AN EXPERIMENTAL AND NUMERICAL STUDY ON IMPROVING THERMAL COMFORT AND ENERGY SAVING MEASURES IN A SLEEPING ENVIRONMENT

DU JING

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An Experimental and Numerical Study on Improving Thermal Comfort and Energy Saving Measures in a Sleeping Environment

Du Jing

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Certificate of Originality

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Du Jing

Abstract

Sleep plays an important role in people's daily life, since human beings spend approximately one third of their life time in sleep and can benefit much from sleep. However, sleeping quality can be affected by a number of factors. Among those, thermal environment and indoor air quality in a sleeping environment are generally provided and maintained by building air conditioning (A/C) systems. As a result of the increased people's living standard and long-lasting hot and humid summers in subtropics, more and more people tend to use air conditioning in their bedrooms during sleep. This, therefore, results in a significant increase in building energy consumption.

Due to the excellent performances in achieving thermal comfort and indoor air quality control and energy saving, task\ambient air conditioning (TAC) can be best applied to sleeping environments, as a sleeping person occupies a small space and is immobile during sleep. However, all previous TAC systems developed and applied to sleeping environments predominantly were convection-based to cool an occupied zone in a sleeping environment and the vent supplying outdoor air at a high velocity and a low temperature had to be close to a sleeping person, resulting in a severe cold draft problem. It is commonly acknowledged that draft can not only cause thermal discomfort, but also result in temporary illness or even chronic diseases for a sleeping person. To deal with the draft problem encountered in previous convection-based TAC (C-TAC) systems applied to sleeping environments, a new non-convection based TAC systems should be developed.

On the other hand, energy for air conditioning a bedroom during sleep periods can also be saved by reducing the nighttime cooling load from external envelopes. The cooling load in a bedroom at nighttime mainly comes from occupants and bedroom envelopes, with the latter being responsible for about three quarters of the total nighttime cooling load. Furthermore, since a bedroom is usually not air conditioned at daytime, a large amount of thermal energy can be stored in the thermal mass in its envelope components. At nighttime, the energy stored in the envelopes would be gradually released, contributing significantly to the total nighttime cooling load in the bedroom, in particularly, during the first two hours of A/C operation.

Therefore, in this Thesis, a systematic research program to develop a non-convection based TAC system applied to a sleeping environment and to evaluate its performance in terms of thermal comfort, indoor air quality and energy efficiency, and to investigate the impacts of daytime heat gain/storage of the external envelopes in a bedroom on its nighttime cooling load and the related mitigation measures, is reported. The Thesis, first of all, begins with presenting an experimental study, as the first part of the systematic research program, on the thermal comfort and ventilation performances of using a radiation-based TAC (R-TAC) system applied to sleeping environments. The study results demonstrated that using the R-TAC system could not only result in better thermal comfort and ventilation performances, but also effectively resolve the cold draft problem as compared with using the previous C-TAC systems. In addition, the potential problem of condensation on the surface of a cold radiant panel may be resolved by raising the panel surface temperature, or using a lower fresh air supply humidity.

Secondly, this Thesis reports a numerical study on the effects of design/operating parameters of the radiant panel in an R-TAC system on indoor thermal comfort and energy performances. In this study, four panel surface temperatures, three different distances between the panel and a bed, three emissivities and two panel areas were used to numerically investigate their effects on indoor thermal comfort and energy performances. A CFD method was developed and validated using the measured data obtained in the first part of the systematic research program. With the validated CFD method, indoor thermal comfort and energy utilization efficiency were evaluated in a simulated bedroom, and temperature and air flow fields visualized to analyze the differences in thermal comfort and energy performances when using an R-TAC system with different design/operating parameters. The study results demonstrated: 1) increasing surface temperature can lead to a higher PMV value and a higher EUC value; 2) increasing surface emissivity and area of the radiant panel can result in a lower PMV value and a lower EUC value; and 3) reducing the distance between the bed and the panel can give a lower PMV value and a higher EUC value.

Finally, the impacts of daytime heat gain/storage in the external envelopes in a bedroom on the nighttime cooling load and the related mitigation were numerically studied, and the study results are presented in this Thesis, as the third part of the systematic research program. Firstly, the cooling load characteristics in the bedroom at nighttime A/C operation, with an emphasis on analyzing the impacts of thermal energy gain and storage in the west-facing external wall of the bedroom at daytime on the resultant nighttime cooling load, were studied. Secondly, the impacts of adding an air gap and ventilating the gap in the west-facing external wall on the total nighttime cooling load were evaluated. The simulation results demonstrated that the nighttime cooling load contributed by the heat flows through the west-facing external wall was the highest among all the envelopes, due to its direct exposure to solar radiation, thus the highest heat gain and storage in its thermal mass. Adding an air gap in the west-facing external wall can help reduce remarkably the hourly total nighttime cooling load for the first hour nighttime A/C operating (21:00-22:00), and the total cooling load for a 10-hour A/C period. Furthermore, ventilating the air gap and adhering aluminum foil can help further reduce the nighttime cooling load.

Publications Arising from the Thesis

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Nomenclature

Variable	**Description**	**Unit**
C	CO_2 concentration at the measurement position	ppm
COP	Coefficient of Performance	—
C_r	CO_2 concentration in return air	ppm
C_s	CO_2 concentration in supply air	ppm
D	Globe diameter	mm
D	Distance between the sleeping person and the radiant panel	mm
DR	Draft risk	%
E	Total energy	J
EUC	Energy utilization coefficient	
$\vec{g}$	Gravitational acceleration	m/s^2
h	Enthalpy	J
h_c	Convective heat transfer coefficient	$W/(m^2\ {}^oC)$
h_r	Radiative heat transfer coefficient	$W/(m^2\ {}^oC)$
k_{eff}	Effective conductivity	$W/(m\cdot{}^oC)$
L	The length of the radiant panel	mm
P	Pressure	Pa
p_a	Water vapor	kPa
$PCLI$	Part cooling load index	%
PEE	Potential energy efficiency index	—

PMV	Predicted mean vote	—
Q_{fan}	Electricity consumption of the ventilation fan	W/m^2
$Q_{reduction}$	The total nighttime cooling load reduction during nighttime A/C from 21:00-07:00 in the following day (W/m^2) compared with that in Case 10	W/m^2
$QR_{external}$	Radiative heat gain by the internal surface of the west-facing external wall	W
QR_{others}	Sum of radiative heat gains by other indoor thermal masses	W
QR_{window}	Radiative heat gain by the internal surface of the west-facing window	W
Q_s	Supply air flow rate	L/s
QT_{others}	Sum of total thermal energy gains by other indoor thermal masses	W
RR	Reduction rate of *SUM* compared with that in the baseline case (Case 1)	%
R_t	Total resistance of a bedding system	m^2 ^{o}C /W
SUM	Sum of the hourly total nighttime cooling load from 21:00-07:00 in the following day	W/m^2
SUM_{casej}	The SUM value in Case j (j=1, 2⋯12)	W/m^2
t	Time	s
t_a	Air temperature	^{o}C
$t_{a,OZ}$	The averaged air temperature in the occupied zone	^{o}C

$t_{a,UZ}$	The averaged air temperature in the unoccupied zone	^{o}C
t_g	Globe temperature	^{o}C
$TNCL_i$	The hourly total nighttime cooling load with its subscript *i* indicating the order of the A/C operation hour	W/m^2
t_o	Operative temperature	^{o}C
$t_{o,OZ}$	The averaged operative temperature in the occupied zone	^{o}C
$t_{o,UZ}$	The averaged operative temperature in the unoccupied zone	^{o}C
t_r	Mean radiant temperature	^{o}C
t_{rp}	Surface temperature of the radiant panel	^{o}C
t_s	Supply air temperature	^{o}C
T_u	Turbulence intensity	%
v	Air velocity	m/s
$\vec{v}$	Air velocity	m/s
VE	Ventilation effectiveness	—
W	The width of the radiant panel	mm
T_u	Turbulence intensity	%
ε	Globe emissivity	—
ρ	Air density	kg/m^3
$\bar{\bar{\tau}}$	Stress tensor	Pa

List of Abbreviations

A/C	Air conditioning
CFD	Computational fluid dynamics
CRCP	Ceiling radiant cooling panel
C-TAC	Convection-based task\ambient air conditioning
DO	Discrete ordinates
DX	Direct expansion
FAC	Full-volume air conditioning
HVAC	Heating, ventilation and air conditioning
IAQ	Indoor air quality
MRT	Mean radiant temperature
OZ	Occupied zone
RACs	Room air conditioners
RCP	Radiant cooling panel
R-TAC	Radiation-based task\ambient air conditioning
S2S	Surface-to-surface
TAC	Task\ambient air conditioning
UZ	Unoccupied zone
WRAC	Window-type room air conditioner

Chapter 1

Introduction

Sleep is an essential part of people's daily life. Not only does sleep occupy approximately one third of the life time of a human being, but also its quality may significantly affect a human being's physical strength, ability to think, work efficiency, or even state of mind at daytime. It is commonly acknowledged that sleeping quality is mainly affected by two factors: personal and environmental. The former refers to the mental or physical status of a sleeping person, and the latter the environmental factors in a bedroom. As far as the environmental factors are concerned, apart from lighting and noise level, the thermal environment and indoor air quality (IAQ) in buildings are generally provided and maintained by building air conditioning (A/C) installations. However, the increased use of A/C systems for maintaining a thermally comfortable sleeping environment leads to increased building energy consumption. Hence, to decrease building A/C energy consumption, it is necessary to develop new measures to improve energy utilization efficiency in an air conditioned sleeping environment and to reduce nighttime cooling load in a bedroom.

Since a sleeping person occupies a small space and is immobile during sleep, it is ideal to apply task\ambient air conditioning (TAC) technology to a sleep environment because of its excellent performances in providing thermal comfort control and improving indoor air quality in a specific localized space, and in reducing energy consumption, as demonstrated in previous related studies. Nonetheless, all previous TAC systems developed and applied to sleeping environments were predominantly convection-based to cool an occupied zone in a sleeping environment and the vent supplying outdoor air at a high velocity and a low temperature had to be close to a sleeping person, resulting in a severe cold draft problem. It is commonly acknowledged that draft can not only cause thermal discomfort, but also result in temporary illness or even chronic diseases for a sleeping person. To deal with the cold draft problem encountered in previous convection-based TAC (C-TAC) systems applied to sleeping environments, a new non-convection based TAC systems should be developed.

In addition, energy for air conditioning a bedroom during sleep can also be saved by reducing the nighttime cooling load from external envelopes. The cooling load in a bedroom at nighttime mainly comes from occupants and bedroom envelopes, with the latter being responsible for about three quarters of the total nighttime cooling load as illustrated in previous related studies. Furthermore, since a bedroom is usually not air conditioned at daytime, a large amount of thermal energy can be stored in the thermal mass in its envelope components. At nighttime, the energy stored in the envelopes would be gradually released, contributing significantly to the total nighttime cooling load in the bedroom, in particularly, during the first two hours of A/C operation.

Therefore, it becomes necessary to develop a non-convection based TAC system applied to a sleeping environment and to evaluate its performance in terms of thermal comfort, indoor air quality and energy saving, and to investigate the impacts of daytime heat gain/storage of the external envelopes in a bedroom on its nighttime cooling load and the related mitigation measures. A systematic research program is therefore carried out and its outcomes are presented in this Thesis.

The Thesis begins with an extensive literature review in Chapter 2. Firstly, the fundamentals of sleep and the environmental factors that influence sleep quality are reviewed. This is followed by presenting a review on the previous studies on the thermal comfort and IAQ in a sleep environment, so as to have a better understanding on the impacts of environmental factors on the sleeping quality. Thirdly, a review on the cooling load characteristics and the energy use for air conditioning in residential buildings in the sub-tropics is reported. Fourthly, the types, operating parameters of TAC systems and the features, and problems of the previously developed C-TAC systems applied to sleeping environments are reviewed. Fifthly, a brief review on radiant cooling panel systems and their operating performances related to thermal comfort and energy consumption is included. This is followed by reviewing the previous numerical studies on indoor thermal environment and IAQ. Finally, existing passive measures applied to building envelopes to reduce heat gain for building energy saving are reviewed.

Chapter 3 presents the title, aims and objectives of the systematic research program, and the research methodologies used.

Chapter 4 reports the first part of the systematic research program: an experimental study on the thermal comfort and ventilation performances of using a radiation-based TAC system applied to sleeping environments. The study results demonstrated that using the R-TAC system could not only result in better thermal comfort and ventilation performances, but also effectively resolve the cold draft problem as compared with using the previous C-TAC systems. In addition, the potential problem of condensation on the surface of a cold radiant panel may be resolved by raising the panel surface temperature, or using a lower fresh air supply humidity.

Chapter 5 presents a numerical study on the effects of design/operating parameters of the radiant panel in an R-TAC system on indoor thermal comfort and energy performances, as the second part of the systematic research program. In this study, four panel surface temperatures, three different distances between the panel and a bed, three emissivities and two panel areas were used to numerically investigate their effects on indoor thermal comfort and energy performances. A Computational Fluid Dynamics (CFD) method was developed and validated using the measured data obtained in the first part of the systematic research program. With the validated CFD method, indoor thermal comfort and energy utilization efficiency were evaluated in a simulated bedroom, and temperature and air flow fields visualized to analyze the differences in thermal comfort and energy performances when using an R-TAC system with different design/operating parameters. The study results demonstrated: 1) increasing surface temperature can lead to a higher PMV value and a higher EUC value; 2) increasing surface emissivity and area of the radiant panel can result in a lower PMV value and a lower EUC value; and 3) reducing the distance between the bed and the panel can give a lower PMV value and a higher EUC value.

Chapter 6 reports a simulation study on the impacts of daytime heat gain/storage in the external envelopes in a bedroom on the nighttime cooling load and the related mitigation measures, as the third part of the systematic research program. Firstly, the cooling load characteristics in the bedroom at nighttime A/C operation, with an emphasis on analyzing the impacts of thermal energy gain and storage in the west-facing external wall of the bedroom at daytime on the resultant nighttime cooling load, were studied. Secondly, the impacts of adding an air gap and ventilating the gap in the west-facing external wall on the total nighttime cooling load were evaluated. The simulation results demonstrated that the nighttime cooling load contributed by the heat flows through the west-facing external wall was the highest among all the envelopes, due to its direct exposure to solar radiation, thus the highest heat gain and storage in its thermal mass. Adding an air gap in the west-facing external wall can help reduce remarkably the hourly total nighttime cooling load for the first hour nighttime A/C operating (21:00-22:00), and the total cooling load for a 10-hour A/C period. Furthermore, ventilating the air gap and adhering aluminum foil can help further reduce the nighttime cooling load.

In the final Chapter of the Thesis, the conclusions and the proposed future work are given.

Chapter 2

Literature review

2.1. Introduction

Sleeping quality can have great impacts on the mental and physical health, and work productivity of human beings. On average, approximately one third of a human's life time is spent in sleep, as revealed in a questionnaire survey that the averaged sleep duration of American people is 6.9 hours and 7.5 hours per night for weekdays and weekends, respectively (Sleep in America Poll 2014). Furthermore, the quality of sleep is also very important (Hobson 1995) in improving the performance in daytime activities and helping people recover from tiredness (Engle-Friedman et al. 2003). Therefore, how to improve sleeping quality is becoming an important issue and attracting an increased research attention. For decades, many factors affecting the sleeping quality have been investigated by medical researchers (Cuellar et al. 2006; Kemal Sayar et al. 2002). It has been commonly acknowledged that sleeping quality was mainly affected by the mental-physical factors of a sleeping person and the environmental factors in a bedroom. Among all the environmental factors, except lighting and noise level, the thermal environment and indoor air quality in a bedroom are provided and maintained by building heating, ventilation and air conditioning (HVAC) systems. It has been found that different levels of indoor thermal environment may have significant impacts on the number of awakenings and sleep duration. Furthermore, a CO_2 concentration level higher than 5000 ppm may lead to mental depression, increased heart rate, unconsciousness and even death to human beings (Rice 2014; Priestly 2003).

With a great improvement in people's living standard and increased demand for better indoor thermal comfort and IAQ, A/C systems are increasingly used and become an essential provision in people's daily life. For the past decades, to achieve improved indoor thermal comfort control and better IAQ, different types of novel A/C systems have been developed, such as dedicated outdoor air systems (DOAS), floor/ceiling radiant cooling panel (FRCP/CRCP) systems integrated with mixed ventilation (MV) or displacement ventilation (DV), under-floor air distribution (UFAD) systems, personalized ventilation systems (PV) and task\ambient air conditioning (TAC) systems, etc.

On the other hand, during summer months in tropics and subtropics, the weather can be very hot and humid. Therefore, air conditioning is widely used not only in office buildings at daytime, but also in residential buildings at nighttime to maintain a suitable thermal environment for sleep, as reported in a survey on the situations of using room air conditioners (RACs) and the thermal environments during sleep in bedrooms in residential buildings in Hong Kong (Lin and Deng 2006). It was reported that about two thirds of the respondents kept their RACs on at nighttime during sleep.

Therefore, due to the increased use of A/C in residential buildings at nighttime, residential energy consumption for air conditioning is significantly increased in residential buildings. For instance, in Hong Kong, in 1971 residential A/C used 155 GWh of electricity, or 14.6% of the total residential energy use. In 1996, the consumption was increased to 2467 GWh, or 30.4% of the total (Lam 2000), and in 2011, further to 4983 GWh, or 45% of the total (Hong Kong Energy Statistics Annual Report 2012). Therefore, in order to decrease A/C energy use while still maintaining an acceptable level of human thermal comfort and IAQ, it is necessary to develop new measures to energy-efficiently air conditioning a sleeping environment and to reduce nighttime cooling load in a bedroom.

Task\ambient air conditioning (TAC) is one of novel A/C systems, originally for daytime use. It is defined as a space conditioning system that allows thermal conditions in small, localized zones to be individually controlled by occupants (Bauman and Arens 1996). Over the years, TAC has attracted an increasing research attention and has been considered as one of several emerging energy-saving technologies in a DOE report (Deshmukh et al. 2004). In related earlier studies, the operating performances of TAC systems applied to office buildings at daytime have been extensively investigated (Amai et al. 2007; Bauman et al. 1993b; Faulkner et al. 1995; Pan et al. 2005) and the advantages of applying TAC have been identified, such as better thermal comfort control, improved IAQ and reduced energy consumption, etc. TAC systems can be used to individually control the local thermal environment. Hence, the use of a TAC system can lead to a higher energy utilization efficiency and better occupant thermal comfort control in a small zone than the use of a full-volume air conditioning (FAC) system. Furthermore, IAQ can be improved in an occupied zone when using a TAC system, as the vent to supply fresh air is installed inside the occupied zone, leading to a higher contaminant removal efficiency.

Due to the immobility of, and a relatively small space occupied by a sleeper for a sleeping environment, the use of TAC appears more effective than the use of other types of air conditioning. However, the reported studies on applying TAC to sleeping environments are limited. Pan et al. (2012b) proposed a bed-based TAC system with flexible air ducts and plenums and it was shown that the use of the bed-based TAC system can help greatly reduce energy consumption for air conditioning a sleeping environment in real buildings. Mao et al. (2013; 2014b; 2014a) improved the bed-based TAC system by removing bulky air ducts and plenums. The study results revealed that using this ductless TAC system can efficiently reduce indoor contamination and energy consumption as compared to the use of a full volume air conditioning system. However, all of the previous TAC systems developed for sleeping environments were predominantly based on convection to cool the air in an occupied space, thus the vents supplying air at a high velocity and a low temperature would have to be close to a sleeping person. Therefore, a convection-based TAC system for a sleeping environment can lead to serious cold-draft problems, which is harmful to human health. For example, as reported by Mao et al. (2013), among 12 measurement positions in an occupied space when using the ductless TAC system, the values of draft risk in 8 of the 12 positions were higher than 20% and even that in 4 out of the 8 positions reached to 40%, which were far higher than the permissible value of 20% as suggested by ASHRAE Standard 55 (ASHRAE 2013d). Therefore, to avoid cold drafts, non-convection based TAC systems applied to a sleeping environment have to be developed.

In addition, to reduce the energy use for building air conditioning, it is possible to reduce space cooling load/heat gain through optimizing the structure and composition of building envelopes. For a nighttime air conditioned bedroom, its cooling load mainly comes from occupants and envelopes. As reported in a previous study (Lin and Deng 2004), the cooling load in a bedroom at nighttime from building envelopes dominated the total cooling load at 75%. Among all the bedroom envelopes, an external envelope(s) contributes predominately to the total nighttime cooling load due to temperature difference between outdoor air and indoor air, and the heat storage in the envelope. A bedroom is usually not air conditioned at daytime, which results in a large amount of thermal energy stored in the thermal mass in its envelope components, in particularly its external envelopes due to direct solar exposure. At nighttime, the energy stored in the envelopes would be gradually released, contributing significantly to the total nighttime cooling load in the bedroom. It was shown by Lin and Deng (2004) that the cooling load from envelopes in the first two hours of a nighttime A/C process was approximately 50% more than that in the later part of the process. Therefore, it is necessary to optimize the structure and composition of external envelopes for reducing the thermal storage at daytime, thus helping decrease the nighttime cooling load.

In this Chapter, an extensive review on a number of important issues related to sleep, sleeping thermal environment, and building energy use, in the existing literature is presented. The review covers fundamentals of sleep, environmental factors affecting sleeping quality, thermal comfort in sleeping environments, energy consumption for air conditioning buildings, TAC systems and the application to sleeping environments, the use of radiant cooling panel systems and its performance in thermal comfort control and energy saving, using CFD method to predict indoor thermal environment in buildings, and finally, the passive building energy saving measures.

2.2. Sleep and the environmental factors influencing the quality of sleep

2.2.1. Sleep

Sleep is a basic necessity of human beings and the sleeping duration differs from person to person considerably (Lavie and Berris 1998). Sleep is a naturally recurring state of mind and body characterized by altered consciousness, relatively inhibited sensory activity, inhibition of nearly all voluntary muscles, and reduced interactions with surroundings. It is distinguished from wakefulness by a decreased ability to react to stimuli, but is more easily reversed than the state of being comatose (National Institutes of Health 2017). Normally, human sleep comprises two highly distinct modes known as rapid eye movement (REM) sleep and non-rapid eye movement (NREM) sleep. REM and NREM sleep are very different: the former is associated with desynchronized and fast brain waves, loss of muscle tone, and suspension of homeostasis, accounting for 75-80% of sleep time; the latter is considered to be deep sleep without prominent eye movement or muscle paralysis. There are four stages of NREM sleep, which is divided based on the pattern of brainwaves, as shown in Figure 2.1, the hypnogram for a young adult. Stage 1 is a temporary transition from full awakening to sleep, and the reverse transitions from sleep to wake or after brief body movements can temporarily occur. The second stage is regarded as the emergence of EEG (Electroencephalography) spindles, the first bona fide sleep stage, occupying more than half of sleep time. Stage 3 and Stage 4 in NREM sleep are called deep sleep, delta sleep, or slow wave sleep, which usually emerge together. During Stage 4, intermixed cerebral activity emerges, together with striated muscle atonia and rapid eye movements, which is regarded as an active form of sleep (Rechtschaffen and Kales 1968). Sleep occurs in cycles of about 90 minutes and there are usually four or five such cycles per night. The whole sleeping process proceeds in an organized and definitive order: REM→ Stage 1→ Stage 2→ Stage 3→ Stage 4→ Stage 3→ Stage 2→ Stage 1→ REM, as described in Figure 2.1, recording a hypnogram of sleep (Brebbia and Altshuler 1965; Lavie and Berris 1998).

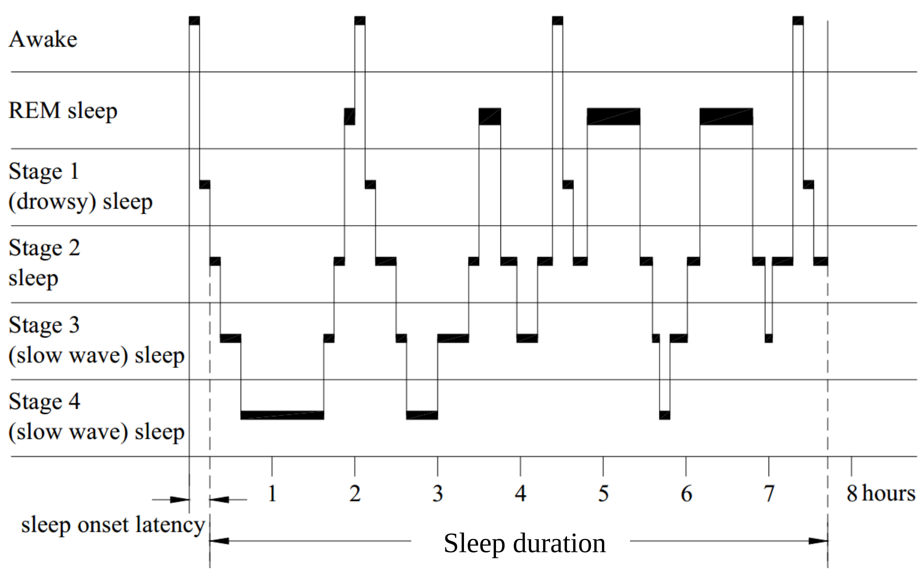


Figure 2.1 The hypnogram of a young adult (Lavie and Berris 1998)

2.2.2. Functions of sleep

Sleep has great influence on people's daily life. This is because sleep occupies about one third of the life time of a human being, as reported in a poll that the averaged sleeping time for American people is 6.9 hours and 7.5 hours every night in weekdays and weekends (Sleep in America Poll 2014). Furthermore, not only just a state of rest, sleep also has many other benefits (Hobson 1995), such as improving the performance in daytime activities and recovering from tiredness (Engle-Friedman et al. 2003).

The sleeping quality may have great effects on the physical health of human beings, for impacting the endocrine and immune systems, or even mortality and morbidity (Åkerstedt and Nilsson 2003). Sleep deficiency for a long time is fatal, which has been tested in rodents and drosophila (Cortelli et al. 1999; Shaw et al. 2002). On the other hand, sleep also has great impacts on mental health, suggesting that sleep in some degree benefits normal neuronal functions. It has been proven by strong evidences that sleep has influence on synaptic plasticity and the ability of learning (Benington and Frank 2003; Stickgold et al. 2001).

However, in recent years, there are an increased number of people suffering from insomnia. For instance, it was reported that 39.4% of Hong Kong residents were significantly troubled by insomnia from a survey study (Wong and Fielding 2011), and about 1 in 4 American workers had insomnia or at least sleep disorder problems, resulting in $63 billion in lost productivity each year (Boyles 2011). Therefore, to improve the sleeping quality, it is necessary to understand the relationships between sleeping quality and the factors influencing sleeping quality.

2.2.3. Environmental factors influencing the quality of sleep

The factors affecting the sleeping quality include the personal factors of a sleeper and the environmental factors during sleep. The latter include lighting, noise level, IAQ and thermal environment in a sleeping environment.

Among these environmental factors, light can have great influence on sleeping quality. Exposure to light may directly make people difficult to fall asleep, and indirectly have effects on the biological clock system of a human being and his/her preferred time to sleep. As reported in a previous study, in the late evening, light may delay the phase of the internal clock, leading him/her to prefer to sleep late (Gooley et al. 2010). Furthermore, in the middle of a night, exposure to light can have adverse impacts, like causing human internal clock to be reset, and make it harder to return to sleep. On the other hand, noise stimuli can be processed by the sensory functions of a sleeper, despite of a non-conscious perception of their presence (Muzet 2007; Basner et al. 2011). It has been reported that, due to noises with intermittent peak levels of not less than 45dB (A), the time duration of falling asleep for human beings can be increased by a few to 20 minutes (Öhrström 1993). The World Health Organization recommended a guideline level of 30 dB for undisturbed sleep (Organization 2009). Apart from light and noise, the effects of IAQ and the thermal environment in a sleeping environment on sleeping quality have been also gradually understood (Haskell et al. 1981b; Muzet et al. 1984; Okamoto-Mizuno et al. 1999; Tsuzuki et al. 2008), which are presented in details in the following paragraphs and Section 2.3, respectively.

In a traditionally air-conditioned room, there may exist many contaminants like formaldehyde, benzene, CO_2, ammonia, coming from building materials and internal pollution sources. However, in many circumstances, it is difficult to detect and monitor the concentrations of all kinds of contaminants in buildings. Therefore, due to the convenience and low cost of detecting, CO_2 concentration has often been used as a representative contaminant in evaluating indoor air quality and ventilation efficiency (ASHRAE 2013e; Hui et al. 2007; Persily 1997). Many studies on indoor CO_2 level have been carried out, including the effects of elevated CO_2 concentrations on human health and determining a reasonable safe level of CO_2 concentration.

Previous studies indicated that the high level of CO_2 concentration may have impacts on normal sleep. For example, Schaefer (1962) conducted a subjective study by exposing to an indoor environment containing 3% CO_2 for 3-6 days. For the first 24 hours, the ability of falling asleep was decreased by general excitement and agitation. Along with the increase of alveoli PCO_2 (partial pressure of CO_2), awakening from sleep finally happened because of the hypercapnic distress. Hedemark and Kronenberg (1982) made an investigation on awakenings caused by CO_2 rebreathing during sleep. The results showed that the partial pressure of CO_2 in alveoli (P_ACO_2) at the range of 49.2 ± 0.4 torr led to wakefulness for all subjects. This was in agreement with the results obtained by Berthon-Jones and Sullivan (1984) who pointed out that awakenings occurred with an increase in P_ACO_2, from 58.6 ± 0.9 torr to 63.8 ± 0.8 torr, as the corresponding sleep pattern changed from Stage 2 to Stage 4 for male sleepers. In summary, exposure to a relatively high CO_2

concentration in sleeping environments would result in awakenings which indirectly affects sleeping quality.

Apart from causing arousal, CO_2 also has negative health effects if too much of it is inhaled. Inhalation of CO_2 may result in an increased heart rate and breathing, sweating, headache, dizziness, lack of breath, shaking, mental depression, blurred vision, unconsciousness and even death. For example, exposure to the air containing 2% CO_2 for several hours may cause exertion dyspnea and headache (Schulte 1964); Exposed to 7-10% CO_2, people may become unconscious or half-unconscious in several minutes (CAT 1953; Dripps and COMROE Jr 1947); When exposed to CO_2 concentrations between 17% and 30% within 1 minute, people would have the symptoms of coma and convulsions, lose the ability of self-control and consciousness and even life (Dalgaard et al. 1972; CAT 1953; Lambertsen 1971).

Since a high level of CO_2 concentration can have negative influence on sleeping quality, or even lead to physical and mental health problems as mentioned earlier, determining safe and reasonable levels of CO_2 concentration was studied. It was previously investigated by Seppänen et al. (1999) that, in office buildings, CO_2 concentrations generally ranged from 350 ppm to 2500 ppm. It was also pointed out by ASHARE that the approximate steady-state indoor CO_2 concentration should be 870 ppm, assuming that indoor CO_2 dissipation rate from an averaged occupant is 0.31 L/min and outdoor CO_2 concentration 350 ppm (ASHRAE 2013e). Based on the recommendation of ACGIH (1999) and OSHA (1978), the safe CO_2 concentration level for an indoor environment should be below 5000 ppm.

2.3. Thermal comfort in sleeping environments

Thermal comfort can be defined as a condition of mind which expresses satisfaction to the thermal environment (ASHRAE 2013d). However, the meanings of “condition of mind” and “satisfaction” are not specified in this definition. This is because that there are many variations, physiologically and psychologically, from person to person, hence, it is difficult to satisfy everyone in a space. Strictly speaking, the term “thermal comfort” does not make too much sense for people during sleep. Basically, comfort was not a state of condition, but rather a state of mind. Therefore, to understand the relationship between the quality of a sleeping for a human being and his / her sleeping thermal environment, in medical or other related subject areas, a ‘thermal comfort zone’ was defined as: a range of ambient temperature around the thermal neutral temperature, within which the quantitative measures of sleep such as sleep stage latencies, time spent in each sleep stage, number and duration of awakening were only slightly modified (Muzet et al. 1984).

2.3.1. Thermal parameters in sleeping environments

Among the four environmental parameters, indoor air temperature can be regarded as the parameter people are most concerned about. Over the past years, the effects of high and low ambient air temperatures on sleep stages for a sleeping person have been experimentally investigated. Kendel and Schmidt-Kessen (1973) found out that a higher temperature may increase the duration and number of awakenings during sleep for a human being. Karacan et al. (1978) demonstrated that the total time of sleep was reduced when the air temperature inside a blanket was heated to 39 ^{o}C using a circulating fluid at a temperature of 49 ^{o}C. H´enane et al. (1977) showed that ambient air temperature between 35 ^{o}C and 39 ^{o}C would not significantly change sleep stage for a sleeper, but awakening time was increased. Parmeggiani et al. (2005) reported that sleeping in an environment with high air temperature reduced REM sleep occurrence times for a sleeper, but the averaged length and duration of REM sleeping intervals did not change. For the effects of low air temperature on a sleeping person, Buguet et al. (1979) carried out a study in the Arctic winter and the results demonstrated that the sleeping time was reduced and sleepers were easy to be interrupted due to awakenings and body movements. Haskell et al. (1981a) conducted a comparison on the impacts of low (21 ^{o}C) and high (34-37 ^{o}C) ambient air temperatures on sleeping quality and it was shown that, in cold environments, the duration of wakefulness of Stage 1 was increased but that of Stage 2 sleep decreased. In summary, a low ambient temperature makes sleepers easier to wake up.

Besides indoor air temperature, other thermal environmental factors like mean radiant temperature (MRT), air velocity and air humidity would also have influence on sleeping quality. Kim et al. (2010) investigated the relationship between sleeping quality and mean radiant temperature, and used % FL (number of flow-limited breaths without snoring) as an index for measuring sleeping quality. The results showed that the best MRT range for good sleep quality depended on seasons, at 19.8–27.08 ^{o}C, 23.4–26.98 ^{o}C and 27.2–30.38 ^{o}C for winter, spring and summer, respectively. Tsuzuki et al. (2008) conducted a subjective study on the effects of air velocity on sleeping quality, by asking 17 males with half-slip sleepwear covered with a cotton blanket to separately sleep under different testing conditions. The results indicated that, with a proper level of air velocity maintained in a sleeping environment, the duration of awakenings can be reduced by lowering skin and rectal temperatures, and furthermore the body mass loss if under a hot and humid condition. Okamoto-Mizuno et al. (1999) demonstrated that exposure to a very hot and humid environment for a sleeper not only increased the thermal load and but also suppressed a reduction in rectal temperature, resulting in an increase in awakenings.

2.3.2. Thermal comfort model and thermal neutral temperature in a sleeping environment

Over the past decades, research works on occupants' thermal comfort have been extensively carried out. For example, Fanger (1970) established the well-known thermal comfort model which has been widely used to evaluate human thermal sensation. De Dear and Brager (2002) proposed a new adaptive comfort standard (ACS) applied to naturally ventilated buildings during summer or in warmer climate zones. Gagge carried out a study on a standard index used to predict human satisfaction level with a thermal environment and discussed three rational indices of thermal comfort (Gagge et al. 1986). In addition, standards or methods of evaluating thermal comfort have been proposed (Olesen and Parsons 2002; Doherty and Arens 1988; de Dear and Brager 2002; Evin and Siekierski 2002; Zhang 2003; Höppe 2002) and field surveys (Newsham and Tiller 1997; Schiller 1990) and experiments in climate chambers (Nakano et al. 2002; Fanger 1970) on human thermal comfort have been carried out. However, most of the previous studies were carried out in the situations when people were awake.

Based on the well-known thermal comfort model developed by Fanger (1970), a modified thermal comfort model applied to sleeping environments was proposed by Lin and Deng (2008a). The model described the heat balance between a sleeping person and his/her sleeping thermal environment as:

$$40 = \frac{34.6 - t_o}{R_t} + \frac{0.06 i_m L_R (p_{sk,s} - p_a)}{R_t} + 0.056(34 - t_a) + 0.6925(5.87 - p_a) \qquad (2.1)$$

Where R_t is the total thermal resistance, affected by many factors including bedding, bed and mattress, sleepwear, air velocity and direction, sleeping posture and the percentage coverage of body surface by bed and bedding, etc (Lin and Deng 2008a).

By solving Equation (2.1), a thermal comfort equation for a sleep environment relating to one personal variable and four environmental variables was obtained:

$$\begin{aligned} 40 = \frac{1}{R_t}\left[\left(34.6 - \frac{4.7\overline{t_r} + h_c t_a}{4.7 + h_c}\right) + 0.3762\left(5.52 - p_a\right)\right] \\ +0.056(34 - t_a) + 0.6925(5.87 - p_a) \end{aligned} \qquad (2.2)$$

In Equation (2.2), there are five variables, R_t, $\overline{t_r}$, t_a, p_a and h_c. Since h_c, the convective heat transfer coefficient, is the function of air velocity, v, Equation (2.2) is similar to the thermal comfort model used at daytime where four variables (i.e., $\overline{t_r}$, t_a, p_a, and v) are thermal environmental variables determining a thermally neutral sleeping environment for a given bedding system.

