of the disturbances in air velocity should be observed at those levels.

Summary

From measurements over a number of cases, differences in terms of the magnitude of the air velocities measured as well as the effect of cycling were found.

In terms of the magnitude of the observed air velocities at the center of the room, for the cases studied, the forced air, the modular and the hydronic system had respectively maximum average values of 0.04, 0.02 and 0.02m/s and maximum values of 0.1, 0.03 and 0.05m/s. The forced air system produced air velocities two times higher than the other two systems.

For the levels studied, the forced air system was found to have a marked discharge height located near the 0.6m level where increases in air velocities were observed with higher cycling operation. This results compares well with temperature profiles observed to vary significantly at the same level.

Discharge heights cannot be similarly established for the modular system and insufficient data was available for the hydronic system.

Based on temperature profiles, the modular system is expected to create higher velocities close the ceiling and the hydronic system closer to the baseboard. For the hydronic system, the maximum air velocity should vary in height since temperature profiles depend on cycling behaviour.

Note however, that higher velocities were observed for the modular system but results were not included here since. The higher velocities were found for few cases however.

From the above, the results tend to indicate the presence of three separate classes of air movements: low (modular), medium (hydronic) and high (forced air). Possible sources of air movement, other than those created by the systems (downdraft and infiltration), were not detected as shown for the case with the hydronic system off and low outdoor temperatures.

6.2 Radiant Temperature

Radiant temperature measurements using the globe thermometers are described along with air temperature measurements. Results refer mostly to the location close to the window where the heating systems should provide the correct compensation of air or radiant temperature to offset the cold window temperatures. A description of the test procedure and results for a few cases studied for each system follows.

Test Procedure

Information for mean radiant temperature was gathered using either the globe thermometers or the radiant temperature sensor on the climate analyzer. The globe thermometers were used at different locations, and at grid points where thermocouples were located. The probe of the climate analyzer was usually located at the center of the room.

The measurements made using the climate analyzer spread over two heating seasons on the first and third floor and its location remained at the center of the room, at a height of 0.6 m.

Due to the expected lower surface temperatures at the window compared to the rest of the room's surfaces, the probe was oriented such that one side was facing the exterior wall and one side faced the back wall of the comfort rooms with the sensor plane parallel to those surfaces.

Prior results from the climate analyzer located at the center of the room showed that at the difference between plane radiant temperature from the front of the room and the back remained less than 0.5°C. This was observed for all systems and for all the cases recorded.

Results for mean radiant temperature from the globe thermometers are based on the equation given in chapter 3, where a low accuracy of ±2°C was calculated. From the small range obtained with the climate analyzer (0.5°C maximum asymmetry at the center), it was expected that the accuracy of Tmr calculated with globe thermometer readings would not be sufficient to reveal small differences between air and radiant temperatures.

The surface of the window is small (less than 5% of room total) and its contribution is less important than walls, ceiling and floor surfaces. Therefore, radiant asymmetry was not expected to be a problem with other surfaces.

Results

For analysis, seven cases were selected to describe mean radiant temperatures in the rooms with two cases for the forced air system and the hydronic system each and three cases for the modular system. The conditions for the tests are listed in Table 6.4. The cases were selected to obtain results for relatively cold days and different globe thermometer locations. The data shown in the Table 6.4 indicate outdoor and indoor air temperatures as well as the surface temperature of the interior surfaces (walls, ceiling, floor).

A coarse estimation of the expected difference between radiant and air temperatures based on surface temperatures can be seen in the table. The temperatures were subtracted to obtain a relative value of the difference between room surface and window temperatures (T1-T2), supply and window (T3-T2), supply and surfaces (T3-T1) and finally, between air and the average of the supply, surfaces and window temperatures.

As a rough estimate of the maximum difference expected between air and mean radiant temperatures is 3°C-the difference between air and all surfaces' temperatures. Surfaces could have been weighted according to their area for a better approximation. For the location close to the window, this method is more accurate. The contribution of the forced air system to radiant temperature is not represented as well in this fashion since this system does not actually have a high temperature body in the room. In the next chapter, more precise angle factors will be used in calculations for thermal comfort assessment.

Radiant asymmetry problems, if any, were expected to be more important due to the difference between window and heater's supply temperatures. The differences between heater and window temperatures ranged between 1°C and 13°C (forced air excluded) and therefore was more important than the differences between supply and other room surface temperatures which ranged between 1°C and 8°C.

Table 6.4: Description of Cases Selected for Globe Temperatures

	Average Temperatures (°C)						
CASES	F1	F2	H1	H2	M1	M2	M3
Floor	1	1	3	3	1	3	3
outdoor	-13	1	-13	0	-7	-7	-6
T1. surfaces	23	22	20	22	22	22	22
T2. window	15	18	15	16	16	14	14
T3. supply	29	29	28	23	25	23	23
T4. air at B3 1.6m	24	23	20	22	23	22	22
(T1-T2)	8	4	5	6	6	8	8
(T3-T2)	14	11	13	7	9	9	9
(T3-T1)	6	7	8	1	3	1	1
T4-(T1+T2+T3)/3	1.7	0.0	-1.0	1.7	2.0	2.3	2.3

For the cases listed in Table 6.4, calculations of the difference between globe and air temperatures were made and the results are listed in Table 6.5.

In very few cases there was a significant difference between globe and air temperatures. In all cases, except one (case Modular case 3, location B1), the globe temperatures were found to be less than the air temperatures. Differences less than -2°C were observed 80% of the time in all cases.

Table 6.5: Difference Between Globe and Air Temperatures

Percentage of Observations

-1 < Tg-Ta > -2 (°C)

CASES	F1	F2	H1	H2	M1	M2	M3
location							
B1 (0.8m)	--	2	--	1	--	--	0
B2 "	14	--	0	0	2	1	--
B3 "	0	0	0	0	0	0	0
B4 "	5	--	0	--	0	18	10
A3 "	--	9	--	--	--	--	--

+1 < Tg-Ta > +2 (°C)

B1 (0.8m)	--	0	--	0	--	--	2
B2 "	0	--	0	0	0	0	--
B3 "	0	0	0	0	0	0	0
B4 "	0	--	0	--	0	0	0
A3 "	--	0	--	0	--	--	--

In terms of the variations from grid points to grid points, we find that in all cases, there was no difference between air and globe temperatures at the center of the room (location B3) as found previously using the climate analyzer.

For locations close to the walls, on the third floor, at the location close to the neighbour wall, there are differences between globe and air temperatures for cases M2 and M3. If we verify the neighbours' wall temperatures for these cases, we find that they were on the average 1°C less than the air temperature at the same location which explains the difference.

For case F1, the lower globe temperatures at location B2 are explained by the interior wall being on the average lower than air temperature. From the data for this day, it was found that the interior wall (close to B2) had averaged 19°C compared to the average of 23°C for the air temperature at location B2. The low interior wall temperatures are due to a 2°C set back which preceded the time period of case F2.

For the locations close to the window, cases F2, H2 and M3 provide information on globe temperatures at location B1. For these cases, graphs showing air, globe and mean radiant temperatures appear in Figures 6.10 to 6.12 for cases F2, H2 and M3 respectively. The results for this location are discussed for each system separately and for cases where significant differences were observed between globe and air temperatures.

The mean radiant temperature was calculated based on the measured globe and air temperatures with air velocities of 0.05 and 0.15m/s. These values were selected to cover the possible range of air velocities expected to occur.

For the forced air system at location B1, (Figure 6.10-a) the variations in globe temperatures followed the variations in air temperatures in time with globe temperature on the average 0.4°C less than the air temperatures and the maximum difference between the two was 1.2°C.

For case F2 (Figure 6.10 b), the difference between the calculated mean radiant temperature and air temperature was less than 2°C (Va=0.15m/s) and was on the average less than 1°C. Therefore, the effect of the low window temperature on mean radiant temperature is to create mean radiant temperature at location B1 on the average 1°C less than air temperatures. This is small considering the ±2°C accuracy.

Hydronic System

For the hydronic system (see Figure 6.11-a), the variations in globe temperatures followed the variations in air temperatures in time with the globe temperatures varying over a smaller range. The globe temperature was on the average 0.8°C less than the air temperatures and the maximum difference between the two was 1.7°C. For case H2, the difference between the calculated mean radiant temperature and air temperature was less than 2.5°C (Va=0.1m/s) and on the average less than 1°C (Figure 6.11-b).

