INVESTIGATION OF A RADIANT COOLING SYSTEM EQUIPPED WITH AN OUTDOOR AIR UNIT FOR AIR CONDITIONING IN HOT AND HUMID CLIMATE

MR. NUPAKON KOTCHARAK
ID: 55300700505

A THESIS SUBMITTED AS A PART OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF ENGINEERING IN ENERGY TECHNOLOGY AND MANAGEMENT

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Thesis Committee

Pipat C. ....

(Asst. Prof. Dr. Pipat Chaiwiwatworakul) Advisor

Surapong C ....

(Prof. Dr. Surapong Chirarattananon) Member

....

(Asst. Prof. Dr. Kuskana Kubaha) Member

S chot ....

(Dr. Surawut Chuangchote) Member

....

(Assoc. Prof. Dr. Apichit Therdyothin) External Examiner

Topic: Investigation of a Radiant Cooling System Equipped with an Outdoor Air Unit for Air Conditioning in Hot and Humid Climate

Name of student: Mr. Nupakon Kotcharak **Student ID:** 55300700505

Name of Supervisor: Asst. Prof. Dr. Pipat Chaiwiwatworakul

ABSTRACT

Radiant cooling is an alternative air-conditioning system that achieves thermal comfort at low energy consumption. However, in hot and humid climates, the potential of the radiant cooling is limited by the radiant panels at higher surface temperature to avoid the condensation of moisture within the air on the panel surfaces.

This thesis investigates air conditioning by using a radiant cooling system in Thailand's hot and humid climate. The study comprises full-scale physical experiments of a radiant cooling system installed in a room of a laboratory building. A dedicated outdoor air unit with run-around-coil heat recovery was installed and integrated to the radiant cooling system for moisture removal from the ventilation air. The experimental results show that the radiant system can provide thermal comfort level within the neutral comfort zone (PMV varies in a range of $\pm$0.5). The condensation can be avoided; even the chilled water was supplied at 18°C. The rate of heat absorption was approximately 40-50 W/m^2 at panel surface temperature was 20°C. TRNSYS simulation program was used to model the radiant room. The results from the simulation do well agree with the experimental measurement.

In the simulation study, the TRNSYS model was used to simulate the interior thermal environment of the radiant room under different periodic climate conditions in a year (i.e. cool and dry, hot and dry, hot and humid, and late rain). A separate model was also established to simulate the same room but equipped with a conventional all-air air-conditioning system. The results show that the radiant system can achieve the same comfort level as the convention system with a smaller cooling load (about 18.6 percent reduction). The outdoor air unit shared about 10 percent of the total sensible load.

The simulations were also performed to examine the influence of air infiltration into the radiant cooling room. The results show that in order to achieve the thermal comfort (more than 90% of time) with no condensation, the infiltration has to be limited to not exceeding 1.0 air change per hour. Under the intense solar radiation in a tropical

region, a radiant room requires insulated walls to minimize external heat gain. For the modeled room, the polyurethane form of 5 cm. thick is sufficient.

Keywords: Dehumidified, PMV index, Radiant cooling, Ventilation air

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LIST OF ABBREVIATIONS AND SYMBOLS

TRNSYS	TRaNsient System Simulation Program
a	weighting factor
A_r	Effective radiation area of body (m^2)
A_D	DuBois surface area of nude body (m^2)
C	Heat loss by convection from the surface of the clothed body (W)
E_d	Heat loss due to water vapor diffusion through the skin (W)
E_{eb}	Normal beam solar radiation (W/m^2)
E_{ed}	Diffuse solar radiation (W/m^2)
E_{eg}	Global solar radiation on horizontal (W/m^2)
E_{re}	Latent heat loss due to respiration (W)
E_{sw}	Heat loss due to sweating (W)
F_j	View factor between the person in a room and a given surface *j*
GRDREF	Ground reflectance
h_c	Convective heat transfer coefficient (W/m^2K)
h_r	Radiative heat transfer coefficient (W/m^2K)
h_{we}	Latent heat of vaporization of water (kJ/kg)
H	Internal heat production (W)
L	Thermal load on the body (W)
l	Dry respiration heat loss (W)
m	Water flow rate, kg/s,
M	Metabolic rate (met)
I_{cl}	Clothing insulation (clo)
PMV	Predicted mean vote
R	Heat loss by radiation from the surface of the clothed body (W)
RC	Radiant cooling system
RH_{amb}	Relative humidity of ambient air (%)
RH_{rm}	Indoor humidity, Room humidity (%)
RH_{vent}	Relative humidity of ventilation air (%)
T_a	Air temperature (°C)
T_{amb}	Ambient air temperature (°C)
T_{dew}	Dew point temperature (°C)

LIST OF ABBREVIATIONS AND SYMBOLS (Cont')

T_{set}	Indoor set point temperature (°C)
T_{sky}	Equivalent temperature of sky (°C)
T_{sw}	Supply water temperature to floor (°C)
T_{suf}	Surface temperature (°C)
T_j	Absolute temperature of surface *j* (K)
T_{mrt}	Mean radiant temperature (°C)
T_o	Operative temperature (°C)
T_{room}	Indoor temperature, Room air temperature (°C)
T_{cw}	Temperature of chilled water (°C)
T_{vent}	Temperature of ventilation air (°C)
v	Air speed (m/s)
x	Humidity ratio (kg water/kg dry air)
ε	Average emissivity of clothing or body surface
ε_j	Emissivity of surface j
σ	Stefan-Boltzmann constant, 5.67×10^{-8} (W/m^{2}K^4)
ρ	Air density (kg/m^3)

CHAPTER 1

INTRODUCTION

1.1 Rationale/Problem Statement

In hot and humid climates, much and more conventional air-conditioners using fan coil units to circulate cooled air from the units throughout the building space are being used in order to attain thermal comfort for the occupants in a building space. The conventional air-conditioning is highly energy intensive and, typically, shares up to 50-70% of the whole building energy consumption. It is thus desirable to develop alternative air-conditioning approaches that thermal comfort can be achieved at low energy consumption. Radiant cooling is an alternative, low-energy-consuming air-conditioning that is successfully implemented in high latitude regions. In tropical region as Thailand, radiant cooling is however quite challenging due to its application and cooling performance being limited by high-moisture ambient air.

Radiant cooling is an alternative, low energy-consuming method of air-conditioning implemented successfully in high latitude regions as shown by many research studies. In tropical region as Thailand, radiant cooling is however quite challenging due to its application and cooling performance being limited by high-moisture ambient air. In hot and humid climate, radiant cooling air-conditioning system has an important issue, that is, condensation on the surface of cooling panel. The study on radiant cooling for avoiding condensation problem with natural ventilation under hot and humid climate countries, like Thailand is found that the temperature of supply cooling water was limited to 24^{o}C [1]. To prevent this problem, there are many researches display how to resolve condensation problem, for instance the application of desiccant wheel [2]. Also, fan coil unit is another approach to prevent condensation on panel surface [3].

The tropical regions have high ambient temperature and humidity. To solve such problems with radiant cooling, the system integrating with the dehumidification system is suitable to apply in hot and humid climates. Generally, the cooling panels do not have dehumidification duty, and condensation does not occur on the surfaces. So, dehumidification of ventilation air supply can prevent the condensation on panels and remove latent heat from the occupied room without the recirculation air fraction. Although the supply of air necessary for ventilation purposes is still distributed throughout ducts, the electrical energy for fans and pumps can be reduced as the amount of air transported through buildings can be significantly reduced [4]. Dehumidification will improve the performance of radiant cooling systems and make thermal comfort easy to achieve. In hot humid climate which it has relative high temperatures and high humidity, these climates usually require both cooling and dehumidification by requiring of low room air temperatures, and supplying cooling water at low temperatures to radiant cooling panels. These limitations are the challenge in apply radiant cooling in tropical climate to achieve thermal comfort and avoid the condensation problem.

Dehumidification units using fan coil units can reduce the humidity of the intaken air before the air's entering a room decreases the performance of the panels by the sharing of the ventilation and total sensible load. Because this dehumidification system supply air of ventilation to avoid condensation problem at low temperature into the room and causes room air temperature is low. Therefore, the radiant cooling system is not able to operate fully performance. Anyway, the alternative dehumidification technology, heat pipe heat recovery, is conducted to replace fan coil unit as ventilation system. Heat pipe run-around-coil heat recovery is able to pre-cool intake air before entering to cooling coil, after air passed through cooling coil, intaken air is pre-heat before supplied to the room. This ventilation air can avoid condensation problem on the panels and sharing load between radiant cooling system and ventilation unit. However, there is no document reported radiant cooling integrated with heat pipe technology in hot and humid climate.

This study investigates the application of radiant cooling systems integrated with dehumidification air ventilation units in a tropical climate country. In addition, thermal comfort of the conventional air-conditioning system, radiant cooling system, and radiant cooling system integrated with dehumidified air ventilation consist of fan coil unit and heat pipe run-around-coil heat recovery were compared. Characteristics and performances, cooling load and energy consumption of each scheme of radiant cooling integrated with dehumidified air ventilation are investigated. Also, parametric study of applicable system is included in this study.

1.2 Literature Review

1.2.1 Performance of Radiant Cooling Systems Integrated with Air Ventilation Units

Behne (1999) presented the influences of displacement flow which interact with the portion of the cooling load being removed by the air supply, the air quality in the occupied zone and the vertical air temperature rise [5]. Novoselac and Srebric (2002) reviewed the studies and design of cooled ceiling and displacement ventilation systems in US buildings [6].

Conroy and Mumma (2001) reported that the dedicated outdoor air system must be designed to control 100% of the space latent loads, and hence, the space dew-point temperature. However, natural ventilation air can be supplied to the room without dehumidification in some situation such as in the winter season [7].

Vangtook and Chirarattananon (2004) studied radiant cooling using natural air for ventilation in a Thai climate. To avoid condensation of moisture on the cooling panel, the temperature of water supplied to the panel was limited to 24^{o}C. The results showed that thermal comfort could be obtained with application of radiant cooling [8]. Furthermore, Vangtook and Chirarattananon (2007) investigated the application of radiant cooling as a passive cooling option in hot and humid climate. They report results of TRNSYS simulation model representing actual experimental room. Chilled water at 10 °C was supplied to ventilation unit to precool ventilation air in daytime, for the nighttime, only radiant cooling system can achieve thermal comfort [9].

1.2.2 Performance Evaluation

Investigations of the efficiency of both active walls and cooling panels and their capacity for heat absorption in the room using chilled water circulation were carried out under laboratory conditions, and the results were compared using computer simulations. In European countries, the preference is for installation of radiant cooling systems in buildings. The testing standard and certification for cooling panels have also been set up. Jiang *et al.* (1992) studied the relationship of the radiant heat transfer between the cooling panel and other radiative heat sources, and then the effects of a radiative heat source and a cooling panel on indoor air flow, temperature stratification, and dispersion of contaminants in a furnished office with displacement ventilation. It was found that the ceiling mounted cooling panel decreased the vertical temperature and that displacement ventilation may have enhanced air movement and contaminate [10].

Kochendorfer (1996) presented the overview of standardized testing methods for the evaluation of the cooling output of room cooling panels, and the result has had influence on the efficiency of cooling panels [11].

Olesen (1997) offered to introduce radiant floor cooling with the upper comfort level for the operative temperature in summer at 26^{o}C in spaces with mostly seated residents. The typical round of a heat exchange coefficient between the floor surface (19^{o}C) and the room (26^{o}C) is 7 W/m^{2o}C, where 5.5 W/m^{2o}C is radiant heat transfer. Meanwhile, a maximum cooling capacity for a floor system is estimated at 50 W/m^{2}, based on the heat exchange between the floor surface and the room. Very often, direct sunshine on the floor in some spaces, such as the atria, halls or other spaces with window facades, have cooling capacities that may approach 100 to 150 W/m^{2}. Moreover, cooling capacity also depends on floor construction, distance between each tube, water flow rate, and floor covering [12].

Olesen *et al.* (2000) showed that according to a test conducted to measure and calculate the heat exchange coefficient between a cooled floor surface and a space, air velocity in the test space should not exceed 0.2 m/s. The results showed that the heat exchange coefficient 7.5 W/m^{2}.K is multiplied by the difference between the floor surface and the operative temperature as a total heat exchange, combined radiant and convective. The separate convection and radiation heat exchange was calculated using the heat exchange coefficient 1.0 and 5.5 W/m^{2}.K with the difference between floor surface and air temperature respectively. The heat exchange coefficient is an important value for calculation of the cooling capacity of a cooling floor system [13].

Ardehali *et al*. (2004) presented a proof for a concept model and a procedure for modeling the heat transfer mechanism of radiant panels for an occupant modeled as a sphere in a thermal zone. It was concluded that the proposed analytical approach is in effective agreement with data available from the literature [14].

Jeong and Mumma (2004) presented a development in their simplified method for accurately estimating the impact of mixed convection on the cooling capacity of ceiling a radiant panel in mechanically ventilated spaces. It was found that the total capacity of ceiling radiant cooling panels can be enhanced in mix convection situations by 5 to 35% under normal operating temperatures [15].

1.2.3 Thermal Comfort Potential of Radiant Cooling Systems

Kulpmann (1993) reported on an investigation of thermal comfort in a test room equipped with a smooth and cooled ceiling surface and supplied with upward displacement ventilation air. The results showed that a high-level thermal comfort was attained and that the temperature of the room surfaces (not only the cool ceiling) was lower or at least equal to the air temperature in the room, which was different from the situation in an air-conditioned room [16].

Simmonds (1996) reported that the traditional design criteria, such as dry-bulb temperature and operative temperature, were not always sufficient. Mean radiant temperature had a large influence on the comfort results. Radiant cooling was a superior means to bring conditions in a space to comfort limits (PMV 0.5) [17].

Meierhans (1996) reported the use of a water-carrying pipe system installed in the core of the concrete ceilings to actively control the thermal mass of an office building in Horgen, Switzerland successfully over three summers. Comfort measurements under actual and simulated conditions confirmed the suitability of the system for small and medium loads [18].

Imanari *et al*. (1999) reported that the radiant ceiling panel system was capable of creating smaller vertical variations in air temperature and a more comfortable environment than can conventional systems [19].

Kitagawa *et al*. (1999) reported on a study on thermal sensation for subjects under temperature asymmetry and different levels of relative air humidity in a climate chamber. They reported that small air movements at 0.1-0.3 m/s under a radiant cooling system could improve thermal comfort for occupants [20].

Nagano and Mochida (2004) reportedly used a rectangular box to represent a reclining person in the calculation of mean radiant temperature sensed by the subjects and found that the resulting temperature was about 1^{o}C lower than the mean radiant temperature obtained from the globe thermometer method. The reclining position is common for human subjects in hospitals and in bedrooms [21].

Carli and Olesen (2001) reported on a field assessment of thermal comfort under a radiant cooling system where pipes were embedded in the building structure. Measurements taken for one office in Austria and two offices in Germany led to the conclusion that acceptable indoor thermal environments were attained during summer [22].

Taweekun and Tantiwichien (2013) developed a radiant cooling system for the thermal comfort of Thai people. They evaluated thermal comfort by PMV value and result was observed that the PMV values are in the comfort ranges during 18:00 to 10:00 for air velocity 0.2 m/s and 0.4 m/s. The results revealed that thermal comfort zones of Thai people are in the ranges of relative humidity 50 – 70% and effective temperature (ET) 24 – 27 °C [23].

1.2.4 Energy Saving Potential of a Radiant Cooling System

Brunk (1993) reported that using a cooling ceiling with ceiling-mounted air outlets and mechanical as well as free cooling and an additional ice storage plant. The total energy costs could be reduced by 50% as compared with a variable-volume system. A study based on TRNSYS simulation compared energy costs of operation of a radiant cooling system with a number of configurations of conventional air-conditioning systems concluded that a radiant cooling system saves much fan energy [24].

Niu *et al.* (1995) discuss the energy consumed by chillers, which was also lower for a radiant cooling system because chilled water is produced at a higher temperature. Using the same simulation program [25], Sodec (1999) presented the results that demonstrated a cooling ceiling system incurs lower energy costs and space requirements than VAV systems under a specific cooling load of 45 - 55 Wm^{-2} [26].

Antonopoulos *et al*. (1998) experimentally and theoretically analysed space cooling using 1.8 x 2.16 m^2 of metal ceiling panels in a 2.0 x 2.5 x 3 m^3 test chamber. The results showed that satisfactory thermal comfort conditions in the climate of Greece and energy savings may be possible given higher temperatures of cooling water, which improve the efficiency of solar-driven absorption chillers [27].

1.2.5 Operation of Radiant Cooling Systems Integrated with Air Ventilation System

Scheatzle, D. G. (1996) describes the use of electronic sensors was made for a new generation of control in a combination radiant/convective system with a radiant floor and ceiling system in a house in Carefree, Arizona, which has caught attention. There are two options for controling the comfortstat and the operative sensor. A comfortstat offers control of additional devices that affect radiant temperature, air motion, and humidity. The operative sensor control by measuring the two primary variables, which in this case is ambient air temperature and mean radiant temperature. The second option was selected by this case as initially sensor and the basis of control. It is a reliable and economical sensor that controls a system to provide more stable comfort at a lower operating cost [28].

Olesen *et al.* (2002) simulated, with a dynamic simulation program TRNSYS 14.2 (1998), each of these control methods for radiant heating and cooling systems, where pipes are embedded in the concrete slabs between each story. The best achievement of the comfort and energy performance was obtained with a combination of time control and water temperature control according to ambient air temperature conditions [29]. In addition, Ryu *et al.* (2004) [30] and Lim *et al.* (2006) [31], similar experiment, also presented studies on these control methods for a radiant floor heating and cooling system in a residential building in Korean. Their results show that water temperature control (outdoor reset with indoor temperature feedback control) is better than the water flow control (on/off control and variable flow control) constancy of room temperature and floor surface temperature. The application of an outdoor reset method to on-off bang-bang control would help reduce fluctuation in average room air temperature. Also, applying pulse-width modulation to on-off bang-bang control improved the stability of room temperature and floor surface temperature.

Song *et al.* (2008) describes a radiant floor cooling system integrated with dehumidified ventilation, which dehumidifies the outdoor air entering to the zone through the cooling coil of a ventilator to prevent condensation on the floor. And, they developed outdoor reset control to adjust supply water temperature of chill water supplying to the floor, and use of indoor temperature feedback control to respond to internal load change. The result from an experiment and TRNSYS simulation showed the acceptable range of comfort conditions [3].

Cejudo Lopez *et al.* (2013) analyzed the interest of an air handling unit in a real building in Madrid and Malaga in Spain under different climates, dry or humid. There are many possible control strategies developing to reach required supply condition. However, the best mode is evaluated by primary energy consumption. Concept of control strategy is the sensible load will be removed firstly, but if the floor cannot remove entire load, the AHU deals with the rest. Moreover, TRNSYS simulation program is used calculate the load [32].

Nutprasert and Chaiwiwatworakul (2014) reported on the results from experiments and simulations of radiant cooling systems could be applicable to achieve thermal comfort in tropical climates by systems integrated with fan coil unit ventilation air systems. In the simulation study, TRNSYS program was used to investigate for annual simulation. But, there is no suitable of control system in the experiment to keep constant of thermal comfort [33].

1.3 Objectives of Study

The main objective of this thesis is to investigate the application of radiant cooling system equipped with an outdoor air unit for air-conditioning in buildings in hot and humid climates. The specific objectives of the study are:

1) To conduct experiments to evaluate performance and thermal comfort that can be achieved by a radiant cooling system integrated with a dehumidified ventilation air unit,

2) To model the radiant cooling system and then evaluate its performance in terms of achievable thermal comfort and cooling load against the typical air-conditioning system.

1.4 Scope of the Study

1) The geographical scope of this study covers a tropical climate, but the basic meteorological information and other information were obtained from the roof deck of the seven-storey Bioresources and Technology building at Bang Khun Tien station in Thailand.

2) The weather data for yearly simulations were taken from the solar radiation and daylight measurement station at the Asian Institute of Technology (AIT).

3) The existing radiant cooling panels and fan coil unit, and installed heat pipes run-around-coil heat recovery at a laboratory building at King Mongkut's University of Technology Thonburi, Bang Khun Tien Campus. The experiment was conducted in a real tropical climate.