Therefore, thermal comfort charts were established at various combinations of five variables mentioned above by solving Comfort Equation (2.2).

Figure 2.2 plots a series of comfort lines combining three variables: wet-bulb temperature, operative temperature (t_o) and the total insulation value for a bedding system, on which thermally neutral sleeping environment can be achieved. It was shown that the effects of relative humidity (RH) on sleeping thermal comfort for a sleeper can be ignored. Specifically, the variation in RH from 0% (absolutely dry air) to 100% (saturated air) in comfort lines can be compensated by only a 0.95-1.63 ^{o}C reduction in operative temperature (within the range of 2.4-0.8 clo total thermal insulation values). In addition, a higher total insulation value leads to a slighter reduction in t_o. For instance, a 1.5 ^{o}C decrease in t_o can be compensated by changing RH from 0% to 100% at the total insulation value of 1.0 clo, as compared to 1.1^{o}C decrease in t_o at the total insulation of 2.0 clo.

The relationships between the total thermal insulation of a bedding system and the operative temperature at different RH values is shown in Figure 2.3. It was obvious that the operative temperature was greatly affected by the total insulation value for a bedding system in a sleeping environment.

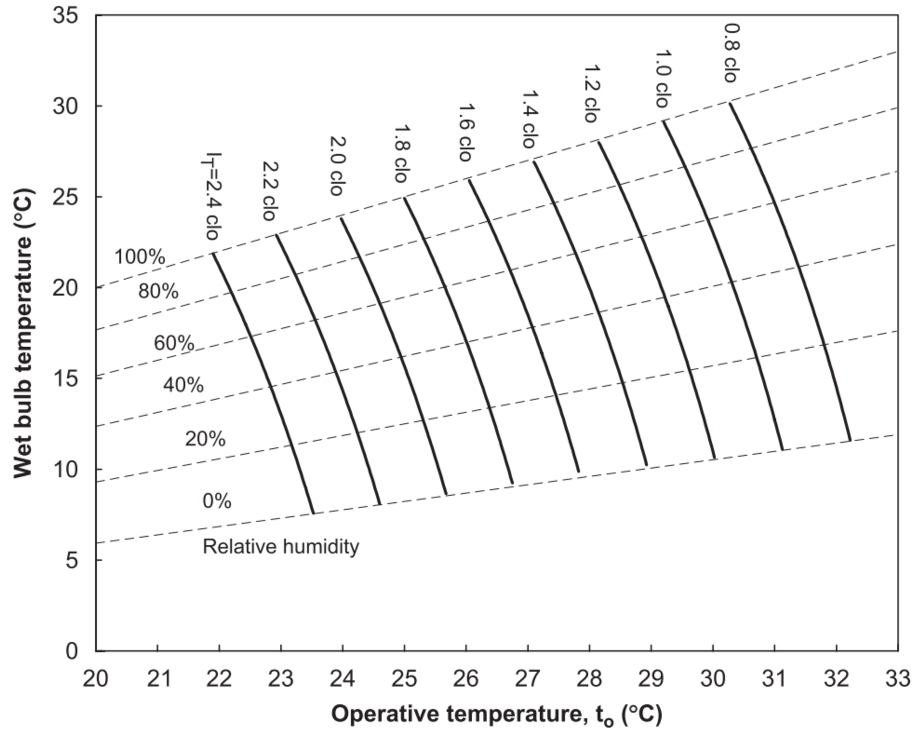


Figure 2.2 Comfort lines at different total insulation values ($v \leqslant 0.15$ m/s) (Lin and Deng 2008a)

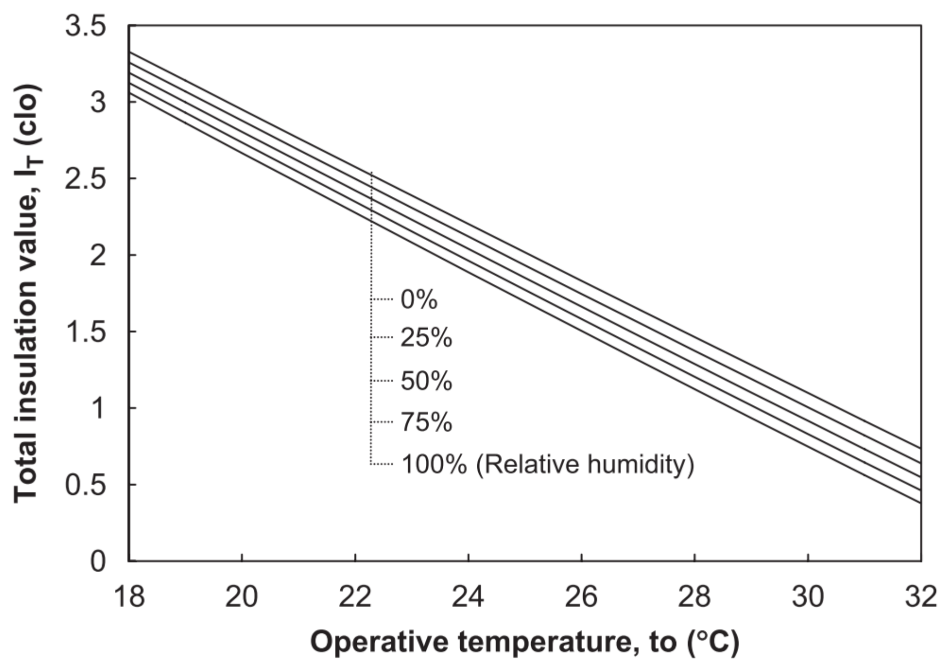


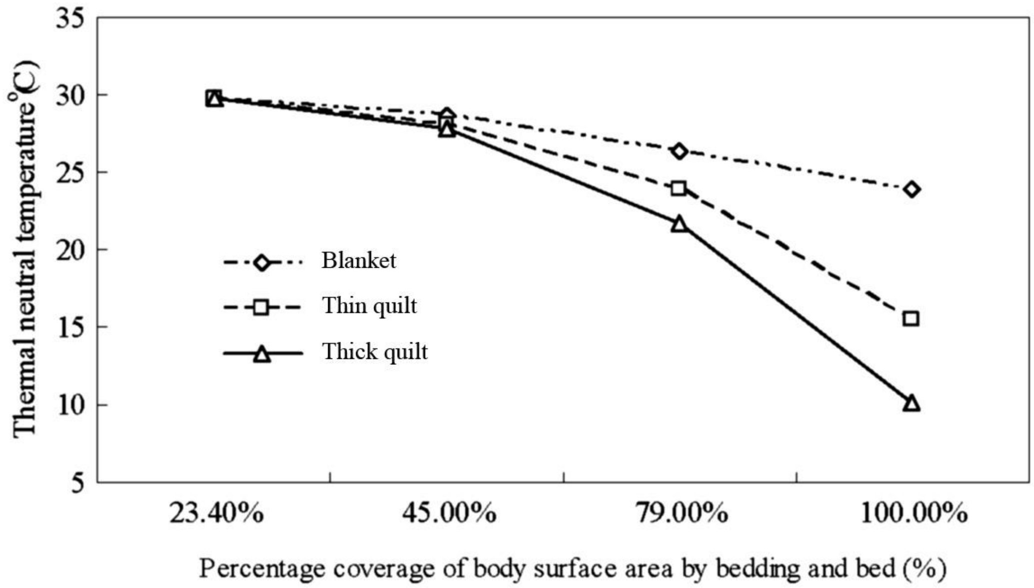
Figure 2.3 Relationships between operative temperature and the total insulation value ($v \leqslant 0.15$ m/s) (Lin and Deng 2008a)

Apart from theoretical studies on thermal comfort, experimental studies have also been carried out to investigate the impacts of high/low ambient temperatures on sleeping stages by using different thermal neutral temperatures ranging from 20 ^{o}C to 32 ^{o}C during sleep (Candas et al. 1982; Karacan et al. 1978; Macpherson 1973; Haskell et al. 1981a; Dewasmes et al. 2000; Di Nisi et al. 1989). In Table 2.1, a summary on the thermal neutral temperatures used in the different experimental studies about sleep is provided. The difference in thermal neutral temperatures used in different studies was significant, suggesting that a well-agreed single thermal neutral temperature for a sleeping environment has not been determined yet. Apart from ambient air temperature, operative temperature or mean radiant temperature were not concerned in most of the studies about sleep. When air temperature of 20-25 ^{o}C was used, the test subjects were covered with bedding, while for other air temperatures, the subjects or sleepers may be nude. On the other hand, although deciding a thermal neutral temperature depends on different experimental conditions used, it is clear that the determined thermal neutral temperatures for a sleeping environment in the previous studies differed from the air temperature (24-26 ^{o}C) usually used in office buildings in summer.

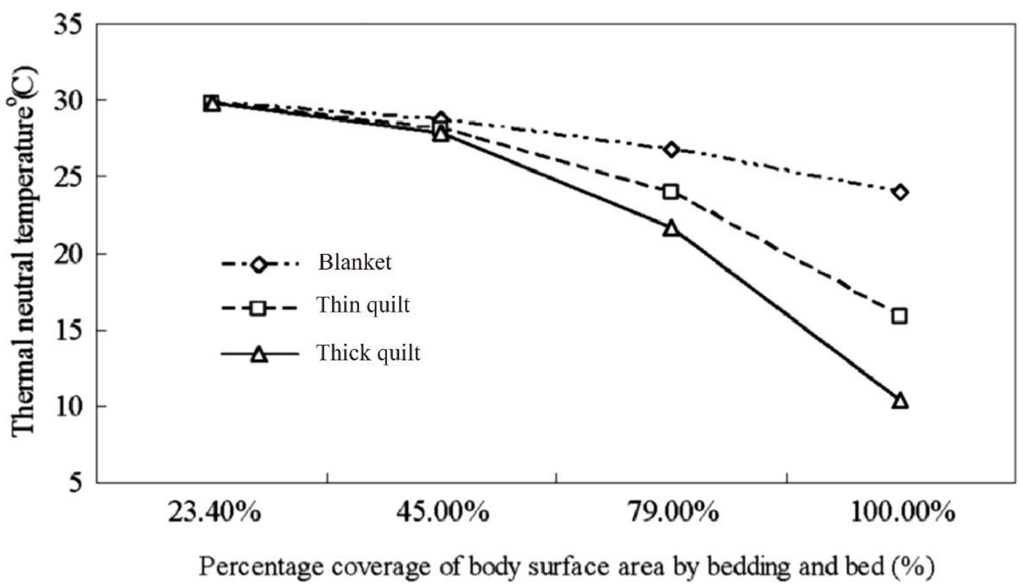
Table 2.1 The determined thermal neutral temperatures in different studies about sleep (Mao 2015)

Literature	Thermal neutral temperature (ºC)	Condition of test subjects	Remark
Macpherson 1973	29-32	Naked	Air temperature
Karacan et al. 1978	22.2	Covered	Air temperature
Haskell et al. 1981	29	Naked	Air temperature
Candas et al. 1982	32	Naked	Operative temperature
Palca 1986	29	Naked	Air temperature
Sewitch et al. 1986	20-22	Covered	Air temperature
Di Nisi et al. 1989	30	Naked	Operative temperature
Dewasmes et al. 2000	28	Naked	Air temperature

In addition, numerical methods were also used to find the thermal neutral temperatures for a sleeping environment under different conditions (Pan et al. 2011). Figure 2.4 shows the obtained thermal neutral temperatures at different percentage coverages of body surface area by beddings at three different bedding systems. As seen, an increase in the thermal insulation of beddings can greatly reduce thermal neutral temperature, suggesting that the thermal insulation of beddings had great impacts on the thermal neutral temperature of a sleeper. However, supply air velocity did not have considerable influence on the thermal neutral temperature for 100% coverage at all three different beddings, when comparing results shown in Figure 2.4(a) and Figure 2.4(b).



(a) at a supply air velocity of 0.12 m/s



(b) at a supply air velocity of 0.24 m/s

Figure 2.4 Relationships between thermal neutral temperature and percentage coverage of body surface area by bed and bedding at the three different beddings (Pan et al. 2011)

2.4. Energy use for air conditioning in residential buildings in the subtropics

With the increased demand on indoor thermal comfort, air conditioning in buildings has become a necessary provision in people’s daily life. A/C systems are used in not only workplaces at daytime, but also sleeping environments at nighttime. Although sleeping environments can be present in many types of buildings, like bedrooms in residential buildings, wards in hospitals and guestrooms in hotels, the bedrooms in residential buildings are the principal form of sleeping environments. Therefore, a review on the energy consumption for A/C in residential buildings in the subtropics is presented in this section.

Figure 2.5 shows the monthly electricity consumption in the commercial residential sectors in Hong Kong from 1979 to 2006. As seen, during the period, electricity consumption in residential buildings had an averaged annual increase rate of 5.9% from 2099 GWh in 1979 to 9841 GWh in 2006 (Lam et al. 2008). Furthermore, the demand for residential electricity always increased, even during the worsen economic period (i.e., the Asian financial crisis) from late 1990s to early 2000s (Lam et al. 2008) and a large proportion of energy consumption was due to air conditioning in residential buildings.

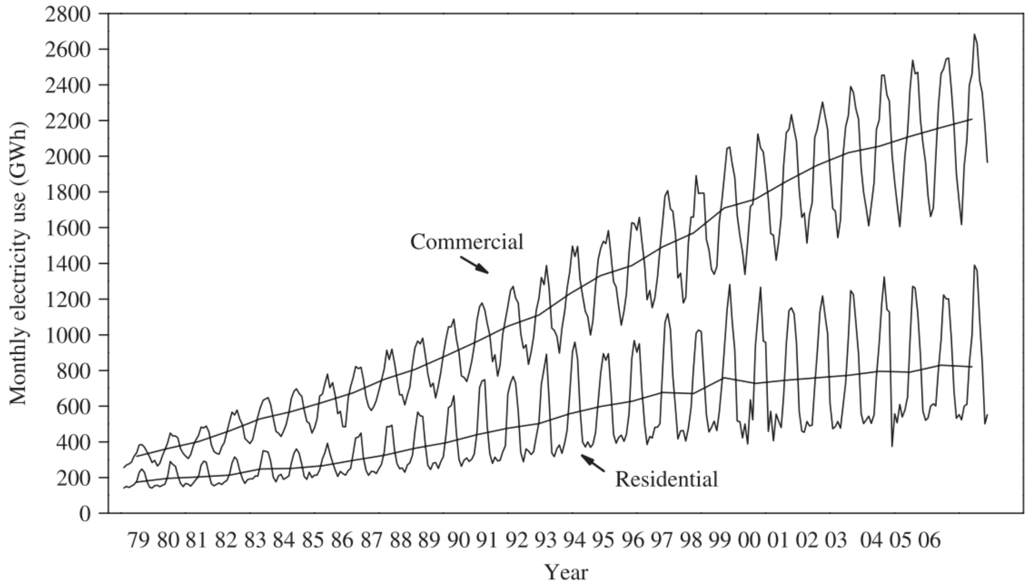


Figure 2.5 Monthly electricity consumption in the commercial and residential sectors from 1979 to 2006 in Hong Kong (Lam et al. 2008)

2.4.1. Cooling loads characteristics in residential buildings in the subtropics

Space cooling loads in residential buildings come from many sources, including building envelopes, occupants, lights and internal appliances, etc. Among these, the cooling load due to heat transfer through envelopes takes the lion's share of the total cooling load. As illustrated in Figure 2.6, in a subtropical region like Hong Kong, envelope heat gains can be responsible for more than two thirds of the total cooling load in a bedroom. Furthermore, during sleep time at night, internal loads is normally kept at a low level, and may only come from occupancy, at about 72 W for an average adult (ASHRAE 2013d). Therefore, the variations of heat gain from envelopes dominate that of the total nighttime cooling load in residential buildings.

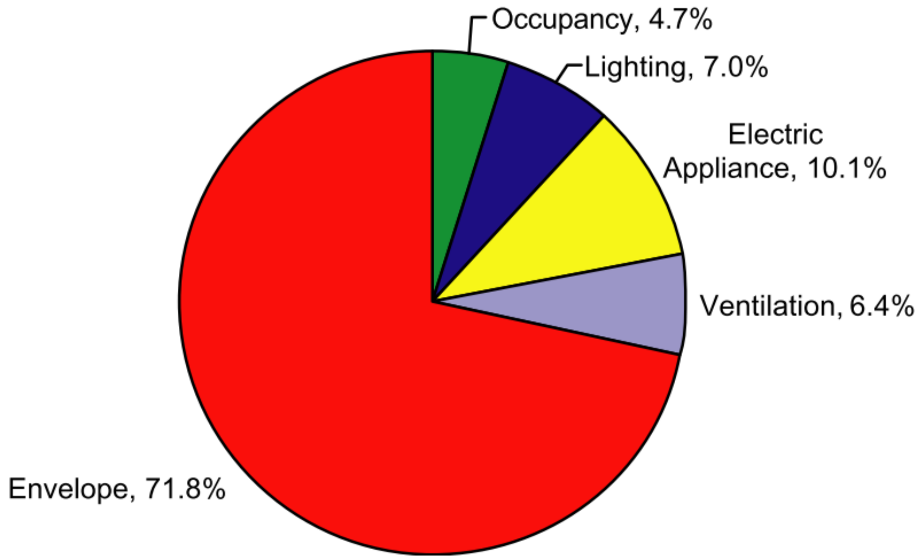


Figure 2.6 Percentage breakdown of the total cooling load for the west-facing bedroom at the A/C starting time (Lin and Deng 2004)

On the other hand, during a nighttime A/C process, the total cooling load at different hours in a bedroom facing different orientations can differ greatly. This can be seen in Figure 2.7, which shows the hourly total cooling loads in bedrooms facing four orientations during 22:00-07:00 in the summer design day. As seen, the highest hourly total cooling load occurred at the first hour of A/C operation. Therefore, the hourly nighttime cooling loads was rapidly reduced in the next 2 hours, and then the decrease rate of cooling load became small from 0:00 to 6:00. In the last hour, the cooling loads was however slightly increased, since space heat gains in bedrooms were increased due to the rising outdoor air temperature and increased internal loads, i.e., from lights and electrical appliances if turned on.

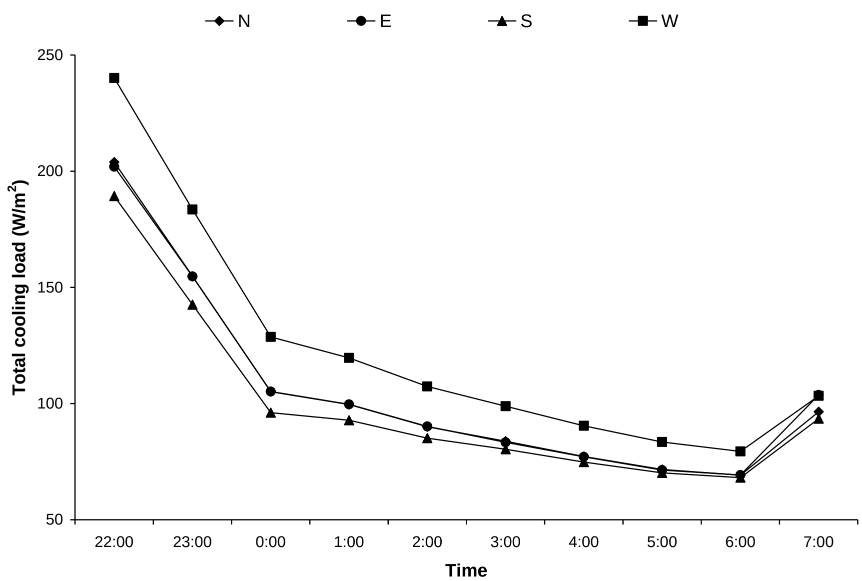


Figure 2.7 Hourly profiles of the total cooling loads for bedrooms facing four different orientations at nighttime operating mode in the summer design day of Hong Kong (Lin and Deng 2004)

2.4.2. Energy use of air conditioning for residential buildings in the subtropics

Summers in the subtropics can be very humid and hot, and may last for more than 7 months. Therefore, to maintain a thermally acceptable indoor environment, RACs are widely used in residential buildings in the sub-tropics. According to the statistics of energy consumption during 1984-1997 in Hong Kong (Chow 2001), as shown in Table 2.2, energy use by air conditioners was increased significantly from 2040 TJ to 9648 TJ, which was a 4.73-fold increase. Furthermore, its percentage share of the household sector consumption was increased from 9.0% to 22.3%. Another survey results (Tso and Yau 2003) showed that, in Hong Kong, there were 95% of households owing at least on air-conditioner with a long operating season and air conditioning was regarded as the largest end-use of electricity, accounting for 46% and 59.1% of the total electricity consumption in the whole year and summer months, respectively.

Table 2.2 Energy use in the domestic sector in Hong Kong, 1984–1997 (Chow 2001)

Years	Total domestic energy consumption (TJ)	Cooking (TJ)	Share (%)	Air conditioning (TJ)	Share (%)	Water heating (TJ)	Share (%)	Lighting and refrigeration (TJ)	Share %	Others (TJ)	Share %
1984	22,718	8381	36.9	2040	9	5423	23.9	4069	17.9	2805	12.3
1985	23,658	8508	36	2386	10.1	5717	24.2	4260	18	2787	11.8
1986	25,088	8566	34.1	3396	13.5	5987	23.9	4501	17.9	2638	10.5
1987	26,457	8540	32.3	4137	15.6	6192	23.4	4683	17.7	2905	11
1988	28,646	8747	30.5	4967	17.3	6509	22.7	4911	17.1	3512	12.3
1989	30,346	8831	29.1	5371	17.7	6786	22.4	5105	16.8	4253	14
1990	32,329	8843	27.4	6665	20.6	6911	21.4	5381	16.6	4529	14
1991	33,831	8823	26.1	7488	22.1	7015	20.7	5632	16.6	4873	14.4
1992	35,678	9300	26.1	7723	21.6	7412	20.8	5885	16.5	5358	15
1993	37,793	9023	23.9	8451	22.4	7598	20.1	6274	16.6	6447	17.1
1994	39,971	9217	23.1	8854	22.2	8023	20.1	6604	16.5	7273	18.2
1995	41,419	9274	22.4	9622	23.2	8364	20.2	6884	16.6	7275	17.6
1996	43,526	9270	21.3	9957	22.9	8510	19.6	7266	16.7	8523	19.6
1997	43,300	9272	21.4	9648	22.3	8632	19.9	7667	17.7	8081	18.7

For details on the energy use of bedroom air conditioning, Wan and Yik (2004) carried out a study on the characteristics of energy end-use in high-rise residential buildings of Hong Kong. The results showed that, for bedrooms in residential buildings, air conditioners were mainly operated at night, beginning from 20:00. Table 2.3 lists the operating hours of RACs in those surveyed households in weekdays.

Table 2.3 Operating hours of RACs in residential buildings in weekdays (Wan and Yik 2004)

Percentage of air-conditioners operated	Living room	Bedroom
80%	18:00–22:00	23:00–05:00
20%	13:00–22:00	20:00–07:00

2.5. Task\ambient air conditioning (TAC) systems

There has been growing research interest on TAC systems since they can perform well in thermal comfort control, IAQ improvement and energy efficiency enhancement. TAC systems can be individually controlled by occupants to provide a thermally comfortable environment in a small, localized zone, leaving the thermal environment outside the zone uncontrolled (Bauman and Arens 1996). Occupants could change supply air velocity and direction, or even supply air temperature in an occupied zone. Therefore, the purposes of designing TAC systems are to allow occupants to have more access to local thermal environment control based on individual preferences, and to offer the potentials to improve work productivity and energy reduction for A/C.

2.5.1. The types of TAC systems and the associated benefits

TAC systems are mostly used in office buildings and other leisure places at daytime and can be categorized into three types according to their installation positions: floor based (Matsunawa et al. 1995), workstation based (Arens et al. 1991; Matsunawa et al. 1995; Zhang et al. 2010) and ceiling based (Yang et al. 2009).

The benefits of using TAC systems have been revealed from the many studies on investigating their operational performances through field measurements and laboratory experiments. The study results demonstrated that the use of TAC systems can lead to excellent performances in thermal comfort, IAQ and energy saving (Bauman et al. 1993a; Faulkner et al. 1995; Amai et al. 2007; Shute 1995).

The main advantage of using TAC systems is that the thermal conditions in an occupied zone can be individually controlled based on occupants' thermal comfort preferences. For example, Tsuzuki et al. (1999) carried out some laboratory tests and designed a TAC system using fan-driven supply vents which can allow individual thermal control in an occupied zone over a considerable temperature range, up to 7ºC for desktop vents and up to 5ºC for floor-based vents. Amai et al. (2007) experimentally studied different types of TAC systems installed in an environmental chamber and experimental results showed better performance in thermal comfort control when using TAC systems. Cho et al. (2001) pointed out that an easy and flexible control for thermal conditions in a local small zone when using an individual environmental module based TAC systems can effectively improve occupant thermal comfort in workplaces. Bauman et al. (1998a) carried out an experimental study on occupant thermal sensations when using different types of A/C systems, and the results demonstrated that the occupants' satisfaction on three aspects, i.e., temperature, temperature control and airflow direction, was greatly improved when using a TAC system based on the mean ratings of respondents.

Apart from good performances in thermal comfort control, ventilation effectiveness and IAQ in a breathing zone can be considerably enhanced since fresh air supplied was closer to occupants when using TAC systems (Nielsen 1996). The results in previous related studies demonstrated that ventilation efficiency can be greatly improved using desk-mounted TAC systems (Melikov et al. 2002; Faulkner et al. 2002). Shute et al. (1995) conducted a study on indoor air quality in a room with a floor-based TAC system and it was found that the concentration of indoor suspended substance was considerably lower as compared with that when using a traditional A/C system. Spoormaker et al. (1990) found out that the concentration of each indoor pollutant can adequately meet the criteria in the relevant standards in a building with a floor-based TAC system.

In addition, the potential of saving energy when using TAC systems is significant as compared to a full volume air conditioning (FAC) system. This was because that a thermally acceptable environment was maintained in an entire room when using a FAC system, but only in an occupied zone when using a TAC system. For the unoccupied zone, air temperature was left uncontrolled, even fluctuating out of comfort limits. Therefore, energy consumption for air conditioning the unoccupied zone when using TAC systems was saved, as compared to the use of FAC systems. Pan et al. (2012b) conducted an experiment and revealed that the performance of a personalized air-conditioning system, called innovative partition fan-coil unit (PCFCU), was better in improving thermal comfort and saving energy, in comparison to using a central A/C system. The experimental results also demonstrated that the time to achieve the same level of occupant thermal comfort can be shortened and energy consumption reduced to 45% by using the personalized A/C system. Mao et al. (2013) used energy utilization coefficient (EUC) (indicating energy saving is possible with EUC value greater than 1) to evaluate and compare the energy utilization performance between a TAC system and a FAC system in a sleeping environment. The results showed that the EUC value was obviously increased when using the TAC system as compared to when using the FAC system, suggesting that a large amount of energy may be saved by not air conditioning the unoccupied zone.

2.5.2. The applications of TAC systems to sleeping environments

Although the operating performances of TAC systems in workplaces at daytime have been extensively and thoroughly studied, there are limited research works on the application of TAC systems to a sleeping environment. These included the work by Pan et al. (2012b) who proposed a new bed-based TAC system with flexible air ducts and plenums. The experimental data supplemented with a theoretical analysis indicated that the thermal environment can be better controlled and energy consumption would be considerably reduced when using the bed-based TAC system. Furthermore, it was shown that, when delivering the air at 24°C with a flow rate of 50 L/s, using a blanket, or at 20-22°C and 50 L/s using a thin quilt, a thermally comfortable environment can be achieved with the bed-based TAC system. However, the flexible air ducts and supply air plenums in this system were very bulky and thus inconvenient to users, and may become impractical in real world, Mao et al. (2013) therefore made improvements to this bed-based TAC system, by removing the flexible ducts and plenums and supplying cooled air through a wall-installed outlet above a sleeping person, as shown in Figure 2.8. The experimental and simulation results showed that, apart from its high energy efficiency, such a ductless TAC system also can efficiently remove the contamination, in addition to maintaining a good thermal comfort in an occupied zone. However, cold-draft is still an unsolved problem in this ductless TAC system, although selecting reasonable design values for supply air temperature, flow rate and vane angle may help resolve the problem.

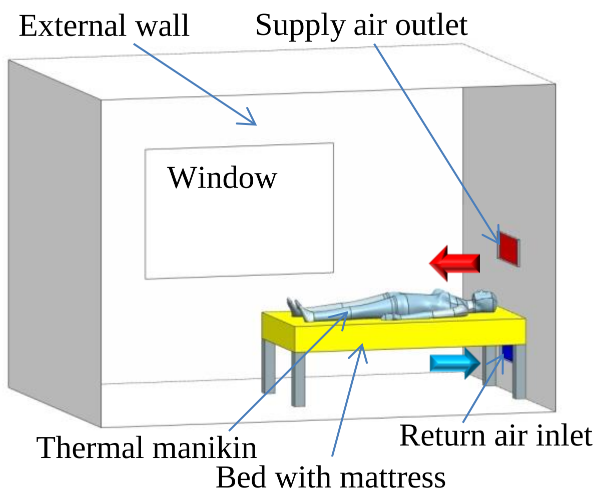


Figure 2.8 The improved bed-based TAC systems (Mao et al. 2013)

2.5.3. The existing problem in the current TAC systems applied to sleeping environments

Currently, most of the TAC systems applied to sleeping environments use predominantly the convection-based method to cool down the air in an occupied space in a sleeping environment. Hence, this will definitely cause the undesirable cold-draft feelings. For example, as reported by Mao at al. (2013), among 12 measurement positions in an occupied space controlled by the ductless TAC system, the values of draft risk in 8 positions inside the occupied space were higher than 20%, and even that in 4 out of the 8 positions 40%, which were far higher than the permissible value of 20% suggested by ASHRAE Standard 55 (ASHRAE 2013d).

Basically, “Draft’’ is defined as “a current of air in confined spaces”. Among the earliest investigations, Houghten et al. (1938) conducted a research on discomfort caused by cold drafts. The term of “draft” was used to describe a local thermal sensation of coolness that occupants could feel in some parts of the body. Fanger et al. (1988) defined the term of cold draft as “an unwanted local cooling effect of the human body caused by air movement”. Cold draft can not only cause thermal discomfort, but also result in some temporary illness or even chronic diseases. For example, Xia et al. (2000) mentioned that when occupants were exposed to airflow for a long time, it can cause headache or giddiness. Griefahn et al. (1997) found out that certain chronic diseases, like pains in the back and joints, rheumatic and bronchitic complaints, colds and hearing problems, were significantly associated with the thermal environment, particularly with drafts, at workplaces. Therefore, for the sake of occupants’ health, it is necessary to develop a novel non-convection based TAC system to mitigate cold draft.

2.6. Radiant cooling panel systems

Currently, for convection-based TAC systems, it is rather difficult to avoid cold draft problem. This is because, when using a convection-based TAC system, an occupied space has to be ventilated and supplied with cooled fast-moving air for the purpose of removing the heat gain in the occupied zone by convective heat transfer. Therefore, a relatively high air velocity around occupants is inevitable, which lead to unwanted draft feelings. To address the cold draft issue, radiative cooling strategy may be considered.

Nowadays, radiant cooling panels (RCP) combined with ventilation systems have been widely used in all types of buildings, with increasing popularity, because of their unique heat transfer mechanism. In an RCP system, as illustrated in Figure 2.9, a cold panel firstly cools down walls by radiation, then the cold panel together with the cooled wall surface exchanges heat with room air by convection. Although internal heat gains are removed by both radiation and convection, the former is the dominant means for heat exchange. Furthermore, heat transfer by radiation is more direct, as compared with convective heat transfer that using air as a medium to exchange heat with occupants. For installation location, a radiant cooling panel can be integrated into ceiling, floor, walls and columns in a room. Generally, ceiling radiant cooling systems are preferred in most cases due to their ability to mitigate discomfort caused by vertical air temperature stratification (Le Dréau and Heiselberg 2014). It is believed that radiant cooling panel systems have two major overwhelming advantages over the traditional full-volume air conditioning (FAC) systems: reduced energy consumption and better indoor thermal comfort control, as detailed in Sections 2.6.1 and 2.6.2, respectively.

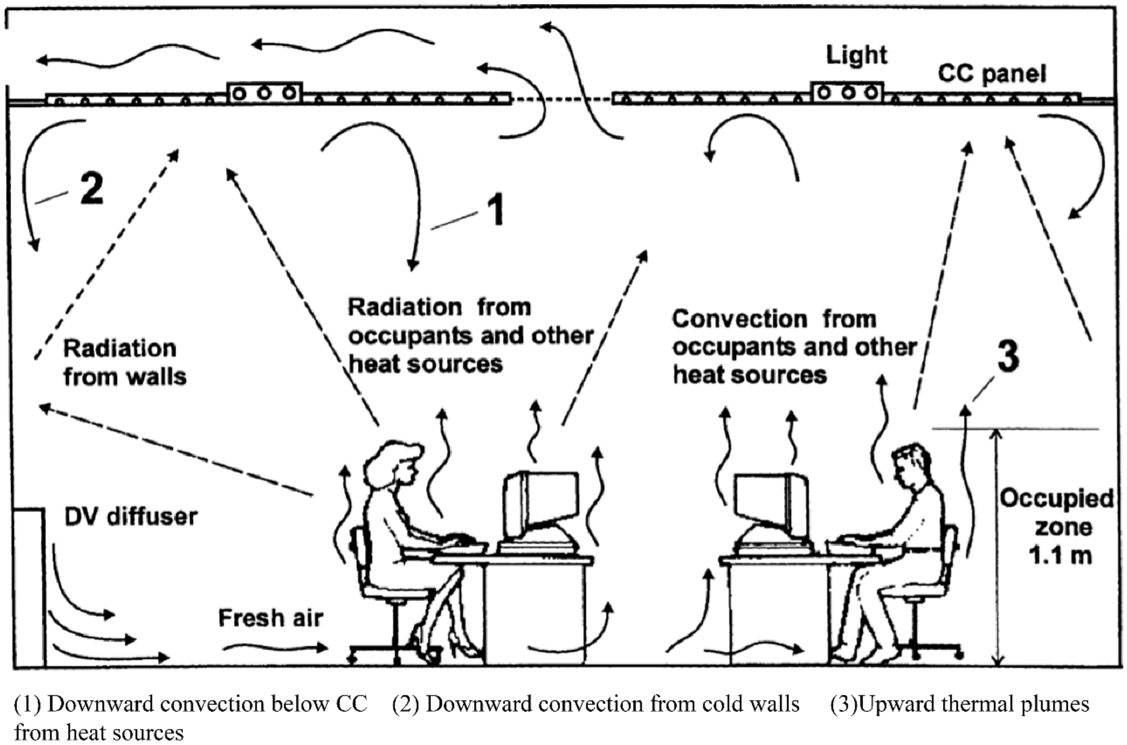


Figure 2.9 The convective and radiative heat transfer inside a room with a ceiling radiant cooling panel system (Novoselac and Srebric 2002)

2.6.1. Energy performance when using radiant cooling panel systems

RCP systems are more energy efficient than FAC systems, because the energy consumed by the water pump for circulating cold water in a RCP system is considerably lower than that by fans in a FAC system. A higher operating temperature in RCP systems allows a chiller to operate more efficiently. Furthermore, renewable energy, such as geothermal energy, can be adopted to provide water at a proper temperature for cooling. Additionally, it is possible to take the advantage of building thermal mass when using RCP systems, so as to not only attenuate fluctuations in room air temperature but also decrease peak energy consumption (Rhee and Kim 2015). For example, Niu et al. (2002) conducted a numerical study on comparing energy performances of four A/C systems, including a ceiling radiant cooling panel with desiccant cooling, an all-air system, an all-air system with heat recovery and a radiant cooling system with an air handling unit. The results showed that the first one could help save up to 44% of primary energy consumption as compared to the all-air system. Furthermore, desiccant regeneration can be used for low-grade heat sources lower than 80 ^{o}C. Feng et al. (2015) claimed that 55% and 25% of energy consumed by the cooling tower and pumping, respectively, can be saved when using a radiant cooling system if its control system is properly configured. Stetiu (1999) found out that a radiant cooling system can be operated in any climate of America at a low risk of condensation and save 30% of the energy consumption and 27% of the peak power demand, as comparison to a conventional all-air system. Zmeureanu and Fazio (1988) conducted a numerical study on a hollow core slab radiant system and found out that on average, it can reduce space cooling load by 28.4-44.2 W/m^2, which is roughly equivalent to occupant heat gains, or half of the heat gains from lights. Chiang et al. (2012) focused on the energy saving potentials of using a cold ceiling system when increasing supply air temperature of the supplementary mechanical ventilation. The simulation results demonstrated that a 6 ^{o}C increase, from 18 ^{o}C to 24 ^{o}C, in supply air temperature can lead to 13.2% of energy saving for chillers and 8.0% for the entire A/C system. Hao et al. (2007) investigated the energy consumption of a ceiling radiant cold panel system with displacement ventilation, combined with a desiccant dehumidification system. It was shown that, compared with traditional air conditioning systems, the use of the radiant cold panel system saved 68.5% of chiller energy and 39.0% of fan energy, but increased 25.6%

of pump energy use and 4 times of boiler energy use due to regeneration of desiccant. It was concluded that the total energy saving was 8.2%.