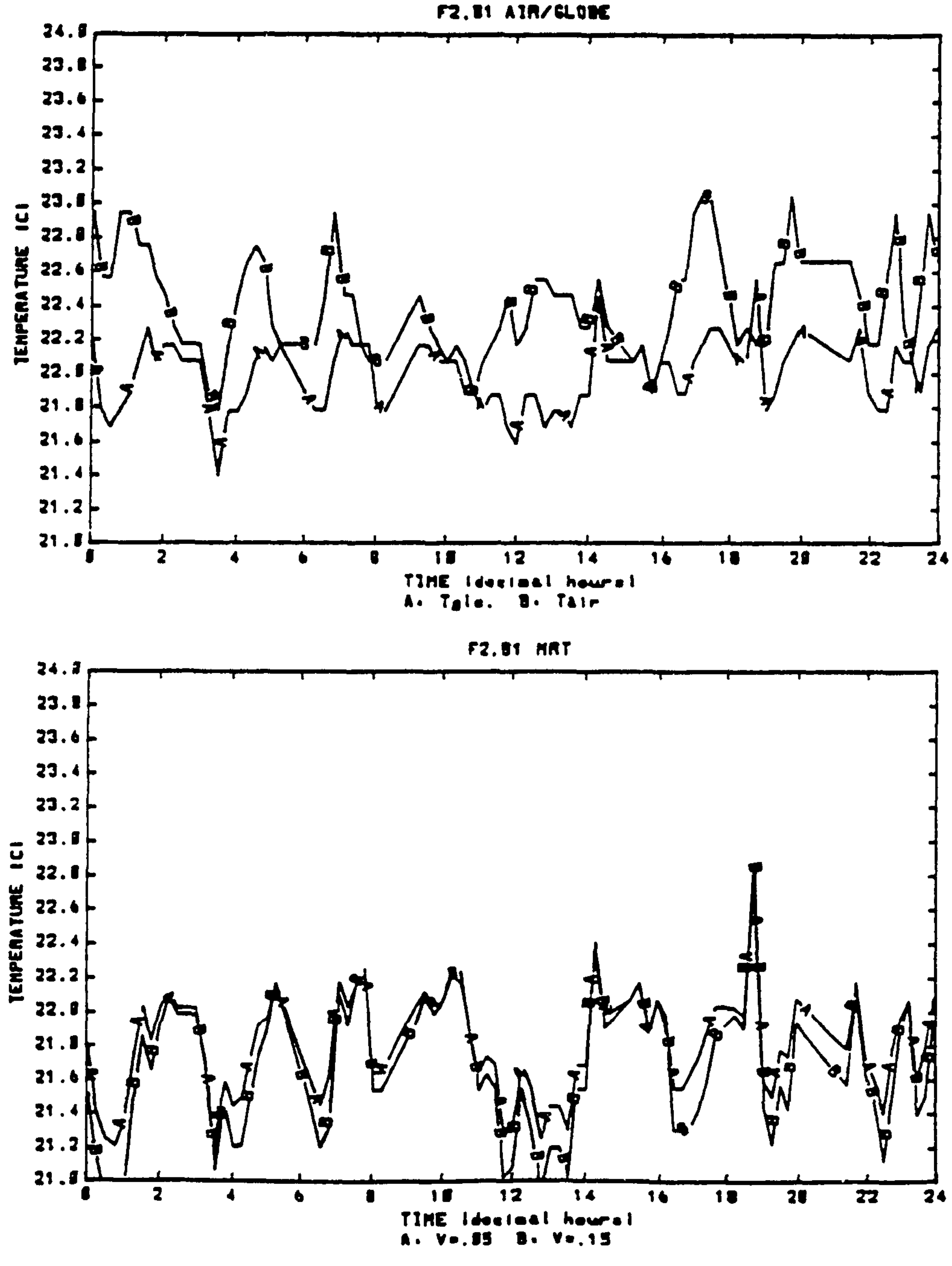


Figure 6.10: Comparison of Air, Globe and Mean Radiant Temperatures, Forced Air

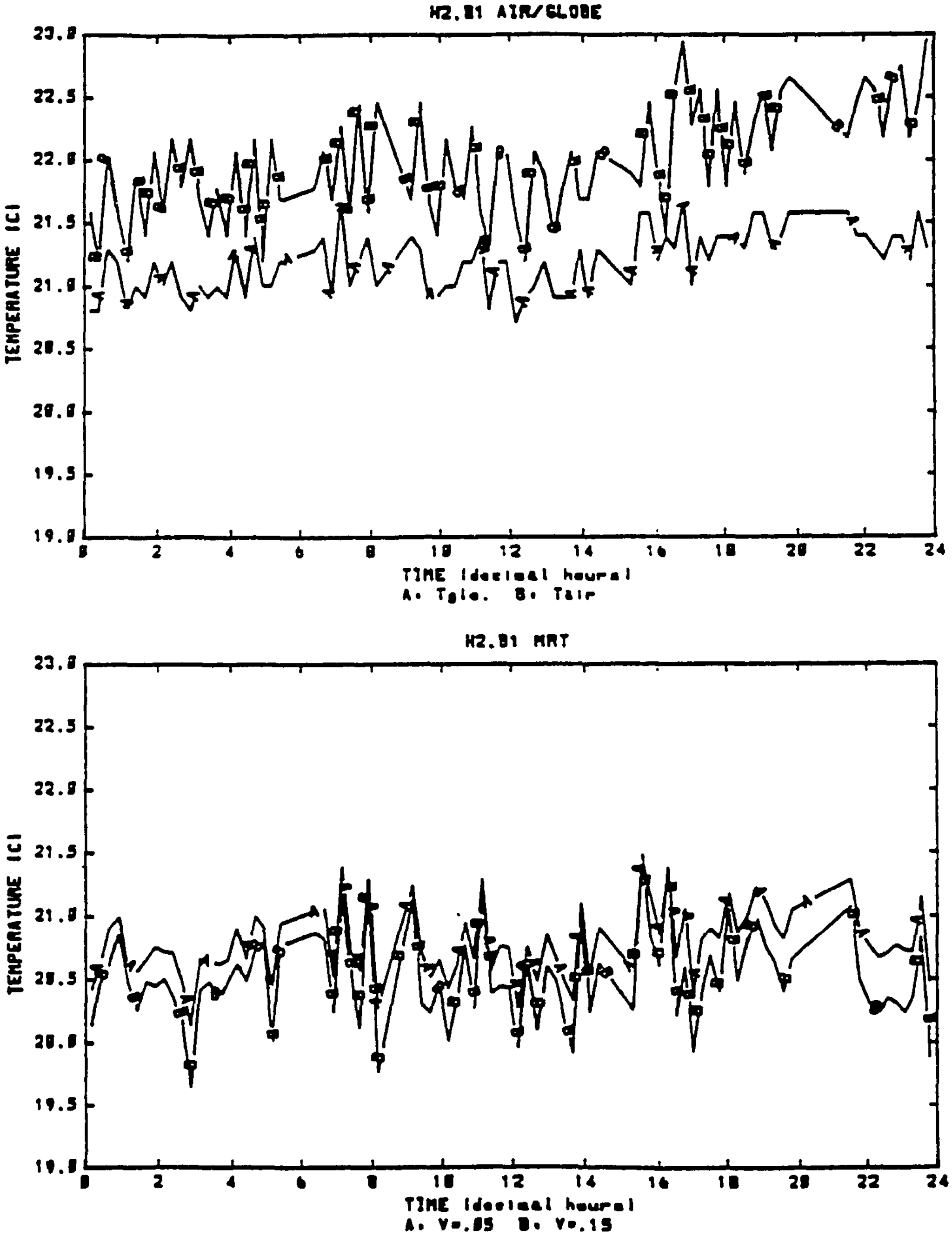


Figure 6.11: Comparison of Air, Globe and Mean radiant Temperatures, Hydronic

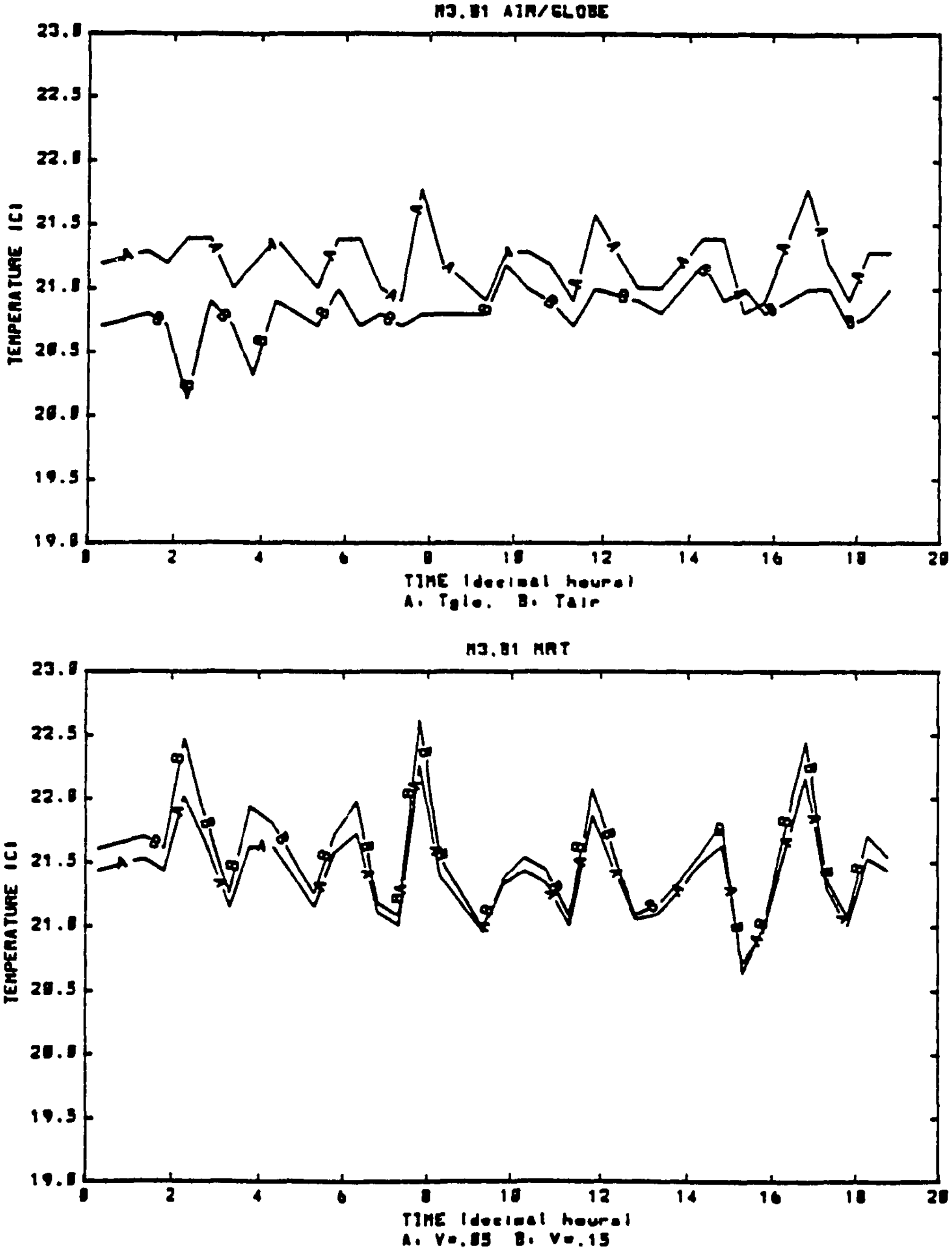


Figure 6.12: Comparison of Air, Globe and Mean radiant Temperatures, Modular

The high surface temperature of the baseboard (average 23°C) did not produce higher mean radiant temperature at the location close to the window contrary to the results expected. The mean radiant temperatures facing the window were not significantly higher than those observed for the forced air system which did not have a high temperature body in the room.

Therefore, the compensation of the baseboard is not significantly different in terms of mean radiant temperature from the compensation of the forced air system even though the radiative components were expected to differ because of the baseboards high temperature.

Modular System

For the modular system on the third floor (see Figure 6.12-a), the variations in globe temperatures follow the variations in air temperatures in time, and again, the globe temperatures varied less in magnitude at location B1. The globe temperature was on the average 0.4°C higher than the air temperatures and the maximum difference between the two was 1.3°C.

For case M3, the calculated mean radiant temperature was higher than air temperature with a maximum less than 2.0°C ($Va=0.05m/s$) and average less than 1°C (Figure 6.12-b). This case is different from the other two since globe temperatures were higher than air temperatures but the difference is small relative to the accuracy of the measurements.

According to the results, the modular system will not contribute to major changes in mean radiant temperatures at the location closer to the window.

Summary

From the above discussion, the mean radiant temperatures observed inside the room and for the different cases were not significantly different from air temperatures at the different locations except close to the window. The measured globe temperatures close to the window indicate that less than 2°C difference between air and mean radiant temperatures are expected for all systems.

Mean radiant temperature rises due to the presence of the hydronic system baseboard were not significant. The radiant heat gains produced by this system were small if limited to the case studied and comparable to those of the forced air system.

Due to the small differences between air and mean radiant temperatures, the thermal environment can be well represented using air temperatures only. Well insulated surfaces in the room combined with the absence of direct solar gains and the relatively low surface temperatures of the systems created small radiant contribution for locations close to the window and negligible at the center. The importance of radiant temperature at the window may have been greater with tne exterior shading removed.

6.3 Relative Humidity

Relative humidity measurements were less extensive than for the other parameters. Results are discussed mainly to show the range found for this parameter and the difference between floors.

Relative humidity was measured in the comfort rooms on the first and third floor of the building. Part of the data were collected using an analog humidity transducer, and part from point measurements with a wet/dry bulb thermometer. The transducer was used during the first year of the project, period beyond which its calibration became less reliable. Afterwards relative humidity measurements were taken using the mercury thermometer.

The results for the analog measurements indicated that relative humidity on the first floor was higher than on the third floor and that variations of this parameter in time were small (inside rooms). From the bulk of the measurements, the relative humidity on the first floor was found to settle near 40% compared to 25% values for the third floor.

Approximately forty wet/dry bulb thermometer readings made at the beginning and at the end of different recording periods during the heating season appear on Figure 6.13. The distribution shown does not reveal a particular difference in humidity with respect to either the first or third floor with overall values varying between 15% and 60%.

Considering the particular time when the measurements were made, humidity on the third floor were either equal or less than on the first floor. On the average the third floor Rh was 15% lower than on the first floor.

The particular value of humidity with respect to the floor of the building depend on the heating system installed, the building's envelope properties (infiltration and permeability), the outdoor conditions (pressure difference) and the water vapour generated inside (occupants, none in this case) and from the neighbours.

Since the forced air and hydronic systems were constantly working during most of the tests (even when the modulars were tested), it is reasonable to assume that they will have a more significant effect on floor humidity. However, the building envelope, different rates of infiltration for each floors and pressurization combined with different outdoor air temperatures, moisture content and wind velocities complicate this assumption.

The final remark is that relative humidity levels for the third floor and first floor should be taken as 40% and 25% respectively in order to account for the values measured in the field.

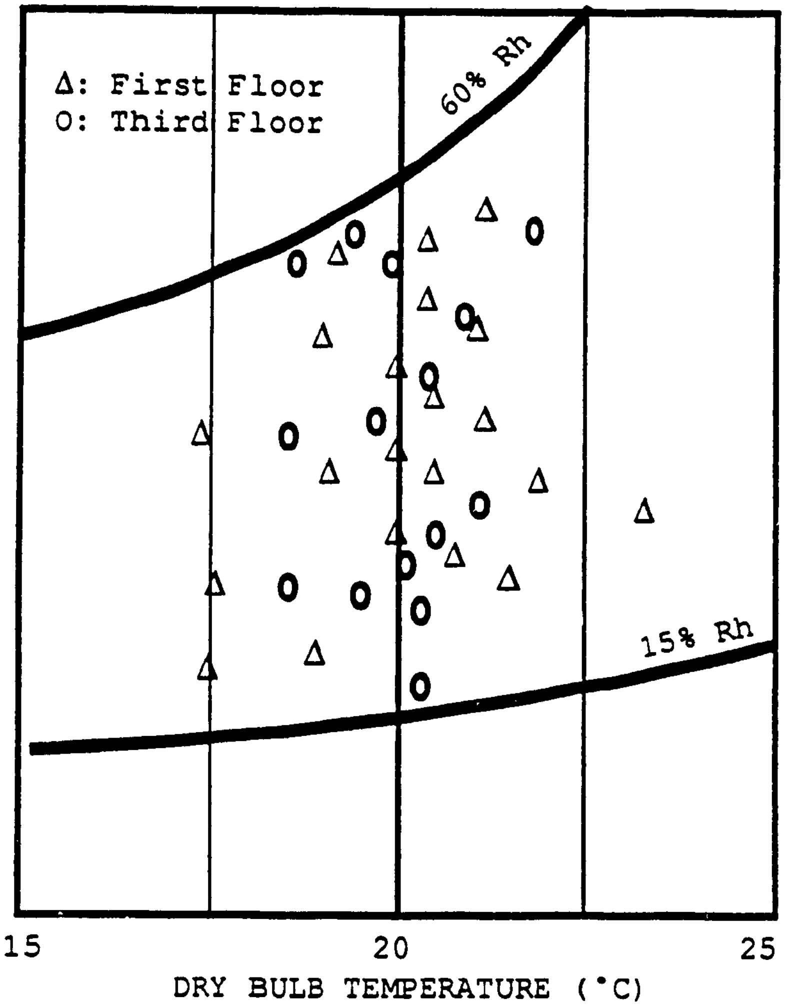


Figure 6.13: Measurements of Floors' Relative Humidity Made at Beginnings and Ends of Some Data Collection Periods

6.4 Summary

The three parameters discussed in this chapter completed the list of the comfort parameters needed to assess thermal comfort and ranges in which these parameters were found was established from selected cases.

