Strategies for Bedroom Air Conditioning in the Subtropics

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Certificate of Originality

I hereby declare that this thesis is my own work and that, to the best of my knowledge and belief, it reproduces no material previously published or written, nor material that has been accepted for the award of any other degree or diploma, except where due acknowledgement has been made in the text.

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Abstract

In the subtropics, air conditioning serves to maintain an appropriate indoor thermal environment not only in workplaces during daytime, but also at nighttime for sleeping in bedrooms in residences or guestrooms in hotels. However, current practices in air conditioning, as well as the thermal comfort theories on which these practices are based, are primarily concerned with situations in which people are awake in workplaces at daytime. Therefore, these may not be directly applicable to air conditioning for sleeping environments.

The thesis reports, first of all, on both a questionnaire survey of the current status of sleeping thermal environments and bedroom air conditioning, and a field monitoring of overnight indoor air temperature and relative humidity in a number of air conditioned bedrooms, in residential buildings in subtropical Hong Kong. This was to gather relevant background information such as the types of air conditioning systems employed in bedrooms, the use of bedding and sleepwear while sleeping and the preferred indoor air temperature settings in bedrooms by occupants, etc.

Secondly, a theoretical study on thermal comfort for sleeping environments is presented. A comfort equation applicable to sleeping thermal environments was derived by introducing appropriate modifications to Fanger's comfort equation. In order to solve the comfort equation, the total thermal insulation provided by bedding systems, which is an important parameter included in the comfort equation, must be obtained. Therefore, an experimental study using a thermal manikin on measuring the total insulation values for a wide range of the bedding systems commonly used in the subtropics has been undertaken. A small-scale database of the total insulation

values provided by the bedding systems has been developed. Comfort charts have been established, and can be used for determining thermally neutral environmental conditions under a given bedding system.

Thirdly, a simulation study to investigate the characteristics of nighttime bedroom cooling load in the subtropics, using a building energy simulation program - EnergyPlus, is presented. The results of the simulation study showed that the characteristics of nighttime bedroom cooling load differed significantly from those at daytime. The total cooling load peaked at the starting time of a nighttime air conditioning process, when the heat stored inside building envelopes and furniture dominated. A method of sizing room air conditioners (RACs) for bedroom was proposed based on the known characteristics of nighttime bedroom cooling load.

Finally, the thesis reports on a study on the outdoor air ventilation rate in bedrooms where RACs are employed. Field measurements of both the indoor overnight carbon dioxide (CO_2) levels and the outdoor air ventilation rates in bedrooms employing RACs, as well as laboratory experiments on the ventilation characteristics of RACs were carried out. The results of the study suggested that the outdoor ventilation rates in the studied bedrooms equipped with RACs could not meet the ventilation requirement specified in ASHRAE Standard 62-2001 even if there was only one occupant in a bedroom. The ventilation damper currently available in a window-type room air conditioner (WRAC) cannot be expected to provide the ventilation rate as required by a ventilation code and its intended function of controlling ventilation was limited. Therefore, an improved design for a WRAC has been proposed to improve its ventilation control and to save energy. A new ventilation rate of 3.0 L/s per person for sleeping environments has also been proposed.

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Nomenclature

Variable	Description	Unit
a_i	Area fraction of segment (i) related to whole body area A_{sk}, $a_i = a_{sk,i}/A_{sk}$	ND
A	Average air change within a time period	h^{-1}
A_C	Percentage coverage of body surface area by bedding and bed	%
A_{cb}	Contact area between bed and body	m^2
A_{cl}	Surface area of clothed body	m^2
A_D	DuBois surface area of nude body	m^2
A_r	Effective radiation area of body	m^2
$c_{p,b}$	Specific heat of body tissue	J/(kg·K)
C	Convective heat loss	W/m^2
C_o	Outdoor air CO_2 concentration	ppm
C_Q	Air flow coefficient of air flow passage	$m^3/s \cdot Pa^n$
C_{res}	Sensible heat loss due to respiration	W/m^2
C_s	Space air CO_2 concentration	ppm
$C+R$	Total sensible heat loss from skin	W/m^2
$C(0)$	Concentration of a tracer gas at the initial time	ppm
$C(T)$	Concentration of a tracer gas at any time, T	ppm
$CLQ_{i,j,t}$	Hourly total cooling load of the bedroom j during one hour time interval, i = 1, 2, …, 1840.	W/m^2
$CLQ_{i',j,t}$	Hourly total cooling load of the bedroom j during one hour time interval in the summer design day, i' = 1, 2,…, 10.	W/m^2
$CONF_A$	Confidence limits on A	h^{-1}
E_{dif}	Evaporative heat loss due to moisture diffusion through skin	W/m^2
E_{max}	Possible maximum evaporative heat loss	W/m^2
E_{res}	Evaporative heat loss due to respiration	W/m^2
$E_{rew,req}$	Evaporative heat loss due to regulatory sweating	W/m^2