4) A fan coil unit, and heat pipe run-around-coil heat recovery were employed as outdoor air units in the experiments to investigate the dehumidification of supply ventilation air.

5) ASHRAE Standard 55 was used to evaluate the thermal comfort of the results of the experiments. Also, ISO 7730 was used in a simulation study according to PMV (Predicted Mean Vote).

6) The dynamic calculations were carried out with a TRNSYS (TRaNsient Systems Simulation) standard program.

CHAPTER 2

THEORIES

2.1 Introduction

In this chapter, all information and applicable contents, including the details for thermal comfort conditions, under the operation of a radiant cooling system is reviewed. The thermal comfort conditions are among the first priorities for the occupants of a room. A radiant cooling system is an alternative technology that can provide thermal comfort with energy savings at the same level as a conventional air-conditioning system. In this study, the radiant cooling systems are evaluated for tropical humid climates, of which there are three types based on the quantity and regime of annual rainfall. The first receives relatively abundant rainfall the whole year long. The second-tropical monsoon climate, experiences a short dry season but a very rainy wet season. The third type-a tropical savanna climate, is classified by a longer dry season and a prominent but not extraordinary wet season. Thailand's climate is defined as a tropical monsoon climate and a tropical savanna climate [11].

2.2 Thermal Comfort Considerations

Thermal comfort is an evaluation by each human of feeling either satisfied or unsatisfied of any environment. Experiments under various conditions in each environment with a comfort sensation indicator called the predicted mean vote (PMV) were first carried by Fanger (1972) [34]. PMV has seven levels indicating a thermal sensation scale as follows:

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

When PMV is zero, thermal comfort is maintained and the condition can be accepted when PMV falls within a ±0.5 range [35]. The most important factors that influence thermal comfort are the indoor air velocity, mean radiant temperature, occupant activity levels, and the clothing worn by people, in addition to air temperature and relative humidity. This thermal comfort condition is based on the methodology presented in the ASHRAE 55 (1992) [36]. All six factors contribute to indoor climate conditions. One of the interesting parameters is the mean radiant temperature which takes the environment temperature and effect of the position of each human in the room into account. For example, a person who works close to a high temperature window will be uncomfortable and hot on the part of the body next to a heat absorbing window. Whereas, another part of the body that faces other walls will experience a lower temperature. Facing both the window and wall simultaneously will make a human uncomfortable. Thus, the room air temperature alone does not indicate thermal comfort; mean radiant temperature is another significant factor of concern.

2.2.1 Measurement and Calculation of Thermal Comfort Parameters

Radiant Temperature of a Surface (T_j), to measure surface temperature, one can resort to the use of a contact thermometer - a resistance or thermocouple one - and an infrared sensor [36].

Air Temperature (T_a): there are many kinds of sensors deployed for the measurement of air temperature. Among them, the resistance thermometer, the thermocouple, and the bimetallic thermometer are common examples. To reduce the relative influence of thermal radiation during use of these sensors however, any observations to be derived ought to undergo a process of shielding the sensor from other sources of thermal radiation, e.g. sunlight, windows and walls [36].

Mean Radiant Temperature (T_{mrt}), in light of the PMV evaluation, the variable mean radiant temperature is required, apart from the other three physical variables. The globe thermometer and two-sphere radiometers are some examples of the instruments used to measure it [36]. According to Fanger, this parameter can be defined as the uniform surface temperature of an imaginary black surroundings which produces radiant heat, loss of an amount identical to that from a person who is in a given body posture and clothing and at a given point in a room [34]. As such, its calculation can approximately be obtained from measured surface temperatures in a room and the corresponding view factors to a person:

$$T_{mrt}^4 = \Sigma_j F_j T_j^4 \qquad (2.1).$$

Air Motion: A few examples of sensors used in measuring this parameter are the hot-wire anemometer, heated sphere anemometer, and vane anemometer. They should be omnidirectional, or else, must be directed very carefully to show the reading of true air speed at a particular test position [36].

Operative Temperature (T_o): The operative temperature is the uniform temperature of an imaginary black enclosure where a person would exchange an amount of heat by radiation and convection with an equal one in an actual non-uniform environment. Accordingly, it can be assessed numerically as the average of air temperature and mean radiant temperature, weighted by their respective heat transfer coefficients (h_c and h_r, respectively). It should be noted that a globe thermometer, which is 5-10 centimeters in diameter, can also be used to measure it directly [36]. The following is the model used to calculate the temperature under discussion [37]:

$$T_o = \frac{(h_c T_a + h_r T_{mrt})}{(h_c + h_{mrt})}, \qquad (2.2a)$$

Where T_o = operative temperature, ºC,

h_c = coefficient of convection heat transfer, W/(K. m^2),

T_a = air temperature (dry bulb), ºC,

h_r = coefficient of radiation heat transfer, W/(K. m^2), and

T_{mrt} = mean radiant temperature, ºC.

As for the convective heat transfer coefficient "h_c", it can be assessed at atmospheric pressure and calculated for a seated person surrounded by moving air using the following equation [36].

$$h_c = 8.3(v^{0.6}) \qquad \text{(For } 0.2 < v < 4.0\text{)} \qquad (2.2b).$$

The linearized radiative heat transfer coefficient "h_r" can be computed by means of the equation below:

$$h_r = 4\varepsilon\sigma \frac{A_r}{A_d}\left[273.2 + \frac{T_{cl} + T_{mrt}}{2}\right]^3 \quad (2.3).$$

As asserted above, the operative temperature can be estimated from the air temperature and the mean radiant temperature. Therefore, it can also be calculated through the following formula:

$$T_o = aT_a + (1 - a)\, T_{mrt} \quad (2.4).$$

where the weighting factor (a) depends upon the air speed (v) in approximately the following

v(ms-1)	*0-0.2*	*0.2-0.6*	*0.6-1.0*
a	*0.5*	*0.6*	*0.7*

Humidity: some instances of instruments in aid of the measurement of air humidity include the electrical conductivity or capacity hydrometer. Calibrating the humidity level in the air can be measured in a variety of ways - for example, via dew point, relative humidity, wet bulb, or vapor pressure and converted into other kinds of measurements utilizing comparison tables or certain psychrometric charts [36].

Physical/Personal Variables: there are two categories to be discussed under this designation. The first is metabolic rate (M), and the other is clothing insulation (I_{cl}).

The metabolic rate denotes the energy-production rate of a body. Table 2.1 shows some typical metabolic rates. Under this standard, metabolism, which is subject to type of activity, is incorporated. If a person is seated and immobile, the metabolic rate of a unit surface area will amount to 58.2 W/m^2, based on an overall surface area of a person at 1.8 m^2 [36].

Table 2.1 Metabolic rates of different activities

Activity	Met	W/ m^2	W (average)
Sleeping	0.7	40.4	70
Reclining	0.8	46.6	80
Seated and quiet	1.0	58.2	100
Sedentary activity (office, lab, school)	1.2	69.8	120
Standing, relaxed	1.2	69.8	120
Light activity, standing (shopping, laboratory experiment, light industry)	1.6	93.1	160
Medium activity, standing (shop assistant, domestic work, machine work)	2.0	114.4	200
High activity (heavy machine work, garage work, if sustained)*	3.0	174.6	300

Note: Typically, rest breaks (schedule or hidden) or other operational factors (get parts, move product, etc.) combine to limit individual work to the time-weighted average level of about 2 met.

The latter relating to clothing insulation refers to the variable governed by increased resistance to sensible heat transfer occurring as a result of putting an individual piece of clothing over an undressed body. The resistance is equal to the effective increase in total insulation characterized by that particular garment and is usually rated in clo units [36]. Table 2.2 gives the clo-values of various pieces of garments.

Table 2.2 Insulation value of clothing elements

Item	I_{clo} (clo)
Garment	
T-shirt	0.08
Men's briefs	0.04
Ankle length sock	0.02
Shoes	0.02
Long - sleeve dress shirt	0.19
Thin trouser	0.15
Thick trouser	0.24
Single breasted jacket (thin)	0.36
Single breasted jacket (thick)	0.42
Ensemble	
Brief, long-sleeve shirt, thin trouser, socks, shoes	0.60
Brief, T-shirt, long sleeve shirt, single-breasted jacket, trouser, socks	1.20

2.2.2 ISO 7730: Moderate Thermal Environments - Determination of the PMV and PPD Indices and Specification of the Conditions for Thermal Comfort

The comfort standard for the International Standards Organization (ISO) is entitled: "Moderate thermal environments - determination of the PMV and PPD indices and specification of the conditions for thermal comfort". This standard is used by most of the world outside the U.S. and incorporates the PMV-PPD model.

PMV is an empirical function derived from the physics of heat transfer and the thermal responses of people in climate chamber tests. PMV establishes a thermal strain based on environmental conditions and attaches a comfort vote to that amount of strain. If the environmental conditions combined with the activity and clothing of the person you are modeling produce a PMV within the range of -0.5 to +0.5, then the condition meets the ISO comfort sensation recommendation [38].

The PMV equation for thermal comfort is a steady-state model. It is an empirical equation for predicting the mean vote on an ordinal category rating scale of the thermal comfort of a population of people. The equation uses a steady-state heat balance for the human body and postulates a link between the deviation from the minimum load on heat balance effector mechanisms, e.g. sweating, vaso-constriction, vaso-dilation, and thermal comfort vote. The PMV equation only applies to humans exposed for a long period to constant conditions at a constant metabolic rate. It is written as:

$$\mathrm{PMV} = f(M, L), \qquad (2.5)$$

$$L = H - E - E_d - E_{sw} - l - R - C, \qquad (2.6)$$

where L is the thermal load on the body,
H is the internal heat production,
E_d is heat loss due to water vapor diffusion through the skin,
E_{sw} is heat loss due to sweating,
L is dry respiration heat loss,
R is heat loss by radiation from the surface of a clothed body,
C is heat loss by convection from the surface of a clothed body.

The equation is expanded by substituting each component with a function derivable from basic physics. All of the functions have measurable values, with the exception of clothing surface temperature and the convective heat transfer coefficient, which are functions of each other. To solve the equation, an initial value of clothing temperature is estimated, the convective heat transfer coefficient computed, a new clothing temperature calculated etc., by iteration until both are known to a satisfactory degree [38].

2.3 Radiant Cooling System

2.3.1 Basic Principles

Radiant cooling systems have been employed successfully in northern Europe for more than twenty years [39]. The system is comprised of panels installed on the ceiling of a room, or in some cases, hung from a high ceiling. Cooling water is supplied to the panels at a temperature above the dew-point temperature of the air in the room to avoid condensation of moisture in the air or on the panels. In radiant cooling, heat is transferred between the space and the cooling panels through a temperature differential. The cool ceiling panels absorb heat through a combination of radiation and convection. Radiative heat transfer occurs through a net emission of electromagnetic waves from the warm occupants and their surroundings to the cool ceiling. On the other hand, the convection of the heat of the room air to the cool panels creates convection currents within the space.

The complexity associated with the overall system necessitates an understanding and incorporation of three heat transfer modes and air infiltration. Figure 2.1 schematically shows these modes of heat transfer in a radiantly cooled room. A radiant cooling panel is centered on the ceiling. The emitted radiant energy is either absorbed by the other room surfaces, transmitted from the room through non-opaque surfaces such as a window, or reflected between room surfaces. This continues until the radiant energy is completely absorbed. Likewise, the reflected energy incident on other room surfaces is partially reflected, transmitted, and absorbed. The temperature gradient between the room surfaces and the room air provides natural convection heat transfer, while air circulation is provided by displacement ventilation (Figure 2.2). The ventilation air is introduced into a room at low levels, and flows by natural means to replace the existing air (typically ≤ 0.5 m/s). In a typical radiantly cooled office building, two to three air exchanges per hour are required. The ventilation air drawn from outdoor should be dehumidified in order to reduce latent load since the cooling panels remove sensible load only. A portion of the energy inside the room is lost by way of conduction through the room boundaries, such as walls, floors, ceiling, and windows. The characteristics of the conduction depend on the temperature of the adjacent rooms or the environment.

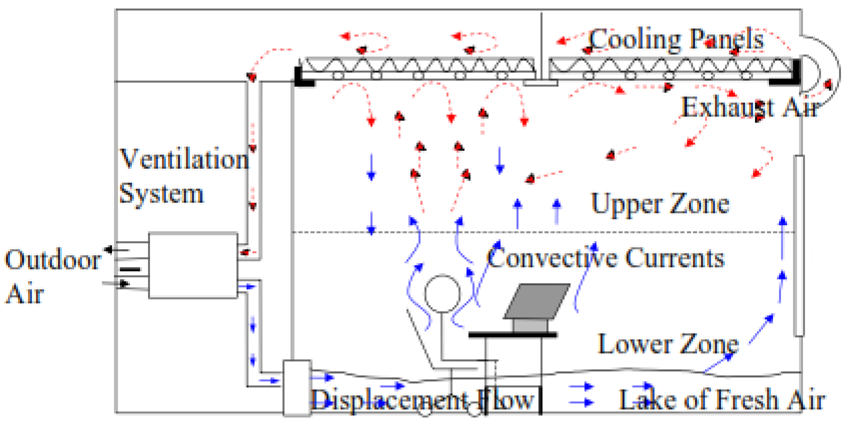


Figure 2.1 Radiant mode of heat transfer in radiantly cooled room.

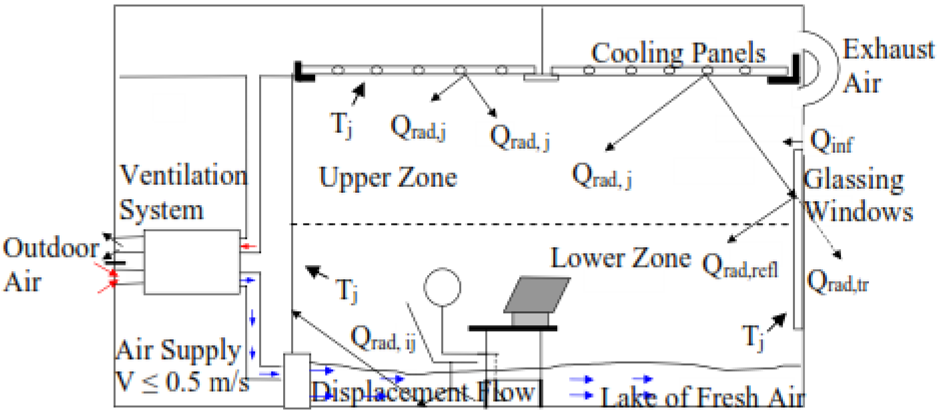


Figure 2.2 Convection mode of heat transfer in radiantly cooled room.

The performance of a radiant cooling system depends on its ability to receive heat convected to the panels by air and to receive heat radiated to the panels by surfaces in the room. The quantity of radiant energy transfer involves mainly the continuous exchanges of radiation between all bodies in a built environment. The rate at which radiant energy is transferred depends on the following factors: temperature (of the emitting surfaces and the receivers), emittance (of the radiating surfaces), reflectance, absorptance, and transmittance (of the receivers), and view factors between the emitting surfaces and the receivers (viewing angle of the occupant to the radiant sources). Any hindrance in the panel to heat transfer to or from its surface (and eventually the cooling water) will reduce the performance of the system. Also, the lower the panel's inlet temperature causes the higher potential of heat-transfer of the cooling system. Therefore, the inlet temperature should be controlled to be as close as possible to the room's dew point temperature. Consequently, the cooling capacity of a radiant cooling system is generally limited by the minimum allowable temperature of the inlet water relative to the dew point temperature of the room air and the size (area) of the panel [40, 41].

2.3.2 Principal Advantages and Disadvantages

The familiar air conditioning in Thailand is the conventional air-conditioning system in which cooling is almost exclusively based on convection. In such a system, cooled and dehumidified air is circulated uniformly in the space that this system relies on mixing with the room air. Heat is convected from all surfaces in the space to the cooled dehumidified air and becomes the cooling load. This load must be occurred by the air-conditioning system. The system will work to extract heat from all sources in the space. Typically a supply air velocity from diffusers is 2-6 m/s. The volume of a supply air is giving between 5 and 15 air changes per hour [42]. This system is continuously developed to resolve many disadvantages when compared to a radiant cooling system.

ASHRAE (1996) [43] reports many advantages of the radiant cooling system. Comfort levels can be better than those of other conditioning systems because radiant loads are treated directly and air motion in the space is at normal ventilation levels. Supplied air quantities do not exceed those required for ventilation and dehumidification. This provides a draft free environment. Noise associated with fan coils or induction units is eliminated. Draperies and curtains can be installed on the inside walls without interfering with the heating and cooling system. Operational costs are reduced for the mechanical chilling system since cooled ceilings operate at relatively high temperatures (average surface temperature of 15^{o}C or above). Chillers can operate at higher temperatures resulting in an increase in efficiency and reduction in energy costs.

The other principal advantages of radiant cooling systems are listed as follows, [42].

- A 100% outdoor air system may be installed with less severe penalties in terms of refrigeration load because of reduced air quantities.
- A common central air system can serve both the interior and perimeter zones.
- Wet surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- Radiant cooling and minimum supply air quantities provide a draft-free environment. This is also confirmed by the study of Imanari *et al.* (1999) [19] and Stetiu (1999) [44] in which it was shown that the radiant ceiling panel system creates a more comfortable environment than conventional systems due to smaller vertical variations of air temperature. This creates less draft.
- Peak loads are reduced as a result of thermal energy storage in the panel structure, and exposed walls and partitions.
- Panels can be coupled with other conditioning systems for heat loss (gain) compensation for cold or heat floors, windows, *etc.*

However, a radiant cooling system has some disadvantages as shown here:

- Slow response is a characteristic of high mass systems that can be ameliorated by careful selection and installation of heating elements and controls.
- Non-uniform surface temperatures due to improper element sizing, piping installation spacing or insufficient heat capacity.

- Options may be restricted for building materials selection, floor coverings, furnishings location, and heater proximity.
- When radiant cooling surface temperature is lower than indoor dew point temperature, condensation occurs on cooling surface, which leads to cooling ceiling damage and indoor microbial growth, because of the limited cooling surface temperature, the cooling ability of radiant terminal device is limited.
- In some wet areas, outdoor air infiltration increases the possibility of condensation, thus doors and windows should be kept closed as much as possible, which limits the applications of natural ventilation.
- When a ventilation system is not employed simultaneously, the indoor air velocity will be too low and lead to sweltering sense [45].

2.3.3 Design of Radiant Cooling Systems

Designing a radiant cooling system to achieve high efficiency of heat transfer for the objective of providing thermal comfort must take several factors into consideration. The cooling structure that is used for the removal of the heat load from the room by cooling water circulates in a tube. The structure installed comes in one of two types - either an active wall (Figure 2.3) or a cooling panel (Figure 2.4). The heavyweight construction active wall is likely to store heat and slow the response for the removal of the heat load so that its optimal performance will be in the buildings that operate continuously for 24 hours.

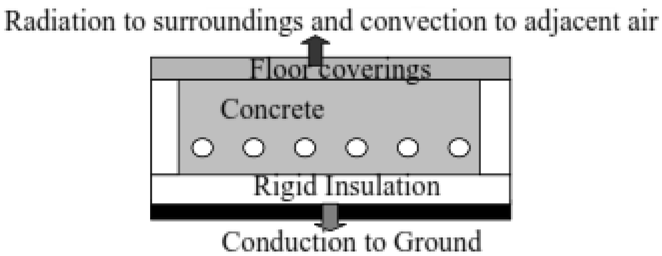


Figure 2.3 Schematic description of a floor slab radiant system as an active wall [46].

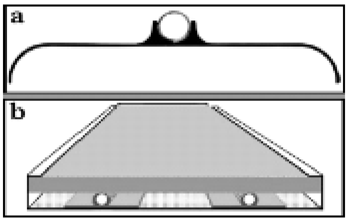


Figure 2.4 Typical ceiling panel construction: (a) free hanging panel; (b) “drop-in” panel with back insulation and acoustical perforation [39].