Air velocity measurements in the rooms showed that the systems belonged to different classes. The modular system produced almost no air currents, the hydronic system produced somewhat more (air velocity less than 0.1m/s), and the forced air the highest (peaks of 0.23m/s). The results for the hydronic system being restricted to the center of the room, higher values should be taken for the location close to the heater.

The differences between air and mean radiant temperature observed for all cases were small indicating that heaters temperature and room surfaces temperatures were not providing major radiant gains or losses in the enclosure. The compensation of the systems is therefore adequate on this point due to the well insulated surfaces and the relatively small importance of the window in terms of area and surface temperature. This does not apply to other cases for different possible levels of insulation and window size.

Finally, the relative humidity measurements indicated lower humidity levels on the third floor (25%) compared to the first floor (40%) and results were limited due to the number of parameters involved which were not measured and because of the small influence of relative humidity on comfort.

CHAPTER 7

7.ASSESSMENT OF THERMAL COMFORT

In the previous chapters, environmental parameters were discussed one parameter at a time and without addressing thermal comfort in a specific manner. With the analysis performed, the range and fluctuations of air temperature, mean radiant temperature, humidity as well as air velocity produced by the different systems have been established separately. These parameters are reviewed in two sections: one on global comfort and the other on local discomfort.

To establish global comfort in the rooms, the PMV, PPD and LPPD indices will be used. These indices provide the thermal sensation vote (PMV), the number of persons expected to be dissatisfied from the thermal environment (PPD) and a measure of thermal uniformity (LPPD). A comparison of the calculated versus the measured values for the PMV index and a description of the observed values for all three indices and each systems will be made, based on seasonal observations and individual cases.

The second section on local discomfort consists in a table summarizing the observed thermal gradients, floor temperatures, radiant asymmetry, temperature fluctuations and air velocities observed for each system. This table lists the performance of each system with respect to the ISO and ASHRAE standards' limitations.

Both global and local discomfort will be reviewed in the last section which precedes the final chapter summarizing results and providing some recommendations.

7.1 Global Comfort

In this section, the accuracy of PMV measurements is first established. The results on calculation of the thermal comfort indices are reported for each system.

The first part on accuracy does not contain elaborate details on how the PMV was calculated and concentrates mainly on showing how calculations from the measurements compare with the readings from the PMV meter.

The second section details the calculation procedure, the cases studied, and finally, the evaluation of the comfort indices (LPPD, PMV, PPD) is accomplished based on measurements made during the season.

7.1.1 Predicted Mean Vote Index: Comparison of Measured and Calculated Values

In order to accurately calculate PMV for different locations in the comfort rooms the measurements obtained from the PMV-meter were used as a reference. This reference was selected since it provided a quick and integrated calculation of the PMV index which allowed to verify our calculations and proceed with confidence to a more elaborate comfort assessment.

For several reasons, the use of the comfort equation to calculate PMV can be assumed at first to be a less accurate method than using the results from the PMV-meter. A main disadvantage with the comfort equation is that the error in the results obtained contains the sum of the error in the values of the four physical variables (Ta, Tmr, Va, Rh) and the error found in the estimation or calculation of the three human parameters (clothing insulation, activity levels and angle factors or body shape). These seven sources of error combined, the accuracy of the results from the equation is therefore expected to be small.

The PMV-meter, on the other hand, has high accuracy specifications (± 0.5°C for temperature measurement) and has the advantage of operating on a sensor which simulates a human being in terms of both radiant and convective heat exchange. Furthermore, the meter was also compared for accuracy with another meter and the results from the two meters agreed with a difference in PMV less than 0.01.

The PMV meter readings are the "measured" PMV values which were compared with the "calculated" values from a computer program we developed. The program was partly taken from a listing in the ISO standard [9] and used experimental measurements of air and mean radiant temperatures. Some small differences existed in terms of the assumptions used for the calculation method and the measurements method. As will be seen, these differences created no significant effect on the results.

The PMV meter was located in the center of the room at a height of 0.6 m with its parameters set for a clothing insulation of 1.2 clo, activity level of 1.0 met and vapour pressure of 1.2 KPa (winter clothing, low activity). Air velocity was measured indirectly by a heat balance on the probe of this instrument. The orientation of the probe simulated a seated person, facing the window.

Recordings of PMV for the modular and the hydronic systems lasted one day, after this period, the PMV was calculated for the same location with the same values used for the PMV meter (1.0 met, 1.2 clo and 1.2 KPa). The measured air and surface temperatures were used in the calculations with the mean radiant temperature calculated from the same surface temperatures combined with the angle factors from the Fanger diagrams [3].

The orientation selected for calculation was a 360° average instead of the window orientation for a seated person (0.6 m). This was done in order to use the calculations for the next sections, to deal with a more general case. Fixed air velocity values of 0.1, 0.15 and 0.2m/s were used to cover a wide range of actual air velocities, and since data were not available for this parameter on the third floor.

Results showed that measurements and calculations were in excellent agreement with calculated PMV higher than measured by less than 0.30 on the thermal sensation scale. For the cases' conditions, a difference in 1 degree in air and mean radiant temperature would cause the PMV to vary by only 0.30. This results was obtained for both the hydronic and the modular system as shown in Table 7.1 (see Figures 7.1 and 7.2). A study by Spain reported similar average differences (0.33 PMV) for a similar comparison with calculated values higher than measured values [28].

Table 7.1: Comparison of Calculated and Measured PMV

		PMVcal-PMVmeas.(avg)	
Case	Va	Hydronic	Forced
	(m/s)	Therm. Sens. Scale	
1	0.10	0.25	0.26
2	0.15	0.14	0.15
3	0.20	0.06	0.07

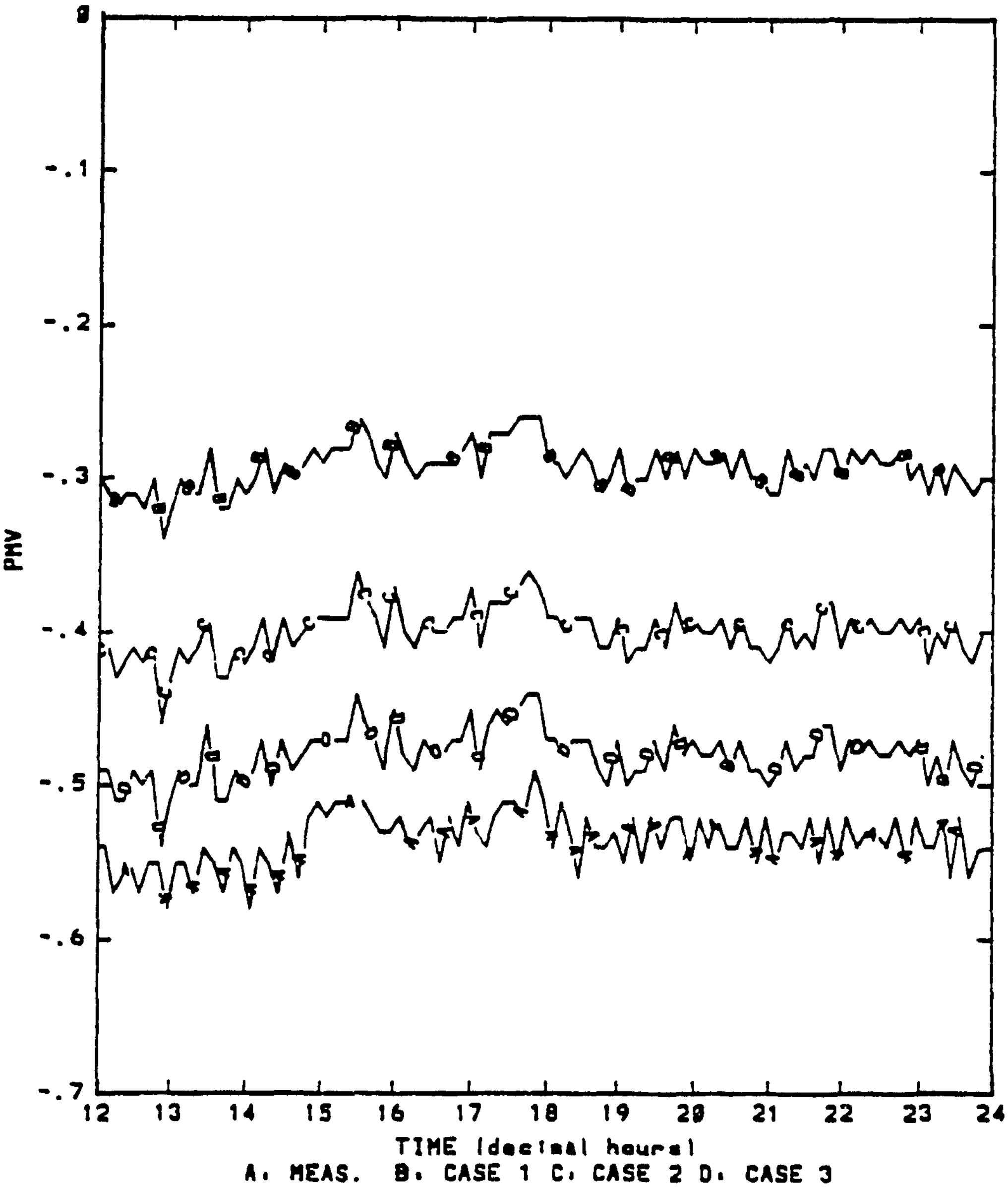


Figure 7.1: Comparison Between Calculated and Measured PMV, Hydronic System

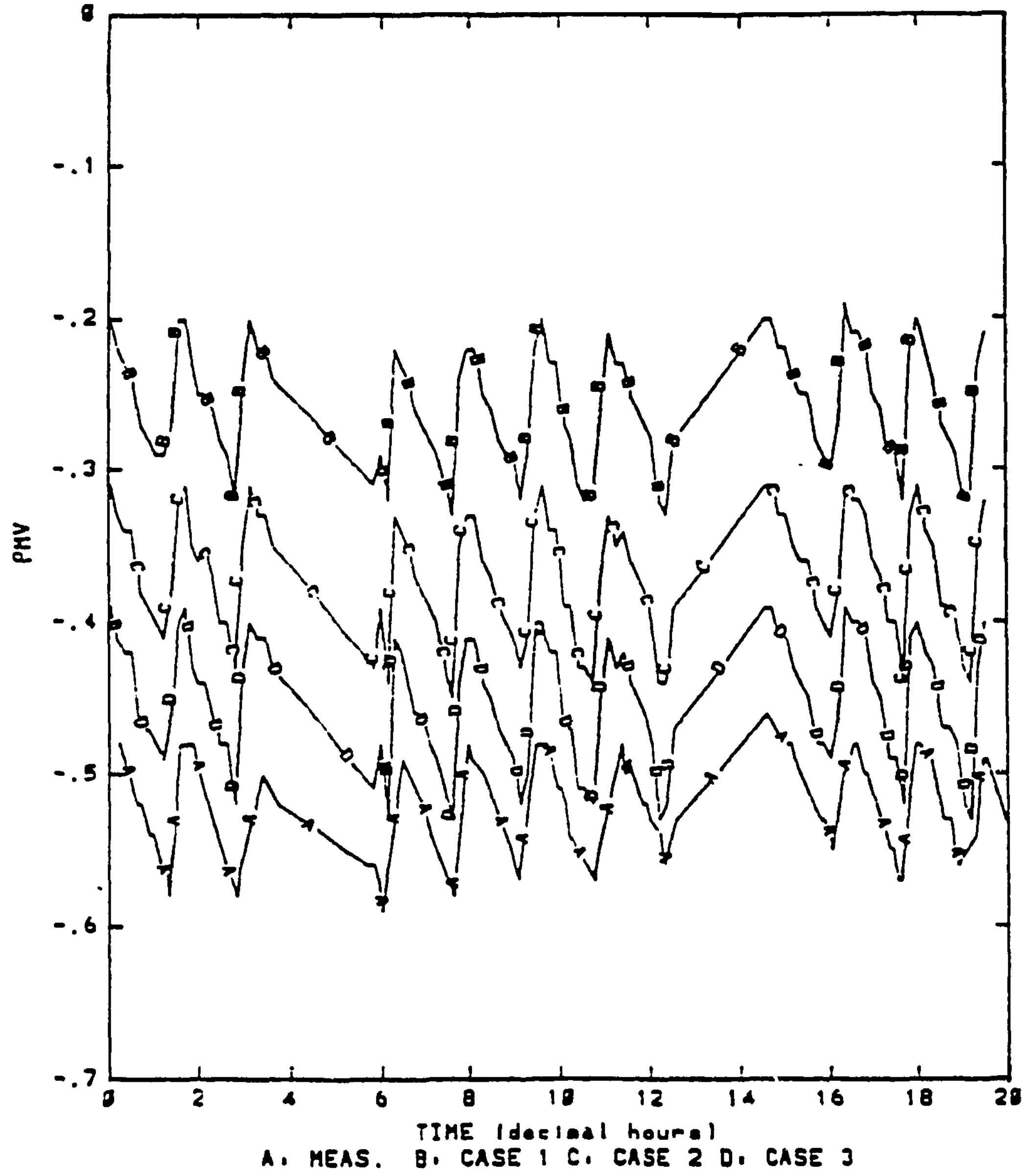


Figure 7.2: Comparison Between Calculated and Measured PMV, Modular

For the periods considered in the tests, the values of PMV for the modular system varied more (0.1 PMV range) than for the hydronic system (0.05 PMV range) but the difference between calculated and measured PMV values is not significantly higher in one case or the other. As will be seen later, these ranges are representative of the tests performed over the heating season.