	required for comfort	
E_{rsw}	Evaporative heat loss due to regulatory sweating	W/m^2
E_{sk}	Total evaporative heat loss from skin	W/m^2
ESE_A	Estimated standard error of A	h^{-1}
ET^*	Effective temperature, based on 50% RH	°C
f_{cl}	Clothing area factor, A_{cl}/A_D	ND
F_{P-j}	Angel factor between manikin and surface j	ND
h	Total sensible heat transfer coefficient	$W/(m^2·K)$
h_c	Convective heat transfer coefficient at surface	$W/(m^2·K)$
h_r	Radiative heat transfer coefficient	$W/(m^2·K)$
h_e	Evaporative heat transfer coefficient at surface	$W/(m^2·kPa)$
H_i	Local heat flux of segment (i)	W/m^2
H_{sk}	Whole body heat flux	W/m^2
i	One hour time interval when the WRACs are operated, i = 1, 2,…, 1840 (i.e., from 22:00 to 7:00 of the following day from 1 May to 31 October, totally 1840 hours)	ND
i'	One hour time interval when the WRACs are operated in summer design day, i' = 1, 2,…, 10	ND
i_m	Total vapor permeation efficiency	ND
I_T	Total insulation of a bedding system including the air layer around the covered body	clo
j	Number of surface (for Equation 5.44 only)	ND
j	Number of bedroom, j = 1, 2, 3	ND
K	Unit constant (6.45 clo·W/(m^2·°C))	ND
l	Body height	m
L	Thermal load on body	W/m^2
L_R	Lewis ratio	K/kPa
m	Body mass	kg
m_{cr}	Mass of body core	kg
m_{sk}	Mass of skin	kg
M	Metabolic heat production	W/m^2
$MRV_{j,t}$	Rate of variation in the maximum total cooling load	%

	for bedroom j at t	
n	Number of testing points	ND
n	Air flow exponent, n≈ 0.5 (for Equation 7.5 only)	ND
N	CO_2 generation rate per person	L/min
p_a	Water vapor pressure in ambient air	kPa
$p_{ET*,s}$	Water vapor pressure in saturated air at $ET*$	kPa
$p_{sk,s}$	Water vapor pressure in saturated air at t_{sk}	kPa
q_{res}	Total heat flow due to respiration	W/m^2
q_{sk}	Total heat flow from skin	W/m^2
Q	Airflow rate	m^3/s
Q_C	Convective heat exchange	W/m^2
R	Radiative heat loss from skin	W/m^2
R_a	Thermal resistance of an air layer	$m^2·°C /W$
R_{cl}	Thermal resistance of clothing	$m^2·K/W$
$R_{e,cl}$	Evaporative resistance of clothing	$m^2·kPa/W$
$R_{e,t}$	Total evaporative resistance	$m^2·kPa/W$
R_t	Total thermal resistance	$m^2·K/W$
R_T	Total resistance of a bedding system including the air layer around a covered body	$m^2·°C /W$
RH	Relative humidity	%
S	Heat storage	W/m^2
S_{cr}	Heat storage in core compartment	W/m^2
S_{sk}	Heat storage in skin compartment	W/m^2
t	Indoor design air temperature, t = 21, 22,…, 27	°C
t_a	Ambient air temperature	°C
t_{cl}	Clothing surface temperature	°C
$t_{current}$	Air temperature at the current hour in design day	°C
t_j	Surface temperature of surface j	°C
t_{Max}	Maximum dry-bulb (DB) temperature in design day	°C
$t_{multiplier}$	Daily range multiplier shown in Fig. 6.2	ND
t_o	Operative temperature	°C
t_{range}	Daily temperature range in design day	°C
$\bar{t}_r$	Mean radiant temperature	°C

t_{re}	Body core temperature	°C
$\bar{t}_{sk}$	Skin temperature	°C
$t_{sk,i}$	Local skin temperature of a segment (i)	°C
$\bar{t}_{sk,req}$	Skin temperature required for comfort	°C
T	Time	s
$TCE_{i,j,t}$	Yearly total cooling energy of the bedroom j	kWh/m^2/Yr
$TRV_{j,t}$	Rate of variation in the yearly total cooling energy for bedroom j at t	%
v	Air velocity	m/s
V_o	Outdoor air flow rate per person	L/s·person
w	Skin wettedness	ND
W	External work accomplished	W/m^2
Y_i	=lnC(T)$_i$, (i=0~n)	ND
$\hat{Y}_i$	Estimate of lnC(T)$_i$, (i=0~n)	ND
α	Sensitivity coefficient for evaluating *PMV*	ND
α_{sk}	Fraction of total body mass concentrated in skin compartment	ND
ε	Emissivity	ND
σ	Stefan-Boltzmann constant (= 5.67×10^8)	W/(m^2·K^4)
ΔP_{AO}	Static pressure different between A (A’) point and O (O’) point (shown in Fig. 7.11)	Pa

