PERFORMANCE STUDY OF RADIANT PANEL SYSTEMS IN COOLING AND HEATING

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Submitted in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

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April 2003

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ACKNOWLEDGEMENTS

By completing this work, Thanking God is my utmost aim. I would like to thank my family for their unwavering patience and love through the past years.

Further, I would like also to thank my supervisor Professor M. Hammad for providing me with so many great opportunities. He was always patient, encouraging and helpful.

Many thanks also go to my friends and colleagues in Al Wathba Investment Company who have helped me a lot during the last years.

Many thanks also go to the numerous, but important, other people at the Jordan University, Professors, librarians, secretaries, machinists and students.

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NOMENCLATURE

A	Panel area (m^2)
AUST	Area weighted average temperature of uncooled, unheated surfaces (° C)
C	Specific heat of panel material (kJ/kg .° C)
C_p	Specific heat of water (kJ/kg.° C)
D	Outside diameter of the tube (m)
f	Fin coefficient
F_{1-2}	View or shape factor
g	Gravity acceleration (9.81 m/s^2)
Gr	Grashof number
H	Characteristic length (m)
h	Convection heat transfer coefficient ($W/m^2.K$)
k	Thermal conductivity (W/m.K)
M	Tube spacing (m)
m	Mass flow rate (kg/s)
n	Number of layers

Nu	Nusselt number
P	Panel perimeter (m)
Pr	Prandtl number
q	Combined heat flux (W/m^2)
Q	Combined heat transfer (W)
$\mathbf{q_c}$	Convection heat transfer flux (W/m^2)
$\mathbf{Q_c}$	Convection heat transfer (W)
$\mathbf{q_r}$	Radiation heat transfer flux (W/m^2)
$\mathbf{Q_r}$	Radiation heat transfer (W)
$\mathbf{Q_s}$	Stored heat (W)
Ra	Rayleigh number
$\mathbf{r_u}$	Characteristic panel thermal resistance($m^2.K/W$)
t	Time (s)
$\mathbf{T_a}$	Ambient temperature (° C)
$\mathbf{T_d}$	Mean skin temperature (° C)
$\mathbf{T_{in}}$	Water inlet temperature to the panel (° C)
$\mathbf{T_{out}}$	Water outlet temperature from the panel (° C)
$\mathbf{T_p}$	Panel temperature (° C)
$\mathbf{T_w}$	water temperature (° C)
$\mathbf{X_i}$	Characteristic thickness (m)
$\boldsymbol{\omega}$	Half of net spacing between tubing (m)
$\boldsymbol{\eta}$	Fin efficiency
$\boldsymbol{\beta}$	Air coefficient of expansion (K^{-1})
$\boldsymbol{\nu}$	Kinematic viscosity (m^2/s)
$\boldsymbol{\sigma}$	Stefan-Boltzmann constant (5.67×10^{-8} $W/m^2 . K^4$)
$\boldsymbol{\varepsilon}$	Emissivity ($0 < \varepsilon < 1$)
$\boldsymbol{\rho_p}$	Density of panel material (kg/m^3)

ABSTRACT

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In this work, the suitability of radiant panel systems to Amman climate, both in heating and cooling modes was studied. Performance study was carried out experimentally. This study is based on the difference between the outdoor and the indoor temperatures. The behavior of the indoor temperature during the daytime was also been analyzed.

A laboratory prototype was established in the University of Jordan to study the behavior of the systems under the actual working conditions. Outdoor, indoor and panel temperatures were measured and then analyzed to study the system response.

Radiant panel system was found to be fit and compatible for Amman weather conditions in both heating and cooling modes. On the other hand, it is expected that the use of radiant panel system in Jordan will reflect positively on the heating and cooling cost especially in public and commercial buildings.

CHAPTER ONE

INTRODUCTION

1.1 General

Keeping thermal comfort conditions during summer time has long been a human preoccupation. For thousands of years, people have used a variety of architectural techniques (thermal mass, shading, strategically placed vents…etc.) to adapt dwelling design and cultural practice to local climate conditions. After the industrial revolution, many of these techniques were adapted to the new requirements of large buildings.

In 1902, while searching for a method to control humidity in a printing plant, Carrier invented the refrigerative chiller. Within a few years, the world had an access to a device that could cool any boxy, sealed building, regardless of how much heat it gained and trapped.

During the past decades, building occupants have developed a critical attitude towards all-air systems. Terms such as "Complaint building" and "Sick building" were born. Several studies on the subject of occupant satisfaction in air-conditioned and naturally-ventilated buildings came to conclusion that the number of unsatisfied occupants in air-conditioned buildings is significantly higher than in naturally ventilated buildings. (Stetiu, 1998)

1.2 Thermal comfort

Thermal comfort is that condition of mind, which express satisfaction with the thermal environment. The subjective sensation of warmth or thermal comfort of the subject has traditionally been measured using a seven-point scale; Descriptive scales such as the ASHRAE or the Bedford scales are normally used. (Giacomini, 2002)

Table (1.1): ASHRAE and Bedford scales

ASHRAE		Bedford
Hot	+3	Much too warm
Warm	+2	Too warm
Slightly warm	+1	Comfortably warm
Neutral	0	Comfortable neither warm nor cool
Slightly cool	-1	Comfortably cool
Cool	-2	Too cool
Cold	-3	Much too cool

Another comfort measurement scale is Fanger's model that forms basis for the international ISO standard 7730. Fanger has related the percentage of dissatisfied occupants to the air velocity and the temperature inside the conditioned space; He concluded that the percentage of dissatisfied occupants increases whenever the air velocity inside the conditioned space increases at the same space temperature. (Giacomini, 2002)

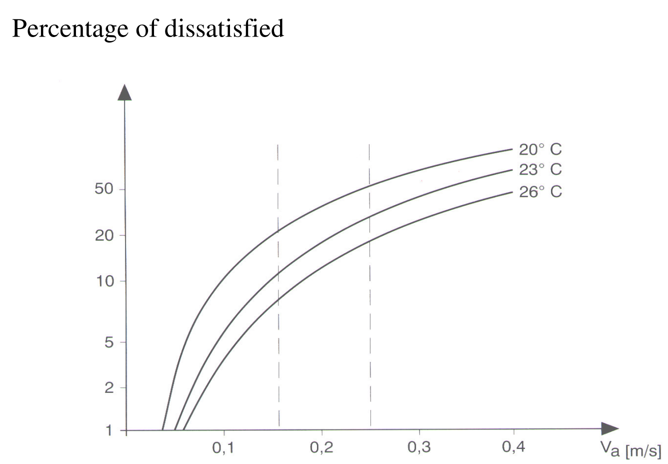


Figure (1.1): Fanger's model

Regardless of the thermal comfort measurement methods, in this work, the comfort zone as per ASHRAE shown in figure (1.2) is considered as a judge for the performance of the radiant panel system.

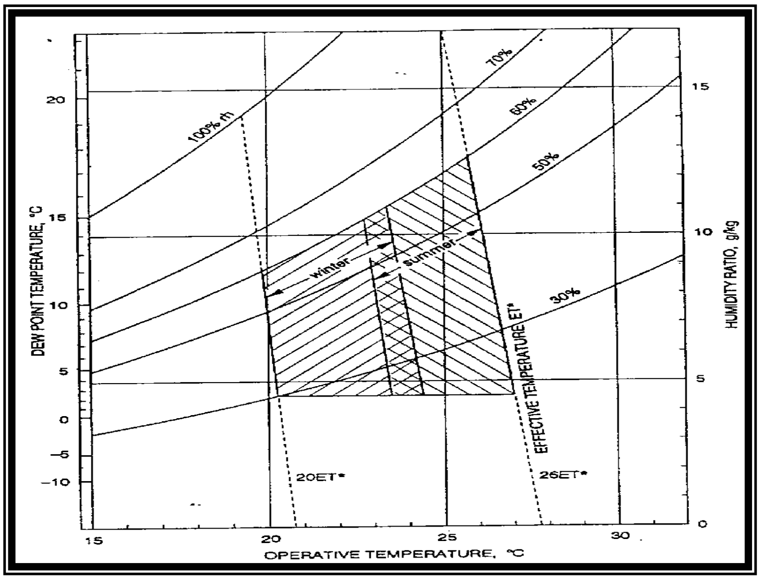


Figure (1.2): Psychometric chart

As per the ASHRAE, the comfort zone for human beings varies between 20˚C and 27˚C and between 25% to 60% relative humidity, depending on season time. (ASHRAE, 2000)

1.3 Literature review

Radiant panel system uses controlled temperature surfaces on the floor, walls, or ceiling in which the temperature is maintained by circulating water, air, or electrical current through a circuit embedded in the panel. A controlled temperature surface is called a radiant panel if 50% or more of the heat transfer is by radiation to other surfaces around the panel.

The radiant panel systems employ long-wave (infrared) radiation to the space to remove or provide heat to the space and maintain acceptable indoor air quality by controlling indoor air humidity through supplying fresh, filtered and humidity traced air among their distribution system. In its operation as an air-conditioning system, the radiant panel system thus separates the task of sensible load recovery from those of humidity controls and ventilation.

Stetiu (1998) mentioned that most radiant panel systems use water as a transport medium to connect the interior radiant surface with an exterior heat sink. He concluded that the thermal properties of water allow radiant surface to (1) remove a given amount of heat from a building which using less than 25% of the transport energy necessary for an all-air system to remove the same amount of heat, (2) shift the peak load demand to later time in the day, and (3) more easily interface with thermal energy storage system. Because radiant panel systems need to use large surface for heat exchange (usually the radiant surface occupies most of the ceiling or the floor), the temperature of the panel water must be only a few degrees different from the room air temperature. This small temperature difference allows the use of heat pumps with very high coefficients of performance.

International Performance Measurement and Verification Protocol (2001) recommended that radiant floor tubing can be used to cool a house, but it is only appropriate for dry climates. The floor temperature is held at 20˚ C by using a small chiller connected to the floor tubing of the steady 13˚ C water temperature. In arid climate, the cold floor can be used to supplement or replace standard ducted air systems. However, in humid climates, problems with over cooling could lead to wet slippery surfaces and fungus growth.

Eriksson et al (1998) concluded that it is practically realistic to use low supply temperatures. Technically they estimated that it is possible to use as low as 35˚ C water temperature with floor heating systems in new buildings. At temperature lower than that, the heat supplying areas will be too large and expensive.

Feustel et al (1994) stated that radiant panel systems are particularly suited to the dry climates. They have been used for more than 30 years in hospital rooms, to provide a draft- free, thermally stable environment. The energy saving and peak-load characteristics of these systems have not yet been systematically analyzed. Moreover, adequate guidelines for design and control of these systems do not exist. This has prevented their widespread application to other building types.

Feustel (1996) studied the requirements of building envelopes, size of the radiant surface, and thermal storage capacity of the building, as well as the energy savings potential of constant temperature conditioning for different climates and combination of building usage. Furthermore, he predicted that systems for future buildings would not distinguish between heating and cooling. Thermally perfect envelops, sufficient window shading, and thermal storage will keep thermal loads in such a narrow band that hydronic thermal conditioning will be performed at a single supply water temperature all year long. If room temperatures drop below the water temperature, the system will heat; if room temperatures exceed water temperature, it will cool. The integration of zones within a building or of buildings with different load patterns will help to reduce energy consumption.

De Carli (2000) has implemented field measurements of thermal comfort conditions in several buildings with radiant cooling systems. Long term measurements of operative, air, surface, system and external temperatures have been carried out.

The analysis of the data shows that, for major part of the time of occupancy, the operative temperature is inside the comfort range. The analysis has been lead for different classes of comfort according to existing standards. The data show an increase in space temperature during the day, which is counterbalanced by a corresponding decrease during the night. This study shows that hydronic radiant cooling systems in many buildings are an interesting alternative to full air conditioning systems, for obtaining acceptable indoors thermal environments during summer.

Mumma (2001) studied the reasons behind the popularity of radiant panel systems in Europe and summarizes three main reasons for dismissing panel cooling in many parts of the world; (1) condensation problems, (2) limited capacity and (3) the capital cost.

Finally he concluded that when a dedicated outdoor air system is used to decouple the space sensible and latent loads, the panels are left only with a portion of the space sensible loads. And if the occupancy exceeds the design by a factor of two to three, it could take hours for the condensation thickness to equal the diameter of a human hair.

Furthermore, he mentioned that panel cooling can meet its capacity duty, and only use 50% of the ceiling in most cases.

Al-Maaitah (2002) developed a mathematical model of the transient response of floor heating systems to ambient temperature variation based on equivalent thickness method. Then he solved the model using numerical technique. Moreover, he investigate for the optimum pipe configuration that would result in the lowest time constant of the response. He finally concluded that the steady state condition is reached within two hours based on Jordanian construction style and materials, which matches the general observations of actual floor-heating systems in Jordan.

1.4 Current work objectives

Radiant panel technology is still in research and development phase. This field is rapidly gaining popularity in Europe where cooling needs are generally small and heating is more required. In this work a study will be carried out to investigate the effectiveness of radiant panel systems in Jordan climate.

Although radiant panel technology in heating mode is well known in Jordan, the system is rarely used as cooling facility due to condensation problems and control lack of knowledge. Despite the fact that the idea of the system is simple as concept, but it has it's own complexity in practice, which result in more expensive set-ups. Never the less, if this system is installed in large buildings and in multi-story buildings, this additional expenses would be recovered.

In this work, the design of a suggested prototype will be carried out along with implementation. Using data available from previous works in different countries and data available from European manufacturers, the prototype is established as an extension to the university's laboratory.

This work is different from the previous works in that it considers the performance of the radiant panel system in cooling and heating mode for Amman climate based on the radiant heat generated or absorbed with only the natural ventilation without any subsidiary air system.

CHAPTER TWO

CONCEPT OF RADIANT PANEL SYSTEM

2.1 Introduction

Radiant panel system uses a combination of radiation and convection. This amount of radiant heat transfer can be as high as 50% of the sensible load while convection accounts for the reminder, (ASHRAE, 2000). With radiant panel system, the radiation heat transfer occurs through a net emission of electromagnetic waves from the occupants and their surroundings to the cool or from the warm panel. On the other hand, convection first cools or heats the room air due to contact with the panel creating convection currents within the space which transfers the energy between its source and the panel where it is absorbed or emitted.

2.2 Ceiling panel

Ceiling panel that used in the present work for cooling is the basic component in the radiant panel system. In its basic characteristic consists of four main parts.

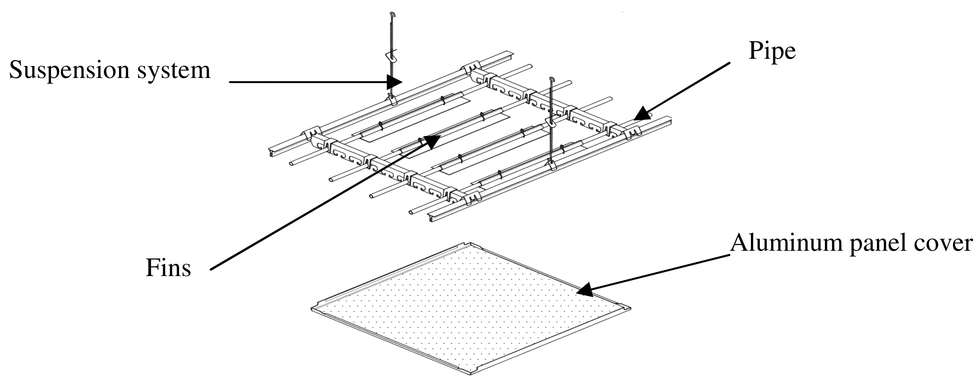


Figure 2.1 Ceiling panel

i. Aluminum panel cover, which behaves as a false ceiling in addition to its function as the surface in which heat will be transmitted or absorbed, increase the heat transfer area. These aluminum sheets must be perforated to allow natural or forced convection. Perforated aluminum panels are manufactured in different sizes, and some models can be ordered with built-in pipes and fins.

ii. Pipes, in which the chilled water circulates. Pipes could be made from different kinds of materials such as polybutylene, copper, cross-linked polyethylene or aluminum. Some conditions shall be fulfilled in the type of pipes which one wants to use such as to withstand minimum pressure needed to circulate chilled water, reliable wall thickness to maximize conductive heat transfer coefficient and easy to be installed.

iii. Fins, which many manufacturer catalogues mention as diffusers. This part of the radiant panel is essential to assure enough contact surfaces between the pipes and the perforated panels.

iv. Suspension system, Since the perforated panel, pipes, water and fins have a noticeable weight, a suspension system should be capable to hang this weight in addition to its flexibility to be adjusted in order to have precise level for the radiant panel.

2.3 Floor panel

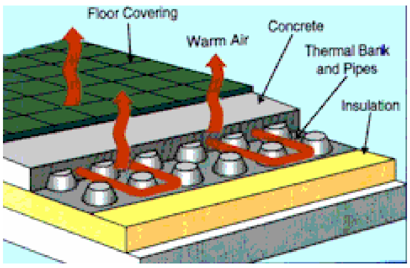


Figure (2.2): Floor panel (Giacomini, 2002)

Floor panel consists of four main layers as follow:

i. Insulation layer: for every under floor panel, a layer of thermal insulation is required to reduce heat loss downward. Many types of thermal insulation could be used including; polystyrene or polyurethane. The major differences between these two types are: method of installation and the value of thermal conductivity; for polystyrene, the thermal conductivity is about 0.03 W/ m. ˚ C and installed as boards, where as the polyurethane has 0.017 W/ m. ˚ C as thermal conductivity and mainly installed as sprayed foam or prefabricated boards.

ii. Piping layer: it is well known that copper and cross-linked polyethylene pipes are mainly used for floor panel piping. Cross-linked polyethylene pipe is preferable due to its flexibility, long life, easy to install and reasonable prices, whereas copper pipes have better thermal conductivity.

iii. Concrete screed: this layer forms the thermal mass of the floor panel and the connection layer between the piping layer and the tiles. In some cases, additives are used to increase thermal conductivity that improves the system response. In most cases steel mesh must be used to reinforce and to improve the mechanical properties of the concrete screed.

iv. Tiles layer: type of tiles needed to be installed must be defined before design, as it affects the design tangibly. In wide floors, expansion joints must be considered to overcome thermal expansion effects.

2.4 Pumping unit

In the present work, centrifugal pumps are used to circulate chilled or hot water between the heat sink/source and the pipes in the radiant panel system. The pump shall be selected according to pre-calculated flow rates and pressure drop.

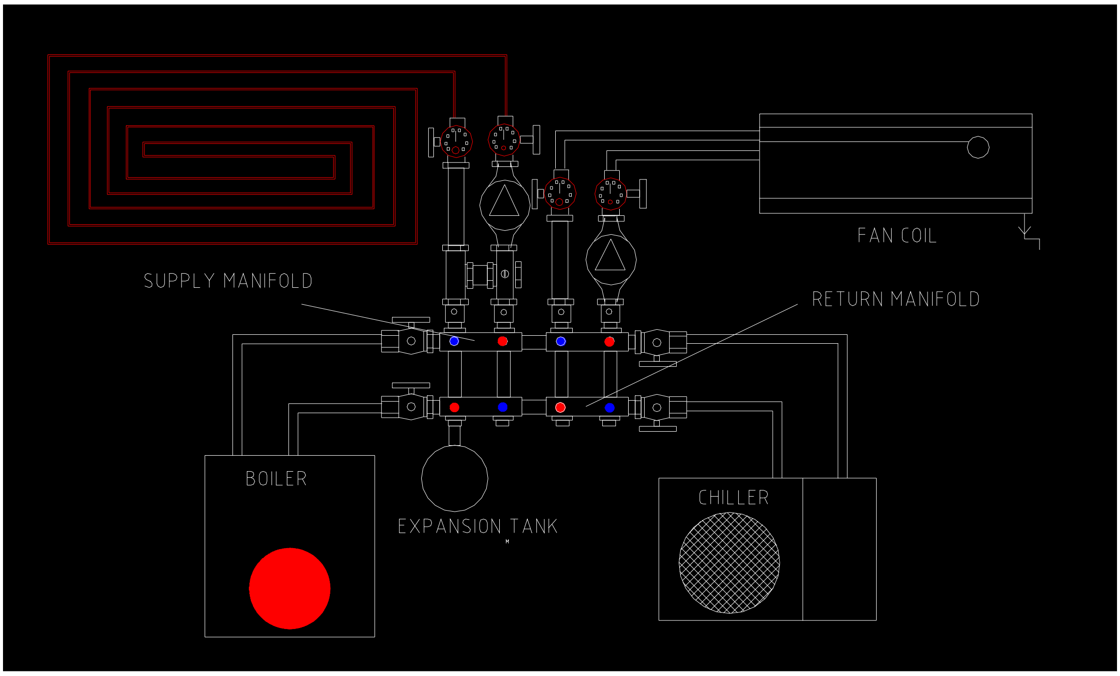


Figure (2.3) Schematic diagram for system components

Other components are used to complete the radiant panel system; chiller and boiler as heat sink and source, valves, fittings and in many cases, the use of a fan coil is a must so as to treat and humidity trace inlet air as shown in figure 2.3.

2.5 Design procedures for radiant panel system in heating and cooling

As mentioned in ASHRAE handbook, (ASHRAE, 2000). Either hydronic or electric circuits can be used to control panel surface temperature. The required effective surface temperature T_p for a combined heat transfer $q = q_r + q_c$ (where; q is the total heat flux emitted or absorbed by the panel, q_r is the amount of heat flux that take place by radiation and q_c is the amount of heat flux that take place by convection) can be calculated by using applicable heat transfer equations for q_r and q_c depending on the position of the panel. At a given ambient temperature T_a, the area weighted average temperature of uncooled (unheated) surfaces (AUST) could be predicted first. Figure (2.4) and figure (2.5) can be used to find T_p when q and AUST are known. The next step is to determine the required mean water temperature T_w in hydronic system. It depends primarily on T_p, tube spacing M and the characteristic panel thermal resistance r_u.