Lightweight cooling panels are another option that offer fast response rates for variable load conditions, and can be run for any length of time. Notably, the lead time for cooling panels is only from three to five minutes. For example, the aluminum sheet thickness of cooling panels is only about 0.76 mm, and the thermally bonded copper cooling water piping is generally 12 mm in diameter or less with 150 mm centers. Panel piping arrangements are generally in a serpentine pattern [39]. The metal cooling panels are used and frequently installed on T-bar grids designed to support the dropped acoustical ceiling. The panels in the dropped ceiling are top loaded with insulation to prevent heat gain from the plenum space [15].

One factor to consider for different cooling loads is the area of active wall or cooling panel necessary. Whenever the rate of heat transfer is limited, the controlling of surface temperature could be higher than the dew point temperature. In such cases, the active wall or cooling panel area should be large enough to handle the cooling load. Design of the appropriate area is complicated; however, the integration of such systems into the building simulation program is an option for calculation of the active wall and cooling panel area required to achieve conditions of thermal comfort. Positioning an active wall or cooling panel also must be approached with the aim of preventing any obstacles from coming between human occupants and the wall or panel.

Another consideration is temperature control of the cooled water inlet to prevent water condensation on cool surfaces. It was recommended that the lowest temperature for cooled water should be at least 1^{o}C above the room air’s dew point. If the dew point of room air is decreased with a dehumidification air system, then the cooled water temperature inlet can be reduced in accord with the decrease in dew point temperature. In this way, the cooling capacity of cooling panels can be increased.

2.4 Dehumidified Air Ventilation System

2.4.1 Heat Pipe Heat Recovery [47]

The alternative technology, heat pipe, is a passive device with the potential for energy recovery. Outside of heat pipe is similar to plate-finned water or steam coil but differ at the tube is not interconnected and separated as evaporator and condenser sections by the partition plate that is shown in Figure 2.5.

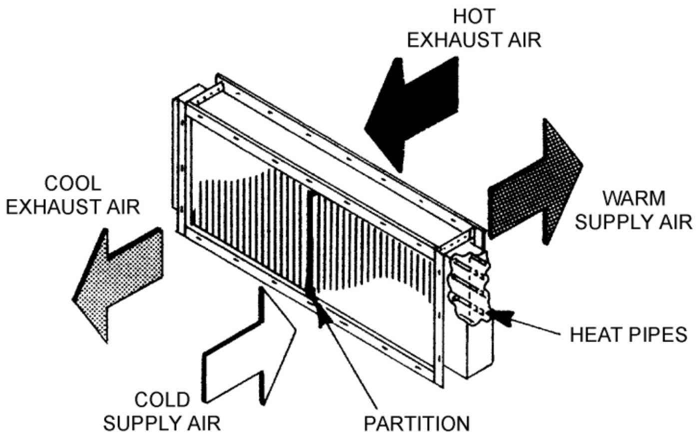


Figure 2.5 Heat pipe heat exchanger.

A heat pipe is a sensible heat transfer device; furthermore, latent heat can be transferred with condensation on the fins, resulting in better performance. Heat pipe is manufactured with an integral capillary wick structure and permanent sealed, inside the tube is evacuated and filled with suitable working fluid shown in Figure 2.6. Working fluid normally is refrigerant or fluorocarbons, water, and other compounds affects to temperature of outlet air temperature.

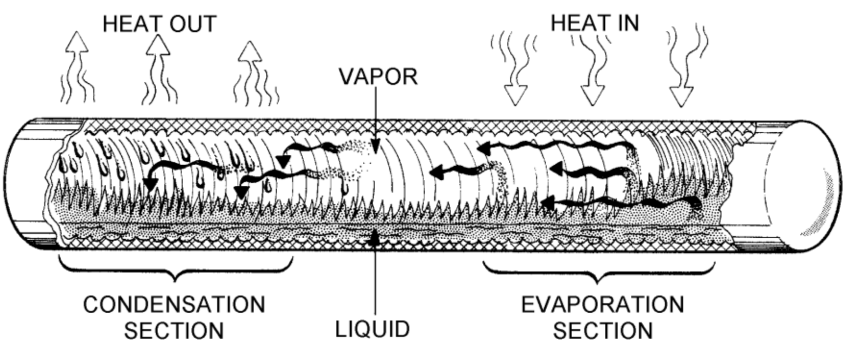


Figure 2.6 Heat pipe tube.

2.4.1.1 Principle Operation of Heat Pipe Heat Recovery

Hot air entering into the exchanger at the inlet side which is the, evaporation side, causing working fluid in heat pipe tube vaporize, and then the differential pressure drive vapor of working fluid into the condensation side of the exchanger that makes the vapor of the working fluid condense and release latent heat, and then flow back to the evaporator. So, the working fluid operates in closed loop as long as the process is droved by a differential temperature.

In actually, around the surface tube and at the working fluid, there is a small temperature drop. However, energy transferring in heat pipe heat exchanger is considered as isothermal. Heat transfer capacity of heat pipe depends on such factors as wick design, tube diameter, working fluid, and tube orientation.

2.4.1.2 Performance of Heat Pipe Heat Recovery

Design and orientation have an effect on heat pipe heat transfer capacity. Figure 2.7 presents heat pipe heat exchanger effectiveness which show the relationship between face velocities and rows of tubes found that the number of rows of tube is direct proportional to effectiveness and the face velocities are inverse proportional to effectiveness. The effectiveness of a counter flow heat pipe heat exchanger depends on the total number of rows. For example, two units in series has an effectiveness equal to a single unit when has a same total number of rows. However, Series units are often used because it easy for handling, cleaning, and maintenance.

The diameter of the pipe has an effect on the heat pipe. Heat transfer capacity increases roughly with the square of the inside diameter of the pipe. For example, when compared a 25 mm inside diameter with the heat transfer capacity of a 25 mm inside diameter more than a 16 mm inside diameter 2.5 times approximately because the diameter of the pipe relate to amount of flowing airstream that make heat can transfer better.

The heat transfer limit of the heat pipe heat exchanger is independent of the length of the tube, excluding a very short heat pipe. For example, a 1 m long heat pipe has heat transfer capacity as same as a 2 m long heat pipe when the diameter is identical because the 2 m heat pipe has twice the external heat transfer surface area of the 1 m pipe, it will reach its capacity limit sooner thus, the length of the tube cannot increase heat transfer capacity. Nevertheless, heat transfer capacity can be improved by reconfiguration the tube and addition the number of rows of the tubes but make heat pipe shorter resulting in getting the same surface area but improve the system performance.

Design of fins and spacing inside the heat pipe heat exchanger base on dirtiness of the two airstreams that relate to cleaning and maintenance requirement. Therefore, in designing not only acceptable pressure drop should be realized but also facilitate to cleaning and maintenance.

2.5 TRNSYS Simulation Program

The University of Wisconsin developed a commercial software package named TRNSYS. This program is a complete and extensible simulation environment for the transient simulation of systems, including multi-zone buildings. It is used by engineers and researchers around the world to validate new energy concepts, from simple domestic hot water systems to the design and simulation of buildings and their equipment, including control strategies, occupant behavior, and alternative energy systems (wind, solar, photovoltaic, hydrogen systems). The main applications of this program consist of solar systems (solar thermal and photovoltaic systems), low energy buildings and HVAC systems with advanced design features (natural ventilation, slab heating/cooling, double façade, etc), renewable energy systems, cogeneration, fuel cells.

A TRNSYS project is typically setup by connecting components graphically in the simulation studio. Each type of component is described by a mathematical model in the TRNSYS simulation engine, and has a set of matching proforma’s in the simulation studio. The proforma has a black-box description of a component consisting of inputs, outputs, parameters, etc.

A standard component of TRNSYS, a Multi-zone building (Type 56), is used to calculate the load, thermal comfort (PMV values based on the concept of EN ISO 7730) [35], temperature and humidity.

CHAPTER 3

METHODOLOGY

The main objective of this study is to investigate the application of a radiant cooling system equipped with an outdoor air unit for air-conditioning in a building in a hot and humid climate.

In the conduction of the experiments, the radiant cooling system was installed with a separated ventilation system in 2 cases, including (i) fan coil unit, and (ii) heat pipe at an experimental room at Bang Khun Tien Campus. The experiments took place under real weather conditions for 24 hours operation and were conducted in 2014. A well-known simulation program, TRNSYS, can be used for the dynamic calculations. The program was validated to the results derived from the experiments in the two cases. Also, UC Berkeley Thermal Comfort Program was employed to compare thermal comfort from results of experiment and simulation.

3.1 Experimentation and Facilities

To conduct experiments with a radiant cooling system integrated with a dehumidification unit, a single-story building and an additional chilled-water radiant cooling system was installed in a room used for the experiments. The collected data consist of surface temperature, room air temperature and relative humidity, inlet and outlet chilled water temperature at cooling panels, fan coil unit and heat pipe, chilled water flow rate of cooling panel and dehumidification units were the operating parameters in the experiments. A data acquisition system connected to a personal computer was used to collect the measured data every minute automatically. The data were detected and obtained follow to changes in behavior of the system. There are two types of the data collectors include continuous record collector and instantaneous record collector.

A) Continuous record collector

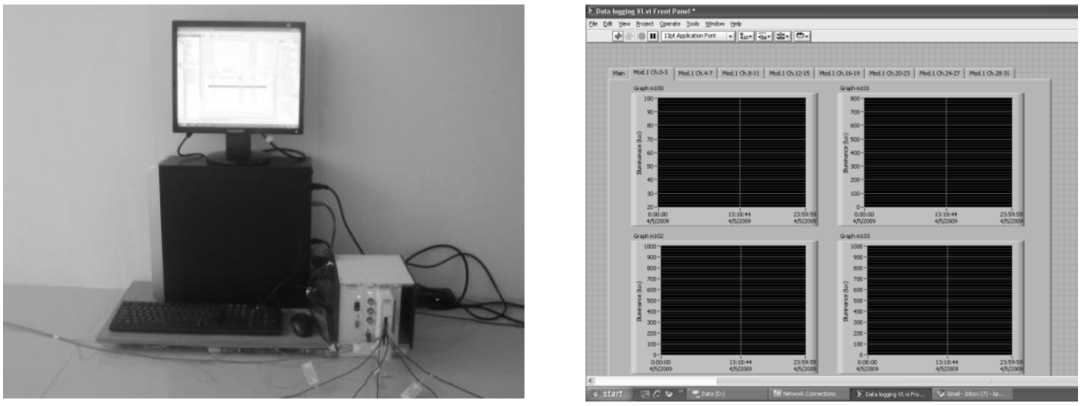
A thermocouple type T was used to measure interior and exterior surface temperatures of the building. PT-100 was adopted to measure cooling panel surface temperature of radiant cooling, inlet and outlet chilled water was also measured by this equipment. Globe temperature was the equipment used to measure mean radiant temperature. In addition, heat flux parameters on the panels were measured by heat flux sensors. The measurement of surface temperature of radiant cooling panels, water temperature of heat pump and heat gain by the experimental room were recorded continuously every minute by the data acquisition system.

B) Instantaneous record collector

Humidity and air movement in the experimental room were measured by TESTO humidity sensor and anemometer sensor. The water flow rate of the chilled water supply to the panels and dehumidification unit was measured by rotameter and electronic flow meter respectively.

In the building, a National Instrument (NI) data logging system is provided to serve intensive measurements of temperature and relative humidity of room air and heat gained on interior wall surfaces, interior daylight distribution, chilled water flows, etc. The software program of the computer controller was developed by the use of the data acquisition toolbox of National Instrument (NI) Lab View software. Figure 3.1(a) is a photograph of the NI computer data controller with a SCXI 1000 chassis and a signal conditioner SCXI-1102 mounted with a front-end SCXI-1303 and a signal conditioner SCXI-1102 mounted with a front-end SCXI-1503 for RTD. Figure 3.1(b) exhibits a font panel of the real time data monitoring. All the measured data from the sensors have been acquired by the system and recorded onto the computer hard disc at every one minute. A new record file can be generated automatically at the beginning of each day.

The weather data were recorded at the roof deck of a seven-storey building of the School of Bioresources and Technology. The sensors included those measuring global, diffuse horizontal, and beam normal components of daylight and solar radiation. The station is also equipped with sensors to measure vertical illuminance and solar radiation, zenith luminance, air temperature and humidity sensors, and sky luminance distribution. All data from the sensors have been logged and recorded on a personal computer. Figure 3.2 shows a photograph of the station.



(a) Data acquisition (DAQ) system (b) Font panel the DAQ system

Figure 3.1 Data acquisition system for the experiment.

Figure 3.2 A photograph of the daylight and solar radiation station.

Due to the minimum infiltration capacity of the experimental building, it was considered a closed system and a ventilation system was set up for the occupant of the experimental room in accordance with the ASHRAE standards. In the initial stage, experimentation included only the physical results of the radiant cooling system integrated with dehumidified ventilation – for example, room air temperature, humidity values and wall surface temperature – with particular attention to thermal comfort achievement. The standard TRNSYS simulation program was used for reach validation. Regarding points of comparison the experimentation brought two cases under study:

1) Radiant cooling system integrated with fan coil unit (Case 1).

2) Radiant cooling system integrated with heat pipe run-around-coil heat recovery (Case 2).

The weather data including ambient air temperature, relative humidity, solar radiation, sky temperature, wind velocity, etc used for this study is derived from the weather station in Bang Khun Tien Campus. The data from station were used to be input in the simulation program in order to receive results to validate the data from the two cases. The simulation program received the input and accordingly simulated each of the two experimental cases, yielding the results for validation.

3.1.1 Radiant Cooling System Equipped with Fan Coil Unit

In this case, experiments of the radiant cooling system integrated with fan coil unit as the air ventilation unit for the dehumidification ventilation air before entering the room were conducted for 24 hours operation. The experiment was conducted in October. During the early rain period as described in Chapter 4.

3.1.2 Radiant Cooling System Equipped with Heat Pipe Run-Around-Coil Heat Recovery

The second case is the radiant cooling system integrated with a heat pipe run-around-coil as the air ventilation unit for reducing the humidity and reheated ventilation air in order to decrease the sharing of the sensible load between the panels and the ventilation system before being introduced into the room. This case was conducted for 24 hours, as described in Chapter 4.

The obtained data from such cases were compared to the results of the TRNSYS simulation program. The points validated the accuracy comprised (a) room air temperature, (b) surface temperature of cooling panels, (c) Heat absorption by radiant panels, (d) thermal comfort level, i.e., PMV index, and (e) Mean radiant temperature. To compare the results in thermal comfort level, the UC Berkeley comfort program was used to calculate PMV values from the results of experiment and simulation from relating parameters.

3.2 Experimental Setup and Measurement Setup

The physical facilities, devices, and measuring sensors used for the experiments is described in this section. In this study, a laboratory building is located at Bang Khun Tien Campus, King Mongkut’s University of Technology, Thonburi (KMUTT). A radiant cooling (RC) system was existed in the building. The details of the building, the RC system is provided below. A number of measuring sensors are described next.

3.2.1. Experimental Building and Cooling System

The experimental building was a single-story, cross-shaped building with gable roofs constructed in an outdoor area reserved for energy research at Bang Khun Tien Campus. The building itself was located at latitude 13.57°N and longitude 100.44°E Figure 3.3 exhibits a pictorial view of the building facing east. A room on the right side was towards north and was the room equipped with radiant cooling (RC) panels for radiant cooling study.

A detailed plan of the building is shown in Figure 3.4. The building had a common room at its center and had four identical side rooms with north, east, south and west orientations. All side rooms had the same internal dimensions of 2.4 m. width by 2.8 m. length and a height from the floor to the ceiling of 2.4 m. Each of the side rooms also had one exterior windowed-wall facing the main cardinal orientations (N, S, E, and W).

Figure 3.3 The experimental building at Bang Khun Tien Campus, KMUTT.

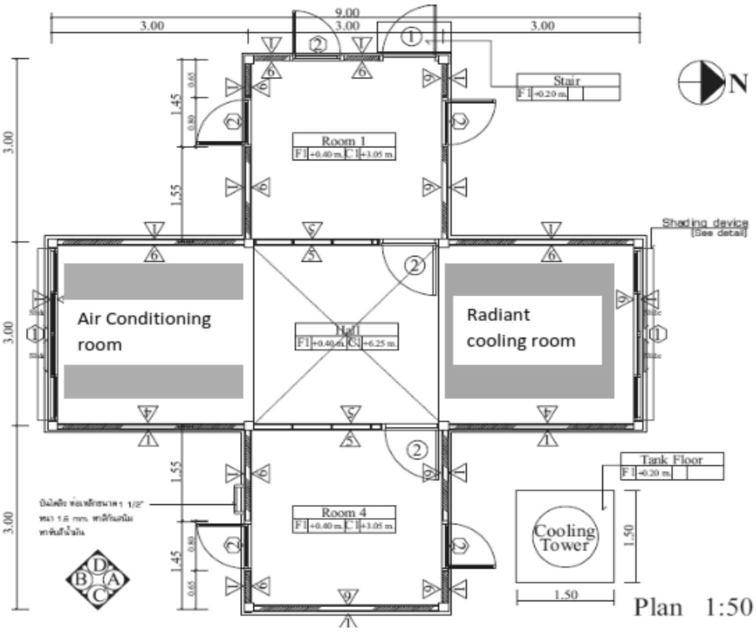


Figure 3.4 A floor plan of the experimental building.

The exterior walls of the building were brick wall with exterior mortar plastering. In order to reduce external heat gain and resulting cooling load, three-inch polyurethane (PU) foam was included to be a layer after brick layer of the walls. The innermost layer of the walls was 10 mm. smart boards. All interior walls of the building were also insulated with the same type of insulation of one-inch thickness. For the building roof, three-inch fiberglass matt was laid over the ceiling to prevent the heat gain. Figure 3.5 shows the detailed configuration of the building walls. The windows were 6 mm. green tinted glass with external shading slats.

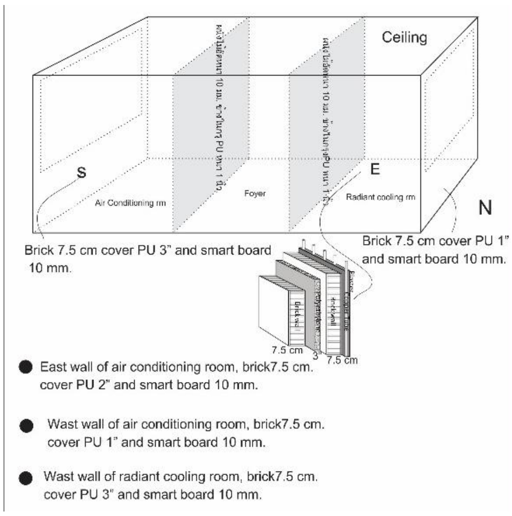


Figure 3.5 Configuration of the walls of the experimental building.

Figure 3.6 shows the north room as experimental rooms. Two radiant cooling (RC) panels were installed: the one on the west wall and the other one under the ceiling. In addition, an outdoor air unit was connected with the north side of the room.

Figure 3.6 The radiant cooling room.

The configuration of the RC panels is shown in Figure 3.7. The RC panels were actually the industrial products of flat-plate solar collectors. The panels were aluminium plate with black-colored font surface. Back side of the plate was affixed with a series of parallel straight copper tubes whose ends were connected with two headers. One header functioned to distribute evenly chilled water over the panels and the other one was to collect the chilled water from the panels. The RC panels were light-weight and thus were fast thermal responsive. Thermal insulation was placed to cover the tubes and beside of the panel in order to allow a privileged font surface heat transfer, to reduce heat loss, and to improve the acoustics of the room. Each RC panel had a gross heat transfer surface of 6.8 m^2.

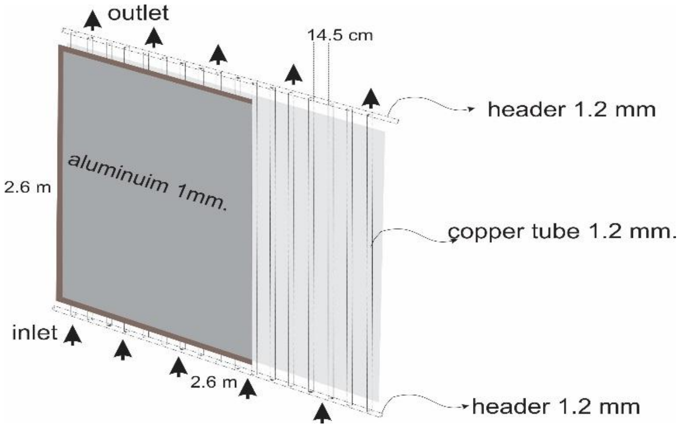


Figure 3.7 Configuration of the RC panels.

