รายงานการวิจัยฉบับสมบูรณ์

โครงการพัฒนาและสาธิตการใช้แผงเย็นให้ความเย็นในอาคารโดยวิธีธรรมชาติ

คณะวิศวกรรมศาสตร์ มหาวิทยาลัยสงขลานครินทร์
วิทยาเขตหาดใหญ่

ผู้วิจัย ผศ.ดร.จันทกานต์ ทวีกุล

งานวิจัยนี้ได้รับทุนอุดหนุนการวิจัย จากเงินรายได้
มหาวิทยาลัยประเภททั่วไปประจำปีงบประมาณ 2548

บทคัดย่อ

บริเวณประเทศที่มีภูมิอากาศแบบร้อนชื้นโดยเฉพาะประเทศไทย จะมีการใช้พลังงานไฟฟ้าสำหรับสำหรับระบบปรับอากาศในบ้านพักอาศัยในปริมาณที่สูงมาก จากข้อมูลการใช้พลังงานตั้งแต่ ปี ค.ศ. 2001-2005 พบว่ามีการใช้พลังงานในระบบปรับอากาศเพิ่มขึ้นอย่างต่อเนื่อง ค่าเฉลี่ย 4.8% ต่อปี ซึ่งระบบปรับอากาศโดยทั่วไปมีวัตถุประสงค์เพื่อสร้างสภาวะสบายและการเพิ่มคุณภาพของอากาศให้ดียิ่งขึ้น การสร้างสภาวะสบายสามารถนำระบบการแผ่รังสีความเย็นมาใช้ทดแทนระบบปรับอากาศได้ ซึ่งระบบนี้จะมีการใช้พลังงานค่อนข้างต่ำ และถือเป็นทางเลือกหนึ่งของการเพิ่มความสบายให้กับมนุษย์

การศึกษางานวิจัยนี้เพื่อออกแบบและทดสอบการใช้แผงเย็นและทดสอบการทำงานอุปกรณ์สำหรับระบบทำความเย็นแบบธรรมชาติภายในบ้านอยู่อาศัย การทดสอบจะใช้ห้องทดลองบริเวณชั้น 2 ของบ้านประหยัดพลังงานในมหาวิทยาลัยสงขลานครินทร์ จังหวัดสงขลา ซึ่งเป็นพื้นที่อยู่บริเวณภาคใต้ตอนล่างของประเทศไทยมีลักษณะอุณหภูมิของอากาศและ ความชื้นสูง ตลอดทั้งปี การออกแบบระบบการแผ่รังสีความเย็น จะควบคุมการทำงานที่สภาวะอัตราการไหลคงที่ด้วยการทำงานแบบอัตโนมัติ สามารถแบ่งได้ออกเป็น 2 ขั้นตอน คือการผลิตน้ำเย็นที่อุณหภูมิ 25 °C และการจ่ายน้ำเย็นเข้าสู่แผงเย็นเพื่อทำการแลกเปลี่ยนความร้อนโดยการแผ่รังสีความร้อนจากคน และพื้นผิวอุณหภูมิสูงเข้าสู่แผงเย็น ภายในห้องทดลองจะมีการหมุนเวียนของอากาศเพื่อช่วยให้มีการถ่ายเทความร้อนโดยการพาความร้อนเข้าสู่แผงเย็นด้วยความเร็ว 0.5 เมตร/วินาที ภาระความร้อนที่ออกแบบในห้องทดลองมีค่า 1,450 วัตต์ และค่าความสามารถในการรับภาระความร้อนของแผงเย็น 66.13 วัตต์/ตารางเมตร หอผึ่งเย็นขนาด 3.27 ตัน ทำหน้าที่ผลิตน้ำเย็นและปั๊มจ่ายน้ำเย็นให้กับแผงเย็นที่ติดตั้งบริเวณผนังและ ฝ้าเพดานของห้องทดลองด้วยอัตราการไหลคงที่ 0.289 กิโลกรัม/วินาที ความแตกต่างของอุณหภูมิน้ำเข้าและออกจากแผงเย็นประมาณ 1.2 °C มีการติดตั้งเครื่องมือตรวจวัดเพื่อเก็บข้อมูลประกอบด้วยอุณหภูมิอากาศภายในและภายนอก ความชื้นสัมพัทธ์ภายในและภายนอก อุณหภูมิผนังทึบ อุณหภูมิแผงเย็น และ อุณหภูมิเฉลี่ยรอบๆ ห้องทดลอง พร้อมทำการบันทึกเก็บข้อมูลทุกๆ 5 นาที

จากข้อมูลการทดลองของวันที่ 19/07/06, 5/08/06 และ 20/09/06 ตั้งแต่เวลา 20:00-10.00 น.พบว่าค่า PMV ของทั้ง 3 วัน มีค่าอยู่ในช่วงสภาวะสบาย คือ -0.5-0.5 ผลการทดลองที่ได้แสดงว่าการทำงานของระบบการแผ่รังสีความเย็นจะสามารถทำงานได้ดีในช่วงเวลากลางคืน และช่วงเช้าเวลาไม่เกิน 10:00 น. นอกจากนี้ยังพบว่าระบบการแผ่รังสีความเย็นมีความต้องการ

กำลังไฟฟ้าเฉลี่ย 541.06 วัตต์ หากเปรียบเทียบกับระบบปรับอากาศแบบทั่วไปพบว่าสามารถประหยัดพลังงานได้ 40% ของการใช้พลังงานทั้งหมด และจากการทดสอบโดยการจำลองการใช้แผงเย็นด้วยโปรแกรม EnergyPlus Version 2.2.2 พบว่าตัวแปรที่มีผลต่อความรู้สึกสบายประกอบด้วยค่าความเป็นฉนวนของเสื้อผ้า ประเภทของกิจกรรมและความเร็วลม เช่น หากกิจกรรมที่มีการใช้พลังงานสูงก็จะทำให้อยู่ในสภาวะไม่สบาย เป็นต้น

ABSTRACT

Thailand is located in hot and humid region, thermal comfort can be achieved by using air-conditioning system. Energy consumption due to air-conditioning system is increasing approximately 4.8% per annum for residential building. The use of air conditioner has two main purposes namely, thermal neutrality and the purification of fresh air. Thermal neutrality can be achieved by using radiant cooling system. This system consumes less energy consumption comparing with the use of conventional air conditioner and it can be used as the alternative option for human thermal comfort.

This study focused on design and investigation of radiant cooling system and relevant equipment for passive cooling in a building. The experiment was set up at the 2nd floor of the low energy house in Prince of Songkla University, Songkhla Province, Thailand. Songkhla Province is located in the southern part of Thailand which has hot and humid throughout the year. The designed radiant cooling system was operated with constant flow rate and varying temperature with two step function control. The radiant cooling panel absorbed radiant heat transfer from occupants and other surfaces from the experimental room. The convective heat transfer to radiant cooling panel by circulate air with speed 0.5 m/s. Heat gain in the experimental room was 1,450 W and radiant panel cooling is 66.13 W/m^2. Cool water was supplied from cooling tower with cooling capacity of 3.27 Ton to wall and ceiling radiant cooling panels with mass flow rate 0.289 kg/s. Temperature difference of inlet and outlet water flow rate was 1.2 °C. Temperature of interior and exterior air, relative humidity of interior and exterior air, surface temperatures of opaque wall, radiant cooling panel and mean radiant temperature were recorded every five minutes.

Experimental results showed that the Predict Mean Vote (PMV) values on 19 July 2006, 5 August 2006 and 20 September 2006 during 20:00 to 10:00 were in the ranges of -0.5 – 0.5 which are in the comfortable ranges for all selected days. Results revealed that the use of radiant cooling system is appropriate in the night time and early morning (until 10:00). The radiant cooling system consumes power consumption approximately 541.06 W. Energy savings can be obtained at 40% by the use of radiant cooling system instead of the use of conventional air conditioner. The sensitivity analysis for various parameters namely, clothing level, metabolic rate and air velocity, was also examined through simulation. Simulation results revealed that thermal comfort could not be achieved at the higher metabolic rate.

ACKNOWLEDGEMENTS

The author wishes to thank the staffs of Department of Mechanical Engineering, Faculty of Engineering, Prince of Songkla University for theirs assistances. Grateful acknowledgements are due to the scholarship donor, Prince of Songkla University, for providing the research scholarship Finally, no word can express the author's deepest gratitude and appreciation to her parents for their continuous encouragement and love throughout the period of research project.

CONTENTS

CONTENTS (Continued)

LIST OF ABBREVIATIONS AND SYMBOLS

A_f = Area of transparent wall, m^2
A_L = Air leakage area, cm^2
A_r = Area for roof, m^2
A_{rad} = Radiant panel cooling, m^2
A_w = Area for opaque wall, m^2
C_p = Specific heat of cool water, kJ/kg °C
C_s = Stack coefficient
C_w = Wind coefficient
D = Globe diameter, m
Eff_c = Performance of cooling tower, %
F = Coefficient of solar radiation
F_{cl} = The ratio of clothing area to body skin area
F_{sa} = lighting special allowance factor
F_{ul} = lighting use factor
h_c = Interior surface heat transfer coefficient from iteration, $W/m^2.K$
$h_{ex,r}$ = Exterior surface heat transfer coefficient for roof, $W/m^2.K$
h_i = Interior surface heat transfer coefficient, $W/m^2.K$
h_o = Exterior surface heat transfer coefficient, $W/m^2.K$
h_1 = air inlet enthalpy, kJ/kg
h_2 = air outlet enthalpy, kJ/kg
h_3 = water inlet enthalpy, kJ/kg
h_4 = water outlet enthalpy, kJ/kg
I_{cl} = Clothing insulation value, clo
I_{facade} = Solar radiation on each facade, W/m^2
I_{global} = Global solar radiation, W/m^2
K = Thermal conductivity for materials construction, W/m.K
L = Imbalance between (M-W) and the rate of heat dissipation, W/m^2
$\dot{m}$ = Cool water flow rate, kg/s
M = Metabolic rate, W/m^2
M-WR = Net Metabolic and Work Rate, W/m^2
M_{a1} = air flow rate at the inlet, kg/s
M_{a2} = air flow rate at the outlet, kg/s
M_3 = water flow rate at the inlet, kg/s
M_4 = water flow rate at the outlet, kg/s
P_v = Vapor pressure of moisture in the air, kPa
P_1 = Pump transfer cool water for step 1
P_2 = Pump transfer cool water for step 2
PMV = Predicted Mean Vote
PPD = Predicted percent dissatisfied
Q = Radiant panel cooling, W/m^2
Q_{al} = Heat flow required to increase moisture content of air leakage into building from W_o to W_i, W

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)

Q_{as} = Heat flow required to raise temperature of air leaking into building from t_o to t_i, W
Q_c = Radiant panel cooling by convection, W/m^2
Q_f = Heat gain due to solar radiation through transparent wall, W
Q_{gain} = Heat gain in the room, W
Q_L = Heat gain due to lighting, W
Q_r = Solar heat gain for roof, W
Q_{rad} = Radiant panel cooling by radiation, W/m^2
Q_w = Solar heat gain for opaque wall, W
R_{cl} = Clothing resistance, m^{2}K/W
RH_{ex} = Exterior relative humidity, %
RH_i = Interior relative humidity, %
SC = Shading Coefficient
T_{cl} = Clothing temperature, °C
T_{eq} = Equivalent temperature, °C
T_{ex} = Exterior temperature, °C
T_g = Globe temperature, °C
T_i = Interior temperature, °C
T_m = Mean monthly exterior temperature, °C
T_n = Neutral interior temperature, °C
$T_{n,p}$ = Average temperature of the non-radiant panel surface of the room, °C
T_p = Mean panel surface temperature, °C
T_r = Mean radiant temperature, °C
T_{wb} = Wet bulb temperature, °C
T_{wi} = Inlet temperature of cool water, °C
T_{wo} = Outlet temperature of cool water, °C
Δt = Average interior and exterior temperature, K
U_f = Overall heat transfer coefficient for transparent wall, W/m^2.K
U_r = Overall heat transfer coefficient for roof, W/m^2.K
U_w = Overall heat transfer coefficient for opaque wall, W/m^2.K
V = Wind speed, m/s
$\dot{V}$ = Air flow rate, L/s
V_a = Interior air velocity, m/s
V_1 = Solenoid valve#1
V_2 = Solenoid valve#2
V_3 = Solenoid valve#3
V_4 = Solenoid valve#4
W_l = Total light wattage, W
W_i = Humidity ratio of interior air, g/kg (dry air)
W_o = Humidity ratio of exterior air, g/kg (dry air)
W_1 = Humidity ratio at air inlet, $kg_{moisture}/kg_{dry\ air}$
W_2 = Humidity ratio at air outlet, $kg_{moisture}/kg_{dry\ air}$

LIST OF ABBREVIATIONS AND SYMBOLS (Continued)

WR = Work rate, W/m^2
Δx = Surface thickness, m
ε = emissivity
α = Solar Absorptivity
τ = Transmisstivity of solar radiation through transparent wall

Final Report

1. Project Name

Development and Demonstration of Use of Radiant Cooling Panel for Passive Cooling in Buildings

2. Project Investigator

Assistant Professor Dr.Juntakan Taweekun
Department of Mechanical Engineering,
Faculty of Engineering,
Prince of Songkla University
Tel. 074-287035-6 Fax.074-212893

3. Budget 196,200 Baht

4. Project Start October 2005

5. Objectives of the Project

The main objectives of this study are design and investigation of the use of radiant cooling panel in building under the southern climate of Thailand. The specific objectives are in details as followings:

- Investigate the use of radiant cooling panel by apply the cool water from cooling tower directly to radiant cooling panel
- Examine thermal comfort under ASHRAE condition by using Predicted Mean Vote and Predicted Percent Dissatisfied
- Simulate the use of radiant cooling panel system using EnergyPlus program
- Compare the use of radiant cooling system with conventional air-conditioning system
- Demonstrate and disseminate the results of the use of the radiant cooling system

6. Scope of the Study

The weather data of Songkhla Province is used for this study. Material used is applicable in Thailand.

7. Progress

7.1 Background and Rationale

Urbanization is developing very rapidly along with the change in the industries in Thailand. The percentage of people working in buildings is increasing with urbanization and decreasing share of agriculture in the economy. With increasing use of electric appliances and equipment, per-capital energy use initially increases especially in the households. But advance in technologies for energy-efficient building design and construction, and programs for promotion of energy efficiency in developed countries and many emerging economies are responsible for a gradual decline in overall energy use per floor area.

The trend of population and social development also contributes to the change in the pattern of energy use in dwellings. The average family comprised six persons in 1980 in Thailand. By 2000, the family size has reduced to four, and may gradually reduce further, as the present family size of the USA of 3.2 may offer some indication. There is a steady increase in the number of household per capital. On the other hand, air conditioning steadily penetrates room by room in urban households. The ranges of popular household appliances also increase. Table 1 provides summary information on electricity use in a remote rural household and in an urban household of Thailand. Electricity consumption in the urban household is more than 10 times of that in the rural household.

Table 1 Electricity use in households, Thailand

Appliance	Size (kW)	Hours/day	kWh/month	Percentage (%)
Remote Rural Household				
Lamps	0.031	5.5	26.1	37.8
Others	N/A	N/A	42.9	62.2
Total	-	-	69.0	100
Urban Household				
Air conditioner	2.25	12	607.5	70.9
- One in bedroom				
- One in common room				
Refrigerator	0.3	24	162	18.9
Lamps	0.4	4-8	72	8.4
Television	0.007	3	6	0.7
Wasting machine	0.36	0.5	5.4	0.6
Rice cooker	0.6	0.25	4.5	0.5
Total			**857.4**	**100**

Source: Surapong (2005)

Residential households in urban and suburban areas use air-conditioning for thermal comfort increasingly. When air-conditioning is used, it contributes 70% of electricity consumption in a household. This phenomenon of penetration of air-conditioning occurs rising urbanization and decreasing size of family that leads to increasing number of households in urban areas.

It is clear that for most economics in Asia, urbanization steadily rises. The number of appliances especially air-conditioning in households increase but these increases will not reach saturation in the near future. Electricity consumption in residential buildings in Thailand increases approximately 4.8% as shown in Fig.1.

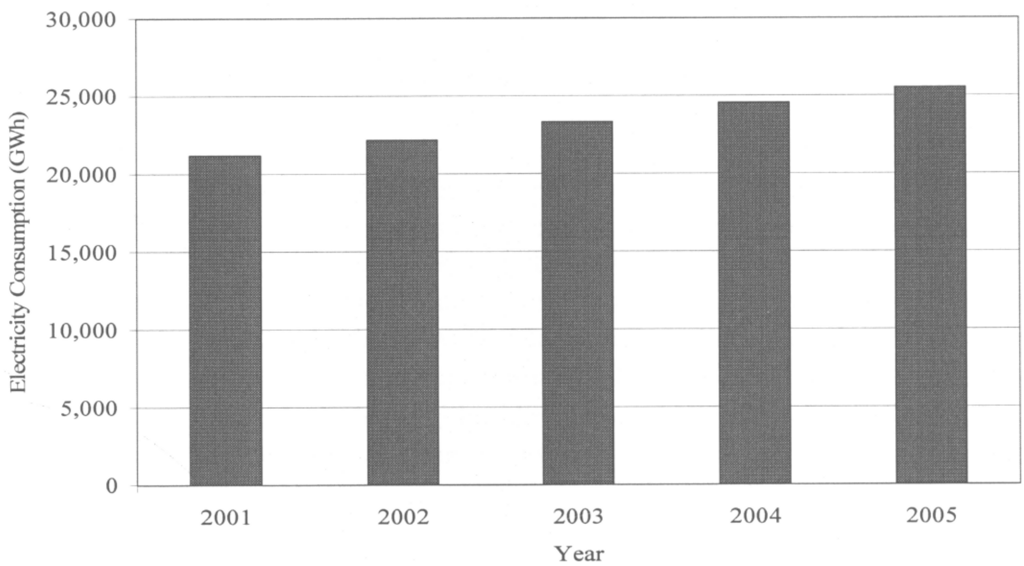


Figure 1 Electricity consumption in residential buildings, Thailand (Source: EPPO, 2006)

In commercial buildings, air-conditioning has reached saturation in dynamic Asian economies which are situated in tropical climate. Although there is a gradual increase in the range and intensity of equipment use in commercial buildings, energy use per unit floor area may already reach its peak as equipment are increasingly more energy-efficient. The potential for energy conservation in buildings in Asia is well-recognized. Many countries create individual country programs for energy efficiency in buildings, and carry them out through activities of state-supported energy conservation centers. These centers are actually technical units entrusted to provide mainly advisory services related to energy conservation. Energy conservation for buildings is also included in demand-side-management programs under implementation by electric utilities in many Asian countries. Multilateral programs have also been promoted. Among ASEAN countries (including Brunei, Cambodia, Indonesia, Laos, Malaysia, Myanmar, the Philippines, Singapore, Thailand, and Vietnam.) cooperation programs on energy conservation for commercial buildings were earlier set up by the earlier member states with dialog countries such as USA and Australia.

In an ASEAN-US program (Levine, 1992), it was estimated that potential electricity savings due to adoption of energy performance standards per commercial building in ASEAN cities ranged from 18 to 36%. Recent study in Thailand produces similar results (Surapong, 1994; Surapong 1996). The ASEAN-US program is accredited for the impetus, it gives to promotion of energy conservation for commercial buildings in ASEAN. Through the program, energy performance requirements for commercial buildings were established in each country (except Brunei). In the Philippines and Thailand, both mandatory and voluntary programs at national scale are now under implementation. In Thailand, programs for conservation for commercial buildings is incorporated into an energy conservation plan which is under implementation as a part of the stipulation of the Energy Conservation Promotion Act (ECP Act) promulgated in 1992. The ECP Act also creates an energy conservation promotion fund to facilitate implementation of the Act. A Demand-Side-Management (DSM) plan has also been implemented by the Thai electric utilities since 1992. The DSM plan complements the activities created by the ECP Act.

The use of air conditioner has two main purposes namely, thermal neutrality and the purification of fresh air. Thermal neutrality can be achieved by using radiant cooling system. This system consumes less energy consumption comparing with the use of conventional air conditioner and it can be used as the alternative option for human thermal comfort. Air conditioners work nearly full capacity for the tropical zone. Average temperature in Thailand is approximately 25-35 °C and there has mostly high humidity along the year. In residential buildings, the use of energy in air-conditioning system, dehumidification and ventilation systems consume up to 70% of the total consumption.

Thermal comfort of local parts of the body and the whole body, including the effects of humidity and small air movements, by subjective experiments under a radiant cooling system was investigated and found that small air movement with the radiant cooling system had a possibility of improving the comfortable sensation votes (Koichi et al., 1999). This cooling technique is suitable for office buildings with low thermal loads and for which no additional air treatment system is provided (Mumma, 2001). The capacity of radiant cooling panel was grossly inadequate during the hot period, even for night time application only.

Prapapong and Surapong (2004) studied the radiant cooling using natural air for ventilation under Thai climate. To avoid condensation of moisture on the cooling panel, the temperature of water supplied to the panel was limited to 24 °C. Figure 2 shows schematic diagram of radiant cooling in the experimental room. Experiments were conducted over the cool period of December, the hot and dry period of March and the humid period of May. The room was served by radiant panels with a total area of 7.5 m^2. Its capacity was grossly inadequate during the hot period, even for night time application only. The TRNSYS program was used to simulate the use of cooling panels and conventional air-conditioning in the experimental room. The results from experiments and simulation showed that thermal comfort could be obtained with application of radiant cooling. The results also implied that radiant consumes less energy than application of conventional air-conditioning system.