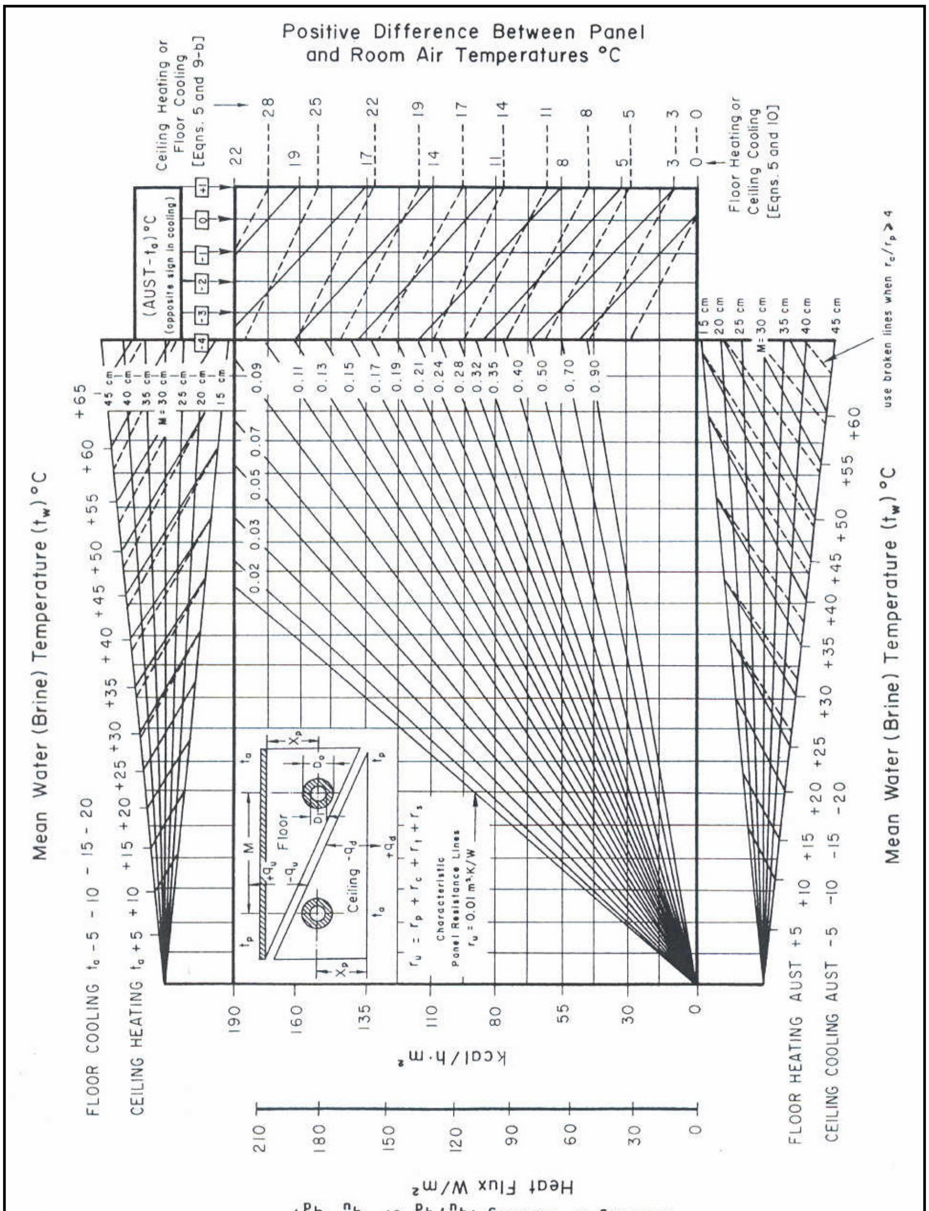


Figure (2.4): Design graph for heating and cooling with floor and ceiling panels (ASHRAE, 2000)

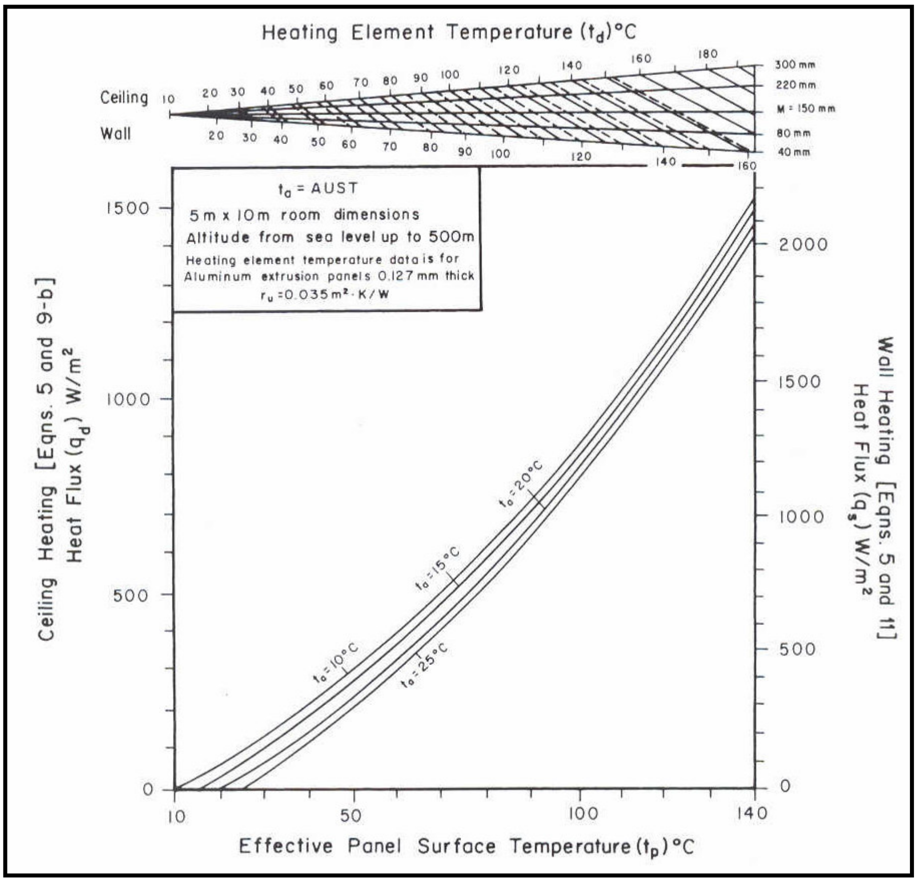


Figure (2.5): Design graph for heating with aluminum ceilings and wall panels

(ASHRAE, 2000)

The following algorithm may also be used to design and analyze panels (ASHRAE, 2000):

$$T_d = T_a + \frac{(T_p - T_a)M}{2\omega\eta + D_\circ} + q(r_p + r_c + r_s M) \text{----------------} (2.1)$$

Where

T_d = Pipe's outer surface temperature, ˚ C

q = Combined heat transfer flux for panel surface, W/m^2

T_a = Ambient design temperature, ˚ C

D = Outside diameter of tube or characteristic contact width of the tube with the panel, m

M = On-center spacing of circuit, m

2ω = Net spacing between tubing, M-D, m

η = Fin efficiency, dimensionless

Fin efficiency can be calculated using the following equation:

$$\eta = \frac{\tanh(f\omega)}{f\omega} \cong \frac{1}{f\omega} \; for f\omega > 2 ------------------(2.2)$$

f = Fin coefficient, dimensionless

$$f = [\frac{q}{m(t_p - t_a)\sum_{i=1}^{n} k_i x_i}] for t_p \neq t_a ---------------------(2.3)$$

Where

m = 2+rc/2rp

rc = Thermal resistance of panel cover, m^2.K/W

rp = Thermal resistance of panel, m^2.K/W

n = Number of layers with different materials, including panel and surface covers.

x_i = Characteristic thickness of each layer i, m

k_i = Thermal conductivity of each layer I, W/(m. K)

The required mean water temperature is

$$T_w = (q + q_b)Mr_t + T_d ------------------(2.4)$$

Where q_b is the flux of back and perimeter heat loss (positive) in a heated panel or gains (negative) in a cooled panel and r_t is the thermal resistance of the tube wall per unit tube spacing, m.K/W.

2.6 Heat transfer

Applying energy conservation principles for the panel shown in figure (2.6) will yield the following:

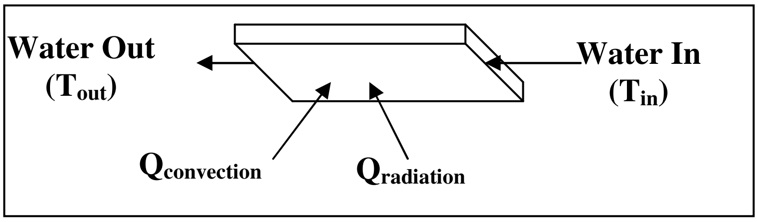


Figure (2.6) Heat Transfer Schematic

Heat convection between the room and the panel

As air moves due to a difference in density in the vicinity of the cold panel compared with that away from it, then, we consider it as natural convection, which is calculated using the following equation:

$$Q_c = hA(T_p - T_r) \text{-----------------------------} (2.5)$$

Where

h = Convection heat transfer coefficient, W/(m^2. K)

A = Panel area, m^2

T_p= Panel temperature, K

T_r = Room temperature, K

The heat transfer coefficient can be written as follows:

$$h = \frac{k}{H} Nu \text{------------------------------------}(2.6a)$$

$$H = \frac{A_p}{P} \text{------------------------------------}(2.6b)$$

Where

k = Conductivity of air, W/(m.K)

H = Characteristic length, m

Nu = Nusselt Number

P = Panel perimeter, m

Nusselt number for convection heat transfer on upper heated or lower cooled plates, can

$$Nu = 0.54 Ra^{1/4} \cdots\cdots 10^4 \le Ra_L \le 10^7 \text{----------------------}(2.7a)$$

$$Nu = 0.15 Ra^{1/3} \cdots\cdots 10^7 \le Ra_L \le 10^{11} \text{----------------------}(2.7b)$$

be calculated using the following equations (Incropera, 1996):

Where

Ra = Rayleigh number, and could be calculated using the following equation:

$$Ra = Gr\,\Pr = \frac{g\beta(T_p - T_r)L^3}{\nu\alpha} \text{----------------------------}(2.8a)$$

$$Gr = \frac{g\beta(T_p - T_r)L^3}{\nu^2} \text{--------------------------------}(2.8b)$$

$$\Pr = \frac{\nu}{\alpha} \text{--}(2.8c)$$

Where

Gr = Grashof number

Pr = Prandtl number

g = 9.81 m/s2

β = Air coefficient of expansion, K^{-1}

L = Characteristic length, m = H in case of rectangular panel

α = Constant

υ = Kinematic viscosity, kg m^{-1} s^{-1}

Convection in panel systems is usually natural; that is, the warming or cooling of the boundary layer of air generates air motion. In practice, many factors, such as a room's configuration, interfere with or affect natural convection. Infiltration, the movement of persons, and the mechanical ventilating systems can introduce some forced convection that disturbs the natural process.

Min et al. Equation could be used to calculate the natural convection heat transfer for cooled ceiling and hot floor radiant panel systems, (ASHRAE,2000) as follows:

$$Q_c = 2.42\frac{\left(abs\left[T_p - T_r\right]\right)^{0.31}\left(T_p - T_r\right)}{H^{0.08}} \text{------------------------} (2.9)$$

Heat radiation between the room and the panel

Radiant heat transfer is an electromagnetic radiation, which is of the same nature as solar radiation. In contrast to heat transfer by convection that require a medium for its existence, heat transfer by radiation takes place in complete vacuum. The radiation heat exchange between two objects is proportional to the fourth power of their absolute temperatures. The net heat exchange by radiation between two bodies is given by the following equation:

$$Q_r = \sigma A_p F_{1-2}\varepsilon(T^4 - T_p^4) \text{------------------------------} (2.10)$$

Where

σ = The Stefan-Boltzmann constant = 5.669 x 10^{-8} W/m^2 . K^4

A_p= Surface area, m^2

F_{1-2}= The view or shape factor, which indicates the fraction of energy leaving body 1 and reaching body 2.

ε = The emissivity of the objects.

T = The average temperature of surfaces that exchange heat with the panel, K

Heat Stored in the panel

Due to the change in the panel temperature, the amount of heat stored varies according

$$Q_s = \rho_p C A_p \frac{\partial T_p}{\partial t} \text{-----------------------------------} (2.11)$$

to the following equation:

Where

Q_s = Heat stored in the panel on changing its temperature.

ρ_p =Density of the panel material per unit area, kg/m^2

C = Average specific heat of the panel material, J/kg.K

A_p = Panel area, m^2

T_p = Panel temperature, K

t = Time, s

Heat removed or rejected by the water

The recommended design value of the rise in the cold water temperature across the radiant panel is four degrees, and the drop in the hot water temperature in heating mode is four to eight degrees. The heat absorbed or rejected as a result of this change in temperature cools or heats the space. The piping system must be sized so that it should be adequate to carry the heat needed for the space it services.

The heat balance equation for the water is given as follows:

$$Q_w = \dot{m} C_p (T_{in} - T_{out}) \text{-----------------------------------} (2.12)$$

Where

Q_w = The heat rate required by the room, kW

m = The mass flow rate flowing through the panel, kg/s

C_p = The specific heat for water, kJ/ kg. ° C

T_{in} = Water inlet temperature, ° C

T_{out} = Water outlet temperature, ° C

Heat balance equation of the panel

The heat balance equation of the panel is summarized by the following equation:

$$Q_s + Q_r + Q_c = Q_w \text{ -----------------------------------(2.13)}$$

$$\rho_p C A_p \frac{\partial T_p}{\partial t} + \sigma A_p F_{1-2} \varepsilon (T^4 - T_p^4) + h A_p (T_p - T_r) = \dot{m}\, C_p (T_{in} - T_{out}) \text{ ------(2.14)}$$

During the steady state condition, the changing in panel temperature with respect to time, could be considered equal to zero, the heat balance equation become as follow:

$$\sigma A_p F_{1-2} \varepsilon (T^4 - T_p^4) + h A_p (T_p - T_r) = \dot{m}\, C_p (T_{in} - T_{out}) \text{ -------------(2.15)}$$

CHAPTER THREE

DESIGN AND CONSTRUCTION OF THE PROTOTYPE

3.1 Purpose

The prototype has been designed to be used as a classroom for 25 students. Then it has been implemented as an extension to the Mechanical Engineering Department laboratories

3.2 Structure

3.2.1 Walls

As shown in figure (3.1), the prototype consists of four walls as follow:

- **Eastern wall;** that faces non-heated and a non-cooled room and contains a window and an entrance door.
- **Western wall**; that faces the out door atmosphere with neither doors nor windows. As it is the writing board wall.
- **Northern wall**; that contains three windows and faces the out door atmosphere.
- **Southern wall;** that contains neither windows nor doors, but faces the out door atmosphere in its upper part and non-heated and non-cooled room in its lower part. Cross section views for all structure components are available in appendix C.

3.2.2 Ceiling

A standard ceiling was constructed to consist of hollow bricks and other components mentioned hereunder in section 3.3, with 52.2 m^2 area.

3.2.3 Floor

The floor consists of two parts, the first part is the ceiling of the old building, and the other is new flooring for floor panel. All details are mentioned hereunder in section 3.3.

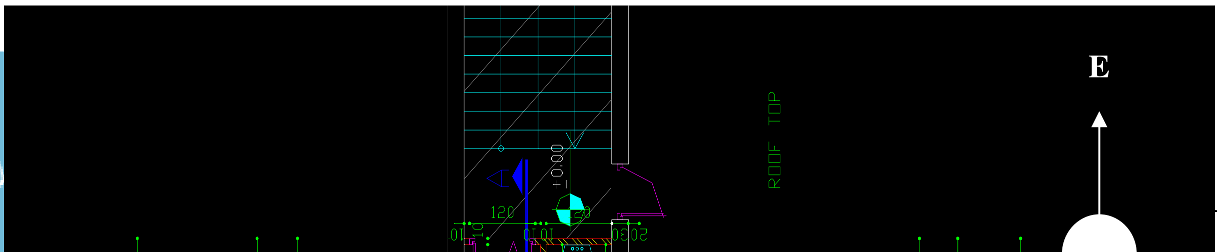


Figure (3.1): Prototype structure

3.3 Structure components

All walls that face the outer atmosphere are composed of double-bricked layers of 10 cm each and three cm of polyurethane insulation sandwiched in between and both bricklayers are covered with two cm of plaster. Where as the walls that face non-cooled and non-heated rooms are composed of single brick layer covered with plaster on both sides.

The ceiling is composed of a layer of hollow bricks of 18 cm thickness covered with a layer of concrete that has been sprayed with a layer of three cm of polyurethane foam, and another five cm concrete, and finally with a layer of one cm asphalt rolls as water proofing.

The floor composed of the ceiling of the old building and a layer of mortar for leveling purposes, then with a layer of 10 cm concrete that covered with three cm polystyrene and the floor panel.

Table (3.1): Structure Components Characteristics

Description	Area (m^2)	U-Factor
Northern wall	30.80	0.64
Eastern wall	16.65	2.40
Southern wall (lower part)	15.60	1.88
Southern wall (upper part)	15.20	0.64
Western wall	18.65	0.64
Ceiling	52.20	0.65

3.4 Thermal insulation

Two types of thermal insulation were used in this prototype; sprayed polyurethane foam and polystyrene boards.

3.4.1 Polyurethane

Polyurethane is a two component, fluorocarbon blown spray system, based on polymeric, which produces rigid, closed cell urethane foam having a nominal core density of 40 kg/m^3. Thermal conductivity of polyurethane varies from 0.017 W/m. ˚C as an initial value up to 0.023 W/m. ˚C as an aged value. In this prototype the polyurethane foam was used to form a thermal envelop of three cm thickness that sandwiched in-between two bricks layers in walls and in-between brick layer and concrete layer in the ceiling. Figure (3.2) shows the polyurethane as insulation for walls and ceiling.

Figure (3.2): Sprayed polyurethane

3.4.2 Polystyrene

Polystyrene is mainly known as boards with different thickness and different densities. As the polystyrene density increases the thermal conductivity decreases. As a building material, the most usage for polystyrene is for walls, ceilings and floors insulation using densities of 15, 25 and 30 kg/m^3 that produce thermal conductivity of 0.037, 0.034 and 0.03 W/m. ˚ C respectively. In this prototype, the polystyrene was used to insulate the floor panel and formed the first layer in the floor panel. Figure (3.3) shows the polystyrene boards as insulation for the floor.

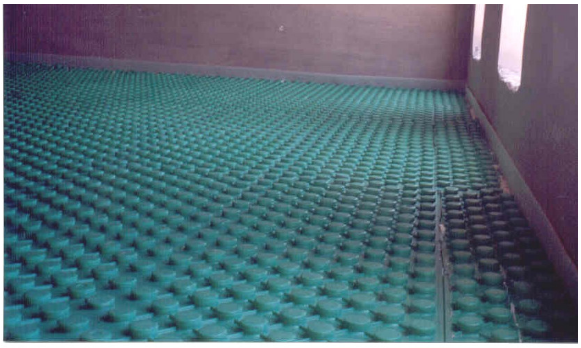


Figure (3.3): Polystyrene boards

3.5 Windows

Two groups of windows were used in this prototype; the first group are exposed to outdoor atmosphere and the second group are part of internal partition with non-heated and non-cooled spaces.

The first group consists of three double glazed windows with similar areas in the northern wall, but with different shading factors. Where as the second group consists of one single glazed window in the eastern wall. The following table shows all windows characteristics values.

Table (3.2): Windows characteristic values

Window No.	Position	Area (m^2)	Shading coeffi.	U value
Window 1	Northern wall	2	0.49	2.08
Window 2	Northern wall	2	0.33	2.00
Window 3	Northern wall	2	0.81	3.14
Window 4	Eastern wall	2	1.00	4.60

3.6 Radiant ceiling panel

3.6.1 Ceiling panel

As previously mentioned, the design procedures are followed to find out the main properties for the needed ceiling radiant panel. First, the cooling load depending on 37 ° C outdoor temperature and 45% relative humidity and 25 ° C indoor temperature estimated. Taking into consideration that the number of occupants will be 25 students and the prototype will be used as a classroom facility, then the needed flow rate and the inlet water temperature were selected using figure (2.4).

After scanning manufacturer's catalogues, the model Giacoklima has been found to be the best compatible to the prototype's needs. (Giacomini, 2002)

3.6.2 ceiling panel specifications

The ceiling panel composed of; suspension system, fins, pipes and aluminum panels as described before.

The system was installed as per the manufacturer's recommendations, first electrical wiring was installed, then the suspension system was found to hang the other components, next, the spacers were connected to the suspension system in order to install the pipes. The fins were adhered to the pipes in order to maximize heat transfer area and finally the aluminum panels were fixed to the system.

After installing the system, the following parameters became fixed:

- The whole room treated as a single zone.
- Four water passages (loops) were used.
- Parallel water distribution system was used, i.e. the pressure drop through all passages is the same.
- Flow rate through all passages is the same and equal to a quarter of the total flow rate.
- The ceiling panel composes of sub panels. Dimension for each is (0.6 x 0.6) m.
- Flow rate through each sub panel is equal to the total flow rate.
- Number of sub panels is 126.

- Two types of sub panels were raised, the first type is called the active sub panel and the second type is the complimentary sub panel. Active sub panel is that contains pipes, whereas the complimentary sub panel is that used for decoration reasons and to complete the ceiling area.
- Number of active sub panels is 118.
- Total area of active sub panels is 42.48 m^2
- Total area of the ceiling is 52.2 m^2
- Percentage of active sub panels to ceiling area is % 81.4.

3.6.3 Ceiling panel piping

Polybutylene pipes of 14x1 mm size with 0.22 W/(m. K) thermal conductivity were used. Different characteristics for Polybutylene pipes are available in appendix D. The pipes were installed as four loops, each of 83 meters length and 12.5 cm apart from each other as shown in figure (3.4).

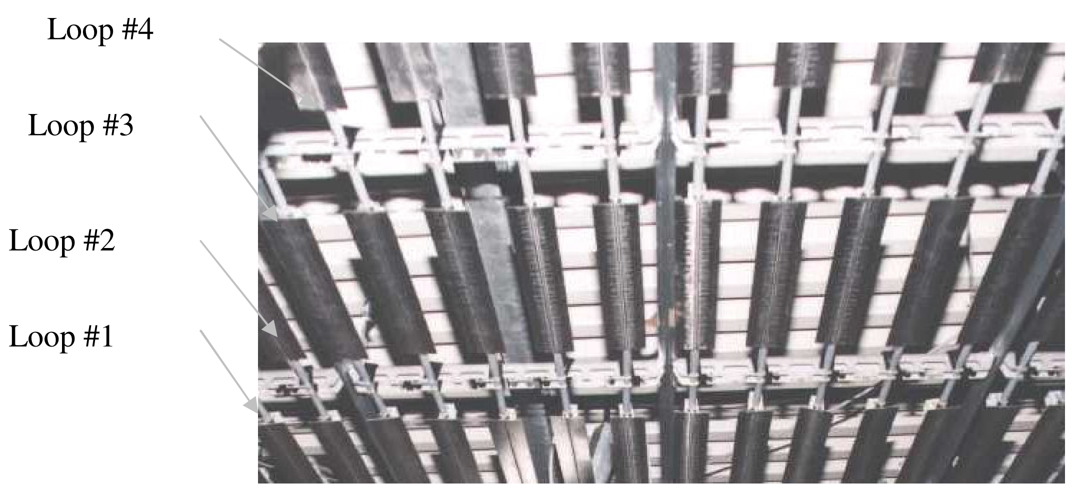


Figure (3.4): Ceiling panel piping

Aluminum fins were installed to maximize heat transfer area of the pipes and to assure enough contact with the aluminum panels. Finally the aluminum panels were installed to form the final component of the ceiling panel as shown in figure (3.5).

Figure (3.5): Ceiling panel

3.6.4 Water distribution in the ceiling panel

As water is the heat transfer media between the chiller and the ceiling panel, a stand-alone distribution network was established. The water network composed of the chiller that will be studied in next section, pumping unit, strainer, check valve, filter, expansion tank, flow detector and distribution manifold.