2.6.2. Thermal comfort performance when using radiant cooling panel systems

It has been proved in many related studies that thermal comfort control by radiant cooling systems can be greatly improved, since radiative heat transfer to occupants is rapid and would not cause the feeling of cold drafts. Compared with FAC systems, RCP systems can help achieve better thermal comfort control due to less air movements, smaller vertical temperature gradient and decreased local occupant discomfort due to staying in cooled indoor environment for a long time (Rhee and Kim 2015). Tian and Love (2008) carried out a field study on examining thermal comfort in a building with radiant slab cooling in Canada. The measured results showed that the averaged operative temperature was approximately 22 ^{o}C, ranging between 20.3 ^{o}C and 23.6 ^{o}C, and the averaged PMV value was -0.53 in summer, which was within the comfort zone. Kim et al. (2005) conducted a numerical study to compare the thermal comfort in an office space using three different cooling systems: a cooling panel system, an all-air system and a mixture system (i.e., cooling panel + natural ventilation). It was demonstrated that, compared with the all-air system, the radiant cooling panel system can remove more heat from a human body by radiative heat transfer, leading to a lower MRT and operative temperature, and thus a better cooling efficiency. Chiang et al. (2012) evaluated the thermal comfort characteristics using a ceiling radiant cooling panel (CRCP) system combined with ventilation and revealed that, whether the radiant panel temperature was at 18 ^{o}C or 22 ^{o}C, a thermally acceptable environment with the PMV values of -1.0 - 1.0 could be met, even at a high supply air temperature of 24 °C. Catalina et al. (2009) found that air velocity was not greater than 0.1 m/s for most of the zones when using a CRCP system and the vertical temperature gradient was less than 1 °C/m, which can adequately meet thermal comfort requirements recommended by ASHRAE (ASHRAE 2013d). It was also proved by Tian and Love (2009) that radiant cooling systems can help achieve a low draft rate at 4%, with the mean air velocity lower than 0.06 m/s during summer months.

2.7. The CFD method used for studying indoor thermal environments

Due to the significant development in computing speed and capacity, computational fluid dynamics (CFD) has been increasingly used in various scientific fields, including mechanical, aerospace, building and structural engineering. Among these, the CFD method can be considered as a valuable and useful tool in the research area related to HVAC, since air flow pattern, air temperature distribution and heat transfer, etc., can be accurately predicted with the CFD method (Liddament 1991; Chen 1996; Emmerich 1997; Stamou and Katsiris 2006; Chen and Zhai 2004; Zhang et al. 2007; Zhai 2006).

2.7.1. Governing equations

The 3D, viscous and steady-state Reynolds averaged Navier-Strokes (RANS) equations, combined with continuity and energy equations, for a flow field are as follows:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot \left(\rho \vec{v} \right) = 0 \quad (2.3)$$

Momentum equation:

$$\frac{\partial}{\partial t}\left(\rho \vec{v} \right) + \nabla \cdot \left(\rho \vec{v} \vec{v} \right) = -\nabla P + \nabla \cdot \left(\bar{\bar{\tau}} \right) + \rho \vec{g} \quad (2.4)$$

Where P is static pressure, $\bar{\bar{\tau}}$ stress tensor (Pa) and $\vec{g}$ gravitational acceleration (m/s^2).

Energy equation:

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = \nabla \cdot (\rho k_{eff} \nabla T - h + (\bar{\bar{\tau}}_{eff} \cdot \vec{v})) \quad (2.5)$$

Where k_{eff} is the effective conductivity W/(m·^{o}C), and E total energy (J), expressed as:

$$E = h - \frac{p}{\rho} + \frac{v^2}{2} \quad (2.6)$$

2.7.2. Turbulence models and applications

Turbulent flows can be predicted using three turbulence modeling methods: Large Eddy Simulation (LES), Direct Numerical Simulation (DNS), and Reynolds-Averaged Navier-Stokes (RANS) equation simulation by turbulence transport models, as shown in Figure 2.10.

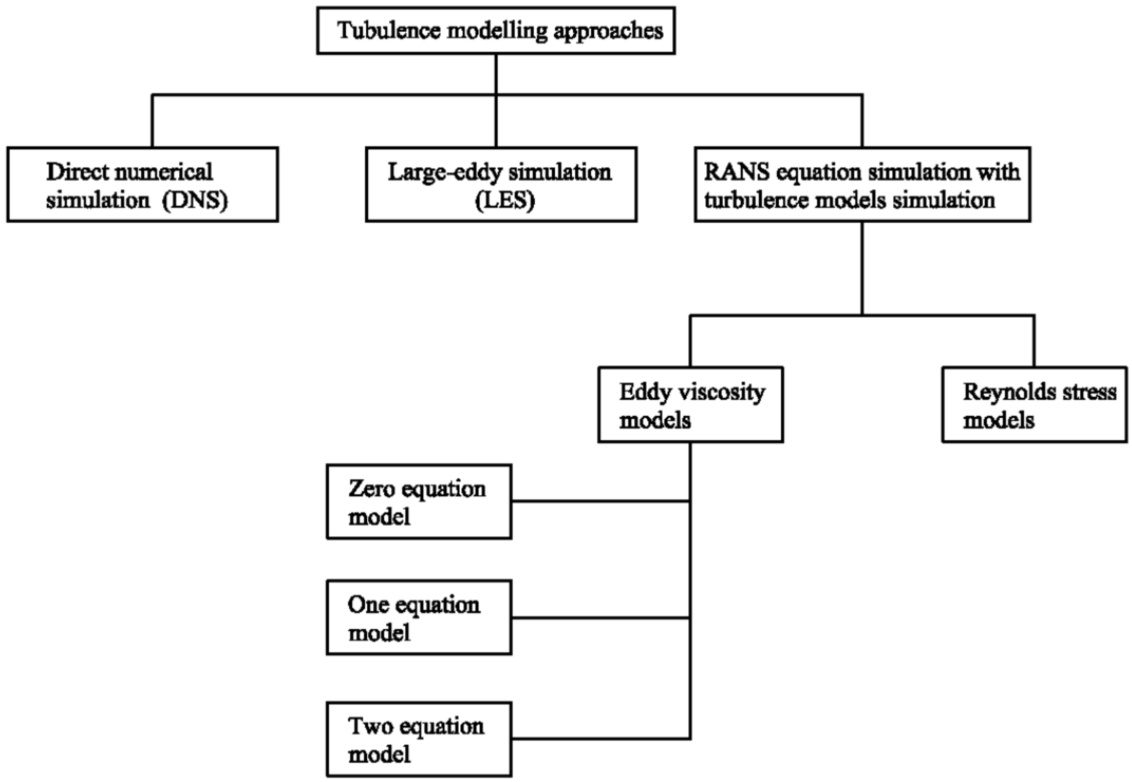


Figure 2.10 The classification of different turbulence modeling methods (Zhai et al. 2007)

Air flow in an enclosed environment can be characterized by numerous flow elements. For example, potential flow and jet flow are usually used to describe the air flow pattern at a return vent and a supply vent, respectively. For indoor air flow, it can be classified as forced convection, natural convection and mixed convection based on its driving force. Generally, different air flow elements should be dealt with different turbulence modeling methods for better accuracy. However, so far, it has been impossible to develop one turbulence modeling method, which allows to accurately and economically predict all of the air flow patterns in a room. Therefore, several factors should be considered when choosing a turbulence modeling method: the level of computational accuracy required, the time needed for simulation, the available computational capacity, and the physical mechanism of the flow, etc.

So far, to deal with practical engineering problems, k-ε turbulence modeling method is still the most frequently used one. A standard k-ε model is only appropriate when solving fully turbulent flow problems. It was pointed out by Niu and Van der Kooi (1992) that, for a zone surrounding a solid surface, the k-ε model should work together with wall functions, since the computational accuracy would be significantly affected by the thickness of the first mesh layer near the boundary walls. The low Reynolds number k-ε and the RNG k-ε modeling methods are two commonly used models in simulating the air flow patterns around an occupant. Murakami et al. (1995) claimed that, to accurately reflect the convection between an occupant and its ambient air, the low Reynolds number k-ε modelling method with local finer grids near an occupant performed better than the standard k-ε turbulence model if no wall functions were applied. However, due to the extra fine grids close to a human body, it was hard to reach convergence in the simulations with the low Reynolds number k-ε model (Voigt et al. 2001).

Recently, another turbulence modeling method, the k-ω model (Menter 1994; Wilcox 1994), is increasingly popular in solving many industrial problems. However, it performs poor in predicting wake region and free shear flows. Therefore, combining the advantages in both the k-ω and the k-ε turbulence models, a shear stress transport (SST) k-ω model was proposed by Menter (1994) and satisfactory performance was achieved. For instance, Arun and Tulapurkara (2005) used three turbulence modeling methods, including

Reynolds stress model, RNG k-ε model and SST k–ω model, to study the air flow in an enclosed environment with an internal partition. The numerical results showed that the SST k-ω model was good at predicting various complicated flow features, such as the exit of swirling flow, the reversed flow at the top part and the downstream vortices near the partition. Stamou and Katsiris (2006) carried out a numerical study on predicting air flow and temperature fields in a simulated workplace with a TAC system using different turbulence models. The results demonstrated that the SST k-ω turbulence modeling method performed the best in accurately capturing the major qualitative flow features as compared to the experimental data. Kuznik et al. (2007) also made a comparison between the experimentally measured velocity and temperature and the corresponding predicted values using four advanced turbulence model: the RNG k-ε model, the Realizable k-ε model, the standard k-ω model and the SST k-ω model in a mechanically ventilated room. It was concluded that using the k-ω turbulence modeling method to simulate indoor air flows can achieve better results without compromising numerical stability. In general, it was suggested in various previous studies that the overall performance of the SST k-ω model (Menter 1994) was superior to those of the RNG k-ε and the standard k-ε models.

2.7.3. Mesh generation

To achieve a high level of correctness and accuracy of CFD simulation results, mesh quality is a key factor. Apart from the mesh size, the mesh quality is also affected by many other mesh factors, like mesh shape, mesh ratio, mesh topology and the mesh consistency for a geometry, etc.

To discretize a domain, four methods are commonly used: cylinder co-ordinates, cartesian co-ordinates, body-fitted co-ordinates and unstructured meshes. For generating meshes around a complex geometry like a thermal manikin, the last two out of the four methods are better to represent the complicated boundaries. However, unstructured meshes are more flexible in generating and distributing meshes, although the body-fitted co-ordinates method has its own advantage in dealing with internal/external flows with smoothly varying abnormal boundaries.

To simulate the thermal environment in an occupied room, two mesh generation methods are commonly used: unstructured grids for a cuboid space enclosing an occupant and structured grids for rest of the space. This is because that unstructured meshes can be automatically generated around the complex geometry of a human body while structured meshes for the rest of space can save much of computational resources and has a higher mesh quality (Gao and Niu 2005; Mao et al. 2014a).

2.7.4. Related numerical studies on the thermal environment around a human body

The knowledge of the thermal environment surrounding an occupant greatly helps correctly evaluate the occupants' thermal comfort. When a thermal manikin was used, the heat loads from the thermal manikin can be fixed to a constant value or adjusted based on different activity levels, various related studies were carried out based on a thermal manikin (Holmér 2004).

Gao and Niu (2005) conducted a review on several issues related to using CFD method to predict the thermal environment around a human body, which includes mesh generation, boundary condition settings, geometric complexity and turbulence model selection. Firstly, the posture, size and the level of geometric complexity of thermal manikins used are different in different related studies, depending on the purpose and required accuracy of a particular study. The geometry for a thermal manikin can be generated using computer aided design, even the laser scanning technique, capable of capturing the details of a thermal manikin (Gao and Niu 2004; Sørensen and Voigt 2003; Pan et al. 2011). Although using a geometry-simplified thermal manikin can save a large amount of computational resources and time for mesh generation and modeling, the use of a more detailed geometry of a thermal manikin is more helpful in reflecting the reality, and can generate more accurate numerical results. Secondly, the boundary condition of a thermal manikin is commonly set at a fixed surface heat flux or a fixed surface temperature. In the previous studies, the convective heat flux was usually set within the range from 20 W/m^2 (Hayashi et al. 2002) to 25 W/m^2 (Brohus and Nielsen 1995). The skin temperatures of a thermal manikin used previously ranged from 31°C (Murakami 1997) to 33.7°C (Sørensen and Voigt 2003). Finally, many numerical studies on convective and radiative heat transfers from a thermal manikin have been carried out, and the studies were useful to evaluate the occupant thermal comfort in an enclosure (Silva and Coelho 2002; Murakami et al. 2000; Sørensen and Voigt 2003; Yang et al. 2002; Gao and Niu 2004).

2.8. Passive energy saving measures in buildings

Energy use for maintaining a thermally comfortable environment accounts for a large proportion of building energy consumption. Hence, it is necessary to improve building energy efficiency by applying either active or passive energy saving measures. Improving the operating performances of HVAC systems can be regarded as active measures, while the improvements to the materials used and construction of building envelopes as passive measures (Sadineni et al. 2011). The application of passive measures to building envelope components can be regarded as the intrinsic feature of a building, which can essentially reduce the total space cooling load. Therefore, to certain extent, passive measures are more important and efficient than active ones in energy achieving.

In cold climates, building envelopes are expected to have high thermal mass, so as to store more solar energy and lose less energy from indoor environment to outdoor. These can be realized by passive measures. For example, Trombe Walls can effectively absorb and transfer solar energy in buildings with a glazing layer and store the solar energy with a high thermal mass wall (Mazria 1979). There are a number of new developments from the classic Trombe wall. For example, Trombe wall combined with a steel panel backed with polystyrene insulation mounted on a south façade can greatly improve the operating efficiency (Ji et al. 2009); PV integrated Trombe wall can solve the problem of overheating of the PV panel in summer (Jie et al. 2007); Phase change material based Trombe wall can reduce the thickness of building envelopes without yielding the energy performance (Tyagi and Buddhi 2007). Moreover, Transwall is a transparent modular wall, comprised of water enclosed and a semi-transparent glass between two parallel glass panes, which can turn the transmitted solar radiation into both heating and illumination in a dwelling space (Nayak 1987). Building envelopes with phase change materials (PCM) can improve the capacity of storing thermal energy and help considerably reduce the heating demand during night (Athienitis et al. 1997).

In hot climates, on the contrary, many passive measures are used to reduce thermal energy storage in building envelopes or solar heat gain. For example, cooling load saving can be achieved by using phase change materials (PCMs) as the components of building envelopes (Waqas and Ud Din 2013), which can utilize cooling energy storage at phase change point to cool down building spaces. A previous study (Nagano 2007) showed that the cooling energy stored in PCM during nighttime reduced the cooling demand by up to 92% on the following day in comparison with a conventional storage system which could only reduce the cooling demand up to 50%. Reflective materials (e.g., aluminum foil, light color paintings) may be commonly adhered to external surfaces of building envelopes (Xamán et al. 2010; Shariah et al. 1998; Ascione et al. 2010), to reduce solar radiation heat gain and thermal irradiance from the surroundings. Akbari et al. (2005) conducted a field study on six different types of buildings retrofitted with high reflectance white coatings or white PVC single-ply membrane. The results demonstrated that, with the high reflective materials covering on building envelopes, cooling load could be reduced by 5-40% and peak demand by 5-10%. Building envelopes can also be covered with a layer of vegetation, called as green wall/roof, which not only improves solar radiation reflectivity, but also act as an extra thermal insulation layer. A field measurement using thermal infrared camera showed that the surface temperature of green wall can significantly lower than that of a bare wall, with a maximum reduction of 4.67 ^{o}C (Yin et al. 2017).

In addition, an air gap can be commonly applied to external opaque walls, or to curtain walls to form something called double skin façades to increase their overall thermal transfer resistances. In some studies, an air gap was ventilated naturally or mechanically (Chan et al. 2009; Hien et al. 2005). The energy-saving characteristics by adding an air gap in an external envelope element have been extensively studied. For example, it was reported that in Hong Kong, the space cooling load in a new building using double skin façades was decreased by 26%, as compared to that in a conventional building (Chan et al. 2009). Zhou and Chen (2010) pointed out that applying ventilated double skin glazing façades would be an effective way to save energy for commercial buildings in hot-summer and cold-winter regions in China. Ciampi et al. (2003) also demonstrated that, with an increase in width of an air gap (not more than 0.15m), energy savings for all the wall designs were increased and 40% of summer cooling energy use can be saved. In addition to energy saving, the use of an air gap may also help improve indoor thermal comfort (Hien et al. 2005). Furthermore, detailed thermal properties of air gaps in building envelopes were also studied (Anderson 1981; Aviram et al. 2001). However, most of the previous studies focus on daytime energy performances of adding an air gap to external walls in office buildings. Therefore, the passive measure of applying an air gap to the opaque external wall of a bedroom may be used to reduce nighttime space cooling load.

2.9. Conclusions

Currently although air conditioning is mainly for daytime activities in different buildings, such as offices and shopping malls, in tropics and subtropics, A/C is provided to sleeping environments at nighttime because of hot and humid weather conditions. Therefore, to provide people with a thermally acceptable sleeping environment, it is necessary to study the related factors affecting sleeping quality, different air conditioning methods applied to sleeping environments and approaches to improve sleeping thermal comfort.

In this Chapter, an extensive review on the previous related studies on above issues is presented. The key findings from this review are summarized as follows. Firstly, as shown in Section 2.2, the factors influencing sleeping quality may include the personal factors of a sleeper, and the environmental factors in a sleeping environment, such as noise level, lighting, IAQ and thermal environment. However, as presented in Section 2.3, there have been only a limited number of previous studies on investigating the appropriate thermal parameters in a sleeping environment, developing thermal comfort models applied to sleeping environments and determining thermally neutral temperature for a sleeping environment.

Secondly, as seen in Section 2.4, the characteristics of nighttime cooling loads from envelopes in a bedroom of a residential building in subtropical Hong Kong were previously studied. It is also shown that the increased use of air conditioning at nighttime during sleep led to the increased energy consumption in residential buildings. Therefore, it becomes obvious to develop new measures to reduce energy use for air conditioning a sleeping environment.

Thirdly, as TAC systems perform excellently in improving local thermal comfort control and decreasing energy use, various TAC systems used at daytime and their advantages are reviewed in Section 2.5.1. However, as pointed out in Sections 2.5.2 and 2.5.3, there were only limited previous studies on TAC systems used for sleeping environments. And although currently the convection-based TAC systems applied to sleeping environments could save more energy and fairly maintain a thermally comfortable sleeping environment, the problem of cold-draft remain unsolved. Therefore, a novel radiation-based TAC system that combined the advantages of the TAC systems and radiant cooling systems reviewed in Section 2.6 should be developed.

Fourthly, Section 2.7 presents a review on using CFD method to study the micro thermal environment around an occupant suggested. The CFD method is powerful to study thermal comfort, energy utilization efficiency and ventilation effectiveness in the field of HVAC engineering, and can therefore be used to study a radiation-based TAC system applied to sleeping environments.

Finally, in Section 2.8, the review on the previous related studies suggested that building energy use can be saved by applying passive measures to building envelopes. However, there were little studies on applying passive measures to envelopes of a bedroom which is primarily used as a sleeping environment, to reduce its nighttime space cooling load.

Therefore, the passive measure of adding an air gap to the opaque external wall was proposed.

Through the extensive literature review reported in this Chapter, future studies on addressing the cold draft problem in the existing C-TAC system applied to sleeping environments and on reducing nighttime space cooling load in a bedroom using passive energy saving measures have been identified. These future studies formed a systematic research program reported in this Thesis.

Chapter 3

Proposition

3.1. Background

From the review on previous related studies presented in Chapter 2, it can be understood that a TAC system may be best applied to a sleeping environment, due to its excellent performance in providing thermal environment control in a local area, and the immobility of a sleeper who occupies a small space. However, the current TAC systems applied to sleeping environments are convection based and the problem of cold drafts has been identified. Therefore, it is necessary to develop an improved non-convection based TAC system applied to sleeping environments and carry out further experimental and numerical studies on evaluating the performances of the improved TAC system.

While to maintain a thermally comfortable sleeping environment in bedrooms, it is inevitable that energy consumption in residential buildings will be increased. In order to reduce energy use for air conditioning sleeping environments or bedrooms at nighttime, passive energy-saving measures, such as adding an air gap, may be applied to external envelopes of the bedroom. Hence, it becomes necessary to study the impacts of daytime heat gain stored in external envelopes of a bedroom on its nighttime cooling load, and the energy saving potentials of applying passive measures in external envelopes.

Therefore, a systematic research program on developing a non-convection based TAC system and on applying passive energy-saving measures to the external envelopes of a bedroom to reduce its nighttime cooling load has been carried out and the results are presented in this Thesis.

3.2. Project title

This systematic research program reported is divided into three parts: Firstly, experimentally evaluating indoor thermal comfort and indoor air quality performances when using a radiation-based TAC (R-TAC) system applied to a sleeping environment; Secondly, numerically studying the effects of design/operating parameters of a radiant panel in the R-TAC system on indoor thermal comfort and energy saving potential; Finally, numerically studying the impacts of daytime heat gain/storage of the external envelopes in a bedroom on its nighttime cooling load and passive energy-saving measures. Therefore, the systematic research program presented in this Thesis is entitled "An Experimental and Numerical Study on Improving Thermal Comfort and Energy Saving Measures in a Sleeping Environment".

3.3. Aims and objectives

The aims and objectives of the systematic research program reported in this Thesis are as follows:

1. To experimentally evaluate the operating performances of the R-TAC system applied to a sleeping environment in terms of thermal comfort control, ventilation efficiency and draft risk in an experimental setup to be specifically established;

2. To numerically investigate the effects of design/operating parameters of the radiant panel in the R-TAC system on indoor thermal comfort and energy saving, using the CFD method which will be experimentally validated, for a sleeping environment;

3. To numerically study the impacts of daytime heat gain/storage in the external envelopes in a bedroom on its nighttime cooling load, and evaluate the effects of applying passive measures including adding an air gap and ventilating the gap in a west-facing external wall in a bedroom on its total nighttime cooling load.

3.4. Research methodologies

Experimental and numerical methods are employed in this systematic research program. Firstly, experimental methods will be used to evaluate indoor thermal environment and indoor air quality in an experimental bedroom of the experimental setup when using the R-TAC system. A thermal manikin will be employed in this part of the research program. Air dry bulb temperature, air wet bulb temperature, globe temperature, surface temperature, indoor air velocity and CO_2 concentration in the experimental bedroom, and skin temperature and heat flux of the thermal manikin will be measured. The thermal environment in the experimental bedroom when using the R-TAC system, in terms of ventilation effectiveness and draft risk, will be evaluated, using the C-TAC system as a basis for comparison.

Secondly, numerical methods will be used to study the effects of design/operating parameters of the radiant panel in the R-TAC system on indoor thermal comfort and energy performance. A CFD method for predicting indoor air temperature and air flow fields will be developed and experimentally validated using the experimental results obtained in the experimental study. Then, various design/operating parameters of the radiant panel, including surface temperature and emissivity, area of the radiant panel, and the distance between the radiant panel and a bed, are varied in CFD method, so that their effects on thermal comfort control and energy performances when using the R-TAC system can be numerically evaluated.

Finally, the well-known building energy simulation software package, EnergyPlus, will be used to study the impacts of daytime heat gain/storage of the external envelopes in a bedroom on its nighttime cooling load and passive measures. A hypothetic west-facing bedroom in the subtropical Hong Kong will be used to study the effects of various passive measures such as adding an air gap to the west-facing external wall in the bedroom and ventilating the gap on reducing the thermal energy stored in the wall and consequently the nighttime cooling load.

Chapter 4

An experimental study on the performances of a radiation-based task/ambient air conditioning system applied to sleeping environments

4.1. Introduction

As people's living standard is being improved, A/C becomes a necessary provision in people's daily life. In summer months in the subtropics, air-conditioning is usually necessary to maintain a suitable indoor thermal environment in an occupied bedroom at nighttime. As a result, energy use for air-conditioning sleeping environments contributes significantly to the total annual electricity use in residential buildings, as reported in Chapter 2.

To reduce energy consumption for A/C while still maintaining a suitable level of indoor thermal comfort and IAQ, TAC systems have been developed and are widely used in office buildings. From the review presented in Chapter 2, it is seen that TAC systems used in office buildings at daytime were investigated in a number of previous studies and many benefits associated with using TAC systems revealed, such as better thermal comfort control and IAQ, reduced energy consumption, etc. On the other hand, in a sleeping environment, since a sleep person occupies a smaller space due to immobility during sleep, TAC systems are considered to be the best air conditioning method applied to a sleeping environment. Therefore, Mao et al. (2013) developed a ductless bed-based TAC system, as shown in Figure 2.8 in Chapter 2, and the study results revealed that the ductless TAC system can efficiently remove the contamination in an occupied zone by supplying fresh air directly to the zone and perform reasonably well as far as thermal comfort was concerned. However, the issue of cold draft remained unsolved when using this ductless TAC system, although using a proper supply air temperature and flow rate, as well as a proper supply vane angle may help ease off the cold draft problem. This was because that the TAC systems so far developed for sleeping environments used predominantly the convection-based method to cool an occupied zone in a sleeping environment and the vent supplying air with a high velocity and a low temperature had to be close to occupants. Therefore, cold-draft could hardly be avoided when using a convection-based TAC (C-TAC) system for a sleeping environment.

Draft can not only cause human thermal discomfort, but also result in temporary illness or even chronic diseases for occupants, as reviewed in Chapter 2. Therefore, it is urgently necessary to develop a non-convection based TAC system that can help ease off draft problem in air conditioned sleeping environments.

Currently, the use of CRCP complete with a dedicated fresh air system has been often seen in all types of buildings. In a CRCP-based A/C system, radiation is the dominant means to exchange heat with building envelopes and internal heat sources including occupants. Therefore, occupants' thermal comfort when using a CRCP-based A/C system may be greatly improved, since cooling was directly and more evenly provided to the occupants without causing cold draft.

Therefore, to address the cold-draft issue encountered in the previously developed C-TAC systems applied to sleeping environments, a radiation-based TAC (R-TAC) system that integrated the advantages of energy saving when using TAC systems and better occupants' thermal comfort control when using CRCP-based A/C systems has been proposed.

In this Chapter, an experimental study on the performances of a prototype R-TAC system in terms of thermal comfort, ventilation effectiveness and draft risk analysis, as the first part of the systematic research program, was carried out, and the study results are presented in this Chapter. Firstly, an experimental setup for the prototype R-TAC system is described, and the measurement methods and experimental conditions specified. Secondly, the thermal comfort, ventilation performances when using the prototype R-TAC system for sleeping environments were experimentally evaluated, and the evaluation results are presented. Thirdly, draft risk performance, using that of the C-TAC system as a base for comparison, and the issue of potential condensation on the radiant panel in the experimental setup are discussed.

4.2. Experimentation

4.2.1. Experimental setup

The experimental setup for the proposed prototype R-TAC system was purposely established. The photos for the experimental setup are shown in Appendix A. The setup consisted of three parts: a plant room, a simulated outdoor space and an experimental bedroom, as shown in Figure 4.1. These three parts are separately described in the following sub-sections.

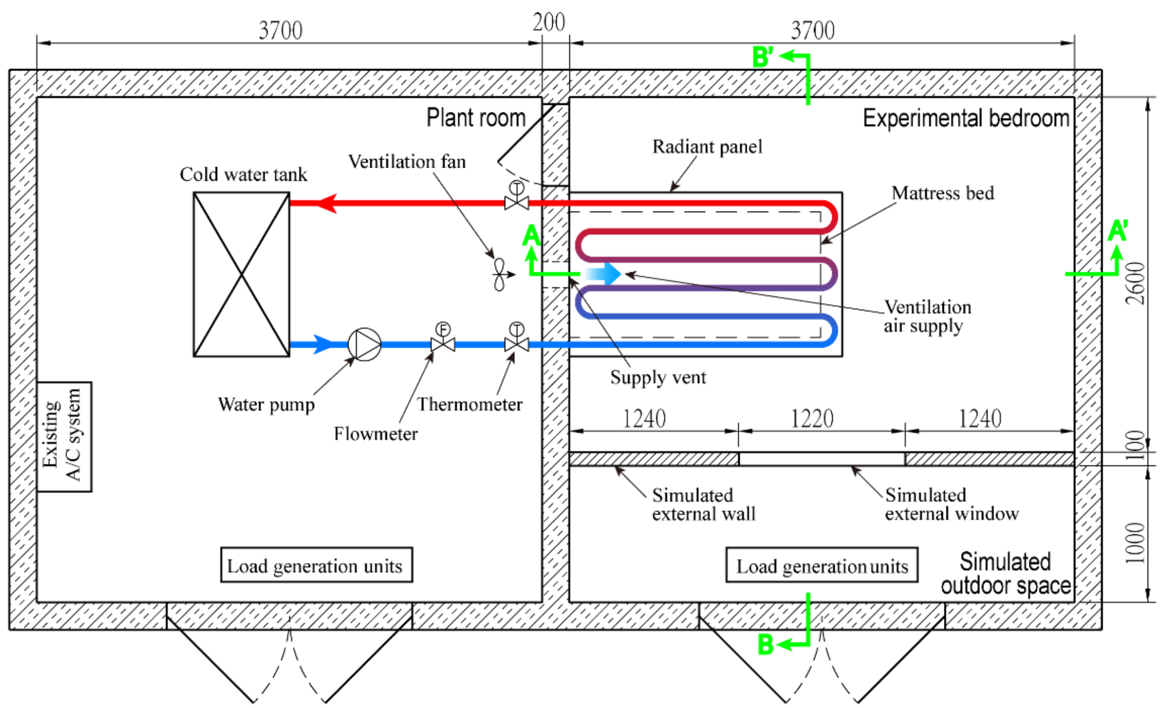


Figure 4.1 The schematics of the complete experimental setup for a prototype R-TAC system (unit: mm)

4.2.1.1. The plant room

In the plant room, a vapor compression based cold water tank to provide the required cold water to the R-TAC system was installed. Cold water at different temperatures can be provided for various experimental purposes. A circulating water pump was used to circulate cold water to a radiant panel of the R-TAC system in the experimental bedroom.

Also in the plant room, there was an existing direct expansion air conditioning (DX A/C) system and load generation units (LGUs), as shown in Figure 4.2. The DX A/C system included a variable-speed rotor compressor, a tube-louver-finned DX evaporator, an electronic expansion valve, an air-cooled tube-plate-finned condenser, and a supply air fan, as well as auxiliary air ducts. The DX A/C system was used to supply cooled and dehumidified air to the plant room during air conditioning operations. Furthermore, the sensible heat and moisture LGUs were provided in the plant room and their generation rates can be adjusted either manually or automatically using a controller accordingly on operator's programming. By jointly using the DX A/C system and LGUs, the air temperature and humidity in the plant room can be maintained at the constant values during experiments.

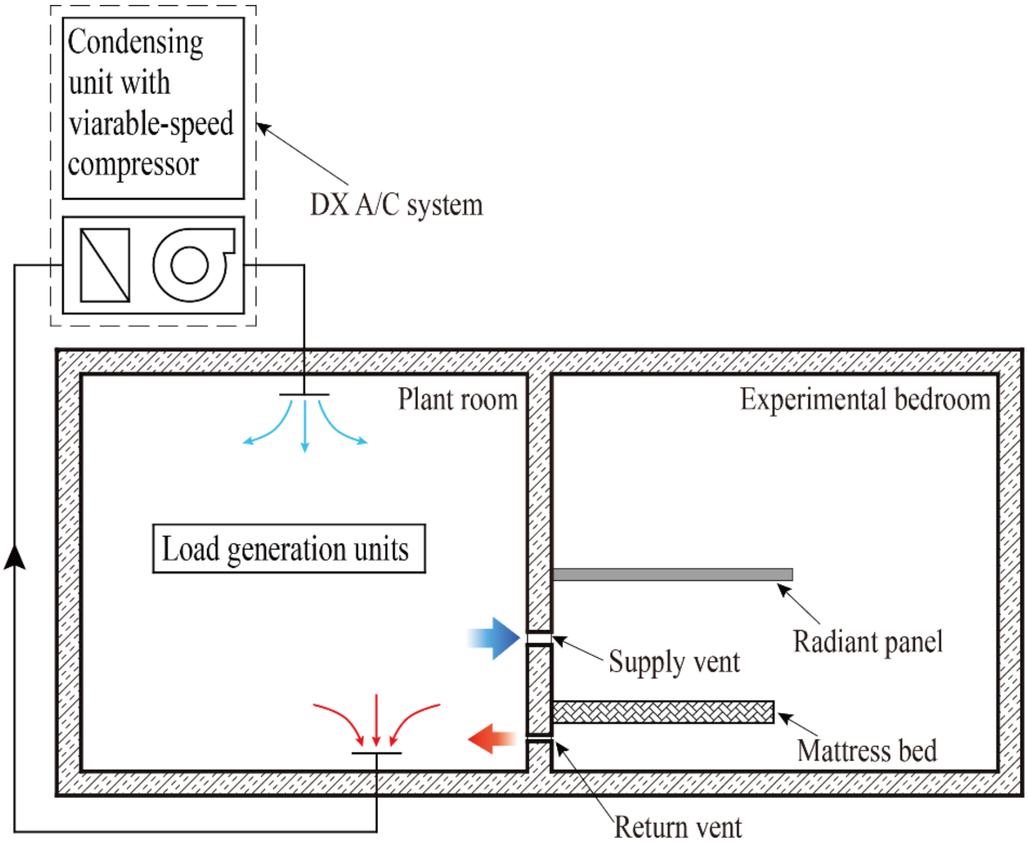


Figure 4.2 The existing DX A/C system in the plant room

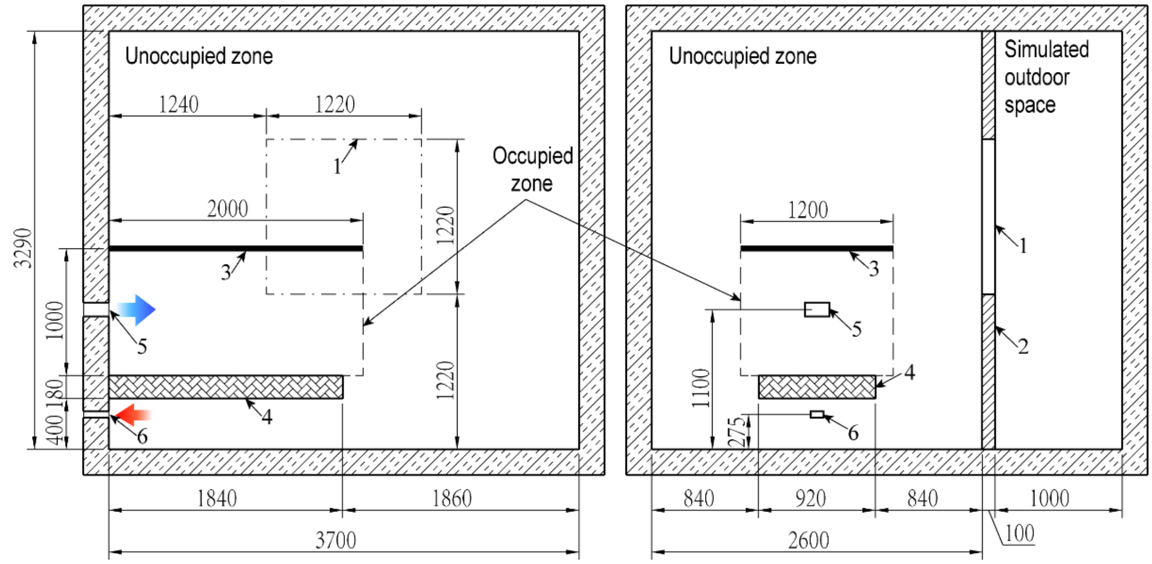
4.2.1.2. The simulated outdoor space

The simulated outdoor space was used to resemble a real outdoor thermal environment to the experimental bedroom, and a few heat generation units were placed there, so that heat transfer between the two would take place through a simulated external wall and a simulated external window of the bedroom.

4.2.1.3. The experimental bedroom

4.2.1.3.1. The layout in the experimental bedroom and its dimensions

The experimental bedroom was used to resemble a real sleeping environment where the prototype R-TAC system was installed. Two sectional views and a three-dimensional view of the experimental bedroom where the prototype R-TAC system for sleeping environments was installed, are shown in Figure 4.3 and Figure 4.4, respectively. The experimental bedroom measured at 2600 mm × 3700 mm × 3290 mm (W×D×H) was built in an existing environmental chamber and there was a simulated external wall with a simulated external window (1220 mm × 1220 mm) on the wall. Except the simulated external wall and the simulated external window, all other envelope surfaces of the bedroom can be regarded as adiabatic since they were well insulated. Inside the bedroom, on the left hand side, there was a mattress bed (1840 mm × 920 mm ×180 mm) placed at 400 mm above the floor level, as shown in Figure 4.3 and Figure 4.4. A thermal manikin was laid on the mattress bed. Between a radiant cold panel and the mattress bed, a cuboid (2000 mm × 1200 mm × 1000 mm) above the mattress bed was designated as an occupied zone (OZ). The rest of the experimental room, on the other hand, was designated as an unoccupied zone (UZ).