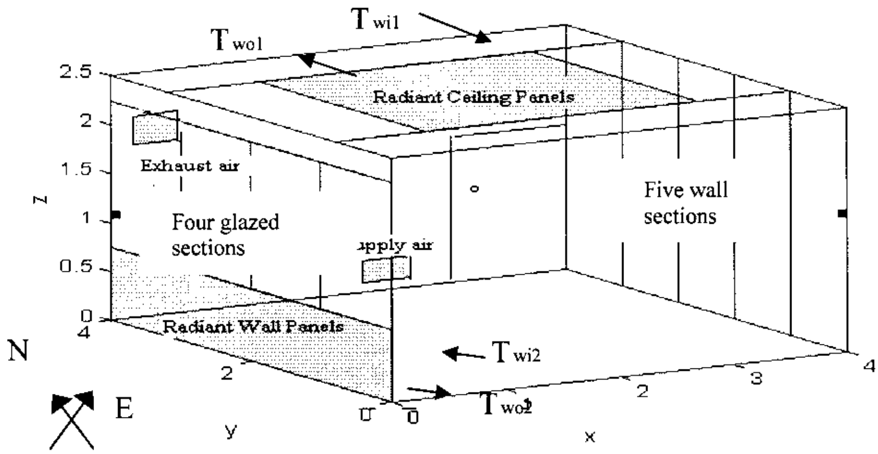


Figure 2 Radiant cooling system in the experimental room
(Source: Prapapong and Surapong, 2004)

From the previous studies, most of researchers concentrated on the use of chilled water supplied from chiller plant directly to radiant cooling panel. Condensation can be occurred. This study concentrates on the use of radiant cooling panel by apply the cool water directly from cooling tower to radiant cooling panel under weather of the southern part of Thailand. EnergyPlus program Version 1.2.2 is used to simulate the use of radiant cooling panel system in the experimental room. Simulation results are compared with results from the experiment.

7.2 Literature Reviews

In conventional air-conditioning system, cooled air flows by forced circulation over a person to convect heat and removes moisture from the body and the surrounding surfaces directly. Heat gain in the room is removed to outside by using air-conditioning system. Figure 3 shows conventional air- conditioning system. The solar radiation falls on the external facade. At the same instant, a part of solar radiation is transmitted through glazing into the building. Solar radiation falling on an opaque surface is partly reflected and the remaining is absorbed on the exterior surface of the opaque wall. Ambient temperature also contributes to heat gain into the building.

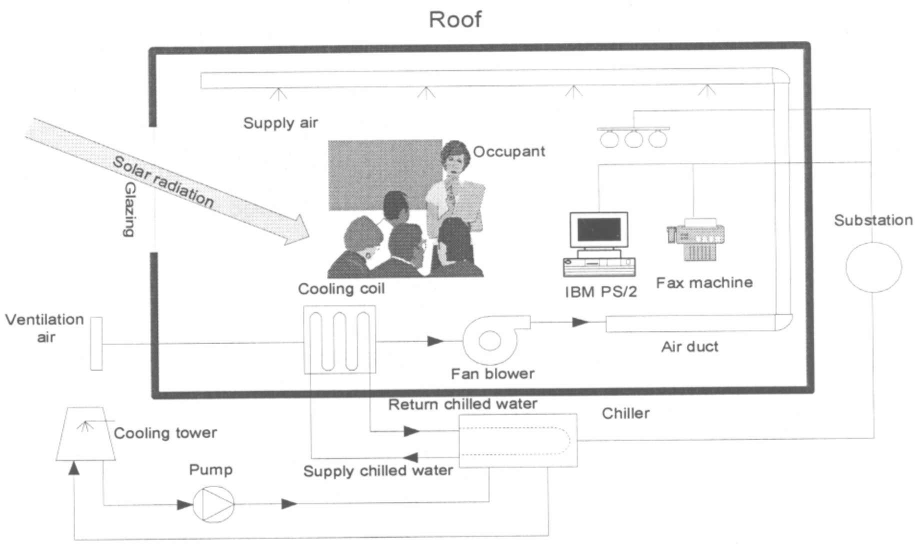


Figure 3 Conventional air-conditioning system

While heat is being conducted through the wall material from the exterior surface to the interior surface, the interior surface of the external wall exchanges thermal radiation with other surfaces in the room. Air in the room also convects heat away from the interior surface. Some equipment and occupants in the room contribute both sensible and latent heat. Sensible heat is transferred as thermal radiation to other surfaces in the space and a part of this sensible heat is also convected to air. Latent heat is directly transferred to air.

7.2.1 **Internal and External Heat Gain**

Heat gains in the building compose of external heat gain from solar radiation through opaque wall, transparent wall and roof; and internal heat gain i.e. occupant, equipment and lighting. The total heat gains in the building also depend on weather data, building type and location.

7.2.1.1 Internal Heat Gain

- **Internal Heat Gain Due to Lighting**

Since lighting is often the major space load component, accurate estimate of the space heat gain is needed. Calculation of this load component is not straightforward. The rate of heat gain at any given moment can be quiet different from the heat equivalent of power supplied to those lights. Only part of the energy from lights is in the form of convective heat. The remaining portion is in the form of radiation, which affects the conditioned space only after having been absorbed and re-released by walls, floors, furniture, etc. Heat gain due to lighting, Q_L, can be calculated using the following equation (ASHRAE, 1997).

$$Q_L = W_l F_{ul} F_{sa} \quad \text{(Eq. 1)}$$

where W_l = Total light wattage, W
F_{ul} = Lighting use factor
F_{sa} = Lighting special allowance factor

- **Internal Heat Gain Due to Occupant**

Table 2 shows rates at which heat and moisture are given off by human beings in different states of activity. These sensible and latent heat gains constitute a large fraction of the total load. Even for short-term occupancy, the extra heat and moisture brought in by people may be significant.

Table 2 Sensible and latent heat gain from occupant

Degree of Activity	Place	Heat Gain (W)		
		Sensible	Latent	Total
Seated at theater	Theater	65	30	95
Moderately active	Offices, hotels	75	55	130
Standing, walking	Department store	75	55	130
Walking, standing	Drug store, bank	75	70	145
Sedentary work	Restaurant	80	80	160
Light bench work	Factory	80	140	220
Moderate dancing	Dance hall	90	160	250
Bowling	Bowling alley	170	255	425
Heavy work	Factory	170	255	425
Athletics	Gymnasium	210	315	525

Source: ASHRAE Fundamentals Handbook (1997)

The conversion of sensible heat gain from people to space cooling load is affected by the thermal storage characteristics of that space and is thus subject to application of appropriate room transfer functions. Latent heat gains are considered instantaneous.

- **Internal Heat Gain Due to Equipment**

The nominal power rating is usually close to the power drawn in actual use. But for office equipment, that would be quite misleading: the actual power has been measured to be much lower (Norford et al.,1989). Some typical values are indicated in Table 3. In recent year, the computer revolution has brought a rapid increase in electronic office equipment, and the impact on loads has become quite important, comparable to lighting. For special equipment such as laboratories or kitchens, it is advisable to estimate the heat gains by taking a closed look at the inventory of the equipment to be installed, paying attention to the possibility that much of the heat may be drawn directly to the outside by exhaust fans.

Table 3 Typical heat gain for several kinds of equipment

Equipment	Heat Gain (W)	Remarks
Television set	50-100	
Refrigerator	100-200	Recent model more efficient
Personal computer(desktop)	50-200	Almost independent of use while turned on
Impact printer	10-30	Increases about twofold during printing
Laser printer	150	Increase about twofold during printing
Copier	150-300	Increase about twofold during printing

Source : Norford et al (1989)

7.2.1.2 External Heat Gain

- **External Heat Gain Due to Solar Radiation Through Opaque Walls and Roof**

A part of solar radiation on an opaque surface is reflected. The absorbed radiation raises the temperature on the surface of the wall section. Heat is transferred to the interior across the wall by conduction. The intensity of solar radiation on roof is higher than on any vertical wall so that the roof temperature is high. In most case, there is a plenum between the roof and the ceiling. Because the roof is at higher temperature during the day, warm stagnant air is trapped in the plenum. A large part of heat transmission from the roof to the ceiling is due to thermal radiation. In this respect, thermally reflective sheet placed between the roof and ceiling acts as good barrier against heat gain when it is used in addition to bulk insulation. In term of the sol-air temperature, the total solar heat gain for opaque wall, Q_w, and roof, Q_r, are calculated from the following equation.

$$Q_w = \frac{U_w A_w I_{facade} \alpha}{h_o} \qquad \text{(Eq. 2)}$$

$$Q_r = \frac{U_r A_r I_{global} \alpha}{h_{ex,r}} \qquad \text{(Eq. 3)}$$

$$U_w = \frac{1}{\left[\left(\frac{1}{h_o}\right) + \Sigma\left(\frac{\Delta x}{K}\right) + \left(\frac{1}{h_i}\right)\right]} \qquad \text{(Eq. 4)}$$

$$U_r = \frac{1}{\left[\left(\frac{1}{h_{ex,r}}\right) + \Sigma\left(\frac{\Delta x}{K}\right) + \left(\frac{1}{h_i}\right)\right]} \qquad \text{(Eq. 5)}$$

where U_w, U_r = Overall heat transfer coefficient for opaque wall and roof respectively, $W/m^2.K$

A_w, A_r = Area for opaque wall and roof respectively, m^2

I_{facade} = Solar radiation on each facade, W/m^2

I_{global} = Global solar radiation, W/m^2
α = Solar Absorptivity
h_o = Exterior surface heat transfer coefficient, W/m^2.K
$h_{ex,r}$ = Exterior surface heat transfer coefficient for roof, W/m^2.K
Δx = Surface thickness, m
K = Thermal conductivity for materials construction, W/m.K
h_i = Interior surface heat transfer coefficient, W/m^2.K

- **External Heat Gain Due to Solar Radiation Through Transparent Wall**

A part of solar radiation falling on transparent is reflected. Another part is absorbed. The absorbed radiation raises internal energy of the transparent that increases temperature of the transparent. This increases additional heat transfer from the transparent into the interior both by convection and thermal radiation. This is called the inward-flowing faction of absorbed solar radiation. A glazing is usually very thin comparing with other components of building envelope. Its mass is small and its thermal inertia is neglected. A large part of solar radiation is transmitted through the glazing. In quantitative terms, heat gain from solar radiation is typically five times the size of heat gain across opaque elements in the same position.

The transmitted radiation is partly absorbed, partly reflected on the surface. However, it is assumed that all the energy of transmitted solar radiation is absorbed in the space. Heat gain due to solar radiation through transparent wall, Q_f, can be calculated using the following equations.

$$Q_f = A_f (SC) F I_{facade} \qquad \text{(Eq. 6)}$$

$$F = \tau + \alpha \frac{U_f}{h_o} \qquad \text{(Eq. 7)}$$

$$U_f = \frac{h_i h_o}{h_i + h_o} \qquad \text{(Eq. 8)}$$

where A_f = Area of transparent wall, m^2
SC = Shading coefficient
F = Coefficient of solar radiation
τ = Transmissitivity of solar radiation through transparent wall
U_f = Overall heat transfer coefficient for transparent wall, W/m^2.K

7.2.2 Air Leakage

Large air leakage is common for residential houses in hot climate. Even when windows and doors are closed, air can leak through seams between a window frame or door frame and the wall and between the window and the door with the corresponding frames. Table 4 provides information on the size of leakage areas at joints and seams. The information was adapted from ASHRAE Handbook of Fundamentals (ASHRAE, 2001).

Table 4 Leakage areas at reference pressure difference of 4 Pa

Point	Details	Leakage Area		
		Typical	Maximum	Minimum
1	Wooden window or door frame and wall on the lower and upper storey, cm^2/m	0.3	0.5	0.3
2	Wooden window or door frame on the lower and upper storey, cm^2/m	1.0	1.2	0.4
3	Ceiling, general, cm^2/m^2	1.8	2.8	0.79
4	Light fixture, cm^2 each	10.0	21.0	1.5

Source: Surapong (2005)

Air leakage can be calculated by LBNL model (Sherman and Grimsrud, 1980). The approach requires the effective air leakage area at 4 Pa, which can be obtained from a whole–building pressurization test. Using the effective air leakage area, the air flow rate due to air leakage is calculated according to Eq. 9.

$$\dot{V} = A_L \sqrt{C_s \Delta t + C_w V^2} \qquad \text{(Eq. 9)}$$

where $\dot{V}$ = Air flow rate, L/s
A_L = Air leakage area, cm^2
C_s = Stack coefficient
Δt = Average interior and exterior temperature, K
C_w = Wind coefficient
V = Wind speed, m/s

The values of stack coefficient for one story, two story and three story houses are 0.000145, 0.000290 and 0.000435, respectively (ASHRAE, 1997). The value of the wind coefficient depends on the local shielding class of the building and the building height. Shielding classes and wind coefficient are shown in Table 5.

Table 5 Local shielding classes

Class	Description	Wind Coefficient, C_w		
		1 storey	2 storey	3 storey
1	No obstructions or local shielding	0.000319	0.000420	0.000494
2	Light local shielding; few obstructions, few trees, or small shed	0.000246	0.000325	0.000382
3	Moderate local shielding; some obstructions within two house heights, thick hedge, solid fence, or one neighboring house	0.000174	0.000231	0.000271
4	Heavy shielding; obstructions around most of perimeter, buildings or trees within 10 m in most directions; typical suburban shielding	0.000104	0.000137	0.00016
5	Very heavy shielding; large obstruction surrounding perimeter within two house heights; typical downtown shielding	0.000032	0.000042	0.000049

Source: ASHRAE Fundamentals Handbook (1997)

Heat flow required to increase moisture content of air leakage into building from W_o to W_i, Q_{al}, and heat flow required to raise temperature of air leaking into building from t_o to t_i, Q_{as}, can be calculated using the following equations.

$$Q_{as} = 1.2\dot{V}\Delta t \qquad \text{(Eq. 10)}$$

$$Q_{al} = 3\dot{V}(W_i - W_o) \qquad \text{(Eq. 11)}$$

where W_i = Humidity ratio of interior air, g/kg (dry air)
W_o = Humidity ratio of exterior air, g/kg (dry air)

7.2.3 Radiant Cooling System

Feustel and Stetiu (1994) studied hydronic system and found that the system is suited to the dry climates. The evaluation of the theoretical performance of hydronic systems could most conveniently be made by simulation. The development of a model that can accurately simulate the dynamic performance of hydronic/radiant cooling systems is announced. The developed model can be used to calculate cooling loads, heat extraction rates, room air temperature and room surface temperature distributions, and used to evaluate issues i.e. thermal comfort, control system, system sizing, system configuration and dynamic response. These authors also found that hydronic system can reduce the amount of air transported through the building by separating the tasks of ventilation and thermal conditioning (Feustel and Stetiu, 1995). Due to the physical properties of water, hydronic system can transport a given amount of thermal energy and consumed less energy approximately 5% of the otherwise necessary fan energy. This improvement alone significantly reduces the energy consumption and peak demand requirement of the air-conditioning system. The report described the development, thermal comfort issues, and cooling performance of the hydronic systems. The peak demand requirement of hydronic system was also compared to conventional all air systems as seen in Fig.4.

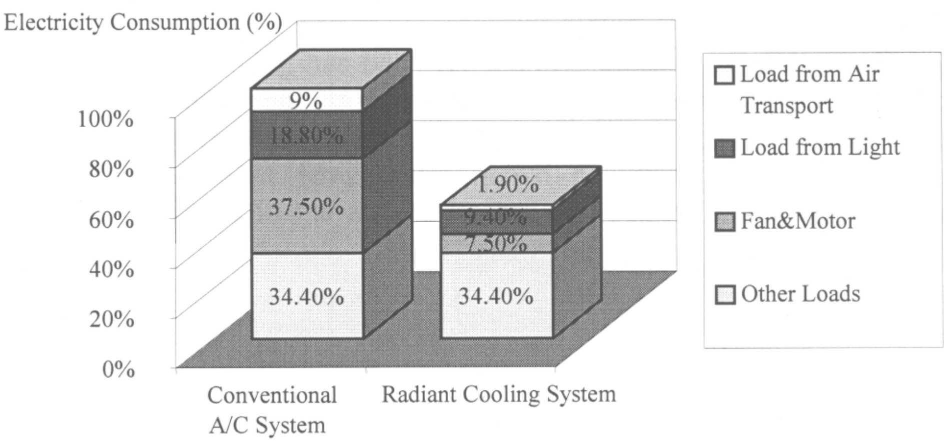


Figure 4 Comparison of electrical consumption for conventional air-conditioning system and radiant cooling system
(Source: Feustel and Stetiu, 1995)

When radiant cooling is used with displacement ventilation, where ventilation air is introduced at low level and flows by natural means to replace ventilation air, such system has been suggested to offer quiet comfort and energy efficiency superior to those of conventional air-conditioning system (Feustel and Stetiu, 1995; Imarori et al., 1999). In Europe, it was reported that cooling tower could be used to cool the water for supply to the radiant panel on the ceiling (Oliveira, 2000). This is considered a passive cooling option and has even higher potential for energy and peak power saving.

Radiant cooling is an alternative option since it has energy and peak power saving potential (Stetiu, 1999). The climatic conditions are less favourable in France than in the European continental regions. Imanari et al (1999) studied the radiant system by avoiding condensation risks under keeping the ceiling surface temperature at a minimum of 17 °C. Besides the fundamental qualities of ceiling water panels cooling systems as far as acoustical and thermal comfort were concerned, this technique allow a 10% reduction of the energy consumption.

Roulet et al (1999) used large panels to control the indoor temperature by cooling as well as by heating in several types of buildings. The panels were made from two corrugated stainless steel foils, seam welded on the perimeter and spot welded at many places on the area. Water at controlled temperature circulated in this cushion. These panels were either installed at ceilings or on walls. In well-insulated buildings, the power required to control indoor temperature was rather low, and a small temperature difference between the panels and the indoor environment was sufficient to deliver or absorb the required heat.

Conroy and Mumma (2001) applied radiative and convective heat transfer equations to the rate of heat removal. The fundamental heat transfer equations govern the radiant cooling panel mean temperature as a function of geometry, materials, flow rates, coolant temperature, and space temperature. The radial panels had no dehumidification duty, and no condensation on the surfaces. The dedicated outdoor air systems must be designed to control 100% of the space latent loads.

Ceiling panel radiant cooling has been refined and used successfully in Europe for more than 20 years (Mumma, 2001). The system comprises panels installed on the ceiling of a room, or in some cases hung from a high ceiling. Cooling water was supplied to the panels at temperature above dew-point temperature to avoid condensation of moisture in the air on the panels. Figure 5 illustrates the radiant cooling system.

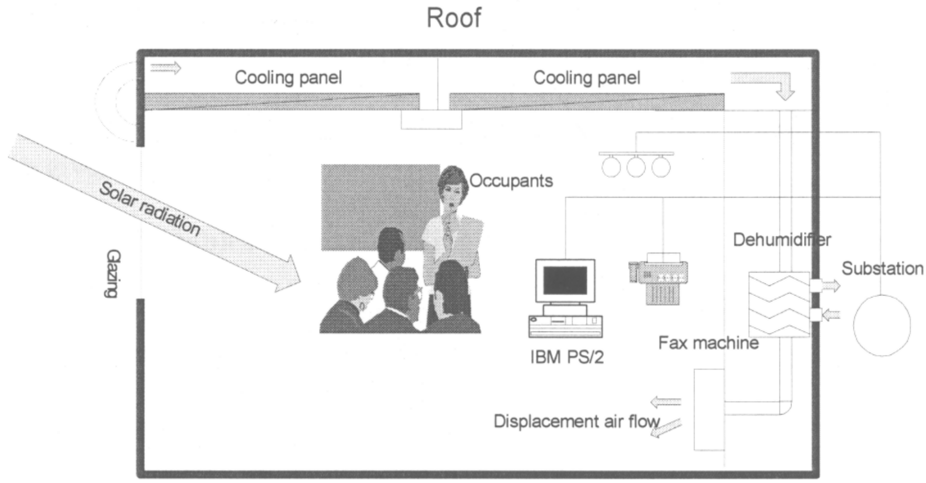


Figure 5 Radiant cooling system

Miriel et al (2002) studied a water ceiling panel system and a monitory data acquisition system was installed in a test room. The ceiling panel was made of copper pipes with rigid aluminium diffusion fins. The material used presented good heating conductivity and the ceiling radiant panel was fast-acting. This cooling technique is suitable for office buildings with low thermal loads and for which no additional air treatment system is provided.

Nakano (2002) developed a radiant ceiling panel system integrated with ice storage system which makes ice using electricity at night and uses it for air-conditioning in the daytime. The system needed no supply air duct in plenum above suspended ceiling and 20% electricity cost was reduced comparing with conventional air-conditioning system. It was also capable of lowering peak demand in the daytime and enables electric utilities to efficiently operated power generation facilities. As the system air-conditions a room with radiant ceiling panels, it was much more comfortable than conventional air-conditioning systems. It was increasingly important not only for electric utilities but also for society as a whole to spread thermal storage air-conditioning systems, which control room temperature using stored cold or hot heat, and promote load leveling between daytime and night.

7.2.4 Thermal Comfort

In order to maintain normal functions, the human body needs to maintain the balance between heat gain and heat loss. Heat can be lost in different ways: radiation to surrounding surfaces, convection to the ambient air, conduction, evaporation, respiration and excretion. The most important loss is due to radiation, followed in order by convection and conduction. Respiration and excretion have less influence on the heat loss of a human body (Feustel and Stetiu, 1995).

The body internal temperature is automatically regulated to 36.8 °C in healthy body. The average skin temperature is taken to be 33.7 °C. The skin temperature at different part of the body varies i.e. head 36.8 °C, chest 34 °C and feet 27 °C. The rate of the heat generation in the body varies with the level of vigor of the activity. The rate of dissipation of generated heat is dependent on the physical environment i.e. temperature and humidity. Four physical variables and two variables related to a person are given as followings.

7.2.4.1 Physical Variables of Surrounding

Four physical variables are air temperature, relative humidity of air, air speed and radiant temperatures of surrounding surfaces.

- **Air Temperature**

Air temperature is perhaps the most important variable. When air temperature is high, convective and conductive heat transfer from the body to the air are not effective. The inverse situation is when air temperature is low, in which convection and conduction heat transfer may become excessive.