Part of the above components is important to complete the system such as the chiller, the expansion tank, and the others are necessary to protect the system.

Figure (3.6) shows the distribution manifold

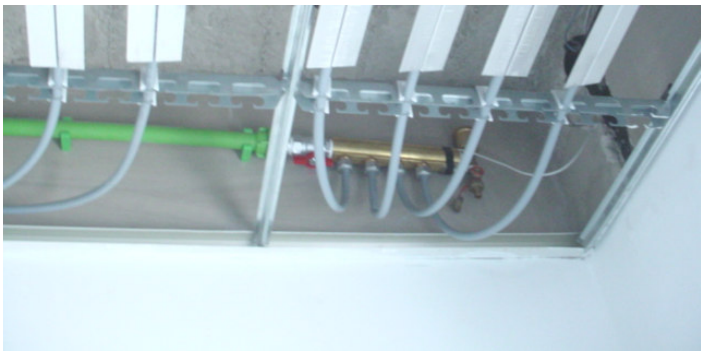


Figure (3.6): Distribution manifold

3.7 Radiant floor panel

3.7.1 Floor panel

The floor panel is different from the ceiling panel since there is no ready panels to be installed, then for each application, there is it's own floor panel. Following the design procedures suggested by the ASHRAE, and taking into consideration that the floor panel might be used for cooling in other studies. The design took place based on the cooling load and not the heating load, since the cooling load is greater than the heating load, and once the cooling load is recovered, the heating load will definitely be recovered using same panel.

3.7.2 Floor panel specifications

Under floor panel that mainly used for heating is implemented according to available knowledge from manufacturers (Giacomini, 2002) and the ASHRAE (ASHRAE, 2000). It composes of pipes in form of loops sandwiched between layers of insulation, concrete and tiles as shown in the next figure.

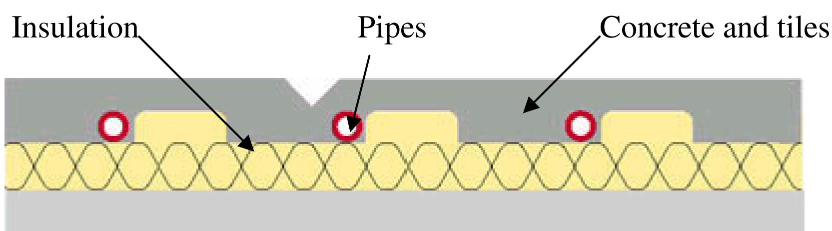


Figure (3.7): Components of the floor panel

After installing the system, the following parameters became fixed:

- The whole room treated as a single zone.
- Seven water passages (loops) with 7.5 cm pipe's spacing were used.
- Parallel water distribution system was used, i.e. the pressure drop through all passages is the same.

- Two pipe sizes were used, for outer loops the pipe size is 20x2 mm and for the inner loops, the pipe size is 18x2 mm.
- Flow rate through all inner loops is the same and for the outer loop is different.
- 80% of the floor area was considered as an active panel.

Figure (3.8) shows the pipes distribution and loops.

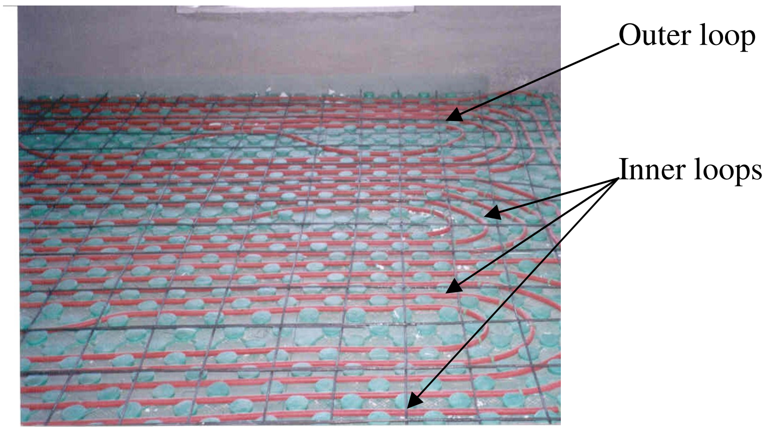


Figure (3.8): Floor panel pipes and loops

3.7.3 Floor panel piping

After scarping three cm of polystyrene insulation, cross-linked polyethylene pipes of 0.35 W/(m. K) thermal conductivity were used to form the floor panel. Two sizes of the polyethylene pipes were used; the first one is 20X2 mm which was implemented to form the first loop that adjacent to the northern wall, and the second size is 18x2 mm which formed the other six loops that cover the rest of the floor area.

The reason behind this distribution of the loops is to assure maximum heat output in the area near to the northern wall, as it is the only wall that has windows and exposed to outer atmosphere.

The next step for formulating the floor panel was to cover the pipes with steel mesh and concrete then finishing with tiles and its requirements as shown if figure (3.9).

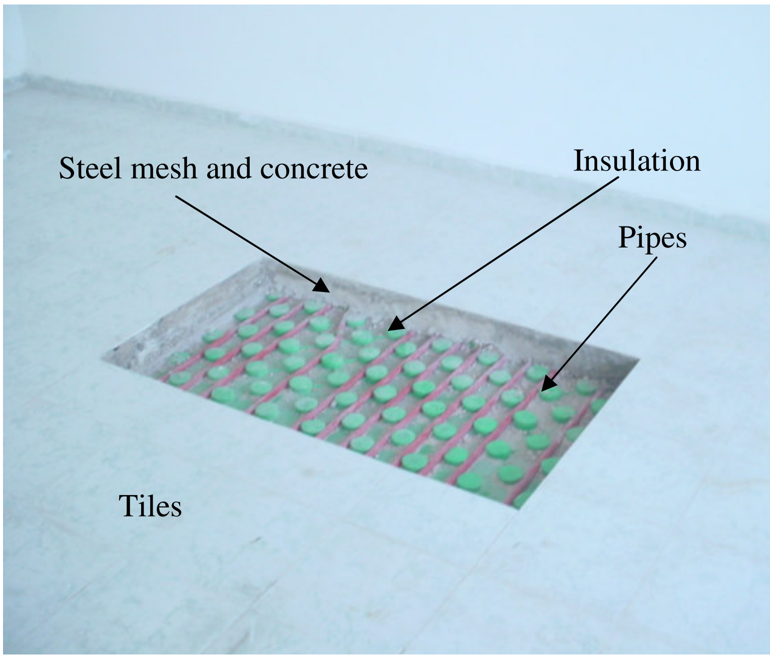


Figure (3.9): Floor panel

3.7.4 Water distribution in the floor panel

As mentioned in section 3.6.4 for the ceiling panel, same components were used for the floor panel except the chiller, in floor panel a boiler has been used since it's common use is for heating.

The distribution manifold for the floor panel is shown in figure (3.10).

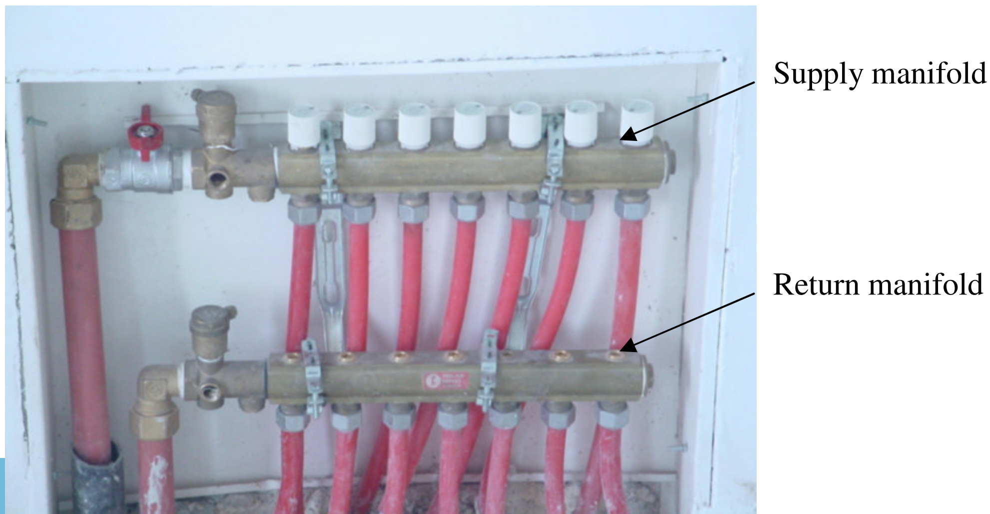


Figure (3.10): Floor distribution manifold

3.8 Heat sinks and sources

3.8.1 Cold water

For cooling mode a constant temperature source of water is required with temperature no less than the dew point.

Refrigerative chillers are the most common equipment for this purpose, while many other sources are some times used such as cooling towers. Another cost free source is the cold water from rivers; in the British parliament building, which is cooled by means of radiant panel system, the Times River was found to be capable to provide water at 14 ˚ C all summer time.

In other cases, where district cooled water networks are available, mixing units with three-way motorized valves are used so as to maintain inlet water at 16˚ C maximum to assure escaping of condensation problems.

In this work a refrigerative chiller of four tons was used to provide the system with cooled water at the needed flow rate.

3.8.2 Hot water

As condensation problems are not exist in heating mode, the system is less sensitive for water temperature. Even the water temperature exceeds 80 ˚ C in district heating systems, the use of heat exchangers might be enough to reduce the water temperature to the acceptable ranges.

In this work, a steel sheet boiler of 30-kcal capacity was used to provide the system with hot water at the required flow rate.

3.9 Pumping unit

Two pumps were used in this work to provide the system with the needed flow rates and to overcome friction in both the ceiling panel and the floor panel. For the floor panel, and as per the design results, a pump that is capable to circulate up to 80 liters per minute and to overcome pressure losses equal to 6.8 m H_2o was used. While for ceiling panel, the selected pump is capable to circulate water up to 96 liters per minute and to overcome pressure drop equal to 6.2 m H_2o.

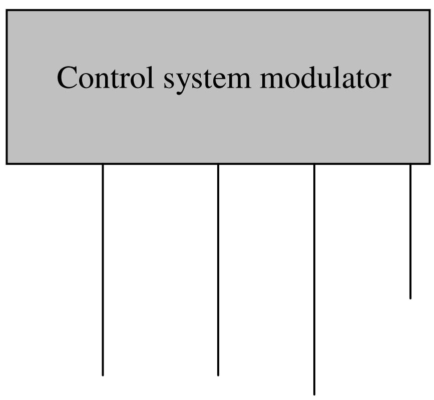
3.10 Control system

As previously mentioned, a control system was used for the following reasons:

- To protect the system from any over heating or over cooling situations.
- To control the flow rate in case of condensation problems.
- To close/open the system during the steady state situations.

Detailed information for the control system operation modes and settings are available in appendix E.

The following figure describes the control system components.



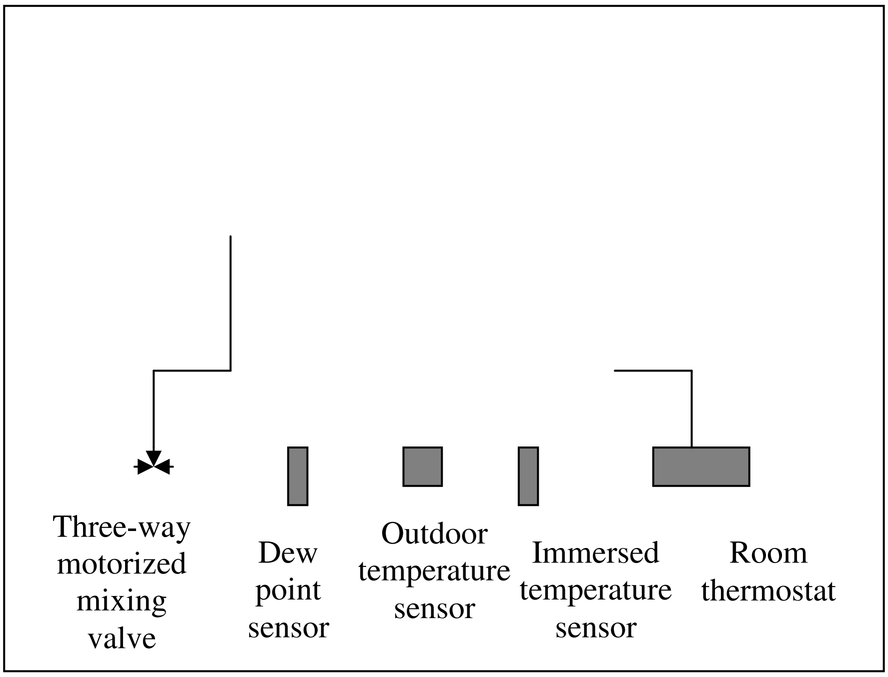


Figure (3.11): control system's components

CHAPTER FOUR

EXPERIMENTAL SETUP

4.1 System Components

The experimental setup in this study consists of the radiant panel system that previously described in chapter three, control and measurement elements. A refrigerative chiller and a boiler are also used as sources for cold and hot water respectively.

4.2 Instrumentation

The instrumentation for this experiment consisted of copper-constantan thermocouple for panel temperature measurement, dry and wet bulb thermometers for room and outside temperature measurements.

A dew point sensor was allocated on the most critical point inside the room, which is the inlet manifold for cold water in the ceiling panel system.

A pre-adjusted control system was installed for both the ceiling and the floor panels to assure bypassing any over heat or over cool situations to protect the system from any undetectable damage. Another purpose for installing the control system is for controlling the water temperature in case of over cooling cases by increasing the bypass water quantity.

Detailed schematic diagram for measurement locations is presented in figure (4.1).

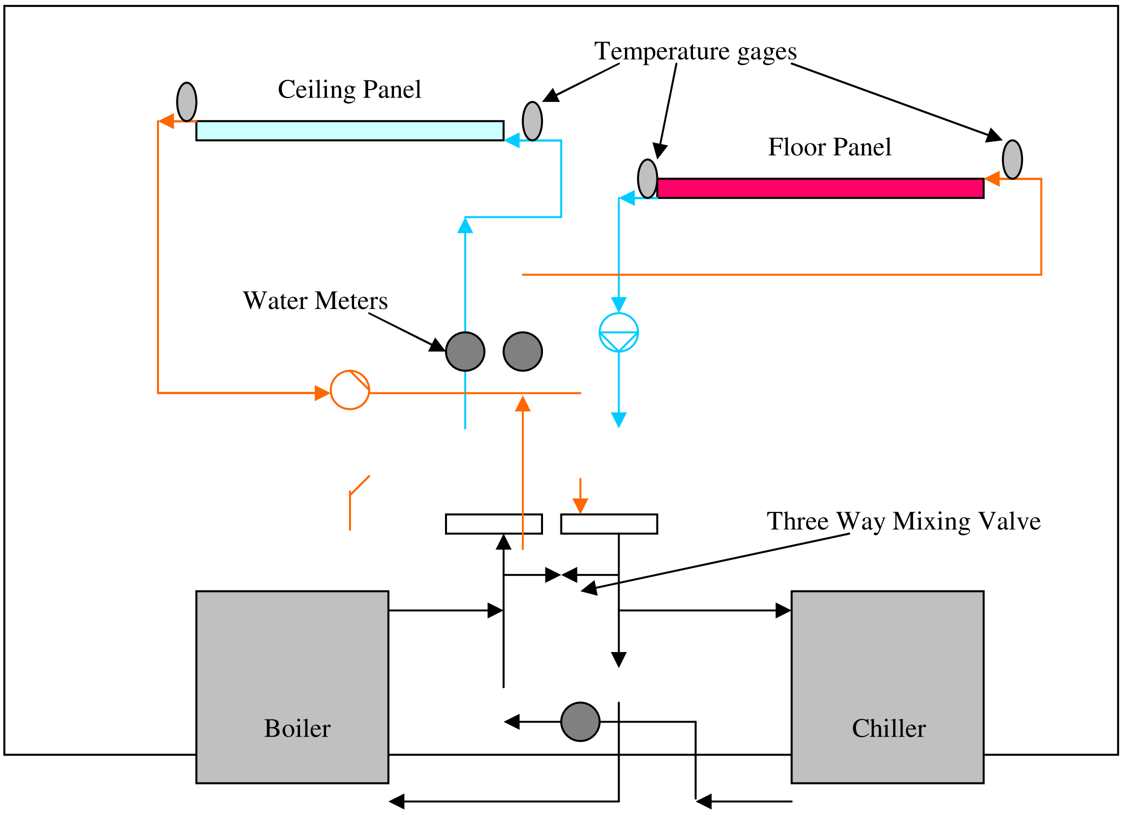


Figure (4.1): Schematic diagram for measurement locations

Temperature measurement

Two measurement instruments were used to measure the required temperatures; the first instrument is copper-constantan thermocouple that has been used to measure the panel, fins and pipes temperature, also the same instrument was used to trace the temperature gradient along the vertical axis of the prototype in three different points; the first point is half meter far from the northern surface, the second point exactly at the middle of the prototype hall and the third point is at half meter far from the southern wall and all the three points are on the same axis.

The other instrument was a normal dry bulb thermometer that has been used to measure the room temperature and the outdoor temperature.

Humidity measurement

Humidity has been measured using a wet bulb thermometer and a dry bulb thermometer, then using the psychometric chart, the relative humidity has been read out. Despite the fact that the prototype elevation is about 800 meters above the sea level, the measured data were plotted on a psychometric chart that designed for sea level in order to improve accuracy level.

4.3 Operating procedures

As this work concerns in heating and cooling of buildings, the operating procedures are divided into two groups; the first group is for cooling mode and the second is for heating mode.

4.3.1 Operating procedures for ceiling panel

After charging the system with water, and making the necessary ventilating actions, the system was connected to the control system that is adjusted according to the needed alert settings.

The chiller was turned on and left for two hours before starting measurement procedures. After testing the control system and the flow detector, measurement actions were took place.

The selected day for taking readings was 9 August 2002, in which the early morning temperature exceeded the indoor design temperature.

Considering a time span between 9:25 AM to 17:00 PM, readings for pipe skin, fins, panels, supply water, return water, outdoor and room temperatures were taken every 10 to 20 minutes. As the system were set for a constant water flow rate, the only action taken for detecting the flow rate was done by noticing the three way mixing valve.

In order to assure the functioning of the control system, the dew point sensor was detected after each reading has been taken.

4.3.2 Operating procedures for floor panel

After charging the system with water, and making the necessary ventilating actions, the system was connected to the control system that has been adjusted according to the needed alert settings

The selected day for taking readings was 14 February 2003, in which the temperature was low enough to run the system near it's peaks.

Considering a time span between 8:30 AM to 19:00 PM, readings for panels, supply water, return water, outdoor and room temperatures were taken every 10 to 20 minutes. As the system was set for a constant water flow rate, the only action taken for detecting the flow rate was done by noticing the three way-mixing valve.

CHAPTER FIVE

COOLING AND HEATING LOADS

5.1 Outdoor and indoor design conditions

Outdoor and indoor design conditions for summer time was assumed as follows:

- Dry bulb temperature: 37˚ C for outdoor and 25˚ C for indoor
- Relative humidity: 45% for outdoor and assumed the same for indoor for calculation reasons.

Where as for wintertime, the design conditions were assumed to be as follows:

- Dry bulb temperature: 2˚ C for outdoor and 22˚ C for indoor.

5.2 Cooling load calculation

5.2.1 Heat gain through walls and ceiling

Using the heat transmission equation, the heat gain through walls and ceiling was obtained. (ASHRAE, 2000)

$$Q_{w,c} = UA_{w,c}(CLTD)_{corr} ------------------------------(5.1)$$

Where

$Q_{w,c}$ is the heat gain through walls and ceiling

U is the overall heat transfer coefficient

$A_{w,c}$ is the area for the walls and ceilings

$$(CLTD)_{corr} = (CLTD + LM)K + (25.5 - T_a) + (T_o - 29.4)F ---------(5.2)$$

$(CLTD)_{corr}$ is the corrected cooling load temperature difference

Where

$CLTD$ is the cooling load temperature difference

LM is the latitude correction factor

K is the color adjustment factor

T_a is the indoor temperature

T_o is the out door temperature

F is the attic or roof factor

Applying the above equation using the design conditions on the structure components mentioned in table (3.1), the results are summarized in table (5.1).

Table (5.1): Cooling load through walls and ceiling

Description	Net Area (m^2)	Cooling load (W)
Northern wall	**24.80**	**310.6**
Eastern wall	**14.81**	**675.3**
Southern wall (lower part)	**15.60**	**583.6**
Southern wall (upper part)	**15.20**	**193.6**

Western wall	**19.21**	**209.0**
Ceiling	**51.2**	**1309.5**

5.2.2 Heat transmitted through glass

5.2.2.1 Transmission heat gain through glass

This portion of heat gain could be calculated using the following equation:

$$Q = A(SHG)(SC)(CLF) ------------------------ (5.3)$$

Where

A is the glass area

SHG is solar heat gain factor

SC is the shading coefficient

CLF is the cooling load factor

Applying the above equation will result with the following table:

Table (5.2): Transmission heat gain through glass

Window	Area (m^2)	SHG	SC	CLF	Cooling Load (W)
1	2	104	0.49	0.82	83.6
2	2	104	0.33	0.82	56.3
3	2	104	0.81	0.82	138.2

5.2.2.2 Conduction heat gain through glass

The same equation in section 5.2.1 can be used here with different $CLTD_{corr}$ values, and the results as per the following table:

Table (5.3): Convection heat gain through glass

Window No.	Position	Area (m^2)	U value	Cooling load (W)
Window 1	Northern wall	2	2.08	54.5

Window 2	**Northern wall**	**2**	**2.00**	**52.4**
Window 3	**Northern wall**	**2**	**3.14**	**82.3**
Window 4	**Eastern wall**	**2**	**4.60**	**56.1**

5.2.3 Internal heat gains

5.2.3.1 Heat gain due to lights

This amount of heat gain could be calculated considering 25 W /m^2 as a unit heat gain, then the total heat gain due to light would be 1280 W.

5.2.3.2 Heat gain due to people

Considering 70 W is the amount of sensible heat that generated by a human body, the total amount of heat generated would be 1750 W.

5.2.4 Latent heat gain

5.2.4.1 Latent heat gain due to air exchange

Assuming that the air change rate is one volume per hour, and using the next equation, then the total latent heat gain due to air exchange is 2428 W.

$$Q_L = \rho V_{oa}(\varpi_a - \varpi_o)h_{fg} \text{ -------------------------------(5.4)}$$

Where

Q_L is the latent load, W

ρ is the air density, kg/m^3

V_{oa} is the air change rate, m^3/s

ω_a is the indoor humidity ratio, kg/kg dry air

ω_o is the outdoor humidity ratio, kg/kg dry air

h_{fg} is the enthalpy of evaporation, kJ/kg

5.2.4.2 Latent heat gain due to people

Latent heat gain per person could be considered equal to 30 W, then the total latent heat gain due to people inside the prototype is equal to 750 W.