Not only did the average difference between calculated and measured PMV remained small but also, the calculation method provided the same results in time. This can be observed in Figures 7.1 and 7.2. for the hydronic and the modular systems tests respectively.

Therefore, the variations in PMV calculated values are as sensitive as the meter variations. When the values measured by the meter increase, the calculated values also increase; this is the same for the decreases. The small differences can be explained by the difference in accuracy between measured and calculated values, the sensitivity of the meter, the difference between orientation selected for each method.

The main conclusion from the above results is that the calculation method compared well with the measurement method and can therefore be used with confidence. Not only the accuracy, but also the sensitivity of the calculations, are similar to the measurement method. The comparison on the third floor being satisfactory, the same results are expected to be applicable for the first floor.

7.1.2 Calculation of LPPD, PMV and PPD Indices

Now that we have established how accurate our measurements of the PMV index are, we can proceed with the calculation of LPPD, PMV and PPD indices to compare the thermal environment created by the three heating systems throughout the season. The first index we will use is the LPPD index since it is a better index to describe in general the thermal environment throughout a room. The more specific indices of PMV and PPD will be treated last to obtain more detailed information. Before the results are listed, a description of the conditions studied and the calculation procedure will be made.

Cases Studied and Calculation Method

To calculate the comfort indices, and hence to study the acceptability of the heating systems throughout the season, a series of calculations were performed based on the measured comfort parameters and some assumptions concerning the type of occupants likely to occupy the building in the future. The calculation of the PMV, PPD and LPPD comfort indices were performed using four different cases for each system; each case combining thermal comfort parameters in a different way. Below, the cases studied are described as well as the assumptions and the parameters which were used.

For each heating system, the physical and human parameters used for calculation of the comfort indices had to be estimated not only from measurements but also using certain assumptions on cases likely to appear in the field.

To be meaningful and to take into account their time variations, directly measured parameters (Ta, Tmr, Rh, Va) had to satisfy the condition of being collected simultaneously. Since all the measured parameters were not obtained simultaneously some of them needed to be fixed or assumed as will be explained later. The human parameters (M,Icl) could not be measured directly and had to be assumed; no data on the future building occupants was available. Since some of these assumptions must be made over a range in order to be realistic, sixteen different cases (four per heating system) were selected for calculation with some fixed and assumed parameters in addition to the available measured values (see Table 7.2).

Table 7.2 : Cases and Parameters Selected for Comfort Assessment

I- Measured Parameters

Ta and Tmr (Tmr calculated from surfaces temperatures and angle factors)

II-Fixed and Assumed Parameters

a) Case Dependant Parameters:

	M (W/m^2)	Icl (m^2°C/W)	PMV*	PPD* (%)
Case 1	58	0.11	-1.1	31
Case 2	58	0.11	-1.1	31
Case 3	70	0.11	-0.5	10
Case 4	70	0.11	-0.5	10

*: PMV and PPD for Ta=Tmr=21°C, Va=0.1m/s, Rh=50%.

b) System Dependant Parameters:

System	Case	Va (m/s)	Pa (KPa)
Forced	1	0.10	1.2
	2	0.20	1.2
	3	0.00	1.2
	4	0.20	1.2
Hydronic and Modular 3rd floor	1	0.10	0.7
	2	0.15	0.7
	3	0.00	0.7
	4	0.15	0.7
Modular 1st floor	1	0.10	1.2
	2	0.15	1.2
	3	0.00	1.2
	4	0.15	1.2

There are several reasons why only the data collected for Ta and Tmr are used for calculation while the other environmental parameters are fixed. The main reason is related to the fact that the other parameters (Va,Pa) were collected at different times during the tests and therefore, in any case, calculations would have to use average figures for these parameters if they were not fixed. Another reason is that general information on Va for each system and Pa for each floor was available from chapter six. Finally, setting Va and Pa to fixed values allowed to limit the comparison on comfort indices results based on temperature distribution first and always have the possibility to proceed with a more detailed investigation depending on the trends in the results thus obtained.

The values of each of the parameters shown in Table 7.2 were selected according to different criteria to obtain results as meaningful as possible and taking into account the information acquired on the behaviour of the systems. These criteria and some of the points considered in the calculations can be briefly summarized for each parameter as follows:

Ta: Thermocouple measurements at 0.6 m for grid locations type B and 0.8 m for type A. From gradients measurements, it can be assumed that temperatures at 0.8 m level are close to those of the 0.6 m level.

Tmr: Calculated from measured surface temperatures and angle factors averaged for a seated person. Average angle factors are sufficient for good approximation when exact person orientation is not fixed. Calculated angle factors appear in appendix D. Calculations showed that the angle factors were not notably different from floor to floor because of the small differences in room dimensions. Surfaces considered where walls, floor, ceiling, window and heater surface temperature (except forced air). Assumption of isothermal surfaces necessary in calculations were verified by a small experiment which showed that surfaces temperatures measured using the climate analyzer probe at more than 10 points on these surfaces varied by less than 1°C from the average temperature for each surface. Infrared scanning of the rooms also showed that surfaces were uniform in temperature.

Va: Values were selected to correspond to a range between 0.0m/s and a maximum. Based on the results of chapter six where average air velocity was found to be higher for the forced air system, maxima of 0.15m/s for the hydronic and the modular systems were selected and 0.20m/s for the forced air system. The value chosen for the modular system is higher than the results listed in chapter six since lower values of air velocity will not show a significant differences on results for thermal comfort indices and because the results of chapter six were limited to a small number of cases, higher air velocities are not totally excluded.

Pa: Values correspond to relative humidity of 50% and 30% at 21°C for the first and third floor respectively. Again, these values were chosen according to the results mentioned in chapter six showing that the first floor was generally more humid than the third floor. In any case, thermal comfort indices are not significantly influenced by relative humidity.

M: The activity levels selected correspond to "seated and resting" (58 W/m^2, 1.0 met) and "light sedentary" (70 W/m^2, 1.2 met) which can be reasonably assumed to represent the most frequent type of activities performed in an apartment. The higher level of activity of 1.2 met was selected to obtain PMV and PPD values less than ±0.5 and 10% respectively. This is to comply with ISO criteria for these indices for 21°C operative temperature which is the air temperature setting for the thermostats of the systems.

Icl: This value was fixed for all the cases to correspond to "light working clothes" (0.11 W/m^2, 0.7 clo). It was also fixed in order to limit the number of cases treated and is a compromise between selecting lighter clothing (such as summer clothing) and heavier clothing (interior winter clothing) assuming that the people prefer to wear lighter clothes at home than at work but will still adopt warmer clothes than in summer.

The above list of factors considered in the calculations is not exhaustive. Some other factors come into play, but considering the comparison with the PMV meter readings, the accuracy obtained from the calculations is sufficient for our purpose. Note that in the development of the thermal comfort indices assumptions are already present and using a comfort model will provide approximate results on people's thermal sensation in any case. The model is a useful tool in the sense that subject testing is not necessary to determine if the systems provide comfortable conditions.

Broadly speaking the combination of the above factors in the four cases selected (table 7.2) allows to compare thermal comfort provided by the heating systems mainly for two types of activities (M), two values of Va and the actual Ta and Tmr observed in the rooms. The other parameters for each cases are fixed based on the occupants (Icl) or related to the floor on which the systems are installed (Pa).

Period Considered: Normal Conditions, All Data Collected During the Season

For the cases and conditions mentioned above, we decided to calculate the thermal comfort indices for each system, using all the data collected during the heating season and using only days when conditions of "normal conditions of operation" prevailed.

The calculations of PMV, LPPD and PPD were performed by two computer programs, one which selected the days satisfying the "normal conditions of operation" and the other which summarized the results in terms of seasonal averages.

The analysis covered a large number of observations made during a number of periods of more than five hours per day. The exact number of readings and periods are listed in Table 7.3. The number of observation and periods is similar for all the systems.

Table 7.3: Statistics for Number of Observations Used in Calculations

System	Number of Periods (5h. to a day)	Number of Observations
Forced	26	1465
Hydronic	28	1411
Modular 1st	22	1341
Modular 3rd	24	1298

In this case, using the data for the whole heating season for each system provides a convenient way to define the general trend in the variations of the different comfort indices. When the results were compiled, the day to day variations were observed allowing to check for possible differences between average and instantaneous thermal conditions in the rooms or instrumentation malfunction such as thermocouple break down or data collection interruptions.

It is true that even though the number of readings are similar for all systems the outdoor conditions were not exactly the same at the time of the tests. Nevertheless, as was described in chapter four, the outdoor conditions were only slightly different for each system when the data for the whole season and conditions of "normal operation" are considered and a general trend can still has significance.

Results on LPPD Index

The results for the LPPD index calculations are reported first because this index provides the evaluation of thermal comfort which we require to (1) evaluate the thermal uniformity in the rooms, (2) make this evaluation independent of the temperature maintained by the thermostat, (3) allow to compare the systems with respect to each other and (4) takes into account the fact that there will always be a predicted minimum of 5% dissatisfied in any room. It is a general index which can be used before examining the PMV index results which are case dependant.

In this case, the description of the results is relatively simple. For all the cases selected and for all the periods considered, the LPPD index has remained between 5.0% and 6.0% for the forced air, the hydronic and the modular systems. In other words, all the systems maintained the rooms in conditions such that it would be expected that a maximum of 6.5% of the persons would feel dissatisfied thermally.

This range being small, and remembering that the calculations are subject to inaccuracies, the difference between the values calculated for each system separately is not significant. Nevertheless it can be mentioned that LPPD values observed for the forced air system were close to 5% compared to the other two systems which had values closer to the 6.0% limit.

For some periods, values slightly higher than 6.0% were found for the modular system (7.0%) on both floors but these occurred at the beginning of the season when it was difficult to select a proper setting for the control of the units since high outdoor temperatures prevailed and since the control settings on the units could not be modified continuously. When these days are discarded, the 6.0% maximum applies.

Before concluding on these results, one assumption which was incorporated in the calculation of LPPD must be emphasized. The LPPD was calculated in four steps:

Step 1: Calculation of PMV at all nine grid locations for 0.6 m level.
Step 2: Finding the room average PMV.
Step 3: Second calculation of PMV for all grid locations using the PMV values found in step 1 minus the average calculated in step 2.
Step 4: Calculating the LPPD is by taking the average of the PPD values found in step 3.

In these calculations there is the assumption that the PMV found in step 1 can in fact be raised (step 3) by the same value throughout the room (average calculated in step 2). This has to be checked further when the values of PMV and PPD at different locations in the rooms are discussed.

If we recall that Fanger qualified his recommendation of a maximum of 6% for LPPD as a "severe" demand [3], then we can state that the systems performed very well in terms of thermal uniformity. Also, by extending the analysis to a large number of days with different outdoor conditions we fulfilled another of Fanger's recommendations which was to judge the thermal environment not only as a function of the heating systems but also as a function of the outdoor climate which influences the thermal field in the rooms.

Results on PMV and PPD Indices: Room Averages

As mentioned in the previous section, the results for LPPD indices have to be further inspected using the individual results for the PMV and PPD indices in order to describe the rooms' thermal uniformity. In the following, we will discuss results obtained from these two indices with respect to the cases selected and in terms of room averages. The results for the different systems will be listed and compared.

For the different cases listed in Tables 7.2 and 7.3, the results for PMV and PPD calculations appear in Table 7.4 where the room seasonal averages are shown.

For all the systems, cases 3 and 4 were the ones for which the thermal comfort parameters combined in the best way as represented by the corresponding PMV and PPD values. Recall that cases 3 and 4 had the same activity level and clothing insulation values (1.2 met, 0.7 clo) but case 4 had air velocity values higher. Consequently higher PPD values were observed for case 4 than for case 3. Cases 1 and 2 were set at the lowest activity level of 1.0 met and the lower PMV and higher PPD was expected.