- **Relative humidity of air**

Relative humidity of air is also relevant and affects the rate of evaporation of moisture from the skin. Perspiration is an effective means human body relies on for rapid heat removal. But under high humidity, evaporation of perspiration is slowed and a sense of stuffiness occurs. Koichi et al (1999) investigated the thermal comfort of local parts of the body and the whole body, in particular, including the effects of humidity at 45%, 65% and 85% and small air movements, by subjective experiments under a radiant cooling system. Subjects were seated on a chair under the radiant cooling panels, and voted their thermal sensation and comfortable sensation. Small air movement with the radiant cooling system had a possibility of improving the comfortable sensation votes in the radiant cooling.

- **Air Speed**

Air speed affects the rate of convection heat transfer from the body. At low temperature, high air speed causes excessive heat removal and often creates a sensation of differential temperatures between the upper and lower parts of the body. At high temperature, higher air speed will increase rate of heat removal to convect heat and removes moisture from the body and the surrounding surfaces directly.

Alamdari et al (1998) studied the displacement airflow patterns when the thermal loads are matched to the cooling capacity of the displacement ventilation system. However, there would be a risk of occupant thermal discomfort at low level for thermal loads higher than 25 W/m^2. Additional cooling can be provided by chilled ceiling panels to offset higher thermal gains. However, the addition of these devices affected the air distribution characteristics of displacement ventilation systems. Chilled ceiling panels changed the air temperature near the ceiling, which created downward convection. Furthermore, radiation heat transfer between the chilled panels and walls reduced the room surface temperature below the room air temperature, which caused downwards convection near the wall. The predictions indicated very high thermal comfort conditions within the studied space.

- **Mean Radiant Temperature**

Radiant temperatures of surrounding surfaces affect the rate of radiation heat transfer from the body to the surfaces. In the case where there is a substantial between the radiant temperatures of different surfaces or between that of the air and that of surrounding surfaces, a sense of discomfort will ensue. A globe thermometer is used to measure a temperature which approximates the average temperature of surrounding surfaces. The mean radiant temperatures, T_r, are calculated from the following equation.

$$T_r = \left[\left((T_g + 273)^4\right) + \frac{1.10x10^8 V_a^{0.6}}{\varepsilon D^{0.4}}\left(T_g - T_i\right)\right]^{1/4} - 273 \qquad \text{(Eq. 12)}$$

where T_g = Globe temperature, °C
V_a = Interior air velocity, m/s
ε = Emissivity (0.95 for black globe)
D = Globe diameter, m
T_i = Interior temperature, °C

Yamtraipat et al (2005) presented the result of a thermal comfort survey using 1,520 Thai volunteers from different climatic regions of Thailand. The survey was conducted using different types of air-conditioned buildings from the private and public sectors. Apart from common thermal comfort factors such as air dry-bulb temperature, relative humidity and air velocity, two non-quantifiable factors namely the acclimatization to the use of air conditioner at home and the education level were considered. A general database for thermal comfort studies in Thailand was created, and different thermal comfort standards were developed for the three climatic regions of Thailand. Twenty six degree Celsius and 50-60% relative humidity could be used as a comfortable environment condition for the whole country. The data was then used to generalize an earlier concept for setting thermal comfort standard using data from non-conditioned building. Figure 6 illustrates Thailand ventilation comfort chart.

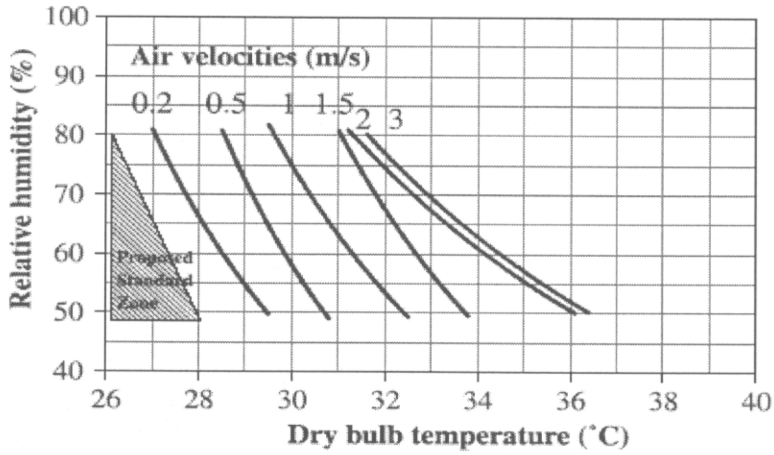


Figure 6 Thailand ventilation comfort chart
(Source: Khedari et al., 2001)

7.2.4.2 Personal Variables

The metabolic rate can be measured accurately by measuring the rates of oxygen in take and generation of carbon dioxide. Generally the work performed by body function is very small. The unit of "met" is devised for thermal comfort studies. One met equals 58.2 W/m^2. Table 6 shows the rates of metabolic rates corresponding to different activities.

Table 6 Metabolic rates of different activities

Activity	met	W/m^2
Sleeping	0.7	40.0
Reclining	0.8	46.6
Seated and quiet	1.0	58.2
Sedentary activity (office, dwelling, lab, school)	1.2	69.8
Standing, relaxed	1.2	69.8
Light activity, standing (shopping, lab, light industry)	1.6	93.1
Medium activity, standing (domestic work, machine work)	2.0	114.4
High activity (heavy machine work, garage work)	3.0	174.6

Source: Surapong (2005)

Clothing level affects the rate of convection, conduction, radiation and evaporation of moisture from the body. A unit of measure of clothing level is "clo". This unit is related to resistance to heat transfer through clothing. Table 7 lists values of clothing insulation. One unit of "clo" is equivalent to a thermal resistance of 0.155 m^2 K/W or a coefficient of thermal transfer of 6.45 W/m^2 K.

Table 7 Insulation values of some garments and ensembles

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, Light slacks and short sleeve shirt	0.5

Source: Surapong (2005)

7.2.4.3 Assessment of Thermal Comfort

The assessment of thermal comfort with human subjects has been conducted by Houghten and Yagloglou (1923). Typically a subject will be situated in a controlled environment and asked to respond to questions on thermal sensation. Thermal sensation has been adopted and some statistical pattern has now been recognized.

- **ASHRAE Thermal Sensation Scale**

Historically, different scales of thermal sensation have been used. But consensus has been reached on the used of a seven-point scale. Each ASHRAE comfort scale has now included both a word ascribed to denote the degree of sensation and a numerical value as shown in Table 8.

Table 8 ASHRAE comfort scale

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

Source: Surapong (2005)

- **Predicted Mean Vote and Predicted Percent Dissatisfied**

From a number of studies, Fanger (1970) related the likelihood of the vote on the ASHRAE comfort scale by a person to the imbalance of a given environmental condition from the neutral condition. Fanger used the term Predicted Mean Vote, PMV, to describe the vote and imbalance condition in the following relationship.

$$PMV = [0.303 \exp(-0.036M)+0.028]L \qquad \text{(Eq. 13)}$$

where M = Metabolic rate, W/m^2
L = Imbalance between (M-WR) and the rate of heat dissipation, W/m^2
(M-WR)= Net metabolic and work rate, W/m^2
WR = 5.5-15(Met-0.8), W/m^2

Ranges of PMV for thermal comfort conditions are shown in Table 9.

Table 9 Ranges of PMV for thermal comfort conditions

Item	Details	Value
1	Comfortable	-0.5 – 0.5
2	Warm	0.5 – 1.0
3	Cool	-1.0 to -0.5
4	Unacceptably warm	Over 1.0
5	Unacceptably cool	Under -1.0

Fanger (1970) also statistically derived a relationship between the value of predicted mean vote and the percentage number of dissatisfied, PPD, at the given environmental condition as shown in Eq.2.14.

$$PPD = 100\text{-}95 \exp[-(0.03353PMV^4 + 0.2179\, PMV^2)] \qquad \text{(Eq. 14)}$$

For an environment condition closed to neutral condition, there is no heat storage either in the core or in the skin compartment. The metabolic rate and work must match heat dissipation as a prerequisite for thermal neutrality to be reached. Fanger (1970) combined heat transfer equations and obtained the following equation for evaluation of a steady environment near neutral condition.

$$\begin{aligned} L = {} & (M\text{-}WR) - 3.96 \times 10^{-8} f_{cl}[(T_{cl}+273.15)^4 - (T_r+2.73.15)^4] \\ & - f_{cl}\, h_c\, (T_{cl}-T_i) - 3.05\, [5.73 - 0.007(M\text{-}WR) - P_v] \\ & - 0.42\, [(M\text{-}WR)-58.15] - 0.0173\, M\, (5.87-P_v) \\ & - 0.0014\, M\, (34-T_i) \end{aligned} \qquad \text{(Eq. 15)}$$

where f_{cl} = The ratio of clothing area to body skin area
= $1.0 + 0.2\, I_{cl}$ for $I_{cl} < 0.5$
= $1.05 + 0.1\, I_{cl}$ for $I_{cl} > 0.5$
I_{cl} = Clothing insulation value, clo
T_{cl} = Clothing temperature, °C
P_v = Vapor pressure of moisture in the air, kPa
T_i = Air temperature, °C

T_{cl} is obtainable from the following equation

$$T_{cl} = 35.7 - 0.0275\,(M-W) - R_{cl}\,\{3.96 \times 10^{-9}\, f_{cl}[(T_{cl} + 273.15)^4 - (T_r + 273.15)^4] + f_{cl} h_c (T_{cl} - T_i)\} \qquad \text{(Eq. 16)}$$

where R_{cl} = Clothing resistance, m^2K/W
= $0.155\ I_{cl}$,
h_c = Interior surface heat transfer coefficient from iteration, $W/m^2.K$
= $2.38\,(T_{cl} - T_a)^{0.25}$ for $2.38\,(T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$, $W/m^2\,K$
= $12.1\sqrt{V_a}$ for $2.38\,(T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$, $W/m^2\,K$

The above relationships have also been incorporated into ISO 7730-1994, development of Predicted Mean Vote and Predicted Percent Dissatisfied Indices and specification of conditions for thermal comfort.

- **Adaptive Thermal Comfort**

Auliciems (1981) used and enlarged data base to obtain a similar relationship between neutral interior temperature, T_n, and mean monthly exterior temperature, T_m, in the form:

$$T_n = 17.6 + 0.31T_m \qquad \text{(Eq. 17)}$$

The results were obtained from studies where respondents wore light clothing in sedentary activities in unconditioned indoor spaces. Szokolay (1980) proposed to use wind to offset higher temperature. He advocated that the equivalent reduction from the prevailing, DT, temperature can be calculated from:

$$DT = 6V_a - V_a^2 \qquad \text{(Eq. 18)}$$

Where V_a is the air speed and the above equation can be used for air speed between 0 and 3 m/s. This is in contrast in stipulation in ASHRAE Standard 55-1992 that air speed should not exceed 0.8 m/s.

Dear and Brager (1998) undertook ASHRAE RP-884 project to examine adaptive hypothesis and its implication for ASHRAE Standard 55-92. The project assembled results of worldwide field surveys comprising more than 21,000 respondents in over 160 buildings. The ASHRAE static predicted mean vote model was shown to be partially adaptive and could be used to explain adaptation occurring in air-conditioned buildings. But occupants in naturally ventilated buildings were found to be tolerant to a wider range of temperatures. The authors claimed that the results formed a basis for a new variable indoor standard for thermal comfort.

Table 10 shows the equivalent temperature, T_{eq}, sensed by a person under an outdoor air temperature and a given wind speed. It can be seen from this Table that for a house in suburban when natural air flow is assisted by electric fan to increase the speed of flow at acceptable level, adaptive thermal comfort or natural comfort could be achieved for outdoor air temperature 33 °C (with a wind speed of 1m/s, DT is 5 °c).

Table 10 Equivalent temperature at wind speed 1 m/s

T_{ex} (°C)	T_n (°C)	T_{eq} (°C)
28	26.3	23
29	26.6	24
30	26.9	25
31	27.2	26
32	27.5	27
33	27.8	28
34	28.1	29

A significant study was made in Bangkok, Thailand during cool season (Jitkhajorwanich et al., 1998). The study focused on respondents in transitional spaces, between indoor and outdoor and spaces with natural ventilation and with air-conditioning. The neutral temperature for all respondents was obtained as 27.1 °C. For naturally ventilated spaces, the neutral temperature obtained was 27.6 °C. For other groups, the results were not reliable statistically.

Another study in Bangkok was under taken more recently (Busch, 1992). The study reported a study of two groups of subjects in their working environments. One group comprises those working in unconditioned buildings. The people in the other group worked in fully air conditioned buildings. It was reported that the neutral temperature for those working in air conditioned buildings did not differ substantially from the neutral zone for summer of ASHRAE, while the majority of people working in unconditioned buildings vote for a neutral temperature of slightly over 28 °C.

7.2.5 EnergyPlus Simulation

EnergyPlus is an energy analysis and thermal load simulation program. Based on a user's description of a building from the perspective of the building's physical make-up, associated mechanical systems, etc. EnergyPlus is used to calculate the heating and cooling loads necessary to maintain thermal control set points, conditions throughout an secondary Heating Ventilation and Air Conditioning (HVAC) system and coil loads, and the energy consumption of primary plant equipment as well as many other simulation details that are necessary to verify that the simulation is performing as the actual building would.

EnergyPlus has its roots in both the Building Loads Analysis and System Thermodynamics (BLAST) and DOE–2 programs. BLAST and DOE–2 were both developed and released in the late 1970s and early 1980s as energy and load simulation tools. Their intended audience is a design engineer or architect that wishes to size appropriate HVAC equipment, develop retrofit studies for life cycling cost analyses, optimize energy performance, etc. The configuration of EnergyPlus program is given in Fig.7.

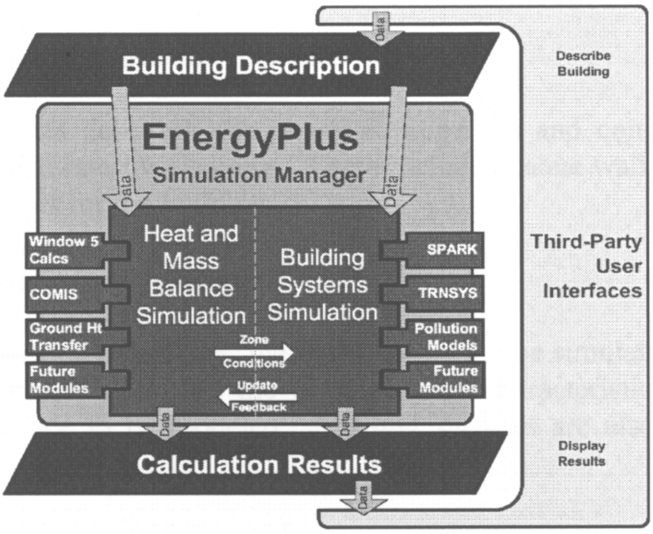


Figure 7 Configuration of EnergyPlus Program

EnergyPlus program composes of various groups for model development i.e. simulation parameter, schedules, surface construction elements. Details of each group for simulation are given as followings.

- Group-Simulation Parameters

This group of objects describes Version, Building, Time-step in Hour, Zone Volume Capacitance Multiplier.

- Group-Location, Climate, Weather File Access

This group of objects is relevant to the ambient conditions for the simulation.

- Group-Schedules

This group of objects allows the user to influence scheduling of many items (such as occupancy density, lighting, thermostatic controls, occupancy activity). In addition, schedules are used to control shading element density on the building. EnergyPlus schedules consist of three parts: a day description, a week description, and an annual description. An optional element is the schedule type. The day description is simply a name and the values that span the 24 hours in a day to be associated with that name. The week description also has an identifier (name) and twelve additional names corresponding to previously defined day descriptions. There are names for each individual day of the week plus holiday, summer design day, winter design day and two more custom day designations. One type of schedule reads the values from an external file to facilitate the incorporation of monitored data or factors that change throughout the year. Schedules are processed by the EnergyPlus Schedule Manager, stored within the Schedule Manager and are accessed through module routines to get the basic values (time step, hourly,etc). Values are resolved at the Zone Time Step frequency and carry through any HVAC time steps.

- Group-Surface Construction Elements

This group of objects describes the physical properties and configuration for the building envelope and interior elements. These include opaque wall, transparent wall, roofs, floors and doors for the building.

- Group-Thermal Zone Description/Geometry

Without thermal zones and surfaces, the building can not be simulated. This group of objects (Zone, Surface) describes the thermal zone characteristics as well as the details of each surface to be modeled. Shading surfaces are also included in this group.

- Group-Internal Heat Gains

Not all the influence for energy consumption in the building is due to envelope and ambient conditions. This group of objects describes other internal heat gains i.e. occupant, lighting, equipment.

7.3 Research Methodology

This report focuses on investigation the use of radiant cooling system in building. Weather data of Songkhla Province is used for system design. The use of the radiant cooling system from the experiment is compared with simulation results using EnergyPlus program Version 1.2.2. The followings describe the methodology in this study.

7.3.1 Design of Radiant Cooling System

- **Weather Data**

Weather data of Songkhla Province, year 2000, are used in this study for design of radiant cooling system.

- **Design of Radiant Cooling Panel and Water Supply System**

The radiant cooling system is designed using cool water supplied from cooling tower passing through the radiant cooling panel which made from copper tube bond with aluminium sheet. There are two panels installed at the wall and the ceiling of the experimental room. In preliminary, the weather data of Songkhla Province, external and internal heat gain in the experimental room are used for design stage of radiant cooling panel. Cooling tower is also investigated to use with the developed radiant cooling panel.

- **Design of Control System**

Firstly, cooling tower was assumed to produce cool water and stored in the storage tank with temperature 25 °C. Secondly, pump was operated using solenoid valve received signal from temperature sensor in storage tank and the cool water lift to the panel by pump to the room and circulated between cooling tower and panel.

7.3.2 Experiment

- **Set Up the Experiment**

The experiment room is set up at the low energy house in Prince of Songkla University, Hatyai campus, Songkhla Province. Temperature sensors, humidity sensors, thermocouple, globe thermometer, energy consumption sensor are installed to measure temperatures of interior air and exterior air, relative humidity of interior air and exterior air, surface temperatures of opaque walls and radiant cooling panel, mean radiant temperature of interior air and electricity consumption, respectively. Arrangements of the various sensors in the experimental building are illustrated in Fig.4.9 (Chapter 4).

- **Data Collection**

The measurement of interior and exterior air temperatures, relative humidity of interior and exterior air, surface temperature, mean radiant temperature, electricity consumption are recorded every five minutes using data logger. The air flow rate in the room is measured instantaneous by anemometer. Table 11 shows the recorded data and the equipment used in this study.

Table 11 Recorded parameter and equipment

Item	Parameter	Equipment Use	Remarks
1	Interior and Exterior Temperature	Temperature Transmitter	Continuous record every 5 minutes
2	Interior and Exterior Relative Humidity	Humidity Transmitter	Continuous record every 5 minutes
3	Interior air speed	Anemometer	Instantaneous record
4	Electricity consumption	Continuous Power Meter	Continuous record every 5 minutes
5	Surface Temperatures	Thermocouple Type K	Continuous record every 5 minutes
6	Mean Radiant Temperature	Globe Thermometer	Continuous record every 5 minutes
7	Water Flow Rate	Tank and Stopwatch	Instantaneous record

- **Data Analysis**

Measured data obtained in five minute interval are analyzed and observed. This study focuses on thermal comfort evaluation. The value of PMV is considered in this study.

Economic analysis for the use of radiant cooling panel and conventional air conditioner was also considered.

7.3.3 EnergyPlus Simulation

- **Construction of Library Building Model**

As mentioned earlier in Chapter 2, EnergyPlus requires detailed physical description of building shape, orientation, composition of opaque wall, transparent wall, floor and roof. Version 1.2.2 of the program is used in this study. The developed model requires details of lighting device, equipment. Details of architectural plan, occupancy schedule, equipment and lighting usage schedule are also required in this model.

- **Prediction of PMV**

The developed model is used with the weather data of Songkhla Province to predict PMV.

- **Validation of Model**

The value of PMV obtained from EnergyPlus model is validated using experimental results. The diagram of research methodology is given in Fig.8.

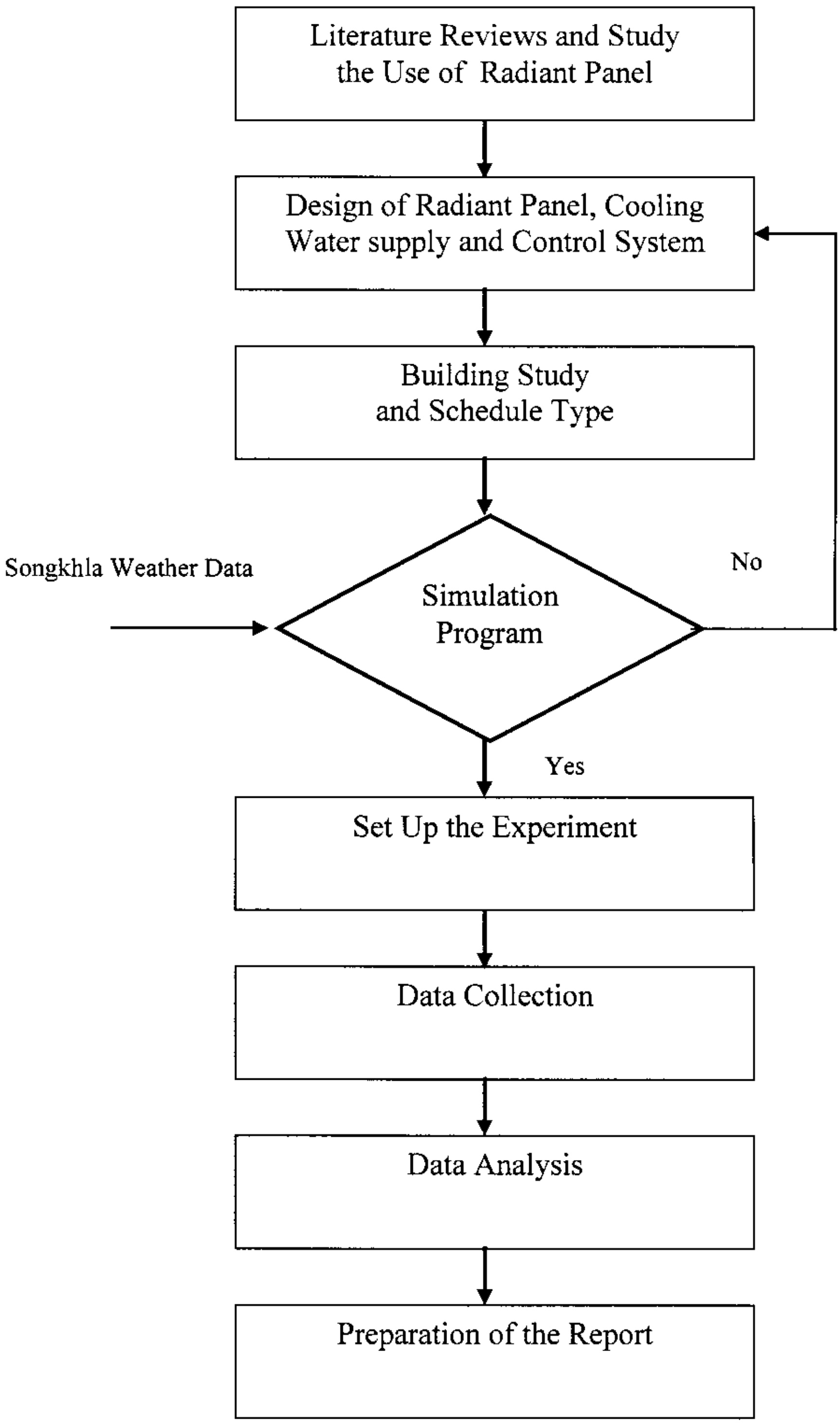


Figure 8 Research methodology

7.3.4 Material and Equipment

- **Temperature and Humidity Transmitter**

Temperature and humidity transmitters-Model TRH-303 were used in this study for measurement of temperature and relative humidity of interior and exterior air as shown in Fig 9. This model is wall mount type. The measurement ranges for relative humidity and temperature are 0-100 %RH with accuracy ± 2 %RH and 0-100 °C with accuracy ± 0.3 °C, respectively. The sensors for humidity and temperature are thin-film capacitor and RTD Pt100 Ω, respectively. The output signal of these transmitters are 4-20 mA two wire.