5.3 Heating load calculation

5.3.1 Heat loss through all exposed areas

Using the heat transmission equation

$$Q_l = UA(T_o - T_a) ----------------------------------(5.5)$$

Where

Q_l is the heat loss, W

A is the exposed area, m^2

T_o is the outdoor temperature, $^\circ C$

T_a is the room temperature, $^\circ C$

The above equation was used to calculate heat losses through windows, walls and ceiling, and the results are summarized in the following table.

Table (5.4): Heat loss through exposed areas

Description	Net Area (m^2)	Heat loss (W)
Northern wall	24.80	317.5
Southern wall (lower part)	15.60	586.6
Southern wall (upper part)	15.20	194.6
Western wall	19.21	245.9
Ceiling	51.20	665.6
Window 1	2.00	83.2
Window 2	2.00	80.0
Window 3	2.00	125.6

5.3.2 Heat loss by infiltration

Infiltration is the leakage of outside air through cracks and clearances around the windows. The amount of infiltration depends mainly on the tightness of the windows and on the outside wind velocity or the pressure difference between the outside and inside.

5.3.2.1 Sensible heat load

The sensible heat load is given by the following equation:

$$Q_s = m_{oa} C_p (T_a - T_o) ------------------------------ (5.6)$$

Where

m_{oa} is the mass of infiltrated outside air per unit time, kg/s

C_p is the specific heat at constant pressure of air, kJ/kg.K

Using the air change method (ACH) to estimate the volume of air change per hour and as per the ASHRAE for rooms with windows or exterior doors on one side only, then the number of air change is 1.5 volume per hour.

Applying the above equation and noting that the room volume is 173.8 m^3 then the sensible load due to infiltration was calculated to be 1694 W.

5.3.2.2 Heating load due to door opening

The average number of door opening can be calculated using the following equation.

$$Door - opening / hr = \frac{P.F}{t.n} ---------------------------- (5.7)$$

Where

P is the number of people

F is a factor for arrivals and departures, 1.33

t is the time occupancy in hours

n is the number of doors

Applying the above equation, the number of openings per hour is 66.5. To calculate the heat load due to 66.5 number of openings per hour and knowing that the infiltration rate is 4.757 m^3/hr, the following equation could be used.

$$Q_f = (\frac{1250}{3600})V_f(T_a - T_o) \text{------------------------} (5.8)$$

Where

Q_f is the heating load due to infiltration through door opening, W

V_f is the volumetric infiltration rate, m^3/hr

The result heating load due to door opening is 2167 W.

5.4 Summery of heating and cooling loads

As a result for the above mentioned segments of heating and cooling loads, the total sensible heating load is 6160 W and the total cooling load is 10013 W that consists of 6835 W as sensible load and 3178 W as latent load.

CHAPTER SIX

RESULTS AND DISCUSSION

6.1 General

In this chapter, the temperatures of six different points were measured. The room temperature, outside temperature, panel temperature, fins temperature, water temperature and pipe temperature were measured and traced to study the effect of each of them on the others.

The vertical temperature profile was also traced and compared with the ideal temperature gradients for heating and cooling systems.

The cooling and heating loads were calculated according to the outside and room temperatures to study the effect of their values on the room temperature.

6.2 Relation between outdoor and room temperatures

- **Cooling:** As shown in figures 6.1 and 6.2, it is clear that the outdoor and room temperature difference varies according to the change in the outdoor temperature during the day time. As the room temperature (at 1.8 m height) varies within a range of one degree, the system could sustain an acceptable range of variation that assured a good stability characteristics. Moreover, the system is able to achieve temperature difference between outdoor and room up to 11.5 $^\circ$ C.

- **Heating:** Figures 6.3 and 6.4 show that the system achieved a room temperature of 21 $^\circ$ C, which could be considered as an excellent temperature for heating, taking into consideration that in situations when the space is fully occupied, a room temperature higher than that could be achieved highlighting that overheating situations might appear, so a good control for water temperature is required. A difference of up to 14.5 $^\circ$ C was achieved between room and ambient temperature.

6.3 Panel and room temperatures

- **Cooling:** Reviewing the mechanism of heat transfer starting from heat gain through structure and windows, the room temperature would increases, but due to the radiant ceiling panel, the room will maintain a temperature within a certain range since the ceiling panel will absorb the sensible cooling load. Figure 6.5

through figure 6.7 show the behavior of pipe, fin and panel temperatures through the daytime.

Figure 6.8 shows that the outdoor-room temperature difference affects directly the room-panel temperature difference. As the outdoor-room temperature difference increases, the room-panel temperature difference decreases, because the panel temperature starts to increase. This relation proves the importance of using a room thermostat that is coupled to a three way mixing that adjusts the mixing ratio of the water.

- **Heating:** The panel temperature in floor heating system showed a better stability than the ceiling cooling system as shown in figure 6.9. The reason for that is the thermal mass of the floor panel.

Figure 6.10 shows that, as the outdoor-room temperature difference decreases, the room-panel temperature difference decreases too, and this behavior was expected since the heating load is decreases as the outdoor-room temperature difference decreases.

6.4 Vertical temperature gradient

- **Cooling:** Figure 6.11 through figure 6.13 show the vertical temperature distribution at three points in the room. It is evident that the variation of temperature between height 0.4 m and height 2.8 m is not more that 0.5 ˚ C which maintain uniform vertical temperature distribution compatible to with international standards.

- **Heating:** Same as cooling, figure 6.14 through figure 6.16 show the same uniform vertical temperature distribution between the two mentioned heights in heating mode.

Comparing figure 6.11 through figure 6.16 with vertical temperature distribution in all air cooling and heating systems will prove an advantage for radiant panel systems over air conditioning systems.

6.5 Cooling and heating flux

- **Cooling:** The cooling load was calculated for each reading for outdoor and room temperatures along the day time. Figure 6.17 shows the change of the sensible cooling flux per unit area. The convection portion of the heat transfer between the room and the panel was calculated for each reading of the room and panel temperature using equation 2.9 as shown in figure 6.18. As per the expectations, the system showed that the majority of heat transfer was conducted by radiation. Figure 6.18 results were calculated using heat transfer analysis showed in previous chapters. Excel algorithm was used based on heating and cooling loads formulas as per the ASHRAE.

- **Heating:** The heating load was also calculated per each reading through the daytime using equation 2.9 as shown in figure 6.19. After calculating the portion of convection heat transfer between the floor panel and the room, the results were countered the expectations since the majority of heat transfer was conducted by convection rather than radiation. The reason for that is the relatively high panel-room temperature difference which transformed heat transfer from free to mixed convection. Figure 6.20 shows the percentage of convection heat transfer in comparison with radiant heat transfer.

6.6 Outdoor and room relative humidity and humidity ratio in cooling mode

Figure 6.21 and figure 6.22 show the change of relative humidity and humidity ratio between outdoor and room through the daytime. It is clear that the relative humidity and the humidity ratio in the room is higher than that in outdoor. The reason for the increase is that the wet bulb and dry bulb temperatures are different in the room than that in the outdoor, i.e. the atmosphere inside the room became as a new one.

Figure 6.23 shows that no condensation occurred due to the temperature difference between the panel and the dew point temperatures. Moreover, this figure shows that the radiant panel system is capable to provide occupancies with thermal comfort since the system is able to reduce the space temperature to ranges within the comfort zone. In fact, the traced points as per figure 6.23 are out of the comfort zone as no humidity treatment was implemented.

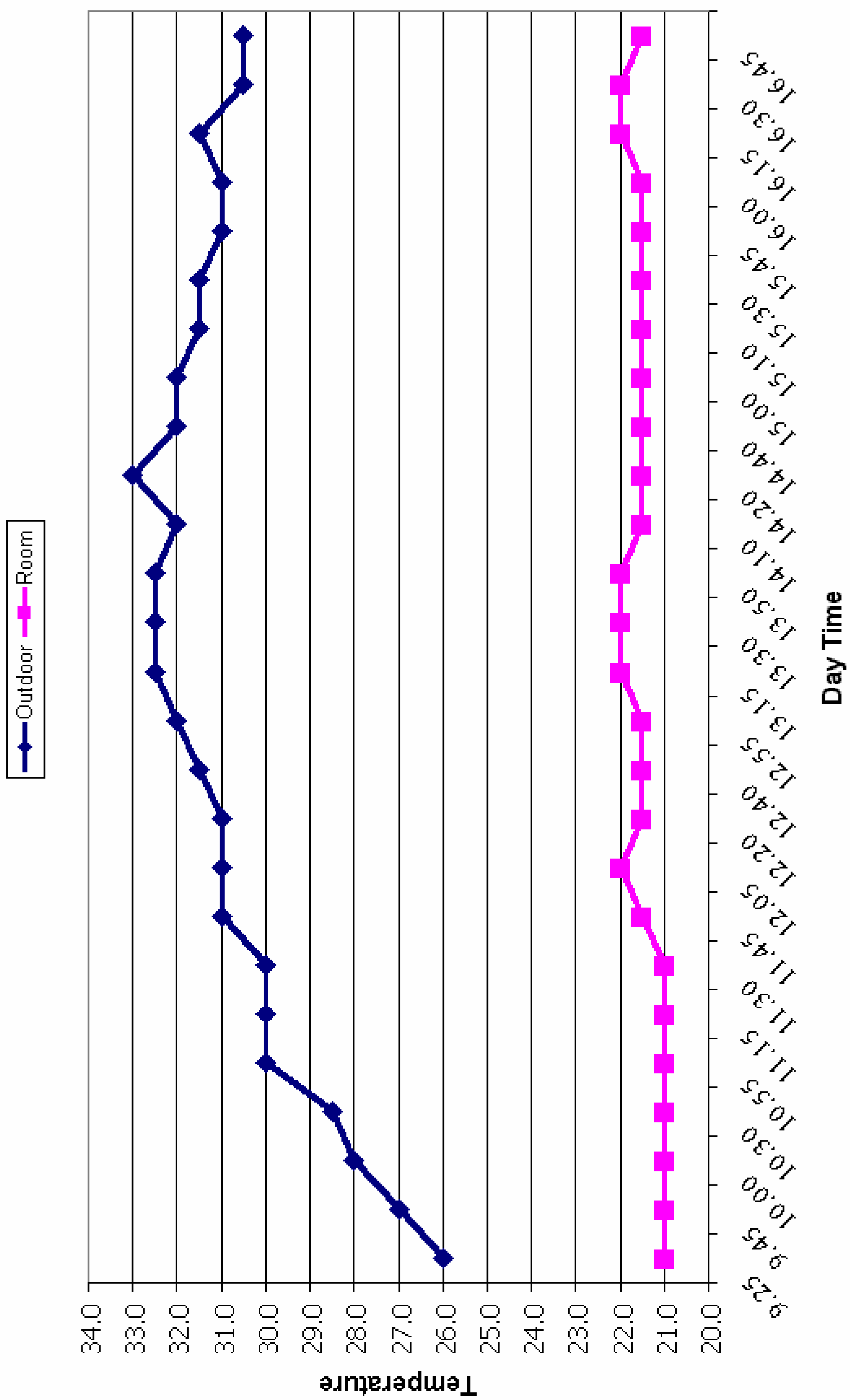


Figure (6.1): Outdoor and room temperatures versus time

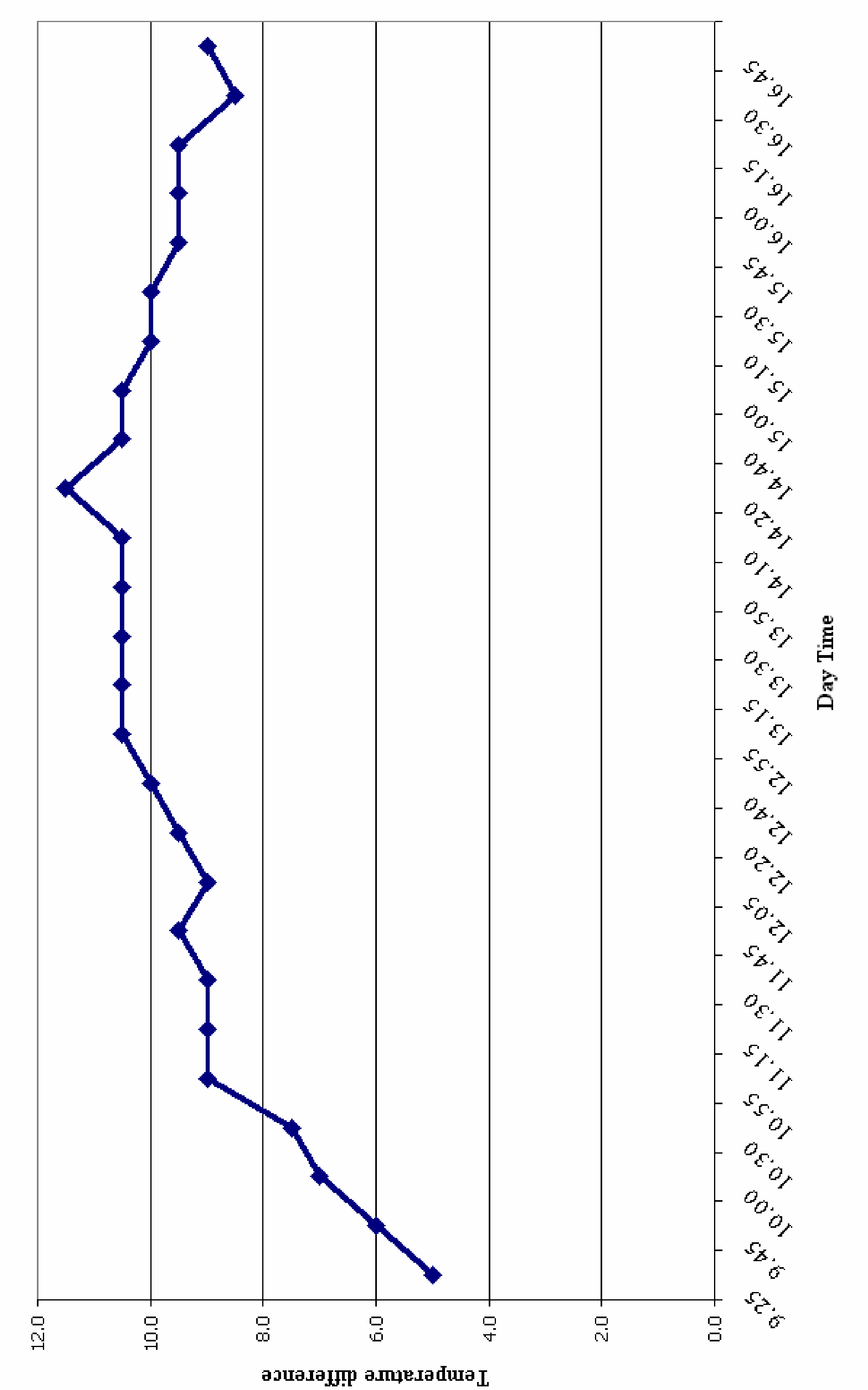


Figure (6.2): Difference between outdoor and room temperature versus time

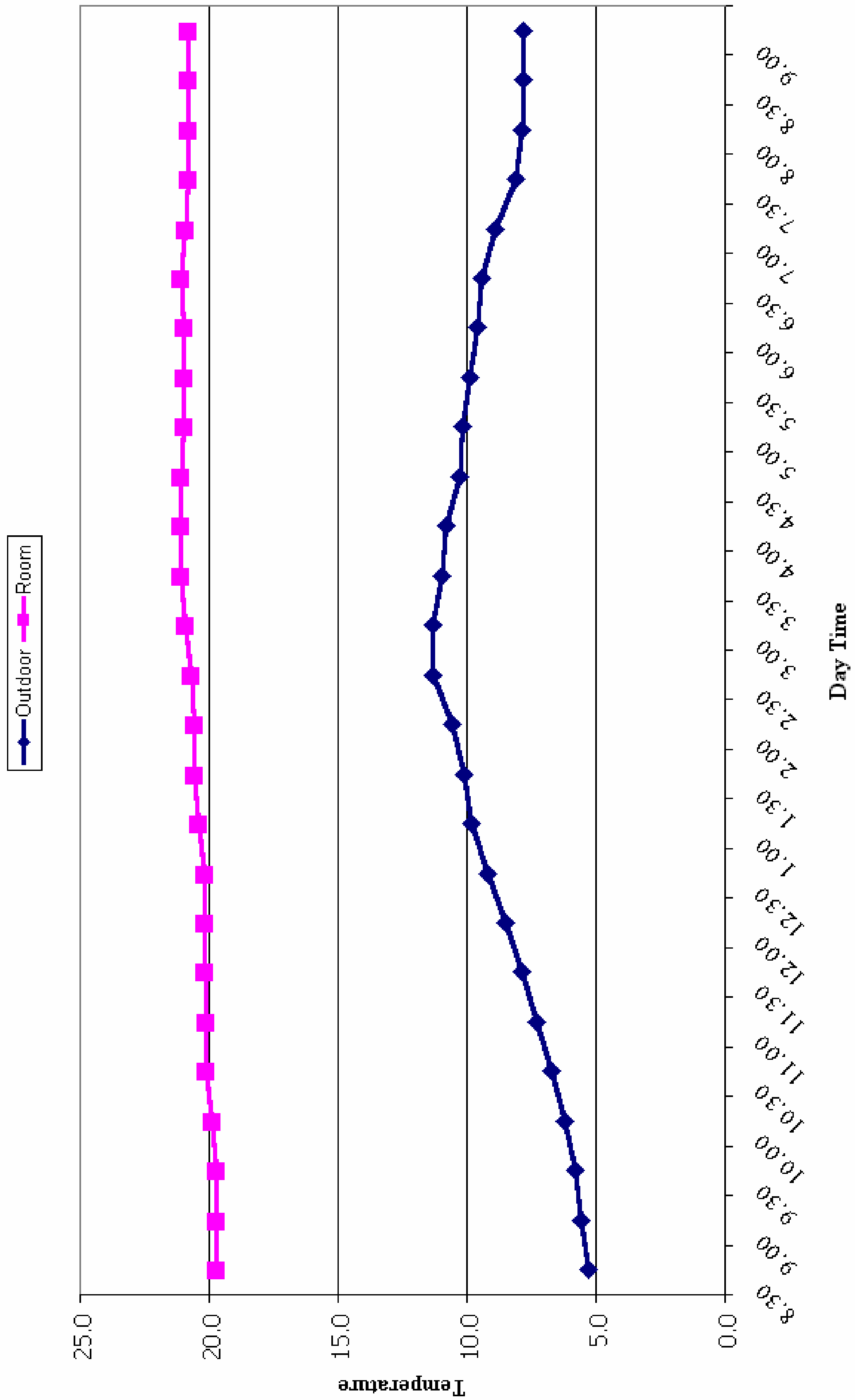


Figure (6.3): Outdoor and room temperatures versus time (heating)

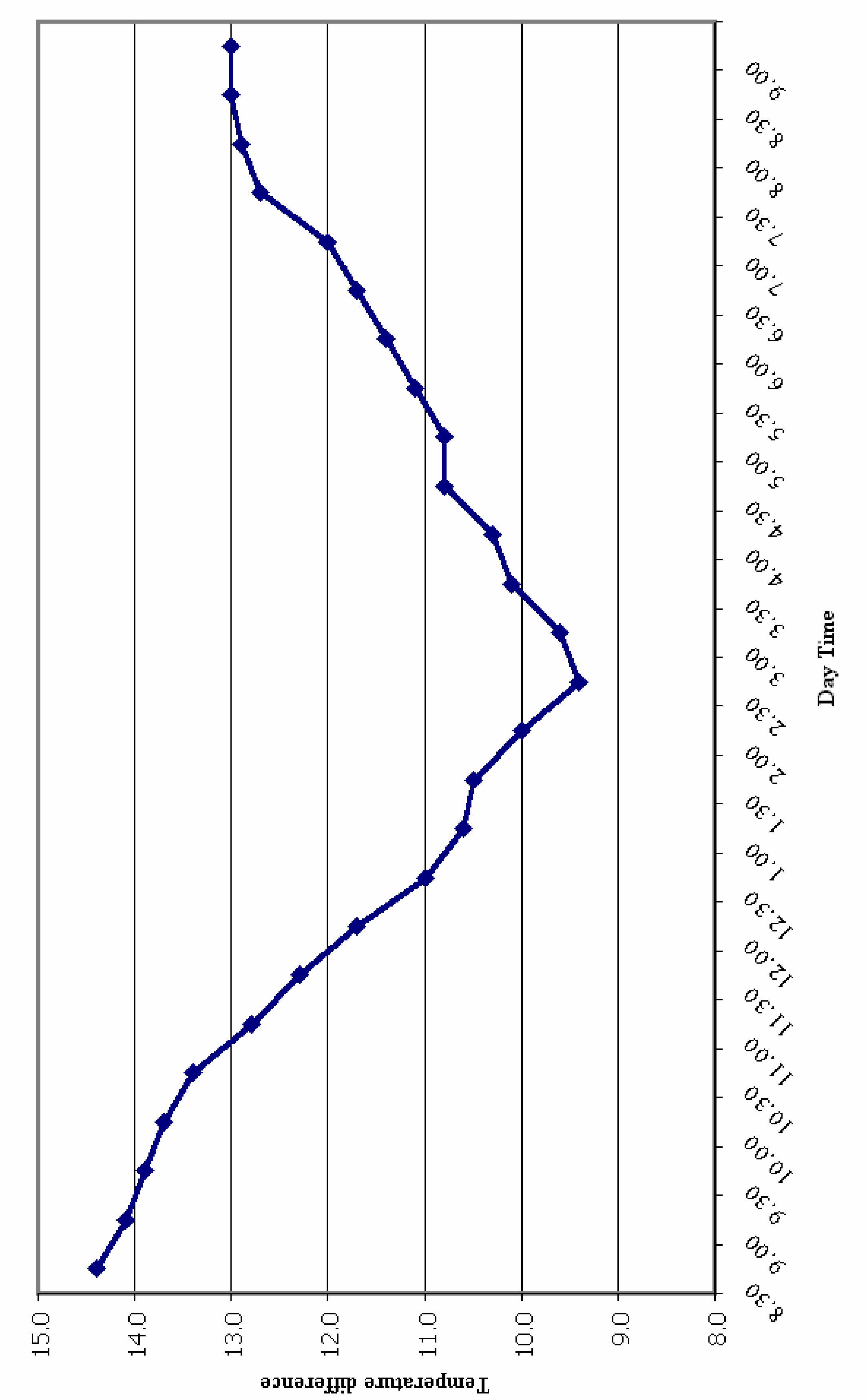


Figure (6.4): Difference between outdoor and room temperature versus time (heating)