Table 7.4: Room PMV and PPD Averages

	PMV				PPD (%)			
	cases							
SYSTEMS	1	2	3	4	1	2	3	4
Forced	-0.9	-1.1	-0.2	-0.5	20	32	6	10
Hydronic	-1.5	-1.7	-0.8	-0.9	50	60	17	24
Mod. 1st	-1.1	-1.3	-0.5	-0.6	30	39	9	13
Mod. 3rd	-1.3	-1.4	-0.6	-0.7	37	46	12	17

Results for the average value of the comfort indices reflect the values already observed in chapter four for room air temperatures maintained. As expected from the distribution of all cases and all the systems, PMV values were found to be below zero, with thermal sensation predicted to be between slightly cool (PMV=-1) and cold (PMV=-2) and PPD ranging from 9% to 60%. Recall that in chapter four, the average room temperatures observed for the hydronic system and the modular system on the third floor were lower than for the other two systems. This explains why these systems performed less well on the PMV and PPD indices. In this sense, the results would have been better if the thermostat for these systems had been set higher.

Therefore, to be meaningful, the value of the PMV and PPD indices must be discussed by considering the relative changes observed between the different cases and assuming that all four cases could occur during the season. It can be assumed that the occupants would ideally expect their systems to provide comfort consistently throughout the season even when the activity they performed are modified.

It can be supposed that at a given time, the occupants's activity and clothing correspond exactly to the description of one of the four cases selected. For all systems, the conditions would become more uncomfortable if the conditions are changed from one case to another. The change would be more important if activity levels are changed (going from case 2 to 3).

For all the systems, if activity level and clothing insulation parameters are fixed, the change in air velocity will cause the comfort of the occupants to be changed significantly and in different proportions for each system.

The range of air velocity observed for the forced air system was higher than for the other systems (maximum of 0.20m/s compared to 0 15m/s) and these higher values were fixed in the calculations. The PPD changes for the forced air system would therefore be more important than for the other systems due to the higher air velocities fluctuations.

Now, if the activity level is varied from 1 2 to 1.0 met then, for all systems, the new conditions will not be satisfactory For all the systems this type of change in activity level will cause the number of persons dissatisfied to be between two times and five times higher if the activity is lowered at some point in time

Knowing the high dependency of the PMV and PPD indices on air temperatures, this indicates that the systems maintain conditions which are correct for comfort within a small range and if activity level or another parameter is modified, changes in thermostat settings are needed to satisfy the new conditions. This signifies that air temperatures being fairly constant for the periods considered (see chapter 4) and for each system, the environment created was suited for very specific conditions

If air temperature had varied more, then the difference between PPD obtained with one set of parameters and another set would be less important In time, the low air temperatures would satisfy one case and the higher temperatures other cases However, it is preferable that the systems satisfy very specific cases knowing that changes in the condition of the occupants are expected to occur only a few times during the day The best would be to have the thermostat linked to know, assume or anticipate these changes.

In this section, we have shown the results for PMV and PPD indices which were averaged values of all room locations. As was expected the systems maintained average conditions more suited for light sedentary activities (cases 3-4) than for the lower activity cases (1-2).

The comparison of the results for each system using Table 7.5 showed that the values selected from chapter six for the range of air velocity for the forced air system introduced significant changes in PPD for a fixed activity level than the other systems.

Finally, it was discussed that comfort conditions maintained by each systems in the rooms were not flexible enough to allow changes in activity levels (1.2 to 1.0 met) without the occupants having to change the thermostat setting. In the next section, the particular values of PMV and PPD at different locations in the rooms will be discussed.

Results on PMV and PPD Indices: Room Variations

In this section we will provide further details on the uniformity of thermal distribution in the rooms based on the spatial and time variations of PMV and the corresponding PPD indices for each system. Using the data for the whole season will enable us to identify how the rooms were maintained from a comfort point of view and decide if, for example a location is always colder or warmer than another or if comfort conditions vary more for one system than for another.

The room distribution and range of PMV and PPD values shown below are based on the steps for calculating LPPD mentioned earlier. For all the systems, values of PMV and PPD were calculated using the following steps:

Step 1: Calculation of PMV for a period at all nine grid locations
Step 2: Average PMV is found for period of data collection

Step 3: Calculation of a corrected PMV, PMV_{∞}, for all grid locations using the PMV values found in step 1 minus the PMV average calculated in step 2.

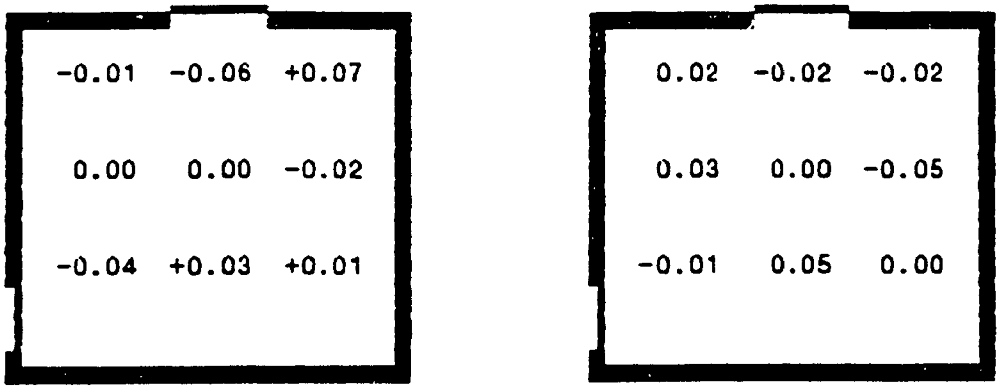
Using this approach allows us to present the results only in terms of the variations of PMV for each location with respect to room average.

Results for PMV and PPD calculated in this manner for the different cases were similar for a given system with differences less than 0.02 PMV between cases For example, the average PMV_{∞}, at location B1 for the forced air were calculated as -0.07, -0.08, -0.06 and -0.06 for cases 1 to 4 respectively These differences being small, it was decided to present results for case 3 only or for the conditions when all systems performed the best

As you will note, the results from PMV and PPD calculations using the method mentioned above resulted in values close to zero PMV and 5% PPD, the thermal environment for all systems is close to neutral and only small differences were observed from location to location

Because room variations are small for a given system, the results of calculations for room PMV, PPD, period averages and maximum PMV variations are presented in Figures 7.3 to 7.6 for all four systems

As can be seen, from the results, all four systems provided environments which are fairly uniform throughout the rooms The results for PMV_{∞} are all between ± 0.2 PMV and PPD is less than 6% The room variations were less than 0.40 PMV for all systems and maximum variations were less than 0.80 PMV for all the systems for the periods considered

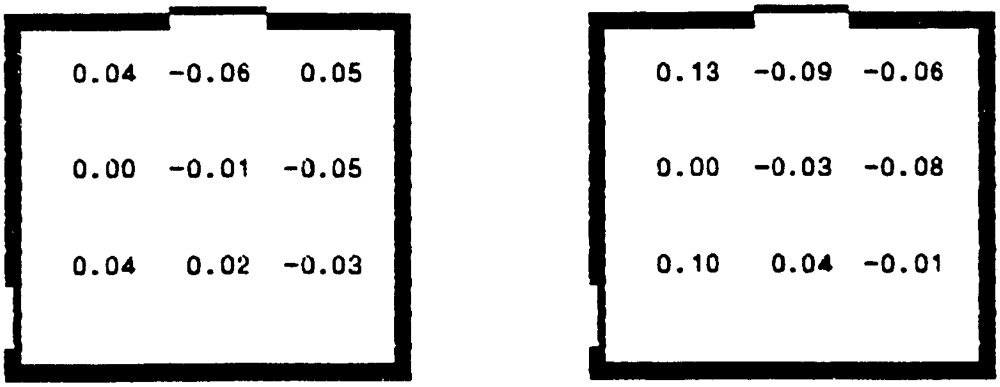


-0.01	-0.06	+0.07
0.00	0.00	-0.02
-0.04	+0.03	+0.01

a) Forced air

0.02	-0.02	-0.02
0.03	0.00	-0.05
-0.01	0.05	0.00

b) Hydronic



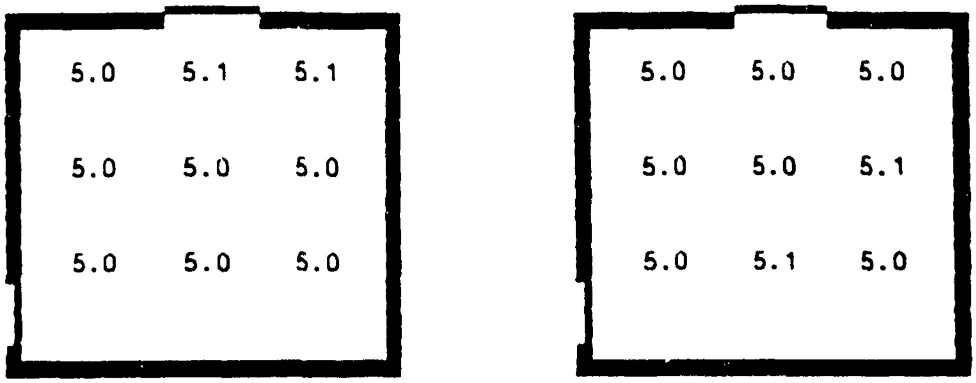
0.04	-0.06	0.05
0.00	-0.01	-0.05
0.04	0.02	-0.03