A heat pump with a cooling capacity of 3.1 refrigeration tons was installed next to the experimental building. The heat pump produced chilled water at 8°C, supplying it to the building for cooling. Figure 3.8 shows a photograph of the heat pump.

Figure 3.8 Heat pump of the building cooling system.

Figure 3.9 exhibits a diagram of the building's cooling system with the heat pump. In the chilled water loop, a 100-liter tank was installed to store chilled water in order to reduce temperature fluctuation of the chilled water supply due to starts and stops of the compressor of the heat pump. In the figure, Pump P1 functioned to circulate chilled water with a constant rate between the heat pump and the tank. Chilled water from the tank was supplied directly to a dehumidified air ventilation unit (fan coil unit or heat pipe run-around-coil heat recovery) by a Pump P2.

For the radiant cooling (RC) system, chilled water from the storage tank could not be used directly. The chilled water was about 8°C from the tank was first supplied to the primary side of a heat exchanger in order to produce chilled water at higher temperature on the secondary side of the heat exchanger. A modulating solenoid valve was equipped on the primary side to regulate the chilled water flow from the tank based on a signal of a thermostat measuring temperature of the chilled water in the secondary side. In this study, chilled water in the secondary side supplied to the RC panels was at 19°C for Case 1 and 18°C for Case 2 in order to investigate the efficiency of dehumidification unit. As shown in the figure, the secondary chilled water was circulated at a constant flow by Pump P3.

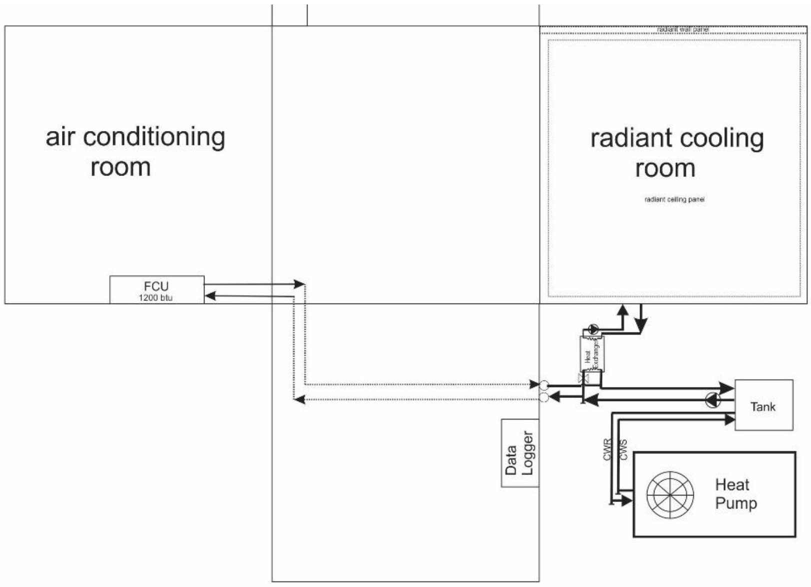


Figure 3.9 A single line diagram of the cooling system of the experimental building.

Figure 3.10 shows the dehumidification system (fan coil unit) that was used to pre-cool and dehumidify the outdoor air before it entered the room. For this experimental study, the ventilation air flow rate was used as 30 kg/h. The ventilation system is also equipped with another cooling coil of capacity of 9,000 Btu to pre-cool the ventilation air as defined conditions.

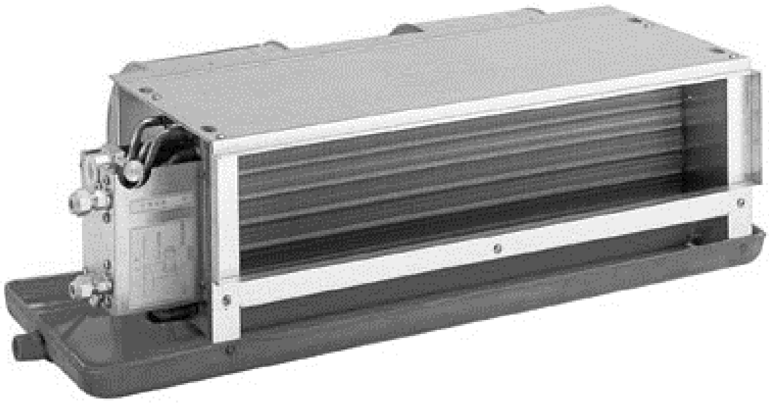


Figure 3.10 A Dehumidification unit (Fan coil unit).

Heat pipe run-around-coil heat recovery is also a dehumidification system, as shown in Figure 3.11. This unit was used to pre-cool the outdoor air, and then passed through cooling coil for dehumidification. After that the dehumidified air was pre-heat before entering into the room. For this experimental study, ventilation air flow rate was used as 30 kg/h. This equipment was used to reduce ventilation load and sensible load share between radiant cooling and dehumidified ventilation system.

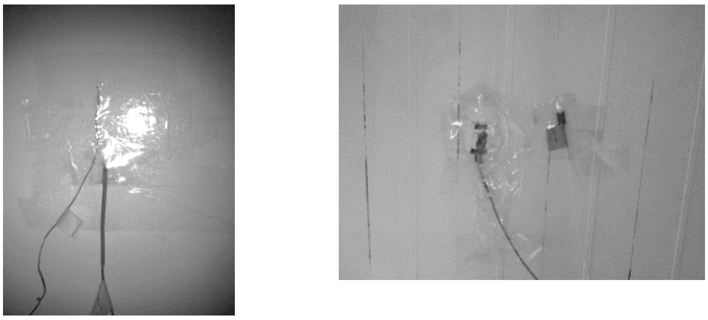
Figure 3.11 A Dehumidification unit (Heat pipe run-around-coil heat recovery).

3.2.2 Measurement Setup

Measurements were taken during the experiments to determine the rates of heat extraction at the RC panels and thermal comfort in the RC. The measurements included surface temperature of the inside walls of the room and the RC panels, flow rate and temperature of chilled water supply and return, and temperature and relative humidity of the room air. The outdoor environment of solar radiation and ambient air were obtained from the recorded data at the meteorological station located in the same area.

(a) Temperature of Surfaces and Heat flow

The temperature of all surfaces (walls, floor, ceiling, and radiant panels) were measured by PT-100 sensors and thermocouple Type T. Each sensor was placed at the center of the surfaces and the measured values were considered as representative temperature of the whole surface. In addition, the rates of heat flow through RC panels were measured by heat flux sensors attached on the panel surfaces. The sensors were placed close to the temperature sensors, as shown in Figure 3.12.



(a) Wall (b) Radiant panel

Figure 3.12 PT-100 sensors to measure the surface temperature.

(b) Air Temperature, Relative Humidity and Globe Temperature of the Room

Air temperature, relative humidity and globe temperature of the room (in RC room) were measured by electronic sensors, including TESTO and the Wet Bulb Globe Temperature sensor (WBGT). The sensors were placed at the center of the room.

(a) TESTO (b) WBGT

Figure 3.13 Sensor to measure temperature and relative humidity of room air.

(d) Water Flow

A water meter was installed to measure the amount of chilled water consumed during the experiments by the two RC panels. Two rotameters were installed later to measure the rates of chilled water flow supplied to each RC panel. In addition, the electronic flow meter was used to measure the rates of chilled water supplied to both dehumidification units (fan coil unit and heat pipe run-around-coil heat recovery), as exhibited in Figure 3.14.

(a) Rotameter (b) Water meter (c) Electronic flow meter

Figure 3.14 Equipment to measure chilled water flow rate**.**

(e) Air Flow

A hot-wire anemometer was used to measure the air velocity of ventilation air supplied into the room.

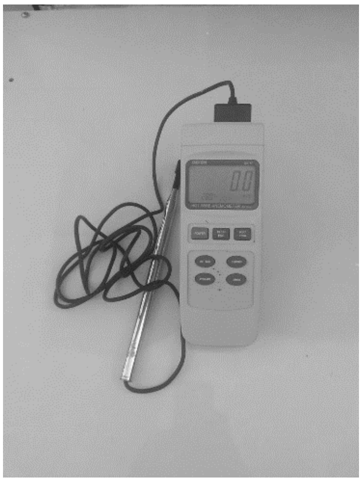


Figure 3.15 Hot-wire anemometer to measure rate of air flow.

3.3 Simulation of the Radiant Cooling System (Whole Year Simulation)

The simulation of the operation of the radiant cooling panels used the TRNSYS simulation program for investigation according to the specific objectives mentioned in chapter 1. There are 4 cases to study and compare comprise (i) conventional air-conditioning system, (ii) radiant cooling system integrated with natural air ventilation, (iii) radiant cooling system integrated with fan coil unit, and (iv) radiant cooling system integrated with heat pipe run-around-coil heat recovery. In each case was operated for 24 hours.

The detailed results of the TRNSYS simulations for all of these cases are shown in Chapter 4. The input data used for whole year simulation ware meteorological conditions data in 2000 from the AIT weather station. The station was installed on the flat roof, at a height of 10 m from floor, at the Energy building. This station measures and records global, beam, and diffuse solar radiation, infrared radiation, air temperature, relative humidity, and wind speed. The data were recorded from the sensors at intervals of one minute and averaged to 15 minutes of data for the radiant cooling system. The weather data were input data for the TRYSYS program reported in the simulation by 15-minute time-steps. Each system of studying was simulated separately in a model room representing the experimental room. The internal load was a sedentary person and an artificial light. Details of each scheme for radiant cooling system are as follows.

Case 1: The application of a conventional air-conditioning system. The simulation was operated in whole day all the year from data of year 2000. The temperature of supply air was set at 12 °C. Moreover, hysteresis controller was used to control the system in the simulation.

Case 2: The application of a radiant cooling system integrated with natural air ventilation. In this case, supply-chilled water temperature to the panels was set at 18°C to achieve high-percentage of thermal comfort. Actually, condensation problem occurred on the panels. This scheme is the based case showing why the dehumidified ventilation is necessary.

Case 3: The application of a radiant cooling system integrated with a fan coil unit dehumidified air ventilation. This application operated by setting supply-chilled water temperature at 18°C to achieve high-percentage of thermal comfort, flow rate and temperature of the ventilation air were set at 30 kg/hr and 12°C respectively to avoid condensation problem.

Case 4: The application of a radiant cooling system integrated with heat pipe run-around-coil heat recovery dehumidified air ventilation. Supply-chilled water temperature was set at 18°C similar to the previous scheme, flow rate of ventilation was set at 30 kg/hr and temperature of the ventilation air were varied depend on ambient temperature. This dehumidified air ventilation unit is an auxiliary device of radiant cooling to avoid condensation problem and achieve high-percentage of thermal comfort.

Case 5: To achieve a higher percentage of thermal comfort, Case 3 was improved by the adjusting the chilled water temperature to be appropriate to the conditions of each season.

Case 6: Case 4 was modified by the application of a radiant cooling system integrated with a heat pipe run-around-coil heat recovery dehumidified air ventilation and adjustment of chilled water temperature for reasons similar to those of Case 5.

3.4 TRNSYS Program and Computation of Comfort Indices

In the simulation study of the operation of the radiant cooling system and the dehumidification units under the conditions, the TRNSYS simulation program was used in the study of both the experiments and other conditions. TRNSYS was developed to the new version of TRNSYS 17 for many energy applications including operation of radiant cooling system. It also performs computation of comfort indices.

3.4.1. TRNSYS Program

TRNSYS is a transient system simulation program that connects utilities to produce the process. This program was employed to simulate the radiant cooling system and air ventilation system in the experimental building. Figure 3.16 to 3.19 illustrates an information flow diagram of conventional air-conditioning system, radiant cooling system with natural air ventilation, and two systems of radiant cooling system integrated with dehumidified ventilation system, respectively.

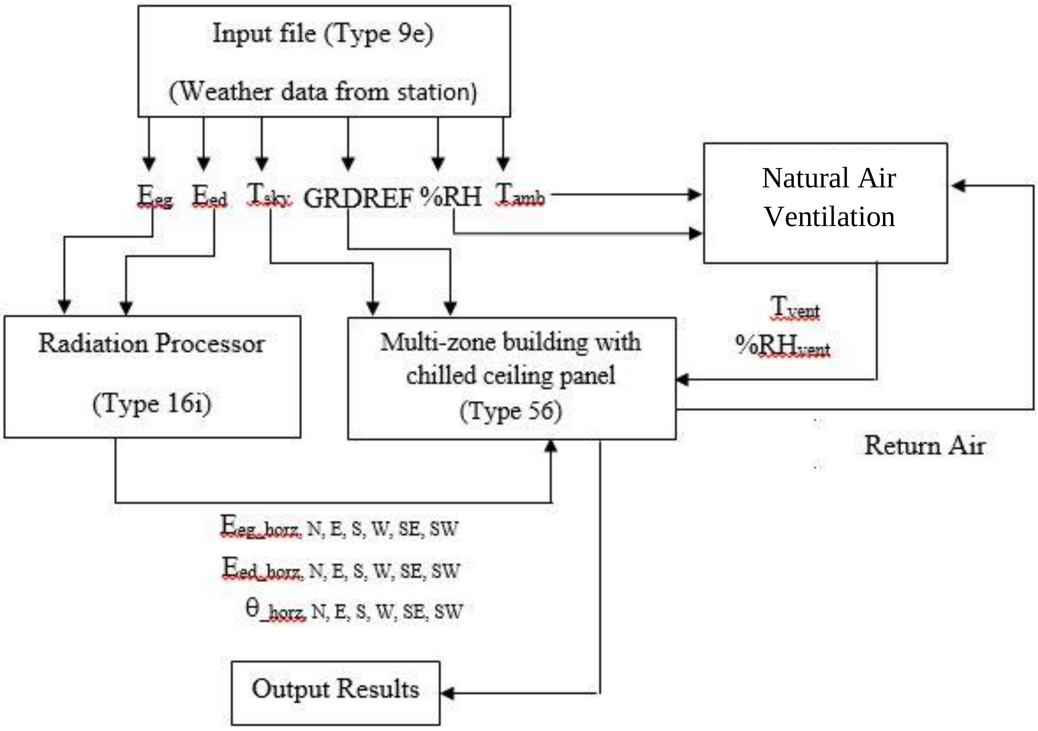


Figure 3.16 An information flow diagram of conventional air-conditioning system.

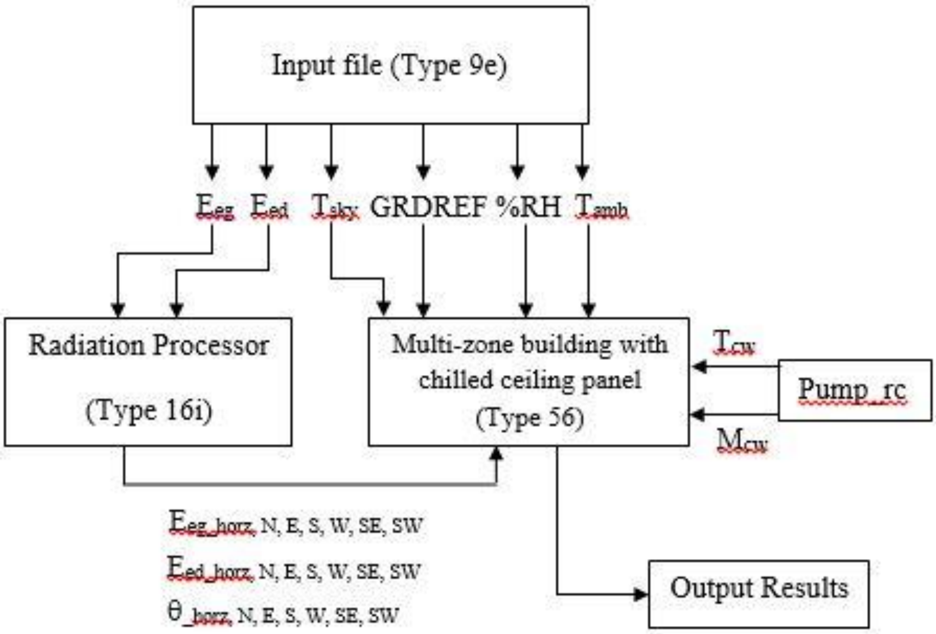


Figure 3.17 An information flow diagram of radiant cooling system integrated with natural air ventilation.

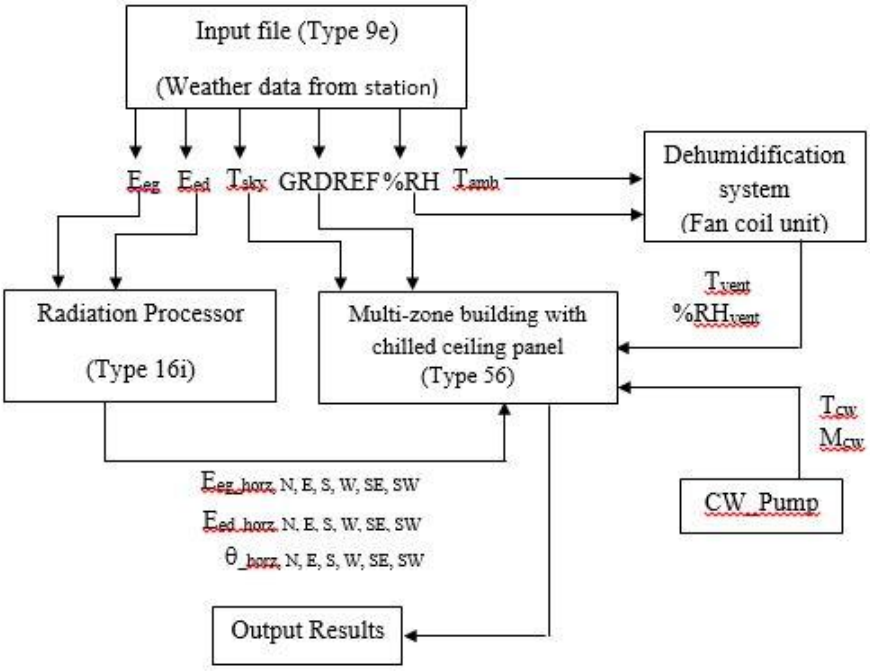


Figure 3.18 An information flow diagram of radiant cooling system integrated with fan coil unit dehumidified air ventilation.

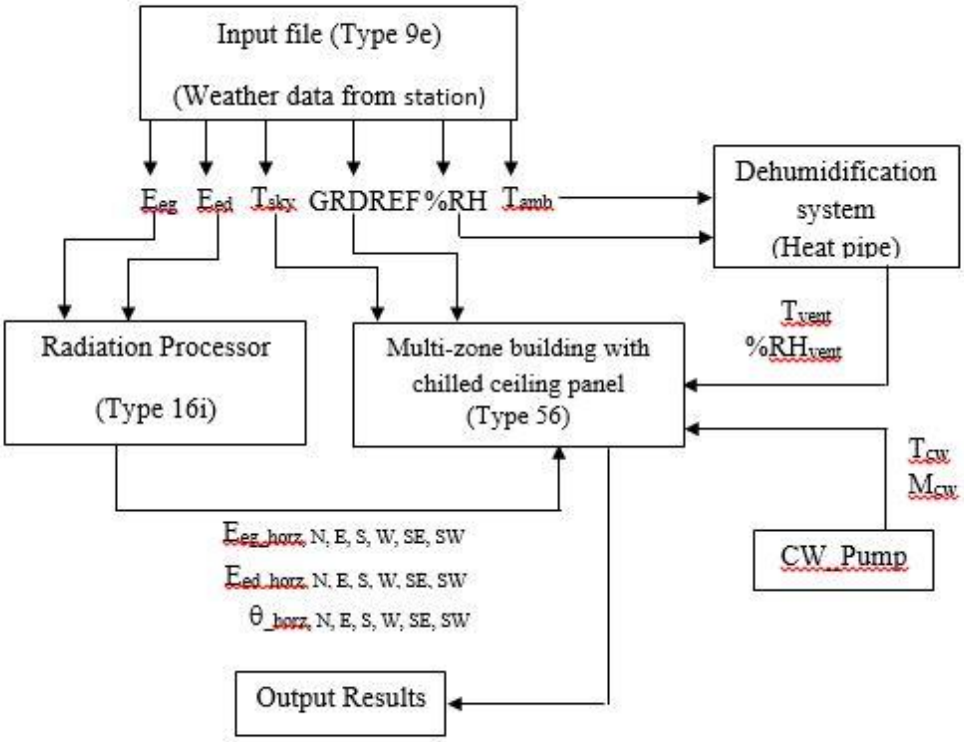


Figure 3.19 An information flow diagram of radiant cooling system integrated with heat pipe run-around-coil heat recovery dehumidified air ventilation.