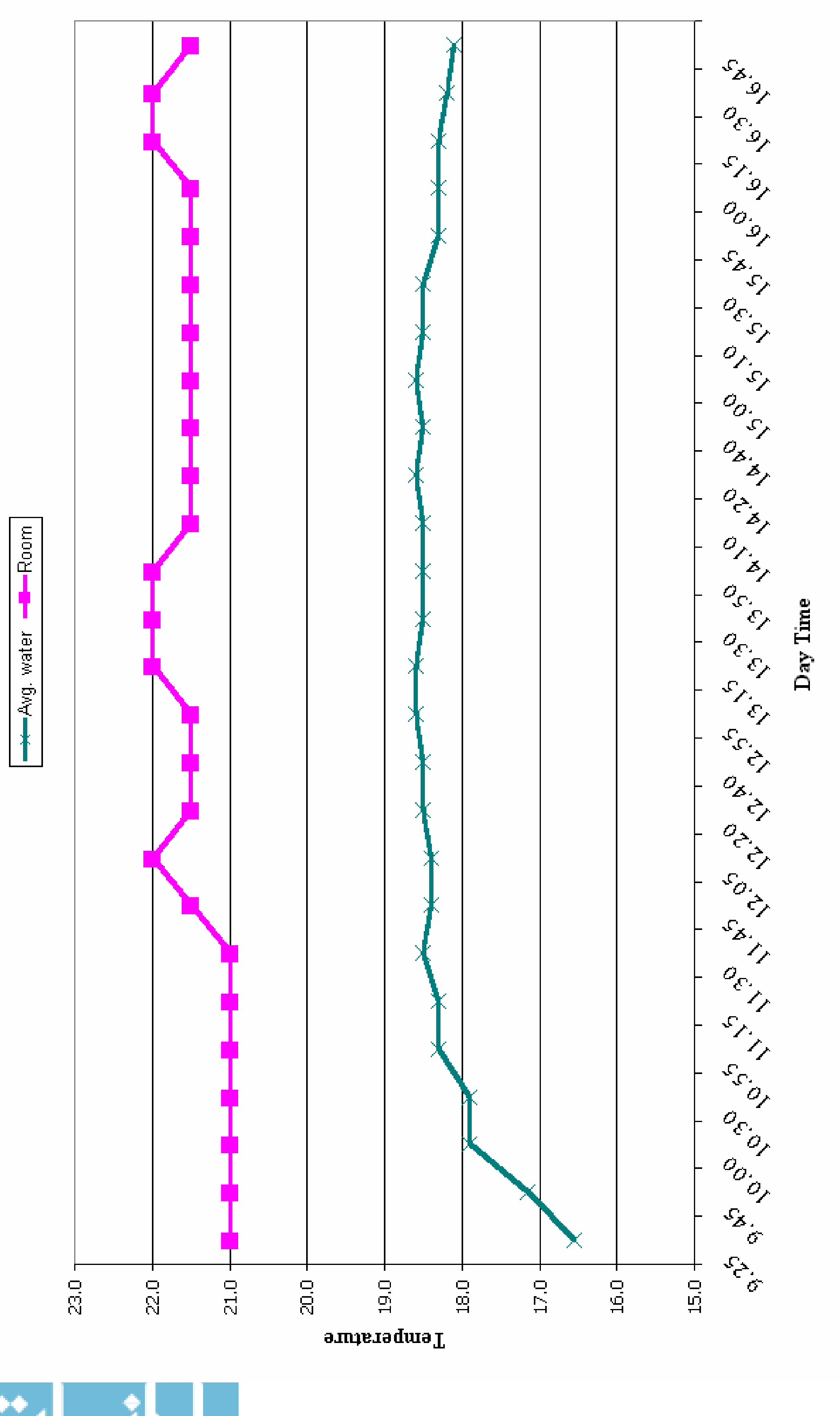


Figure (6.5): Average water and room temperatures versus time

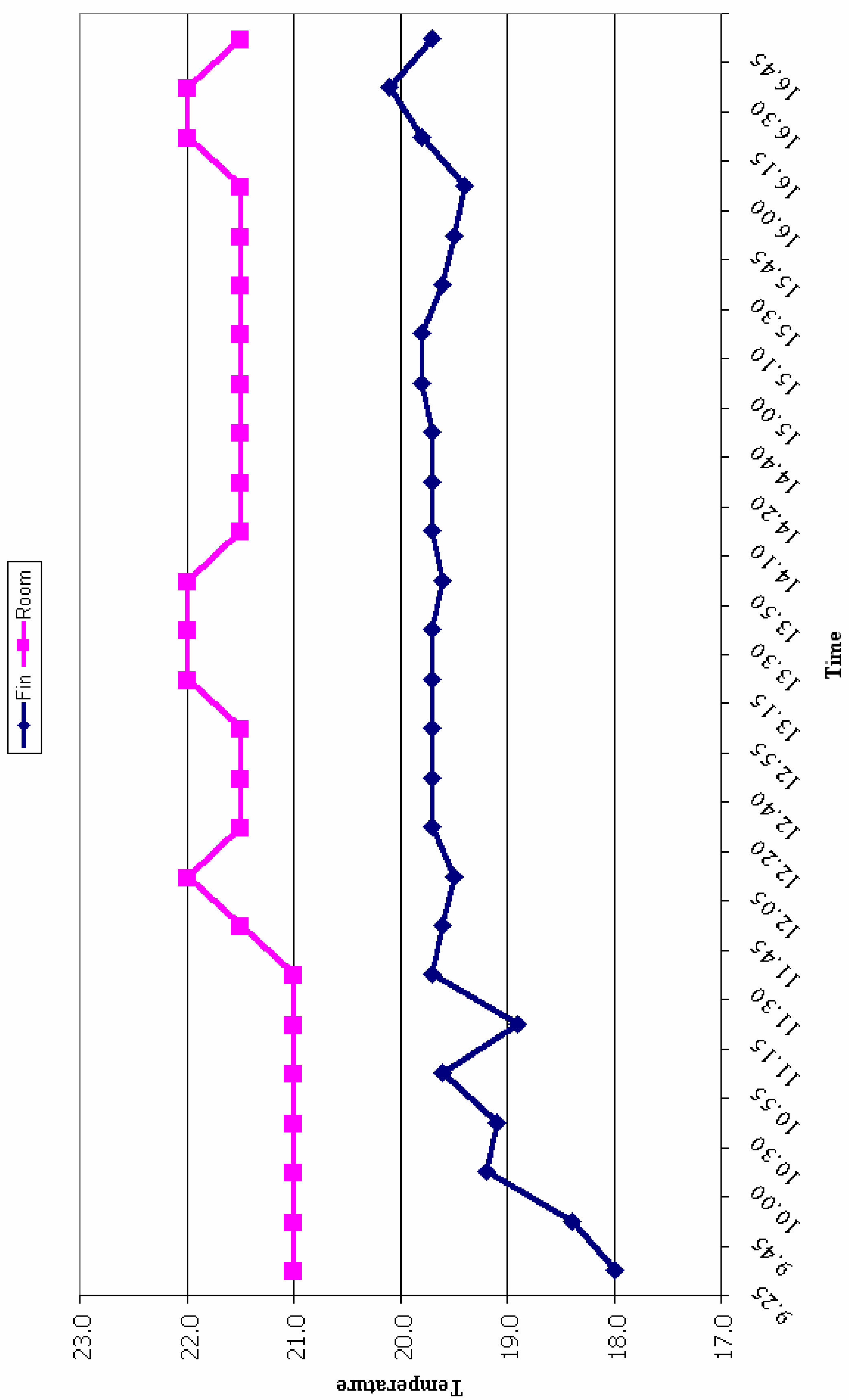


Figure (6.6): Room and fin temperatures versus time

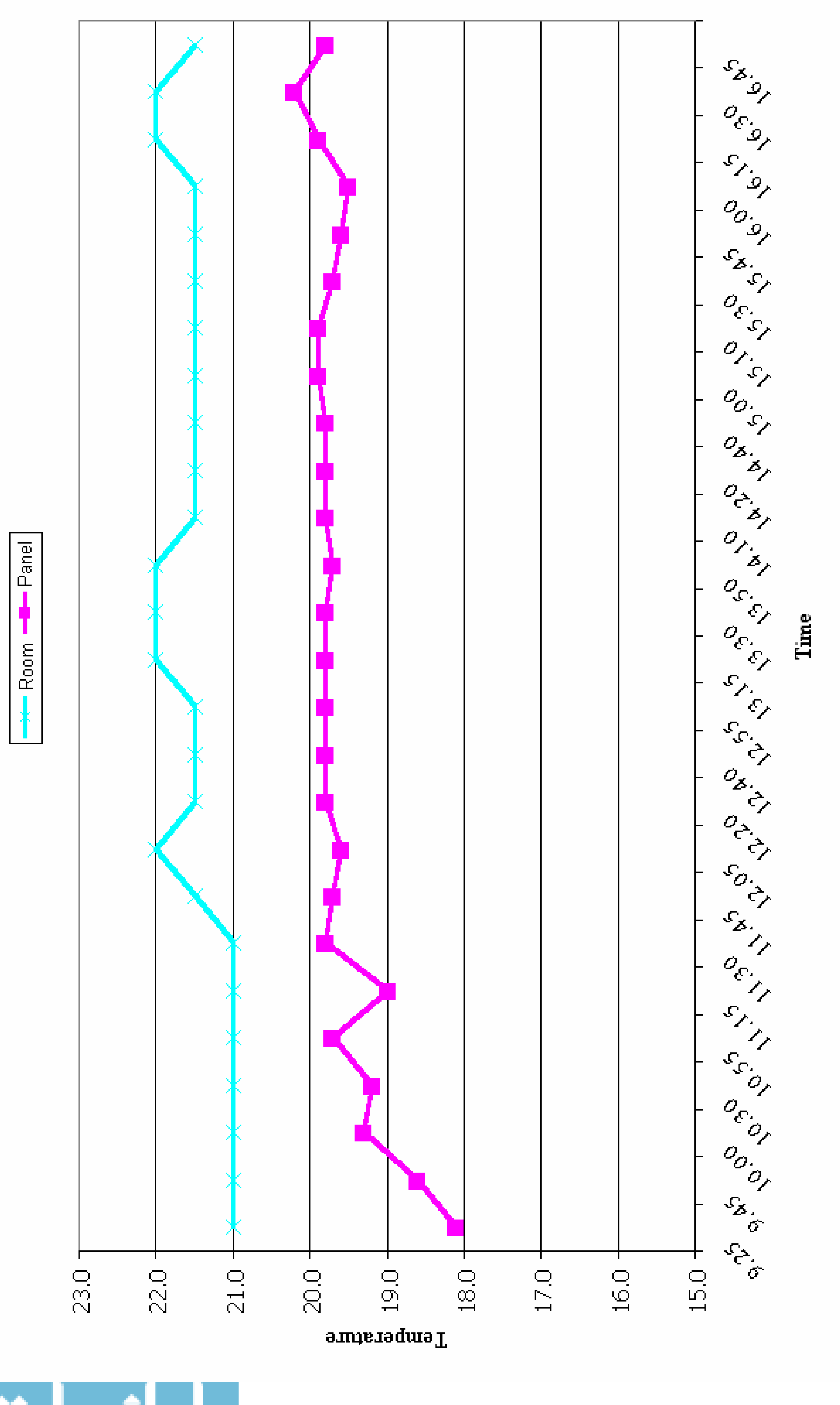


Figure (6.7): Room and panel temperatures versus time

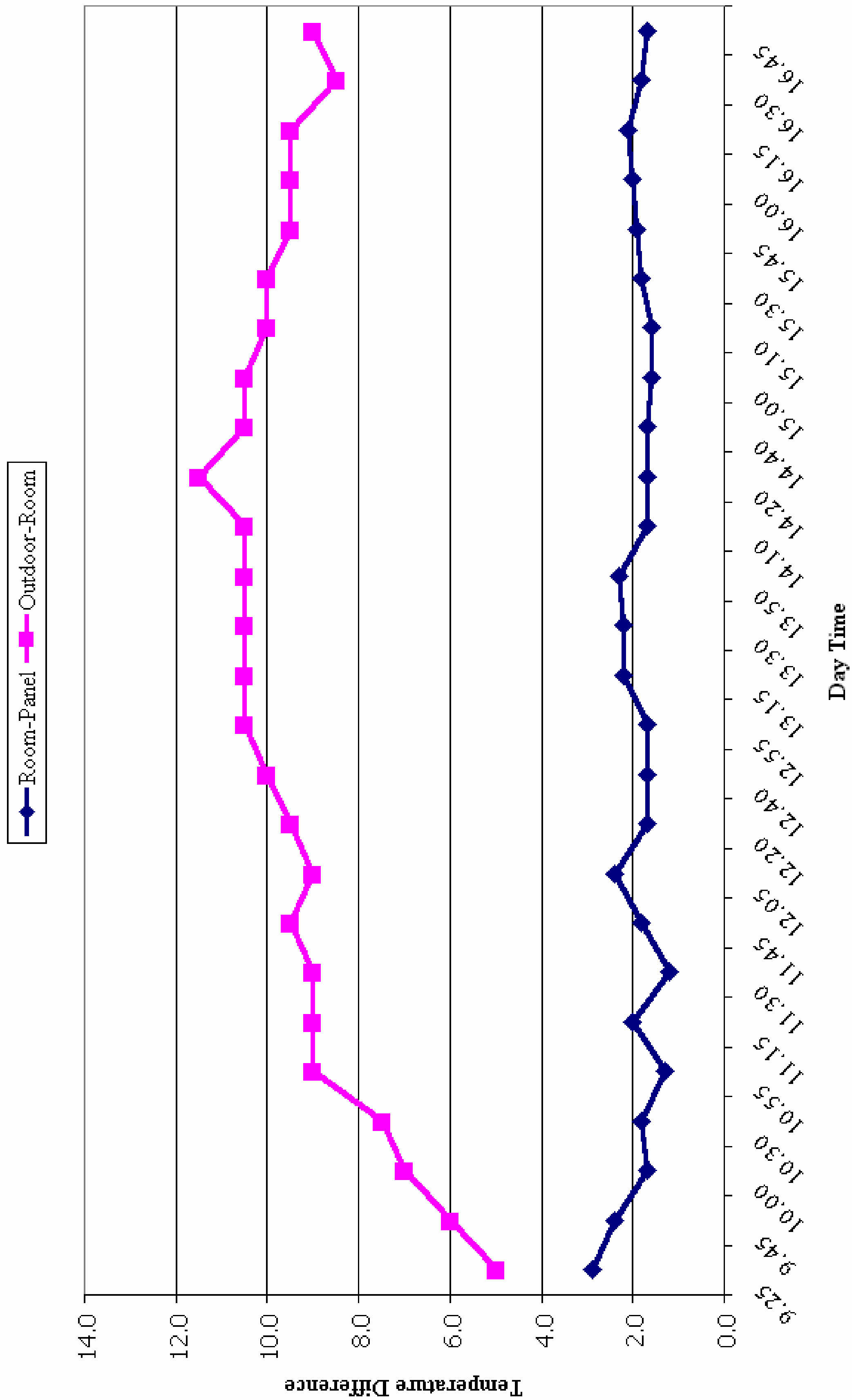


Figure (6.8): Difference between outdoor-room and room-panel temperature versus time

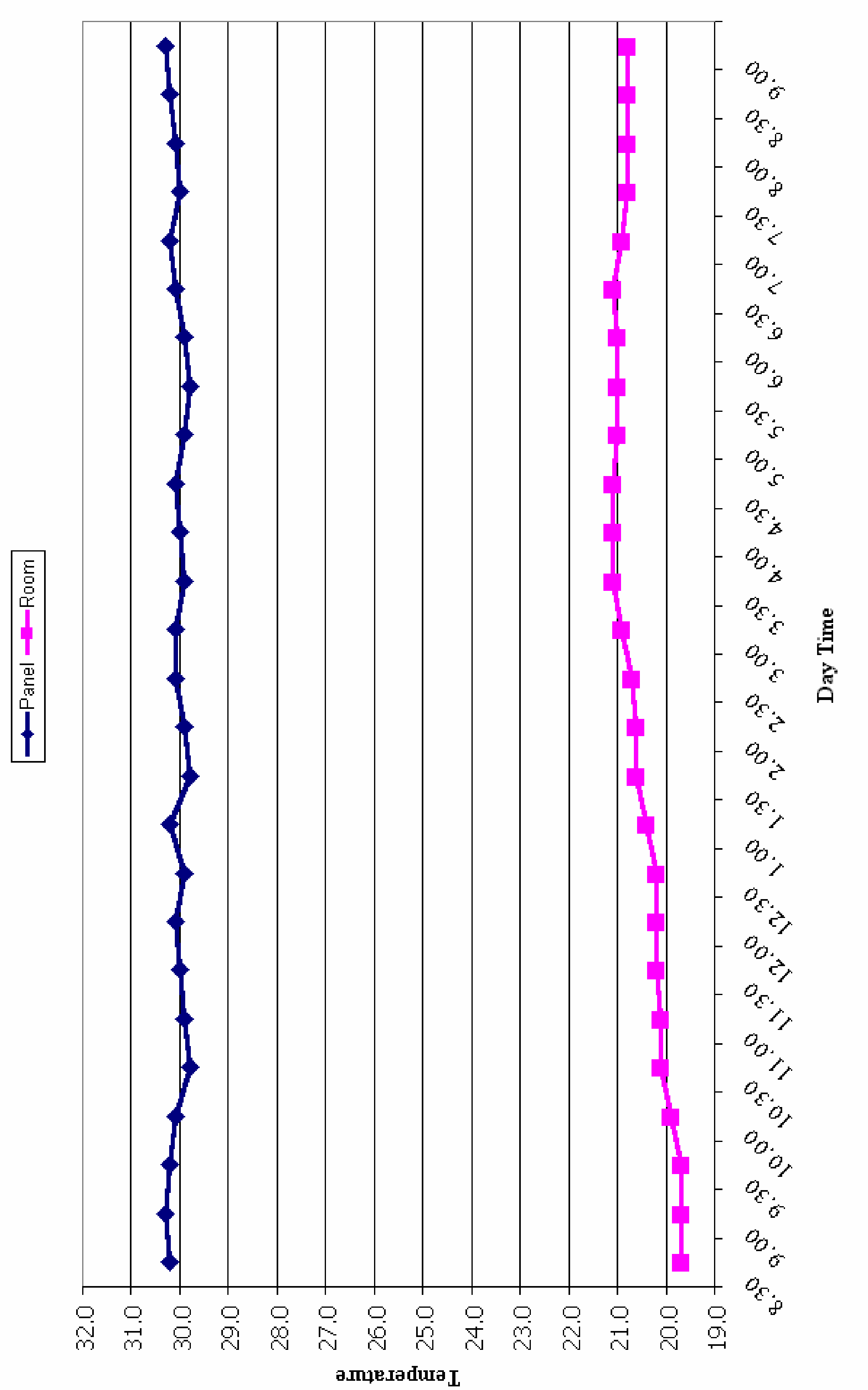


Figure (6.9): Room and panel temperatures versus time

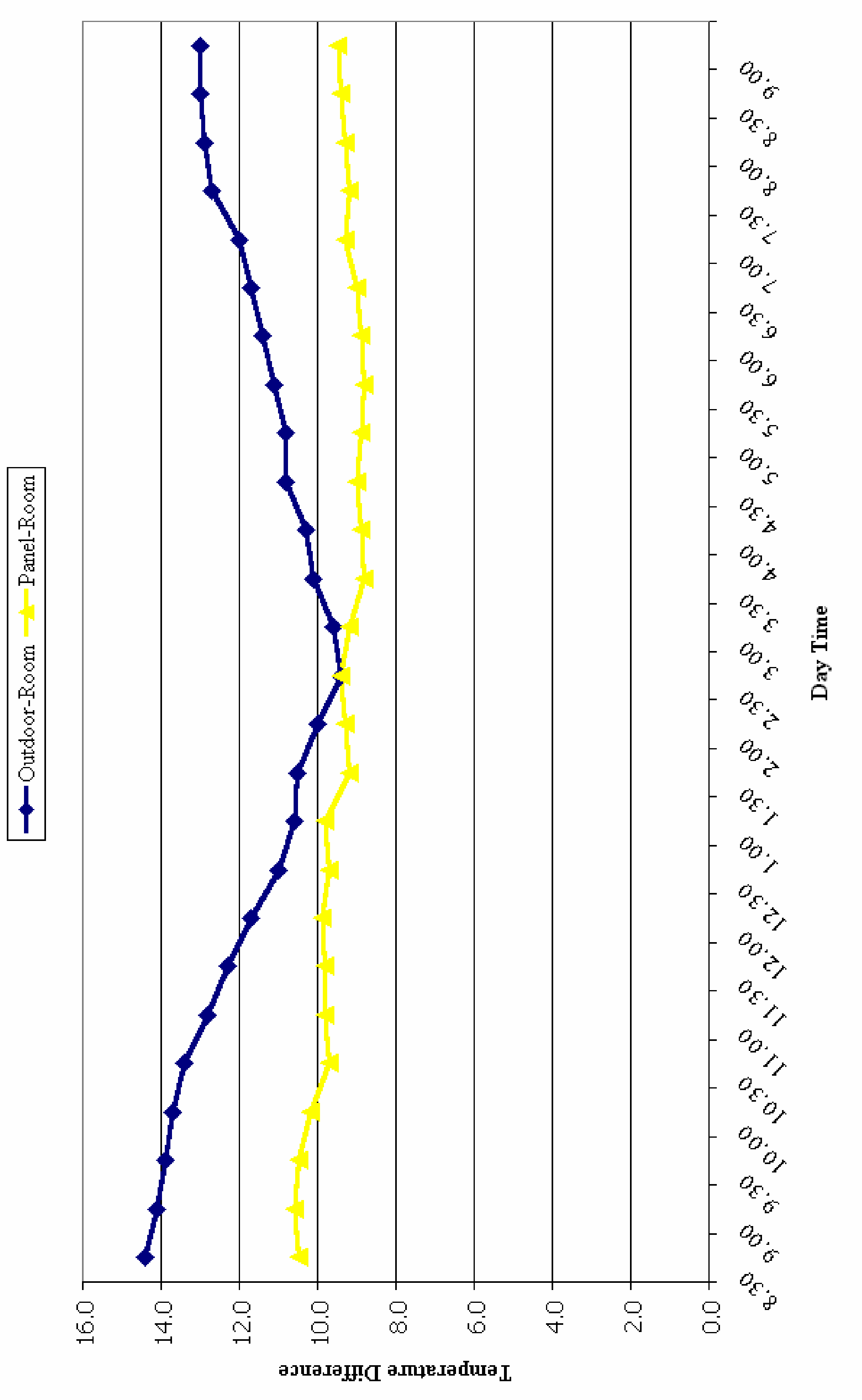


Figure (6.10): Difference between outdoor-room and room-panel temperatures versus time (heating)