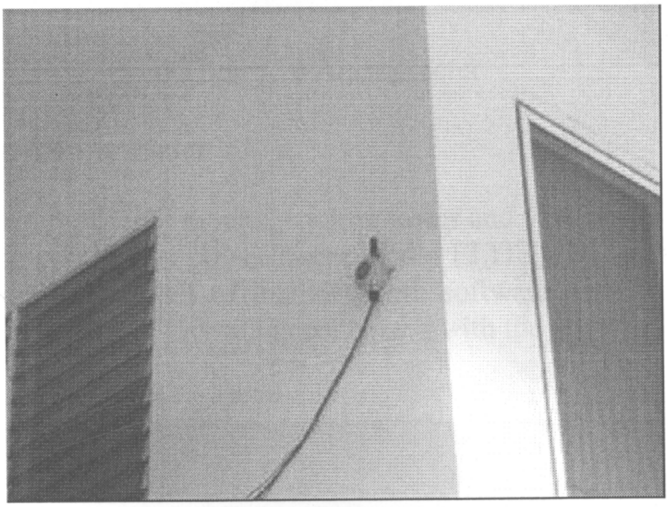


Figure 9 Temperature and humidity transmitter

- **Anemometer**

The portable anemometer is used for measurement air flow of the ventilation air in the experimental room. The measurement range is 0.4 to 30 m/s with accuracy ± 2 %. The sensor is conventional twisted vane arms and low-friction ball-bearing design as given in Fig.10.

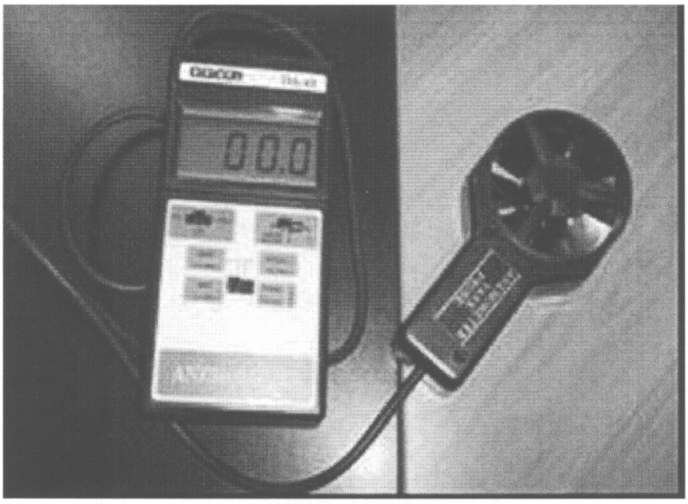


Figure 10 Anemometer

- **Continuous Power Meter**

The electricity consumption of pumps, cooling tower and ventilation fan are measured and recorded using continuous power meter model ELITE PRO as shown in Fig. 11. The power recorder has 128kB of memory with software and RS-232 cable. The small split core model SCT-0750-005 is connected with the power recorder.

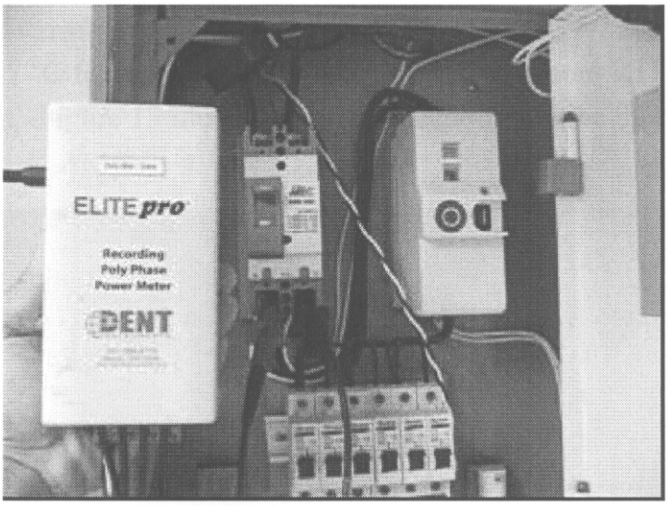


Figure 11 Continuous power meter

- **Thermocouple**

The temperatures of each surface in the experimental room are measured using thermocouple type K installed 14 points. Two thermocouples are placed on the surface of the wall radiant cooling panel and six are placed on the ceiling radiant cooling panel. The remaining thermocouples are placed at the surface of opaque wall as shown in Fig.12.

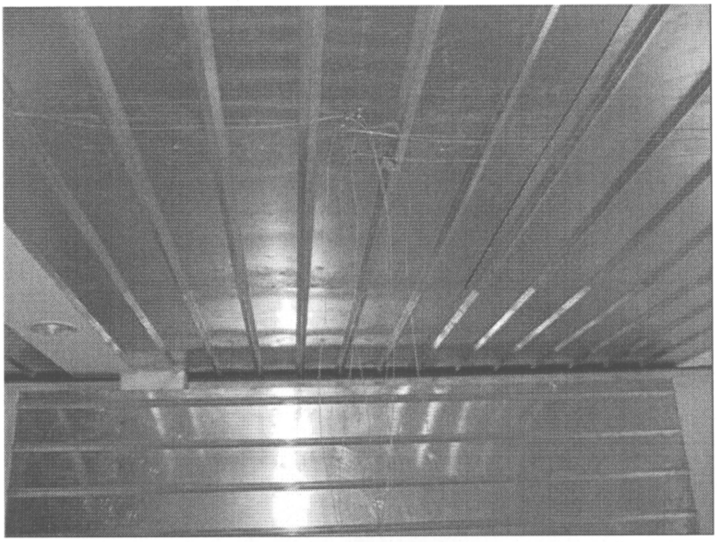


Figure 12 Thermocouple type K

- **Globe Thermometer**

Globe Thermometer model QUESTEMP ° 15 is used for measurement of globe temperature to determine the mean radiant temperature. The accuracy of the equipment is ± 0.5 °C for temperature range 0-100 °C. Sensors RTD Pt100 Ω is used for this equipment. Figure 13 illustrates the installation of globe thermometer.

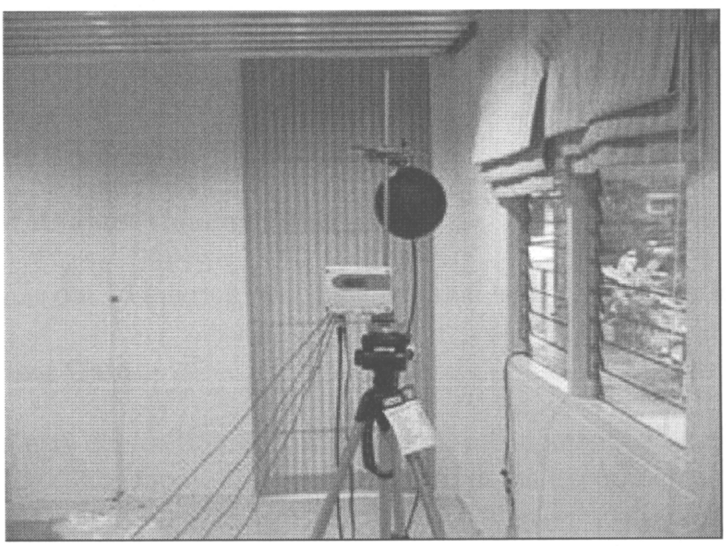


Figure 13 Globe thermometer

- **Data Logger**

The data logger model "GRAPHTEC GL450" is used for data recording. The data from the sensors are transmitted to data logger and recorded every five minutes. A software is used to process the data that allowed graphs of data values to be viewed continuous. The number of analog input is 40 channels with PC interface USB (Version 2.0). Memory capacity is 2 M words internal memory and 1 G PC card slot. The measurement accuracy for Thermocouple type K in the range $-100 < Ts \leq 1370$ °C is ± 0.05%.

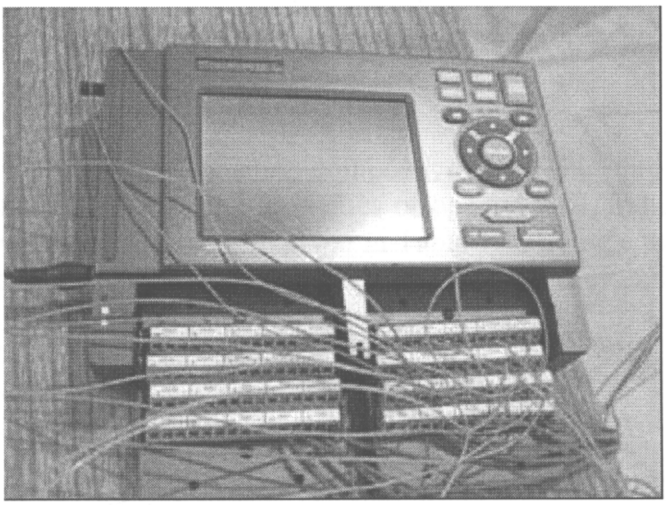


Figure 14 Data logger

7.4 Design and Experimental Set Up

Firstly, the radiant cooling system and control system are designed. Secondly, the design values in the first stage are used to construct the radiant cooling system i.e. radiant cooling panel, cooling tower. Finally, the simulation tool is used for comparison between the experimental results and simulation results. The followings are the details in each stage.

7.4.1 Design of Radiant Cooling Panel

Heat gain, Q_{gain}, is due to internal heat gain, external heat gain and air leakage.

- **Internal Heat Gain**

Internal heat gain is due to lighting, occupant and equipment. Equation 1, Tables 1 and 2 are used for calculation of internal heat gain in the experimental room as shown in Table 12.

Table 12 Summarization of Internal Heat Gain

Equipment	Q'ty	Heat Gain (W)	Total Heat Gain (W)
1.Lighting	4 lamp	18	72
2.Occupants			
- Sensible Heat Gain	2 persons	75	150
- Latent Heat Gain	2 persons	55	110
3.Equipment			
- Television	1 set	100	100
- Computer	1 set	150	150
Total Internal Heat Gain			**582**

- **External Heat Gain**

External heat gain due to solar radiation composes of heat gain through opaque wall, transparent wall and roof. Material properties of opaque wall, transparent wall, roof and door in the experimental room are given in Table 13. Using equations 4, 5 and 8, the overall heat transfer coefficient for opaque wall, door, roof and transparent wall are 1.168, 2.355, 0.444, 5.867 $W/m^2.K$, respectively. Calculations are given in details in A.1 (Appendix A).

Table 13 Material properties of opaque wall, transparent wall and roof

Item	Wall/Roof	Opaque Wall		Transparent Wall		
		Density (kg/m^3)	Conductivity (W/m.K)	τ	α	SC
1	Opaque Wall					
	- 10 mm-Cement Motar	1,182	0.305			
	- 75 mm-Light Weight Concrete	600	0.121			
	- 10 mm-Cement Motar	1,182	0.305			
2	Transparent Wall					
	- 6 mm SMG-III	-	-	0.32	0.63	0.59
3	Roof					
	- 12 mm-Cpac	2,400	0.978			
	- 2 mm-Insulation	80	0.03			
	- 150 cm-Air Gap	1.177				
	- 75 mm-Insulation	130	0.038			
4	Door					
	- 30 mm-Wood	495	0.118			

Equations 2, 3 and 6 are used to calculate solar heat gain. The values of solar heat gain through opaque wall, roof and transparent wall are 73.59 W, 43.53 W and 233.21 W, respectively. Detailed calculations are given in A.2 (Appendix A).

- **Air Leakage**

Large air leakage is common for residential houses in hot climate. Even when windows are closed, winds and other effects can cause a large variation of air leakage. Tables 14 and 15 show the information of air leakage area on a door and a window in the experimental room.

Table 14 Information on a door

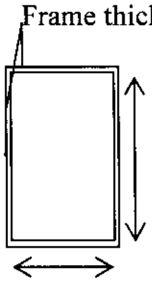
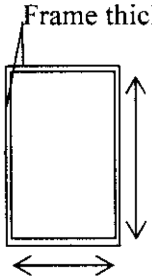
Dimension	Details	Unit	Values
Frame thickness 4 cm 	Total frame length = 2(2.1+0.9)	m	6
	Leakage between frame and wall	cm^2/m	0.3
	Leakage area between frame and wall	cm^2	1.80
Height 2.02 m	Leakage length of door = 2(2.02+0.82)	m	5.68
	leakage between door and frame	cm^2/m	1.0
	Leakage area between doors and frame	cm^2	5.68
Width 0.82 m	Total air leakage area per door	cm^2	7.48

Table 15 Information on a window

Dimension	Details	Unit	Values
Frame thickness 4 cm 	Total frame length = 2(1.2+0.8)	m	4
	Leakage between frame and wall	cm^2/m	0.3
	Leakage area between frame and wall	cm^2	1.20
Height 1.12 m	Leakage length of window = 2(1.12+0.72)	m	3.68
	leakage between door and frame	cm^2/m	1.0
	Leakage area between doors and frame	cm^2	3.68
Width 0.72 m	Leakage length of louver	m	0.72
	leakage in each louver	cm^2/m	1
	Leakage area of louvers (11 louvers)	cm^2	7.92
	Total air leakage area per window	cm^2	12.8

The total air leakage area in each facade and the quantities of doors, windows and open channel are given in Table 16. There has no air leakage in east and south facades due to well sealing.

Table 16 Leakage areas

Direction	Point	Q'ty	A_L (cm^2)
North			
- Door frame and wall	①	1	1.80
- Door frame	②	1	5.68
Total leakage areas for North direction			**7.48**
West			
- Door frame and wall	③	1	1.80
- Door frame	④	1	5.68
- Open channel	⑤	1	900
- Window frame and wall	⑥	2	2.40
- Window frame	⑦	2	7.36
- Louver	⑧	22	15.84
Total leakage areas for West direction			**933.08**

Referred to Table 16, the sample points for air leakage are illustrated in Fig.15.

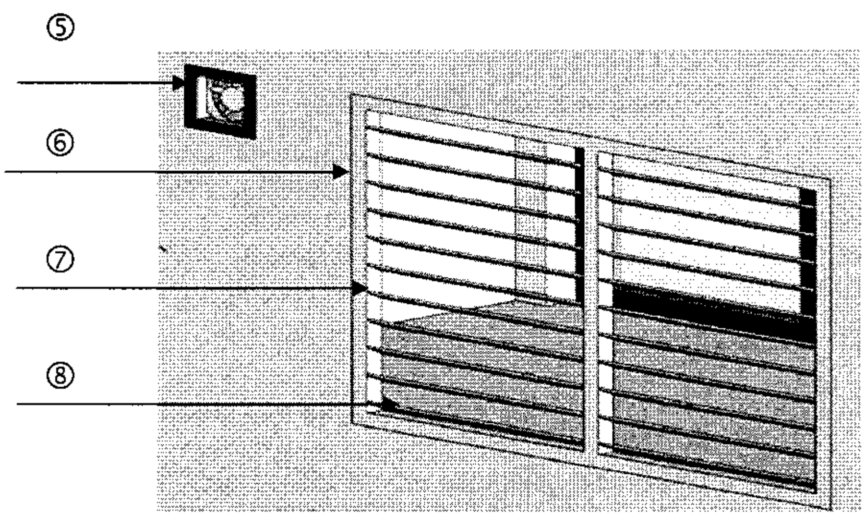


Figure 15 Leakage areas in the experimental room

The data from Table 16, equations 10 and 11 are used to calculate energy required to warm exterior air entering by infiltration to the temperature of the room, energy associated with net loss of moisture from the space and total loss due to air leakage which is equal to 80.07 W as shown in Table 17. Sample calculation is shown in A.3 (Appendix A).

Table 17 Loss due to air leakage

Direction	Q_{as} (W)	Q_{ai} (W)	Total Loss (W)
North			
- Door frame and wall	0.12	0.03	0.15
- Door frame	0.39	0.10	0.49
Loss due to air leakage, North facade			**0.64**
West			
- Door frame and wall	0.12	0.03	0.15
- Door frame	0.39	0.10	0.49
- Window frame and wall	0.16	0.04	0.20
- Window frame	0.50	0.13	0.63
- Louver	1.07	0.27	1.34
- Open channel	61.05	15.57	76.62
Loss due to air leakage, West facade			**79.43**
Total loss due to air leakage			**80.07**

Heat gain in the experimental room is obtained from the calculated values of internal heat gain, external heat gain and air leakage as shown in Eq.19. Table 18 summarizes the total heat gain:

$$Q_{gain} = \text{internal heat gain} + \text{external heat gain} + \text{air leakage} \qquad \text{(Eq. 19)}$$

Table 18 Total heat gain

Item	Details	Unit	Values
1	Internal Heat Gain	W	582.00
2	External Heat Gain	W	350.33
3	Loss due to air leakage	W	80.07
	Total heat gain		1,012.40

- **Radiant Panel Cooling**

Radiant panel cooling occurs by the combined heat transfer mechanisms of radiation and convection. The details are as followings.

Radiant Heat Transfer: Radiant heat transfer is governed by the Stefan-Boltzmann equation. For most building enclosure cases encountered in practice, the enclosure emittance are approximately 0.9 and the view factor between the ceiling and the balance of the enclosure is at least 0.87 (ASHRAE 1996). Placing these common values into the Stefan-Boltzmann equation results is shown as:

$$Q_{rad} = 0.15 \times 10^{-8} \left[\left(T_{n,p}^4 \right) - \left(T_p^4 \right) \right] \qquad \text{(Eq. 20)}$$

where Q_{rad} = Radiant panel cooling by radiation, Btu/ft^2.hr
$T_{n,p}$ = Average temperature of the non-radiant panel surface of the room, °R
T_p = mean panel surface temperature, °R

Convective Heat Transfer: The rate of heat transfer by convection is a combination of natural and forced convection. Natural convection results from the cooled air in the boundary layer just below the panels being displaced by warmer air in the room. This natural process can be altered or even changed to forced convection by infiltration, human activity and the mechanical ventilation systems. Min (1956) suggested that for practical panel cooling applications without forced convection, the cooling convective heat transfer is given by the following equation (ASHRAE 1996).

$$Q_c = 0.31(T_{n,p} - T_p)^{0.31}(T_{n,p} - T_p) \qquad \text{(Eq. 21)}$$

Where Q_c is radiant panel cooling by convection (in unit of Btu/ft^2.hr). Thus, radiant panel cooling, Q, can be calculated by using the following formulation:

$$Q = Q_{rad} + Q_c \qquad \text{(Eq. 22)}$$

In this design, the mean panel surface temperature and average temperature of the non-radiant panel surface of the room are assumed to be 25 °C (536.4 °R) and 32 °C (549 °R), respectively. The calculated values of radiant and convective heat transfer of radiant cooling panel are 38.11 W/m^2 and 27.02 W/m^2, respectively. Total radiant panel cooling is 65.13 W/m^2. The detailed calculation is given in A.4 (Appendix A).

- **Area of Radiant Cooling Panel**

The performance of radiant cooling panel depends on radiation and convection heat transfer. In this system, water is used for absorb heat gain in the experimental room. Area of radiant cooling panel, A_{rad}, is calculated using.

$$A_{rad} = \frac{Q_{gain}}{Q} \qquad \text{(Eq. 23)}$$

Cool water flow rate, $\dot{m}$, can be calculated using the following equation.

$$Q_{gain} = \dot{m} C_p (T_{wo} - T_{wi}) \qquad \text{(Eq. 24)}$$

where C_p = Specific heat of cool water, kJ/kg °C (equal to 4,179 kJ/kg °C)
T_{wo} = Outlet temperature of cool water, °C
T_{wi} = Inlet temperature of cool water, °C

Water is cooled by using cooling tower and this cooled water is supplied to radiant panel installed at the opaque wall and the ceiling. In the previous calculation, the cool water supplied at temperature 25°C will absorb heat gain 65.13 W/m^2. The design heat gain for this experiment is 1,450 W_{th} which calculated from total heat gain 1,012.40 W_{th} with allowance factor 40%. The sizing of radiant cooling panels installed at the ceiling and the opaque wall are 16.83 m^2 and 5.44 m^2, respectively.