3.4.2. Computation of Comfort Indices

TRNSYS adopts the International Standards Organization's procedure for computation of predicted mean vote or PMV for moderate thermal environments, as given in Standards Document EN ISO 7730-1995 [34]. This standard in turn adopts Fanger's recommended equations for calculation of PMV based on four given physical variables of dry–bulb temperature, relative humidity, mean radiant temperature, and air speed. Two personal variables of clothing insulation value and metabolic rate are also required for PMV evaluation.

The ranges of PMV for thermal comfort conditions are:

Comfortable	-0.5 to 0.5
Warm	0.5 to 1.0
Cool	-1.0 to -0.5
Unacceptably warm	Over 1.0
Unacceptably cool	Under -1.0

The multi-zone building module calculates the values of three physical variables in a simulation run. The air speed, the fourth variable, was entered as 0.05 m/s for TRNSYS simulation as well as for manual calculation of PMV of a condition using measured values. With user-input values of the two personal variables, the module produces a value of PMV for the environment in the zone at each time step. TRNSYS permits the calculation of view factor between the flat surfaces in a room, for the purpose of calculating the radiant heat exchange [47]. These algorithms cannot be utilized for evaluating the radiant exchange between a human being and the surfaces of the room [47].

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Experimental Study

This chapter discusses the measured results of the two experiments on the radiant cooling system equipped with an outdoor unit. For the first experiment, the outdoor air unit was a conventional fan coil unit while that in the second experiment was a fan coil unit integrated with a run-around-coil heat pipe heat recovery.

For both experiments, chilled water was supplied to each of the radiant panels at a rate of 180.0 kg/hr at a temperature of 19°C. The outdoor air unit supplied the dehumidified ventilation air into the radiant room at a rate of 30 m^3/hr, equivalent to 1 air change per hour (ACH). The internal heat gain in the room was approximately 276 W. The heat was generated by using six fluorescent lamps contained inside a wooden box. The surface temperature of the box was measured at 33°C.

In the experiments, TRNSYS software was used to model the radiant system and the experimental room. The calculated results from the TRNSYS model were also compared to the measurements from the experiments.

4.1.1 Radiant Cooling System Equipped with a Fan Coil Unit

In this experiment, the radiant cooling system was equipped with a fan coil unit functioning as a dedicated outdoor air unit. To handle the latent load of the ventilation air and those within the radiant room, the outdoor air was cooled down by the fan coil unit to a temperature of 12°C before being supplied into the radiant room.

This experiment was undertaken on 15 October, 2014. Figure 4.1 shows the solar radiation of global, diffuse horizontal, and beam normal components measured at the meteorological station. The temperature and absolute humidity of the outdoor air are also shown. On this day, the sky was quite clear and the global radiation reached 900 W/m^2 during noon. At that time, the diffuse radiation was measured at 350 W/m^2. The outdoor air temperature varied between 26.0°C and 33.0°C. The absolute humidity of the outdoor air was about 11-17 g/kg_{da}. The weather today was also rather hot and humid.

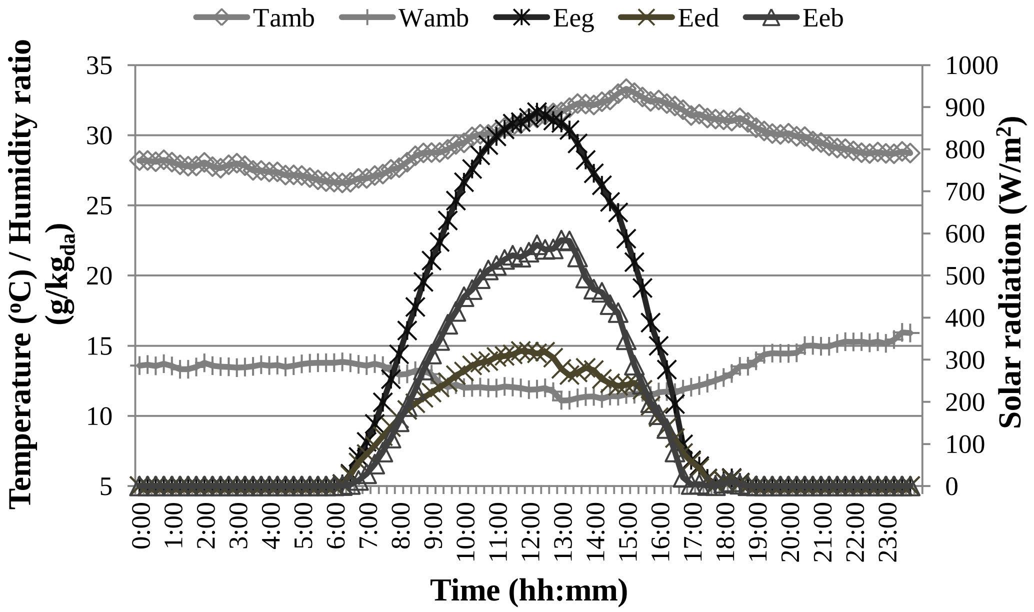


Figure 4.1 Solar radiation and outdoor air conditions on 15 October 2014.

Figure 4.2 shows the variations of the room air temperature, and the surface temperatures of the radiant panels on the ceiling and on the walls. It can be observed that the room air varied within a narrow range of 24.0-25.5°C. The temperature was lowest at 6:00 and highest at 15:00. The temperatures of both radiant panels were almost constant at 22.0°C. In the figure, the temperature values simulated by TRNSYS software were also plotted. The calculated values of the room air temperature were quite comparable with the measurements. However, the temperature values of the radiant panels from the calculation were slightly higher that the measurements of about 1°C.

Figure 4.3(a) shows the measured results of the heat flux on the surface of the wall radiant panel. It can be observed that the heat flux had varied between 110W and 170 W. The calculated heat flux using TRNSYS was also shown in the figure. The variation of the heat absorption by the panel was in the similar trend with the room air temperature. As the calculated values of the surface temperature of the panel were higher than the actual measured values, the predicted heat flux was also higher.

Figure 4.3(b) is similar to Fig. 4.3(a), but prepared for the ceiling radiant panel. It can be observed that the heat absorbed by the ceiling panel was higher than that of the wall panel. The heat absorption varied between 160W and 250 W. For the ceiling panel, the calculated heat flux from the TRNSYS model agreed well with the measurements. The measurements of the heat flux indicate that the capacity of heat absorption of the radiant panel was about 20-50 W/m^2.

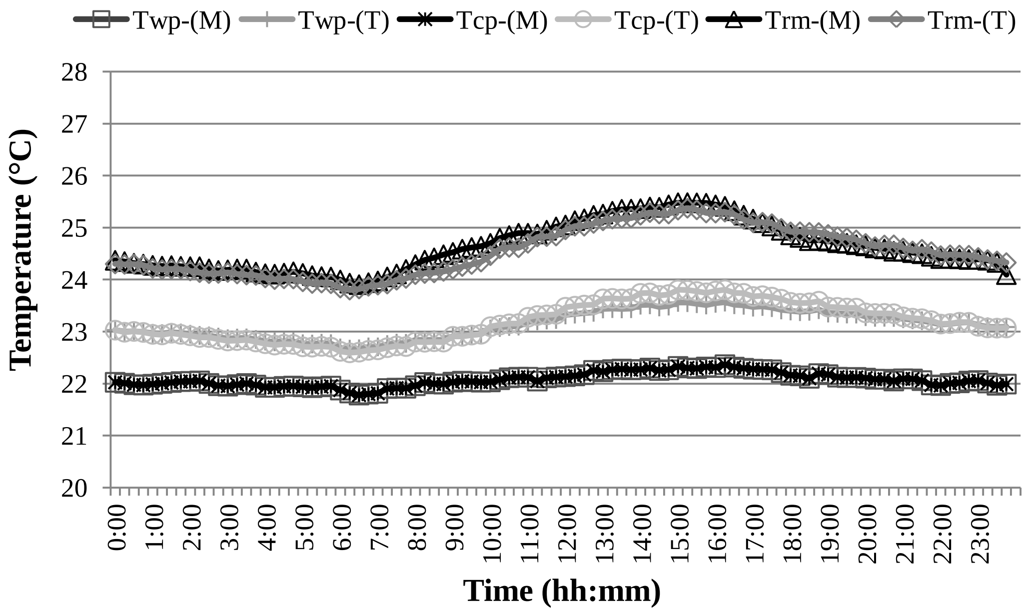
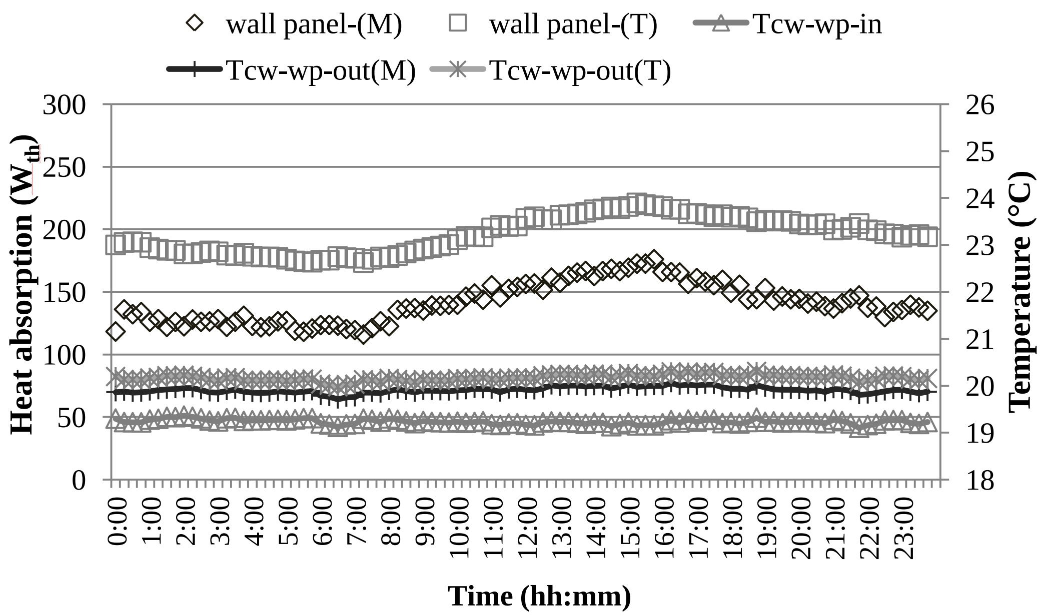
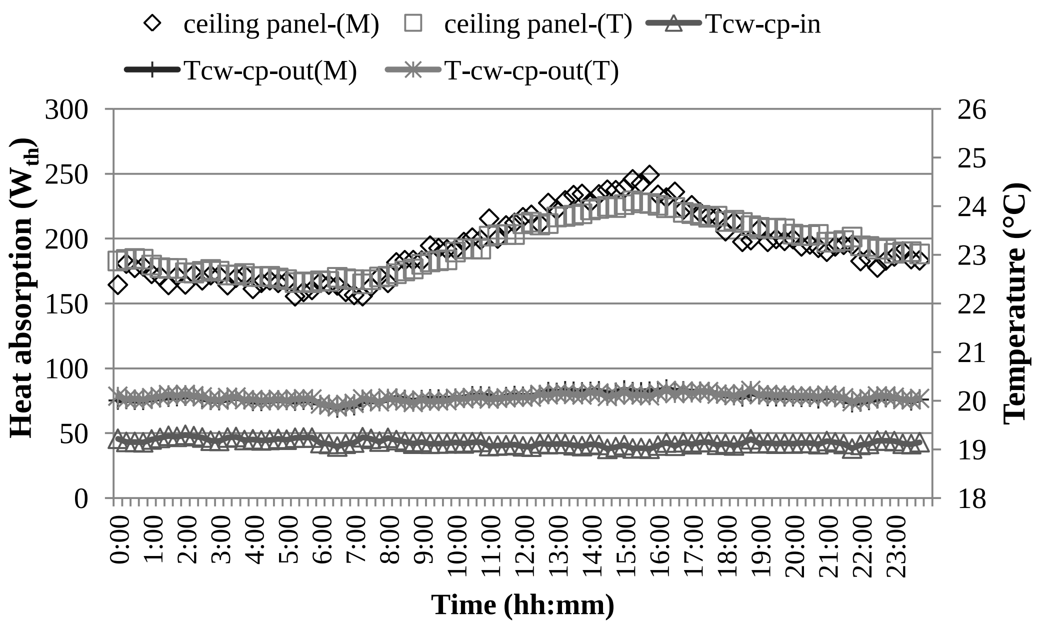


Figure 4.2 Temperature of the room air and surface temperature of the radiant panels on 15 October, 2014.



(a) Wall radiant panel



(b) Ceiling radiant panel

Figure 4.3 Heat absorption by the radiant panels on 15 October, 2014.

Figure 4.4 shows the variations of the dew-point temperature of the room air during the course of the experimental day. The values of the dew-point temperature were derived by using the measured values of the temperature and the relative humidity of the room air. The plot shows that the dew-point temperature was lower than the surface temperature of the radiant panel all the time. There was no condensation of moisture from the room air on the radiant panels.

Figure 4.5 shows the thermal comfort evaluation of the radiant room. In the figure, the plots of the mean radiant temperature and the PMV index were given. The mean radiant temperature values were derived from the measured data of the globe temperature and the temperature of the radiant panels. Compared to the room air temperature, the mean radiant temperature was slightly higher. This results from the radiant heat from the wooden box, and from the wall surfaces with their temperature higher than that of the radiant panels.

In the plot, the measured PMV values were derived from the measured values of the surface temperatures of the walls and radiant panels, and the temperature and relative humidity of the room air. The values of personal variables of metabolic rate and the clothing insulation were set at 1.1 Met and 0.5 Clo, respectively. The plot shows that the PMV values varied within $\pm$0.5 of the neutral comfort. The calculated PMV values from

TRNSYS were also shown in the plot. The calculated results do agree well with the measurements. For the mean radiant temperature, it was found that its values were higher than those the room air. This would result from the thermal radiations from the wooden box and the wall and glazing.

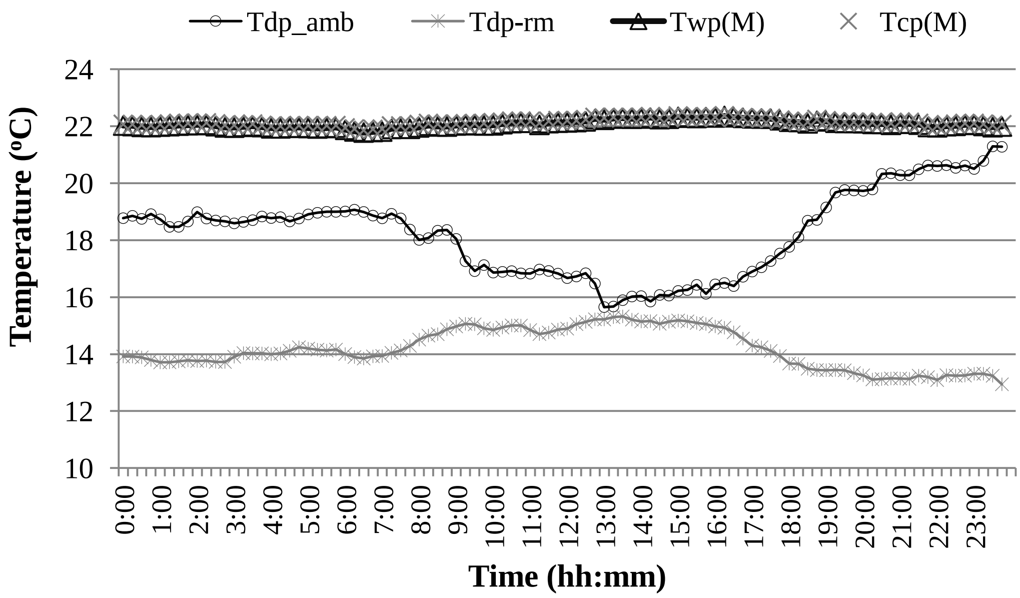


Figure 4.4 Room dew point temperature and temperature of radiant panels on 15 October, 2014.

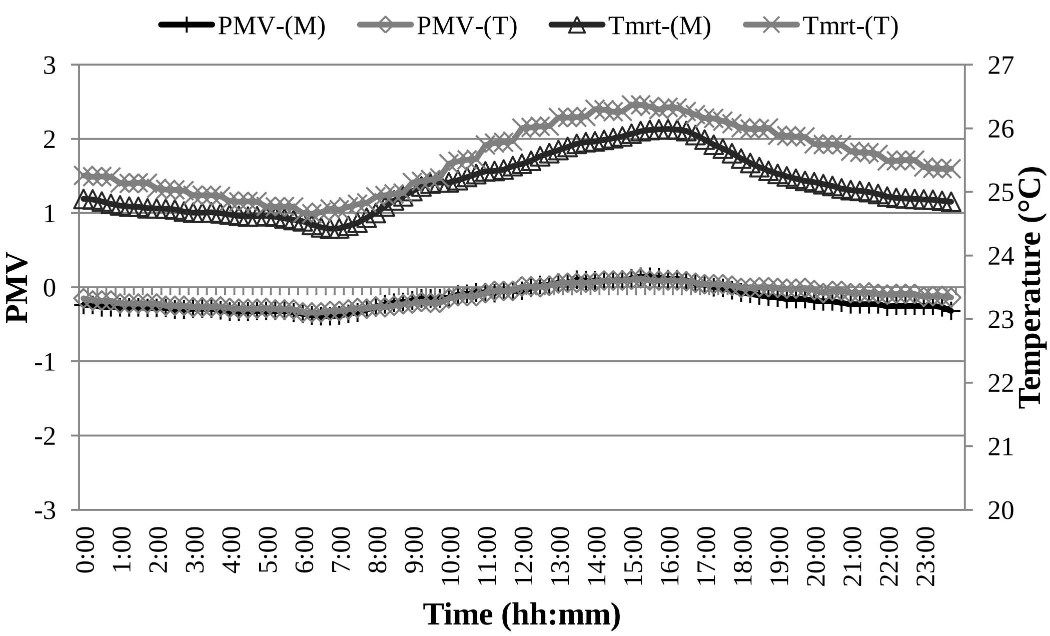


Figure 4.5 PMV and mean radiant temperature on 15 October, 2014.

4.1.2 Radiant Cooling System Equipped with a Fan Coil Unit with Run-Around-Coil

This experiment was identical to the previous experiment, but the conventional fan coil unit was now replaced with a fan coil unit that integrated with a run-around-coil heat recovery. With this additional feature, the fan coil unit supplied the dehumidified ventilation air at higher temperature of 19-25°C.

This experiment was conducted on 26 November, 2014. Figure 4.6 shows the measured solar radiations on the experimental day. In the figure, maximum global radiation was about 800 W/m^2 while the corresponding diffuse radiation measured at the same time was 500 W/m^2. Today, the sky was more clouds compared with the day of the previous experiment. Variations of outdoor air temperature and absolute humidity were also plotted. The outdoor air temperature varied from 26°C up to 35°C while the absolute humidity was between 12-16 g/kg$_{da}$. Although this experiment was performed on a different day but the radiation and the outdoor air condition were comparable.