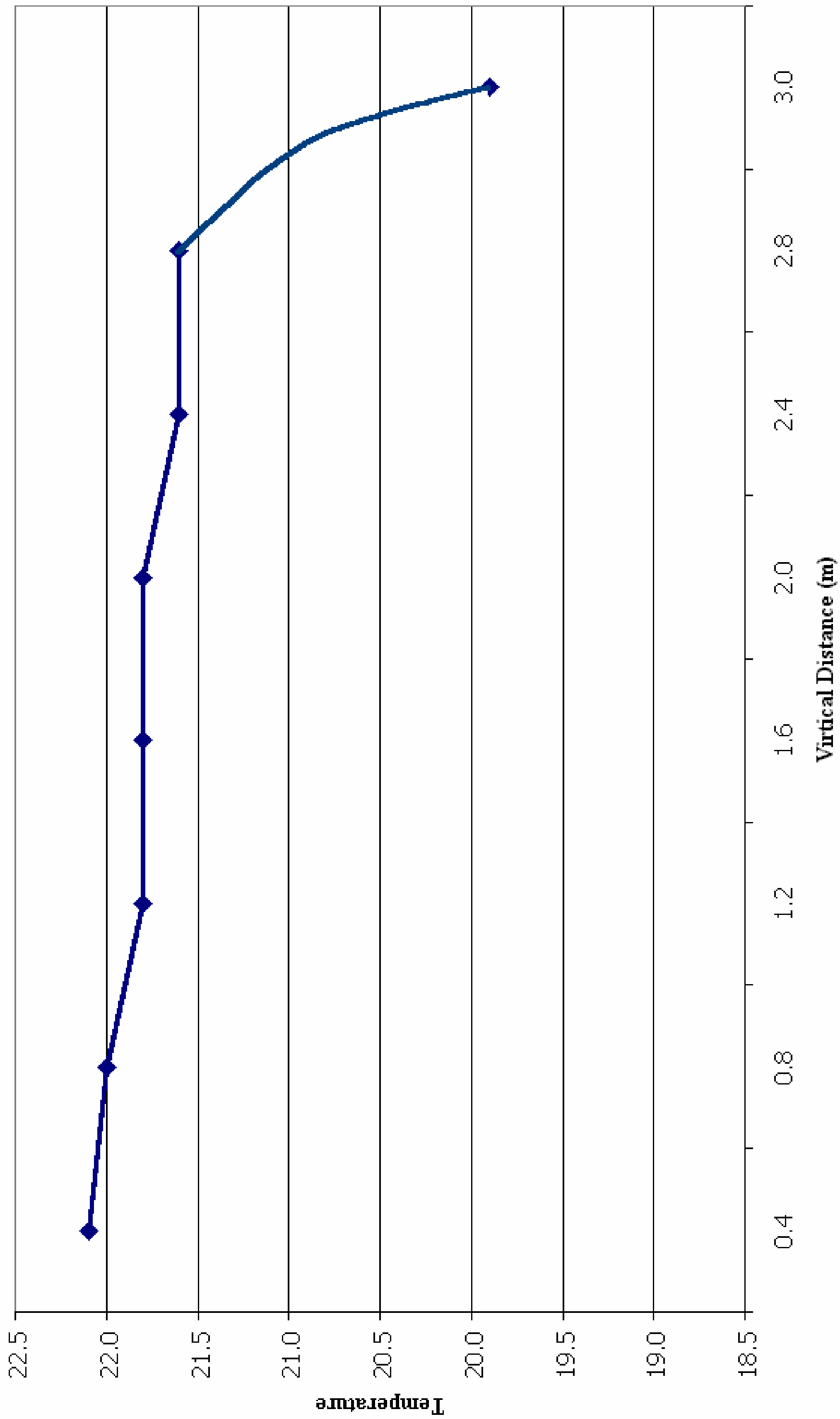


Figure (6.11): Vertical distance versus temperature at point 1

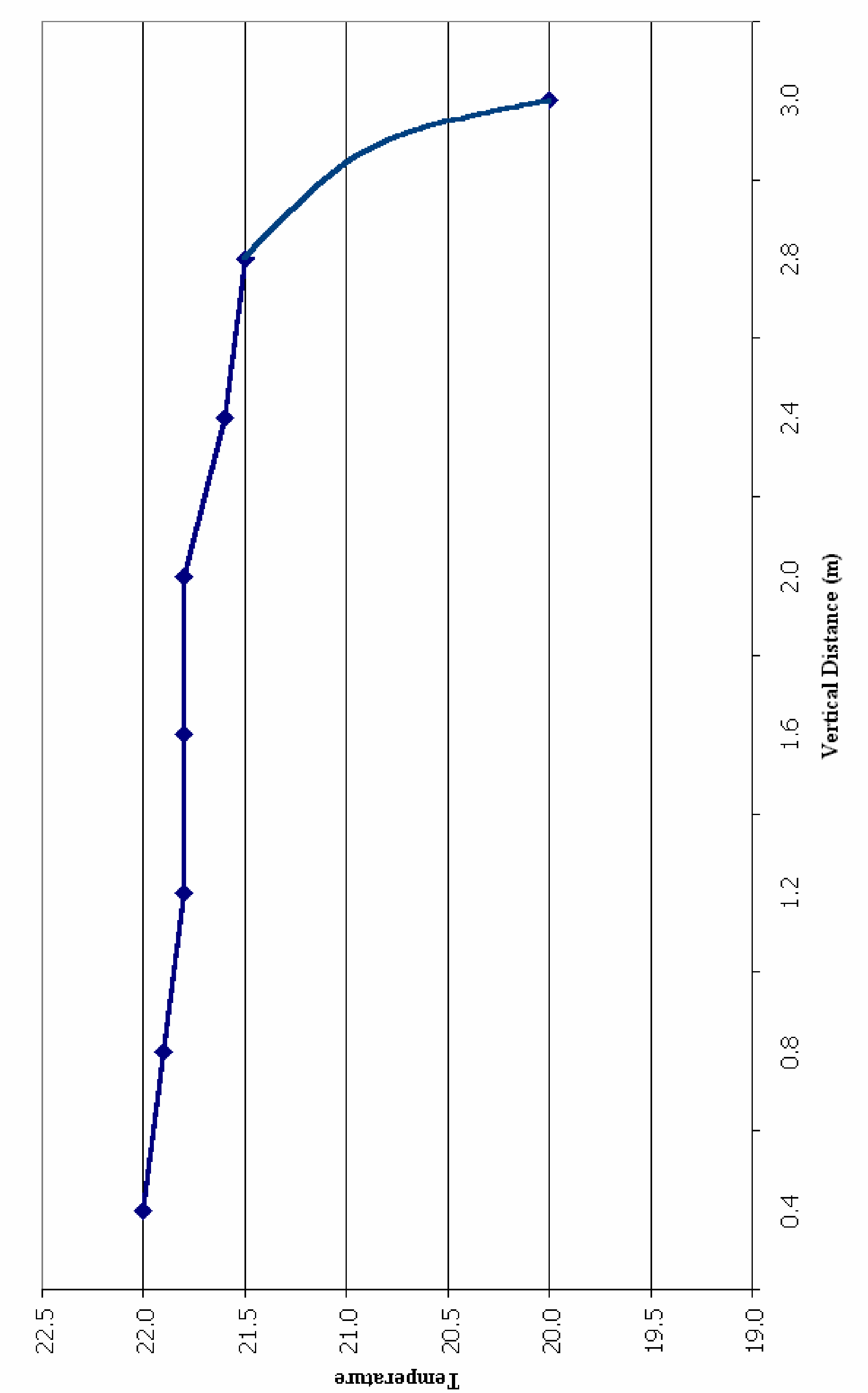


Figure (6.12): Vertical distance versus temperature at point 2

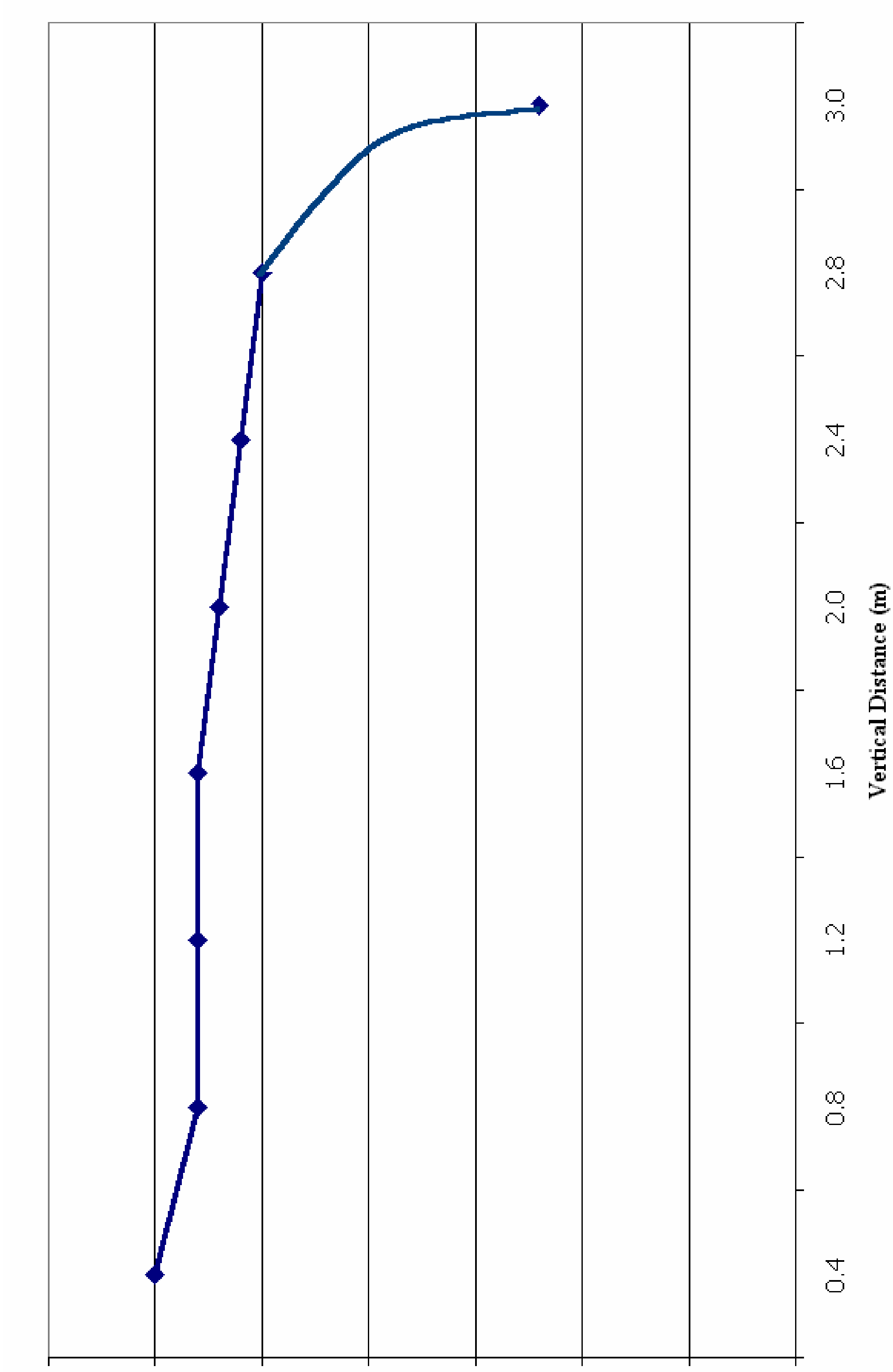


Figure (6.13): Vertical distance versus temperature at point 3

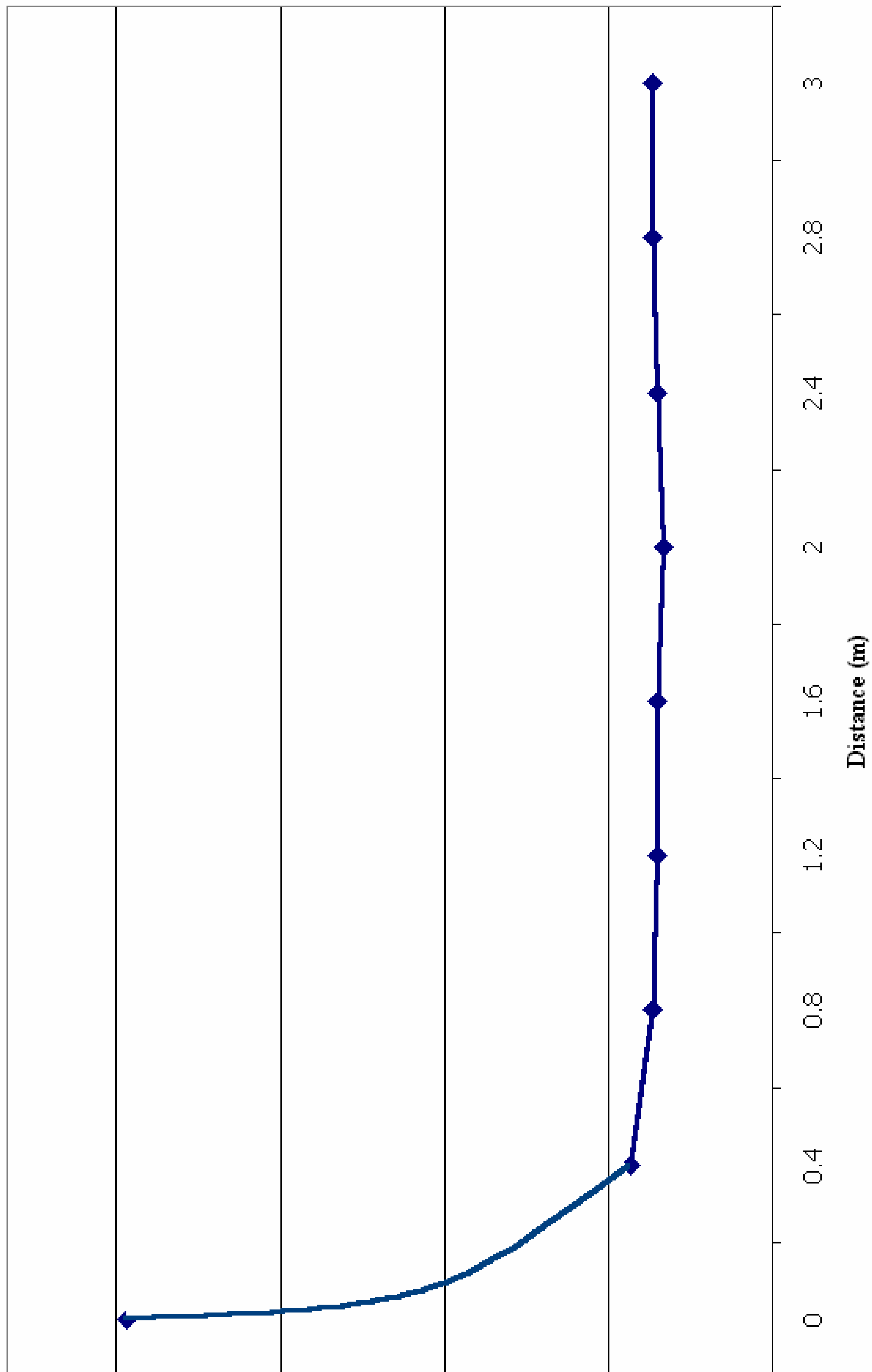


Figure (6.14): Vertical distance versus temperature at point 1 (heating)

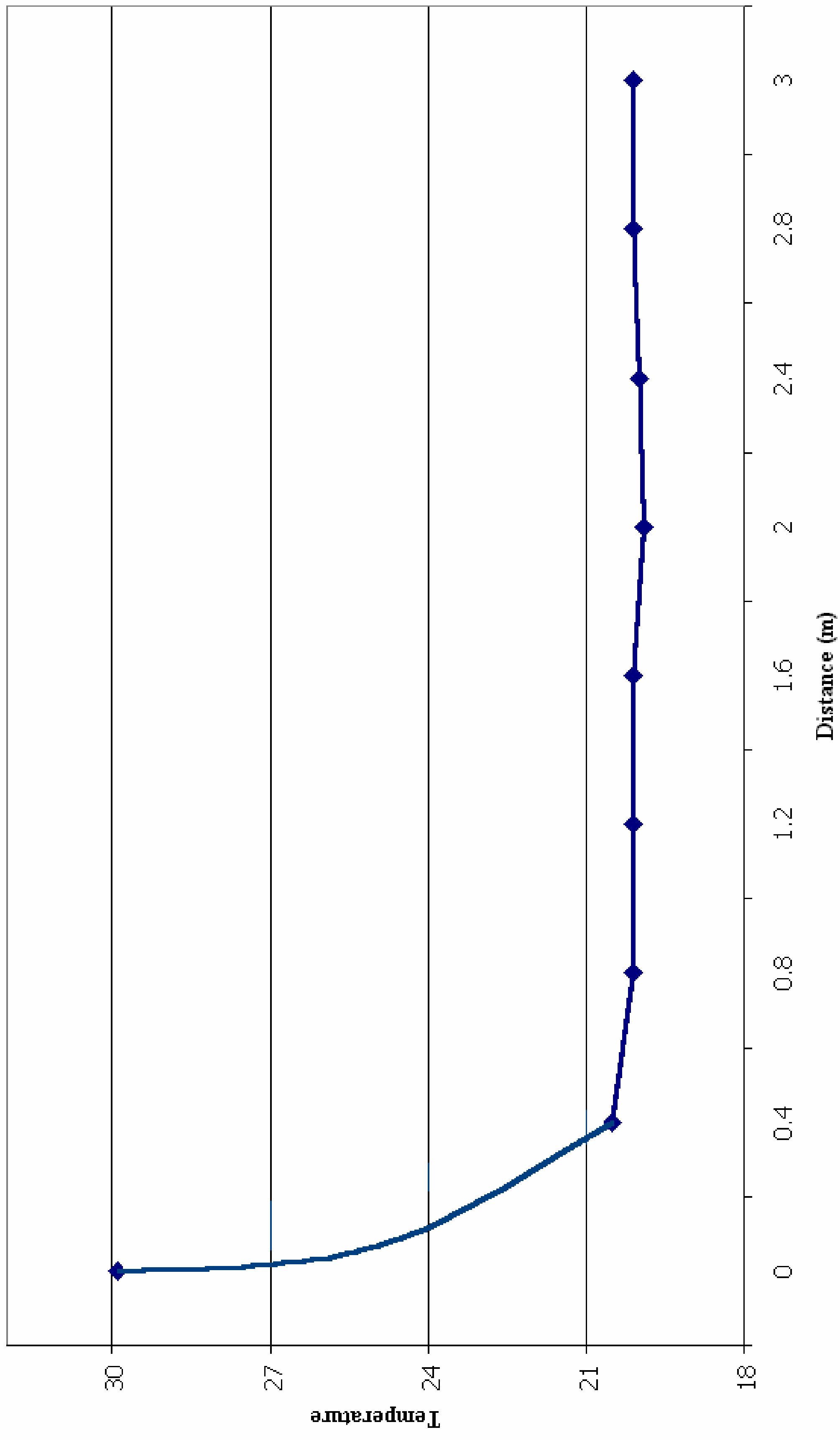


Figure (6.15): Vertical distance versus temperature at point 2 (heating)

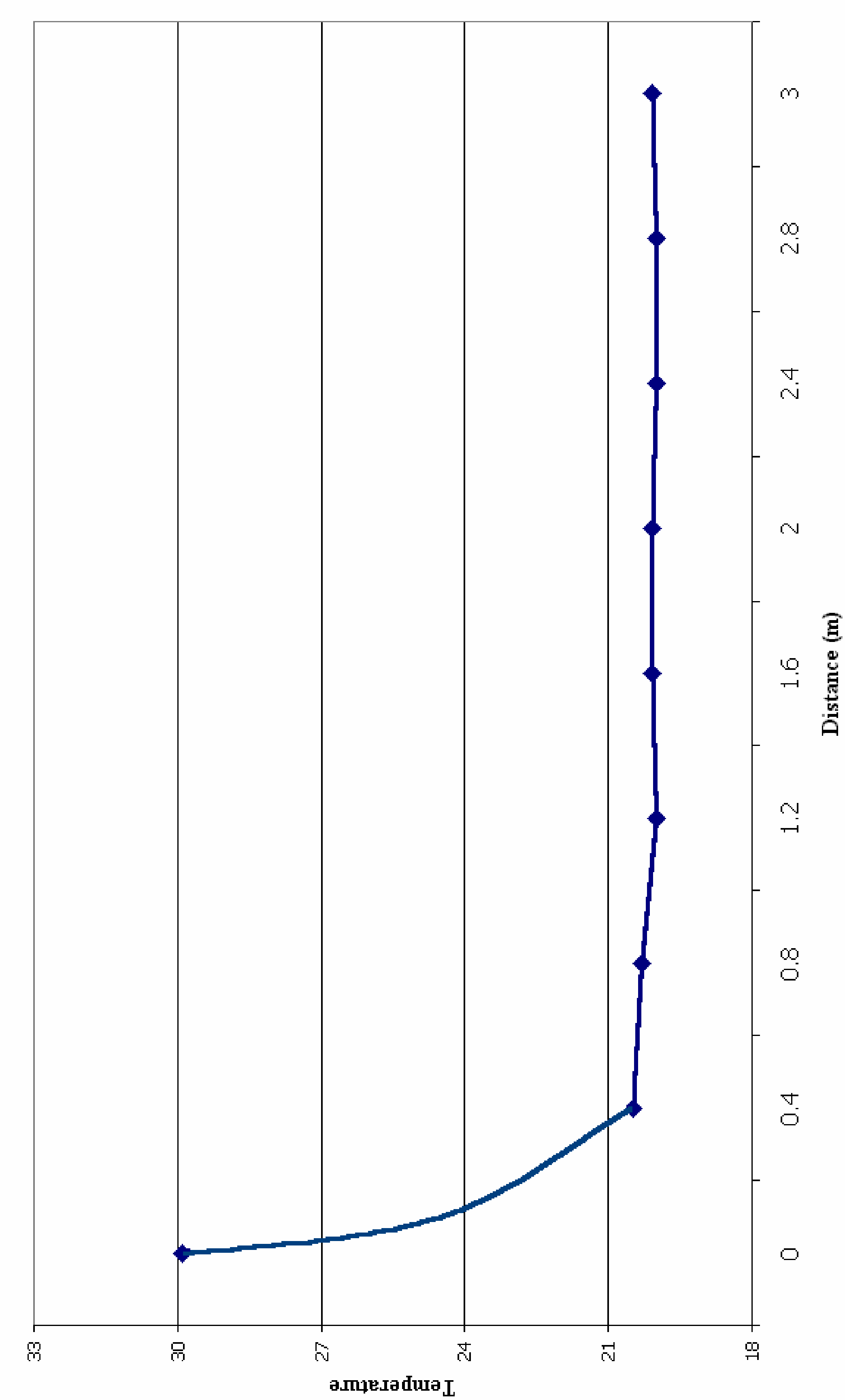


Figure (6.16): Vertical distance versus temperature at point 3 (heating)