Table 19 shows the area of radiant cooling panel, cool water flow rate and heat gain. Detailed calculation is given in A.5 (Appendix A).

Table 19 Design Parameter of Radiant Cooling Panel

Parameter	Unit	Radiant Panel		
		Ceiling	Wall	Total
Area	m^2	16.83	5.44	22.26
Cool water flow rate	kg/s	0.219	0.070	0.289
Heat gain	W	1,098.24	351.76	1,450

Thailand is located in the tropical zone which high relative humidity and temperature throughout the year. Songkhla is located in the southern part of Thailand. Table 20 shows average, minimum and maximum temperatures of Songkhla Province in year 2000 for each period. Each reference day are given in details in Appendix B.

Table 20 Weather climate of Songkhla Province, year 2000

Period	Reference Day	Dry Bulb Temperature (°C)		
		Average	Min	Max
16 February – 31 May	12 March	29.97	23.58	38.33
1 June – 15 August	8 July	29.25	26.6	33.01
16 August – 31 October	23 September	27.78	24.29	36.71
1 November – 15 February	5 December	27.12	21.45	33.02

Source: Meteorological Department (2000)

It was observed that there is the highest average dry-bulb temperature in the period of 16 February–31 May. Thus, this period will be used in preliminary design of radiant cooling system.

7.4.2 Design of Cooling Tower

The performance of cooling tower depends on relative humidity and wet bulb temperature of exterior air. If relative humidity of exterior air is low, the performance of cooling tower is higher. Properties of cooling tower are presented in Table 21.

Table 21 Cooling tower Properties

Description	Unit	Values
Cooling Capacity	Ton	3.27
Water flow rate	kg/s	0.55
Volumetric air flow rate	m^3/s	0.4167
Differential temperature	°C	7

Source: Liang Chi Industry Co., Ltd.

7.4.3 Radiant Cooling Operation and Control System

When the cool water is supplied to radiant cooling panel, pump is operated. Pump P_1, P_2 and fan of cooling tower are controlled by controller (on-off function). Temperature sensors installed at storage tank and discharge pipe of cooling tower transfer signal for on/off control valve. The closing and opening of control valve status are given in Table 22. Process operation and control of cool water producing and cool water used are illustrated in Fig.16.

Table 22 Function of pump and control valve

Operation	Status of pump		Status of control valve			
	P_1	P_2	V_1	V_2	V_3	V_4
Step 1	Open	Close	Open	Open	Open	Open
Step 2	Close	Open	Open	Open	Close	Close

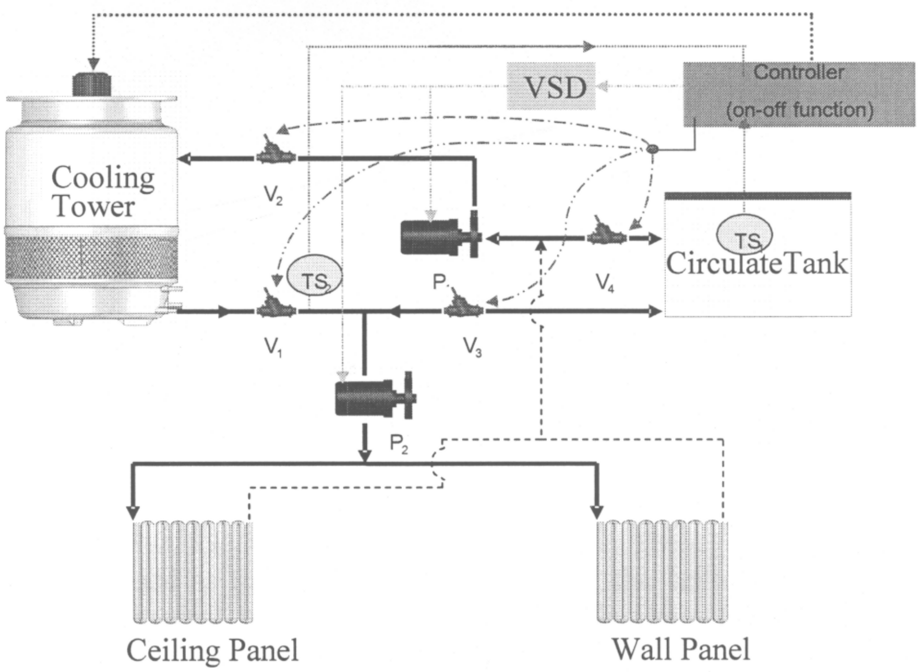


Figure 16 Radiant cooling operation and control system

The steps of operation and control are in details as following.

Step 1 Cool water producing: Pump is circulated high temperature water from storage tank and heat exchange with air from cooling tower until temperature below the set point. The set point of this process is 25 °C and then the system will be switched to step 2.

Step 2 Cool water usage: Cool water is supplied to four panel (one wall radiant cooling panel and three ceiling radiant cooling panel) with inlet temperature 25 °C and continuously circulate to cooling tower with closed system. The system is operated until outlet water temperature from cooling tower reaches 25.5 °C . Then, the system will be shut down.

7.4.4 Experiment

The experiment was set up at the 2nd floor of the low energy house in Prince of Songkla University, Hatyai Campus, Songkhla Province. The location of the experimental room is shown in Fig.17. The total floor area of the experiment room is 19.25 m^2 and height 2.8 m.

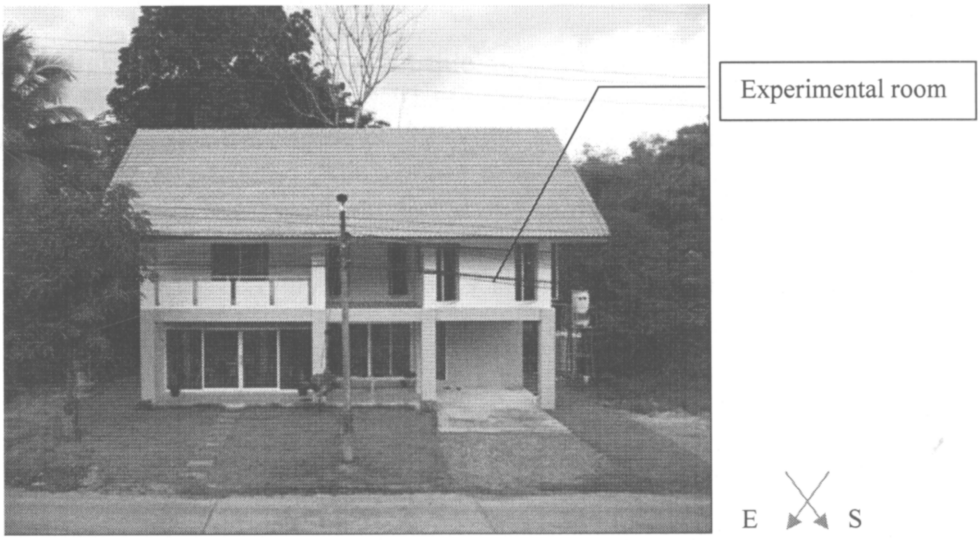


Figure 17 Location of the experiment room

Fiber Glass was used as insulation and placed above the ceiling of the experiment room to reduce external heat gain due to solar radiation. Figure 18 shows the installation of the fiber glass in the experimental room.

Figure 18 Fiber glass insulation

The opaque wall is light weight concrete with thickness 75 mm and surface coated with cement motar. Transparent wall-SMG-III is used in this study and it has ultra violet transmittance 13%, solar transmittance 32% and solar absorptance 63%. The radiant cooling panels are constructed from copper tube bonded to aluminium sheet. The cooling panels are installed at the ceiling and at one side of the opaque wall in the experimental room. The designed area of the ceiling and wall radiant cooling panels are 16.83 m^2 and 5.44 m^2, respectively. Figure 19 shows the fabrication of copper tube and the radiant cooling panel after bond with aluminium sheet in the experimental room is illustrated in Fig.20.

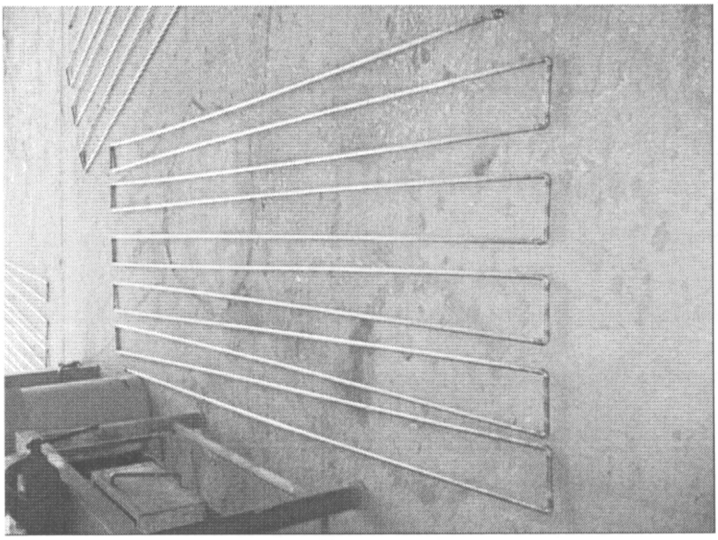


Figure 19 Fabrication of copper tube

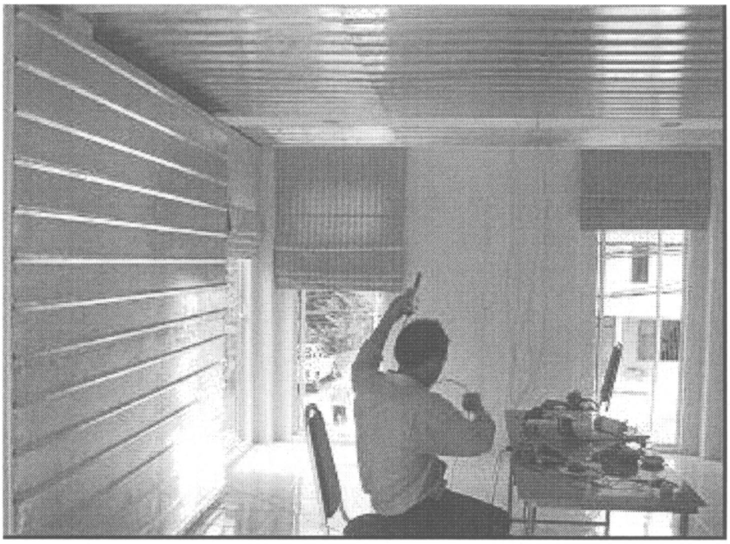


Figure 20 Experiment room and the radiant cooling panel

The designed cooling tower, storage tank and piping system are installed beside the experimental room and connected with the ceiling and wall radiant cooling panels as seen in Fig.21.

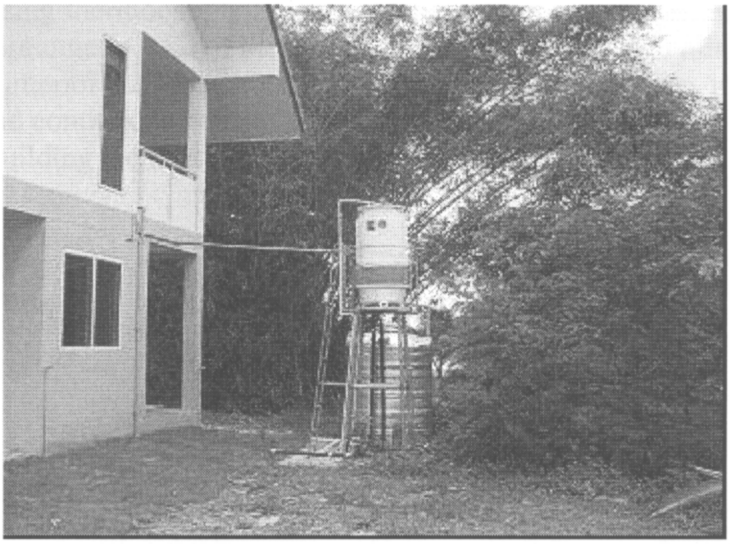


Figure 21 Cooling tower, storage tank and piping system

As mentioned earlier, the design system requires the control system for step 1 and step 2. Thus, the control circuit is designed and the control panel can be seen in the following figure.

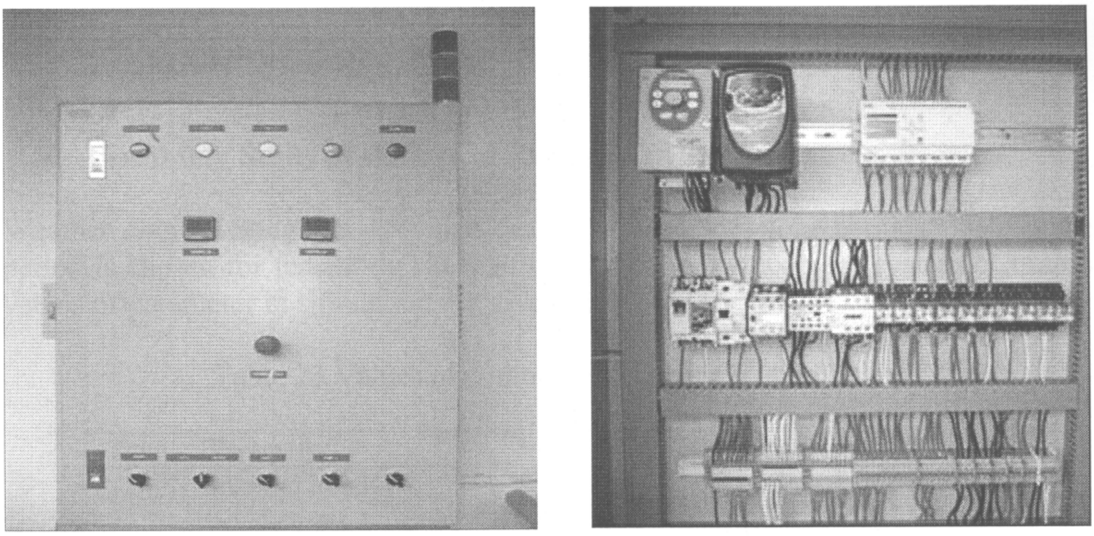


Figure 22 Control panel

7.4.5 Data Collection

Temperature and humidity sensors are installed at the interior and exterior of the experimental room to measure the temperature and relative humidity of interior and exterior air. Surface temperatures of opaque wall and radiant cooling panels are measured by using thermocouple type K. Mean radiant temperature is also recorded using globe thermometer. All of these data are recorded every five minutes using data logger. The interior air speed and electricity consumption are measured using anemometer and continuous power meter. Arrangements of the various sensors in the experimental building are illustrated in Fig.23.

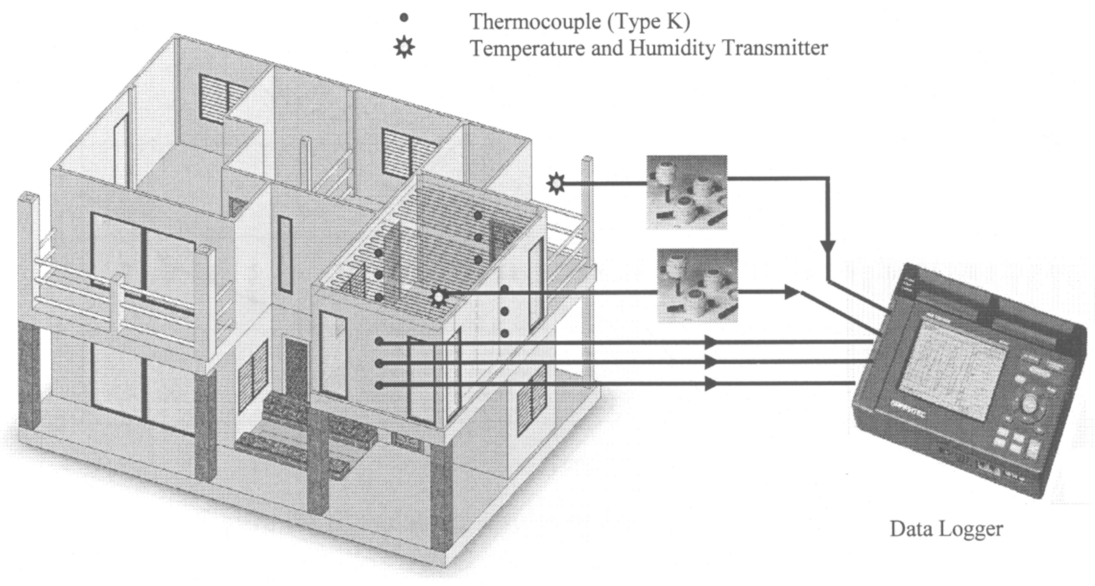


Figure 23 Arrangement of various sensors in the experimental room

7.5 Results and Discussion

- **Feasibility Study of the Use of Radiant Cooling System**

Weather data of Songkhla Province can be divided into four periods. The hottest period is chosen for feasibility study of the use of radiant cooling system. Common values of parameter in simulation are given in Table 23.

Table 23 Values of common parameters in simulation

Item	Parameter	Unit	Value
1	Internal Load	W	582
	- Occupant	W	260
	- Lighting	W	72
	- Equipment	W	250
2	Flow Rate of Cool Water	kg/s	0.289
3	Area of Cooling Panel	m^2	22.26
4	Interior Air Velocity	m/s	0.5
5	Occupancy Parameter		
	- Metabolic Rate	met	1.2
	- Clothing Insulation	clo	0.5

The developed model of the experimental room and reference day on 12 March 2000 is used for simulation to predict PMV value. Simulation result is shown in Fig.24.

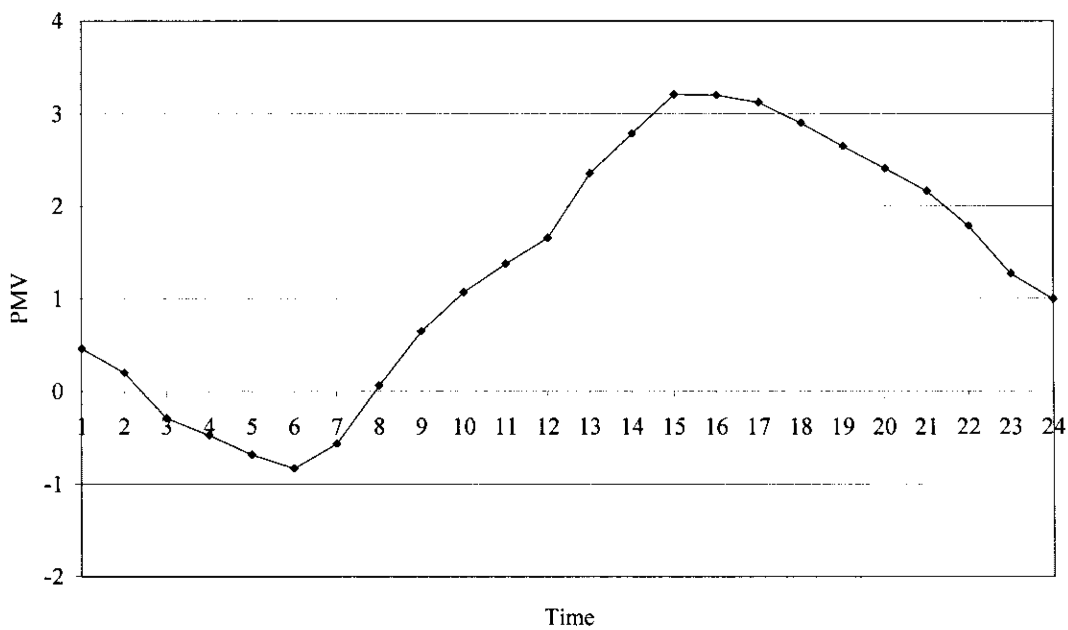


Figure 24 Simulation results of PMV, 12 March 2000

It was observed that thermal comfort can be obtained during the night time. During the day time after 10:00, the value of PMV is more than one which is in the range of unacceptably warm. Then, this study focuses only on the night time application.

- **Sensitivity Analysis**

The sensitivity analysis is also examined by varying the clothing level, metabolic rate and interior air velocity through the simulation. Nevertheless, the trend results are in the same pattern.

Varying Clothing Level: The clothing level is varied in the range of 0.2-0.8 clo. Results from simulation are given in Fig.25. The weather data on 20-21 September 2006 is used for simulation. It was observed that the pattern of PMV values by varying clothing level are in the same trend. The simulation reveals that the comfort can not be obtained for higher clothing level at 0.8 clo which is in the range of unacceptably warm. For the lower clothing level at 0.2 clo, the comfort can be obtained during 18:00-20:00, occupants feel very cold and the comfort can not be obtained during 20:00-9:00. The comfort can be achieved for clothing level at 0.5 clo during 23:00 to 8:00.

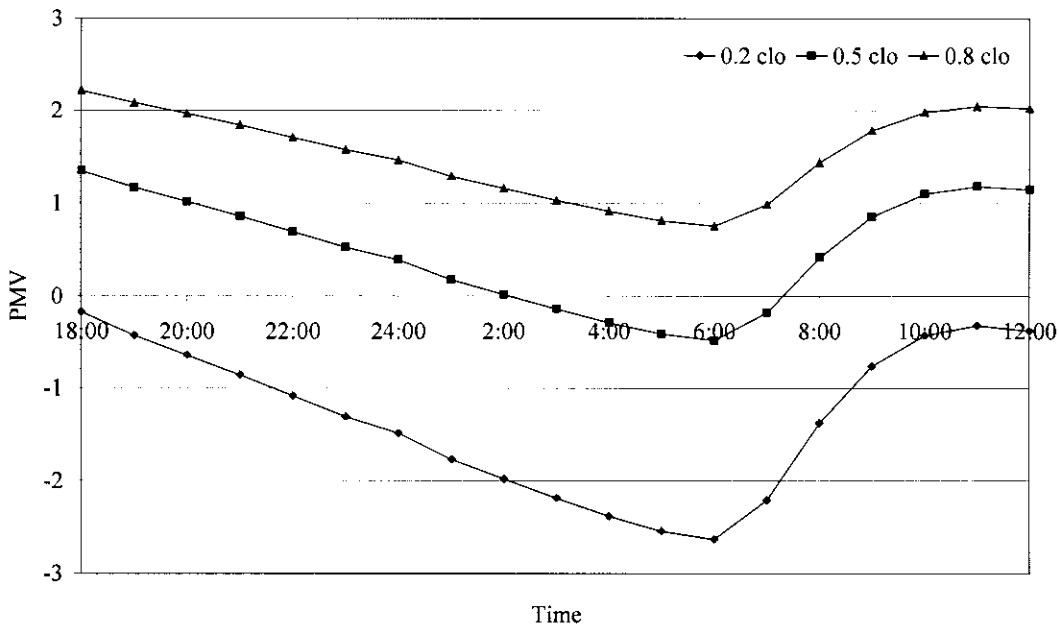


Figure 25 Values of PMV by varying clothing level

Varying Metabolic Rate: The metabolic rate is varied in the range of 1.0 met – 1.6 met. Figure 5.11 shows the resultant PMV values by varying metabolic rate in the rang 1.0-1.6 met. It can be seen that thermal comfort can be achieved at metabolic rate 1.2 met during 23:00-8:00. The radiant cooling system is not appropriate with higher activity or higher metabolic rate i.e. at the metabolic rate 1.6 as seen in Fig.26. In contrast, lower activity also falls in the unacceptable cool during 1:00 to 7:00.