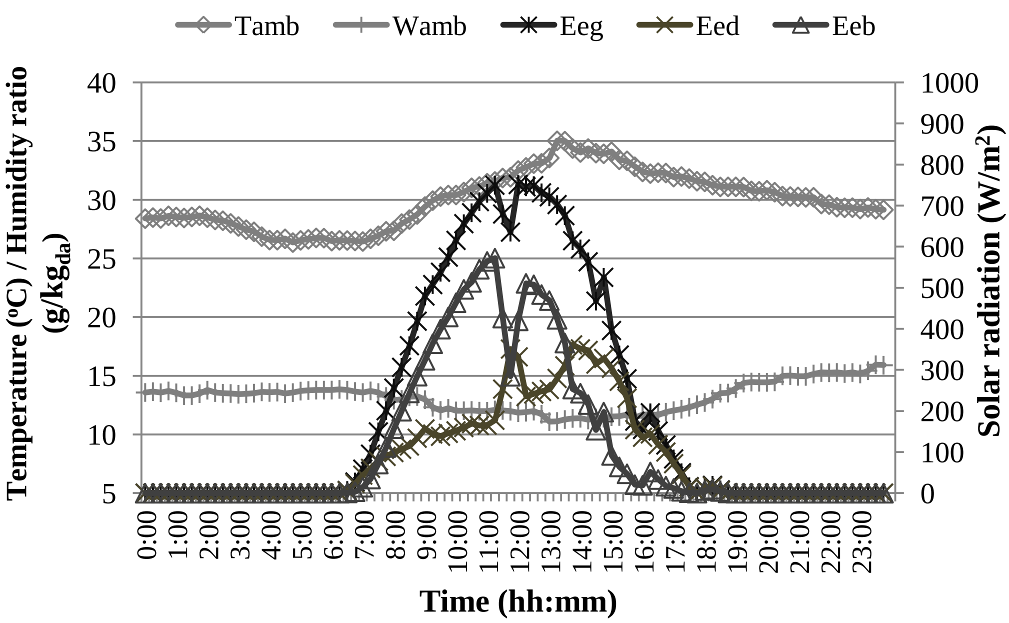


Figure 4.6 Solar radiation and ambient air condition on 26 November, 2014.

Figure 4.7 shows the measured temperature of the room air and of the radiant panel. It was observed that the condition of the room air was quite similar to that of the previous experiment. The air conditioning system kept the room air temperature within a range of 23-25°C, even though the outdoor air in this experiment was supplied to the room at 19-

25°C; much higher compared to that of the previous experiment. As the chilled water was still supplied at 19°C, the surface temperatures of the radiant panels were about 21°C.

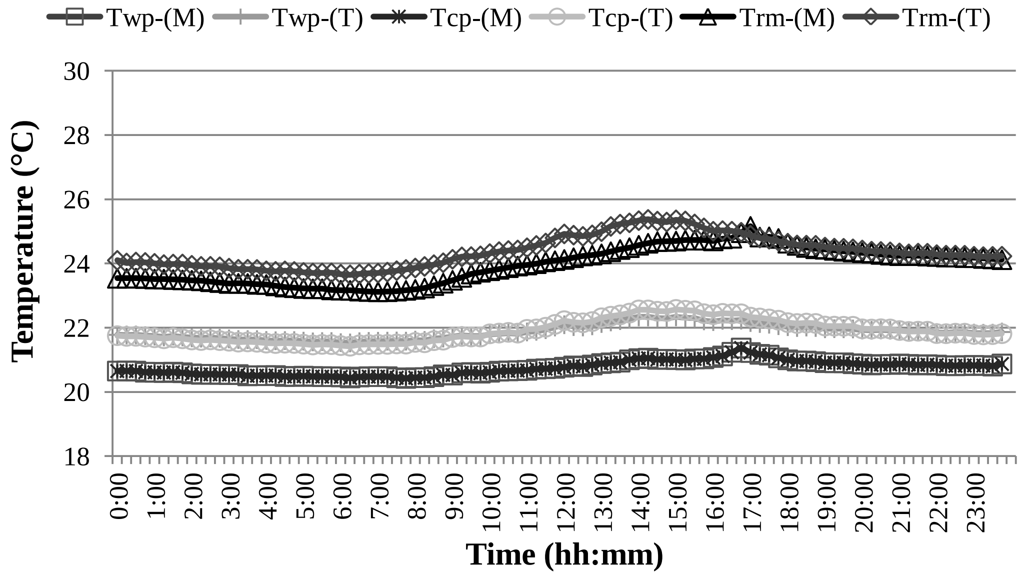
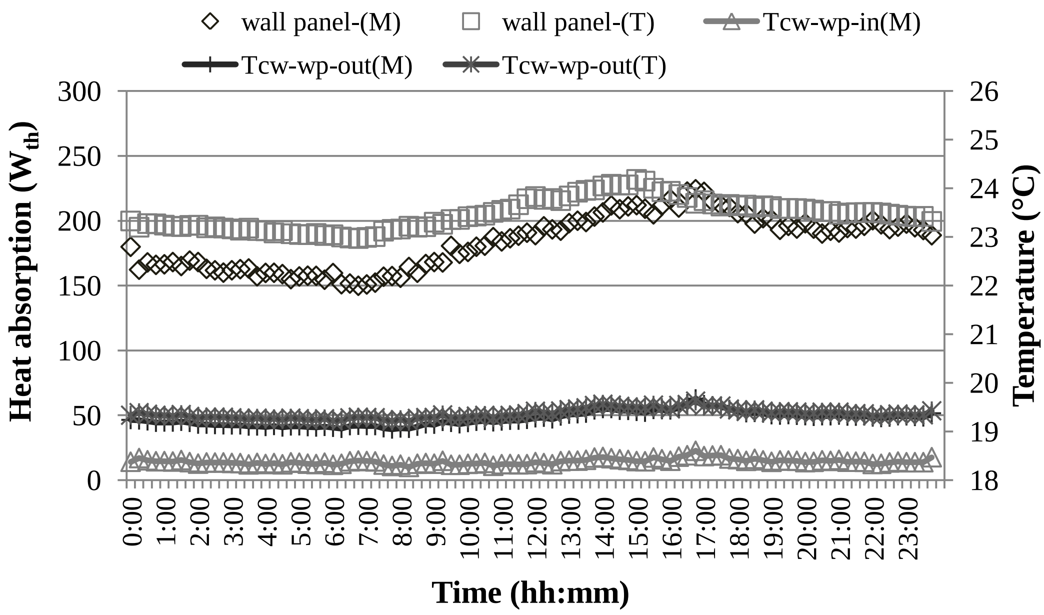


Figure 4.7 Temperature of the room air and surface temperature of the radiant panels on 26 November, 2014.

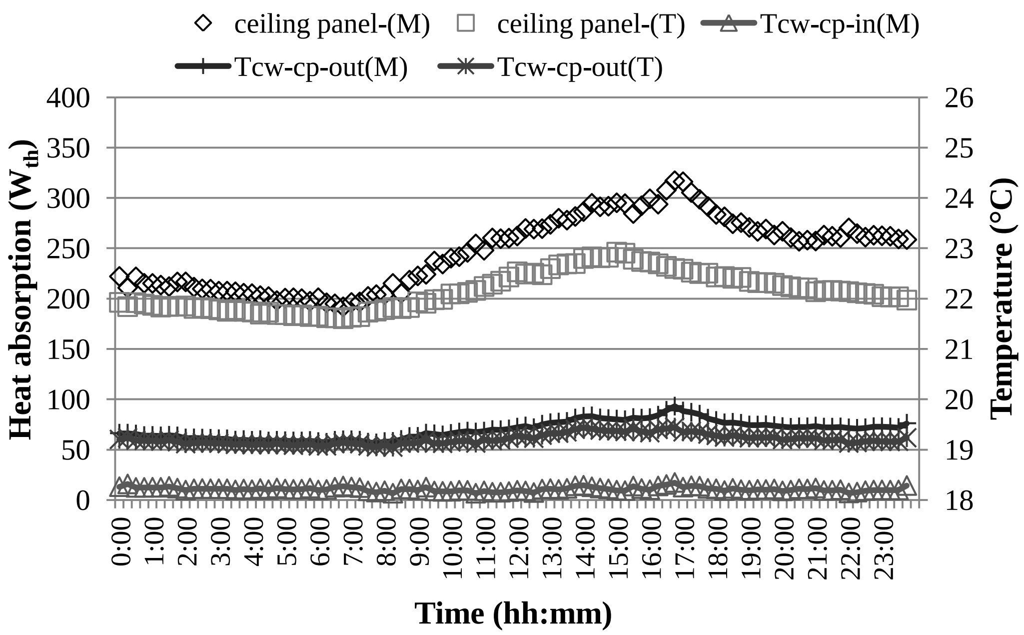
In Fig. 4.7, the calculated values of the room air temperature and the surface temperature of the radiant panels from with those of TRNSYS are also shown. It was observed that the calculated values do agreed well the room air. TRNSYS predicted the surface radiant temperature would be higher than the measurements about 1°C.

Figs. 4.8 (a) and (b) show the heat absorption of the radiant panels. The total load was in a range of 350 W to 500 W. Although the variations of the heat absorption by the panels are similar to the previous case, the amount of the absorbed heat is larger. This would result from the higher room air temperature.

To avoid condensation problems, the temperatures of the cool panels have to be above the room dew point temperature. Figure 4.9 shows the room dew point temperature, temperature of chilled panels, and outside air dew point temperature. Again, the condensation did not occur on the panel surfaces.



(a) Wall panel



(b) Ceiling panel

Figure 4.8 Heat absorption by the radiant panels on 26 November, 2014.

Thermal comfort is evaluated, as shown in Fig. 4.10. The PMV values from the experiment and the simulation are similar to those of the previous case, implying that radiant cooling integrated with heat pipe run-around-coil heat recovery could achieve thermal comfort for the whole day and show good agreement between the experiment and the simulation. The PMV values are in range of -0.5 to 0.5 representing the thermal comfort zone.

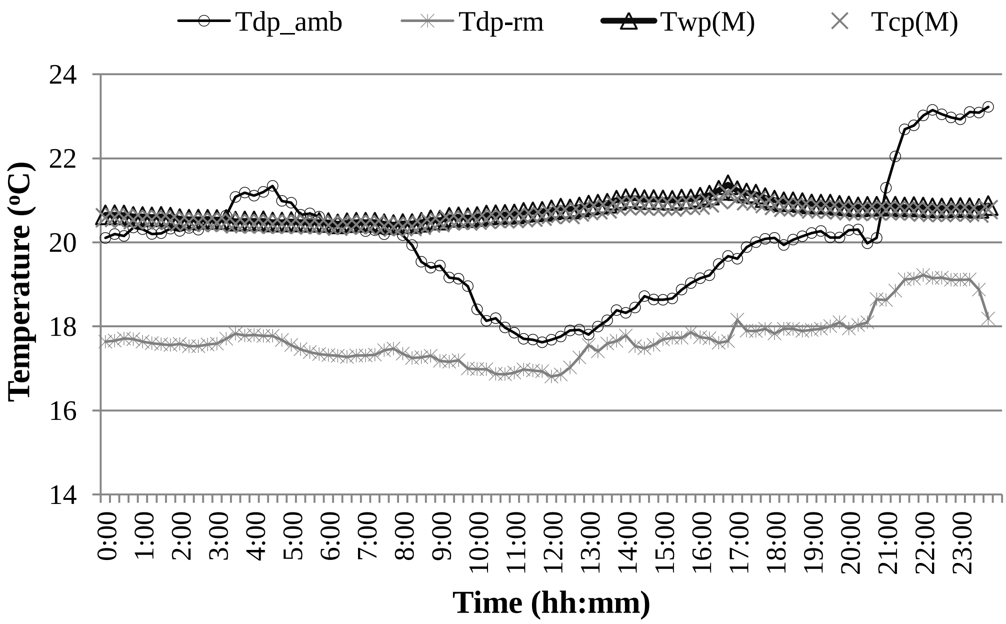


Figure 4.9 Room dew point temperature and temperature of radiant panels on 26 November, 2014.

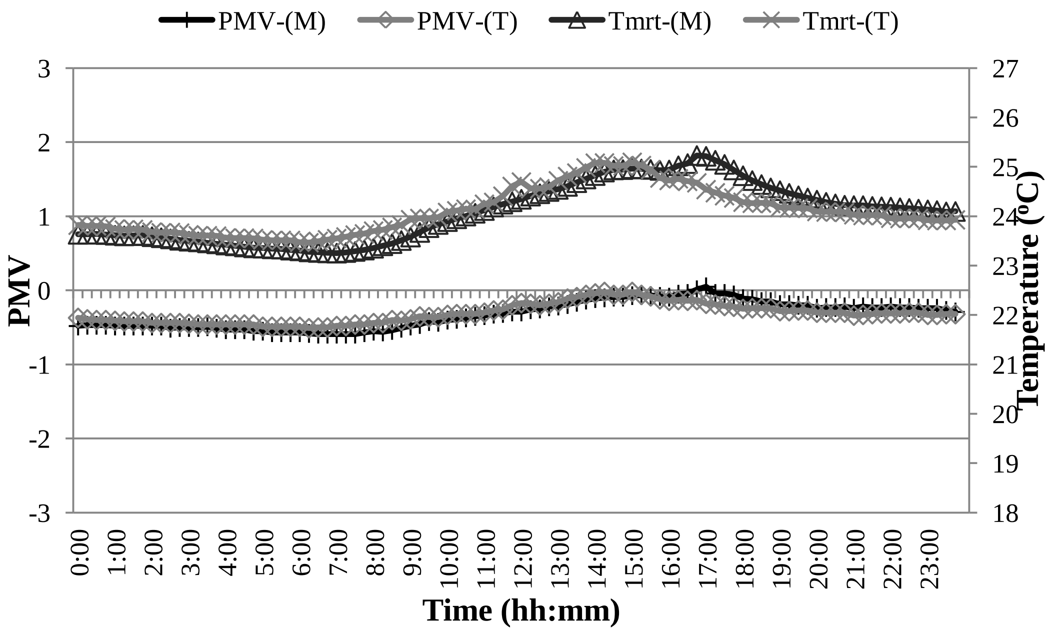


Figure 4.10 PMV values from measurements and TRNSYS simulation on 26 November, 2014.

4.2 Simulation Study

In the simulation study, TRNSYS was used to simulate the radiant room for the whole year. The simulations were separated into four systems (cases):

(i) conventional air-conditioning system,

(ii) radiant cooling system integrated with natural air ventilation,

(iii) radiant cooling system integrated with fan coil unit, and

(iv) radiant cooling system integrated with heat pipe run-around-coil heat recovery.

The results of each system were compared to evaluate the system's performance.

4.2.1 Thailand's Tropical Climate

The climate of Thailand can be classified into four patterns: Cool and dry, Hot and dry, Hot and humid and Late rain. Table 4.1 presents the temperature and absolute humidity statistics of the weather record in year 2000. The weather data was obtained from solar radiation and daylight measurement station in Bangkok Metropolis.

Table 4.1 Statistics of temperature and humidity ratio of the ambient air

Description		Period			
		Cool & Dry (1Nov-15Feb)	Hot & Dry (16Feb-31May)	Early rain (Hot&Humid) (1Jun-15 Aug)	Late rain (16Aug-31Oct)
24 hours operation (0:00-24:00)					
Dry-bulb temperature (°C)	Mean daily minimum	16.85	21.32	23.40	23.54
	Mean daily average	27.55	29.36	29.11	28.66
	Mean daily maximum	38.16	40.66	38.79	38.96
Absolute humidity (kg/kg_{da})	Mean daily minimum	0.00530	0.00750	0.01380	0.01228
	Mean daily average	0.01369	0.01725	0.01755	0.01766
	Mean daily maximum	0.01908	0.02160	0.02080	0.02049

4.2.2 Whole Year Simulation

The previous experimental results already showed that radiant cooling could be applied to provide thermal comfort in one day in a hot and humid climate. In this section, simulation will be performed to investigate use of the radiant cooling in comparison with conventional air-conditioning for the whole year.

TRNSYS was again used for the study. The input data of the TRNSYS simulation for 24 hours were the weather data of central Thailand in the year 2000 in a room model represented the experimental room in the earlier investigation. The internal load was consisted of 65 W of sensible load, 55 W of latent load, and 40 W of electric lighting load. The ventilation air flow rate was identical to that used in the experiment, at 30 kg/hr. The represented room model as the experimental model has shown in prior experiences and experimental results obtained could be used in a simulation results.

This simulation also assumed that the occupant was in the seat in a very light writing position. The room was equipped with 2 radiant cooling panels: the one at the west wall and the other one at the ceiling. Each of the panels has an area of 5.76 m^2. Results of a series of whole-year simulations using TRNSYS computer code on applications of conventional air-conditioning system, radiant cooling system integrated with natural ventilation and integrated with the outdoor air units are described as follows. Table 4.2 shows the parameters values for yearly simulation study for all case.

Table 4.2 Assumptions of the simulation for all cases

Category	**Input data**
Geographical location	13.57°N, 100.5°E
Weather condition	Weather data of year 2000
Infiltration rates	0.5 ACH, constant rate
24-hours occupancy	
Duration hours (hrs)	0:00 to 24:00
Human (Seated, very light writing)	
Sensible heat (W/person)	65
Latent heat (W/person)	55
Light (1 lamp, W)	40
Radiative part	60%
Convective part	40%
Humidity	0
Metabolic rate of occupant (met)	1.1
Clothing insulation (clo)	0.5

- **Case 1: Conventional air-conditioning system**

In the simulation, the air supply temperature was set as a constant at 12°C, and hysteresis controller was applied to switch the system on and off to maintain the room air temperature at 24.5°C. The rate of air circulation was 200 kg/hr.

- **Case 2: Radiant Cooling with Natural Ventilation**

This case was set as the base case for radiant cooling by compared to other the cases. Chilled water temperature was set constant at 18°C. The simulation aimed to investigate the performances of radiant cooling without dehumidified ventilation in tropical climate although the condensation problem actually occurred.

- **Case 3: Radiant Cooling with Fan Coil Unit**

In Case 3, the fan coil unit was equipped with the radiant system. It was used as the outdoor air unit to dehumidify the outdoor air before it is supply into the room so that the condensation problem can be avoided. The temperature of the dehumidified outdoor air was 12°C.

- **Case 4: Radiant Cooling with Fan Coil Unit Integrated with Run-Around-Coil**

Case 4 was similar to Case 3, but the fan coil unit was integrated with the heat pipe run-around-coil. In this case, the dehumidified outdoor air was supplied into the room with higher temperature than that in Case 3.

a) Simulation results

Table 4.3 summarizes the simulation results for the four study cases. In general, all systems achieved thermal comfort. The conventional air-conditioning system (Case 1) can provide thermal comfort for more than 90% of time in each period and in whole year.

For Case 2, as the outdoor air was not dehumidified, the thermal comfort was achieved mainly in the cool and dry periods. For Case 3, thermal comfort condition improves from that of Case 2 since the outdoor air is dehumidified. However, Case 3 can achieve neutral thermal comfort only 67.2% in the period of cool and dry. The people feel too cool since the outdoor air is supplied at 12°C. Examining Case 4, the radiant room can achieve the neutral thermal comfort for more than 80% of each period. The results also show that for Case 3, if the thermal comfort is apart from the neutral, the condition tends to be slightly cool or cool. This is opposite to Case 4.

Table 4.3 The simulation results for thermal sensation achieve by the four cases of the air-conditioning systems.