c) Modular 1st

0.13	-0.09	-0.06
0.00	-0.03	-0.08
0.10	0.04	-0.01

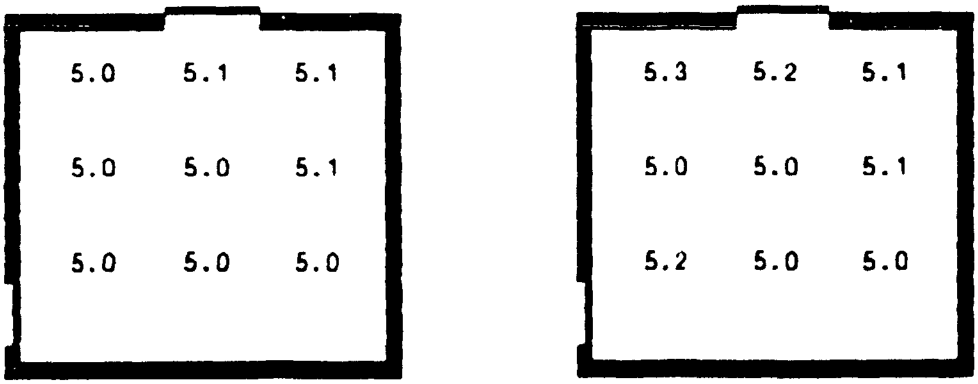
d) Modular 3rd

Figure 7.3: PMV_{cor} Room Distributions, Seasonal Average



a) Forced air

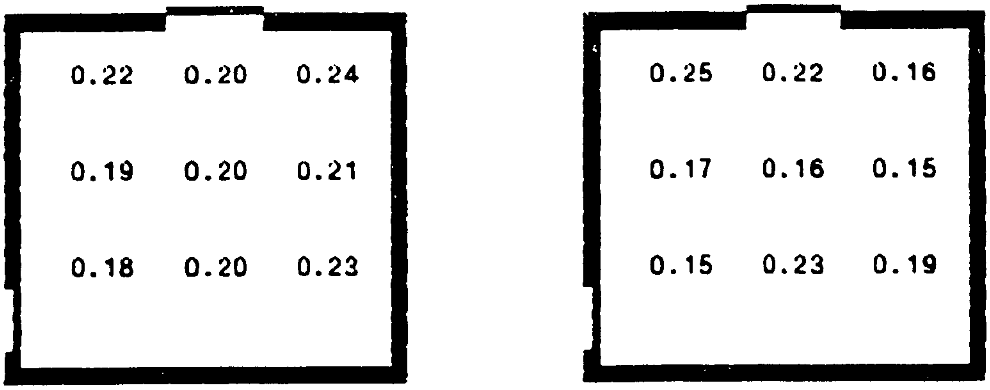
b) Hydronic



c) Modular 1st

d) Modular 3rd

Figure 7.4: PPD$_m$ distributions, Seasonal Average



0.22	0.20	0.24
0.19	0.20	0.21
0.18	0.20	0.23

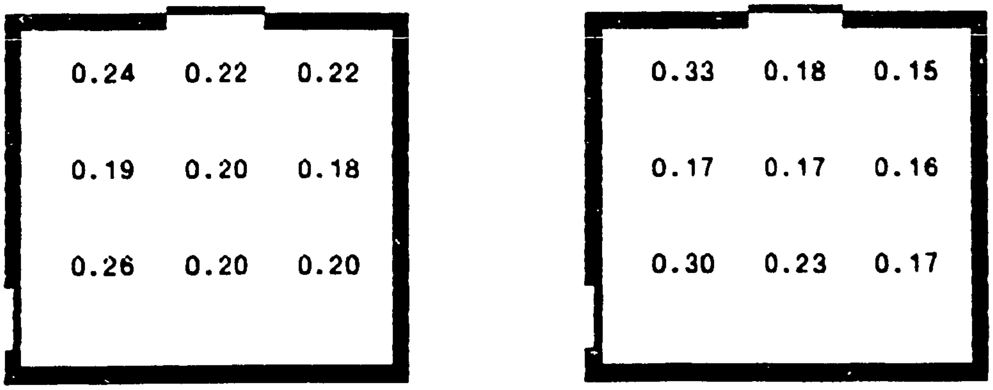
room average= 0.21

0.25	0.22	0.16
0.17	0.16	0.15
0.15	0.23	0.19

room average= 0.19

a) Forced air

b) Hydronic



0.24	0.22	0.22
0.19	0.20	0.18
0.26	0.20	0.20

room average= 0.21

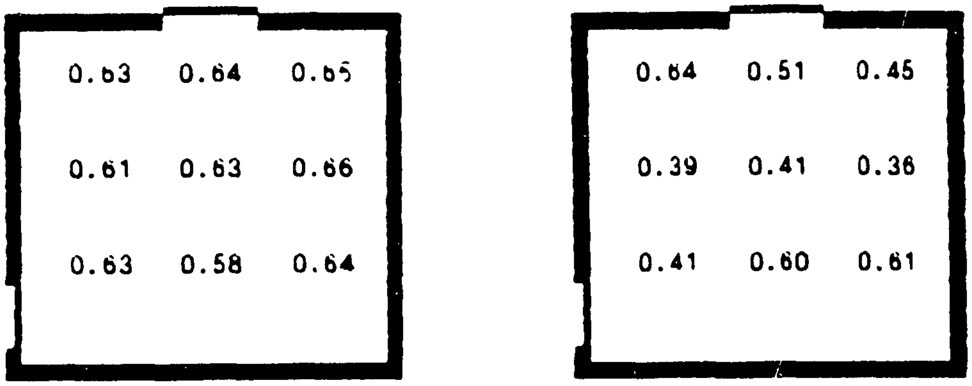
0.33	0.18	0.15
0.17	0.17	0.16
0.30	0.23	0.17

room average= 0.21

c) Modular 1st

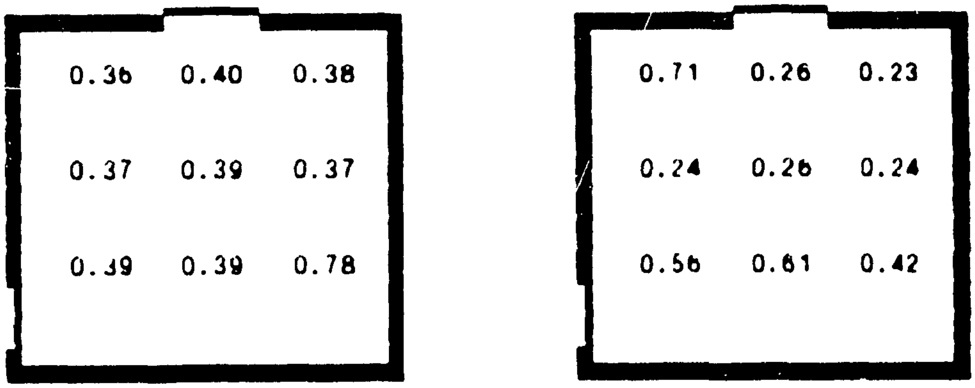
d) Modular 3rd

Figure 7.5: PMV_{av} Average Variations Per Period, Seasonal Average



a) Forced air

b) Hydronic



c) Modular 1st

d) Modular 3rd

Figure 7.6: PMV_{w} Maximum Variations per Period, Seasonal Average

Results obtained for all the systems indicate small variations both in space and in time and it is difficult to actually point to any major differences between the environment created as rated on the PMV index. This is also further justified since the accuracy of the measurements may have been responsible for the differences, the values calculated being so similar for a given system or if systems are compared together.

From Figure 7.3, the PMV at the location facing the window is negative for all three systems, indicating that this zone is expected to be slightly colder than the rest of the room This is due to the colder temperatures (air and radiant) created by the window. The compensation from the hydronic system is slightly better than for the other systems with a PMV of -0 02 compared to -0.06, -0.06 and -0.09 for the forced air, the modular system on the first floor and the modular system on the third floor respectively.

Let us consider the room distribution of PMV for the forced air system It can be seen from Figure 7.3 that PMV varied by 0 11 (0.07 to -0.04) for the locations in the room. The general trend is to have the warmer locations at the lower right and in the upper right corner of the room The center location is at zero PMV.

For the hydronic system, the room distribution of PMV is spread over a range of 0 10 PMV (-0.05 to 0.05) and colder points are spread on either sides of a diagonal going from the upper left side to the lower right side of the room. The PMV at the center of the room is at zero.

The room distributions for the modular systems are similar on both floors notwithstanding a few exceptions. The distribution is characterized on both floors by the cooler locations located on the right side of the room and has 0.11 range (-0.06 to 0.05) on the first floor and a 0.22 range (-0.09 to 0.13) on the third. A higher PMV is found at the location close to the heater (+0.13) on the third floor which is not so important on the first floor. This is due to the fact that the modular system on the third floor was closer to the thermocouple grid at that location than on the first floor.

The PPD distributions for all the systems, (Figure 7.4) show that values greater than 5.3% were not observed Therefore, the thermal acceptability of the room can be considered to be fairly good for all the systems based on the measurements. The higher PMV values at the different locations are the reflection of the PMV values shown in Figure 7 3 For all the systems, the PPD at the center of the room is 5 0%, the smallest value possible.

The average PMV and PPD values throughout the room describe the general thermal comfort expected to be found in the room, the average and maximum variation of PMV help to determine where and by how much the thermal sensations are expected to fluctuate in time or over a period such as the ones we observed in our measurements

For the forced air system, the average and maximum room variations of PMV are similar for all points, indicating again the effect of the forced air system which is to distribute heat uniformly throughout the room The average PMV point variations observed were less than 0 25 PMV for all the periods considered and the maximum variations were less than 0 66 PMV The average room variation was at 0 21 PMV Notice how the local average and maximum variations in the rooms are close to each other

For the hydronic system, the variations in PMV were in general slightly higher close to the heater than for the back locations The average PMV point variations observed were less than 0 25 PMV for all the periods considered and the maximum variations were less than 0 64 PMV The room average variation was at 0 19 PMV The average variations observed close to the heater (front of window) were approximately 1 4 times higher than the locations at the center of the room (0 22 PMV compared to 0 16 at the center) For one location, the back center location, variations observed were higher than the front location This can be explained by the effect of the enclosure at the back wall since similar variations were observed for the modular system on the third floor

The variations in PMV observed for modular systems on both floors were on the average similar for the room average (0 21 PMV in both cases) but differed in terms of points measurements. The greatest difference is again for the location close to the heater where average and maximum variations were higher on the third floor (average=0.33, maximum=0.64) than on the first floor (average= 0.24, maximum=0.36). This is due to the location of the modular system unit located closer to the measurements location on the third floor.

For all the systems, the maximum variations in PMV being small over a period, if we consider the corresponding changes in PPD. A person in the room would not be expected to feel drastic changes in his or her thermal sensation. Taking the maximum change observed for all the systems at the center of the room, which is 0.63 PMV (forced air), then, assuming that a person is at the neutral state of 0.0 PMV then his or her thermal sensation would be expected to vary at most from neutral to slightly cool.

Furthermore, this maximum range is actually calculated based on variations occurring about a zero or neutral value and therefore includes positive and negative ranges. Therefore, the maximum variation of 0.63 PMV should be divided by two (0.32 PMV) and the thermal sensations would vary at the most between -0.32 to +0.32 PMV (between slightly cold and slightly warm)

In the end, it can be concluded from measurements of PMV and PPD that all systems can maintain thermally comfortable environments with only slight differences from the point of view of distribution and variations in these indices. The forced air system, just as was the case for temperature distribution, produced a fairly uniform environment with small spatial variations throughout the room but high variations in time. The hydronic system also produces a slightly better compensation at the front of the window, with a spatial distribution less uniform than for the forced air system but with smaller time variations. In terms of spatial distribution, the modular system was found to create a zone slightly colder on the side of the room opposite to where the system was installed and created a significantly warmer zone close to the heater. Time fluctuations for the modular system were observed to be similar to the hydronic system. Some of these comments are contained in Table 7.6.

Table 7.5: Comparison of Systems for PMV and PPD Indices

	Forced	Hydronic	Modular
PPD at center	5.0	5.0	5.0
PMV room range	0.11	0.10	0.22
Avg PMV period change	0.21	0 19	0.21
Max PMV period change	0 66	0 64	0.71
Warmer zone	none	none	heater
Colder zone	window	window, neighbour side	window neighbour side

7 1 3 Summary

This section has dealt with the calculation and measurement of the PMV, PPD and LPPD indices for the three types of heating systems studied and the calculations have covered all the data available collected for the heating season when the systems were in normal operation

In the first section we showed how the measurement and calculation method compared in order to establish the sensitivity and accuracy of our calculation method The results from this section showed that our calculation method produced results which were in good agreement with the measurement technique available in terms of accuracy and sensitivity

In the second section, our calculations of the LPPD, PMV and PPD indices showed that all the systems produced fairly good environments in terms of global thermal comfort with only small differences between the systems The results did not show any major problems which could occur from the point of view of comfort if the center of the rooms is considered Small differences pointed to the fact that the forced air system introduced higher variations in the environment, while the distribution created by the modular system may be problematic close to the heater. For the hydronic system the general room distribution and ranges of variations were acceptable.

7.2 Local Discomfort

The possible problems of local discomfort are listed in Table 7.7 where each systems are checked for: asymmetric thermal radiation, draught, vertical air temperature differences, cold floors, temperature drifts and cycling.

The problems of cold floors, draught and radiant temperature asymmetry did not appear in the observed conditions for several reasons For heating, these problems are likely to occur in rooms with low insulation levels and high infiltration rates to either cause cold floors, cold air currents or low surface temperatures and this was not the case for the rooms studied Second, these problems are usually eliminated when the heaters are able to maintain desired indoor temperatures within a small range and with only a small drift as was the case here Finally, the boundary conditions of the interior surfaces were such that gains from other floors or the neighbouring building were sufficient to keep them high as seen in the transient response case study.

For air velocities, only the forced air system created peak values higher (0 23m/s) than the standard requirements of 0.15m/s but only for the location close to the supply and 0 10m/s peaks were found elsewhere in the room. The system is within standard requirements for the central locations in the room but the location close to the supply may be a problem Note that the measurements close to the supply were made approximately 40cm away from the exterior wall and the standard consider the occupied zone 60cm away from the walls. Hence, it would be necessary to measure beyond the 60cm zone to verify if the standards requirements are fulfilled. If the results for peaks at location B1 (0.23m/s) and B3 (0.10m/s) are interpolated, the zone close 60cm away from the supply should encounter peak values close to 0.20m/s which is higher than standard.

Only for the hydronic system did vertical temperature differences go over the standards' limits since for the other systems, the gradients remained small, (forced air) or occurred higher than 1.2m (modular system). The baseboard of the hydronic system caused increases close or higher than 3°C/m between the 0 1m and 0 6m levels in some of the cases discussed in chapter 5. However, this problem was not observed for locations away from the heater where gradients were well below 3°C/m and were less than the forced air or the modular systems' values

From results obtained earlier, the gradients of the hydronic system are not only high close to the baseboard but they vary in shape according to the cycling of the system and therefore these variations may cause more discomfort than if they had been constant and some permanent adjustments could be made by the occupants

From the table 7 7, of local discomfort, the major problem for all the systems is mainly the rate of temperature changes which are either exceeding or close to the standards requirements The rates observed for the modular and the forced air systems (chapter 4) were higher than the limit of 2 2°C/h but the peak to peak variations were less than 1 1°C and therefore, the combination of rate and time changes observed were within standards requirement The hydronic system performed less well for the location close to the heater where rates of up to 10°C/h were observed for peaks higher than 1 1°C For other locations the hydronic system performed as well as the other systems

The present review showed that the rates of temperature changes and the temperature gradients were the two main local discomfort problems which could be encountered in the space The forced air and modular systems created temperature fluctuations which were close to the standards requirements in the center of the room while the hydronic system produced both high gradients and high temperature fluctuations for the location close to the baseboard.

For the forced air system, values of air velocity close to the supply were found to exceed standards requirements for the peak values observed but were acceptable for the center of the room.