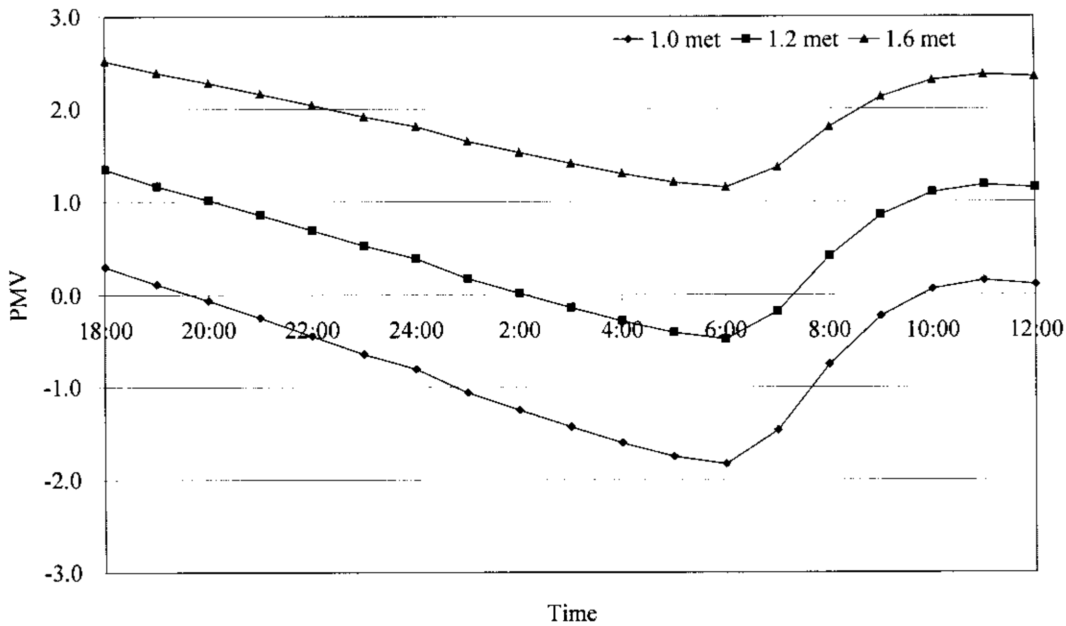


Figure 26 Values of PMV by varying metabolic rate

Varying Interior Air Velocity: Figure 27 illustrates the PMV values by varying interior air velocity. Interior air velocity is also the main parameter to be considered for thermal comfort. In this study, the simulation is done by varying the range of interior air velocity 0.15-1.0 m/s.

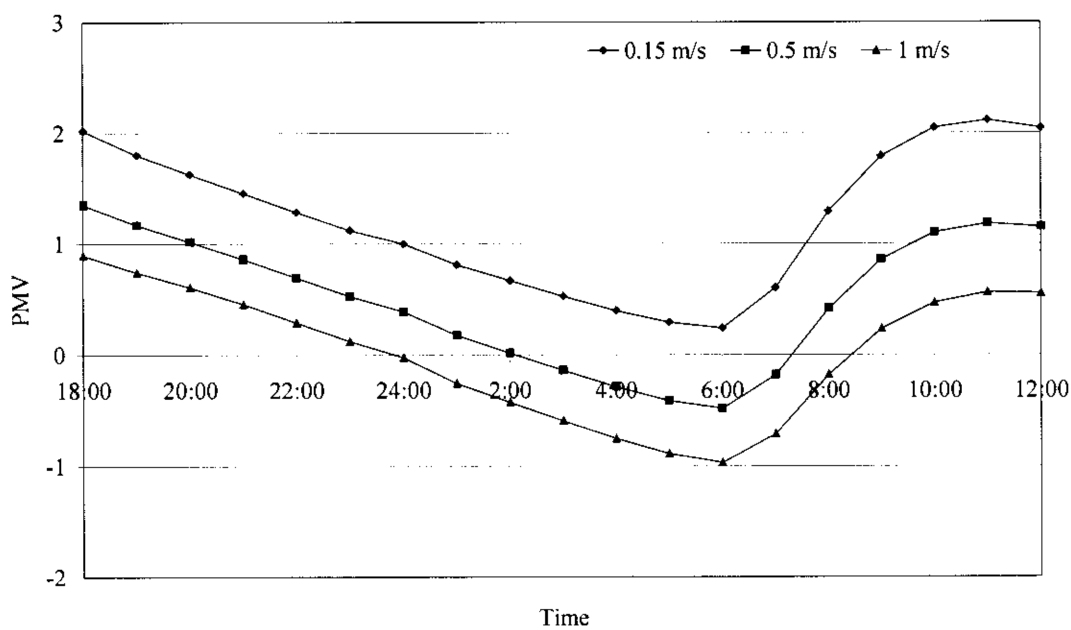


Figure 27 Values of PMV by varying interior air velocity

It can be seen from the results that thermal comfort can still be fulfilled at the lower air velocity in the narrow ranges during 3:00-7:00. The higher air velocity helps at the beginning for thermal comfort achievement. Nevertheless, it is not proper for the late night time after 4:00 which occupants feel cool. The interior air velocity at 0.5 m/s can give the widely range for thermal comfort.

- **The performance of Cooling Tower**

In this study, the performance of cooling tower is also evaluated. The followings are the measurement data of cooling tower.

Table 24 Measurement values of cooling tower

Item	Measurement Data	Abbreviation	Unit	Values
1	Water inlet temperature	T_{wi}	°C	25.6
2	Water outlet temperature	T_{wo}	°C	24.7
3	Air inlet temperature:	T_{a1}	°C	25.2
4	Air outlet temperature	T_{a2}	°C	25.7
5	Wet bulb temperature	T_{wb}	°C	22.9
6	Air inlet relative humidity	RH_1	%	82.2
7	Air outlet relative humidity	RH_2	%	87.8
8	Average air inlet velocity	V_a	m/s	0.6
9	Area	A	m^2	0.56

The data obtained from Table 24 are used to examine the performance of cooling tower. Figure 28 illustrates the configuration of cooling tower.

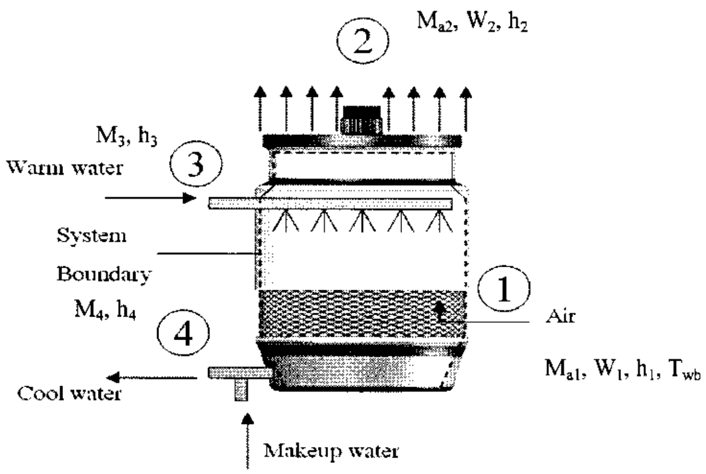


Figure 28 Configuration of cooling tower

The performance of cooling tower can be calculated using mass and energy balance as shown in the following equations.

Mass Balance

$$M_{a1} = M_{a2} \quad \text{(Eq.25)}$$

$$M_3 + M_{a1}W_1 = M_4 + M_{a2}W_2 \quad \text{(Eq.26)}$$

Energy Balance

$$M_{a1}h_1 + M_3h_3 = M_{a2}h_2 + M_4h_4 \quad \text{(Eq.27)}$$

$$\text{Cooling Tower Performance} = M_{3,\,actual} \times 100\%/M_{3,Theoritical} \quad \text{(Eq.28)}$$

Where:
- M_{a1} = air flow rate at the inlet, kg/s
- M_{a2} = air flow rate at the outlet, kg/s
- M_3 = water flow rate at the inlet, kg/s
- M_4 = water flow rate at the outlet, kg/s
- W_1 = Humidity ratio at air inlet, $kg_{moisture}/kg_{dry\ air}$
- W_2 = Humidity ratio at air outlet, $kg_{moisture}/kg_{dry\ air}$
- h_1 = air inlet enthalpy: kJ/kg
- h_2 = air outlet enthalpy: kJ/kg
- h_3 = water inlet enthalpy: kJ/kg
- h_4 = water outlet enthalpy: kJ/kg

The performance of cooling tower is 59.96 % using mass and energy balance. Detailed calculation is given in Appendix C. It was observed that the wet bulb temperature, relative humidity of exterior air and cool water flow rate affect the performance of cooling tower. The higher performance of cooling tower can be obtained if the water outlet temperature and wet bulb temperature are slightly difference.

- **Evaluation of Radiant Cooling Panel Temperature**

Interior air temperature, average opaque wall temperature and radiant cooling panel temperature from measurement are shown in Fig.29. It can be seen that the temperature of interior air drops from 27.7 °C to 25.6 °C during 18:00-7:00. The temperatures of interior air and opaque wall are slightly difference. It was also observed that the interior air, opaque wall and radiant cooling panel temperatures are in the same pattern, which is decreasing during 18:00-7:00 and increasing during 7:00-12:00.

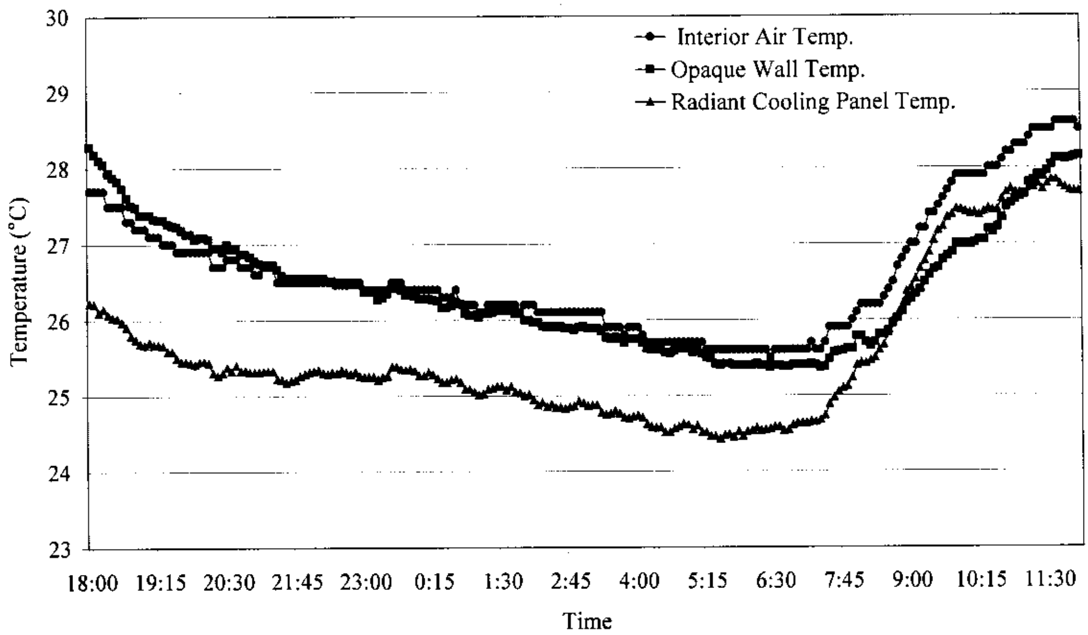


Figure 29 Interior temperature, opaque wall temperature and radiant cooling panel temperature from measurement during 20-21 September, 2006

It was observed that the accumulated storage load at the opaque wall will release to the experimental room in the night time. Therefore, the interior air temperature and opaque wall temperature are slightly difference. Meanwhile, the interior air temperature and opaque wall temperature decrease continuously until 7:00 according to radiant cooling panels absorb heat gain in the experimental room. Nevertheless, the interior air temperature, opaque wall temperature and radiant cooling panel temperature increase steadily after 7:00. This is due to the effects of exterior environment i.e. temperature and relative humidity of exterior air.

Figure 30 illustrates interior temperature, mean radiant temperature and radiant cooling panel temperature from measurement during 20-21 September, 2006. The mean radiant temperature is calculated in details in Appendix D. There have the radiant and convective heat transfers from mean radiant temperature to radiant cooling panel during 18:00 to 23:00. The different between mean radiant temperature and radiant cooling panel temperature is nearly zero during 23:00 to 6:00. The interior air temperature decreases in the same pattern as radiant cooling panel temperature.

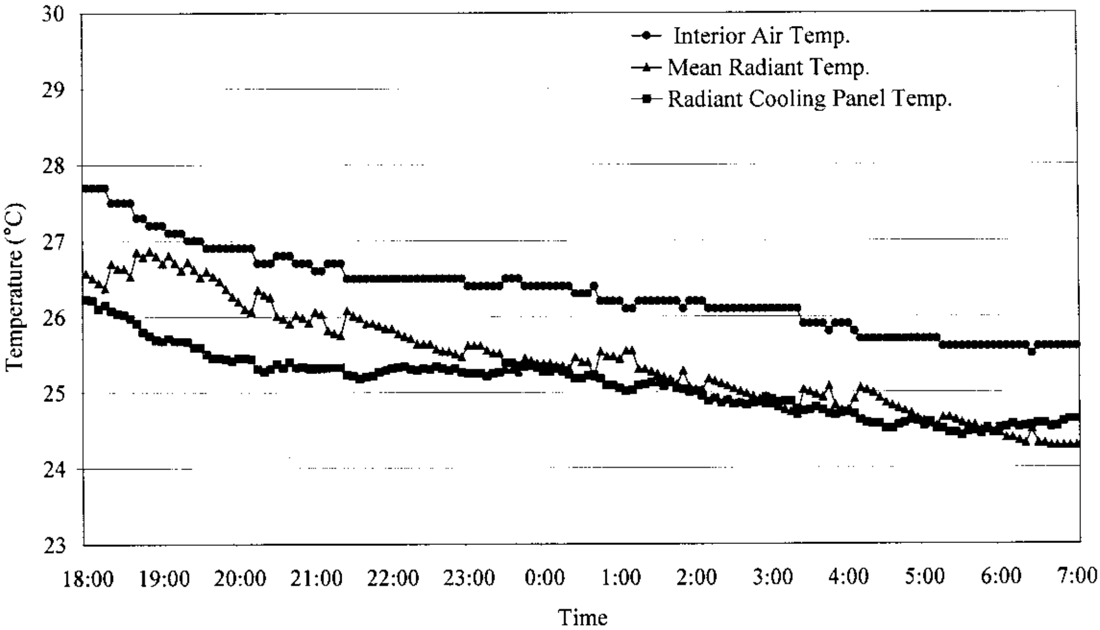


Figure 30 Interior temperature, mean radiant temperature and radiant cooling panel temperature from measurement during 20-21 September, 2006

Figure 31 shows the comparison between interior and exterior temperatures with and without radiant cooling system. It was revealed that in the duration time of 23:00-7:00 which exterior temperature in case of with and without radiant cooling system is rather the same. The interior temperature of the experimental room in case of with radiant cooling system has lower temperature comparing with the case of without radiant cooling system.

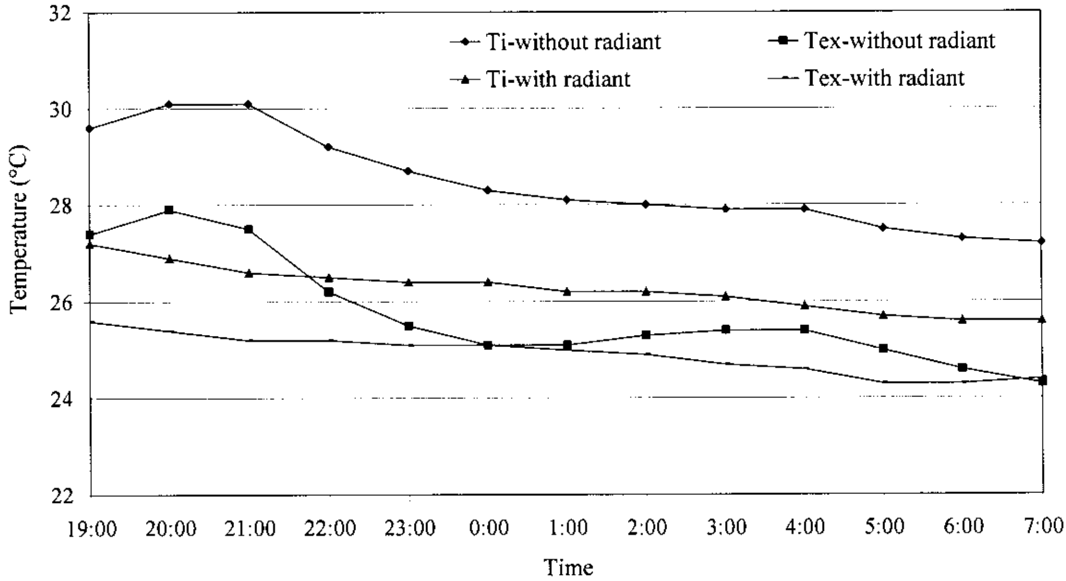


Figure 31 Comparison between interior and exterior temperature in case of with radiant cooling system and without cooling system

- **Thermal Comfort Evaluation**

Evaluation of thermal comfort for the use of radiant cooling panel using PMV value is investigated and the results imply that thermal comfort can be obtained by using radiant cooling panel in the night time. In this study, the system operated at cool water flow rate 0.289 kg/s and air speed in the experimental room is 0.5 m/s. Figure 32 illustrates the PMV values for 19 July 2006, 5 August 2006 and 20 September 2006. The cool water is supplied to wall and ceiling cooling panels from 6:00. It was observed that the PMV values are in the range of -0.5 – 0.5 which is in the comfortable range during 20:00 to 10:00 for all selected days. The PMV value is more than 0.5 after 10:00 which the occupants in the experimental room feel warm.

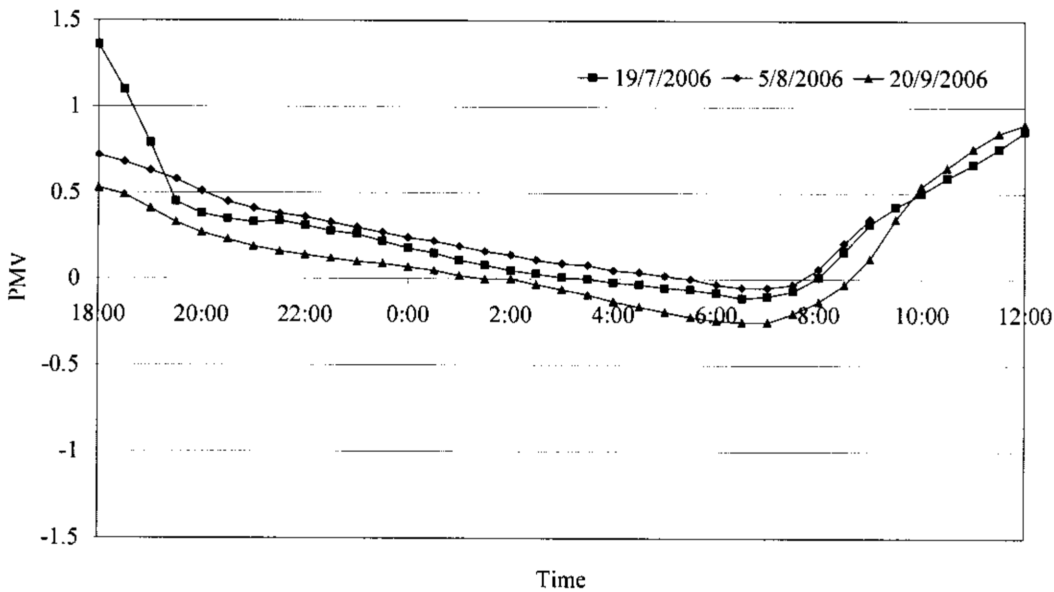


Figure 32 PMV values for the selected days

- **Energy Saving**

The radiant cooling system consumes electrical power approximately 541.06 W. Figure 33 illustrates the power consumption for the use of radiant cooling system during 20-21 September 2006. Normally, the design of split type air conditioner for the experimental room sizing is 12,000 Btu/hr. The specified power consumption is 1,127.60 W for high energy efficient ratio (EER) air conditioner. In the comparison of the use of radiant cooling system and the use of split type air conditioner (at 80% of specification of power consumption which is equal to 902.08 W), energy savings can be obtained approximately 40% when using radiant cooling system compared to air conditioner.