Period	Case	Thermal sensation (%)				
		Too warm	Warm	Comfort	Cool	Too cool
Cool & Dry	1	0.0	0.3	99.7	0.0	0.0
	2	0.1	22.3	71.1	6.3	0.1
	3	0.0	0.0	67.2	30.6	2.1
	4	0.0	1.9	85.5	12.3	0.3
Hot & Dry	1	0.0	6.1	93.9	0.0	0.0
	2	9.4	40.7	49.9	0.0	0.0
	3	0.0	0.7	91.3	7.9	0.0
	4	0.0	16.6	83.1	0.3	0.0
Hot & Humid	1	0.0	5.9	94.1	0.0	0.0
	2	8.4	41.5	50.1	0.0	0.0
	3	0.0	0.2	97.0	2.8	0.0
	4	0.0	15.9	84.1	0.0	0.0
Late Rain	1	0.0	8.7	91.3	0.0	0.0
	2	5.8	37.2	57.0	0.0	0.0
	3	0.0	0.0	93.5	6.5	0.0
	4	0.0	11.9	88.1	0.0	0.0
Yearly	1	0.0	4.9	95.1	0.0	0.0
	2	5.7	34.8	57.7	1.9	0.0
	3	0.0	0.3	85.9	13.2	0.6
	4	0.0	11.2	85.1	3.7	0.1

Table 4.4 exhibits the interior temperature of the air-conditioned room. For Case 1 of the conventional air conditioning, the temperature of the room air can be kept constant at 24.5°C. The mean radiant temperature varied within 27.9-28.6°C across the four different weather patterns.

For Case 2, it was found that the room air was higher than that of Case 1. The room air was cooled by only the radiant panels which are not effective. Temperature of the outdoor air supplied to the room was also rather high. Since Case 2 has no dehumidified process of the outdoor air, the room air is quite humid; the values of absolute humidity are higher than 19 g/kg$_{da}$ for most of the time.

For Cases 3 and 4, the results in the table show that the mean radiant temperature of the room was lower than that of Case 1. This lead Case 3 and 4 can achieve the neutral thermal comfort at higher room air temperature. It should be noted that the air velocity within the room of Case 3 and 4 is also low than Case 1. These results demonstrate the radiant cooling system achieve the neutral thermal comfort by mode of the radiative heat transfer.

Table 4.4 The simulation results of the interior temperature of the air-conditioned room

Period	Case	T_{cw} (°C)	T_{room} (°C)	T_{vent} (°C)	T_{mrt} (°C)	T_{op} (°C)	$T_{dp-room}$ (°C)	Absolute Humidity (g/kg$_{da}$)
Cool & Dry	1	-	24.5	12.0	27.9	26.2	17.2	12.577
	2		25.3	27.5	24.7	25.0	20.7	15.541
	3		23.5	12.0	24.1	23.8	16.5	11.783
	4		24.7	20.6	24.5	24.6	16.9	12.052
Hot & Dry	1	18	24.5	12.0	28.6	26.5	17.9	12.960
	2		26.1	29.4	25.3	25.7	24.0	19.096
	3		24.0	12.0	24.7	24.4	18.2	13.098
	4		25.4	22.7	25.2	25.3	18.4	13.242
Hot & Humid	1	18	24.5	12.0	28.6	26.6	17.9	12.996
	2		26.0	29.1	25.3	25.7	24.3	19.399
	3		24.1	12.0	24.7	24.4	18.4	13.233
	4		25.4	22.7	25.2	25.3	18.5	12.996
Late Rain	1	18	24.5	12.0	28.4	26.5	18.3	13.439
	2		25.8	28.7	25.2	25.5	24.4	19.502
	3		23.9	12.0	24.6	24.2	18.4	13.286
	4		25.3	22.6	25.0	25.1	18.6	13.392

Figure 4.11 shows the percentage of time the moisture had condensed on the radiant panels. It was observed that with the dehumidification of the outdoor air, no condensation occur in the radiant room.

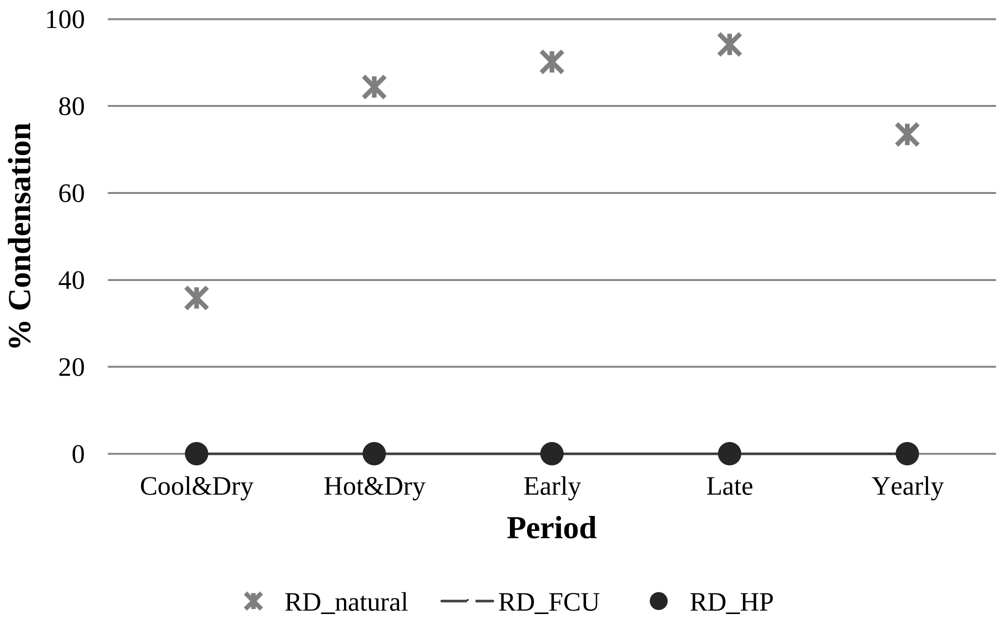


Figure 4.11 Percentage of condensation occurring on the radiant panels.

b) Temperature adjustment of radiant panel to improve thermal comfort

In the previous section, the simulations of radiant cooling had assumed that the temperature of the chilled water supply was constant at 18°C throughout the year. The results show no radiant system can provide the neutral comfort for all periods of the different weather patterns. The radiant system of Case 3 tends to over-cool the radiant room in the cool and dry period. For the system of Case 4 can provide the neutral comfort in the cool and dry period by it does under-cool the room in other periods.

The simulations in this section examine two additional cases. Case 5 is identical to Case 3, but the chilled water was supplied to the panels at the temperatures shown in Table 4.5. Case 6 is identical to Case 4 but the temperature of the chilled water is set as shown in Table 4.6.

Table 4.5 The input data of adjustment of chilled water temperature of case 5

Period	Input data
Cool & Dry	19 (1 Jan – 15 Feb) , 21 (1 Nov – 31 Dec)
Hot & Dry	18
Hot & Humid	18
Late Rain	18

Table 4.6 The input data of adjusted chilled water temperature of Case 6

Period	Input data
Cool & Dry	18 (1 Jan – 15 Feb) , 20 (1 Nov – 31 Dec)
Hot & Dry	17
Hot & Humid	17
Late Rain	17

Table 4.7 shows the simulation results of Case 5 and Case 6 as compared to Case 1. It can be observed that the radiant room can now achieve the neutral thermal comfort more than 90% of time in each weather period.

Table 4.7 Thermal sensation results from the simulation of adjust-temperature chilled water supplied to the panels as compared with conventional air-conditioning systems

Period	Case	Thermal sensation (%)				
		Too warm	Warm	Comfort	Cool	Too cool
Cool & Dry	1	0.0	0.3	99.7	0.0	0.0
	5	0.0	0.7	94.8	5.1	0.0
	6	0.0	1.4	94.4	4.1	0.0
Hot & Dry	1	0.0	6.1	93.9	0.0	0.0
	5	0.0	0.7	91.4	7.9	0.0
	6	0.0	4.7	92.6	2.7	0.0
Hot & Humid	1	0.0	5.9	94.1	0.0	0.0
	5	0.0	0.2	97.0	2.8	0.0
	6	0.0	3.1	96.9	0.0	0.0
Late Rain	1	0.0	8.7	91.3	0.0	0.0
	5	0.0	0.0	93.5	6.5	0.0
	6	0.0	1.8	97.9	0.3	0.0
Yearly	1	0.0	4.9	95.1	0.0	0.0
	5	0.0	0.3	94.0	6.5	0.0
	6	0.0	2.8	95.2	2.0	0.0

Table 4.8 shows the corresponding interior room environment of Case 5 and Case 6. It should be noted that the mean radiant temperature of the radiant room is lower than that of the room using conventional air-conditioning system.

Table 4.8 The room-characteristic simulation results of 4 cases for 24 hours operation in each period after adjustment of chilled water temperature

Period	**Case**	**Troom (°C)**	**Tvent (°C)**	**Tmrt (°C)**	**Top (°C)**	**Tdp-room (°C)**	**Absolute Humidity (g/kg_{da})**
Cool & Dry	1	24.5	12.0	27.9	26.2	17.2	12.577
	5	24.4	12.0	25.1	24.7	16.5	11.783
	6	25.0	19.5	25.0	25.0	16.9	12.084
Hot & Dry	1	24.5	12.0	28.6	26.5	17.9	12.960
	5	24.0	12.0	24.7	24.4	18.2	13.098
	6	24.9	22.0	24.7	24.8	18.4	13.251
Hot & Humid	1	24.5	12.0	28.6	26.6	17.9	12.996
	5	24.1	12.0	24.7	24.4	18.4	13.233
	6	24.9	22.0	24.7	24.8	18.5	12.996
Late Rain	1	24.5	12.0	28.4	26.5	18.3	13.439
	5	23.9	12.0	24.6	24.2	18.4	13.286
	6	24.7	22.0	24.5	24.6	18.6	13.394

Table 4.9 shows the sensible cooling load of the radiant cooling systems of Case 5 and Case 6 as compared to the conventional air-conditioning system of Case 1. It can be observed that the sensible loads of the radiant systems are lower than that of the conventional air-conditioning system.

Comparing the two radiant cooling systems, the sensible ventilation load of Case 6 is just about half of that of Case 5. This is because the use of the run-around-coil to pre-cool and reheat the ventilation air. The temperature of the ventilation air of Case 6 is higher than that of Case 5 (see Table 4.8). Figure 4.12 shows that shares of the sensible load of the ventilation air to the total sensible load of the system.

Table 4.9 Sensible load of the radiant cooling systems and of the conventional air conditioning system

Period	Sensible cooling load (W)						
	Case 1	Case 5			Case 6		
	Ventilation & Circulation	Wall panel	Ceiling panel	Ventilation	Wall panel	Ceiling panel	Ventilation
Cool & Dry	11,447	4,572	4,477	3,123	5,634	5,485	1,604
Hot & Dry	16,563	6,343	6,163	3,489	7,531	7,291	1,484
Hot &Humid	16,839	6,375	6,195	3,438	7,562	7,322	1,433
Late Rain	15,909	6,197	6,021	3,347	7,383	7,147	1,338

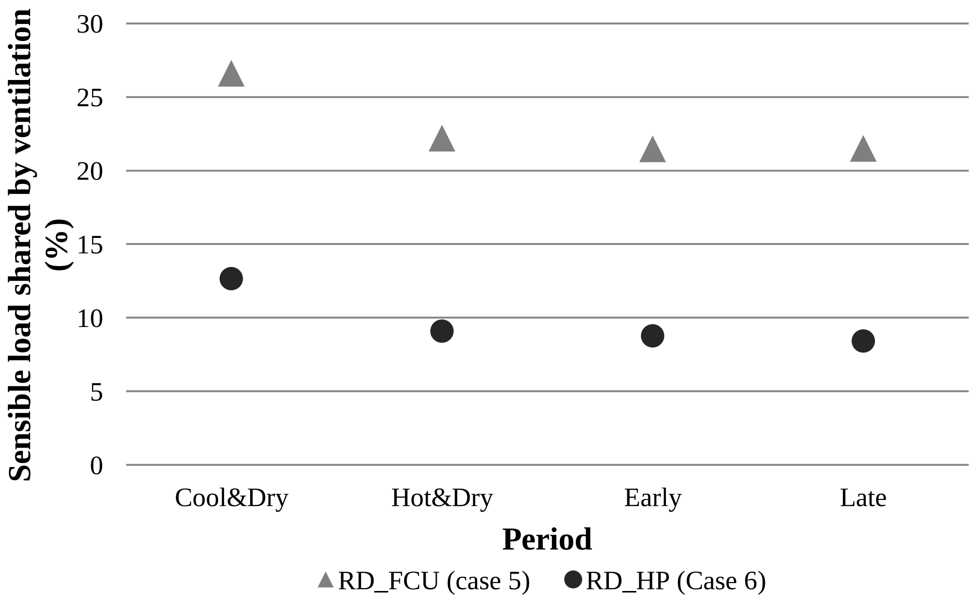


Figure 4.12 Sensible load shared by the ventilation for different operating periods.

The results in this part show that the outdoor air unit with run-around-coil had enhanced the capability of the radiant panel (see Table 4.9).

Table 4.10 examines the latent load of the radiant system and the conventional air conditioning system. It can be observed that the latent loads of the radiant systems are about 15% smaller than that of the conventional system. This should be because of that the radiant systems do not dehumidify the room circulation air.

Table 4.10 Latent load of the radiant cooling systems and of the conventional air conditioning systems

Period	Latent cooling load (W)		
	Case 1	Case 5	Case 6
	Ventilation & Circulation	Ventilation	Ventilation
Cool & Dry	2,363	2,752	2,297
Hot & Dry	5,486	4,393	3,718
Hot & Humid	5,744	4,515	3,837
Late Rain	5,415	4,553	3,848

Table 4.11 shows the resulting energy consumption for air-conditioning by using the conventional system and the radiant systems. In the calculation, the COP of the air-conditioning system is assumed to be 2.7. It is clearly observed that the radiant system consume less energy than the conventional system.

Table 4.11 Evaluation of energy consumption of conventional air-conditioning system and radiant cooling system

Period	Total Cooling load (kWh_{th}) and Energy consumption (kWh_e)							
	Case 1				Case 5		Case 6	
	Cooling Load (kWh_{th})	Energy (kWh_e)	Power of fan (kWh_e)	Energy (kWh_e)	Cooling Load (kW_{th})	Energy (kWh_e)	Cooling Load (kW_{th})	Energy (kWh_e)
Cool & Dry	1477.7	547.3	128.4	675.7	1596.5	591.3	1607.2	595.3
Hot & Dry	2337.1	865.6	127.2	992.8	2160.7	800.2	2123.2	786.4
Hot & Humid	1717.2	636.0	91.3	727.3	1560.1	577.8	1531.9	567.4
Late Rain	1641.8	608.1	92.4	700.5	1548.6	573.6	1518.1	562.2
Year	7767.1	2876.7	219.6	3096.3	6865.9	2542.9	6780.4	2511.2

Figure 4.13 shows the energy saving potential of the radiant system. Compared with the conventional system, the radiant system consumes energy less than 25%.

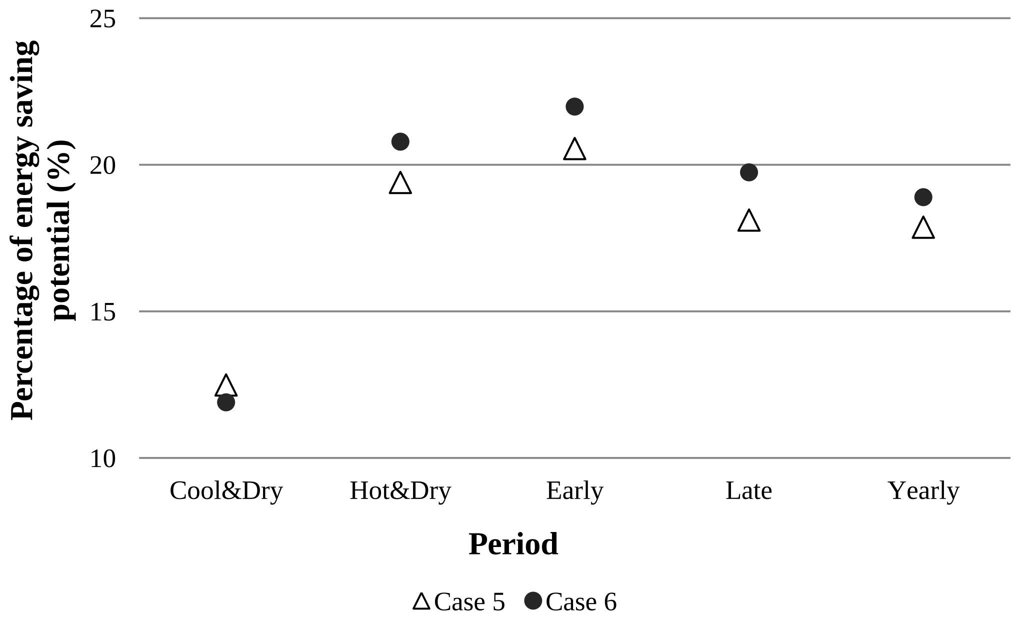


Figure 4.13 Energy saving potential of radiant cooling system.

4.3 Parametric Study of Radiant Cooling System

In this section, parametric studies were performed to determine the influence of air infiltration and of wall insulation on the radiant system. Only the radiant system with the outdoor air unit equipped with a round-around-coil was examined.

4.3.1 Influence of Air Infiltration

In Thailand, buildings are constructed with less concern about the leakage between the indoor air and outdoor air through the building envelope. For the radiant system, the air infiltration could increase the moisture of the indoor air, and eventually cause condensation on the radiant panel surfaces.

In the parameters, the air infiltration was assumed to vary among 0.0, 0.5, 1.0, and 1.5 ACH. Figure 4.14 shows percentage of condensation could occur when there is infiltration to the room. Infiltration should be limited to less than 0.5 ACH to guarantee no condensation.

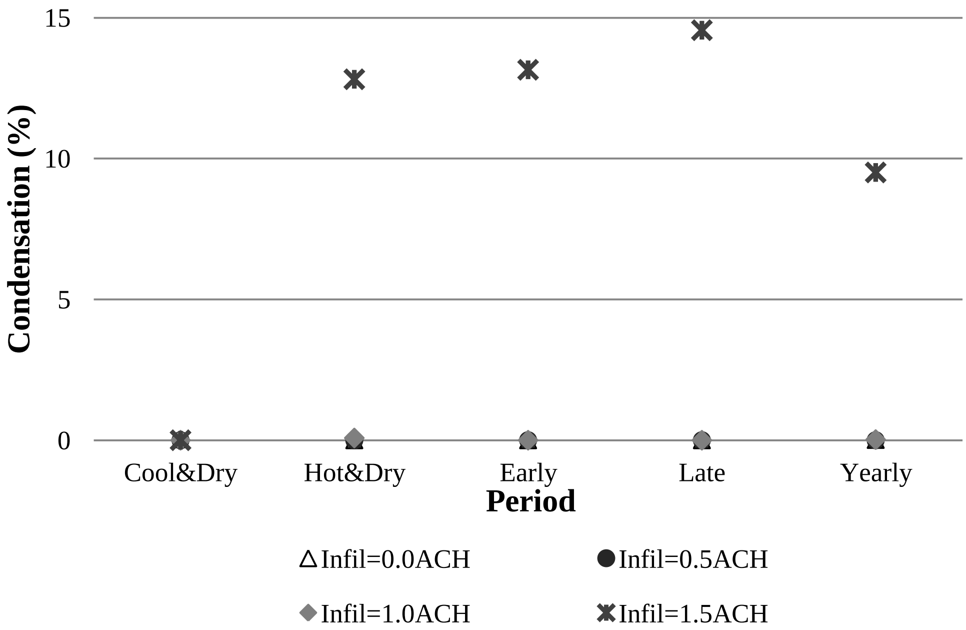


Figure 4.14 Condensation as function of air infiltration.

Figure 4.15 shows the influence of the air infiltration on thermal comfort. In the figure, the increase of air infiltration increases moisture in the room and reduces thermal the comfort level. The infiltration at a rate of 0.5 ACH reduces the neutral thermal comfort level of about 2%.