1. Simulated external window 2. Simulated external wall 3. Radiant panel 4. Mattress bed
5. Supply vent (190×110) 6. Return vent (100×50)

(a) A-A' sectional view (b) B-B' sectional view

Figure 4.3 Sectional views of the experimental bedroom and the prototype R-TAC system (unit: mm)

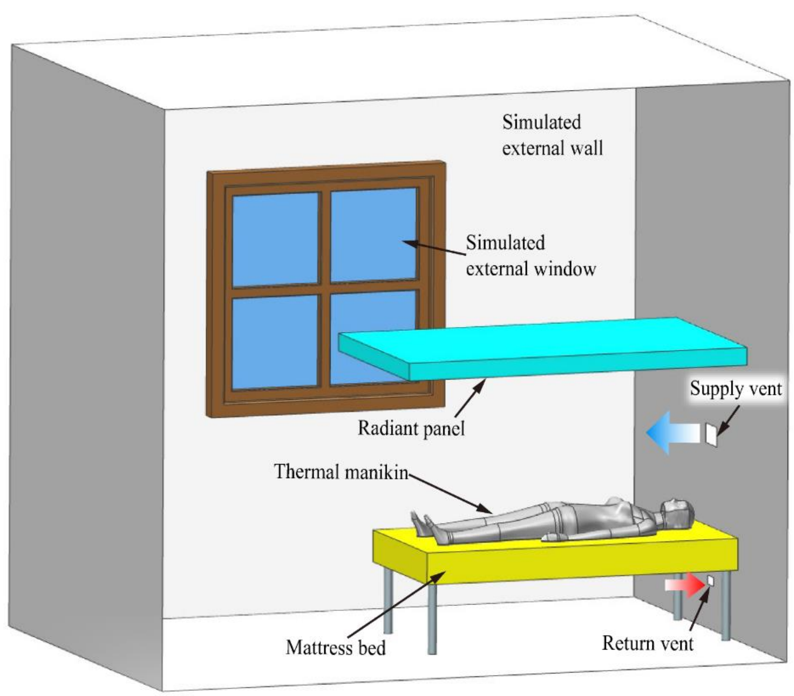


Figure 4.4 A 3-D view of the experimental bedroom and the prototype R-TAC system

4.2.1.3.2. The details of the radiant panel

The radiant panel of the R-TAC system was placed at 1580 mm above the floor level and its construction details are shown in Figure 4.5. The distance between the panel and mattress bed was at 1000 mm, sufficient for an adult to sit on the bed. For the panel itself, an aluminum foil adhered to the water pipes was used to increase heat exchange between the water pipes and an aluminum sheet in the lower part of the panel. On the other hand, in the upper part of the panel, a thermal insulation layer and an external aluminum foil were used to reduce conductive and radiative heat transfer, respectively. Therefore, except for the surface facing the thermal manikin, all other surfaces of the radiant panel were regarded as thermally insulated. Based on the rated cooling capacity of radiant panel (Jeong and Mumma 2003, 2004) and the estimated cooling load for the experimental bedroom (shown Table 4.1), the surface area of the radiant panel was determined at 2000 mm × 1200 mm, larger than the area of the mattress bed, with its output cooling capacity slightly larger than the estimated cooling load.

4.2.1.3.3. The ventilation system

There was also a ventilation system made of a supply vent (190 mm ×110 mm) and a return vent (100 mm ×50 mm), placed on the left hand side wall, at 1100 mm and 275 mm above the floor level, respectively, so that conditioned air at different states from the plant room next door can be supplied to the occupied zone for various experimental purposes. As seen from Figure 4.4, a larger supply vent which was close to a sleeping person was used to avoid cold draft.

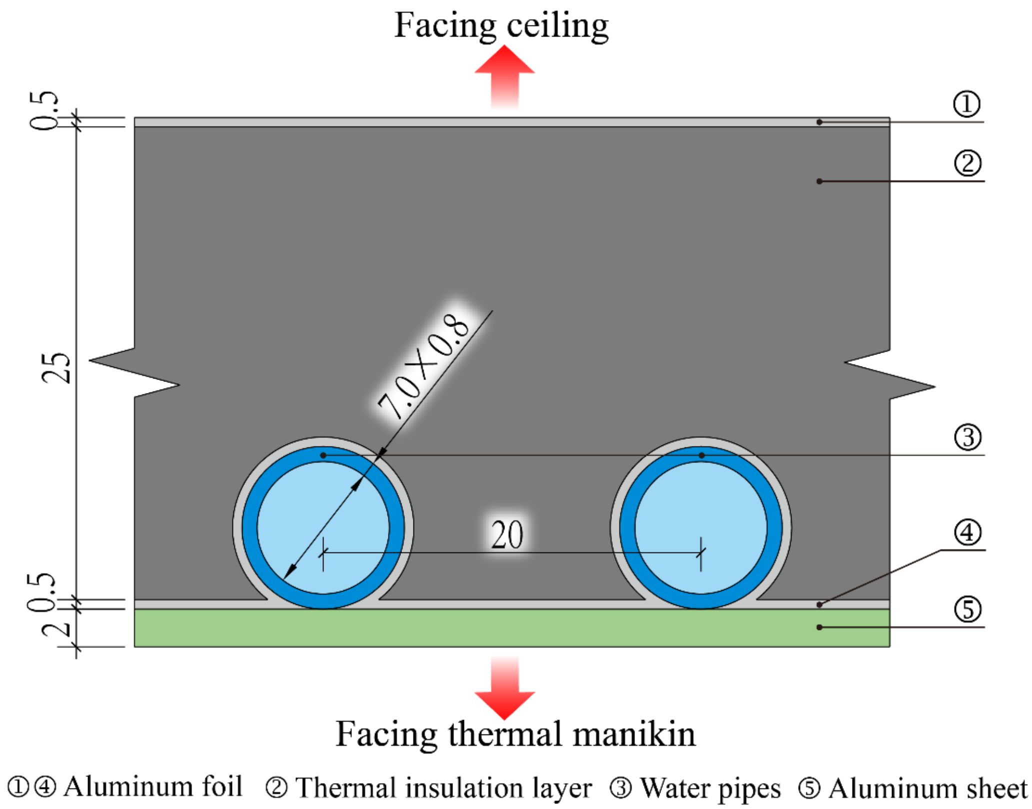


Figure 4.5 Details of components in the radiant panel components (unit: mm)

Table 4.1 Calculation of the cooling load in the experimental bedroom to determine the radiant panel area

<table>
<tr><th colspan="2">Heat gain/loss source</th><th>Simulated external wall</th><th>Simulated external window</th><th>Thermal manikin</th><th>Radiant panel</th></tr>
<tr><td colspan="2">Thickness, δ (mm)</td><td>20</td><td>2</td><td rowspan="7">—</td><td rowspan="7">—</td></tr>
<tr><td colspan="2">Thermal conductivity, λ (W/m· °C)</td><td>0.72</td><td>0.90</td></tr>
<tr><td rowspan="2">Thermal resistance[a] (m^2· °C /W)</td><td>Indoor, R_i</td><td>0.20</td><td>0.20</td></tr>
<tr><td>Outdoor, R_o</td><td>0.41</td><td>0.41</td></tr>
<tr><td rowspan="2">Air temperature (°C)</td><td>Indoor[b], t_i</td><td>23.0</td><td>23.0</td></tr>
<tr><td>Outdoor, t_o</td><td>30.0</td><td>30.0</td></tr>
<tr><td colspan="2">Calculation method</td><td colspan="2">$q = \frac{t_o - t_i}{R_i + R_o + \delta / \lambda}$</td></tr>
<tr><td colspan="2">Heat flux rate, q (W/m^2)</td><td>11.0</td><td>11.4</td><td>40.0 (ASHRAE 2013a)</td><td>-87.5[c]</td></tr>
<tr><td colspan="2">Area (m^2)</td><td>10.7</td><td>1.5</td><td>1.479</td><td>2.4</td></tr>
<tr><td colspan="2">Heat flux (W)</td><td>117.3</td><td>17.0</td><td>59.2</td><td>-210.0</td></tr>
<tr><td colspan="2">Total</td><td colspan="3">117.3+17.0+59.2-210.0=</td><td>-16.6 W</td></tr>
</table>

[a] Including the effects of convective and radiative heat transfer (ASHRAE 2013a).

[b] Indoor design air dry-bulb temperature for Hong Kong climates (Lin and Deng 2004).

[c] Total cooling capacity for a top-insulated metal radiant panel, when its surface temperature at 17 °C and surface air velocity at 0 m/s (Jeong and Mumma 2004).

4.2.1.3.4. The thermal manikin

To simulate a sleeping person, the thermal manikin, Alex, was placed on the mattress bed in a supine position, as shown in Figure 4.6. The surface area of the manikin was divided into twenty independent segments. Each segment had its own temperature sensor, heating element and controller. Consequently, the distribution of skin temperature for a human being may be approximately simulated. Previous studies demonstrated that a real human body can be considered as consisting of two concentric thermal compartments, a core compartment and a skin compartment (Stolwijk and Hardy 2010; Gagge 1971; Wissler 1964). Among the two, the temperature in the core compartment remains basically stable during sleeping while that in the skin compartment varies, based on the ambient thermal environment and internal core temperature to achieve a heat balance. Therefore, in the current study, to simulate a sleeping person, the core temperature setting was 36.5 ^{o}C for all body segments (Kreider et al. 1958), but the skin temperatures were uncontrolled, and can therefore vary in response to ambient thermal environment. There was also an artificial lung in the manikin so that the breathing of a human being may be simulated and the assessment of ventilation effectiveness in the experimental bedroom was made possible. An air transporting system of the artificial lung, which was made of two pumps and valves, controlled the frequency of manikin’s breathing, i.e., the duration of inhalation and exhalation of a simulated pulmonary ventilation. In the current study, a rate of 0.142 L/s of pulmonary ventilation for ethnic Asians was used (Niu et al. 2007). The frequency of breathing was regulated at 17 /min using two connected digital timers (Niu et al. 2007). The air was inhaled through the nose but exhaled through the mouth of the manikin. Furthermore, since the outer surface of the thermal manikin cannot dissipate moisture, a humidifier was used to resemble human moisture dissipation, releasing water vapor at 45 g/h, according to ASHRAE (ASHRAE 2013a).

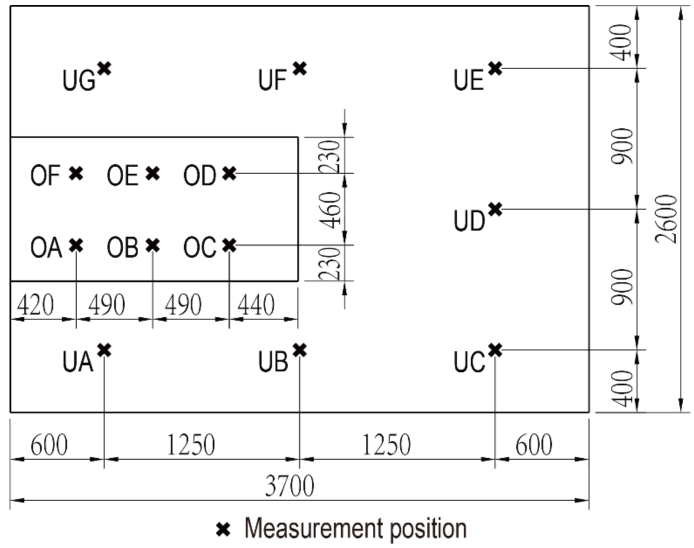
Figure 4.6 The thermal manikin used in the experiments

4.2.2. Measurement methods

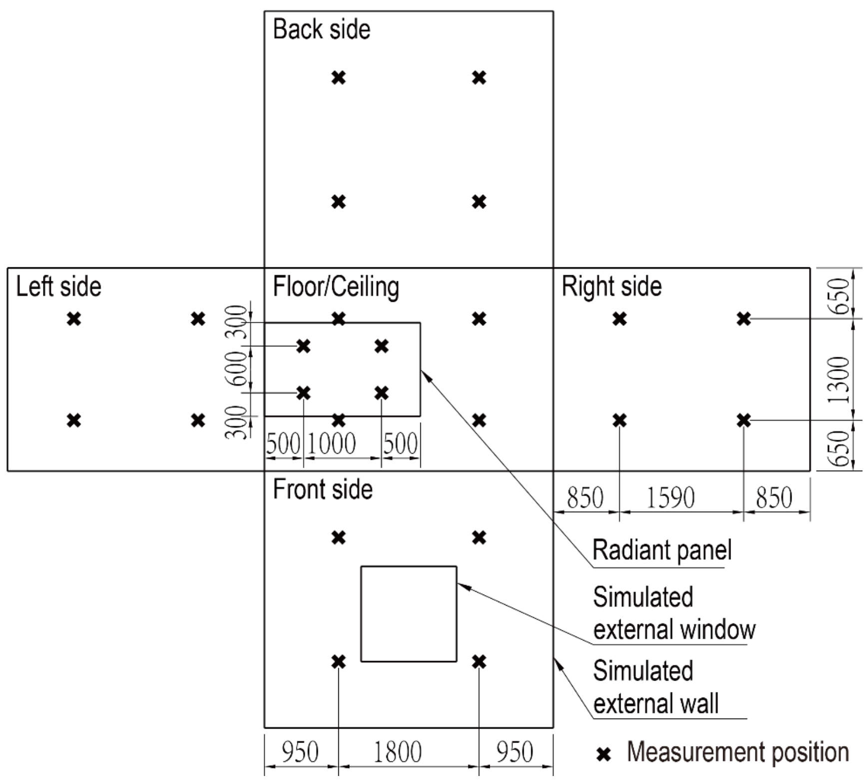
Six measurement locations in the occupied zone and seven in the unoccupied zone were identified and are shown in Figure 4.7(a), so that the thermal environments in both zones can be separately studied.

Inside the unoccupied zone, the following four parameters, air dry-bulb temperature, air wet-bulb temperature, air velocity and globe temperature for obtaining mean radiant temperature (MRT) (Bedford and Warner 1934; Thorsson et al. 2007; Vernon 1932) were measured at four different heights, 100 mm, 600 mm, 1100 mm and 1700 mm above floor level, at each of the seven locations (ASHRAE 2013c). The measured values of each parameter in the seven locations at the four heights, totally 28, were averaged to be the mean value of each of the parameters in the unoccupied zone. Inside the occupied zone, at each of the six locations, the above four thermal environmental parameters and CO_2 concentration were measured at 800 mm and 1000 mm above floor level, as shown in Figure 4.7(a). The measured values of each parameter in the 6 locations at the two heights in the occupied zone were averaged to be the mean value of each parameter in the occupied zone. In addition, for each of the envelopes and the radiant panel, Figure 4.7(b) shows four surface temperature measurement locations on the surface of radiant panel and six bedroom envelopes (including floor, ceiling, left-hand side wall, right-hand side wall, front side wall and back side wall). The measured surface temperatures in the four locations were averaged to be the mean surface temperature of each envelope surface and the radiant panel.

Air dry-bulb and wet-bulb temperatures and all surface temperatures were measured using K-type thermal couples having an accuracy of $\pm$0.4%, and air velocities an air velocity transducer having an accuracy of $\pm$4% (0.05-0.5 m/s). The globe temperatures were obtained using a standard globe thermometer, which consisted of a hollow black-painted copper sphere with a diameter of 150 mm and a thickness of 0.4 mm, and a thermometer with its bulb at the center of the sphere.



(a) For indoor thermal environment



(b) For surfaces of envelopes and the radiant panel

Figure 4.7 Measurement locations inside the experimental bedroom (Plan view, unit: mm)

4.2.3. Experimental conditions and cases

The heat gain inside the experimental bedroom came from two sources: the thermal manikin at fixed core temperature, 36.5 ^{o}C, and external envelope heat gain presented from the simulated outdoor space at a constant air temperature of 30 ^{o}C. Furthermore, previous studies (Jeong and Mumma 2003; Olesen 2008; Imanari et al. 1999) demonstrated that the surface temperature of a radiant panel should not be too low when considering condensation risk. Therefore, experimental surface temperatures of the radiant panel (t_{rp}) were determined at 17 ^{o}C, 19 ^{o}C, 21 ^{o}C and 23 ^{o}C, respectively. Outdoor air flow rate supplied to the experimental bedroom from the plant room was set at 7.5 L/s and 3.0 L/s, based on ASHRAE Standard 62.1 (ASHRAE 2013e) for a sedentary person having an activity level of 1.2 Met and the suggestion from a previous related study (Lin and Deng 2003) for a sleeping person at 0.7 Met activity level, respectively.

Table 4.2 lists all groups of experimental cases for different experimental purposes. Firstly, the purposes for Group 1 experiments were to evaluate the three environmental parameters affecting thermal comfort, i.e., indoor air temperature, MRT and air velocity, and the overall thermal comfort, at different surface temperatures of the radiant panel. In Group 2 experiments, the effects of different fresh air flow rates on ventilation effectiveness and draft risk were studied. Group 3 and 4 experiments were used to investigate the effects of different surface temperatures of the radiant panel and supply air temperatures on draft risk. Finally, the condensation issue that may exist in this R-TAC system was discussed using the results obtained in Group 5 experiments.

Table 4.2 Groups of experimental cases

Group	Case no.	Surface temperature of radiant panel (^{o}C)	Fresh air temperature (^{o}C)	Fresh air flow rate (L/s)	Relative humidity of fresh air (%)	Bedding insulation
1	1.1	17	23	7.5	50	None
	1.2	19	23	7.5	50	None
	1.3	21	23	7.5	50	None
	1.4	23	23	7.5	50	None
	1.5	17	23	7.5	50	Full-slip
	1.6	19	23	7.5	50	Full-slip
	1.7	21	23	7.5	50	Full-slip
	1.8	23	23	7.5	50	Full-slip
2	2.1	17	23	3	50	None
	2.2	17	23	7.5	50	None
3	3.1	19	19	7.5	50	None
	3.2	19	21	7.5	50	None
	3.3	19	23	7.5	50	None
4	4.1	17	21	7.5	50	None
	4.2	19	21	7.5	50	None
	4.3	21	21	7.5	50	None
5	5.1	17	23	7.5	40	None
	5.2	17	23	7.5	50	None
	5.3	17	23	7.5	60	None

4.3. Results and analysis for experimental results in Groups 1 and 2 (for thermal comfort and ventilation effectiveness)

In this experimental study, the measured values in all measurement positions were recorded by two data loggers at an interval of 5 seconds. During experiments, after the air temperature inside the experimental bedroom and the surface temperatures of the envelopes became stabilized, the averaged value in each of the measurement positions in all experimental cases was obtained by averaging 10 minutes' measured values, for a specific measured parameter.

4.3.1. Group 1 experiments for thermal comfort evaluation

In this section, thermal comfort in both the occupied zone and unoccupied zones are evaluated using the PMV index.

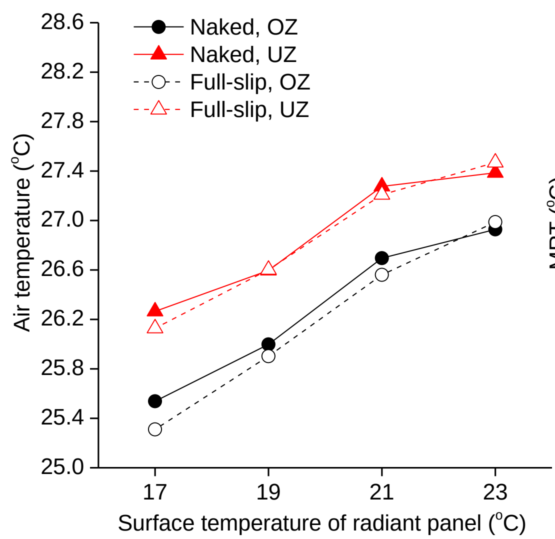
4.3.1.1. Indoor air temperature and MRT

When a C-TAC system was applied, indoor air temperature was usually regarded as the only index to evaluate the indoor thermal environment. However, MRT is also a very important parameter influencing human thermal comfort. Even in a space served by a convection-predominant air conditioning system, the radiative heat loss from an occupant accounted for about 60% of the total (Gao et al. 2006; Pan et al. 2011), not to mention that in the R-TAC system. MRT, or t_r, can be obtained by (Thorsson et al. 2007; Kuehn et al. 1970):

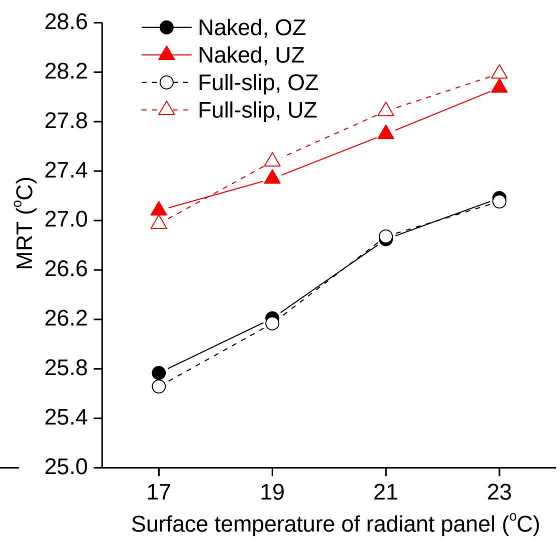
$$t_r = \left[\left(t_g + 273.15 \right)^4 + \frac{1.1 \times 10^8 v^{0.6}}{\varepsilon D^{0.4}} \times \left(t_g - t_a \right) \right]^{1/4} - 273.15 \tag{4.1}$$

where t_g is the globe temperature at a measurement position (°C), t_a the measured air temperature at a measurement position (°C), v the measured air velocity (m/s), D the globe diameter (mm), and ε the globe emissivity.

The averaged air temperatures and MRTs in Group 1 experimental cases in both the occupied and the unoccupied zones, at different bedding insulations and different surface temperatures of the radiant panel, are shown in Figure 4.8. It can be seen that: a) with an increase in radiant panel surface temperature, both air temperature and MRT would be increased; b) the differences between indoor air temperature/MRT at the naked condition and those at the full-slip condition were small, with the maximum temperature difference of 0.2 ^{o}C, suggesting that different beddings did not significantly affect both indoor air temperature and MRT; c) the differences between air temperature/MRT in the two zones were 0.5-0.8 ^{o}C for indoor air temperature and 0.9-1.3 ^{o}C for MRT, demonstrating that radiative heat transfer would play a more important role when using the R-TAC system, and that a great potential of energy saving when using the R-TAC system can be expected as less energy was needed to cool the unoccupied zone.



(a)



(b)

Figure 4.8 Indoor air temperatures and MRTs in Group 1 experimental cases

4.3.1.2. Air velocity

As air velocity in the experimental bedroom was mainly influenced by fresh air supply, there was no remarkable difference in measured air velocity among all cases in Group 1. Therefore, only the results from Case 1.1 are selected for the averaged air velocities in the two zones and air velocity distribution in the occupied zone, as shown in Figure 4.9. As seen, the averaged air velocity in the two zones was very low, at below 0.05 m/s. Also, in the occupied zone, all the measured air velocities at the 12 measurement positions were not more than 0.15 m/s, which can well meet the air velocity requirement for thermal comfort suggested by ASHRAE (ASHRAE 2013d). It is further noted that measured air velocity at 800 mm above the floor level in the occupied zone tended to be higher than that at 1000 mm above the floor level.

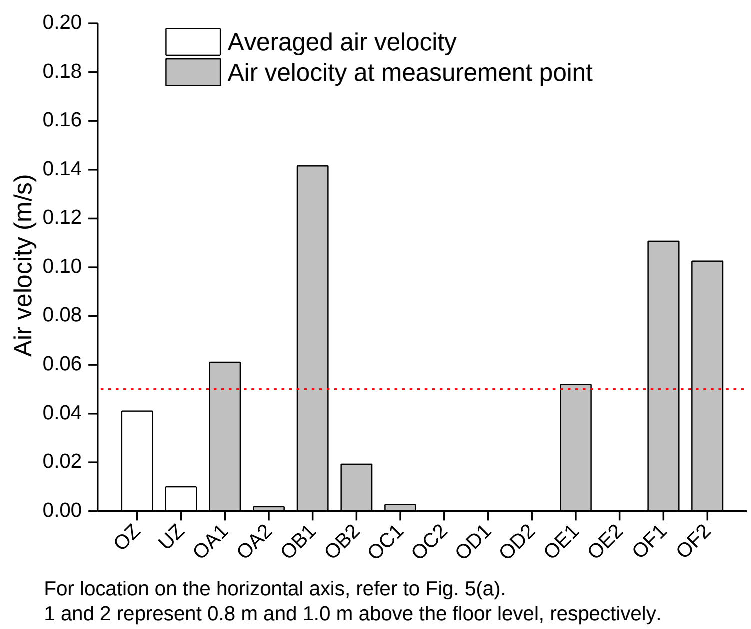


Figure 4.9 Averaged air velocities in the two zones and air velocity distribution in the occupied zone for experimental Case 1.1

4.3.1.3. Thermal comfort evaluation

A previously modified PMV index applicable specifically to sleeping environments, which was developed based on the well-known Fanger's thermal comfort model (Lin and Deng 2008a), was used to evaluate the thermal comfort in the experimental bedroom when using the R-TAC system, as follows:

$$\begin{aligned} PMV = 0.0998\left\{40 - \frac{1}{R_t}\left[\left(34.6 - \frac{4.7t_r + h_c t_a}{4.7 + h_c}\right) + 0.3762\left(5.52 - p_a\right)\right]\right\} \\ -0.0998\left[0.056\left(34 - t_a\right) + 0.692\left(5.87 - p_a\right)\right] \end{aligned} \tag{4.2}$$

where R_t is the total resistance of a bedding system, which is the thermal resistance to sensible heat transfer between a sleeping person and his/her surroundings provided by a bedding system, including bed, sleepwear, bedding and its percentage coverage over a human body ($m^2\,^oC$ /W), 0.1519 $m^2\,^oC$ /W (0.98 clo) at naked state, and 0.24335 $m^2\,^oC$/W (1.57 clo) at full-slip state, respectively, obtained from a previous study (Lin and Deng 2008b); h_c convective heat transfer coefficient (W/(m^2 ^oC)), h_c=2.7+8.7$v^{0.67}$ for 0.15<v<1.5 m/s, h_c=5.1 for 0<v≤0.15 m/s; p_a water vapor pressure in ambient air (kPa).

The evaluated PMV values based on Group 1 experimental results in the occupied/unoccupied zones with two bedding insulations (i.e., the total insulation of a bedding system) at different radiant panel surface temperatures are shown in Figure 4.10. As seen, in both zones, the PMV values were increased with the increase in surface temperature of the radiant panel. On the other hand, as shown in Figure 4.8, although different bedding insulations had little influence on the thermal environment inside the bedroom, it did make a difference in the evaluated PMV values. For example, at the t_{rp}=17 °C condition, PMV was increased from -3.07 at the naked condition to -1.17 at the full-slip condition. That was to say that the predicted mean thermal sensation voted was changed from "cold" at the naked condition to "slightly cool" at the full-slip condition. Furthermore, the difference in evaluated PMV values between the two zones, with the same bedding insulation, was significant, ranging from 0.39 to 0.73, and the difference was gradually narrowed down as the surface temperature of the radiant panel was increased.

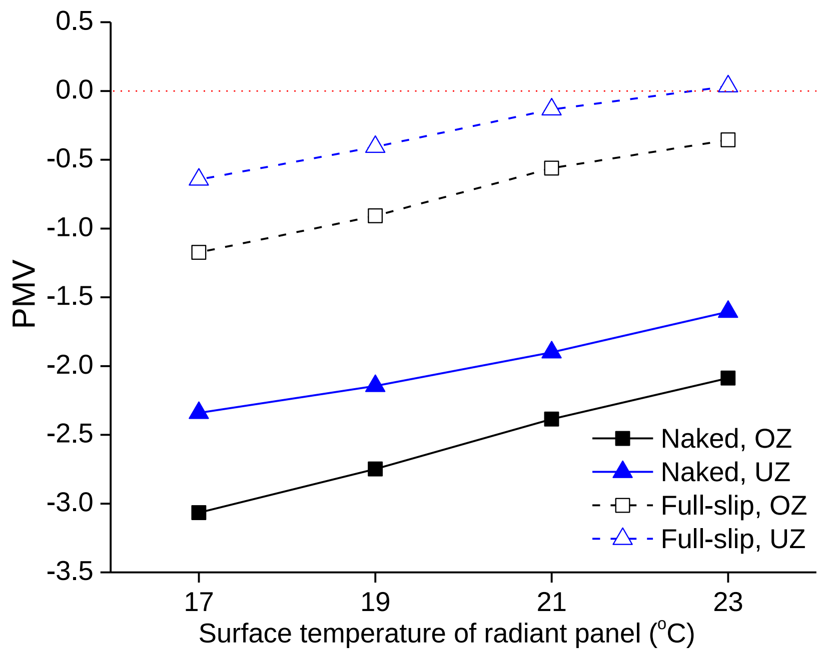


Figure 4.10 The evaluated PMV values based Group 1 experimental results in the occupied/unoccupied zones with two bedding insulations at different surfaces temperatures of the radiant panel

4.3.2. Group 2 experiments for ventilation performance evaluation

In this part of the study, since the breathing zone of a sleeping person was within the occupied zone, the ventilation performances in the occupied zone when using the R-TAC system were evaluated using ventilation effectiveness index (VE):

$$VE = \frac{C_r - C_s}{C - C_s} \tag{4.3}$$

Where C_r is the CO_2 concentration in return air, C_s the CO_2 concentration in supply air, and C the CO_2 concentration at a measurement position. VE reflects the effectiveness of contaminant removal from an occupied zone. The higher the *VE*, the better the ventilation performance.

Figure 4.11 shows the measured ventilation effectiveness in Group 2 experiments, at two different heights in the occupied zone at two different fresh air flow rates (7.5 L/s and 3.0 L/s). The following can be observed from the Figure: a) the ventilation effectiveness at 7.5 L/s fresh air flow rate was better than that at 3.0 L/s fresh air flow rate; b) the *VE* values at the two different heights were similar. Furthermore, Figure 4.12 shows the evaluated *VE* values at three regions near three different body segments of the manikin in the occupied zone, by averaging the *VE* values at four measurement positions near head, pygal and feet of the manikin, respectively. It can be seen that the ventilation effectiveness in the region near head was higher than that near the pygal and the feet at the two fresh air flow rates, because the supply vent for fresh air was close to the head of the manikin.

In addition, when comparing the *VE* value of 1.37 at 7.5 L/s fresh air near head obtained in the current study with that of 0.61 at 15 L/s fresh air in a previously studied C-TAC system with a supply vent (570 mm × 210 mm) installed at the height of 1.1 m above the floor level (Mao et al. 2013), a better ventilation effectiveness was demonstrated. Although the experimental settings in these two TAC systems were slightly different, it may indicate that, when using a C-TAC system, the fresh air was supplied to the non-breathing zone, but the fresh air in an R-TAC system was directly delivered to the breathing zone.

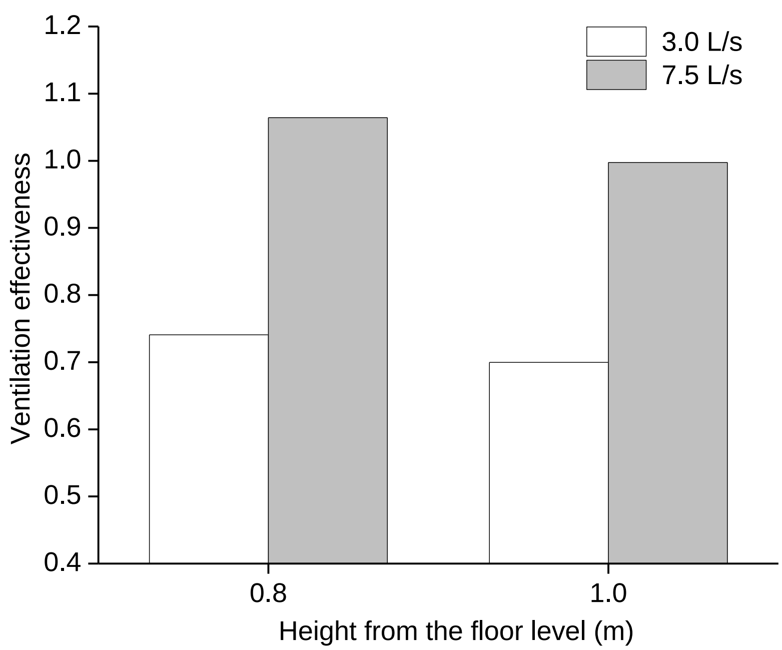


Figure 4.11 Measured ventilation effectiveness based on Group 2 experimental results at two different heights at two different fresh air flow rates

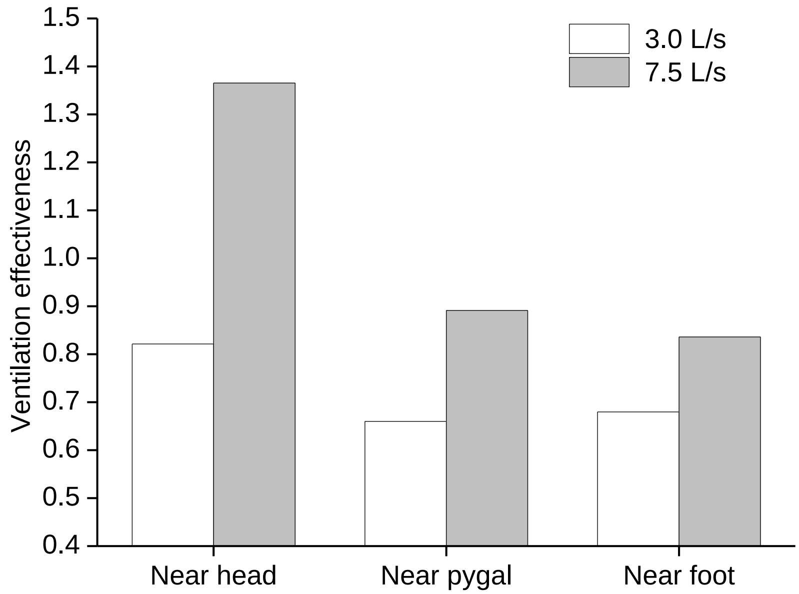


Figure 4.12 Measured ventilation effectiveness at three regions near three different body parts of the manikin at two different fresh air flow rates

4.4. Results of the experimental cases in Groups 2-5 and their related discussions (for draft risk and potential condensation risk)

4.4.1. Groups 2, 3 and 4 experimental results for comparing draft risk performances when using an R-TAC system with that when using a C-TAC system

Draft is an unwanted local cooling for a human body caused by air movement. As mentioned previously, the purpose of developing this prototype R-TAC system was to address the issue of cold draft encountered in a C-TAC system. Therefore, in this Section, the comparison between the draft risk performance in the occupied zone when using the R-TAC system and that when using the C-TAC system is reported.

Draft risk (DR) is a draft rating index used to predict the percentage of occupants feeling draft (International Standard ISO 7730; ASHRAE 2013d):

$$\begin{cases} DR = (34 - t_a)(v - 0.05)^{0.62}(0.37vT_u + 3.14) \\ v = 0.05 \text{ m/s}, \text{ when } v \leq 0.05 \text{ m/s} \\ DR = 100\%, \text{ when } DR > 100\% \end{cases} \tag{4.4}$$

where T_u is the turbulence intensity and may be evaluated at 40% when using mixing ventilation (ASHRAE 2013c). ASHRAE Standard 55 (ASHRAE 2013d) suggests that the *DR* value in each measurement position should be less than a permissible value of 20%.

The averaged DR values for the occupied zone from Groups 2, 3 and 4 experimental cases were obtained by averaging the DR values at the 12 measurement positions shown in Figure 4.7(a) at different surface temperatures of the radiant panel (t_{rp}), supply air temperatures (t_s) and supply air flow rates (Q_s), and are shown in Figure 4.13. It can be seen that the averaged DR values at all conditions when using R-TAC system were below 2.0%, significantly lower than the DR values of 14-22% obtained in the previous study (Mao et al. 2014a) where a C-TAC system was used. This was because when the R-TAC system was used, only fresh air at a higher temperature but also a smaller flow rate was provided, as compared to a higher supply air flow rate and a lower supply temperature used in a C-TAC system. Finally, as seen from Figure 4.13(c), draft risk was not remarkably affected by the radiant panel surface temperature, since it was only affected by the air movement inside the experimental bedroom.