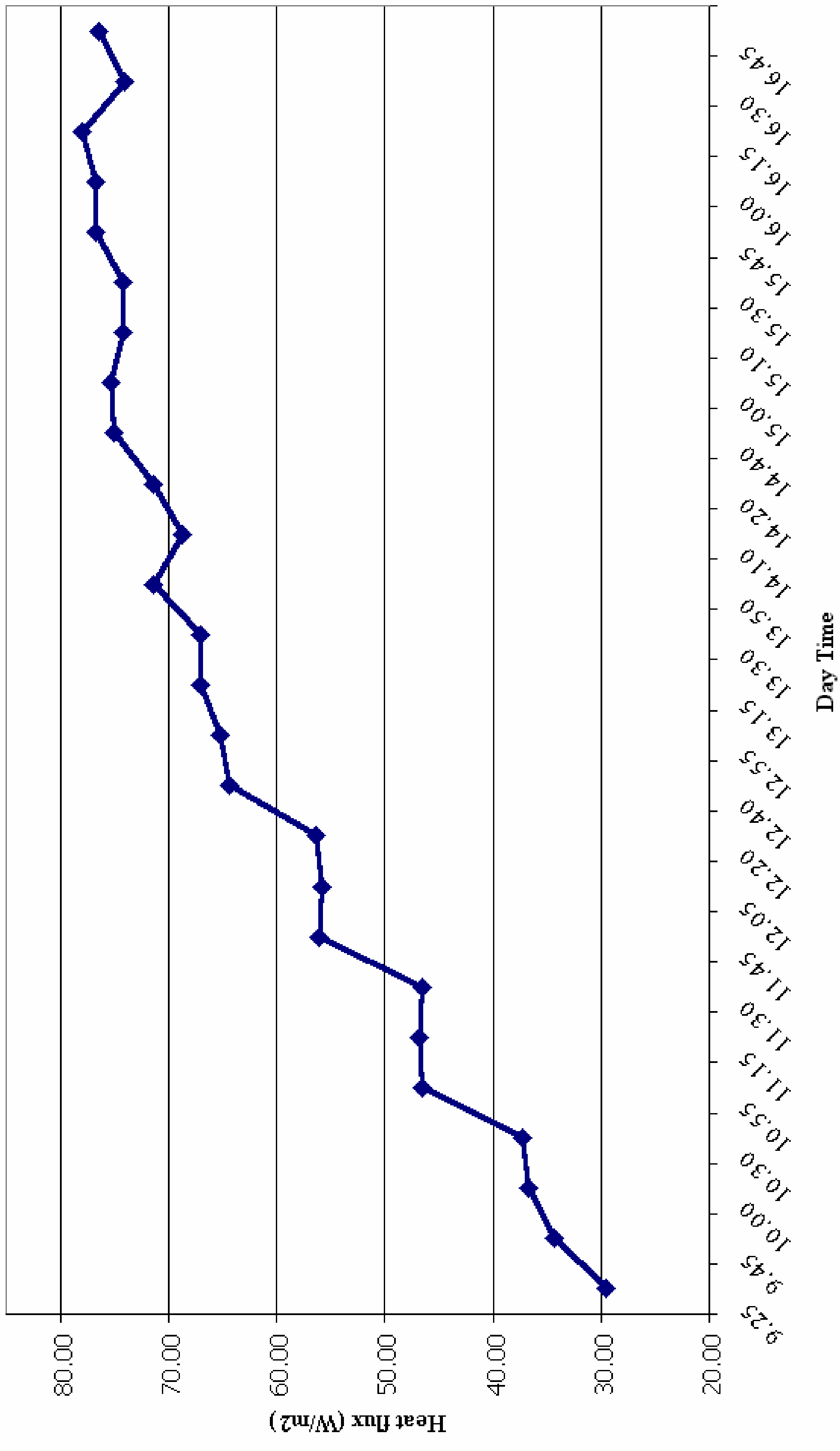


Figure (6.17): Sensible cooling load flux versus time

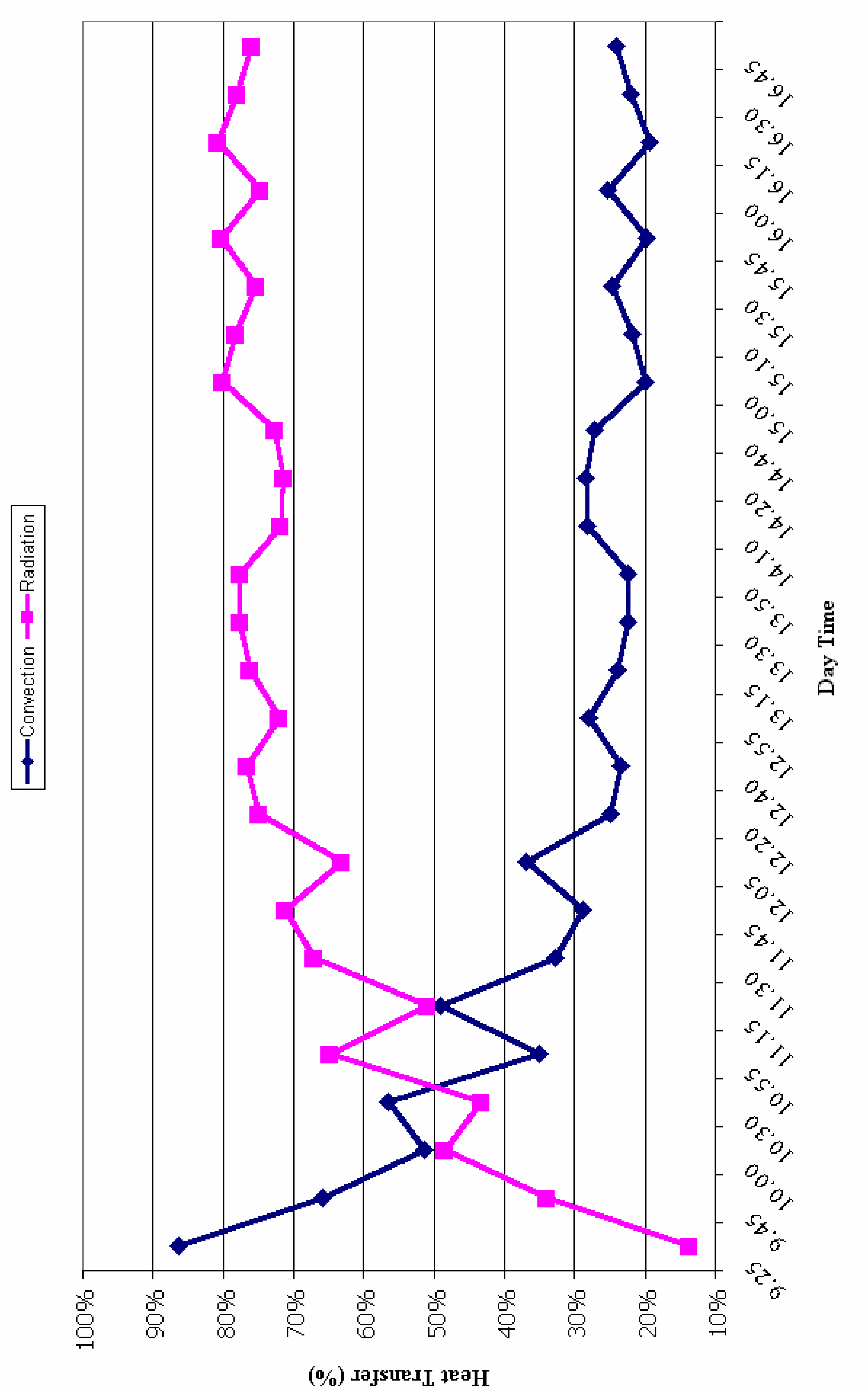


Figure (6.18): Convection and radiation heat transfer ratio versus time

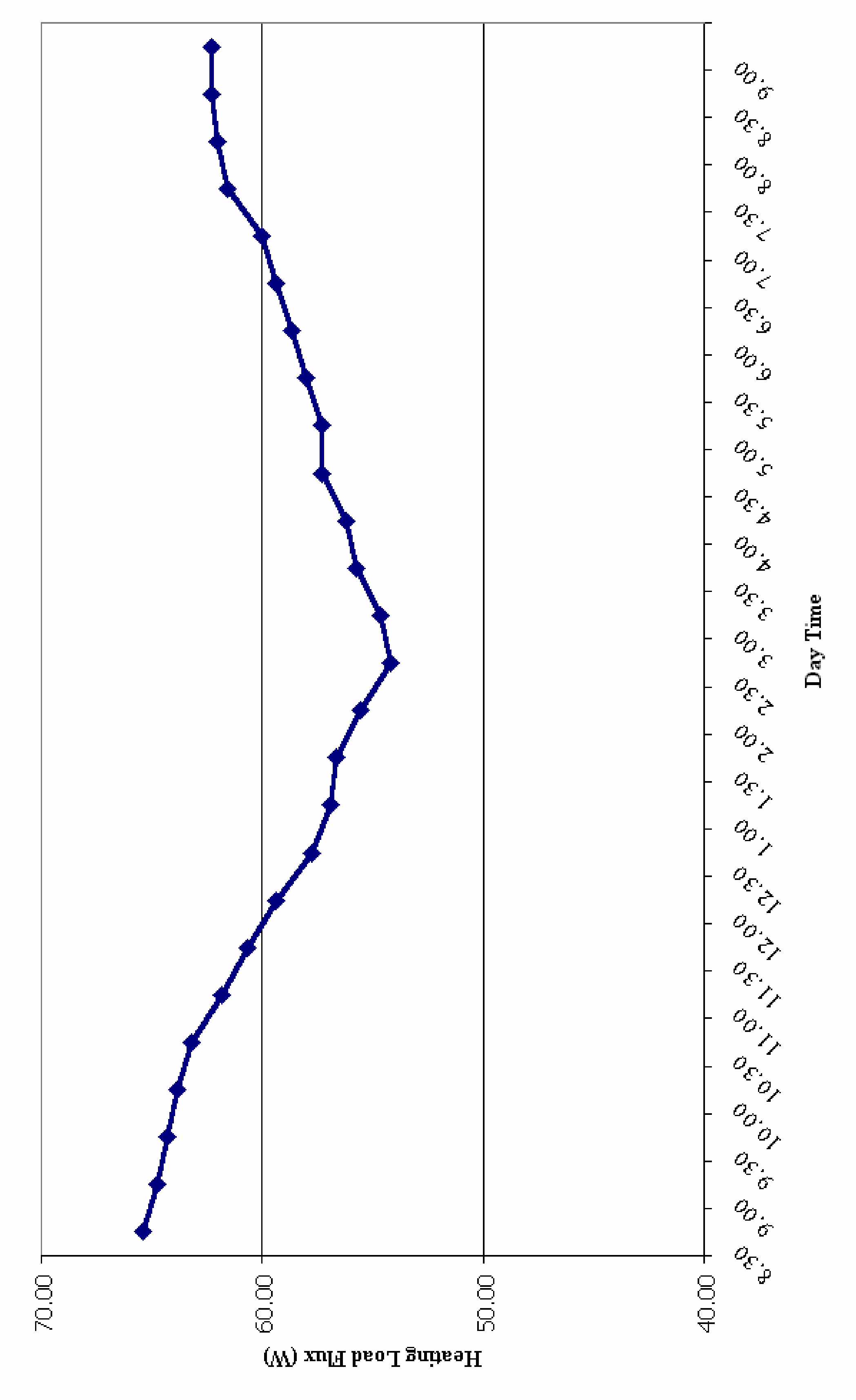


Figure (6.19): Heating load flux versus time

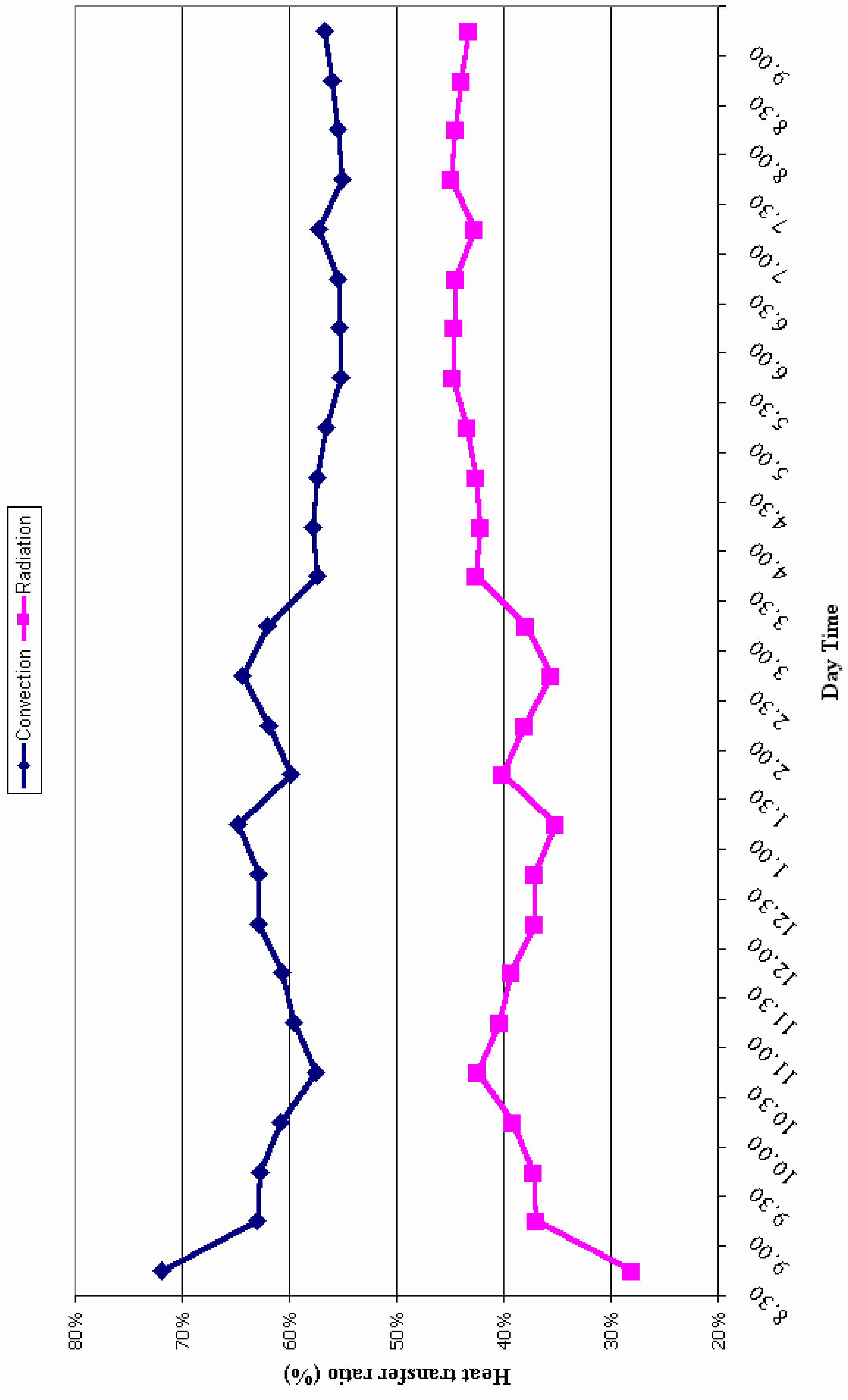


Figure (6.20): Convection and radiation heat transfer ratio versus time (heating)