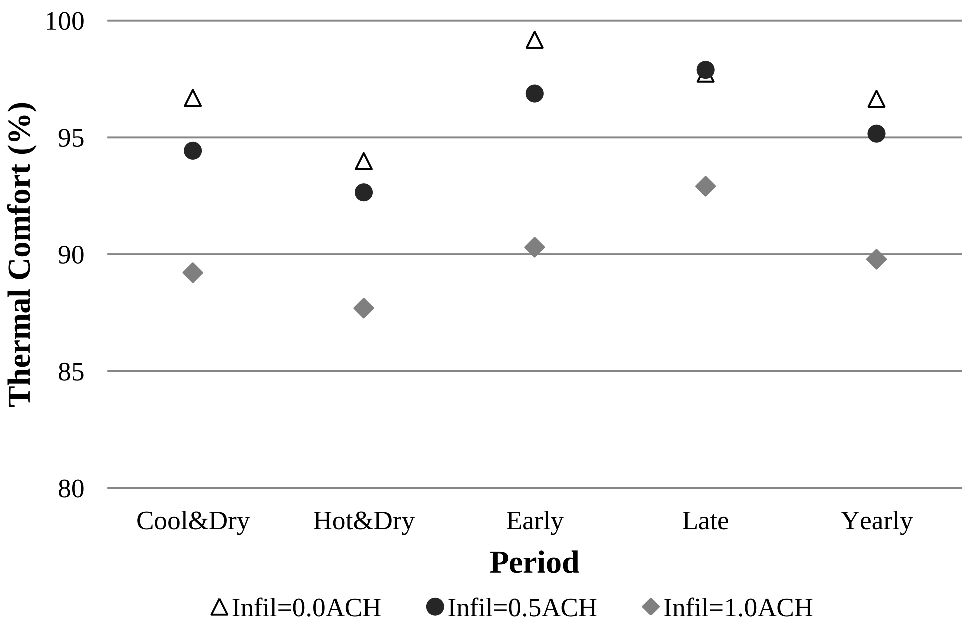


Figure 4.15 Thermal comfort as functions of the operating parameters of infiltration.

4.3.2 Influence of Insulation

The radiant system is rather low and has a slow cooling rate as compared to a conventional system. To solve this issue, the wall of the radiant room required insulation. In this parametric study, the radiant room is assumed to be adhered with polyurethane insulation on north wall (Other exterior wall already insulated). The insulation thickness was varied from 0.05 m, 0.10m, 0.15m, 0.20m, 0.25m, and 0.35m. Chilled water flow rate was supplied at 240 kg/h.

Figure 4.16 shows the rate of the heat absorbed by the radiant panels. It can be observed that increasing the insulation thickness would reduce the heat absorbed by the radiant panels since the heat gained through wall is reduced. However, the thickness of 5 cm. seems to be enough.

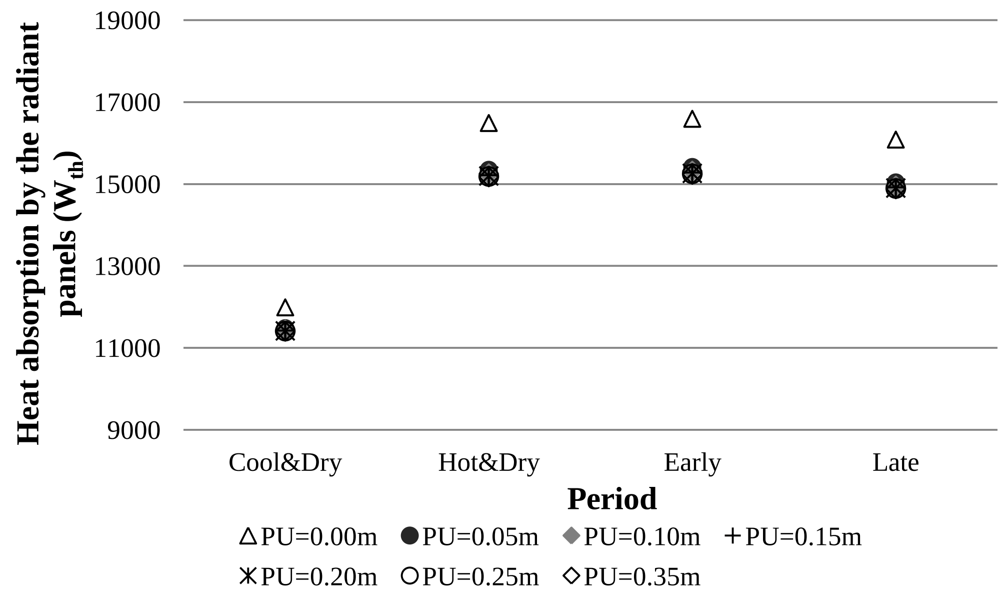


Figure 4.16 Influence of insulation on the heat absorbed by the radiant panels.

Figure 4.17 shows that the insulation is the important factor for achieving thermal comfort. Inclusion of the insulation can improve the neutral comfort level up to 10%.

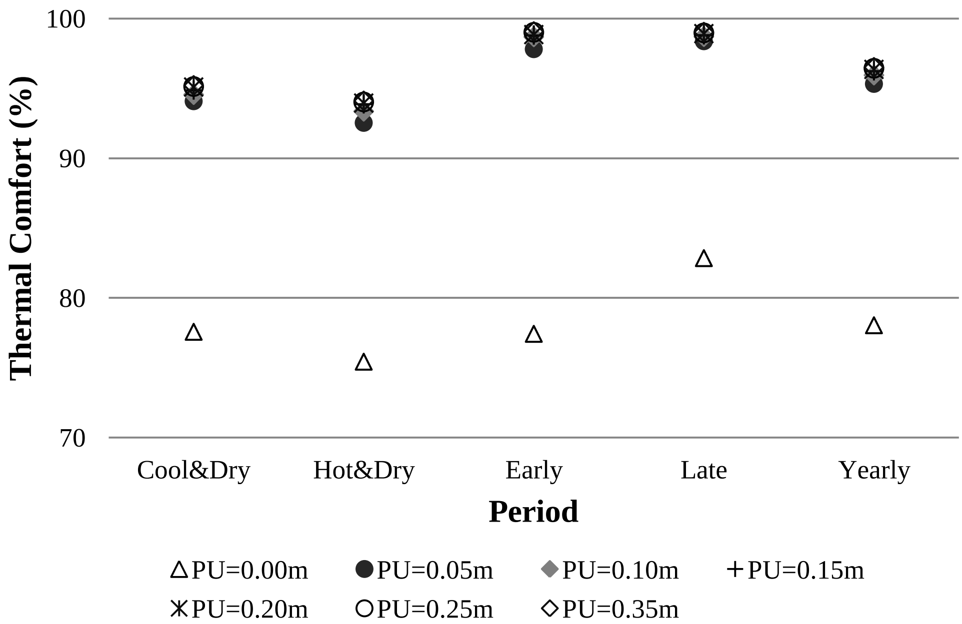


Figure 4.17 Influence of insulation on the thermal comfort.

CHAPTER 5

CONCLUSION

In this thesis, radiant cooling for air-conditioning was investigated in Thailand's hot and humid climate. A radiant cooling system coupled with an outdoor air unit was installed for an office-like room for physical experiments. For the radiant system, the radiant cooling panels were used to absorb the sensible heat within the room, while the outdoor air unit was used to handle the load of the ventilation air. A number of sensors were installed to measure the cooling loads of the radiant panels and of the outdoor air unit. With the measurements, the thermal comfort condition within the room was also evaluated by using Predicted Mean Vote index (PMV).

The experimental results show that the radiant system can provide the neutral thermal comfort level of PMV=$\pm$0.5 throughout the experimental periods. The chilled water could be supplied to the radiant panels at 18°C with no condensation at the panels. At the panel surface temperature of about 20°C, the rate of heat absorption was about 40-50 W/m^2. During the experiments, the radiant room was also modeled by using TRNSYS software. It was found that the calculated results from the model fitted the experimental measurements well.

In the simulation study, the model was used to simulate the interior thermal environment of the radiant room under the different climate conditions in a year (i.e. cool and dry, hot and dry, hot and humid, and late rain). The separate model was also established to simulate the same room, but equipped with the conventional all-air air-conditioning system. The results show that the radiant system can achieve the same comfort level as the convention system with a smaller cooling load (about 18.6 percent reduction). The outdoor air unit shared about 10 percent of the total sensible load.

The simulations were also performed to examine the influence of air infiltration into the radiant cooling room. The results show that in order to achieve the thermal comfort (more than 90% of time) with no condensation, the infiltration has to be limited to not exceeding 1.0 air change per hour. Under the intense solar radiation in the tropical region, the radiant room requires the insulated walls to minimize the external heat gain. For the modeled room, the polyurethane form of 5 cm. thick is sufficient.

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APPENDIX A

Properties of the Wall and Radiant Panels in the Experimental Room

Table A.1 Properties of the compositions of wall materials

Composition	Thickness (m)	Density (ρ), kgm^{-3}	Conductivity (k), $Wm^{-1o}K^{-1}$	Capacity (C_p), $kJkg^{-1}K^{-1}$	Emittance (ε)	Absorbtance (α)
West wall						
- Chilled ceiling panel	-	-	-	-	0.90	0.10
- Plaster	0.015	1858	0.72	0.837		
- Concrete block	0.075	1370	1.02	0.92		
- Polyurethane foam	0.0762	32.04	0.023	0.59		
- Concrete block	0.075	1370	1.02	0.92		
- Plaster	0.015	1858	0.72	0.837	0.82	0.25
East wall						
- Plaster	0.015	1858	0.72	0.837	0.82	0.25
- Concrete block	0.150	1370	1.02	0.92		
- Plaster	0.015	1858	0.72	837	0.82	0.25
North wall						
- Smart board	0.010	1260	0.084	0.95	0.82	0.25
- Polyurethane foam	0.010 (0.0254)	32.04	0.023	1.21		
- Concrete block	0.075	1370	1.02	1760		
- Plaster	0.015	1858	0.72	0.837	0.82	0.25
South wall						
- Plywood	0.010	528	0.138	1.21	0.82	0.6
- Polyurethane foam	0.001 (0.0254)	32.04	0.023	1.21		
- Plywood	0.010	528	0.138	1.21	0.82	0.6
Ceiling						
- Chilled ceiling panel	-	-	-	-	0.90	0.10
- Plaster	0.015	1858	0.72	0.837		
- Concrete slab	0.010	2400	1.442	0.92		
- Fiber glass (Staycool)	0.0762	12	0.038	0.96	0.05	0.05
Floor						
- Granite	0.008	1200	0.338	0.80	0.82	0.9
- Concrete	0.100	1858	1.442	0.837	0.82	0

Table A.1 Properties of the compositions of wall materials (Cont')

Composition	Thickness (m)	Density (ρ), kgm^{-3}	Conductivity (k), $Wm^{-1o}K^{-1}$	Capacity (C_p), $kJkg^{-1}K^{-1}$	Emittance (ε)	Absorbtance (α)
Roof						
- Ultracool (siam fiber glass)	0.0762	16	0.036	0.96	0.05	0.05
- Roof tile	0.013	2400	0.993	0.92	0.82	0.6

Figure A.1 A photograph of chilled water tubes behind the radiant panel.

Figure A.2 A photograph of chilled water tubes inside heat pipe run-around-coil heat recovery.

APPENDIX B

Diagrams and Tables of the Simulation Configurations of TRNSYS

These figures in Appendix B are shown for the TRNSYS diagram used in the simulation study.

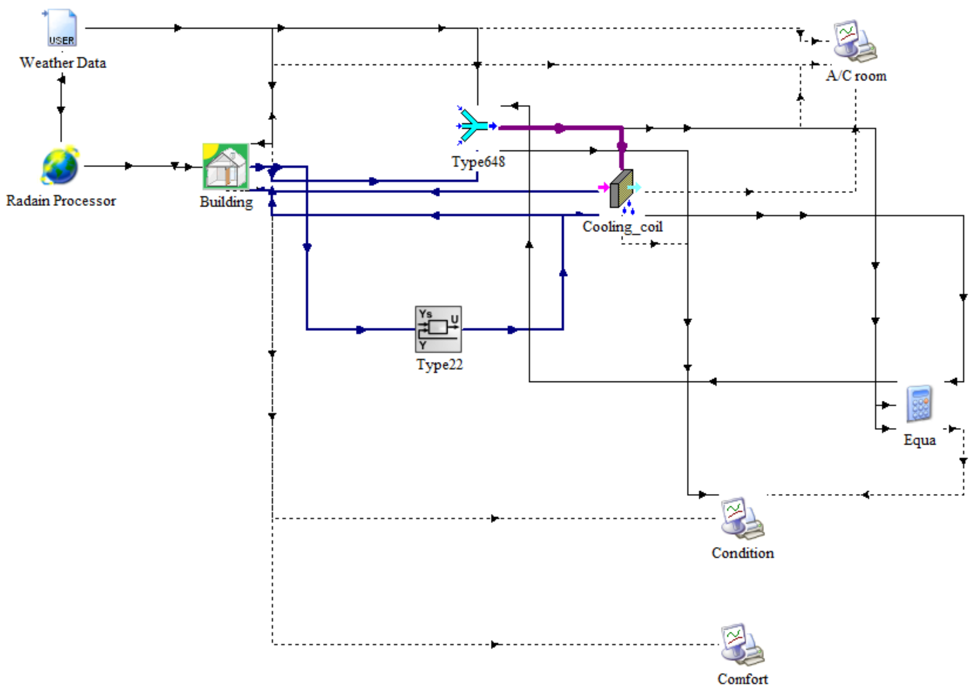

Figure B.1 A diagram of the simulation of a conventional air conditioning system (Case 1).

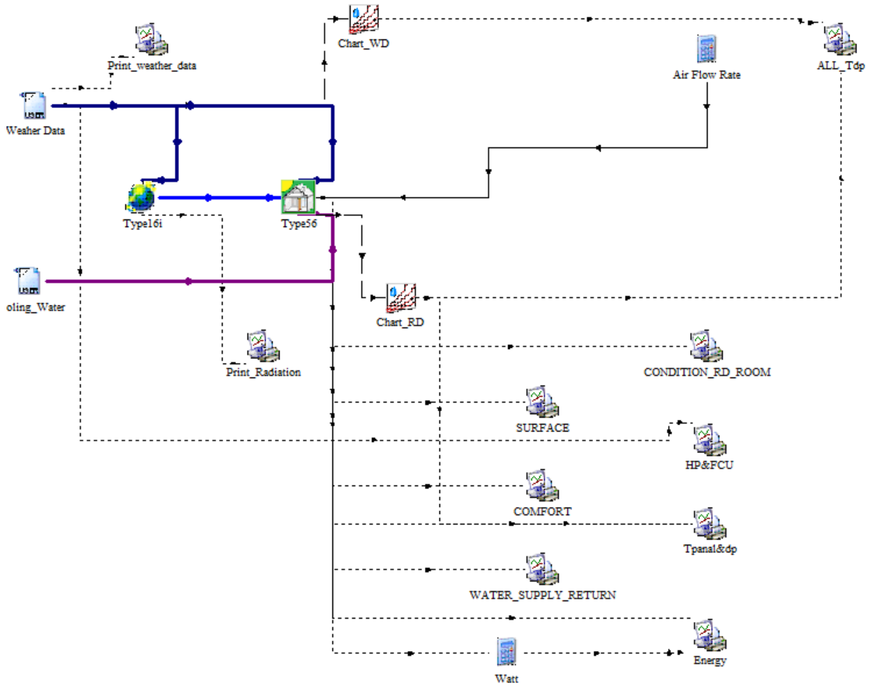

Figure B.2 A diagram of the simulation of radiant cooling integrated with natural air ventilation (Case 2).

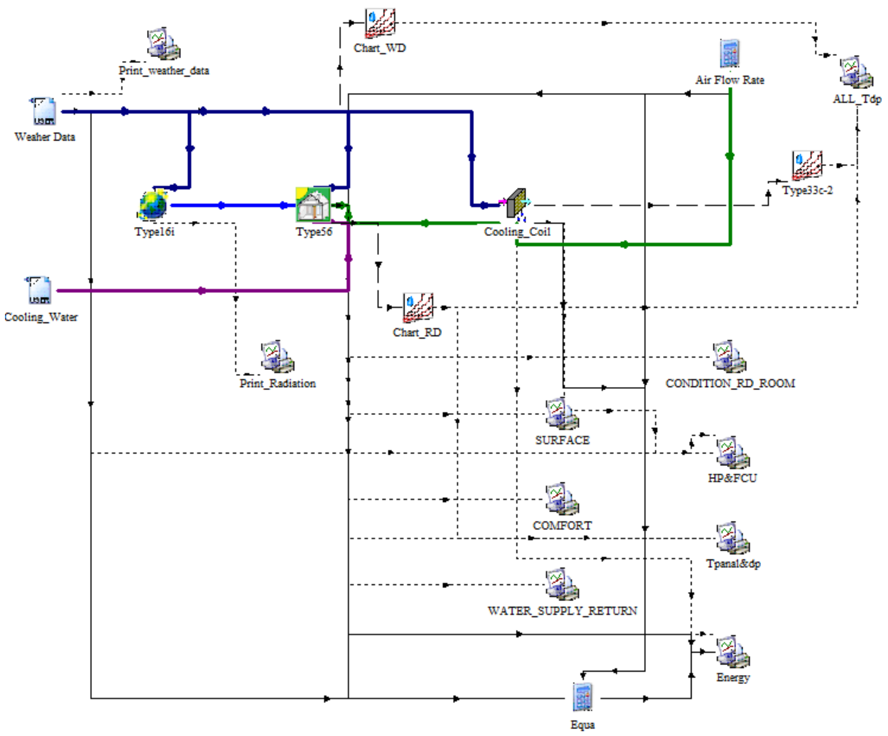

Figure B.3 A diagram of the simulation of radiant cooling integrated with fan coil unit dehumidified air ventilation unit (Cases 3 and 5).

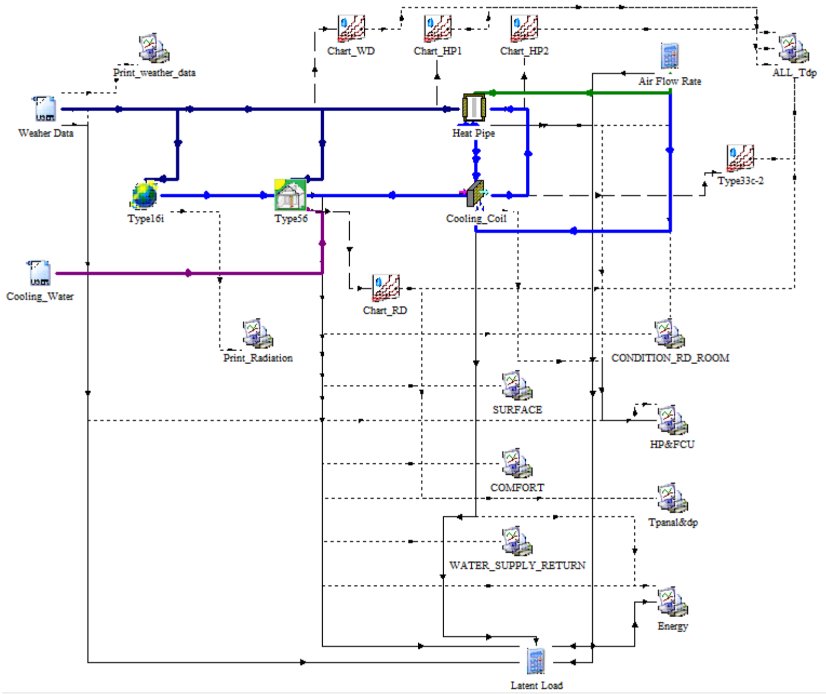

Figure B.4 A diagram of the simulation conventional air conditioning with heat pipe run-around-coil dehumidified air ventilation unit (Cases 4 and 6).

For the TRNSYS base-case simulation, the details of all configurations carried in this study are presented in Table B.1.

Table B.1 The details of configurations of cooling panels

Panels	Details	Thickness (m)
West wall	Aluminium plate	0.001
	Copper tube	0.0005
	Polyurethane foam	0.001
	Air gap	0.03
	Plaster	0.015
	Concrete block	0.075
	Polyurethane foam	0.0762
	Concrete block	0.075
	Plaster	0.015
	Total thickness	**0.2887**
Ceiling	Aluminium plate	0.001
	Copper tube	0.0005
	Polyurethane foam	0.001
	Air gap	0.03
	Plaster	0.015
	Concrete slab	0.010
	Fiber glass (Staycool)	0.0762
	Total thickness	**0.1337**
Pipe spacing		**0.145**
Pipe inside diameter		**0.0012**