Other problems such as cold floors, asymmetric thermal radiation and draught were not encountered due to the relatively good properties of the enclosure and the low heaters' surfaces temperatures. It is expected, however, that for rooms with greater window area and lower insulation levels that radiant temperature asymmetry and draught could cause more serious problems. For a zone located under the floor of the rooms which is maintained at high temperatures cold floors problem may never be present.

Table 7.6: List of local Discomfort Problems for Each System

	FORCED AIR	HYDRONIC	MODULAR
ASYMMETRIC THERMAL RADIATION	not a problem: surfaces temperatures close to air temperatures, ,heater's surface temperature (hydronic and modular) not high enough, window small in size and well insulated		
DRAUGHT	not a problem: air velocity less than 0.20m/s and no major masses of cold air coming into the room		
AIR VELOCITY (standard: less than 0.15m/s)	peak values close to 0.20m/s, average near 0.10m/s, exceeded close to supply only	peak values less than 0.10m/s but assumed higher close to the baseboard	peaks less than 0.05m/s
VERTICAL TEMPERATURE DIFFERENCES (standard: less than 3°C between 0.1m and 1.1m levels)	far from exceeded	serious problems close to the heater, correct elsewhere	close to limits but only at level higher than 1.2m
COLD FLOORS	not a problem: due to boundary conditions, floor temperatures were only slightly lower than air temperatures		
TEMPERATURE DRIFTS	not a problem: systems were able to maintain temperatures which varied by less than 1.5°C		
CYCLING (standard: for peak to peak variations higher than 1.1°C, rates<2.2°C/h	peak to peak variations less than 1.1°C but rates of 6.3°C/h observed for 0.8°C peak changes at center	problem close to baseboard where rates of 10°C/h observed for peak changes of 1.3°C at the 0.8m level, elsewhere rates up to 3.1°C/h but peaks less than 0.8°C	peak variations less than 1.1°C but rates up to 4.7°C/h observed

7.3 Summary

The three systems produced room conditions for which the LPPD indices was close to 5% which is the minimum predicted percentage of dissatisfied possible. Therefore, for global comfort, all the systems performed satisfactorily.

For local discomfort, some problems were identified for all three systems. For the forced air and the modular systems, high rates of temperature changes were found to be near the standards' limits while for the hydronic system, problems of high temperature gradients and high rates of temperature changes were found. The forced air created air velocity above standards requirements for the region close to the supply. For all three systems no local discomfort problems due to cold floors, asymmetric thermal radiation or draught were identified due to the relatively well insulated rooms and no major outdoor air infiltration. The surface temperatures of the hydronic and modular systems did not cause temperatures high enough to cause radiation problems.

With this chapter completed, a final conclusions can be made along with some recommendations with respect to thermal comfort and heat distribution created by the three different heating systems.

CHAPTER 8

8.SUMMARY AND RECOMMENDATIONS

In this final chapter, the results obtained from the tests are reviewed to conclude on thermal comfort produced by the systems (local discomfort and global comfort), the heat distributions they created, the control aspect and the improvements which could be made. Finally, a general summary is made. Numerical results reported earlier are not included here to obtain a more general appreciation of the systems and the reader can refer to earlier chapters for more precise information.

8.1 Thermal Comfort

All three systems were able to provide distributions sufficiently uniform to maintain comfort throughout the room based on the results on global thermal comfort but particular problems of local discomfort were found for each systems.

Global Comfort

The LPPD index indicated clearly that the conditions maintained globally throughout the rooms by the systems were such that less than 6% of the people would have felt dissatisfied throughout the season studied. The LPPD indices, calculated for the seasonal distribution described in section 4.2, remained between 5% and 6% for all three systems.

As described in chapter 3, 5% dissatisfied is the best which can be expected from any system due to the variable nature of human response to thermal environment and therefore the systems performed very well with regards to this minimum. Also, with the LPPD index calculated for a wide range of outdoor conditions the procedure satisfies the recommendations of Fanger for thermal comfort assessment.

The results for PMV and PPD indices showed that all the systems maintained room conditions which were acceptable for specific cases of human occupancy. It was found that comfort can be provided for specific types of activity and clothing but when these two parameters are modified, the environments were not suited any more for the new conditions and adjustments had to be made. In other words, the environments would not satisfy at the same time a number of people with different types of clothing and activity levels. The percentage of dissatisfied would be greater in this case.

Local Discomfort

Local discomfort problems were specific to each system and can be discussed separately. Also, because of the room's envelope properties or the system's surface temperature, problems of cold floors and draught were not present.

The forced air system created local discomfort problems mainly related to high time variations in air temperatures in the room and high air velocities close to the supply Temperature variations remained on the brink of exceeding standards' limitations with high observed rates of change but peak to peak variations sufficiently low to avoid discomfort. Peak air velocities exceeded standards limits close to the supply but averages did not.

The hydronic system created local discomfort problems close to the baseboard in terms of high temperature gradients and high temperature changes but fulfilled standards requirements elsewhere. In addition to exceeding the standards limitations, since the local discomfort problems vary with the system's cycling, the users of this system may be further dissatisfied by the seasonal variations which may prevent them from finding one type of adjustment to cope with local discomfort problems.

For the modular system, only high rates of temperature changes were found for local discomfort. The rates observed were however small compared to those of the forced air system. High gradients were observed for this system but these remained important only close to the ceiling

For asymmetric thermal radiation, no major problems were observed This is due to a relatively small and well insulated window, and because the surface temperature of the systems (hydronic and modular) were close to values for the rest of the enclosure

Draught problems were not present since no major source of cold air entered the room even though high velocities (responsible for draught) were found with the forced air system

For local discomfort, the modular system appears to be the best of the three systems since the other two have more than one local discomfort problem associated to them Note however that the "seasonal" outdoor temperature distribution available for the modular systems was on the average warmer than for the other two systems

8.2 Temperature Distribution and Gradients

The forced air system created the most uniform temperature distribution of all the three systems. The others either produced high local temperature differences (hydronic baseboard) or sent most of the heat to the ceiling region (modular).

The forced air system created small temperature variations in all directions of the room due to the mechanical forcing employed. The mechanical distribution caused the air to be well mixed into the room, and, as was seen for transient warming, the temperature at all points in the room followed more or less the same variations in time.

The hydronic baseboard created high gradients and a concentrated mass of warm air which varied according to cycling operation and tended to divide the room in two major thermal zones. The first thermal zone was at the front of the baseboard where high negative gradients were observed at mild outdoor temperatures, and high positive gradients occurred at cold outdoor temperatures. The distribution for the rest of the room was excellent with low gradients and uniform temperatures throughout.

For transient warming, the localized distribution of the hydronic system was dominant and the room was found to warm up at rates many times higher close to the baseboard. This would mean that occupants would have to be uncomfortable during most of the warming period due to high temperature fluctuations before comfort throughout the room is achieved.

The modular system had a poor distribution performance as far as air temperature gradients were concerned since it heated the region between 1.2m and the ceiling. In other words, the air temperature above this level was warmed before the occupied level at heights between floor and 1.1m. The problem is caused by the location of the system on the side of the window which is not a suitable location to compensate for the cold window air coming from the window-all the heat goes directly to the ceiling. The problem was found to increase as outdoor air temperatures were lower, based on the results on the distribution of gradients throughout the season.

In transient warming the modular system's fast performance was offset by the poor distribution: the room is heated from the back to the front. The warm air has to travel all along the ceiling before it is brought back down.

As can be seen, the comparison of the systems for global comfort made earlier did not explicitly reveal the points mentioned here on heat distribution. In this sense, the systems' performance for local discomfort and heat distribution are more relevant concerns.

8.3 Control Aspects

The hydronic system maintained more constant temperature in the comfort room over the season than the forced air and the modular systems. The lesser performance of the forced air and modular systems on the first floor can be attributed to the difference in room thermal properties compared to the third floor. The performance of the modular system was influenced by the fact that there was not someone at the house to correctly modify the control position to find the optimum setting.

For control purposes, the hydronic and modular systems had a great advantage over the forced air system because they were controlled by individual zones using a control located directly on the system. These controls were the dials on the baseboard and at the front of the modular system. Similar control is not possible with the forced air system without causing balancing problems in the system

The control of the modular system was difficult to use and it performed less well in moderate outdoor conditions typical of the beginning of the season-the correct setting had to be established when the cooling load was high enough for the system to start cycling.

The control aspect certainly deserves more attention when the seasonal performance of the systems is concerned. For example, factors such as the people living in the building, unbalances between different zones, the presence of heat sources and other factors may modify the response of the systems as a whole.

8.4 Suggested Improvements for Each Systems

The problems related to each system can be solved up to a certain extent by improving their distribution mechanism or their cycling behaviour and using some simple means. A list of suggested improvements is therefore made here and could be used to solve some of the problems found.

It must be emphasized that the following actions suggested will create new conditions. These new conditions may create environments with properties different from the ones observed here.

Forced Air System:

-High temperature fluctuations can be solved by modifying the on-off cycles in two ways. The first and more expensive way is to create a progressive beginning for the on cycle which is smoother (longer and less sharp) and sustained longer afterwards using a variable fan speed control modulated by an intelligent controller. The other solution is to maintain a constant minimum supply of heat which could be increased for higher demands.

-Using a longer and wider diffuser would reduce face air velocity and reduce the room air velocities. This would affect temperature distribution and the supply grilles would have to be redesigned if the changes are important.

Hydronic

-Local discomfort due to high gradients and high temperature fluctuations close to the baseboard can be reduced by changing the design of the baseboard in two ways. It can be constructed longer, and with horizontal and vertical louvers diffusing the heat to the sides to reduce heat concentration (louvers could be similar to the ones available for unitary cooling units used in rooms). A second solution would be to include a fan to the baseboard, but in this case louvers would also be required and the cost may be prohibitive.

- The cycling temperature of the hydronic system should be maintained close to a fixed average value instead of being allowed to change. Otherwise the heat distribution characteristics of the baseboard will vary during the season. Maintaining an average cycling temperature should be performed by the boiler which could be controlled to maintain minimum flow during the season irrespective of the loads. It is important to have a system which cycles in more or less the same way at different loads since the occupants can better adapt to the local discomfort problems if they are constant.

Modular System

The main problem with this system is that the air leaving is not sent to the occupied zone but goes to the ceiling. This problem cannot be improved by putting the device under the window due to code limitations and therefore the distribution must be modified in some way. Some improvements would include:

-Covering the top grille of the system with a piece of metal at a 45° angle.

-Changing the location of the grille from the top to the side facing to window or on both sides of the units. This would influence room heat distribution, and, for installation, two types of models (left or right side distribution) would be required.

-Reducing the high difference between air temperature delivered and room air temperature in order to reduce the upward rise due to buoyancy effect on the air leaving the top. This may be performed by using a more conductive front cover, maintained at higher temperature, and thus increase the radiation component.

-Keeping the original design a radiator instead of a convector although the efficiency of radiation distribution is doubtful if the size of the heating body is the same. Human angle factors for the modular are negligible (less than 2% of total at the center) and higher surface temperatures would have to be allowed.

8.5 General Summary

Relatively speaking, the results for thermal comfort and heat distribution obtained indicate that the forced air system produced the best environment in the rooms. This system and the other systems are far from perfect when local discomfort and distribution mechanism are studied further.

However, local discomfort problems are not so important that general room comfort (global comfort) is seriously jeopardized and improvements and modifications can be made to correct most of the problems found.

The results from laboratory and field studies mentioned in chapter 2 also came to the conclusion that all systems performed well from the point of view of global comfort. Local discomfort problems were related to high heat concentration close to the distribution units or high temperature gradients.

In comparison to previous studies, data for a large number of days during the season were used for actual field conditions. This approach was useful to probe further the influence and the trends in the systems' operation for a cold climate and for transient conditions. For example, trends in gradients fluctuations and room temperatures maintained for a large number of days were possible to be observed. Also, the room selected in the building were such that results for systems could be successfully compared between floors, and for similar outdoor temperature variations.

We must also note that a large amount of information gathered is still available from the data collected. It can still be retrieved to be used as reference for actual building design or simulation. For example, typical outdoor wall and window surfaces temperature values are available for a large number of outdoor conditions.

Field studies of this kind could be repeated and improved in order to increase general knowledge on heating systems' operation. For example, the systems modifications suggested above should be applied and tested to complete the research cycle necessary to produce improved products for manufacturers and users of these systems. The modifications should be verified for different buildings and different climates. In another way, the detailed observation of air temperature distribution and variations leads to an understanding of the air movements in a room and this understanding can finally bring accurate models and new techniques for design and analysis with important returns needed to ameliorate comfort and energy consumption.

It is our opinion that this research project has required techniques used to collect and process information which will be routine operations in the near future. In this sense, thermal comfort and comfort in general should always be a prime consideration in all aspects of building design or improvement--buildings are for people. Past and future research in this field will contribute to the advancement required to ameliorate building environments which we feel was often a neglected preoccupation up to now mainly due to the limited amount of information available.