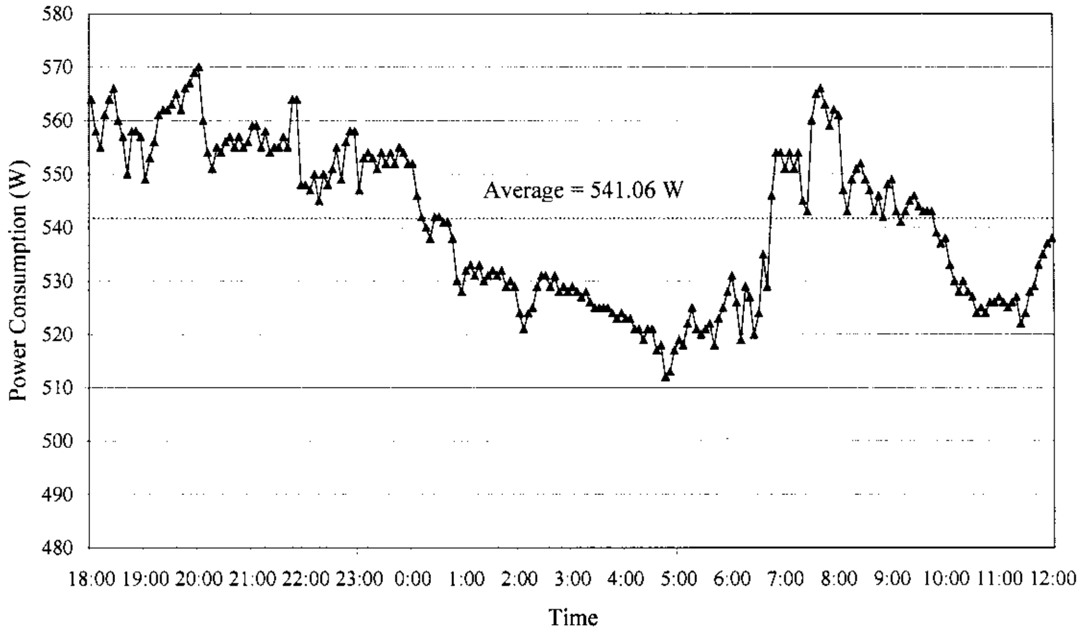


Figure 33 Power consumption for radiant cooling panel system, 20-21 September 2006

- **Comparison between the Experiment and Simulation**

Comparisons of mean radiant temperature between simulation and experiment are given in Fig.34.

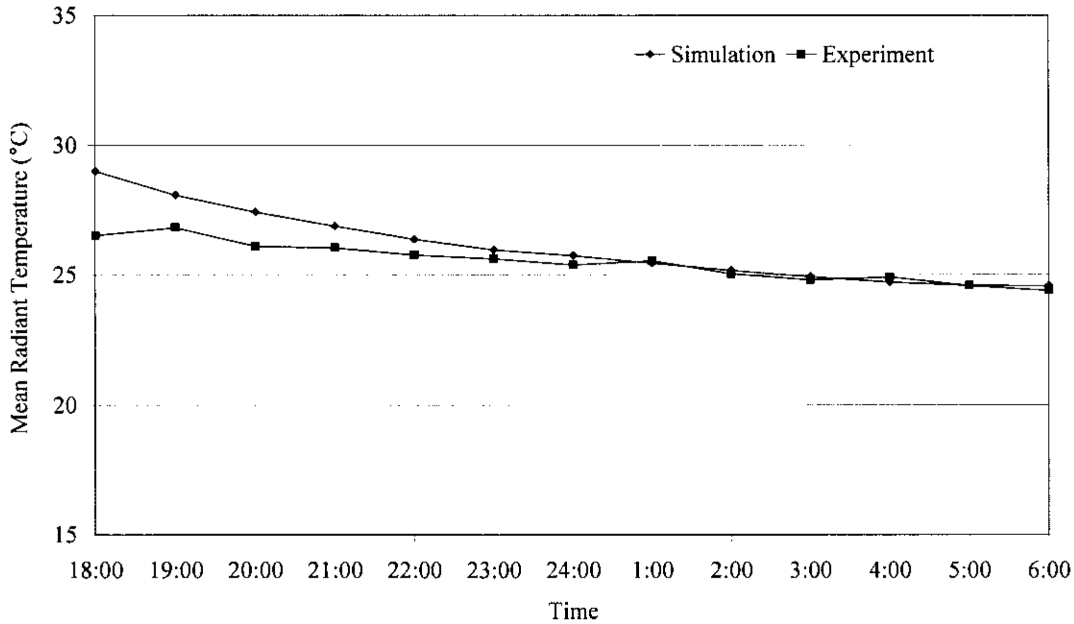


Figure 34 Comparisons of mean radiant temperature between simulation and experimental, 20-21 September 2006

It was observed that the mean radiant temperature prediction from simulation has slightly variation comparing with the results from experiment.

Figure 35 shows the comparison between simulation and experiment on 20-21 September 2006. The input file for simulation is used with the Songkla weather data to predict PMV values.

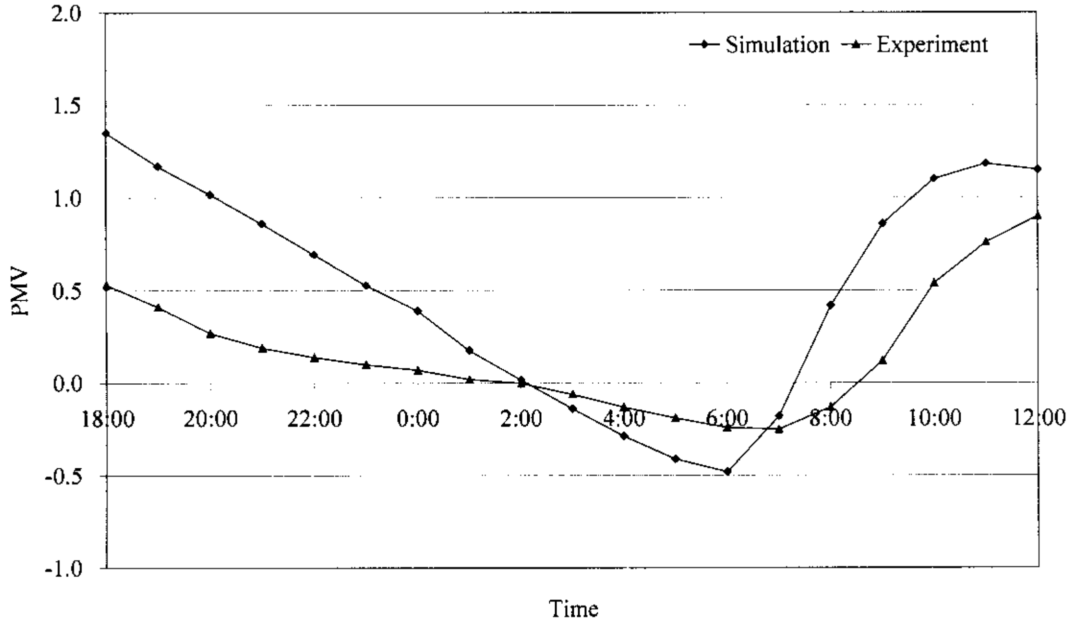


Figure 35 Comparisons of PMV value between simulation and experimental, 20-21 September 2006

It can be seen that results from simulation is over expectation comparing with results from the experiment.

7.6 Conclusion and Recommendation

The radiant cooling system can be applied for buildings under the tropical climate and consumes less energy consumption comparing with the use of air conditioner. Experimental results revealed that energy saving can be obtained approximately 40% using radiant cooling system instead of air conditioner. Cooling tower could be employed to provide cool water for radiant cooling panels. The performance of cooling tower was 59% and depended on the temperature and humidity of exterior air.

Radiant cooling panel temperature decreased continuously by heat transfer between non-cooling surface and radiant cooling panel during 18:00 to 7:00. Mean radiant temperature which is the temperature surrounding the body decreased continuously from 18:00 and this temperature was nearly the same as temperature of radiant cooling panel during 23:00 to 7:00.

Values of PMV were used to examine thermal comfort and found that the use of radiant cooling system is appropriate for the night time and early morning (until

10:00). The sensitivity analysis was also investigated by varying clothing level, metabolic rate and interior air velocity. It was observed that all of these three factors affect strongly to comfort level.

The metabolic rate was varied in the range of 1.0 met – 1.6 met through simulation. Results revealed that thermal comfort can be achieved at metabolic rate 1.2 met during 23:00-8:00. The radiant cooling system is not appropriate with higher activity or higher metabolic rate i.e. at the metabolic rate equal to 1.6 met. On the other hand, lower activity also falls in the unacceptable cool during 1:00 to 7:00. The clothing level was varied in the range of 0.2 clo-0.8 clo through simulation. Results showed that thermal comfort can be achieved at 0.5 clo during 23:00 to 8:00. Thermal comfort is still fulfilled at the lower air velocity in the narrow ranges during 3:00-7:00. The higher air velocity helps for thermal comfort achievement at the beginning. Nevertheless, it is not appropriate for the late night time after 4:00 which occupants feel cool. The interior air velocity at 0.5 m/s gives the widely range for thermal comfort.

This study concentrates only one metabolic rate for sedentary work. Thermal comfort depends on various parameters i.e. metabolic rate, level of clothing, wind speed. Thus, the detailed studies on different activities and level of clothing should be focused for the further researches. The relative humidity of interior air and humidity control in the building are also the important parameters which should be integrated for the future works.

8. List of Publication and Proceeding

- **International Conference**

  1. Ar-U-Wat Tantiwichien, Juntakan Taweekun, Chukiat Kooptanond, Panyarak Ngamsritragul, 2006. An Experimental and Simulated Study of Radiant Cooling Panel in Residential Buildings for Tropical Climate. 2nd International Conference on "SEE 2006" Sustainable Energy and Environment 2006, Bangkok, Thailand, 21-23 November 2006

- **National Conference**

  1. Ar-U-Wat Tantiwichien, Juntakan Taweekun, Chukiat Kooptanond, Panyarak Ngamsritragul, 2006. A Study on Hydronic Radiant Cooling in a Buildings for Tropical Climate. 4th Graduate Symposium, Prince of Songkla University, Songkhla, Thailand, 31 March, 2006.

  2. Ar-U-Wat Tantiwichien, Juntakan Taweekun, Chukiat Kooptanond, Panyarak Ngamsritragul, 2005. Development of Use of Radiant Cooling Panel for Passive Cooling in Buildings. 19th ME-Nett, Phuket, Thailand, 19-21 October 2005.

References

[1] Alamdari, F., Butler, D.J., Grigg, P.F. and Shaw, M.R., 1998. Chilled Ceilings and Displacement Ventilation. Renewable Energy 15, 300-305.

[2] ASHRAE, 1996. 1996 ASHRAE Handbook-HVAC systems and equipment. Atlanta: American Society of Heating, Refrigerating and Air-conditioning Engineering, Inc.

[3] ASHRAE, 1997. Handbook-Fundamental. Atlanta: American Society of Heating, Refrigerating and Air-conditioning Engineering, Inc.

[4] Auliciems, A., 1981.Towards a Psycho Physiological Model of Thermal Perception. International of Biometeorology, 25:109-122.

[5] Bush, J.F., 1992. A Table of Two Population; Thermal Comfort in Air-Conditioned and in Naturally Ventilated offices in Thailand. Energy and Buildings 18; 235-249.

[6] Carpenter, S.C., and Kokko, J.P., 1998. Radiant Heating and Cooling, Displacement Ventilation with Heat Recovery and Storm Water Cooling: An environmentally Responsible HVAC System. ASHRAE Transactions 104(2): 1321-1326.

[7] Conroy, C.L., and Mumma, S.A., 2001. Ceiling radiant cooling panels as a viable distributed parallel sensible cooling technology integrated with dedicated outdoor-air system. ASHRAE Transactions 107(1).

[8] Fanger, P.O., 1970. Thermal comfort analysis and applications in environmental engineering. Mcgraw-Hill, New York.

[9] Feustel, H.E. and Stetiu, C., 1995. Hydronic Radiant Cooling-Preliminary Assessment. Energy and Building 22, 193-205.

[10] Givoni, Baruch, 2000. Climate Responsive Building Design in Hot Humid Regions, Proceedings of a conference on Green Architecture: the Sustainable Built Environment in the New Millenium, organized by the Council of Deans of Architecture Schools of Thailand, 21-22 April 2000, Bangkok.;235-239

[11] Imarori, T., Omori, T. and Bogaki, K. (1999). Thermal Comfort and Energy Consumption of the Radiant Ceiling Panel System: Comparison with All-Air System. Energy and Buildings 30, 167-175.

[12] ISO, 1994. Moderate thermal environments: Determination of the PMV and PPD indices and specification of the conditions for thermal comfort, standard 7730.

[13] Jitkhajornwanich, K, Adrian C. Pitts, Altbert Malama and Steve Sharples. 1998. Thermal Comfort in Transitional Spaces in the Cool Season of Bangkok. ASHRAE Transactions, Part IA: 1181-1193.

[14] Khedari, J., Yamtraipat, N. and Hirunlabh, J. (2001). A New Concept for Setting Thermal Comfort Standards. Conference on Moveing Thermal Comfort Standards into the 21st Century, UK, 472-478.

[15] Koichi, K., Noriko, K., Hiroko, H., and Shin-ichi, T., 1999. Effect of Humidity and Small Air Movement on Thermal Comfort under a Radiant Cooling Ceiling by Subjective Experiments. Energy and Buildings 30: 185-193.

[16] Levine, M.D., Deringer, J.J., and Buscg, J.F., 1992. ASEAN-USAID Building Energy Conservation Project, Find Report, Energy Analysis Program, Energy and Environment Division, Lawrence-Berkeley Laboratory, June.

[17] Material Handbook, 1998. Faculty of Energy and Material, King Mongkut's University of Technology, Thonburi.

[18] Min, T.C., Schutrum, L.F., Parmelee, G.V., and Vouris, J.D., 1956. Natural Convection and Radiation in a Panel Heated Room. ASHRAE Transactions 62, 237.

[19] Miriel, J., Serres, L., and Trombe, A., 2002. Radiant ceiling panel heating-cooling systems: Experimental and simulated study of the performances, thermal comfort and energy consumptions. Applied Thermal Engineering 22, 1861-1873.

[20] Kreider, J.F., Rabl, A. Heating and cooling of building: Design for efficiency. Mechanical Engineering Hand Book.

[21] Mumma, S.A., 2001 Ceiling Panel Cooling Systems, ASHRAE Journal, November, 28-32.

[22] Oliveira, F.J. (2000). Thermal Behavior Closed of Wet Cooling Towers for Use with Chilled Ceilings. Applied Thermal Engineering 20, 1225-1236.

[23] Prapapong Vangtook and Surapong Chirarattananon, 2004. An Experimental Investigation of Thermal Comfort from Application of Radiant Cooling in Thailand. The Joint International Conference on Sustainable Energy and Environment (SEE), 1-3 December 2004, Hua Hin, 512-520.

[24] Roulet, C.A., Rossy, J.P., and Roulet, Y., 1999. Using Large Radiant Panels for Indoor Climate Conditioning. Energy and Building 30, 121-126.

[25] Sherman, M.H. and Grimsrud, D.T., 1980. Infiltration-Pressurization correlation: Simplified physical modeling. ASHRAE Transactions 86(2): 778.

[26] Simmonds, P., 1996. Practical Applications of Radiant Heating and Cooling to Maintain Comfort Conditions, ASHRAE Transactions (1): 659-666.

[27] Stetiu, C., 1999. Energy and Peak Power Savings Potential of Radiant Cooling System in US Commercial Buildings. Energy and Buildings 30, 127-138.

[28] Stetiu, C., and Feustel, H.E., 1994. Development of a model to Simulate the Performance of Hydronic/Radiant Cooling Ceilings, Lawrence Berkeley Laboratory, California.

[29] Surapong, 2005, Building for energy efficiency

[30] Szokolay, S.V., 1980. Environmental Science Handbook. Longman/Construction press, Wiley/Hasted Press.

[31] Yamtraipat, N., Khedari, J. and Hirunlabh, J. (2005). Thermal comfort standards for air conditioned buildings in hot and humid Thailand considering additional factors of acclimatization and education level. Solar Energy 78, 504-517.

APPENDIX A

Heat Gain, Air Leakage and Radiant Panel Cooling Calculation

Appendix A

A.1 Calculation of Overall Heat Transfer Coefficient

In this study, the values of exterior surface heat transfer coefficient, h_o, exterior surface heat transfer coefficient for roof, $h_{ex,r}$, and interior surface heat transfer coefficient, h_i, are 22 $W/m^2.K$, 18 $W/m^2.K$ and 8 $W/m^2.K$, respectively. The followings are the sample calculations of overall heat transfer coefficient for opaque wall, door, roof and transparent wall.

A.1.1 Overall Heat Transfer Coefficient for Opaque Wall

Overall heat transfer coefficient of opaque wall can be calculated as:

$$U_w = \frac{1}{\left[\left(\frac{1}{h_o}\right)+\Sigma\left(\frac{\Delta x}{K}\right)+\left(\frac{1}{h_i}\right)\right]}$$

$$= \frac{1}{\left[\left(\frac{1}{22}\right)+\left(\frac{0.01}{0.305}+\frac{0.075}{0.121}+\frac{0.01}{0.305}\right)+\left(\frac{1}{8}\right)\right]}$$

$$= 1.168 \quad W/m^2.K$$

A.1.2 Overall Heat Transfer Coefficient for Door

The overall heat transfer coefficient of door can be calculated as:

$$U_w = \frac{1}{\left[\left(\frac{1}{h_o}\right)+\Sigma\left(\frac{\Delta x}{K}\right)+\left(\frac{1}{h_i}\right)\right]}$$

$$= \frac{1}{\left[\left(\frac{1}{22}\right)+\left(\frac{0.03}{0.118}\right)+\left(\frac{1}{8}\right)\right]}$$

$$= 2.355 \quad W/m^2.K$$

A.1.3 Overall Heat Transfer Coefficient for Roof, U_r

Using the Eq.5, the following is the sample calculation of the value of overall heat transfer coefficient of roof.

$$Ur = \frac{1}{\left[\left(\frac{1}{h_{ex,r}}\right) + \Sigma\left(\frac{\Delta x}{K}\right) + \left(\frac{1}{h_i}\right)\right]}$$

$$= \frac{1}{\left[\left(\frac{1}{22}\right) + \left(\frac{0.012}{0.978} + \frac{0.002}{0.03} + 0.986 + \frac{0.075}{0.038}\right) + \left(\frac{1}{8}\right)\right]}$$

$$= 0.312 \quad W/m^2.K$$

A.1.4 Overall Heat Transfer Coefficient for Transparent Wall, U_f

$$U_f = \frac{h_i h_o}{h_i + h_o}$$

$$= \frac{(8)(22)}{(8+22)}$$

$$= 5.867 \quad W/m^2.K$$

A.2 External Heat Gain

- **External Heat Gain Due to Solar Radiation Through Opaque Walls, Q_w**

Table A.2.1 Heat gain through opaque walls

Direction	U_w ($W/m^2 \cdot °C$)	A_w (m^2)	I_{facade} (W/m^2)	α	h_0 ($W/m^2 \cdot °C$)	Q_w (W)
North						
- wall	1.168	2.87	73.9	0.57	22	6.42
- door	2.355	1.89	73.9	0.57	22	8.52
East	1.168	12.04	95.87	0.57	22	34.93
South	1.168	6.44	90.69	0.57	22	17.67
West	1.168	2.00	99.90	0.57	22	6.05
Total heat gain through opaque wall						73.59

Sample calculation for Table A.2.1, East Direction

$$Q_w = \frac{U_w A_w I_{facade} \alpha}{h_o}$$

$$= \frac{(1.168)(12.04)(95.87)(0.57)}{22}$$

$$= 34.93 \quad W$$

- **External Heat Gain Due to Solar Radiation Through Roof, Q_r**

Table A.2.2 Heat gain through roof

Detail	U_r (W/m^2 °C)	A_r (m^2)	I_{global} (W/m^2)	α	$h_{ex,r}$ (W/m^2 °C)	Q_r (W)
Roof	0.312	19.25	228.88	0.57	18	43.53

Sample calculation for Table A.2.2

$$Q_r = \frac{U_r A_r I_{global} \alpha}{h_{ex,r}}$$

$$= \frac{(0.312)(19.25)(228.88)(0.57)}{18}$$

$$= 43.53 \quad W$$

- **External Heat Gain Due to Solar Radiation Through Transparent Wall, Q_f**

Table A.2.3 Heat gain through transparent wall

Direction	SC	I_{global} (W/m^2)	F	A_f (m^2)	Q_f (W)
North	0.59	73.90	0.49	-	-
East	0.59	95.87	0.49	3.36	93.13
South	0.59	90.69	0.49	3.36	88.09
West	0.59	99.90	0.49	1.80	51.99
Total heat gain through transparent wall					233.21

Sample calculation for Table A.2.3, East direction

$$F = \tau + \alpha \frac{U_f}{h_o}$$

$$= 0.32 + 0.63\left(\frac{5.867}{22}\right)$$

$$Q_r = A_f (SC) F I_{facade}$$

$$= (3.36)(0.59)(0.49)(95.87)$$

$$= 93.13 \quad W$$

A.3 Air Leakage

Volumetric air flow rate, $\dot{V}$, due to air leakage is calculated according to Eq. 2.9. Stack coefficient, C_s, for two story houses as mentioned in Chapter 2 is 0.00029 and wind coefficient, C_w, from Table 2.4 at two story houses and class 2 is 0.000325. Average interior and exterior temperature, Δt, is 1.96 K and wind speed, V, obtained from the meteorological Khohong station is 0.9 m/s. The calculation of volumetric air flow rate is as following.

$$
\begin{aligned}
\dot{V} &= A_L \sqrt{C_s \Delta t + C_w V^2} \\
&= 7.48\sqrt{(0.00029\text{x}1.96) + 0.000325(0.9)^2} \\
&= 0.2157 \quad \text{L/s}
\end{aligned}
$$

The energy required to warm exterior air entering by air leakage to the temperature of the room, Q_{as}, and energy associated with net loss of moisture from the space, Q_{al}, are calculate using equations 2.10 and 2.11 as following.