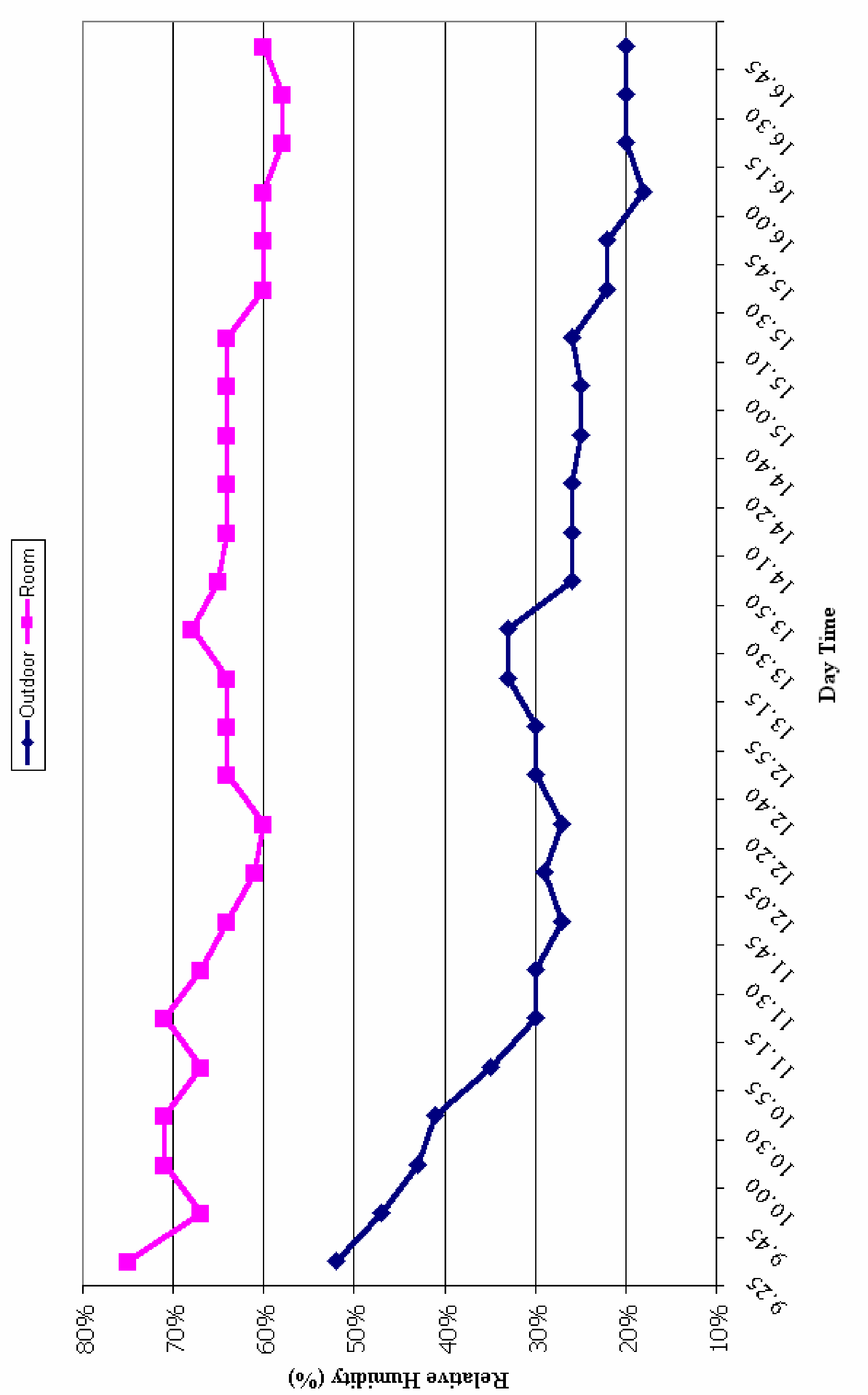


Figure (6.21): Relative humidity versus time

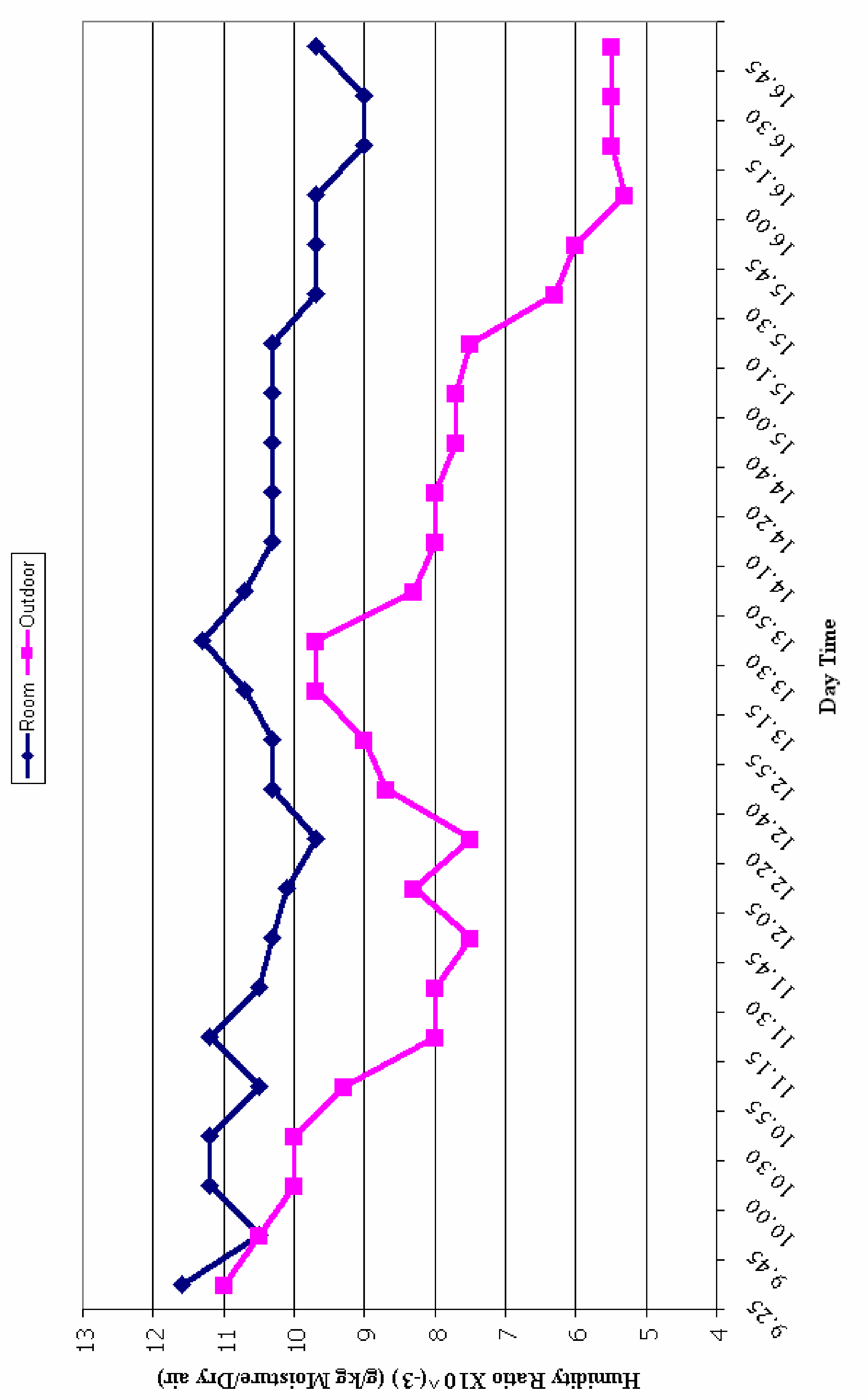


Figure (6.22): Humidity ratio versus time

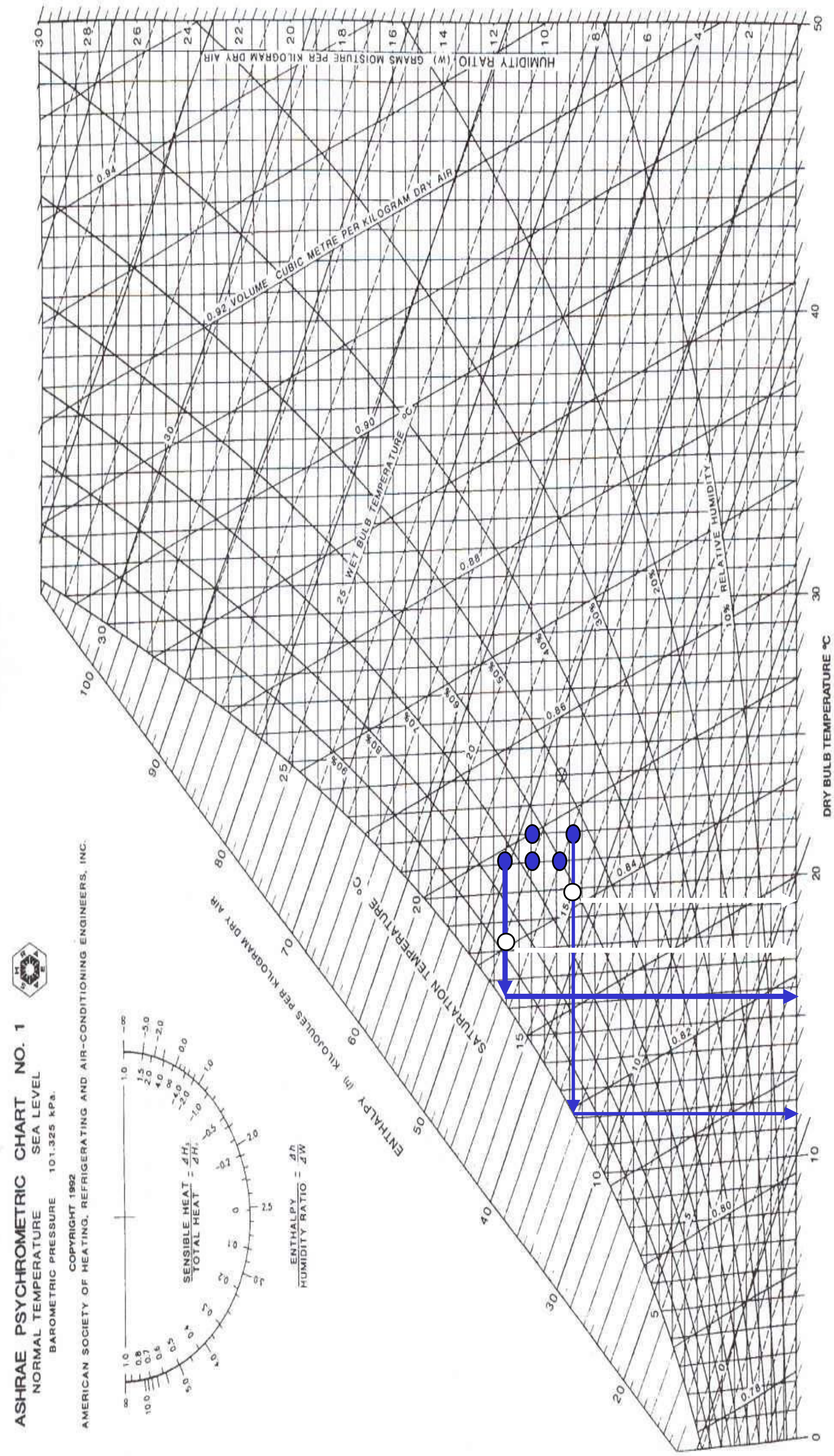


Figure (6.23): Psychometric chart

CHAPTER SEVEN

CONCLUSIONS AND RECOMMENDATIONS

7.1 Conclusions

1. The radiant panel system was successfully operated and tested. It has been found applicable and compatible to weather in Jordan for both heating and cooling. Ceiling panel cooling can easily absorb the sensible cooling load and the floor heating system can subsidy losses during heating mode.

2. In cooling mode, the system operates at water temperature above the dew point where no condensation problems were raised, moreover, average water temperature of 18 ˚ C was found sufficient to provide a panel temperature capable to absorb the sensible cooling load. Whereas in heating mode, an average water temperature of about 48 ˚ C is able to maintain floor temperature that provide space with enough amount of heat to cover the heat losses.

3. An average heat flux of 78 W/m^2 is easily achievable using 75% of the ceiling area in cooling mode, and for higher value of heat flux, increasing the effective area and decreasing the average water temperature could be an effective tools. Whereas in heating, heat flux of 65 W/m^2 was easily covered by the floor panel.

4. The vertical temperature gradient was found to be according to international standards for both heating and cooling that provides occupancies with high comfort level.

7.2 Recommendations

1. The radiant panel system was tested under natural ventilation conditions. Using a Dedicated Outdoor Air System or a Displacement Ventilation System will widen the capacity of the system.

2. The system was used for heating through floor and cooling through ceiling. Consider other researches to study floor cooling and ceiling heating will help in gaining popularity for the system since under floor heating system is well known in Jordan, and switching these systems to floor cooling will definitely improve the cooling market.

3. Testing the effective of hot water from solar cells would decrease the capital cost of the system.

4. Comparing capital cost and running cost between radiant panel system and other conditioning systems will improve occupant's confidence.

5. Developing a mathematical model to estimate panel temperature would empower the experimental results.

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Appendix A

Cooling Mode Data

(A-1) Readings for room, outdoor, panel, fins and pipe temperatures during cooling mode.

Day Time	Room Dry Bulb	Room Wet Bulb	Outdoor Dry Bulb	Outdoor Wet Bulb	Pipe	Panel	Fin
9.25	21.0	18.0	26.0	19.0	17.4	18.1	18.0
9.45	21.0	17.0	27.0	19.0	18.3	18.6	18.4
10.00	21.0	17.5	28.0	19.0	19.1	19.3	19.2
10.30	21.0	17.5	28.5	19.0	19.0	19.2	19.1
10.55	21.0	17.0	30.0	19.0	19.5	19.7	19.6
11.15	21.0	17.5	30.0	18.0	18.8	19.0	18.9
11.30	21.0	17.0	30.0	18.0	19.6	19.8	19.7
11.45	21.5	17.0	31.0	18.0	19.5	19.7	19.6
12.05	22.0	17.0	31.0	18.5	19.4	19.6	19.5
12.20	21.5	16.5	31.0	18.0	19.5	19.8	19.7
12.40	21.5	17.0	31.5	19.0	19.6	19.8	19.7
12.55	21.5	17.0	32.0	19.5	19.6	19.8	19.7
13.15	22.0	17.5	32.5	20.0	19.6	19.8	19.7
13.30	22.0	18.0	32.5	19.0	19.6	19.8	19.7
13.50	22.0	17.5	32.5	19.0	19.5	19.7	19.6
14.10	21.5	17.0	32.0	18.5	19.6	19.8	19.7
14.20	21.5	17.0	33.0	19.0	19.6	19.8	19.7
14.40	21.5	17.0	32.0	18.0	19.6	19.8	19.7
15.00	21.5	17.0	32.0	18.0	19.7	19.9	19.8
15.10	21.5	17.0	31.5	18.0	19.7	19.9	19.8
15.30	21.5	16.5	31.5	17.0	19.4	19.7	19.6
15.45	21.5	16.5	31.0	16.5	19.2	19.6	19.5
16.00	21.5	16.5	31.0	16.0	19.3	19.5	19.4
16.15	22.0	16.5	31.5	16.5	19.3	19.9	19.8
16.30	22.0	16.5	30.5	16.0	19.3	20.2	20.1
16.45	21.5	16.5	30.5	16.0	19.5	19.8	19.7

(A-2) Readings for supply, return and average water temperatures during cooling mode.

Day Time	Inlet Water	Outlet Water	Average Water
9.25	16.1	17.0	16.6
9.45	16.1	18.2	17.2
10.00	16.0	19.8	17.9
10.30	16.1	19.7	17.9
10.55	16.0	20.6	18.3
11.15	16.0	20.6	18.3
11.30	16.0	21.0	18.5
11.45	16.1	20.7	18.4
12.05	16.0	20.8	18.4
12.20	16.1	20.9	18.5
12.40	16.1	20.9	18.5
12.55	16.1	21.1	18.6
13.15	16.0	21.2	18.6
13.30	16.0	21.0	18.5
13.50	16.1	20.9	18.5
14.10	16.0	21.0	18.5
14.20	16.0	21.2	18.6
14.40	16.1	20.9	18.5
15.00	16.1	21.1	18.6
15.10	16.1	20.9	18.5
15.30	16.0	21.0	18.5
15.45	16.0	20.6	18.3
16.00	16.0	20.6	18.3
16.15	16.1	20.5	18.3
16.30	16.1	20.3	18.2
16.45	16.0	20.2	18.1

Appendix B

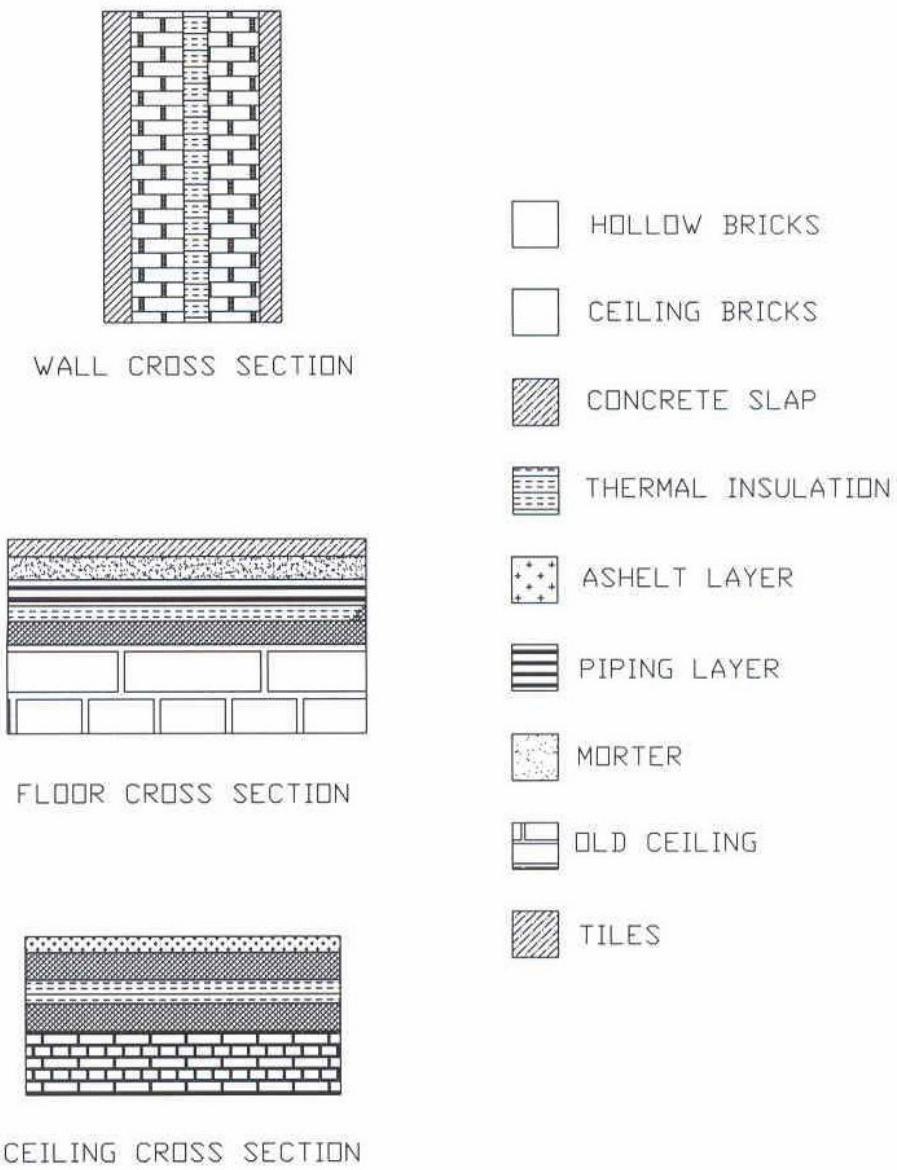
Heating Mode Data

(B-1) Readings for Room, Outdoor and panel temperatures during heating mode

Day Time	Room	Outdoor	Panel	Inlet Water	Outlet Water	Average water
8.30	19.7	5.3	30.2	50.2	43.7	47.0
9.00	19.7	5.6	30.3	50.1	43.8	47.0
9.30	19.7	5.8	30.2	50.1	44.2	47.2
10.00	19.9	6.2	30.1	50.0	44.3	47.2
10.30	20.1	6.7	29.8	50.1	44.3	47.2
11.00	20.1	7.3	29.9	50.2	44.5	47.4
11.30	20.2	7.9	30.0	50.2	44.9	47.6
12.00	20.2	8.5	30.1	50.3	44.8	47.6
12.30	20.2	9.2	29.9	50.1	44.9	47.5
1.00	20.4	9.8	30.2	50.2	45.1	47.7
1.30	20.6	10.1	29.8	50.0	45.2	47.6
2.00	20.6	10.6	29.9	50.2	45.6	47.9
2.30	20.7	11.3	30.1	50.1	45.6	47.9
3.00	20.9	11.3	30.1	50.3	45.7	48.0
3.30	21.1	11.0	29.9	50.3	45.9	48.1
4.00	21.1	10.8	30.0	50.2	45.6	47.9
4.30	21.1	10.3	30.1	50.0	45.1	47.6
5.00	21.0	10.2	29.9	50.2	45.1	47.7
5.30	21.0	9.9	29.8	50.3	45.0	47.7
6.00	21.0	9.6	29.9	50.3	44.8	47.6
6.30	21.1	9.4	30.1	50.3	44.9	47.6
7.00	20.9	8.9	30.2	50.2	44.8	47.5
7.30	20.8	8.1	30.0	50.1	44.6	47.4
8.00	20.8	7.9	30.1	50.2	44.5	47.4
8.30	20.8	7.8	30.2	50.3	44.6	47.5
9.00	20.8	7.8	30.3	50.1	44.2	47.2

Appendix C

Walls, Ceiling and Floor cross-sections



Appendix D

Pipe's specifications

(D-1) Polybutylene pipe's specifications

Physical properties	ASTM method	Unit	Value
Melt flow rate	D1238	g/10 min	0.4
Color	-	-	Grey
Density	D1505	g/cm^3	0.937
Dielectric constant	D150-65T	-	2.5

Mechanical properties	ASTM method	Unit	Value
Yield point	D638	MPa	17.6
Tensile strength	D638	MPa	33.4
Breaking elongation	D638	%	280%
Elastic module	D638	MPa	265
Shore hardness	D2240	D scale	D60
Thermal conductivity	C177	W/m.K	0.22
Embrittlement temperature	D746	°C	-21

(D-2) Cross-linked polyethylene pipe's specifications

Properties	Unit	Value
Degree of XL	%	> 65%
Max. working temperature	°C	95
Breaking elongation	%	300%
Elastic module	MPa	107
Softening temperature	°C	130
Thermal conductivity	W/m.K	0.35

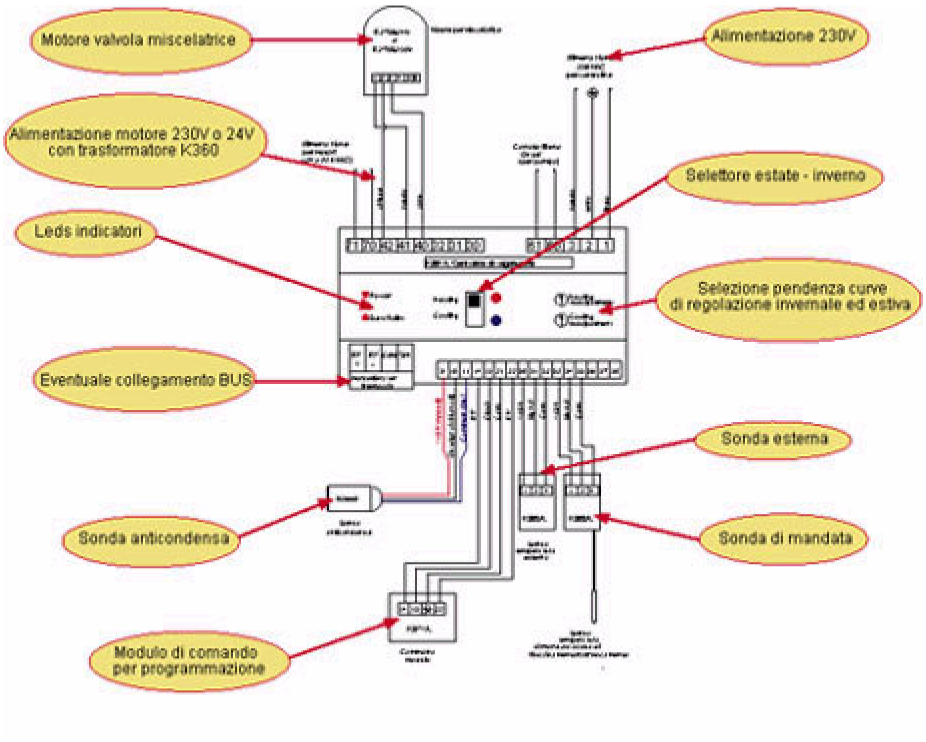
Appendix E

Control system

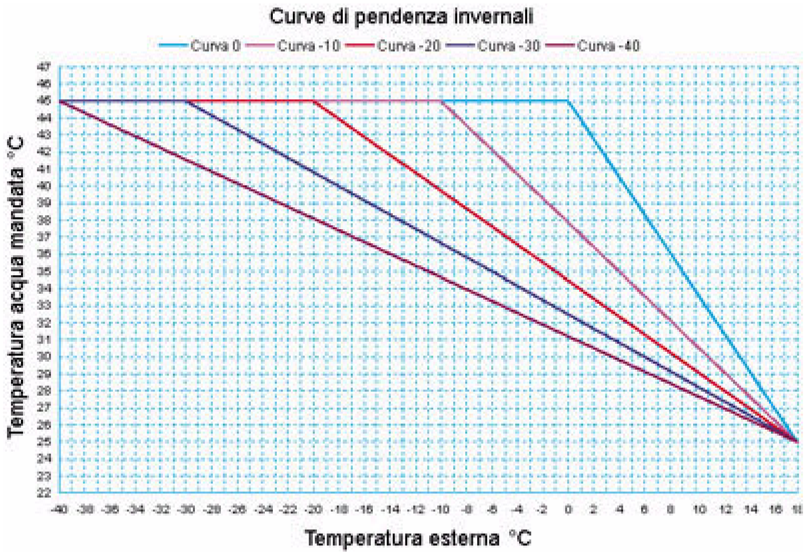
Controls and setting

The installations require two kinds of control:
a) Water supply temperature control
b) Hydraulic circuits balancing

Compensator



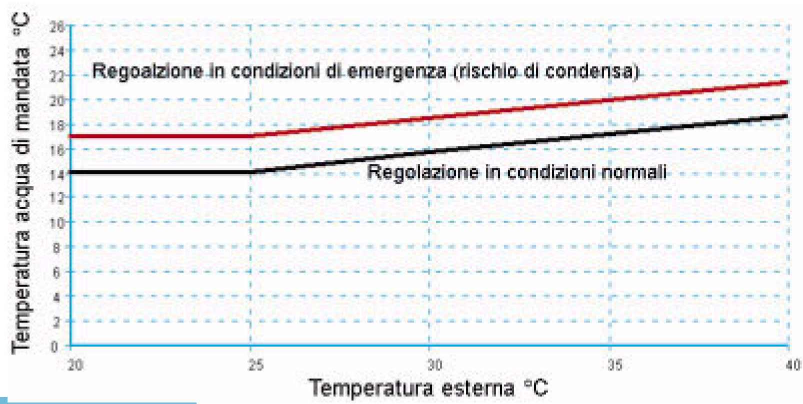
Using a compensator as shown in the above illustration makes the compensation. The electrical connections are at 230 V. It actuates the 3-way mixing valve installed on the main manifold R586, adjusting its position in function of the outside temperature revealed by means of the outside sensor (K365A) and the flow temperature revealed by the flow sensor (K363A). The correct balancing curve is selected in function of the outside temperature, as indicated in the following graph.



The required setting curve has to be selected within the above range, referring to outside design temperature (e.g. outside minimum design temperature –10°C, then 45°C flowing temperature is supplied when the outside temperature reaches really –10°C and the graph is the second starting from right side). From the above graphs it is evident that a higher flow temperature is possible for setting curves located more in the right position in the graph. In fact for the outside temperature of –7 °C in the above graph the flowing temperature is 42.5°C. For higher flowing temperature move the setting towards the line of setting 0 °C (approx. –8°C). It provides higher water temperature and consequently higher heat output.

Summer setting curves

The following graph indicates the function of the compensator for summer conditions:



Flow temperature shall never be below 14°C and the slope of the line starts from the outside temperature of 25°C. The slope (0.2 – 0.8) has to be selected in function of the inside temperature and from the admitted humidity accepted. For example in the case of locations having very high humidity a low slope (0.2) is not recommended, because of the high risk of surface condensation, being the dew-point line easy to reach on the cold surface.

Even in case of no condensation, the risk of having the system constantly working on emergency conditions are rather high.

We suggest the setting at the slope 0.3, which is a good compromise for the majority of European climate. By means of a screwdriver it is then possible to modify it according to the needs.

The compensators are equipped with a dew point sensor, the function of which is to protect the cooled surfaces from the risk of condensation. For this reason they are located in touch with the screed, in the coolest areas of the floor. Its function is the following: when on the cold surface the humidity reaches 95% (attention: the 95% is the humidity of the air film surrounding the cold structure, and not the ambient humidity). The risk of condensation is transmitted to the electronic compensator, which provokes the shifting of the controlled flow temperature upwards (normally 3 K or 5 K) depending on the technical evaluations of the local risk.

ملخص

"دراسة كفاءة نظام الأسطح المشعة في التدفئة و التبريد"

يتناول هذا البحث دراسة كفاءة نظام الأسطح المشعة من الناحية العملية في تطبيقات التدفئة و التبريد. حيث اعتمدت هذه الدراسة على فرق درجات الحرارة بين الجو الخارجي و الحيز المدفأ التي أمكن تحقيقها باستخدام هذا النظام، كما تمت مراقبة درجة حرارة الحيز المدفأ و المبرد باستخدام هذا النظام خلال فترات التشغيل لدراسة التغير الذي يطـــرأ و مسبباته.

تمت هذه الدراسة من خلال إنشاء مختبر في الجامعة الأردنية يعتمد نظام الأسطح المستعارة في تدفئته و تبريده وذلـــك لاعتماد ظروف تشغيل طبيعية و بحسب الظروف الجوية لمدينة عمان، حيث تمت قراءة كل من درجة حرارة الحيز، درجة حرارة الجو الخارجي و درجة حرارة الأسطح المستعارة و من ثم جرى تحليل هذه القراءات للتوصل لجـــدوى كفاءة نظام الأسطح المشعة.

بعد دراسة النتائج وجد أن نظام الأسطح المشعة ملائمة للظروف الجوية في الأردن لاستخدامات التدفئة و التبريـــد و من المتوقع أن تنعكس استخدامات أنظمة الأسطح المشعة بصورة إيجابية على كلفة أنظمة التدفئة و التبريد و خصوصا في المباني العامة و المتعددة الطبقات.