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Appendix A (U-Value Calculations)

Below are listed the envelope property calculations performed according to ASHRAE Fundamentals. Material properties used are from ASHRAE tables or from manufacturer specifications and renovation plans. The dR values indicates the variations found in estimating resistances values for ceratin materials or arrangements.

A) List of Materials Used

Layer No.	Layer Name	Abbre-viation	R value (W/m2 °C)	dR (W/m2°C)
1	Outside film	h	0.030	0.0
2	Wall inside film	h W	0.120	0.0
3	Horiz. surf. film	h F	0.110	0.0
4	Brick	B	0.125	0.0
5	38X89 studs	S1	0.765	0.080
6	Wall insulation	IW	2.460	0.123
7	Roof insulation	IR	5.280	0.264
8	Vapour Barrier	VB	0.000	0.0
9	Gypsum (13mm)	G3	0.079	0.0
10	Gypsum (16mm)	G6	0.099	0.0
11	Air space (39)	A1	0.180	0.0
12	Wood blocks	WB	0.411	0.0
13	Black Paper	P	0.100	0.0
14	Door	D	0.300	0.0
15	New floor tiles	T	0.470	0.030
16	New underfloor	NU	0.140	0.030
17	Old underfloor	OU	0.297	0.060
18	Wood planks	WP	0.347	0.070
19	Floor joist 11"	FJ	2.610	0.100
20	Roof Membrane	RM	0 058	0.003
21	Impregnated Board	B	0.230	0.080
22	Air space (90)	A2	0.160	0.010
23	Roof joists (2X8")	RJ	1.760	0.0
24	Air Space (400)	A3	0.160	0.010
25	Window glass only	SS	0.330	0.0
26	Window Shade	SD	0.048	0.0
27	Window (overall)	ST	0.313	0.020

B) Walls

Exterior Wall		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
1	h	0.030	0.030		
4	B	0.125	0.125		
5	S1	0.765	-----	0.080	-----
6	IW	-----	2.46	-----	0.123
8	VP	0.000	0.000		
9	G3	0.079	0.079		
2	h W	0.120	0.120		
		1.120	2.814	0.08	0.123
		U=0.409		dU=0.040	

Street Facade		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
1	h	0.030	0.030		
4	B	0.125	0.125		
11	A	0.180	0.180		
12	WB	0.411	0.411		
13	P	0.100	0.100		
5	S1	0.765	-----	0.080	-----
6	IW	-----	2.460	-----	0.123
8	VP	0.000	0.000		
9	G3	0.079	0.079		
2	h W	0.120	0.120		
		1.810	3.51	0.080	0.123
		U=0.311		dU=0.027	

Partition inside		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
2	h W	0.030	0.030		
9	G3	0.079	0.079		
5	S1	0.765	-----	0.080	-----
22	A2	-----	0.160	-----	0.010
9	G3	0.079	0.079		
2	h W	0.030	0.030		
		0.983	0.378	0.080	0.010
		U=2.480		dU=0.193	

Neighbour Wall		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
2	h W	0.030	0.030		
9	G3	0.079	0.079		
5	S1	0.765	-----	0.080	-----
4	A1	-----	0.180		
4	B	0.250	0.250		
5	S1	0.765	-----	0.080	-----
9	G3	0.079	0.079		
2	h W	0.030	0.030		
		2.000	0.628	0.160	-----
		U=0.688		dU=0.038	

C) Other Components: Door, Window, Floor

Door		R(m2 C/W) overall	
No.	Name		
2	h W	0.030	R = 0.360
14	D	0.300	U = 2.78
2	h W	0.030	dU= 0.139

Window (shade in & out)		R(m2 C/W) overall	
No.	Name		
26	SD	0.048	U = 1.670
25	SS	0.330	dU= 0.202
26	SD	0.048	

Window (shade in & out)		R(m2 C/W) overall	
No.	Name		
26	SD	0.048	U = 1.885
25	SS	0.330	dU= 0.220
Only		Glass U=2.27, dU=0.11	

Floor 1st floor		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
31	h F		0.110		
15	T		0.470		0.030
16	NU	0.140		0.030	
18	WP	0.347		0.070	
17	OU	0.297		0.060	
19	FJ	-----	-----	-----	-----
3	h F	0.110			
		-----	1.474	-----	0.190
		U=1.474		dU=0.070	

Flooring (2-3)		R (m2 C/W)		dR (m2 C/W)	
No.	Name	at framing	betwe.	at framing	betwe.
3	h F	0.110	0.110		
15	T	0.470	0.470	0.03	0.03
16	NU	0.140	0.140	0.03	0.03
18	WP	0.347	0.347	0.07	0.07
17	OU	0.297	0.297	0.06	0.06
19	FJ	2.610	-----	0.10	----
22	A2	-----	0.160	----	0.01
10	G6	0.099	0.099		
3	h F	0.110	0.110		
		4.183	1.573	0.290	0.281
		U=0.597		dU=0.118	

Appendix B (Programs Used for Analysis)

There were four major programs created to analyze data:

1-General Data Analysis
2-Batch Processing and Weather Data Manipulation
3-PMV Calculations
4-Temperature Profiles Analysis

The second program (batch processing) was responsible to retrieve data from the database to create data files with a format accessible for the other programs. It used a list of all tests periods entered into a calendar file corresponding to each system.

The general data analysis program could perform mathematical operations using a mapping of the location of all the thermocouples and the other instruments used. It could accumulate results to perform data analysis for the season. Graphical outputs included X-Y graphs, histograms and bar charts used to analyze results or for presentation.

The PMV and temperature profiles programs were special programs as their name indicate. They performed calculations using data previously scanned using the general data analysis program.

Some of the option menus for these programs are listed below.

1) General Data Analysis Program

Main Menu:
1-Read data from user specified files
2-File operations
3-Display stored information
4-Statistical analysis
5-Produce X-Y plots from files
6-Produce X-Y plots from memory
7-Produce histograms
8-Produce 3D surfaces
9-Show plane distribution of temperatures

a) Statistical Analysis (option no.4 of main menu)

1-Select points
2-Change output file name
3-Calculate max,min,avg,sdev
4-Distribution
5-Decrease/increase
6-Correlation matrix
7-Planes stats
8-Additive distribution
9-Display results of additive distribution
10-Additive gradients
11-Display results of gradients

b) X-Y Graphs (option no.5 or no.6 of main menu)

0-Return to main
1-Select dates
2-Show parameters
3-Modify parameters
4-Graph page splicing
5-Edit comments box
6-Produce graphs
7-X axis limits
8-Change comment box file

2) Data Retrieval and Weather Program

0-Quit
1-Edit a calendar file
2-Edit list of commands file
3-Edit created command file
4-Command file create
5-Create weather data files
6-Edit weather file
7-Submit data search

3) PMV Calculations

0-Exit
1-Change File: temperatures
2-Change File: angle factors
3-Change File: physical parameters
4-Change File: output
5-Show angle factors/grid/surfaces/points
6-Show physical parameters
7-Select grid points
8-Show grid points
10-Calculate PMV
11-Calculate PMV of a value

Appendix C (Instruments Specifications)

1.Specifications for Air Velocity Transducers

Brand Name : TSI
Model : Omnidirectional Probe no. 1620
Repeatability : ± 0.2 % reading
Response Time : 2 s
Output Voltage : approx. 3 to 6 V DC, non-linear
Velocity Range : 0 to 3m/s
Accuracy : ± 10% reading (0.2 to 3.0m/s) +5% to -20% reading ±0.03m/s (0 to 0.2m/s)

2.Specifications for Thermal Comfort Meter

Brand Name : Bruel & Kjaer no. 1212
Accuracy : ± 0.5°C for Temperatures
Voltage Output : 1.0 V per PMV unit
0.1 V per °C
0.05 V per % of PPD

3.Specifications for Climate Analyzer

Brand Name : B&K Climate Analyzer no. 1213
Output : X-Y graph and digital

Sensors

Air Temperature : ± 0.2°C 20 s response time to 50% of step change

Surface Temperature : ± 0.5°C 2 s response time to 50% of step change

Radiant Temperature : ± 0.5°C 15 s response time to 50% of step change

Dew Point : ± 0.5°C (air temperature within 10 K of dew point) ± 1.0°C (air temperature 10 to 20 K away from dew point)

Air Velocity : 0.05 to 1m/s range 0.2 s response time to 90% of step change ± 5%, ± 0.05m/s for flow direction greater than 15 degrees from the rear of transducer axis

Appendix D (Angle Factors Calculations)

Angle factors were calculated according to the tables for a seated person (0.6m height) and averaged for 360° rotation about the person's central axis (see reference 43). The results shown here are summarized from calculations which considered each surface to be divided in smaller surfaces according to the available thermocouple locations.

A) Summary, Angle Factors, 1st Floor Forced Air System

	GRID POINTS								
Surfaces	A1	A2	A3	A4	A5	B1	B2	B3	B4
Wa. Window	0.182	0.034	0.043	0.027	0.259	0.281	0.053	0.088	0.055
Window	0.020	0.013	0.014	0.014	0.030	0.062	0.020	0.023	0.024
Neigh.Wa.	0.066	0.068	0.122	0.345	0.259	0.101	0.081	0.167	0.374
Back Wa.	0.046	0.187	0.259	0.257	0.040	0.051	0.063	0.088	0.070
Door Wa.	0.376	0.343	0.146	0.083	0.153	0.133	0.357	0.148	0.087
Floor	0.215	0.267	0.305	0.201	0.192	0.285	0.301	0.353	0.276
Ceiling	0.095	0.088	0.112	0.072	0.067	0.087	0.126	0.133	0.114
Total	1.000	1.000	1.001	0.999	1.000	1.001	1.001	1.000	1.000

B) Summary, Angle Factors, 1st Floor Modular

	GRID POINTS								
Surfaces	A1	A2	A3	A4	A5	B1	B2	B3	B4
Modular	0.070	0.003	0.005	0.001	0.002	0.038	0.009	0.010	0.001
Wa. Window	0.112	0.031	0.038	0.026	0.257	0.243	0.045	0.078	0.054
Window	0.020	0.013	0.014	0.014	0.030	0.062	0.020	0.023	0.024
Neigh.Wa.	0.066	0.068	0.122	0.345	0.259	0.101	0.081	0.167	0.374
Back Wa.	0.046	0.187	0.259	0.257	0.040	0.051	0.063	0.088	0.070
Door Wa.	0.376	0.343	0.146	0.083	0.153	0.133	0.357	0.148	0.087
Floor	0.215	0.267	0.305	0.201	0.192	0.285	0.301	0.353	0.276
Ceiling	0.095	0.088	0.112	0.072	0.067	0.087	0.126	0.133	0.114
Total	1.000	1.000	1.001	0.999	1.000	1.001	1.001	1.000	1.000

C) Summary, Angle Factors, 3rd Floor Hydronic

Surfaces	GRID POINTS A1	A2	A3	A4	A5	B1	B2	B3	B4
Baseboard	0.002	0.001	0.005	0.006	0.027	0.083	0.011	0.025	0.011
Wa. Window	0.180	0.032	0.038	0.020	0.238	0.199	0.040	0.064	0.044
Window	0.020	0.014	0.014	0.014	0.024	0.062	0.023	0.023	0.024
Neigh.Wa.	0.066	0.068	0.122	0.345	0.259	0.101	0.081	0.167	0.374
Back Wa.	0.046	0.187	0.259	0.257	0.040	0.051	0.063	0.088	0.070
Door Wa.	0.376	0.343	0.146	0.083	0.153	0.133	0.357	0.148	0.087
Floor	0.215	0.267	0.305	0.201	0.192	0.285	0.301	0.353	0.276
Ceiling	0.095	0.088	0.112	0.072	0.067	0.087	0.126	0.133	0.114
Total	1.000	1.000	1.001	0.999	1.000	1.001	1.001	1.000	1.000

D) Summary, Angle Factors, 3rd Floor Modular

Surfaces	GRID POINTS A1	A2	A3	A4	A5	B1	B2	B3	B4
Modular	0.120	0.003	0.001	0.001	0.005	0.019	0.006	0.013	0.003
Wa. Window	0.062	0.030	0.042	0.026	0.260	0.262	0.044	0.075	0.052
Window	0.020	0.014	0.014	0.014	0.024	0.062	0.023	0.023	0.024
Neigh.Wa.	0.066	0.068	0.122	0.345	0.259	0.101	0.081	0.167	0.374
Back Wa.	0.046	0.187	0.259	0.257	0.040	0.051	0.063	0.088	0.070
Door Wa.	0.376	0.343	0.146	0.083	0.153	0.133	0.357	0.148	0.087
Floor	0.215	0.267	0.305	0.201	0.192	0.285	0.301	0.353	0.276
Ceiling	0.095	0.088	0.112	0.072	0.067	0.087	0.126	0.133	0.114
	1.000	1.000	1.001	0.999	1.000	1.001	1.001	1.000	1.000