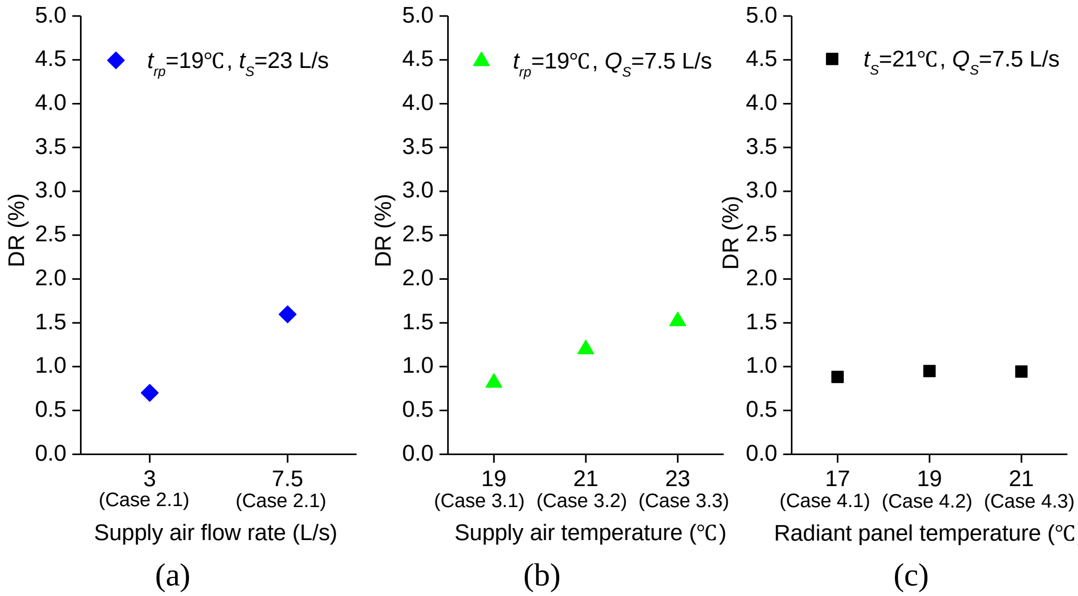
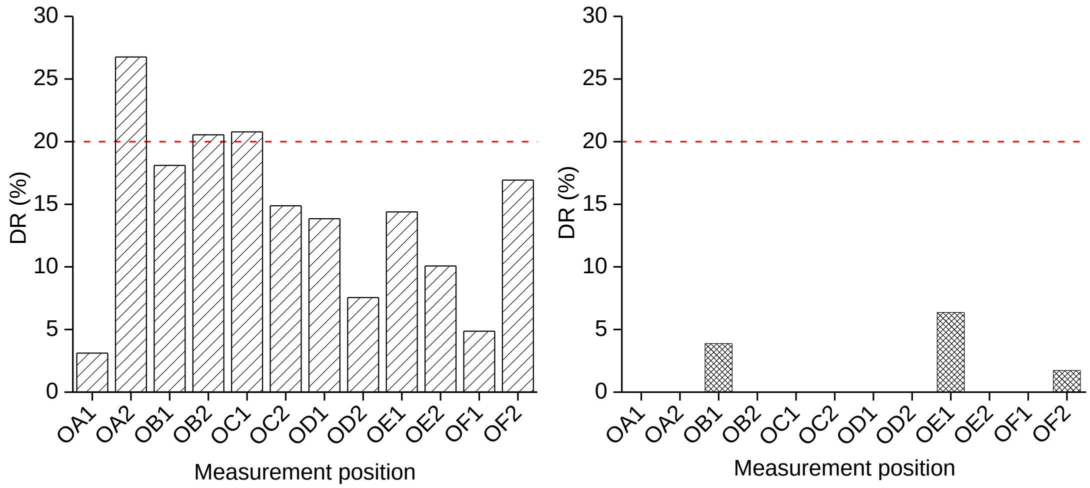


Figure 4.13 Averaged DR values in Group 2, 3 and 4 experiments in the occupied zone when using the R-TAC system

To further examine the draft risk performance when using R-TAC system, in the occupied zone, the measured DR values at 12 positions were compared with that when using a C-TAC system. Figure 4.14 shows that both the measured DR values in Case 2.2 of the current study and that in a study case of a previous study of a C-TAC system (Mao et al. 2014a). The evaluated PMV values in the occupied zone in the two cases were approximately the same, at -3.07 and -3.03 (Mao et al. 2015), respectively. However, as shown in Figure 4.14(a) for the results in the previous study, its DR values ranged from 3.1% to 26.8%, with the DR values at 3 measurement positions higher than the permissible value of 20%. On the other hand, in Figure 4.14(b) for the results of Case 2.2 in the current study, the highest DR value was only at 7%. Therefore, using the R-TAC system can effectively avoid the cold draft problem encountered when using a C-TAC system.



(a) t_s=23 ^oC, Q_s=50 L/s, for the C-TAC system in a previous study (Mao et al. 2014a)

(b) t_{rp}=17 ^oC, t_s=23 ^oC, Q_s=7.5 L/s, for the R-TAC system in the current study (Case 2.2)

Figure 4.14 Evaluated DR values at each of the measurement positions in the occupied zone in both a previous study (Mao et al. 2014a) and the current study

4.4.2. Results of experimental cases in Group 5 on potential condensation risk

Generally, condensation on a radiant panel would occur when its surface temperature is lower than the dew point temperature of the air surrounding the panel. In Figure 4.15, the averaged dew point temperatures in the occupied zone obtained in the experimental cases in Group 5 are shown. As seen, the higher the relative humidity of the fresh air supply, the higher the average dew point temperature and relative humidity in the occupied zone.

It is also seen from the Figure that the averaged dew point temperature in the occupied zone in Case 5.3 was actually higher than the surface temperature of the radiant panel at 17 ^{o}C, suggesting the risk of condensation when using an R-TAC system, and condensation was also observed in Case 5.3. Nonetheless, the problem of potential condensation on a panel surface may be resolved by raising the panel surface temperature, or using a lower fresh air supply humidity. When a higher panel surface temperature is intended, a larger radiant panel surface area may correspondingly be expected to compensate the loss in cooling capacity due to a higher panel surface temperature.

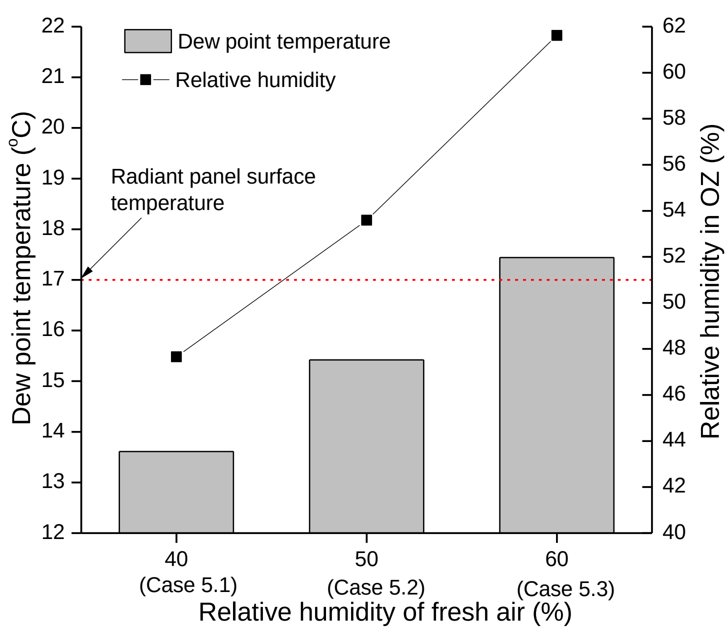


Figure 4.15 Averaged dew point temperature and relative humidity in the occupied zone for Group 5 experiments at different relative humidities of fresh air

4.5. Conclusions

To address the cold draft issue encountered when using the previously developed C-TAC systems for use in sleeping environments, an R-TAC system was developed and its performances experimentally evaluated. Experimental setup and conditions, and measurement methods are firstly presented, and experimental results of performance evaluation from five experimental Groups secondly reported and analyzed. The study results showed that the differences between air temperature/MRT in the unoccupied zone and those in the occupied zone were significant demonstrating that a great potential of energy saving when using the R-TAC system can be expected as less energy was needed to cool the unoccupied zone. A very low level of air velocity in the occupied zone was maintained. The difference between the evaluated PMV values in the occupied zone and that in the unoccupied zone with the same bedding insulation was also significant. In addition, using the R-TAC system could effectively avoid the cold draft problem encountered when using the previously developed C-TAC system (Mao et al. 2013; Mao et al. 2014b; Mao et al. 2014a), since the majority of space cooling load was taken care by the radiant panel, with only a very small amount of fresh air being provided. Finally, the problem of potential condensation on the panel surface has been observed, but may be resolved by raising the panel surface temperature, or using a lower fresh air supply humidity. However, in this experimental study, due to its experimental nature, fixed values of a number of design/operating parameters for the R-TAC system, such as the size and the surface emissivity of the radiant panel, had to be used. Given that these parameters may have significant impacts on the operating performance of the R-TAC system, a follow-up numerical study on the effects of varying design/operating parameters of the radiant panel on the operating performance of the R-TAC system was carried out and the study results are presented in the next chapter.

Chapter 5

A numerical study on the effects of design/operating parameters of the radiant panel in a radiation-based task air conditioning system on indoor thermal comfort and energy saving for a sleeping environment

5.1. Introduction

From the experimental study reported in Chapter 4, when using the R-TAC system, the differences between air temperature/MRT in an unoccupied zone and those in an occupied zone were remarkable, demonstrating that a great energy saving potential can be expected. The ventilation effectiveness in the region near the head of the manikin was higher than that near the pygal and the feet. Furthermore, at the same level of PMV, draft risk in the occupied zone when using the R-TAC system was considerably lower than that when using the C-TAC system.

For this R-TAC system, it can be understood that the heat transfer between a sleeping person and the cold radiant panel is mainly via radiation, which is greatly affected by the following four panel design/operating parameters: surface temperature, emissivity and area of the panel and distance between the bed and the panel. However, in the experimental study reported in Chapter 4, due to the nature of an experimental study, for three of the four design parameters, i.e., surface emissivity, surface area and the distance, fixed values of 0.5, 2.4 m^2 and 1000 mm, respectively, were used. For the operating parameter of panel surface temperature, only four fixed values of 17 ^oC, 19 ^oC, 21 ^oC and

23 ^{o}C, were used. Therefore, in order to optimize the R-TAC system for the best possible operating performance in terms of thermal comfort control and energy saving, it was necessary to further study the operating performance of the R-TAC system when the values of the above four design/operating parameters were varied. On the other hand, CFD based numerical methods have been widely used in optimizing the design of A/C systems and in predicting air flow pattern and temperature distribution inside an air conditioned bedroom or around a human body, as illustrated in Chapter 2.

Therefore, a follow-up CFD based numerical study on the effects of varying the surface temperature and emissivity, area of the radiant panel, and the distance between a radiant panel and a bed when using the R-TAC system on indoor thermal comfort and energy saving in sleeping environments, as the second part of the systematic research program, was carried out, and the study results are reported in this Chapter. Firstly, the details of a geometry model and the CFD method, as well as the boundary conditions and the study cases in the current numerical study are presented. Secondly, the results of validating the CFD method by comparing the predicted air temperatures, velocities and surface temperatures with measured data obtained in the experimental study at a fixed design/operating condition as reported in Chapter 4 are reported. This is followed by presenting the results of the numerical study on the effects of varying design/operating parameters of the radiant panel on thermal comfort and energy saving when using the R-TAC system for sleeping environments. Finally, a conclusion is given.

5.2. Methodology

5.2.1. Geometry model

A geometry model for the experimental bedroom used in the experimental study was built. The bedroom was located inside an environmental chamber, which also included a plant room and a simulated outdoor space, as shown in Figure 4.1 in Chapter 4. The details of the experimental bedroom, the plant room and the simulated outdoor space are described in Chapter 4.

5.2.2. CFD method

A commercial CFD code (ANSYS Fluent 2010) was used to predict the air flow and heat transfer inside the experimental bedroom. For the bedroom under consideration and the temperature difference expected, the Rayleigh number was greater than 10^{10}, suggesting that turbulence was expected. A SST turbulence model would perform better than both a $k-\varepsilon$ model and a $k-\omega$ model in predicting air temperature and velocity fields inside a bedroom, as illustrated in Chapter 2, and was therefore used in this numerical study. Since natural convection occurred in most part of the experimental bedroom, the Boussinesq approximation hypothesis was used for the buoyant force term (ANSYS Fluent 2010; Wilcox 1998). The SIMPLE algorithm was used with a second order scheme for the convective terms. The steady state, viscous and 3D governing equations for the flow field inside the experimental bedroom are given as Equations (2.3) - (2.6) in Chapter 2.

5.2.3. Boundary conditions and numerical cases

The temperature for the simulated outdoor space was set at a constant value of 30 ºC, which is the averaged outdoor air temperature at nighttime in Hong Kong according to ASHARE weather data (ASHRAE 2013a). Outdoor air flow rate supplied to the experimental bedroom from the plant room was set at 7.5 L/s, at a fixed temperature of 23 ºC and a fixed relative humidity of 50%. The boundary conditions for the numerical study are listed in Table 5.1.

Table 5.1 Boundary conditions used in the numerical study

Boundary	Boundary conditions
External wall	Fixed temperature: 30 ºC, emissivity: 0.9
External window	Fixed temperature: 30 ºC, emissivity: 0.94
Floor	Adiabatic, emissivity: 0.3
Other walls	Adiabatic, emissivity: 0.07
Mattress bed	Adiabatic, emissivity: 0.9
Thermal manikin	Fixed temperature: 36.4 ºC, emissivity: 0.9, thermal resistance: 0.054 $m^2 \cdot K/W$
Radiant panel surface facing manikin	Varying
Radiant panel surface facing ceiling	Adiabatic, emissivity: 0.07
Supply vent	Air temperature: 23 ºC, air flow rate: 7.5 L/s
Return vent	Pressure outlet

In the current numerical study, instead of using a fixed distance between the panel and the bed, and a fixed surface area of the panel, three different distances (D) and two different panel areas (L×W) shown in Figure 5.1 were used: D=600 mm, 800 mm and 1000 mm (used in the experiment); L×W=2000 mm × 1200 mm (used in the experiment) and 1840 mm × 920 mm (the same dimension with the mattress bed), respectively. These three distances are marked as “1”, “2” and “3”, respectively, and the two areas as “L” and “S”, respectively, for easy identification in this Chapter. Consequently, six different geometry settings were resulted in, and marked as “L1”, “L2”, “L3”, “S1”, “S2” and “S3”, as shown in Table 5.2. Furthermore, considering condensation risk (Jeong and Mumma 2003; Olesen 2008; Imanari et al. 1999), the same surface temperatures of the radiant panel (t_{rp}) at 17 ^oC, 19 ^oC, 21 ^oC and 23 ^oC, as those in the experimental study reported in Chapter 4, were used in the current numerical study. For the panel surface facing the thermal manikin, its emissivity (ε) was set at 0.1 (for metal materials), 0.5 and 0.9 (for common building materials), respectively.

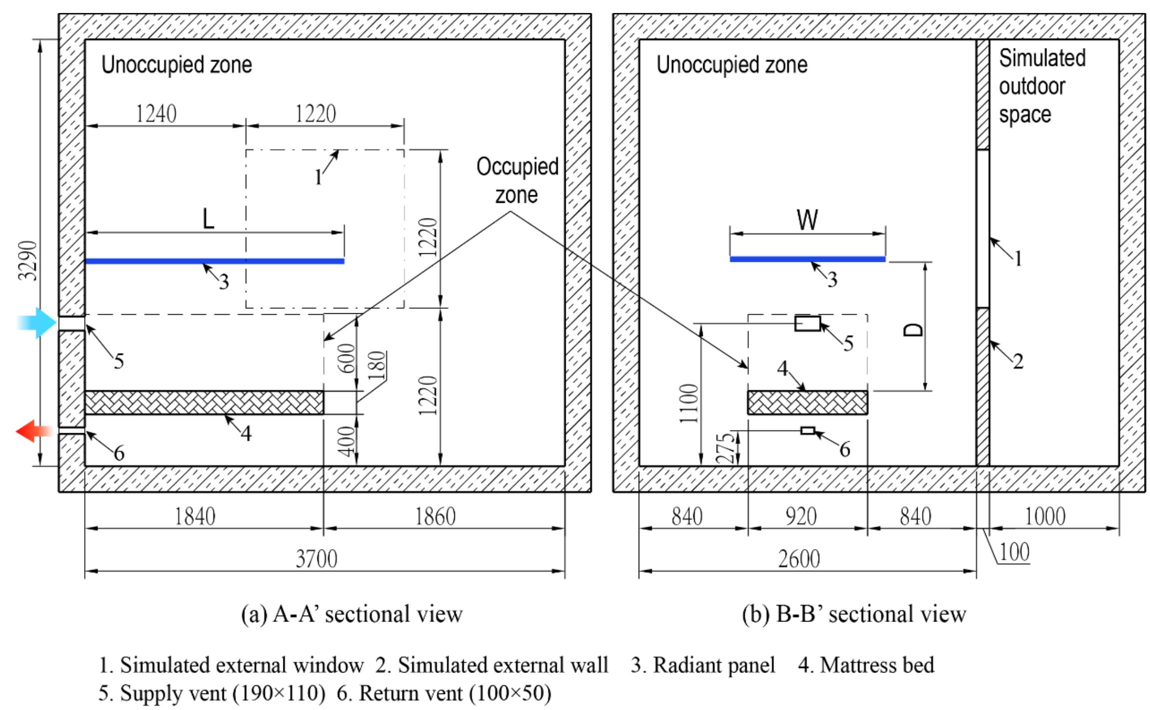


Figure 5.1 Sectional views of the experimental bedroom with different panel settings (unit: mm)

With the above settings, different groups of numerical cases for different purposes were obtained and are shown in Table 5.2. Firstly, the only case in Group 1 or Case L3.0, where all settings were the same as those used in the experimental study, was used to validate the CFD method. Secondly, the cases in Groups 2-5 and 8-11 were used to investigate the effects of different surface temperatures and areas of the panel, and the distance between the bed and the panel on thermal comfort and energy saving when using the R-TAC system. Lastly, the cases in Groups 3, 6 and 7 were used to study the effects of different emissivity of the radiant panel on indoor thermal comfort and energy saving.

Table 5.2 Numerical study cases

Group	Panel area: L×W (mm×mm)	t_{rp} (°C)	ε	D (mm)		
				600	800	1000
1 (for validation)	Large (2000×1200)	17	0.5			L3.0
2		17	0.9	L1.1	L2.1	L3.1
3		19	0.9	L1.2	L2.2	L3.2
4		21	0.9	L1.3	L2.3	L3.3
5		23	0.9	L1.4	L2.4	L3.4
6		19	0.1	L1.5	L2.5	L3.5
7		19	0.5	L1.6	L2.6	L3.6
8	Small (1840×920)	17	0.9	S1.1	S2.1	S3.1
9		19	0.9	S1.2	S2.2	S3.2
10		21	0.9	S1.3	S2.3	S3.3
11		23	0.9	S1.4	S2.4	S3.4

Table 5.3 The number of meshes for two zones at different geometry settings (unit: million)

		Occupied zone	Unoccupied zone	Total
Mesh independence		0.82	1.13	1.95
		0.99	1.45	2.44
		1.29	1.75	3.04
Geometry settings	L1	1.05	1.36	2.41
	L2	0.99	1.42	2.41
	L3	0.99	1.45	2.44
	S1	1.05	1.28	2.33
	S2	0.99	1.36	2.35
	S3	0.99	1.32	2.31

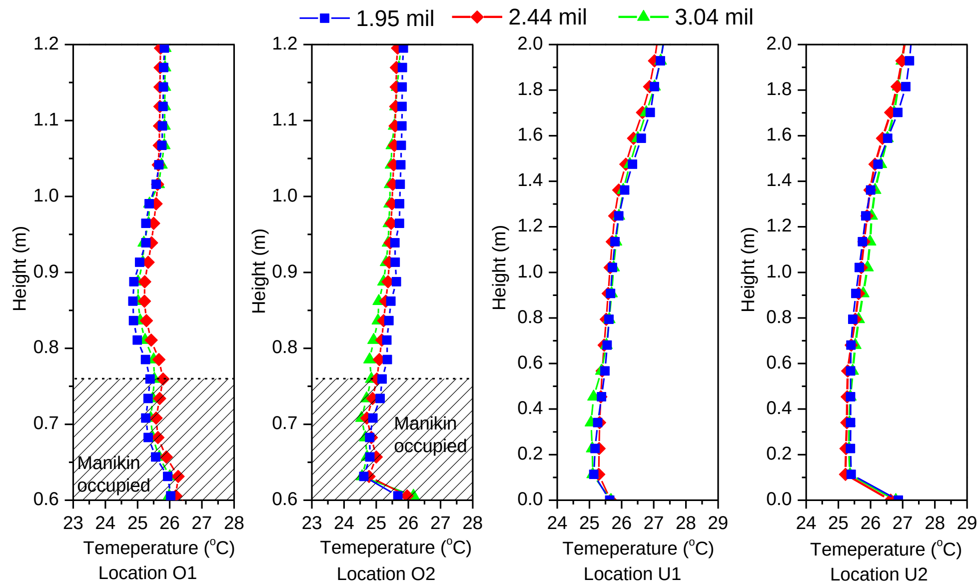
Due to the complicated geometry of the manikin, grids were separately generated for an occupied zone with unstructured grids and an unoccupied zone with structured grids. 0.4 mm and 1 mm were used for the heights of the first layer mesh near both thermal manikin surfaces and wall surfaces, respectively, for better computational results. Table 5.3 lists the number of meshes of the two zones for examining mesh sensitivity and for different geometry settings.

5.3. Validation of the CFD method (Case L3.0)

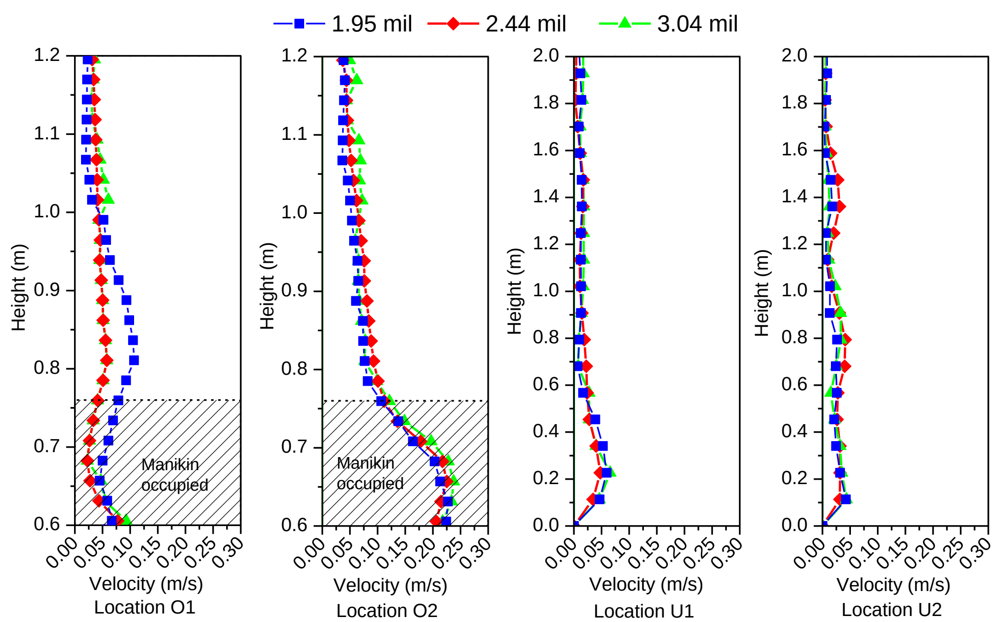
Following the recommended validation procedures and criteria in a previous study (Chen and Srebric 2002), since the flow mechanism and heat transfer characteristics in all the cases were similar (Zhai et al. 2007), the validation of the CFD method only needed to be done for one of the cases, and therefore in this numerical study, Case L3.0 was used to validate the CFD method. In Case L3.0, all the settings were the same as that used in the experimental study. Hence, the CFD method was validated by comparing the numerical results for Case L3.0 with those obtained in the experimental study. In carrying out the validation, the measured and simulated air temperatures and velocities at two points inside the occupied zone (O1 and O2), and two outside the occupied zone (U1 and U2), and the measured and simulated surface temperatures of the six envelopes of the bedroom were compared.

5.3.1. Mesh sensitivity

To ensure that the numerical results obtained were independent of the mesh at an acceptable tolerance, mesh sensitivity was examined. Therefore, the simulated air temperatures and velocities at the above four locations at three different mesh sizes of 1.95 million, 2.44 million and 3.04 million mesh elements are shown in Figure 5.2. As seen, the simulated results at 2.44 million and 3.04 million mesh sizes were very close to each other, within the range of 0.4 ^{o}C and 0.04 m/s. However, the differences between the simulated results at 1.95 million mesh sizes and those at the other two mesh sizes were noticeable. Hence, the numerical results at the mesh sizes of 2.44 million were considered grid independent and thus the mesh sizes of 2.44 million was used in the subsequent numerical study.



(a) Air temperature

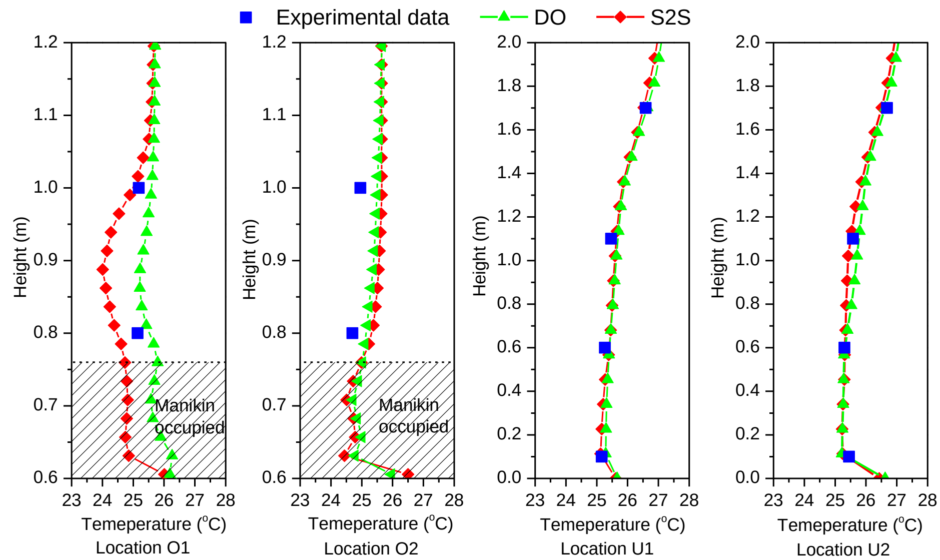


(b) Air velocity

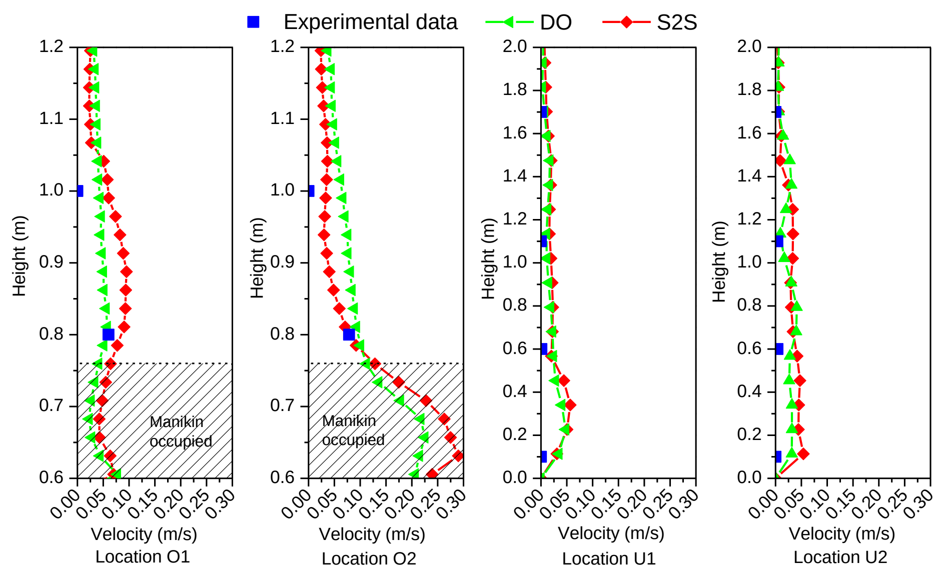
Figure 5.2 The simulated air temperatures and air velocities at four positions using three different mesh sizes

5.3.2. Selection of radiation model

A surface-to-surface (S2S) model is commonly used for radiation heat exchange calculations in buildings, but it only accounts for the radiative heat exchange between surfaces and disregards the presence of air in a room. On the other hand, a discrete ordinates (DO) model, by which the radiant intensity field can be obtained and air treated as a participating media with an extinction coefficient $\beta = 0$, has been recognized for its high accuracy for optically thin media (Modest 2013). Therefore, the simulated results using the above two radiation models were compared with the experimental data. As shown in Figure 5.3, the simulated air temperatures and air velocities using DO model agreed better with the measured data. When using the DO model, the maximum absolute differences between the measured and simulated air temperatures were 0.5 ^{o}C in the occupied zone and 0.2 ^{o}C in the unoccupied zone, and those between the measured and measured air velocities were 0.06 m/s in the occupied zone and 0.03 m/s in the unoccupied zone. Therefore, a DO model for radiation heat exchange calculation was selected in the current numerical study.



(a) Air temperature



(b) Air velocity

Figure 5.3 Comparison between the measured and simulated air temperatures and air velocities using the two different radiation models

5.3.3. Surface temperature validation

Apart from the air temperature and air velocity inside a bedroom, the surface temperature of the bedroom envelope is also an important parameter affecting indoor thermal environment, in particular for a radiation-based air conditioning system. Therefore, the simulated surface temperatures for the six bedroom envelopes were compared with experimental data, as shown in Figure 5.4. It can be seen that the measured and simulated surface temperatures were close to each other, with a maximum difference of 0.6 ^{o}C. Therefore, following the suggestions of validating a CFD method in ASHARE Handbook 2013 (ASHRAE 2013a), the simulated results from this CFD method can be considered as acceptable to reflect the real thermal environment and air flow pattern in a bedroom using the R-TAC system.

5.4. Numerical results and analysis

With the validated CFD method, thermal comfort and energy performance in the experimental bedroom when using R-TAC system have been numerically studied in detail, and the study results are presented in this section. When presenting the study results, a sectional plane shown in Figure 5.5 was used to illustrate the operative temperature field and flow field. The place was perpendicular to X-axis, placed at the middle point of the width of the experimental bedroom. The simulated values of all the parameters were obtained by volume-averaging the data in the different meshes in the occupied/unoccupied zones.

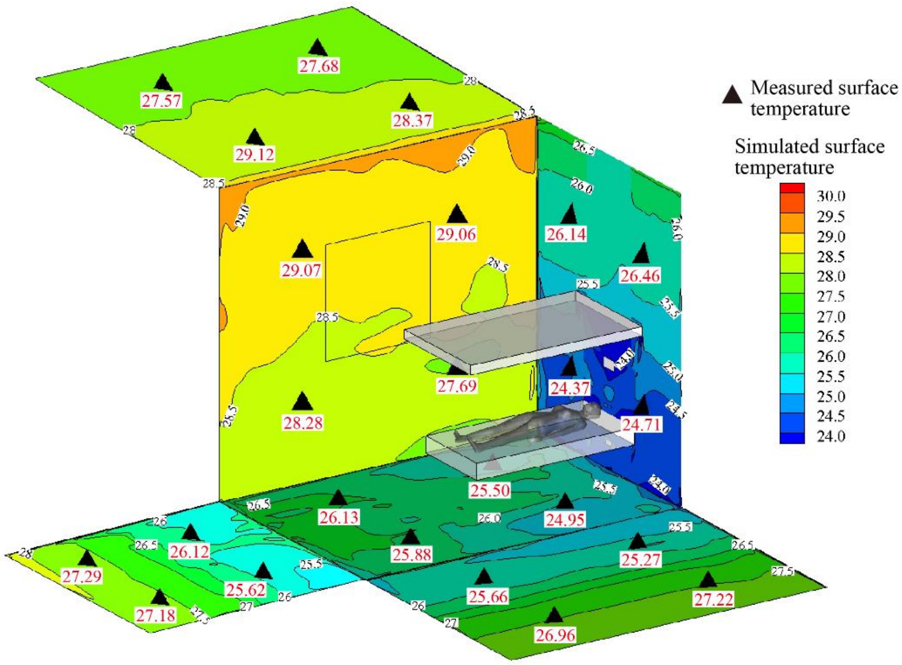


Figure 5.4 Comparison between the measured and simulated surface temperatures of six bedroom envelopes

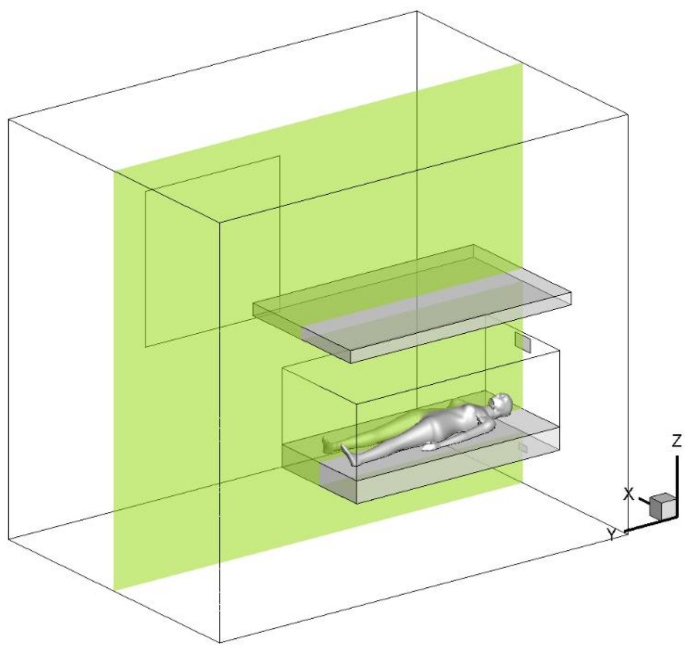


Figure 5.5 The sectional plane for the experimental bedroom

5.4.1. Thermal comfort

In this section, two indexes, operative temperature (t_o) and predicted mean vote (PMV), were used to illustrate the effects of varying surface temperature, area and emissivity of the panel and distance between the bed and the radiant panel on thermal comfort when using the R-TAC system. The operative temperature is a comprehensive index that integrates convective and radiative heat transfer, and thus reflects the thermal environment in the experimental bedroom. On the other hand, PMV was used to evaluate human thermal sensation in the experimental bedroom when using the R-TAC system (Lin and Deng 2008a); The two indexes are expressed as follows:

$$t_o = \frac{h_r t_r + h_c t_a}{h_r + h_c} \tag{5.1}$$

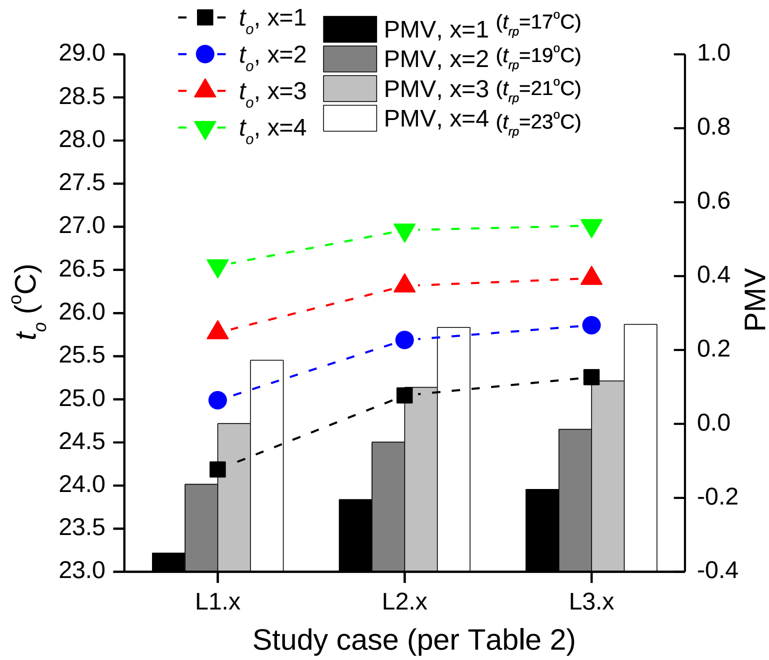
where t_r is mean radiant temperature (°C); t_a air temperature (°C); h_r radiative heat transfer coefficient (W/(m^{2}K)), $h_r = 4.7$ sufficient for most calculations (ASHRAE 2013a); h_c convective heat transfer coefficient (W/(m^{2}K)), h_c=2.7+8.7$v^{0.67}$ for 0.15<v<1.5 m/s, h_c=5.1 for 0<v≤0.15 m/s.

$$\begin{aligned} PMV = 0.0998\left\{40 - \left[\left(C+R\right) + \frac{0.3762\left(5.52 - p_a\right)}{R_t}\right]\right\} \\ -0.0998\left[0.056\left(34 - t_a\right) + 0.692\left(5.87 - p_a\right)\right] \end{aligned} \tag{5.2}$$

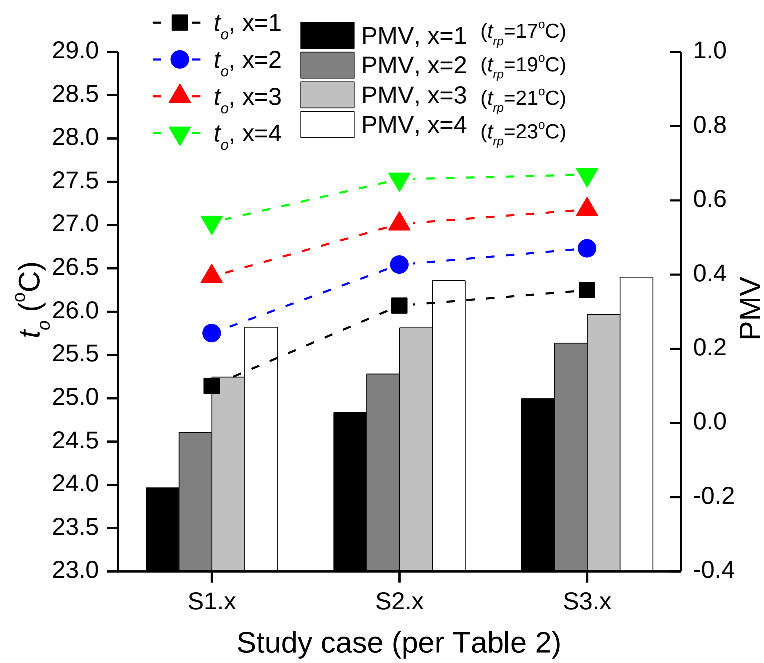
where $C+R$ is sensible heat loss from a human body (W/m^2), R_t the total resistance of a bedding system, which is the thermal resistance to sensible heat transfer between a sleeping person and his/her ambience provided by bed, sleepwear, bedding and its percentage coverage over a human body (m^2 ^oC /W), 0.24335 m^2 ^oC/W (1.57 clo) at full-slip state, obtained from a previous study (Lin and Deng 2008b); p_a water vapor pressure in ambient air (kPa), set at the value when relative humidity was 50%.