$$
\begin{aligned}
Q_{as} &= 1.2\,\dot{V}\,\Delta t \\
&= 1.2(0.2157)(1.96) \\
&= 0.51 \quad \text{W}
\end{aligned}
$$

$$
\begin{aligned}
Q_{al} &= 3\,\dot{V}(W_i - W_o) \\
&= 3(0.2157)(16.99\text{-}16.79) \\
&= 0.13 \quad \text{W}
\end{aligned}
$$

$$
\begin{aligned}
\text{Total loss} &= Q_{as} + Q_{al} \\
&= 0.51 + 0.13 \quad \text{W} \\
&= 0.64 \quad \text{W}
\end{aligned}
$$

A.4 Radiant Panel Cooling Calculation

<u>Assumption Values</u>

$T_{n,p}$ = Average temperature of the non-radiant panel surface of the room
= 32 °C
= 549 °R

T_p = mean panel surface temperature
= 25 °C
= 536.40 °R

<u>Detailed Calculation</u>

Radiant panel cooling by radiation, Q_r

$$
\begin{aligned}
Q_r &= 0.15\times10^{-8}\left[\left(T_{n,p}^4\right)-\left(T_p^4\right)\right] \\
&= 0.15 \text{ x } 10\text{-}8\ [(549)^2 - (536.4)^4] \\
&= 12.09 \quad \text{Btu/hr.ft}^2 \\
&= 38.11 \quad \text{W/m}^2
\end{aligned}
$$

(conversion factor 1 W/m^2 = 0.3170 $Btu/hr.ft^2$)

Radiant panel cooling by convection, Q_c

$$
\begin{aligned}
Q_c &= 0.31\left(T_{n,p}-T_p\right)^{0.31}\left(T_{n,p}-T_p\right) \\
&= 0.31(549 - 536.4)^{0.31}(549 - 536.4) \\
&= 8.57 \quad \text{Btu/hr.ft}^2 \\
&= 27.02 \quad \text{W/m}^2
\end{aligned}
$$

Radiant panel cooling, Q

$$
\begin{aligned}
Q &= Q_r + Q_c \\
&= 38.11 + 27.02 \quad \text{W/m}^2 \\
&= 65.13 \quad \text{W/m}^2
\end{aligned}
$$

A.5 Area of Radiant Cooling Panel, A_{rad}

Area of radiant cooling panel is calculated using:

$$A_{rad} = \frac{Q_{gain}}{Q}$$

$$= \frac{1{,}450}{65.13}$$

$$= 22.26 \quad m^2$$

$$Q_{gain} = \dot{m} C_p (T_{wo} - T_{wi})$$

$$\dot{m} = \frac{Q_{gain}}{C_p (T_{wo} - T_{wi})}$$

$$= \frac{1{,}450}{(4{,}179)(1.2)}$$

$$= 0.289 \quad kg/s$$

Ceiling Radiant Cooling Panel

Design different temperature = 1.2 °C

Ceiling radiant panel area = 16.83 m^2

Ceiling Cool water flow rate = Ceiling radiant panel area x $\dot{m}$/radiant panel area

$$= \frac{(16.83)(0.289)}{22.26}$$

$$= 0.219 \quad kg/s$$

Cooling load, Ceiling

$$= \dot{m} C_p (T_{wo} - T_{wi})$$

$$= (0.219)(4{,}179)(1.2)$$

$$= 1{,}098.24 \quad W$$

Wall Radiant Cooling Panel

Wall radiant panel area = 5.44 m^2

Cool water flow rate = Total cool water flow rate – Ceiling cooling water flow rate

= 0.289 – 0.219

= 0.070 kg/s

Cooling load, Wall = Total cooling load – Cooling load, ceiling

= 1,450 – 1,098.24

= 351.76 W

APPENDIX B

Weather Data of Songkhla Province

Table B.1 Weather data for Songkhla Province, March 12, 2000

Hour	T_{amb}(°C)	RH (%)	T_w (°C)	T_{dew} (°C)	T_{sky}(°C)
0	26.89	84.08	24.74	23.97	19.16
1	26.66	85.56	24.72	24.04	19.01
2	26.32	87.23	24.63	24.03	17.98
3	25.94	88.94	24.49	23.98	17.59
4	25.39	91.57	24.31	23.92	15.68
5	24.83	92.9	23.93	23.60	14.52
6	23.58	94	22.84	22.56	15.75
7	23.7	94.81	23.06	22.82	20.39
8	26.33	73.55	22.69	21.24	15.74
9	29.57	45.16	20.73	16.47	15.5
10	32.06	33.95	20.34	14.24	18.04
11	33.54	27.61	19.94	12.35	19.79
12	34.45	26.47	20.29	12.48	20.95
13	35.83	24.01	20.60	12.16	21.77
14	37.29	23.56	21.47	13.09	22.57
15	38.33	20.4	21.26	11.75	22.29
16	38.04	20.42	21.07	11.53	21.08
17	36.95	20.33	20.33	10.58	19.5
18	33.69	28.55	20.27	12.99	17.4
19	29	40.68	19.42	14.33	15.44
20	28.44	45.34	19.86	15.51	15.24
21	28.23	52.62	21.00	17.65	17.4
22	27.53	56.94	21.15	18.26	17.8
23	26.78	61.26	21.21	18.72	17.28

Table B.2 Weather data for Songkhla Province, July 8, 2000

Hour	T_{amb}(°C)	RH(%)	T_w (°C)	T_{dew}(°C)	T_{sky}(°C)
0	27.88	70.52	23.65	22.03	28.21
1	27.06	79.03	24.17	23.11	28.43
2	27.13	77.53	24.02	22.87	27.64
3	27.29	76.03	23.95	22.70	27.28
4	27.06	76.8	23.85	22.64	26.13
5	26.98	75.4	23.57	22.26	24.67
6	26.6	76.82	23.42	22.20	24
7	28.07	72.53	24.14	22.67	24.39
8	29.55	66.17	24.46	22.57	24.68
9	30.64	59.3	24.24	21.80	26.48
10	31.66	55.12	24.35	21.56	28.54
11	33.01	51.41	24.77	21.66	30.07
12	32.91	50.63	24.53	21.32	30.17
13	30.22	65.63	24.97	23.07	30.98
14	29.9	69.8	25.37	23.79	29.66
15	32.64	55.47	25.25	22.57	30.09
16	31.35	59.33	24.86	22.47	29.66
17	30.97	59.01	24.47	22.03	28.08
18	30.52	62.05	24.62	22.43	25.46
19	28.64	71.21	24.45	22.92	25.79
20	28.3	74.02	24.58	23.23	27.44
21	28	76.19	24.63	23.42	25.62
22	27.92	72.82	24.04	22.60	26.1
23	27.62	69.84	23.31	21.63	24.79

Table B.3 Weather data for Songkhla Province, September 23, 2000

Hour	T_{amb}(°C)	RH (%)	T_w (°C)	T_{dew}(°C)	T_{sky}(°C)
0	24.29	93.94	23.53	23.25	27.13
1	24.29	93.63	23.49	23.20	27.35
2	24.41	93.26	23.57	23.25	27.45
3	24.55	93.1	23.68	23.36	27.55
4	24.65	93.05	23.77	23.45	27.68
5	24.61	93.18	23.75	23.44	27.69
6	24.64	93.62	23.84	23.54	27.65
7	25.56	90.34	24.31	23.86	28.54
8	27.13	80.74	24.49	23.54	29.83
9	29.46	70.81	25.14	23.61	29.4
10	31.82	61.27	25.62	23.44	28.45
11	31.4	61.12	25.23	23.01	30.73
12	31.98	59.16	25.38	23.01	31.12
13	33.57	49.74	24.90	21.64	30.44
14	35.43	43.74	25.13	21.23	31.47
15	36.71	39.82	25.24	20.84	30.62
16	34.07	51.27	25.63	22.59	29.18
17	29.32	70.59	24.97	23.42	26.43
18	26.04	86.07	24.20	23.53	28.18
19	24.36	94.24	23.64	23.38	26.59
20	24.43	93.58	23.63	23.33	26.05
21	24.61	92.5	23.66	23.32	25.99
22	24.63	93.06	23.76	23.44	25.35
23	24.74	93.1	23.87	23.55	25.22

Table B.4 Weather data for Songkhla Province, December 5, 2000

Hour	T_{amb}(°C)	RH (%)	T_w °C	T_{dew} °C	T_{sky}(°C)
0	24.02	81.04	21.59	20.58	16.86
1	23.86	80.03	21.31	20.22	16.74
2	23.16	84.3	21.21	20.38	16.05
3	22.74	83.45	20.70	19.81	15.34
4	22.5	83.18	20.43	19.52	15.03
5	21.94	85.96	20.25	19.50	14.7
6	21.45	87.44	19.96	19.29	14.55
7	22.54	78.93	19.92	18.72	14.63
8	25.86	60.7	20.32	17.71	15.87
9	29.14	49.15	21.13	17.41	18.14
10	31.74	41.1	21.63	16.93	20.72
11	32.56	38.64	21.75	16.69	22.73
12	32.05	39.8	21.60	16.71	23.62
13	33.02	37.35	21.81	16.57	23.75
14	32.84	38.21	21.87	16.76	23.23
15	32.96	37.26	21.75	16.48	22.82
16	32.53	38.05	21.59	16.43	21.94
17	30.86	44.22	21.58	17.30	20.85
18	28.33	56.41	21.74	18.85	19.46
19	26.89	64.98	21.89	19.77	18.6
20	25.97	69.35	21.74	19.94	18
21	25.13	75.47	21.86	20.50	17.14
22	24.65	75.79	21.46	20.11	16.96
23	24.15	79.38	21.49	20.37	16.62

APPENDIX C

Cooling Tower Performance Calculation

Appendix C

C.1 Calculation of Cooling Tower Performance

From Experimental
Cool water flow rate, $M_3 = 0.289$ kg/s

Psychrometric chart data
Humidity ratio at air inlet, $W_1 = 0.0166\ kg_{moisture}/kg_{dry\ air}$
Humidity ratio at air outlet, $W_2 = 0.01833\ kg_{moisture}/kg_{dry\ air}$
Specific air volume, $v_1 = 0.868\ m^3/kg_{dry\ air}$

Mass Balance

$$\begin{aligned} M_{a1} &= M_{a2} \\ &= V_a \times A/v_1 \\ &= 0.6 \times 0.56/0.868 \\ &= 0.387\ \text{kg/s} \end{aligned}$$

$$\begin{aligned} M_3 + M_{a1}W_1 &= M_4 + M_{a2}W_2 \\ M_3 - M_4 &= M_{a1}(W_2 - W_1) \\ &= 0.387 \times (0.01833-0.0166) \\ &= 0.000669\ \text{kg/s} \end{aligned}$$

Energy Balance

$$\begin{aligned} M_{a1}h_1 + M_3h_3 &= M_{a2}h_2 + M_4h_4 \\ M_3h_3 - M_4h_4 &= M_{a2}h_2 - M_{a1}h_1 \end{aligned}$$

$$(M_3 \times 107.4) - (M_4 \times 103.63) = (0.387 \times 72.38) - (0.387 \times 67.5) = 1.888$$

$$\begin{aligned} M_3 &= 0.482406\ \text{kg/s} \\ M_4 &= 0.481737\ \text{kg/s} \end{aligned}$$

$$\begin{aligned} \text{Cooling Tower Performance} &= M_{3,\ actual} \times 100\%/M_{3,\ Theory} \\ &= 0.289 \times 100\%/0.482406 \\ &= 59.9\ \% \end{aligned}$$

APPENDIX D

Calculation of Mean Radiant Temperature, PMV and PPD

Appendix D

D.1 Calculation of Mean Radiant Temperature

Sample calculation on 20 September, 2006 at 9.00 PM

1. Measurement data

$$T_i = 26.60 \quad ^{\circ}C$$
$$T_g = 5.533 \quad V$$
$$V_a = 0.5 \quad m/s$$

2. Globe temperature conversion (the formulation from the specification manual of globe thermometer)

$$\begin{aligned} T_g &= (21.6 \text{ x } T_g) - 40 \\ &= (21.6 \text{ x } 5.533) - 40 \\ &= 79.51 \quad ^{\circ}F \\ &= 26.40 \quad ^{\circ}C \end{aligned}$$

3. Mean radiant temperature

$$\begin{aligned} T_r &= \left[\left((T_g + 273)^4\right) + \frac{1.10x10^8 V_a^{0.6}}{\varepsilon D^{0.4}}\left(T_g - T_i\right)\right]^{1/4} \\ &= \left[\left((26.40 + 273)^4\right) + \frac{1.10x10^8 (0.5)^{0.6}}{0.95x(0.1)^{0.4}}(26.40 - 26.60)\right]^{1/4} \\ &= 26.03 \quad ^{\circ}C \end{aligned}$$

D.2 Calculation of PMV and PPD

Sample calculation on 20 September, 2006 at 21.00 PM

1. Measurement Data

T_i = 26.6 °C
RH_{in} = 80.13 %
T_r = 26.03 °C
V_a = 0.5 m/s
p_v = 2.792 kPa (from psychrometric chart)

2. Occupant parameter

M = 1.2 Met, (metabolic rate, 1 Met = 58.2 W/m^2)
= 1.2 x 58.2 W/m^2
= 69.84 W/m^2
WR = 5.5-1.5(Met-0.8)
= 5.5-1.5(1.2-0.8)
= -0.5 W/m^2
f_{cl} = 1.0 + 0.2 Icl for Icl < 0.5 clo
= 1.05 + 0.1 Icl for Icl > 0.5clo
= 1.0 + 0.2(0.5)
= 1.1
R_{cl} = 0.155 I_{cl}
= 0.155 x 0.5
= 0.0775

3. Solving for T_{cl} (clothing temperature)

In this calculation, the iteration was done by assumption of the values of h_c, T_{cl1} and T_{cl2}

Iteration #1
Assumption
h_c = 0.5 $W/m^2\,K$
T_{cl1} = 30 °C

$$T_{cl1,new} = 35.7-0.0275(M-WR)-R_{cl}\{3.96 \times 10^{-9} f_{cl}[(T_{cl}+273.15)^4 - (T_r+273.15)^4] + f_{cl}h_c(T_{cl}-T_i)\}$$

$$= 35.7-0.0275(69.84-(-0.5))-0.0775 \times \{3.96 \times 10^{-9} \times 1.1[(30+273.15)^4 - (26.03+273.15)^4] + 1.1x(0.5)(30-26.6)\}$$

$$= 30.8519 \text{ °C}$$

$$\Delta T_{cl1} = T_{cl1} - T_{cl1,new}$$
$$= 30 - 30.8519$$
$$= -0.8519 \text{ °C}$$

T_{cl2} = 29 °C

$T_{cl2,\ new}$ = 35.7-0.0275 (69.84-(-0.5))-0.0775 x {3.96 x 10^{-9} x 1.1[(29 + 273.15)^4 – (26.03 + 273.15)^4] + 1.1x(0.5)(29 – 26.6)}
= 31.6525 °C

ΔT_{cl2} = $T_{cl2} - T_{cl2,new}$
= 29-31.6525
= -2.6525 °C

Iteration #2

Substitute T_{cl1} = 30 °C, ΔT_{cl1} = -0.8519 °C and T_{cl2} = 29 °C, ΔT_{cl2} = -2.6525 °C in the following equation.

T_{cl3} = T_{cl2}-(ΔT_{cl2}[T_{cl2}-T_{cl1}]/ (ΔT_{cl2}- ΔT_{cl1}))
= 29-(-2.6525 x [29-30]/ ((-2.6525)-(-0.8519)))
= 30.4731

ΔT_{cl3} = T_{cl3}-[35.7-0.0275 (M-WR)-R_{cl} {3.96 x 10^{-9} f_{cl}[(T_{cl} + 273.15)^4 – (T_r + 273.15)^4] + $f_{cl}h_c$($T_{cl} - T_i$)}]
= 30.4731-[35.7-0.0275 (69.84-(-0.5))-0.0775 x {3.96 x 10^{-9} x 1.1[(30.4731 + 273.15)^4 – (26.03 + 273.15)^4] + 1.1x (0.5) x (30.4731 – 26.6)}]
= 1.17543581

h_c = 2.38 $(T_{cl} - T_i)^{0.25}$ for 2.38 $(T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$
= $12.1\sqrt{V_a}$ for 2.38 $(T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$.

2.38 $(T_{cl3} - T_i)^{0.25}$ = 2.38 $(30.4731 – 26.6)^{0.25}$
= -5.2171

h_c = $12.1\sqrt{0.5}$
= 8.5559

Iteration #3

T_{cl4} = T_{cl3}-(ΔT_{cl3}[T_{cl3}-T_{cl2}]/(ΔT_{cl3}-ΔT_{cl2}))
= 30.0208

ΔT_{cl4} = 30.0208-[35.7-0.0275 (69.84-(-0.5))-0.0775 x {3.96 x 10^{-9} x 1.1[(30.0208 + 273.15)^4 – (26.03 + 273.15)^4] + 1.1x (0.5) x (30.0208 – 26.6)}]
= 0.2225

h_c = 2.38 $(T_{cl} - T_a)^{0.25}$ for 2.38 $(T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$
= $12.1\sqrt{V_a}$ for 2.38 $(T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$.

2.38 $(T_{cl4} - T_i)^{0.25}$ -$12.1\sqrt{v}$ = 2.38 $(30.0208 – 26.6)^{0.25}$ -$12.1\sqrt{0.5}$

$= -5.3192$

$h_c = 12.1\sqrt{0.5}$
$= 8.5559$

Iteration #4

$T_{cl5} = T_{cl4}-(\Delta T_{cl4}[T_{cl4}-T_{cl3}]/(\Delta T_{cl4}-\Delta T_{cl3}))$
$= 29.9151$

$\Delta T_{cl5} = 29.9151-[35.7-0.0275\ (69.84-(-0.5))-0.0775 \times \{3.96 \times 10^{-9} \times 1.1[(29.9151 + 273.15)^4 - (26.03 + 273.15)^4] + 1.1\text{x}\ (0.5) \times (29.9151 - 26.6)\}]$
$= 0.00011$

$h_c = 2.38\ (T_{cl} - T_i)^{0.25}$ for $2.38\ (T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$
$= 12.1\sqrt{V_a}$ for $2.38\ (T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$.

$2.38\ (T_{cl} - T_i)^{0.25} - 12.1\sqrt{V_a} = 2.38\ (29.9151 - 26.6)^{0.25} - 12.1\sqrt{0.5}$
$= -5.3445$

$h_c = 12.1\sqrt{0.5}$
$= 8.5559$

Iteration #5

$T_{cl6} = T_{cl5}-(\Delta T_{cl5}[T_{cl5}-T_{cl4}]/(\Delta T_{cl5}-\Delta T_{cl4}))$
$= 29.9151$

$\Delta T_{cl6} = 29.9151-[35.7-0.0275\ (69.84-(-0.5))-0.0775 \times \{3.96 \times 10^{-9} \times 1.1[(29.9151 + 273.15)^4 - (26.03 + 273.15)^4] + 1.1\text{x}\ (0.5) \times (29.9151 - 26.6)\}]$
$= 1.03 \times 10^{-8}$

$h_c = 2.38\ (T_{cl} - T_i)^{0.25}$ for $2.38\ (T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$
$= 12.1\sqrt{V_a}$ for $2.38\ (T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$.

$2.38\ (T_{cl} - T_i)^{0.25} - 12.1\sqrt{V_a} = 2.38\ (29.9151 - 26.6)^{0.25} - 12.1\sqrt{0.5}$
$= -5.3445$

$h_c = 12.1\sqrt{0.5}$
$= 8.5559$

Iteration #6

$T_{cl7} = T_{cl6}-(\Delta T_{cl6}[T_{cl6}-T_{cl5}]/(\Delta T_{cl6}-\Delta T_{cl5}))$
$= 29.9151$

$\Delta T_{cl7} = 29.9151-[35.7-0.0275\ (69.84-(-0.5))-0.0775 \times \{3.96 \times 10^{-9} \times 1.1[(29.9151 + 273.15)^4 - (26.03 + 273.15)^4] + 1.1\text{x}\ (0.5) \times (29.9151 - 26.6)\}]$
$= 2.49 \times 10^{-14}$

$h_c = 2.38\ (T_{cl} - T_i)^{0.25}$ for $2.38\ (T_{cl} - T_i)^{0.25} > 12.1\sqrt{V_a}$

$= 12.1\sqrt{V_a}$ for $2.38\ (T_{cl} - T_i)^{0.25} < 12.1\sqrt{V_a}$.

$2.38\ (T_{cl} - T_i)^{0.25} - 12.1\sqrt{V_a} = 2.38\ (29.9151 - 26.6)^{0.25} - 12.1\sqrt{0.5}$

$= -5.3445$

$h_c = 12.1\sqrt{0.5}$

$= 8.5559$

4. Calculation the imbalance between (M-WR) and the rate of dissipation

$$L = (M\text{-}WR) - 13.96 \times 10^{-8} f_{cl}[(T_{cl}+273.15)^4 - (T_r+2.73.15)^4]$$
$$- f_{cl}\ h_c\ (T_{cl}-T_i) - 3.05\ [5.73 - 0.007(M\text{-}WR) - p_v]$$
$$- 0.42\ [(M\text{-}WR)-58.15] - 0.0173\ M\ (5.87-p_v)$$
$$- 0.0014\ M\ (34-T_i).$$
$$= (69.84-(-0.51)) - 13.96 \times 10^{-8} \times 1.1[(29.9151+273.15)^4 -$$
$$(26.03+2.73.15)^4] - 1.1 \times 8.559\ (29.9151-26.6) - 3.05\ [5.73 -$$
$$0.007(69.84-(-0.51)) - 2.792] - 0.42\ [(69.84-(-0.51))-58.15] - 0.0173 \times$$
$$69.84\ (5.87-2.792) - 0.0014 \times 69.84\ (34-26.6).$$
$$= 3.6342$$

5.Solving for PMV and PPD

$$PMV = [0.303 \exp(-0.036M+0.028)]L$$
$$= [0.303 \exp((-0.036 \times 69.84)+0.028)] \times 3.6342$$
$$= 0.19087$$
$$PPD = 100-95 \exp[-(0.03353PMV^4 + 0.2179\ PMV^2)]$$
$$= 100-95 \exp[-(0.03353(0.19087)^4 + 0.2179(0.19087)^2)]$$
$$= 5.7553$$