Figure 5.6 and Figure 5.7 show the simulated volume-averaged operative temperatures and PMV values in the occupied zone at different settings. In Figure 5.6 (a) for the large panel and (b) for the small panel, with an increase in radiant panel surface temperature, there was an obvious rise in operative temperature. At the same distance and surface temperature of the radiant panel, the operative temperatures in the occupied zone with the larger panel area were lower than those with the smaller panel area, but the differences were gradually narrowed down, from 1 ^oC to 0.5 ^oC when radiant panel surface temperature was increased from 17 ^oC to 23 ^oC. In terms of the comparisons on the effects of different distances on thermal comfort, the operative temperatures in the occupied zone at the distance of D=800 mm and D=1000 mm were similar, and even overlapped when t_{rp}=23 ^oC. However, the operative temperatures in the occupied zone at D=600 mm were greatly lower than those at the other two distances. This was because that, at D=600 mm, the radiant panel was very close to the supply vent and thus blocked the upward diffusion of fresh air. In Figure 5.7, as emissivity of the large radiant panel was increased from 0.1 to 0.9, there was a clear drop of 2 ^oC in operative temperature in the occupied zone at three different distances between the bed and the radiant panel. The PMV values in all the cases ranged from -0.35 to 0.56, but most of them stayed within the range of -0.5~+0.5, suggesting a thermally comfortable environment. Furthermore, the variation trends of

PMV at different cases were similar to those of operative temperature, but with a smaller change rate.



(a) Groups 2-5 for the large panel



(b) Groups 8-11 for the small panel

Figure 5.6 The simulated operative temperatures and PMVs in the occupied zone

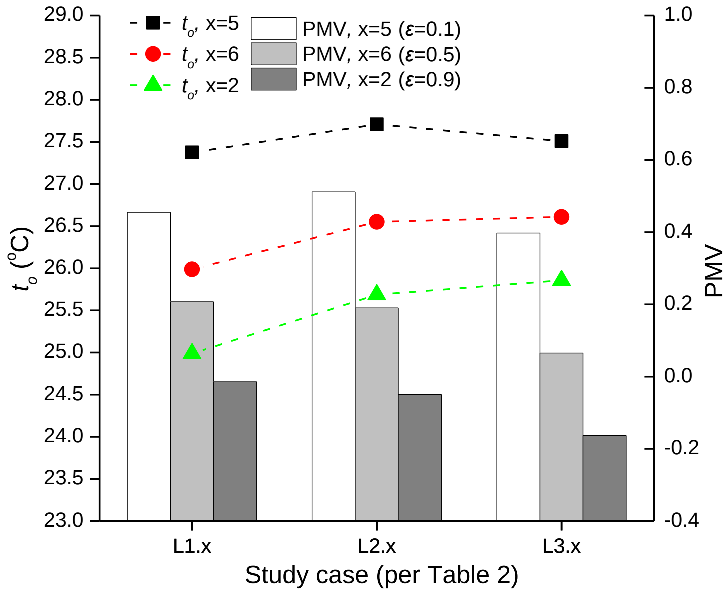


Figure 5.7 The simulated operative temperatures and PMVs in the occupied zone for study Groups 3, 6 and 7

5.4.2. Energy saving

In a TAC system, due to the difference in thermal environments between an occupied zone and an unoccupied zone, an energy saving potential can be expected. Usually, energy saving may be evaluated using an energy utilization coefficient (EUC), which was used by Liu et al. (2008), by Pan et al. (2012b) and Mao et al. (2014a; 2013; 2014b). EUC is defined in Equation (5.3), where $t_{a,UZ}$ and $t_{a,OZ}$ are the averaged air temperature in the unoccupied zone and the occupied zone (°C), respectively, t_s the supply air temperature (°C). However in the current study, considering that radiative heat transfer played a significant role as far as this R-TAC system was concerned, a modified energy utilization coefficient (EUC) reflecting the difference in operative temperature between the occupied zone and the unoccupied zone was used, defined in Equation (5.4).

$$EUC_C = \frac{t_{a,UZ} - t_s}{t_{a,OZ} - t_s} \tag{5.3}$$

$$EUC = \frac{t_{o,UZ} - t_{rp}}{t_{o,OZ} - t_{rp}} \tag{5.4}$$

where $t_{o,UZ}$ and $t_{o,OZ}$ are the averaged operative temperature in the unoccupied zone and the occupied zone (^{o}C), respectively. Equation (5.4) suggests that energy saving becomes possible if EUC is greater than 1, since less cooling energy is used to remove the heat in the unoccupied zone. It is noted that although a higher EUC does not necessarily lead to a less energy consumption, it can reflect the degree of saving if a TAC was used instead of an FAC system.

Figure 5.8 and Figure 5.9 show the EUC values at different study cases. As seen in Figure 5.8, a smaller distance between the bed and the panel would lead to a larger EUC value for both small and large panel. This indicated that varying the distance between the bed and the panel can greatly help improve energy performance. At the same setting of the radiant panel, the EUC value slightly increased with an increase in radiant panel surface temperature. In addition, EUC values when using a large radiant panel were clearly higher than those when using a small radiant panel, suggesting energy utilization efficiency can be effectively improved by using a large radiant panel. Finally, as shown in Figure 5.9, EUC value ascended significantly with an increase in emissivity of the radiant panel, but the increase rates were reduced as the distance between the bed and the radiant panel was increased from D=600 mm to D=1000 mm.

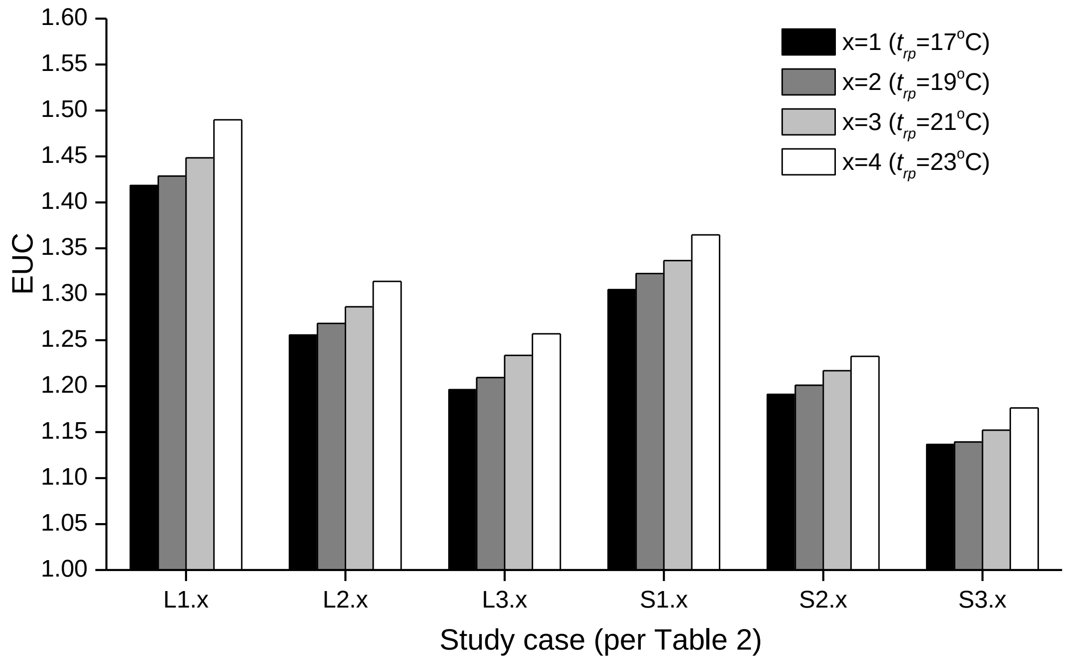


Figure 5.8 EUC values in the experimental bedroom when using R-TAC system at different study cases

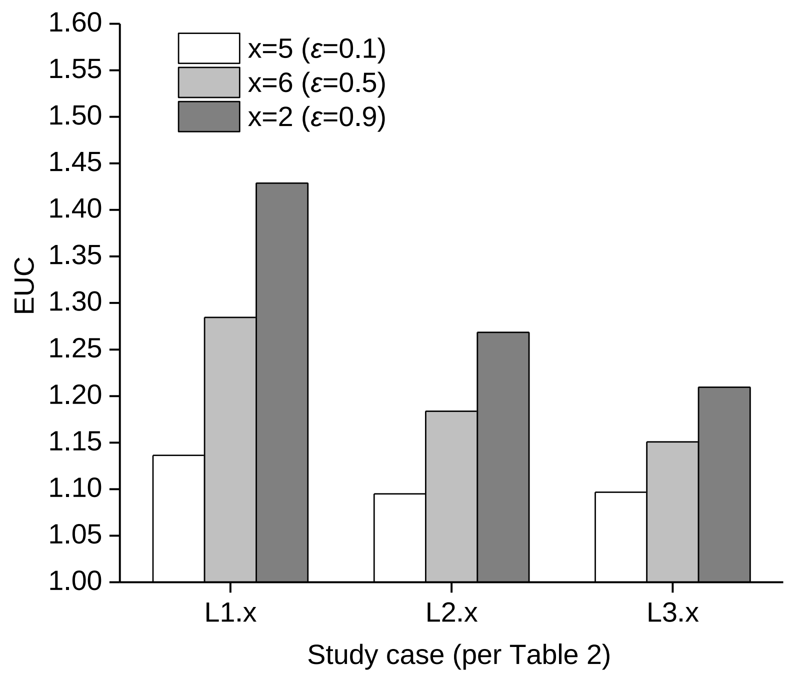


Figure 5.9 EUC values in the experimental bedroom when using R-TAC system at different study cases

5.4.3. Results visualization and comparison at five selected cases

To visualize the effects of three design parameters on thermal comfort and energy performances when using the R-TAC system, the simulated operative temperature fields and air velocity vectors at the sectional plane at five selected cases, Cases L1.2, L1.5, L2.2, L3.2 and S3.2, are shown in Figure 5.10 (a) ~ (e). The use of Cases L1.2, L2.2 and L3.2 was to visualize and compare the differences in operative temperature and air velocity at three different distances between the bed and the radiant panel, shown in Figure 5.10 (a), (c) and (d), respectively. Furthermore, the use of Cases L3.2 and S3.2 was to visualize and compare the differences in operative temperature and air velocity at two different panel areas shown in Figure 5.10 (d) and (e), and Cases L1.2 and L1.5 at two different panel emissivities shown in Figure 5.10 (a) and (b), respectively.

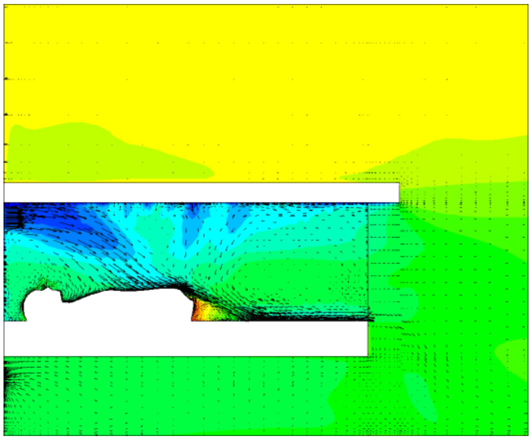
Figure 5.10 (a)~(e) show that there were significant differences in operative temperature between the occupied zone and the unoccupied zone. Also, a weak jet flow from the supply vent was observed. Hence, air velocity in the occupied zone was slightly higher than that in the unoccupied zone. In addition, stratification in operative temperature in the bedroom was obvious, with the stratified line around the height of the radiant panel.

When further comparing operative temperature fields and air velocity vectors among the above cases, the following may be observed:

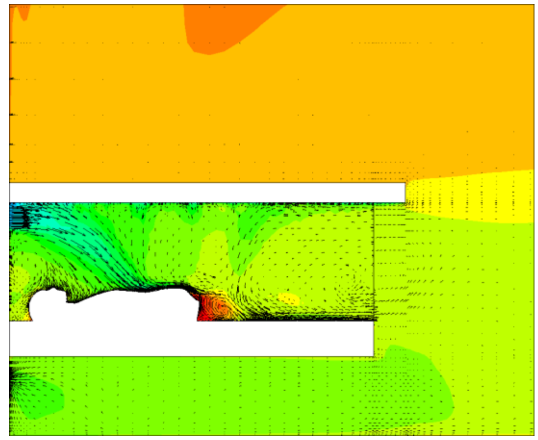
(a) When comparing the effects of distance between the bed and the radiant panel on the thermal environment in the occupied zone (see Figure 5.10 (a), (c) and (d)), best performances were achieved at L1 setting with the shortest distance between the bed and the radiant panel, but there were no significant differences in the operative temperature in the occupied zone between L2 setting and L3 setting. As seen in Figure 5.10 (a), due to the radiant panel being the nearest to the supply vent, the air velocity in the occupied zone at L1 setting was higher than those at L2 and L3 settings. Consequently convective heat transfer between the radiant panel and its surroundings was enhanced, resulting in a higher EUC value.

(b) For the effects of different panel size on indoor thermal environment for Cases L3.2 and S3.2, shown in Figure 5.10 (d) and (e), it seemed that the operative temperatures in both the occupied and unoccupied zones at S3 setting were slightly higher than those at L3 setting, due to the smaller area of the radiant panel.

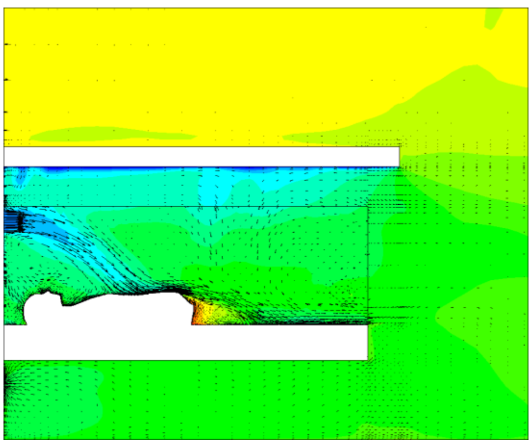
(c) At different emissivities of the radiant panel for Cases L1.2 and L1.5, shown in Figure 5.10 (a) and (b), the operative temperatures in both zones at 0.9 emissivity were obviously lower than those at 0.1 emissivity, although air velocities in this two cases were similar. Furthermore, the operative temperatures in both zones in Case L1.5 shown in Figure 5.10 (b) were even lower than those in Case S3.2 shown in Figure 5.10 (e) with a smaller panel size and a longer distance between the bed and the radiant panel. Therefore, the radiant panel with a low emissivity would severely hinder the heat transfer between the panel and other surfaces of the bedroom, thus resulting in a lower cooling capacity of a radiant panel.



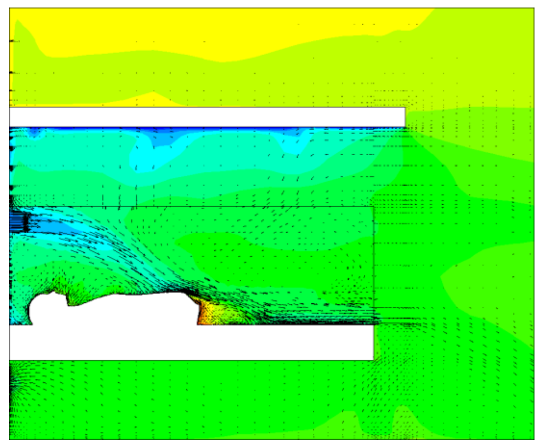
(a) Case L1.2 (t_{rp}=19 ^{o}C, ε=0.9)



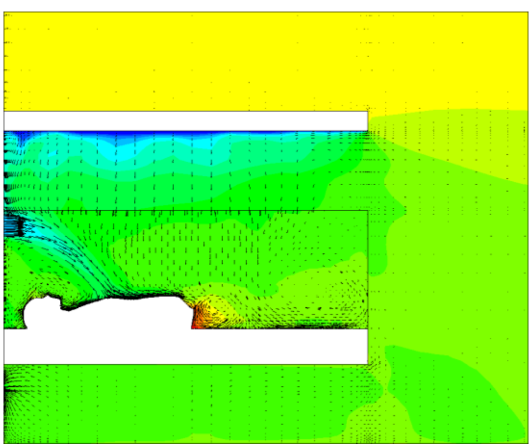
(b) Case L1.5 (t_{rp}=19 ^{o}C, ε=0.1)



(c) Case L2.2 (t_{rp}=19 ^{o}C, ε=0.9)



(d) Case L3.2 (t_{rp}=19 ^{o}C, ε=0.9)



(e) Case S3.2 (t_{rp}=19 ^{o}C, ε=0.9)

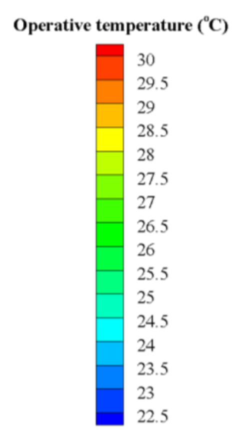


Figure 5.10 Operative temperature fields and air velocity vectors at five selected study cases

Finally, the PMV and EUC values at four different surface temperatures, two different areas of the radiant panel and three different distances between the bed and the panel at an emissivity of ε=0.9 are shown in Figure 5.11. As seen, for the large panel, the R-TAC system can be used to maintain a thermally neutral environment for a sleeping person with the maximum energy utilization coefficient, at Case L1.3 (PMV=0.0009, EUC=1.45). On the other hand, for the small panel, the optimum case was S1.2 (PMV=-0.26, EUC=1.32).

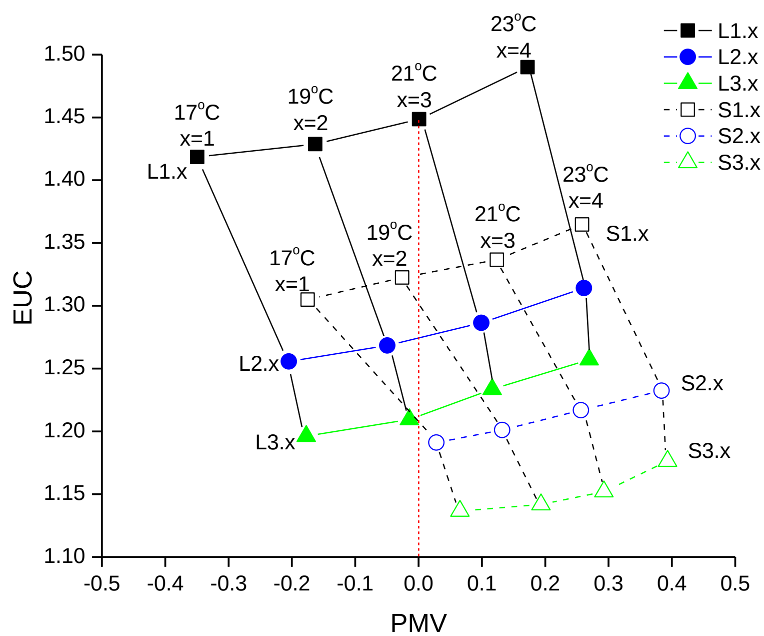


Figure 5.11 PMV versus EUC at different cases

5.4.4. Draft risk

Draft risk (DR) was used to evaluate dissatisfied percentage of occupants caused by cold draft, which has been defined in Equation (4.4) in Chapter 4. Figure 5.12 shows the averaged DR values in the occupied zone at different surface temperatures and areas of the radiant panel, and distances between the bed and the panel. The DR values at all cases were similar, at a very low level of 1.2~2.2%, although the design/operating parameters of the radiant panel did have great impacts on the thermal environment in the bedroom. Overall speaking, as affected by the air temperature in the occupied zone, the DR value tended to be higher at a lower surface temperature and a larger size of the radiant panel, and a shorter distance between the bed and the panel. To further investigate the detailed DR values in the occupied zone, the DR field in the bedroom when using the R-TAC system for Case L1.1 is shown in Figure 5.13. As seen, DR values in the entire occupied zone were lower than 20%, although the DR values were relatively higher near the supply air jet.

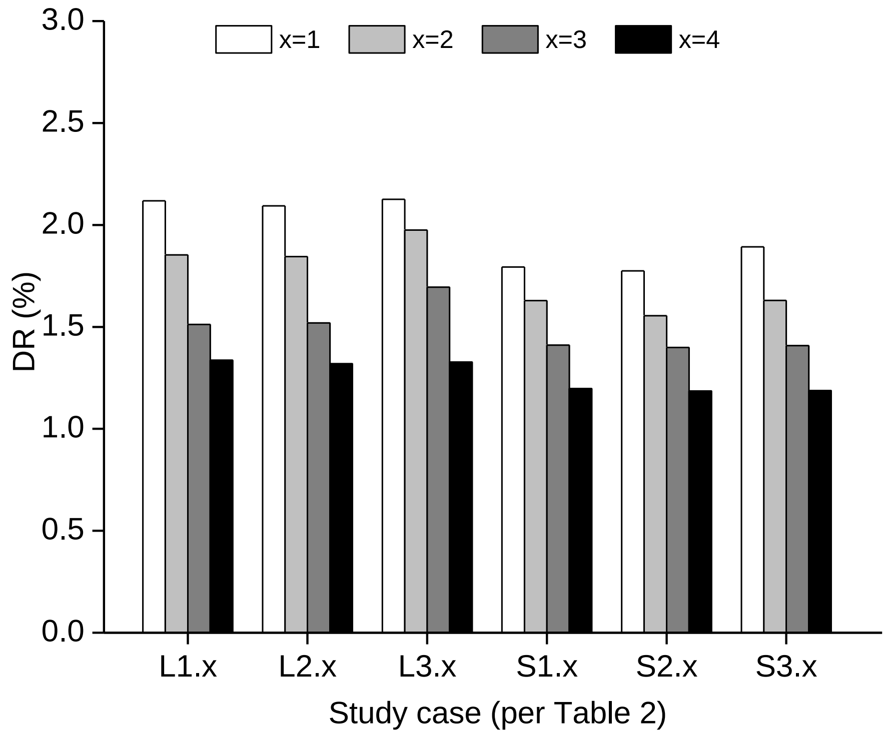


Figure 5.12 DR values in the occupied zone at different study cases

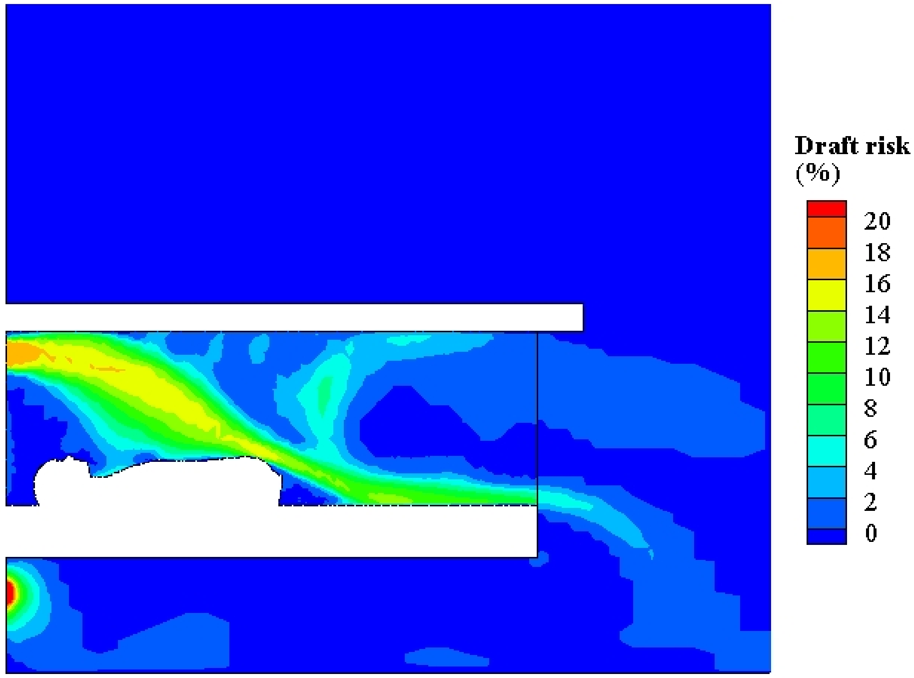


Figure 5.13 DR field in the bedroom when using the R-TAC system for Case L1.1

5.5. Discussions

When the overall indoor environment is within comfort ranges, an occupant may still feel uncomfortable if one or more parts are subject to load heating/cooling. Except cold draft, non-uniformities may also be due to windows or improperly sized or installed radiant panels (ASHRAE 2013a). In this R-TAC system, the radiant panel was designed to be closer to a sleeping person, which may lead to local discomfort caused by non-uniformity. Radiant asymmetry can be evaluated by the difference in radiant temperature of the environment on opposite sides of the person. Taking Case L1.1 with the lowest PMV value as an example, the simulated temperature differences for cold envelope (between the cold panel and the floor) and warm envelope (between the external wall and its opposite wall) were 8.7 ºC and 1.59 ºC, respectively, both within the limits of 10 ºC and 22 ºC, respectively, suggested by ASHARE. Therefore, the local discomfort caused by radiant asymmetry may be ignored when using the R-TAC system.

5.6. Conclusions

This Chapter reports a numerical study on the effects of the design/operating parameters of the radiant panel, including surface temperature, surface emissivity and area of the radiant panel and the distance between the bed and the panel, on thermal comfort inside an experimental bedroom and energy saving when using the R-TAC system. The numerical method was validated using the experimental data obtained in the experimental study presented in Chapter 4, and the simulated results on indoor thermal comfort and energy performances at different surface temperatures, emissivities and sizes of the radiant panel and different distances between the bed and the panel were analyzed and compared. The study results showed that increasing surface temperature can lead to a higher PMV value and a higher EUC value, while increasing surface emissivity and area of the radiant panel can result in a lower PMV value and a lower EUC value. Furthermore, although reducing the distance between the bed and the panel caused a lower PMV value and a higher EUC value, the PMV and EUC values at the distance of D=800 mm and those at the distance of D=1000 mm were similar. It can be seen that the differences in operative temperature between the occupied zone and the unoccupied zone were significant, indicating a higher energy utilization efficiency when using the R-TAC system. Air velocity in the bedroom was at a low level, although it was slightly higher in the occupied zone. Finally, the DR values were at a very low level in all the study cases, indicating the cold draft problem encountered when using a convective heat transfer based TAC system for a sleeping environment can be effectively resolved when using the R-TAC system. Changing the design/operating parameters of the radiant panel may have impacts on ventilation efficiency, therefore, indirectly affecting indoor air quality, which should be studied in the future.

Chapter 6

The impacts of daytime external envelope heat gain/storage on the nighttime cooling load and the related mitigation measures in a bedroom in the subtropics

6.1. Introduction

Apart from developing a novel radiation based TAC system applied to sleeping environments, as presented in Chapters 4 and 5, energy use for air conditioning a bedroom during sleep can also be saved by reducing the nighttime cooling load. For residential buildings in the subtropical Hong Kong, Lin and Deng (2004) found out that the total cooling load in the summer design day in a bedroom facing west during 21:00~22:00 exceeded excessively those facing the other three orientations and was approximately 50% more than that in the later part of a nighttime A/C process, as shown in Figure 2.7 in Chapter 2. They also suggested that the cooling load from the building envelope dominated the total cooling load at 75%. On the other hand, Bojic et al. (2001) suggested that adding thermal insulation to the external walls in residential buildings in hot and humid climates would not lead to a significant cooling load reduction. Furthermore, a comparison in energy performances between using thermally light and massive external walls in residential buildings demonstrated that the use of external walls with high thermal mass was not economical (Chirarattananon and Hien 2011).

For bedrooms in residential buildings, they are normally not occupied, thus no A/C is required at daytime. During the unoccupied period, thermal energy may be stored in the thermal mass in a bedroom such as an external envelope and internal furniture, when they are subjected to solar radiation and heat transfer due to temperature difference. It is noted that in cold climates in winter, the use of Trombe walls can store solar heat at daytime to provide heating at nighttime, as reported in Section 2.8 in Chapter 2. In hot and humid climate in summer, the thermal energy stored in the thermal mass in a bedroom at daytime will on the contrary become part of space cooling load for the bedroom at nighttime when it is occupied and air conditioning turned on, which is also reflected by the load profiles shown in Figure 2.7 in Chapter 2. Therefore, it makes sense to minimize the thermal energy stored in the thermal mass at daytime, such that the resultant total space cooling load at nighttime operation will be lower. However, although cooling load characteristics in bedrooms during sleep at nighttime have been extensively studied, no previous studies on both the impacts of daytime heat gain/storage in the thermal mass on the cooling load in a bedroom at nighttime operation and the measures to minimize the daytime heat/storage in the thermal mass were reported. These two are therefore the issues addressed in this Chapter.

Currently, an air gap is widely applied to external opaque walls or to curtain walls to increase their overall thermal transfer resistances, and the energy-saving potentials by adding an air gap in an external envelope element are presented in Section 2.8 in Chapter 2. However, no previous studies on applying an air gap to the opaque external wall of a bedroom to reduce the thermal energy stored in the wall when not occupied at daytime, leading to a reduced space cooling load at nighttime operation, may be identified.

Therefore, in this Chapter, a simulation study on further analyzing the cooling load characteristics in a bedroom in the sub-tropic Hong Kong at nighttime and on the effects of adding an air gap in the opaque external wall of the bedroom on reducing thermal energy stored in the wall and consequently the space cooling load at nighttime A/C for the bedroom is reported, as the third part of the systematic research program. The west-facing bedroom in a high-rise residential block in Hong Kong, used in a previous related study (Lin and Deng 2004), was also used as a platform to carry out the simulation study, based on which a simulation model using EnergyPlus was established. In this Chapter, firstly, a detailed description of the model and assumptions to simplify the model are presented. This is followed by a detailed account of the cooling load characteristics in the bedroom at nighttime operation focusing on analyzing the impact of thermal energy gain and storage in the west-facing external wall of the bedroom at daytime on the resultant nighttime cooling load. Finally, the effects of adding an air gap and ventilating the gap in the west-facing external wall on the total cooling load at nighttime are investigated.

6.2. Model development and simulation conditions

The well-known building energy simulation program, EnergyPlus (Tanimoto et al. 2008), was adopted in the simulation study, since the current study can be regarded as an extension to the previous related simulation study (Lin and Deng 2004), where EnergyPlus was also used. In fact, since its first release in 2001, EnergyPlus has been widely accepted in evaluating building energy use performances and thermal load calculation (Lin and Deng 2004; Li et al. 2006; Pan et al. 2012a).

6.2.1. Model development

In the previous related study (Lin and Deng 2004), a hypothetic high-rise model residential building was used. Figure 6.1 shows a typical floor plan of the residential building. It was pointed out that all the cooling loads for all the bedrooms facing west at night were the highest among that for the bedrooms in this floor. Therefore, the west-facing bedroom, B3, in Apartment 6_W in Figure 6.1, was used as a study platform in the previous study (Lin and Deng 2004).

In the current study reported in this Chapter, to maintain a continuity to the previous related study (Lin and Deng 2004), the same west-facing bedroom, B3, was also used. The size of the bedroom (W×D×H) was 2275 mm × 2545 mm × 3000 mm, as shown in Figure 6.2. On its west-facing external wall, there was a window (1200 mm × 900 mm) with a generic interior shade (Solar transmittance=0.15; Solar reflectance=0.35; Thickness=0.001 m; Conductivity=0.05 W/K·m) and a concrete overhang of 500 mm width. Table 6.1 lists the construction details and outside boundary conditions for relevant building envelope components/indoor furniture available in the bedroom, and Table 6.2 the physical properties of the materials used in envelope components. For the five envelopes, i.e., internal partitions/walls, and floor and ceiling, there were two surfaces for each of them, i.e., external and internal. Only the five external surfaces were assumed adiabatic, but the internal surfaces were not and can absorb and store solar heat.

Except the overhang, the simulated bedroom was not shaded by any other obstructions in its close vicinity.

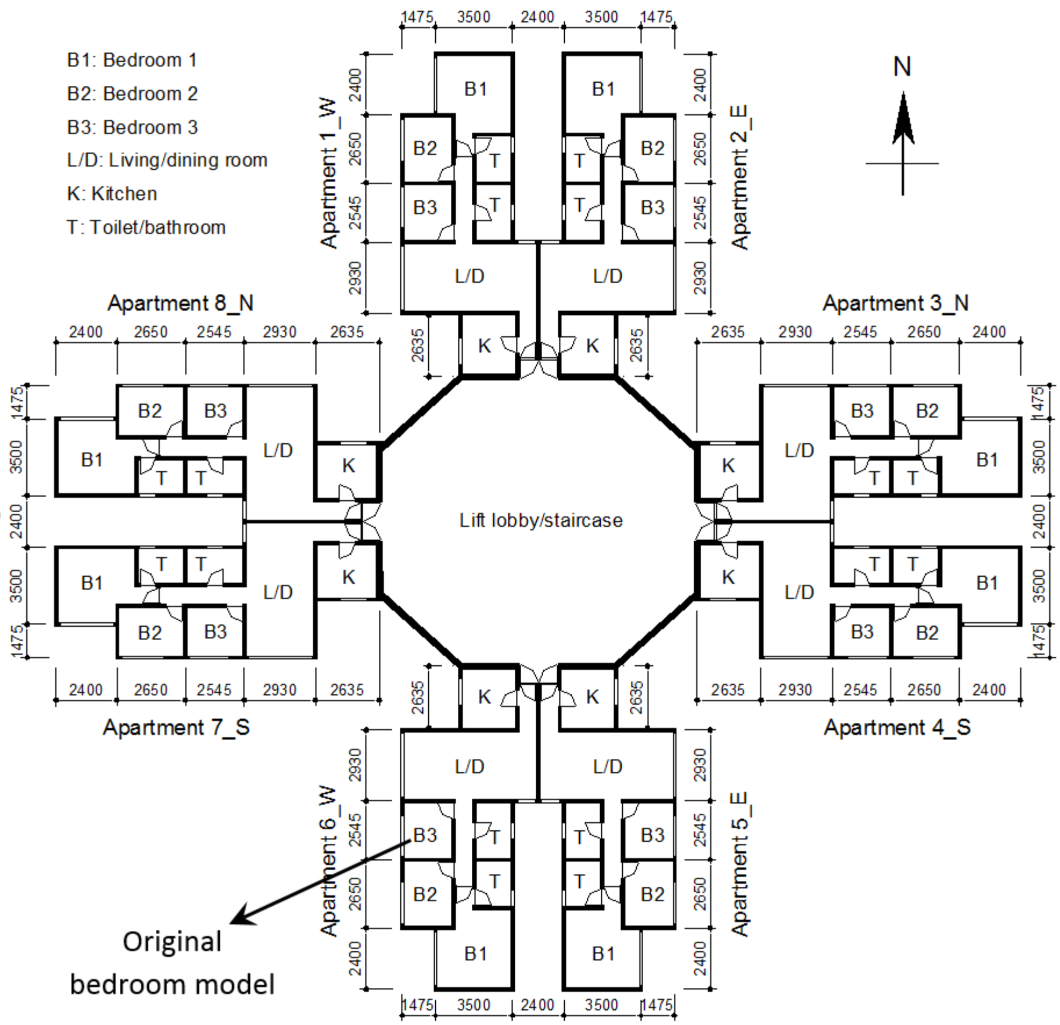


Figure 6.1 Floor plan of a high-rise residential building in Lin and Deng study (2004)

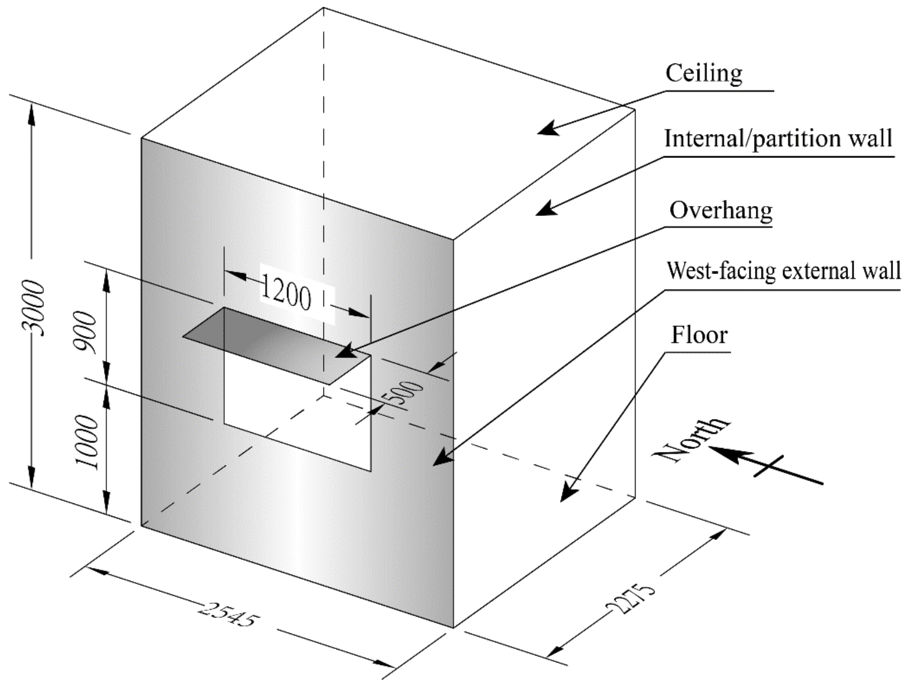


Figure 6.2 The details of the simulated bedroom (B3 in Apartment 6_W)

Table 6.1 Details of building envelope components and indoor furniture in the studied bedroom

Component	Description	Outside boundary conditions (External surface only)
External wall	13 mm cement/sand plaster (outside) +100 mm concrete+13 mm gypsum plaster (inside)	Outdoor environment
Internal partition/walls	50 mm concrete (outside) +13 mm gypsum plaster (inside)	Adiabatic
Floor and Ceiling	50 mm concrete	Adiabatic
Window	An external concrete overhang of 500 mm width+ one sheet of 5-mm-thick glass+ a generic interior window shade	Outdoor environment
Furniture	One sheet of 18-mm-thick wood with 3 m² of surface area	—

Table 6.2 Physical properties of the materials used in envelope components

Material	Specific heat capacity (J/kg·K)	Density (kg/m³)	Thermal conductivity (W/K·m)
Concrete	653	2400	2.16
Cement/sand plaster	840	1860	0.72
Gypsum plaster	837	1120	0.38
Wood	2093	800	0.16
Glass	—	—	0.9

6.2.2. Schedules of A/C operation, internal heat gains and ventilation

In this simulated bedroom, a window-type room air conditioner (WRAC) was installed. Its operating time at night was from 21:00 to 7:00 in the following day. The operating mode of the WRAC was set at "continuous fan cycling compressor", which was commonly adopted by RACs.

The bedroom was occupied by one average adult (of 1.73 m tall, 70 kg, DuBois area= 1.8 m^2) (ASHRAE 2013b). According to ASHRAE Standard 55 (ASHRAE 2013d), the activity levels of the occupant were assumed at 60 W/m^2 (1.0 met) when awake and 40 W/m^2 (0.7 met) when sleeping. The assumed hourly internal heat gains from lighting, electric appliance and the occupant in the bedroom are shown in Table 6.3.

A fresh air flow rate of 7.5 L/(s·person) as recommended by ASHRAE Standard 62 (ASHRAE 2001) was provided when WRAC was in operation. However, when the bedroom was not occupied at daytime, it was naturally ventilated with the window open at an assumed constant air change rate of 2 ACH.

Table 6.3 Hourly internal heat gains in the simulated bedroom

Time period	Lighting load (Lin and Deng 2004) (W/m²)	Electric appliance load (Lin and Deng 2004) (W/m²)	Activity level (W/m²)
00:00-06:00	0	0	40
06:00-07:00	8.165	23.53	40
07:00-18:00	0	0	0
18:00-20:00	0	23.53	0
20:00-21:00	16.33	23.53	0
21:00-23:00	16.33	23.53	60
23:00-24:00	0	0	40

6.2.3. Indoor design air temperature and meteorological data

At 25 ^{o}C operative temperature and 60% air relative humidity, an indoor environment will be situated in the comfort zone (10% PPD) as recommended by ASHRAE Standard 55 (ASHRAE 2013d). Since operative temperature is approximately the mean value of indoor air temperature and mean radiant temperature in most practical situations (ASHRAE 2013d) and indoor air temperature is usually lower than mean radiant temperature, indoor air temperature should be below 25 ^{o}C (Lin and Deng 2004). Furthermore, in Hong Kong, it has been recommended to use 22 ^{o}C air temperature and 50% relative humidity in bedrooms (HK-BEAM May 2003). Therefore, indoor air temperature was set at 23 ^{o}C in this simulation study. Only indoor air temperature was controlled using a WRAC, with indoor relative humidity left uncontrolled (Lin and Deng 2004).

Using the outdoor cooling design conditions in the summer design day in Hong Kong, outdoor design dry-bulb (DB) temperature, wet-bulb (WB) temperature and the daily temperature variation range are 33.2 ^{o}C, 26.1 ^{o}C and 4.5 ^{o}C, respectively (ASHRAE 2013a; Lin and Deng 2004). EnergyPlus can automatically and sinusoidally distribute the 4.5 ^{o}C range over the 24 hours in the design day, so that outdoor air temperature at any time in the summer design day could be determined (Lin and Deng 2004).

6.3. Analysis on nighttime cooling load characteristics in the simulated bedroom

In this simulation study using EnergyPlus, a time step of 1 min was used. Hourly averaged values for each output variable were obtained by averaging the 60 data obtained in every minute.

6.3.1. Nighttime cooling load characteristics in the simulated bedroom

Figure 6.3 shows the simulated total hourly nighttime cooling loads from 21:00 to 07:00 in the following day (normalized by per unit floor area of the bedroom, W/m^2) in the simulated bedroom, which is similar to that previously obtained in (Lin and Deng 2004) shown in Figure 2.7 in Chapter 2, and their percentage breakdowns due to envelopes, internal gains and ventilation, in the summer design day.

As seen in Figure 6.3, the total hourly cooling load at nighttime peaked at 21:00, i.e., the starting hour of a nighttime A/C process, and then decreased significantly in the next two hours, by almost 50%. The variation of the cooling load in the remaining hours was less significant, although a declining trend was still maintained, except a slight increase in the last hour. As seen from Figure 6.3, the cooling loads due to internal gains and ventilation accounted for a small fraction of the total nighttime cooling load and remained stable for nearly the whole night. However, the nighttime cooling load from building envelopes took the lion's share of the total nighttime cooling load in every hour, which was particularly true for the first two hours, taking up more than two thirds of the hourly total cooling load. This therefore implies the significant impact of thermal energy stored in the envelope components and suggests a large potential of energy saving through modifying building envelopes so as to decrease the cooling load at nighttime.

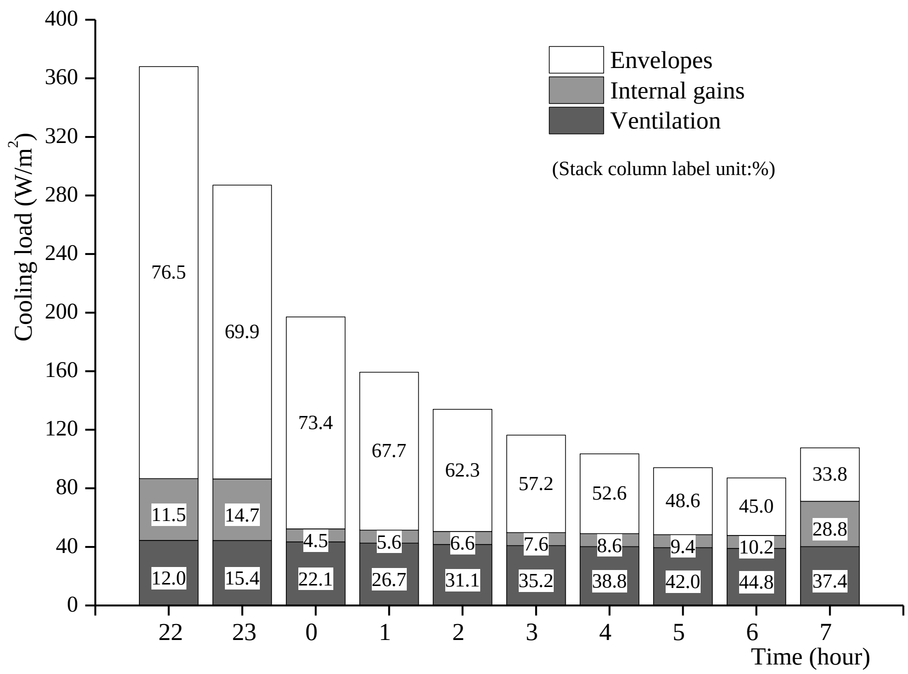


Figure 6.3 The simulated total hourly cooling loads at nighttime and their percentage breakdowns for the studied bedroom in the summer design day

To further understand how the thermal energy stored in envelope components at daytime would impact the cooling load at nighttime, Figure 6.4 shows the simulated 24 hours indoor air temperature variation profile for the studied bedroom in the summer design day. As seen for this west-facing bedroom, during 08:00-14:00 without air conditioning, indoor air temperature gradually increased from 26.2 °C to 30.8 °C. Then from 14:00 onwards in the afternoon, indoor air temperature was increased at a faster pace, reaching 39.6 °C at 18:00 and levelling for the next few hours, but dropping to 23 °C (indoor air temperature setpoint) at 21:00 when the bedroom was occupied and air conditioning turned on. Therefore, there were approximately 5 hours when the indoor air temperature was higher than the outdoor air temperature, as indicated by the shaded area "A" in Figure 6.4. This suggested that during this 5-hour period, indoor air was heated, possibly by the thermal energy stored in the thermal mass.

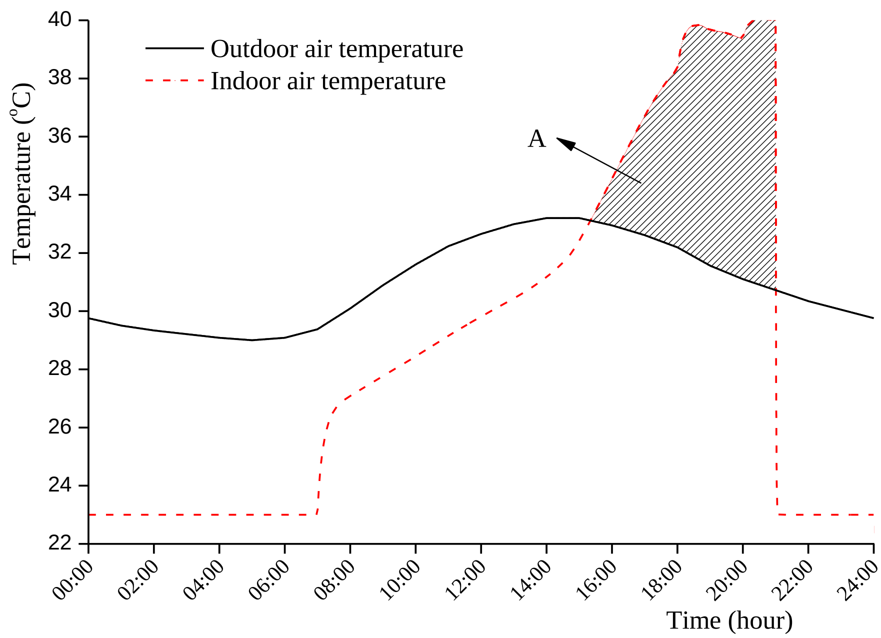


Figure 6.4 Simulated variation profile of hourly indoor air temperature in the west-facing bedroom in the summer design day

An examination on the simulated hourly envelope surface temperatures in the bedroom in the summer design day shown in Figure 6.5 can help further reveal the impact of thermal energy stored in thermal mass on indoor thermal environment. In Figure 6.5, since the west-facing external wall was the only envelope component in direct contact with outdoor environment, its internal surface temperature is separately presented from the averaged temperature of the other five internal surfaces. As seen, the simulated external surface temperature of the west-facing external wall significantly fluctuated, reaching 54 ^{o}C at 17:00, much higher than the outdoor air temperature at that time. This high temperature directly reflected the accumulated solar heat gain by the external wall over the entire afternoon. On the other hand, the simulated variation profiles of the internal surface temperature of the west-facing external wall and the averaged temperature of the other five internal surfaces in the bedroom are also shown in Figure 6.5, with the former always being higher than the latter. As seen from Figure 6.5, although the external surface temperature of the west-facing external wall was at a higher level than that of the internal surface of the external wall at daytime, it quickly dropped after 18:00 to a temperature of 35.0 ^{o}C at 21:00, which was lower than that of the internal surface at 36.1 ^{o}C. This suggested that the west-facing external wall stored a large amount of thermal energy over daytime and began to release it to outdoor air after sunset. As the temperature of outdoor air was lower than that of indoor air, more heat was released to outdoor air, leading to a lower external surface temperature. At 21:00 when air conditioning was turned on, the temperature of the internal surface of the external wall, the averaged temperature of the other five internal surfaces and indoor air temperature were 37.0 ^{o}C, 36.1 ^{o}C and 23.1 ^{o}C, respectively. This explained that since thermal energy stored in the thermal mass began to release, a very high hourly total cooling load in the first two hours of the nighttime A/C process in the simulated bedroom was resulted in.

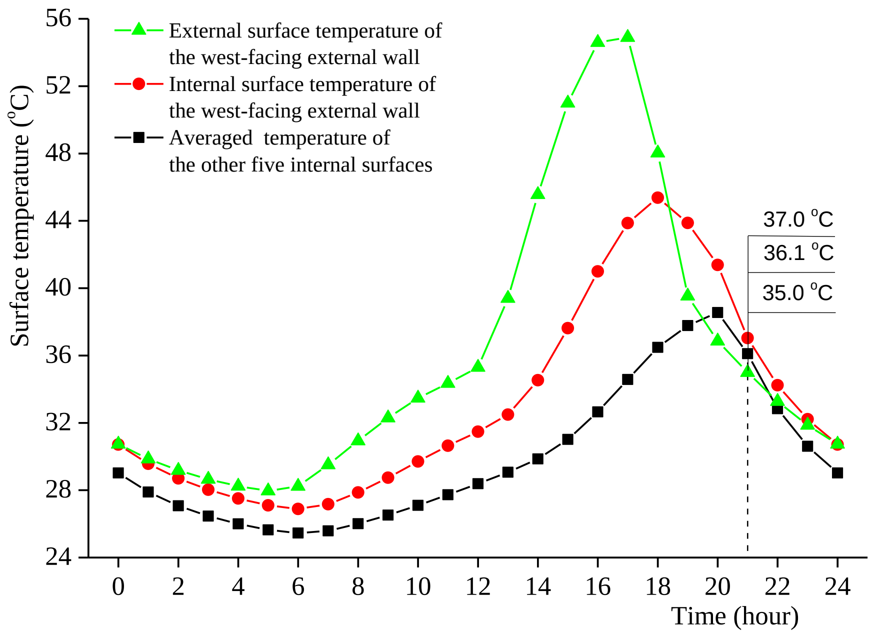


Figure 6.5 Simulated variation profiles for hourly surface temperatures in the west-facing bedroom in the summer design day

In the subtropics, solar radiation is considerably intense at daytime in summer months. Therefore, bedrooms, in particular west-facing bedrooms, can receive and store thermal energy due to solar heat gain and then result in a large nighttime cooling load, as discussed earlier. As shown in Figure 6.6, the simulated total hourly nighttime cooling loads in the simulated bedroom under the a hypothetic condition of no solar radiation in the summer design day are compared with that with solar radiation. As seen, at the start of nighttime air conditioning, i.e., 21:00, the total cooling load with no solar radiation was significantly 48.1% less than that with solar radiation. Then in the remaining hours thereafter, although the difference between the two became smaller, the former was always lower than the latter. This suggested that the effect of solar radiation would significantly impact the total nighttime cooling load as long as thermal mass was available to store the thermal energy from the radiation.

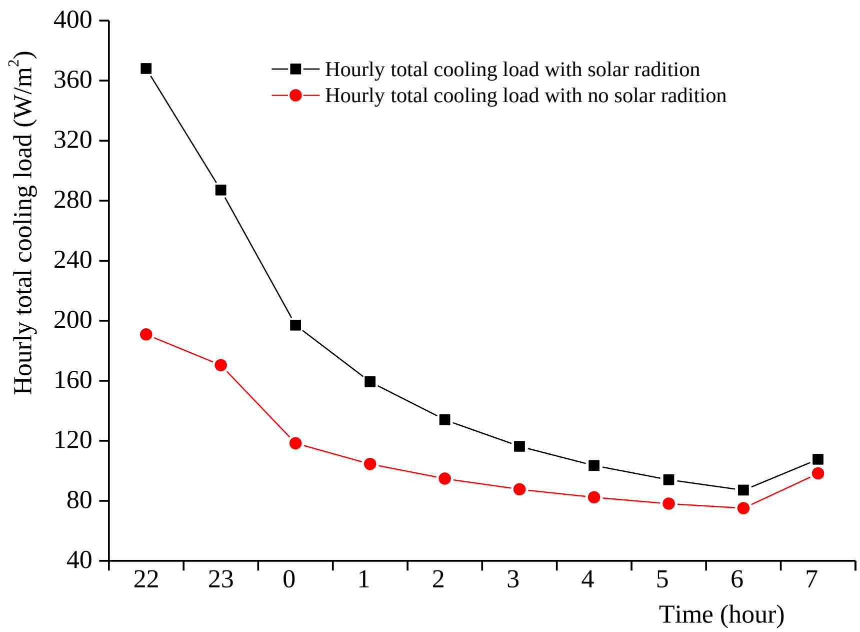


Figure 6.6 The comparison between the total cooling load at nighttime under the condition with solar radiation and that with no solar radiation in the simulated bedroom in the summer design day

6.3.2. Effects of the thermal mass of the west-facing external wall in the simulated bedroom on the nighttime cooling load in the summer design day

As discussed in Section 6.3.1, thermal energy stored in the thermal mass can significantly affect cooling load at nighttime. It is also revealed that solar radiation contributed significantly to the total cooling load since it provided most thermal energy to be stored in the thermal mass, as shown in Figure 6.6. Since the west-facing external wall was the primary receiver of the solar radiation in the simulated bedroom, the impact of the availability of thermal mass in the external wall to store thermal energy on nighttime cooling load was therefore studied by simulation, where the density and specific heat capacity of wall material was set to zero while assuming its value of thermal insulation unchanged.

The study results are shown in Figure 6.7. As seen, at 21:00, the nighttime cooling load from the west-facing external wall when its thermal mass was set to 0 was about half of that with thermal mass. For the remaining hours, the former was always smaller than the latter. It can also be seen that the hourly nighttime cooling load from the external wall at 0 thermal mass was resulted from the air temperature difference between indoor air and outdoor air and, hence, was gradually decreased as outdoor air temperature gradually dropped overnight.

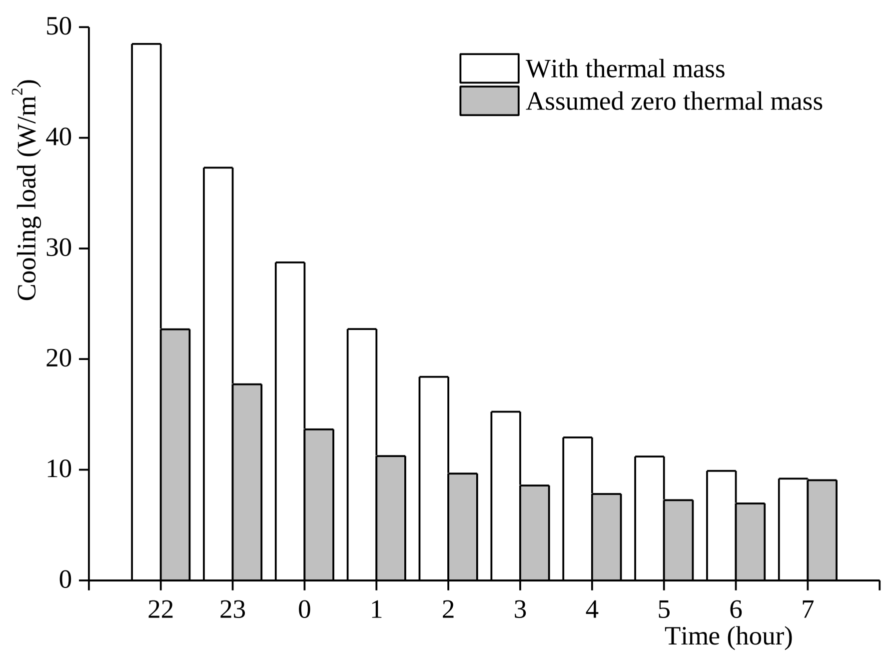


Figure 6.7 The simulated hourly nighttime cooling load due to the west-facing external wall with thermal mass and that without thermal mass

Apart from the west-facing external wall, thermal energy can be also stored in other indoor thermal masses in the bedroom including the other three internal walls, floor, ceiling and furniture, and the thermal energy to be stored there mainly came from radiative heat transfer from the west-facing external wall and the west-facing external window.

At daytime, the internal surface temperatures of both the external wall and the external window were much higher than that of other thermal masses. Based on the principle of energy conservation, the sum of radiative heat gains on all surfaces in an enclosed space is:

$$QR_{\text{external}} + QR_{\text{window}} + QR_{others} = 0 \tag{6.1}$$

Where QR is radiative heat gain by an internal surface (W). A positive QR means that the surface absorbs radiative energy, and a negative QR releases radiative energy. The subscripts "external" and "window" means the internal surface of the west-facing external wall and the internal surface of the west-facing window, respectively, and the subscript "others" means other internal surfaces in the bedroom including the three internal walls, floor, ceiling and furniture. Hence, QR_{others} means the sum of total radiative heat gains by other indoor thermal masses.

Figure 6.8 shows the simulated radiative heat exchanges within the bedroom at daytime in the summer design day. As seen, at t_1 (07:06), the other thermal masses began to store radiative energy while the external wall and the window started to transmit energy out and the process did not stop until t_2 (18:54). However, from t_2 to t_3 (21:00), the external window having a smaller thermal storage capacity started to absorb thermal energy, when the external wall was the only source for emitting thermal energy. A ratio of radiative heat transfer to other indoor thermal masses from the external wall to the total thermal energy gain by other thermal masses, was defined as:

$$\eta = \frac{\int_{t_1}^{t_2} QR_{others}dt \bullet \frac{\int_{t_1}^{t_2} QR_{external}dt}{\int_{t_1}^{t_2} (QR_{external} + QR_{window})dt} + \int_{t_2}^{t_3} QR_{others}dt}{\int_{t_1}^{t_3} QT_{others}dt} \times 100\% \qquad (6.2)$$

Where QT_{others} is the total thermal energy gains by other indoor thermal masses (W), which included direct solar heat gain, thermal radiation from the external wall and window and convective heat transfer from indoor air. Using Equation (6.2), η can be obtained by calculating corresponding areas for QR_{others}, QR_{window}, $QR_{external}$ and QT_{others} in Figure 6.8, at 58.1%. At 21:00, the total nighttime cooling load, the nighttime cooling load from envelopes and the nighttime cooling load from west-facing external wall can be obtained from Figure 6.3 and Figure 6.7, at 368.0 W/m^2, 281.4 W/m^2 and 48.5 W/m^2, respectively. Consequently, the west-facing external wall was responsible for 49.9% ([48.5+ (281.4- 48.5)×η)]/368.0×100%=49.9%) of the total nighttime cooling load in the bedroom from 21:00 to 07:00 in the following day. This suggested that the west-facing external wall contributed significantly to the total nighttime cooling load in the bedroom in the summer design day. Therefore, measures to modify the west-facing external wall for reducing its heat gain and storage have been studied, which is reported in Section 6.4.

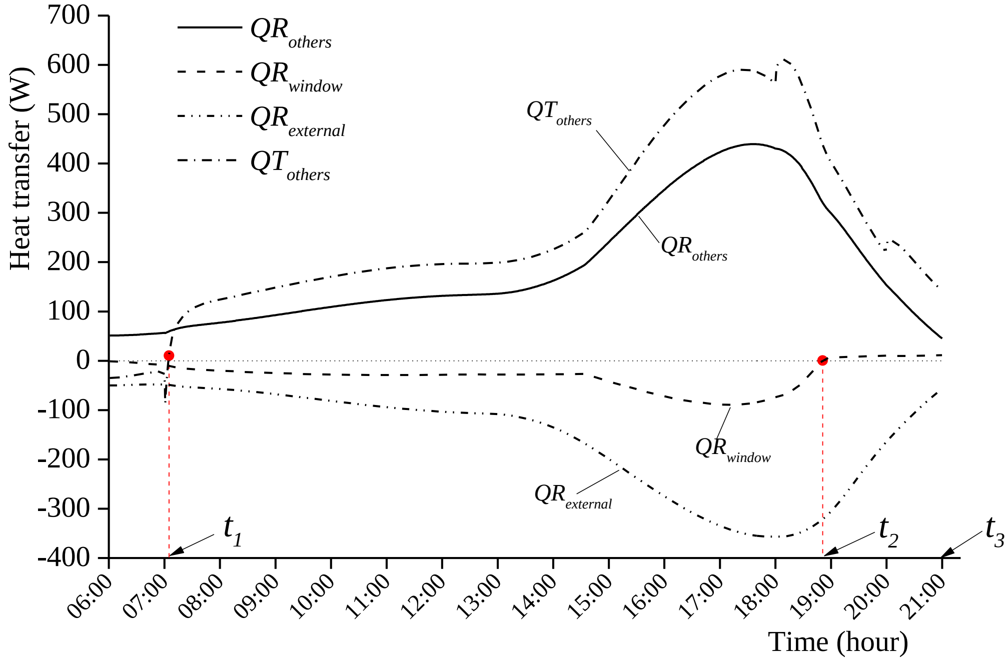


Figure 6.8 Simulated radiative heat exchanges within the bedroom at daytime in the summer design day

6.4. Modifying the external wall to reduce the nighttime cooling load in the simulated bedroom

Various measures to modify the opaque west-facing external wall in the simulated bedroom to reduce the gain and storage of thermal energy in its thermal mass were studied by simulation and the study results are presented in this Section. The modifications were centered in adding an air gap to the external wall, as shown in Figure 6.9. This was because adding an air gap in an opaque wall can be very effective in modifying its thermal insulation but without increasing its thermal mass, as mentioned in Section 6.1. Totally, there were 12 simulation cases for wall modification, as detailed in Table 6.4. The simulated results presented in Section 6.3, were designated as Case 1, as a baseline for comparison purpose. In Cases 2-6, where an air gap of varying width was introduced to the wall and the effects of air gap width on the resultant total nighttime cooling load in the bedroom investigated. Furthermore, the effects of varying the width of inside and outside concrete layers at a fixed air gap width of 20 mm on the total nighttime cooling load in the bedroom were compared in Case 4 and Cases 7-10. Finally, in Cases 11 and 12, the effects of ventilating the air gap of 20 mm width and adhering aluminum foil to the surfaces of the air gap of 20 mm width on the total nighttime cooling load in the bedroom were respectively investigated.

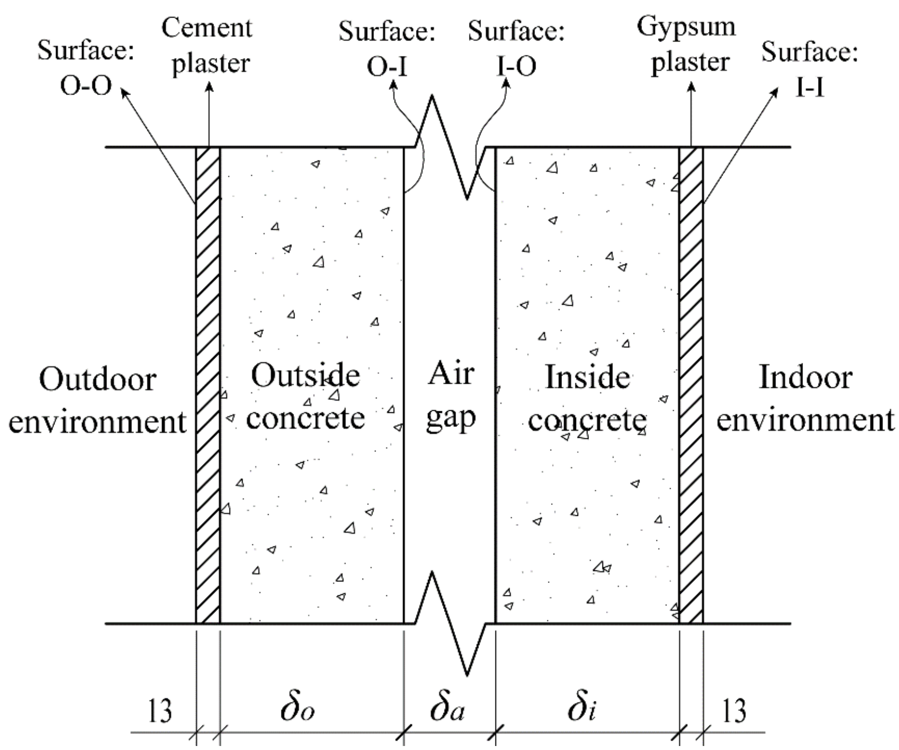


Figure 6.9 Configuration of the west-facing external wall in the simulated bedroom

Table 6.4 The details of the 12 simulation cases

Case	δ_o[a] (mm)	δ_a[a] (mm)	δ_i[a] (mm)	Air gap ventilation	Aluminum foil adhered
1 (baseline case)	50	0	50	N/A	N/A
2	50	10	50	—	—
3	50	15	50	—	—
4	50	20	50	—	—
5	50	30	50	—	—
6	50	40	50	—	—
7	90	20	10	—	—
8	70	20	30	—	—
9	30	20	70	—	—
10	10	20	90	—	—
11	10	20	90	Yes	—
12	10	20	90	—	On Surface I-O,O-I[b]

[a] δ_o, δ_a and δ_i are the width of outside concrete layer, air gap and inside concrete layer of the west-facing external wall (Figure 6.9)

[b] See Figure 6.9

6.4.1. Effects of varying the width of the air gap on the total nighttime cooling load in the studied bedroom

In this Section, the effects of varying the width of the air gap, δ_a, on the total nighttime cooling load in the bedroom were studied through simulations. From Figure 6.10, it can be seen that adding a 10 mm air gap to the external wall would lead to a significant reduction of 21.6% in the cooling load at 21:00 (in Case 2) as compared with that in Case 1. However, as the width of the air gap was gradually increased from 10 mm to 40 mm (in Case 2 to 6), although the total cooling load at 21:00 was also gradually reduced, the magnitude of reduction was not remarkable, as shown in Figure 6.10. This was because increasing the width of an air gap would not significantly further increase the thermal insulation value of an opaque wall after an air gap was added to it (ASHRAE 2013a). Therefore, in the next three sub-sections, when further modifying the external wall such as ventilating an air gap or adhering aluminum foil to the surfaces of an air gap, an air gap of fixed width of 20 mm was used for carrying out simulation.

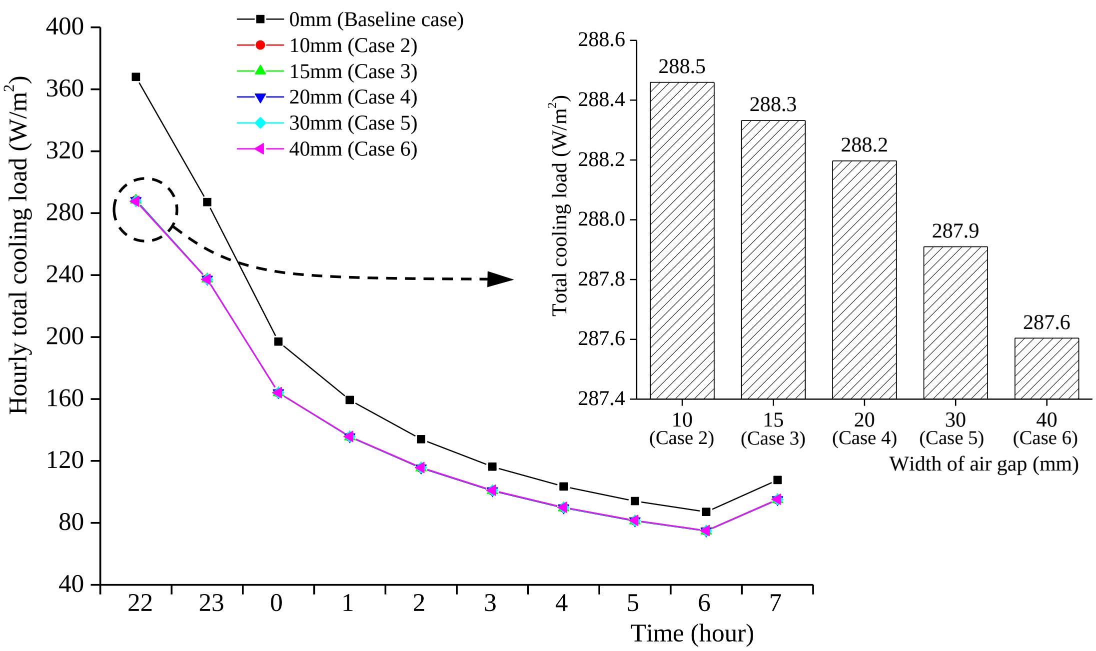


Figure 6.10 The simulated total nighttime cooling load in the bedroom with different widths of air gaps in the external wall

6.4.2. Effects of varying the width of inside and outside concrete layers of the west-facing external wall on the total nighttime cooling load in the studied bedroom

While the sum of δ_o and δ_i was kept constant at 100 mm, both δ_o and δ_i were varied to investigate the effects of varying the width of inside concrete and outside concrete layer with a 20 mm air gap in between, on the total nighttime cooling load in the bedroom and the simulation results are shown in Figure 6.11. As seen, the smaller the width of the outside concrete layer, the more the cooling load reduction. This was because the outside concrete layer was directly exposed to solar radiation, and a thin layer of outside concrete layer would imply less thermal mass available to store thermal energy at daytime. On the other hand, a thicker layer of inside concrete would imply a higher thermal insulation to hinder the thermal energy transfer from the air gap to indoor environment, thus a smaller cooling load. This explains that in Case 10, a lowest total cooling load, among the five cases having the thinnest outside layer and thickest inside layer, was obtained.

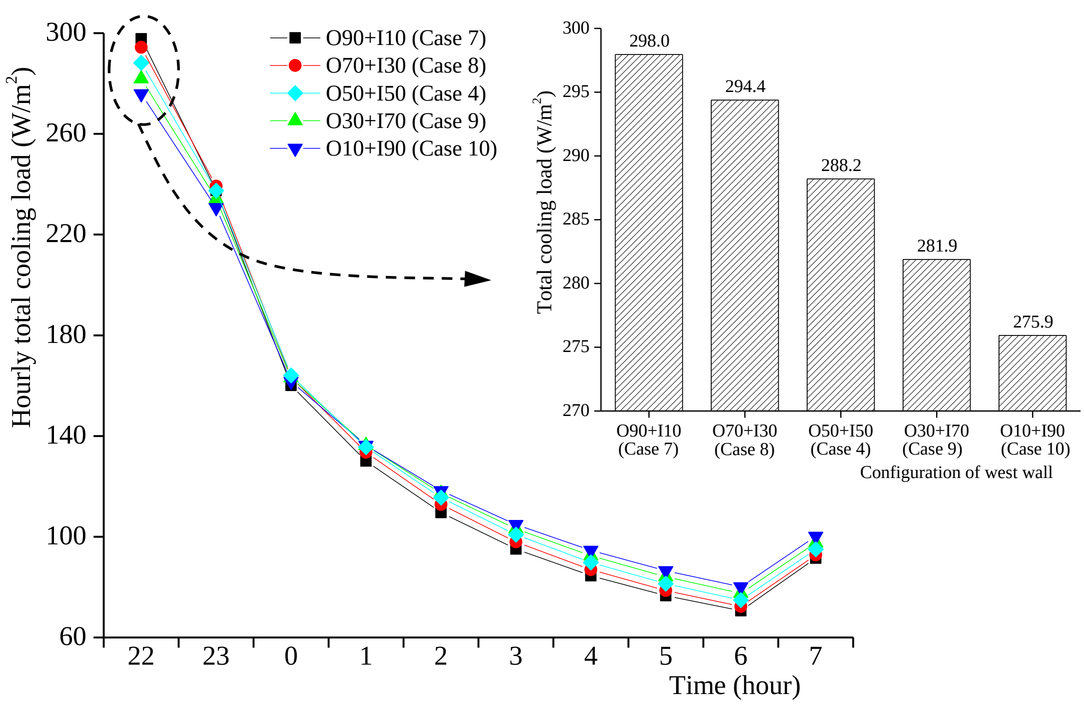


Figure 6.11 The total nighttime cooling load in the bedroom with varying width of inside and outside concrete layers of the west-facing external wall in the studied bedroom in the summer design day

6.4.3. Effects of ventilating the air gap on the total nighttime cooling load in the studied bedroom in the summer design day

Although inserting an air gap in the external wall was effective in reducing the total nighttime cooling load in the studied bedroom, as shown in Figure 6.12 for the results of Case 10, air temperature in the air gap stayed higher than outdoor air temperature between 09:00 and 02:00 in the following day, represented by the shaded area “B” in the figure. The highest air temperature of 49 ^{o}C in the gap occurred at around 17:30, much higher than the outdoor air temperature at that time. Therefore, if the gap could be mechanically ventilated, air temperature in the gap would be lowered to help reduce nighttime cooling load. Therefore, in Case 11, the effects of ventilating the air gap on nighttime cooling load reduction were investigated and compared with that in Case 10. The investigation was divided into two parts. Firstly, the air gap was ventilated from one to nine hours, with one hour increment, prior to the A/C operation. Secondly, the air gap was only ventilated for one hour between 12:00 and 21:00, during the non-occupied period. In Case 11, an axial fan with a fixed air flow rate of 0.2425m^3/s and a medium pressure at a rated power input of 25W was used.

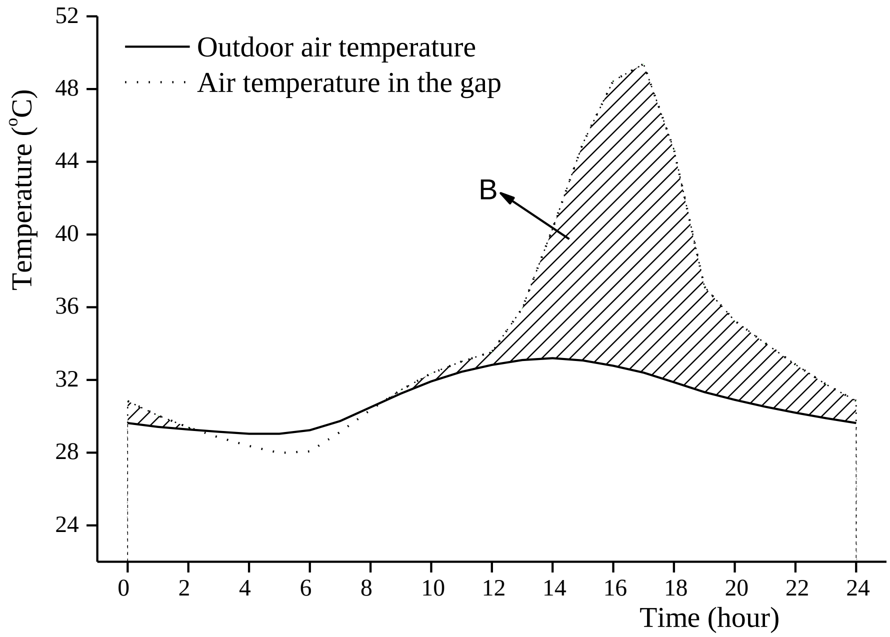


Figure 6.12 The simulated profiles of the air temperature in the air gap and outdoor air temperature in the summer design day in Case 10

Since mechanical ventilation also consumed electrical energy, a potential energy efficiency index (Blondeau et al. 1997), *PEE*, was used to reflect the effectiveness of ventilating the air gap on electrical energy saving for nighttime space cooling:

$$PEE = \frac{Q_{reduction} / COP}{Q_{fan}} \tag{6.3}$$

Where $Q_{reduction}$ is the total nighttime cooling load reduction during nighttime A/C from 21:00-07:00 in the following day (W/m^2) compared with that in Case 10; *COP*, Coefficient of Performance for the WRAC, assumed at 3.0; Q_{fan} the electricity consumption of the ventilation fan (W/m^2). Therefore, $Q_{reduction}$ / *COP* represents A/C electrical energy reduction (A/C EER). Equation (6.3) indicates that when *PEE* > 1, ventilating leads to net electrical energy saving.

The simulation results for the two parts in Case 11 are shown in Figure 6.13 and Figure 6.14, respectively. As seen in Figure 6.13, the longer the ventilation time, the higher the A/C EER. However, *PEE* displayed a different variation pattern, peaking at 1.27 for 5-hour ventilation in advance of the A/C operation. Furthermore, based on *PEE* values, ventilating the air gap for 4-8 hours can lead to net electrical energy saving, with the largest net saving when the gap was ventilated for 5 hours. This suggests that ventilating the air gap for a longer time does not necessarily yield a greater net electrical energy saving since a longer ventilation time increased fan electrical consumption. On the other hand, in Figure 6.14, when ventilating the air gap for only one hour during the unoccupied period between 12:00 and 21:00, the *PEE* values at each hour was significantly varied. The highest *PEE* value of 2.18 was during 17-18 when the temperature in the gap was the highest, as shown in Figure 6.12. Therefore, the higher the air temperature in the gap, the more the net electrical energy saving.

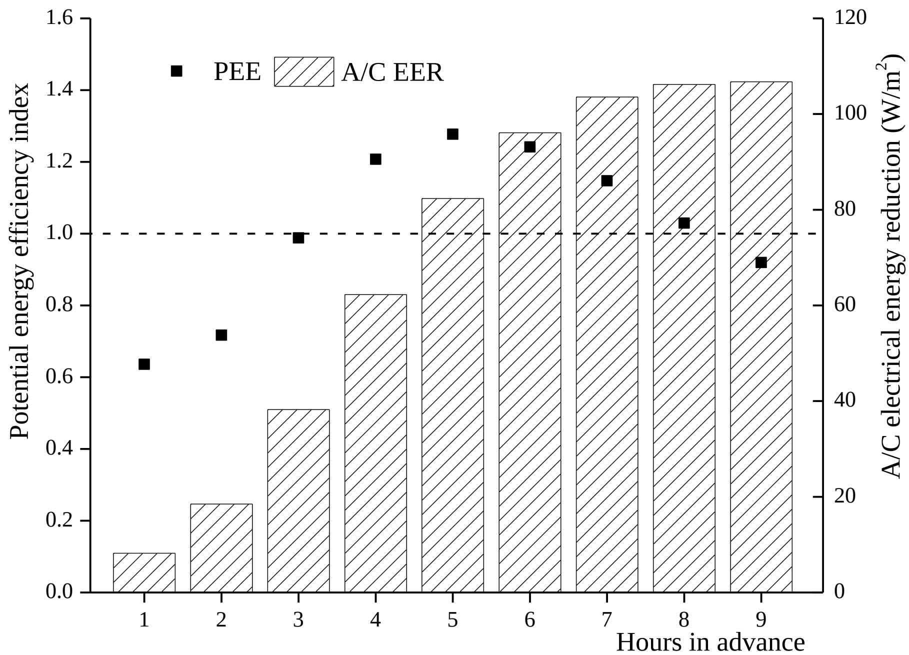


Figure 6.13 The simulation results of ventilating the air gap for different duration in Case 11

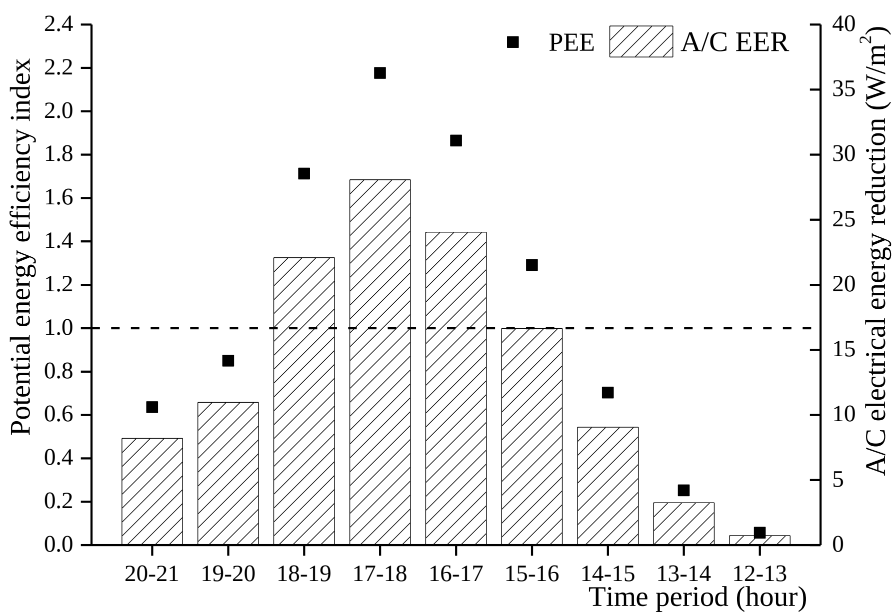


Figure 6.14 The simulation results of ventilating the air gap for only one hour during unoccupied period between 12:00 and 21:00 in Case 11

6.4.4. Effects of adhering aluminum foils on the surfaces of an air gap on the total nighttime cooling load

Inside an air gap, radiation between its two surfaces at different temperatures was the dominating form of heat transfer. For a concrete surface having a high emissivity of 0.9 [135], a high level of radiation heat exchange between the two surfaces of the air gap can be expected. Thus, in order to reduce the radiative heat transfer from outside concrete layer at a higher temperature to the inside one at a lower temperature, thus reducing nighttime cooling load, in Case 12, the effects of adhering aluminum foils to Surface I-O and Surface O-I on nighttime cooling load reduction were studied, and compared with that in Case 10. As suggested (Al-Homoud 2005; Chang et al. 2008; Fairey 1986), the use of aluminum foil with ultralow radiance emissivity of 0.05 can form an effective barrier for radiation heat transfer.

Figure 6.15 shows the simulation results as well as the comparisons with those in Case 1 and Case 10. As seen, when the aluminum foils were adhered on both surfaces of the air gap, there was a considerable reduction in the total nighttime cooling load at 21:00 by 47.2% and 29.6%, as compared with those in Cases 1 and 10, respectively. This was because a relatively higher surface radiative resistance was resulted in after adhering aluminum foil in comparison with that of concrete, to effectively lower the radiative heat transfer from higher temperature outside concrete layer to lower temperature inside concrete layer.

A comparison between the surface temperatures and the total heat gains on the external surface of the west-facing external wall in the studied bedroom, Surface O-O, in Case 10 and that in Case 12 is shown in Figure 6.16. As seen, the temperature of Surface O-O after adhering aluminum foil (Case 12) was higher than that without aluminum foil (Case 10), leading to a significantly increased convective heat transfer to outdoor air and longwave infrared thermal radiation to its surroundings and sky at daytime. Therefore, the total heat gain from Surface O-O in Case 12 was relatively lower than that in Case 10, resulting in a significant reduction on the nighttime total cooling load.

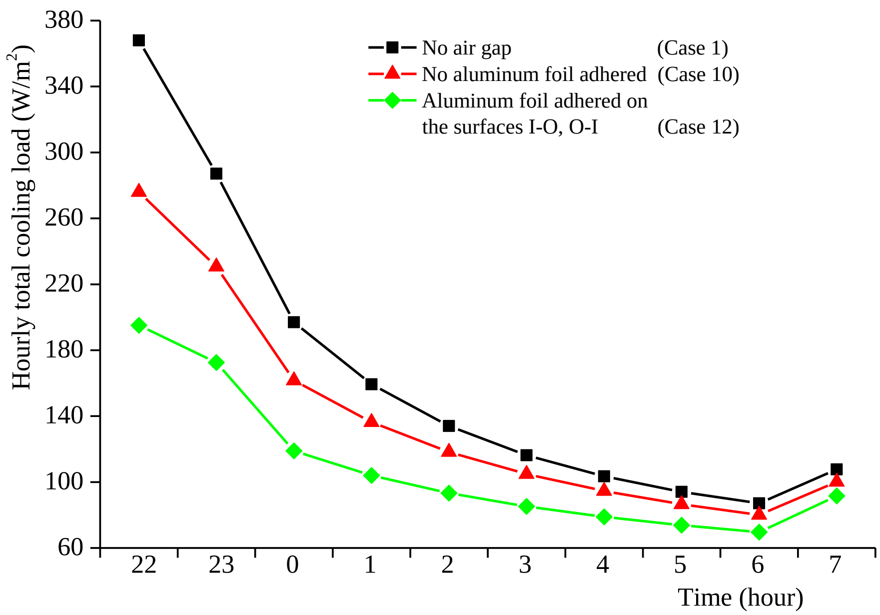


Figure 6.15 The simulated hourly total cooling loads in the bedroom in Case 1, 10 and 12 in the summer design day

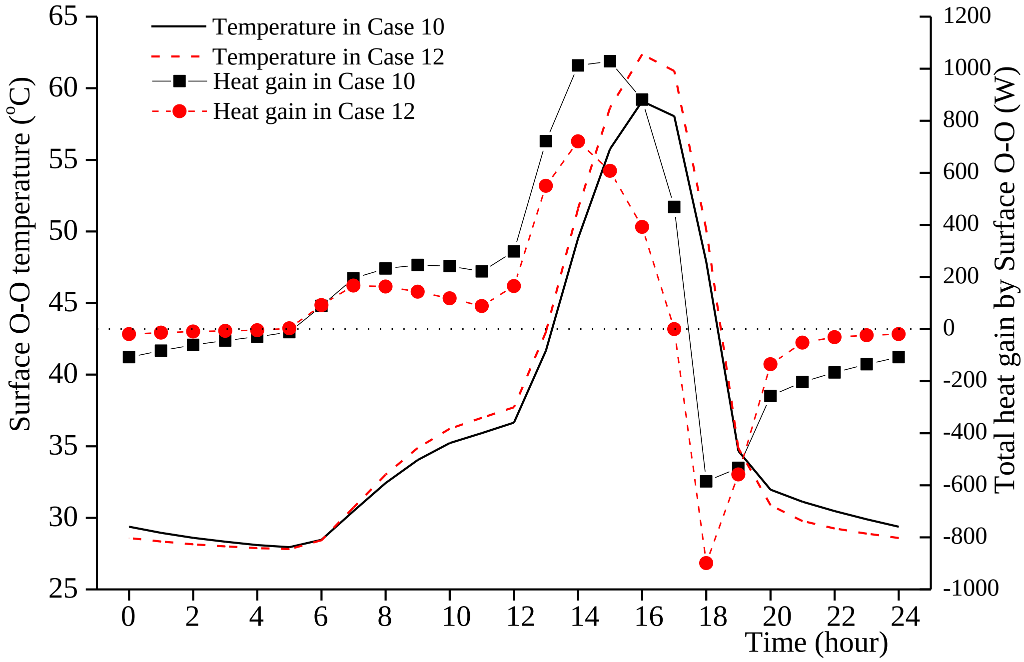


Figure 6.16 The simulated surface temperatures and the total heat gains on Surface O-O in Case 10 and Case 12

6.4.5. Discussions

The effects of different wall modifications on the total nighttime cooling load reduction were separately studied earlier in 12 study cases. In this sub-section, these 12 cases were summarized and compared with one another, shown in Table 6.5, using three comparing indexes *SUM*, *RR* and *PCLI*, defined in Equations (6.4)-(6.6), respectively.

$$SUM = \sum_{i=1}^{10} TNCL_i \tag{6.4}$$

$$RR = \frac{SUM_{case1} - SUM_{casej}}{SUM_{case1}} \times 100\% \tag{6.5}$$

$$PCLI = \frac{\left(\sum_{i=3}^{10} TNCL_i\right)/8}{TNCL_1} \times 100\% \tag{6.6}$$

SUM is the sum of the hourly total nighttime cooling load from 21:00-07:00 in the following day (W/m^2), *RR* the reduction rate of *SUM* compared with that in the baseline case (Case 1) and *PCLI* part cooling load index which reflects the flatness of cooling load variation profile over the 10-hour nighttime A/C process, the greater the *PCLI* , the smaller the difference between the maximum and minimum cooling load, respectively.

In Equations (6.4)-(6.6), *TNCL* is the hourly total nighttime cooling load with its subscript *i* indicating the order of the A/C operation hour (W/m^2); For example, i=1 for the first hour, i.e., 21:00-22:00. SUM_{casej} represents the *SUM* value in Case j (j=1, 2⋯12).

Table 6.5 Summary of the 12 simulation cases

Case	Case description	*SUM* (W/m²)	*RR* (%)	*PCLI* (%)
1	Baseline case	1654.1	0	33.9
2	Varying the width of the air gap	1382.1	16.4	37.1
3		1382.4	16.4	37.1
4		1382.6	16.4	37.2
5		1383.0	16.4	37.2
6		1383.3	16.4	37.3
7	Varying the width of inside and outside concrete layer of the west-facing external wall	1353.6	18.2	34.3
8		1371.8	17.1	35.6
9		1387.6	16.1	38.6
10		1389.0	16.0	40.0
11[a]	Ventilating the air gap during 12:00-21:00	1282.3	22.5	41.3
12	Adhering aluminum foil	1083.1	34.5	45.8

[a] Only using the highest A/C EER value of 106.7 W/m^2.

The results in Table 6.5 show that while there was no remarkable difference by varying the air gap width, adding an air gap would lead to significant cooling load reduction of 16.4%, and slightly improve the flatness of cooling load variation profile, as compared to those in Case 1. In Cases 7-10 with a fixed air gap width of 20 mm, varying the width of both the inside and outside concrete layers would lead to different *SUM* and *PCLI* values. Overall speaking, the effects of reducing cooling load and flatness of cooling load variation profile were very much comparable to those of adding an air gap to the west-facing external wall. However, in Case 11, ventilating the air gap for 9 hours prior to starting A/C would result in a load reduction rate of 22.5%, and a *PCLI* of 41.3%, both being better than that of adding an air gap. Nonetheless, ventilating the gap mechanically would lead to additional electrical energy use for a ventilation fan, which should be considered as shown in Section 6.4.3. Finally, in Case 12, adhering aluminum foil to both sides of the air gap would achieve a significant load reduction of 34.5%, as compared to the baseline case. In addition, its *PCLI* of 45.8% were also remarkably higher than that in the baseline case of 33.9%, indicating a much flatter load variation profile.

It must be also pointed out that the above comparisons were only based on load reduction and the flatness of load variation profile during a nighttime A/C process. There are other factors that can also have a role to play. For instance, although ventilating mechanically an air gap can help reduce cooling load and improve the flatness of variation load profile, additional energy consumption will be required to operate a ventilation fan to achieve the required ventilation rate. In addition, the costs involved in different cases will also be different, and need to be considered when making a final decision in adopting a specific wall modification.

6.5. Conclusions

A simulation study on further analyzing the cooling load characteristics in a bedroom in Hong Kong at nighttime and on the effects of adding an air gap to the opaque west-facing external wall of the bedroom on reducing thermal energy stored in the wall and consequently the space cooling load at nighttime air conditioning for the bedrooms is reported. The results of analyzing cooling load characteristics showed that, the nighttime cooling load from building envelopes took the lion's share of the total nighttime cooling load in every hour of the A/C process. Among all envelope components, the west-facing external wall contributed most significantly to the total nighttime cooling load, because of its direct exposure to solar radiation, thus the heat gain by, and storage in its thermal mass. This suggested there was a high potential to reduce nighttime space cooling load for the bedroom by modifying the west-facing external wall. Therefore, the effects of adding an air gap in the wall on load reduction were studied.

The study results showed that adding an air gap in the external wall can reduce remarkably the hourly total nighttime cooling load for the first hour (21:00-22:00) and total cooling load for the 10-hour A/C period. However, while varying the width of either the air gap or inside/outside concrete layers would not reduce cooling load further, ventilating mechanically the air gap can further increase the cooling load reduction and result in a flatter load variation profile at the expense of consuming additional electrical energy.

Finally, adhering aluminum foil to both sides of an air gap can further remarkably reduce nighttime cooling load, and improve the flatness of the load variation profile.

Apart from adding an air gap to the west-facing external wall in a bedroom, to reduce its nighttime space cooling load, there may be other measures that can be applied to a bedroom in reducing its nighttime cooling load, such as pre-ventilating the bedroom before occupancy and using light-color surface for the external wall to increase its reflectance so as to absorb less solar radiation, etc. These can however be studied separately in future follow-up studies.

Chapter 7

Conclusions and future work

7.1. Conclusions

A systematic research program to develop a non-convection based TAC system applied to a sleeping environment and to evaluate its performance in terms of thermal comfort control, indoor air quality and energy saving, and to investigate the impacts of daytime heat gain/storage of the external envelopes in a bedroom on its nighttime cooling load and the related mitigation measures, has been carried out and is reported in this Thesis. The conclusions of the Thesis are as follows:

1. To address the cold draft issue encountered when using the previously developed C-TAC systems in sleeping environments, an R-TAC system was developed and its performances experimentally evaluated, as reported in Chapter 4. The study results showed that the differences between air temperature/MRT in the unoccupied zone and those in the occupied zone were significant, demonstrating that a great potential of energy saving when using the R-TAC system can be expected as less energy was needed to cool the unoccupied zone. A very low level of air velocity in the occupied zone was maintained. The difference between the evaluated PMV values in the occupied zone and that in the unoccupied zone with the same bedding insulation was also significant. In addition, using the R-TAC system could effectively avoid the cold draft problem encountered when using the previously developed C-TAC system, since the majority of space cooling load was taken care by the radiant panel, with

only a very small amount of fresh air being provided. Also, the problem of potential condensation on the panel surface has been observed, but may be resolved by raising the panel surface temperature, or using a lower fresh air supply humidity.

2. To optimize the operating performance of the developed R-TAC system, the effects of the design/operating parameters of the radiant panel, including surface temperature, surface emissivity and area of the radiant panel and the distance between the bed and the panel, on thermal comfort inside an experimental bedroom and energy saving when using the R-TAC system were numerically studied, and the study results are reported in Chapter 5. The numerical method was validated using the experimental data obtained in the experimental study reported in Chapter 4, and the simulated results on indoor thermal comfort and energy performances at different surface temperatures, emissivities and sizes of the radiant panel and different distances between the bed and the panel were analyzed and compared. It was shown that increasing surface temperature can lead to a higher PMV value and a higher EUC value, while increasing surface emissivity and the surface area of the radiant panel can result in a lower PMV value and a lower EUC value. Furthermore, although reducing the distance between the bed and the panel caused a lower PMV value and a higher EUC value, the PMV and EUC values at the distance of 800 mm and those at the distance of 1000 mm were similar. It can also be seen that the differences in the operative temperature between the occupied zone and the unoccupied zone were significant, indicating a higher energy utilization efficiency when using the R-TAC system. Air velocity in the bedroom was at a low level, although it was slightly higher

in the occupied zone. The DR values were at a very low level in all the study cases, indicating the cold draft problem encountered when using a C-TAC system for a sleeping environment can be effectively resolved when using the R-TAC system.

3. Apart from developing a novel A/C systems, A/C energy use in a bedroom during sleep may also be saved by decreasing the nighttime cooling load from external envelopes. Therefore, a simulation study on further analyzing the cooling load characteristics in a bedroom in Hong Kong at nighttime and on the effects of adding an air gap to the opaque west-facing external wall of the bedroom on reducing the thermal energy stored in the wall and consequently the space cooling load at nighttime air conditioning for the bedrooms was carried out and the study results are reported in Chapter 6. The results of analyzing cooling load characteristics showed that, the nighttime cooling load from building envelopes took the lion's share of the total nighttime cooling load in every hour of a nighttime A/C process. Among all envelope components, the west-facing external wall contributed most significantly to the total nighttime cooling load, because of its direct exposure to solar radiation at daytime, thus the heat gain by, and storage in its thermal mass. This suggested there was a high potential to reduce nighttime space cooling load for the bedroom by modifying the west-facing external wall. Therefore, the effects of adding an air gap in the wall on load reduction were studied. The study results showed that adding an air gap in the external wall can reduce remarkably the hourly total nighttime cooling load for the first hour (21:00-22:00) and total cooling load for the 10-hour A/C period. However, while varying the width of either the air gap or inside/outside concrete

layers would not reduce cooling load further, ventilating mechanically the air gap can further increase the cooling load reduction and result in a flatter load variation profile at the expense of consuming additional electrical energy. In addition, adhering aluminum foil to both sides of an air gap can further remarkably reduce nighttime cooling load, and improve the flatness of the load variation profile.

The completed systematic research program reported in this Thesis has made significant contributions to advancing air conditioning technology for sleeping environments, through developing a novel R-TAC system to achieve a high level of thermal comfort control, ventilation effectiveness and energy efficiency, and to resolve the cold draft problem encountered in the previously developed C-TAC system, and through further studying the impacts of daytime heat gain/storage in the external envelopes in a bedroom on the nighttime cooling load and proposing the effective mitigation measures to help reduce the nighttime cooling load.

7.2. Proposed future work

Following on the successful completion of the systematic research program reported in this Thesis, a number of possible future studies are proposed:

1. In the experimental study reported in Chapter 4, the problem of potential condensation on the panel surface was observed, which may cause mould growth on the panel and inconvenience to the occupants. To avoid the condensation problem, follow-up studies on investigating moisture distribution in the bedroom when using

the R-TAC system and its related mitigation measures, such as raising the panel surface temperature, or using a lower fresh air supply humidity, should be therefore carried out.

2. In both the experimental and numerical studies reported respectively in Chapter 4 and Chapter 5, the experimental setup was built in an environmental chamber where the simulated outdoor air temperature was fixed at 30 ^{o}C, and a thermal manikin was used to assemble a sleeping person. Therefore, the actual thermal sensation for a real person may not be accurately predicted. Therefore, in the future, a subjective survey on the thermal comfort and a field study on energy saving, in bedrooms exposed to real weather conditions when using the R-TAC system should be conducted.

3. The effectiveness in cooling load reduction through adding an air gap to the west-facing external wall in the simulated bedroom was extensively studied by simulation, as reported in Chapter 6. However, there may be other measures that can be applied to a bedroom in reducing its nighttime cooling load, such as pre-ventilating the bedroom before occupancy and using light-color surfaces for the external walls to increase its reflectance so as to absorb less solar radiation, etc. These can also be studied in future.

Appendix A—Photos of the experimental setup

Photo 1 The environmental chamber

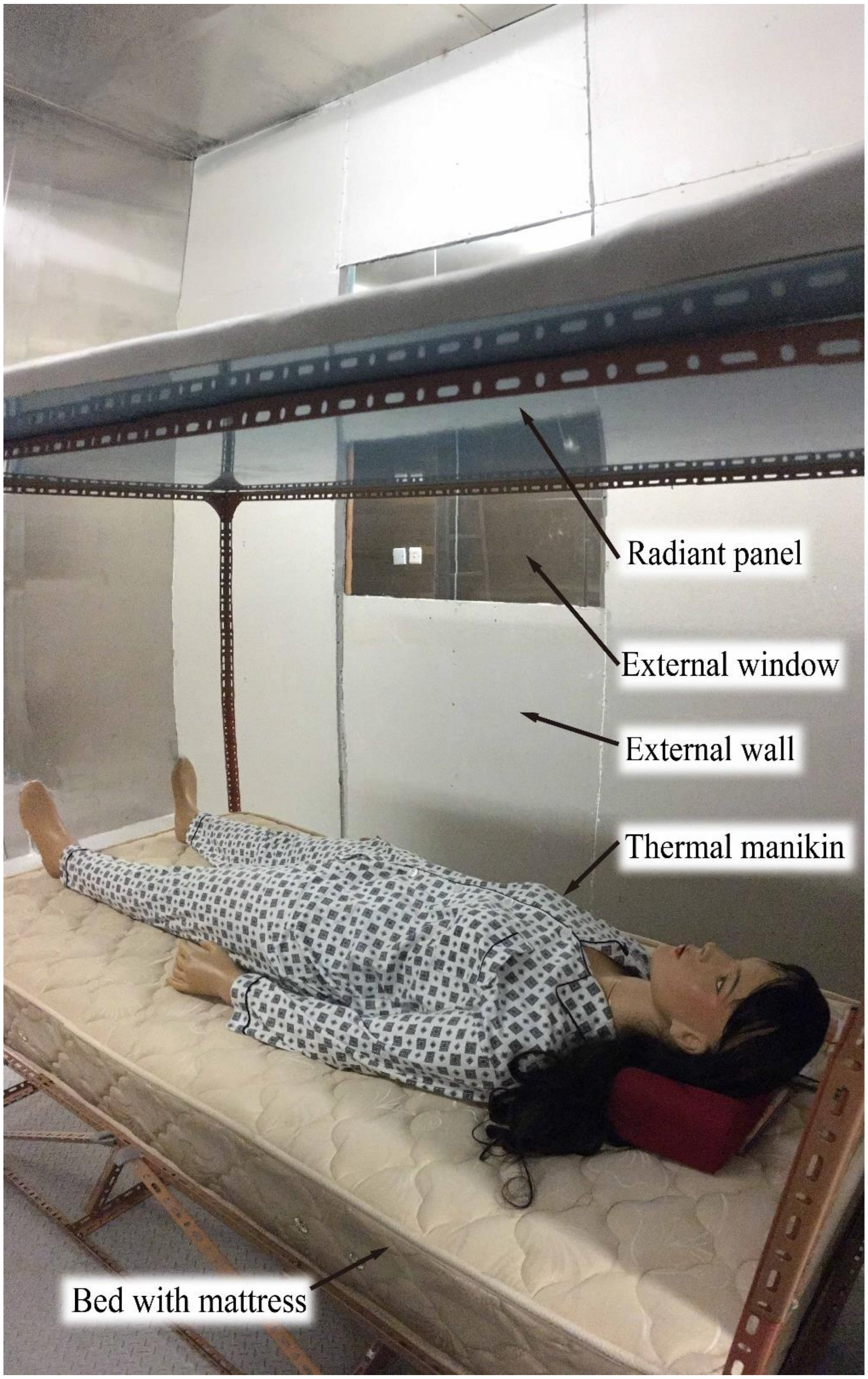


Photo 2 The experimental bedroom

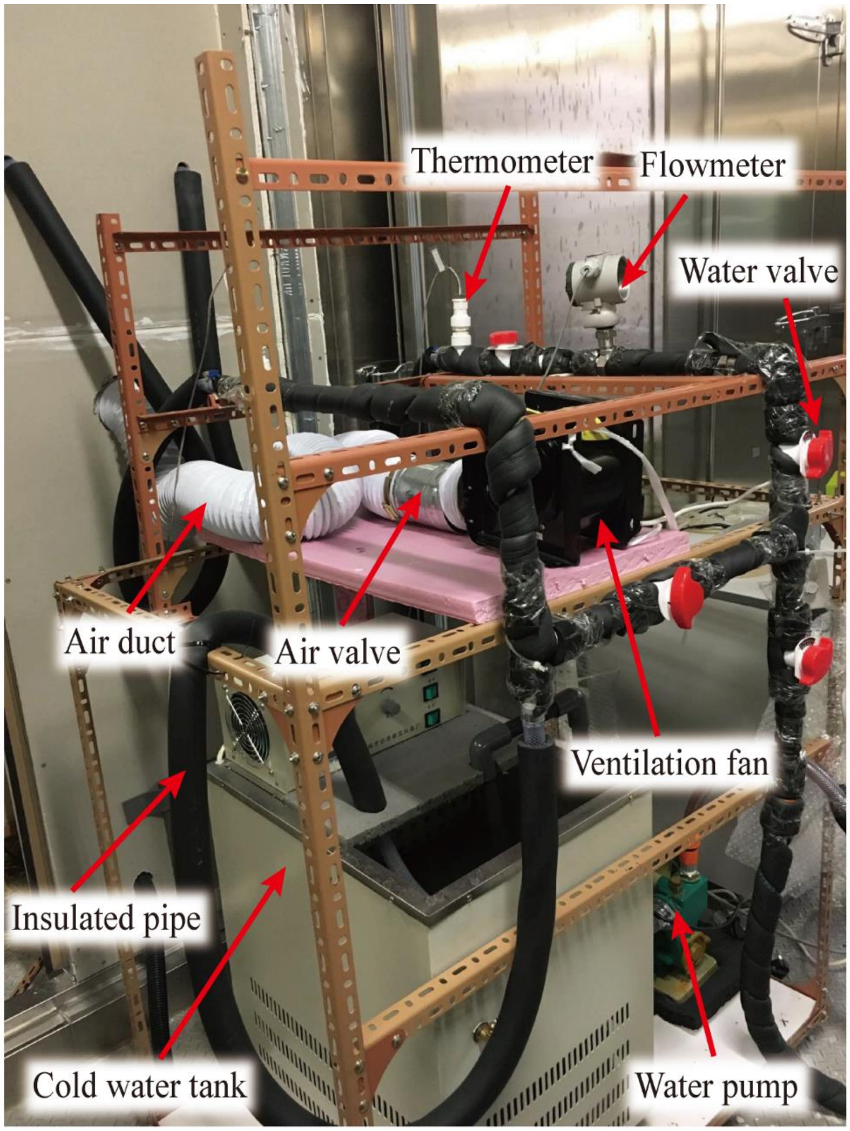
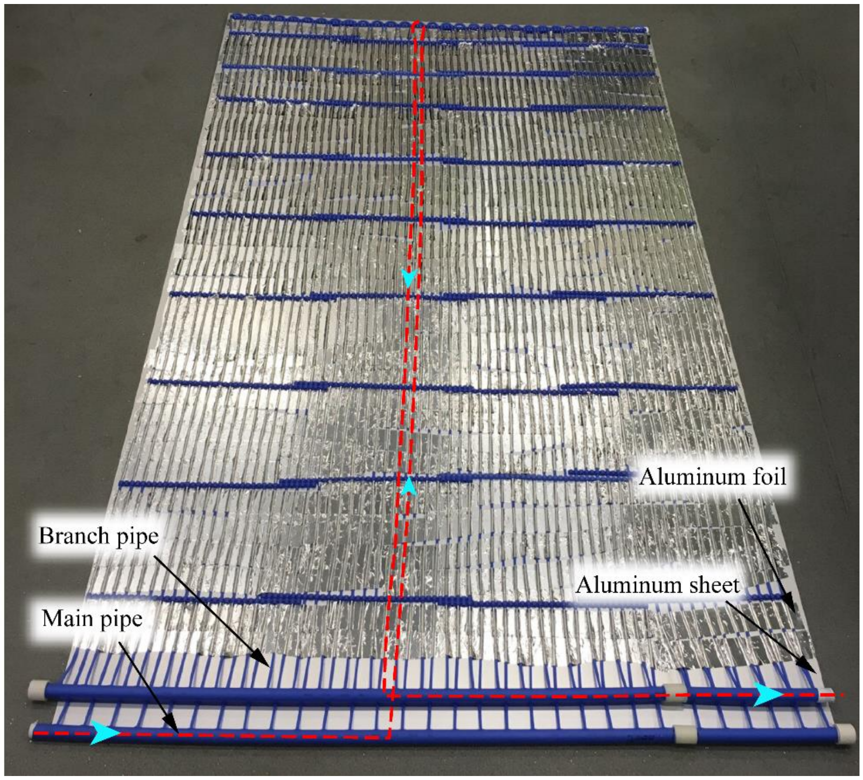


Photo 3 Water cooling system and ventilation system in the plant room



(a) The inner construction of the radiant panel

(b) The surface of the radiant panel facing ceiling

Photo 4 Details of the radiant panel components

Photo 5 Load generation units

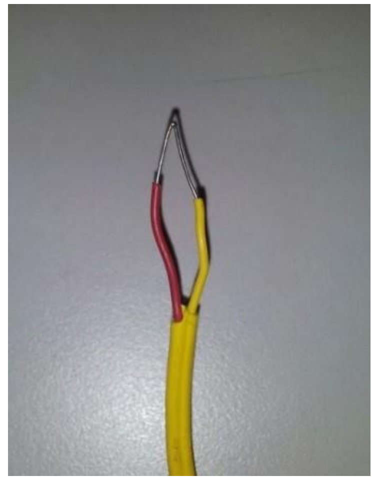


Photo 6 K-type thermal couple

Photo 7 Water flowmeter

Photo 8 Air flowmeter

Photo 9 Air velocity transducer

Photo 10 Data logger

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Sleep duration

External wall

Supply air outlet

Window

Return air inlet

Bed with mattress

Thermal manikin