Note: ND = No Dimensions

Subscripts

a	Air
c	Convective
cb	Body
cl	Clothing
dif	Diffusion
e	Evaporative
i	Body segment (for Chapter 5)
i	Time interval (for Chapter 6)
i	Testing point (for Chapter 7)
j	Number of bedroom or surface
r	Radiative
req	Required for comfort
res	Respiration
rew	Regular sweating
s	Saturated
sk	Skin
t	Indoor air temperature setting

Chapter 1

Introduction

With rising living standards and expectations for thermal comfort, air conditioning has gradually come to be considered a necessity. This can be seen from the fact that residential air conditioning has increasingly become common. Consequently the increased use of residential air conditioning has had a significant impact on the total amount of energy use in residences.

In the subtropics, air conditioning serves to maintain an appropriate indoor thermal environment not only in workplaces during daytime, but also at nighttime for sleeping in bedrooms in residences or guestrooms in hotels. However, current practices in air conditioning, as well as the thermal comfort theories on which these practices are based, are primarily concerned with situations in which people are awake in workplaces at daytime. Therefore, these may not be directly applicable to air conditioning for sleeping environments. Thus, it becomes necessary to develop appropriate strategies for bedroom air conditioning.

One fundamental issue in developing air conditioning technology is the theory on thermal comfort, whether for a workplace at daytime or a sleeping environment at nighttime. Currently the theories on thermal comfort in workplaces at daytime are well established. However, research on thermal comfort for sleeping environments at nighttime is limited and more work concerning the thermal comfort of people during sleep is needed. In addition, both the internal and external conditions for bedroom air conditioning at nighttime are different from those at daytime. These include, for example, that there is virtually no solar heat gain at nighttime, and that the thermal insulation provided by clothing at daytime is replaced by that from a bedding system consisting jointly of bed, mattress, sleepwear, and the bedding used by people.

The thesis begins with an extensive literature review on thermal comfort and energy use for air conditioning in residences environments, focusing on the theories of thermal comfort in workplaces at daytime, the related research work on sleep carried out by medical professionals, and the limited existing research work on thermal comfort for sleeping environments. A brief review of the residential electricity consumption in subtropical Hong Kong is also included. The issues related to air conditioning in bedrooms, such as cooling load calculation and fresh air supply are also reviewed. A number of important issues where further in-depth research work in maintaining appropriate sleeping thermal environments in bedrooms in the subtropics is required have been identified. These are the expected targets of investigation in the project reported in this thesis.

The research proposal covering the aims and objectives, the project title and the methodologies adopted in this project is presented in Chapter 3.

Chapter 4 reports on both a questionnaire survey on the current status of sleeping thermal environments and bedroom air conditioning, and a field monitoring of overnight indoor air temperature and relative humidity in a number of air conditioned bedrooms in residential buildings in Hong Kong. The survey focused on human factors such as sleeping habits, the use of bedding and sleepwear while sleeping, and the preferred indoor air temperature settings in bedrooms by occupants, etc.

A theoretical study on thermal comfort for sleeping environments is presented in Chapter 5. A comfort equation applicable to sleeping thermal environments was derived by introducing appropriate assumptions and modifications to Fanger's comfort equation. In order to solve the comfort equation, the total thermal insulation provided by bedding systems, which is an important parameter included in the comfort equation, must be obtained. Therefore, an experimental study using a thermal manikin was undertaken to measure the total insulation values for a wide range of the bedding systems (through different combinations of beds, bed sheets, bedding and their percentage coverage over the manikin, and sleepwear) commonly used in the subtropics. A small-scale database of the total insulation values provided by the bedding systems has been developed. By solving the comfort equation, comfort charts have been established, and can be used for determining thermally neutral environmental conditions under a given bedding system. Both PMV and PPD indexes can also be calculated for the purpose of assessing thermal comfort, if and when necessary. A number of issues related to the effect of the total insulation value provided by a bedding system on thermal comfort, the use of summer quilts and air conditioning culture were discussed.

The simulation study reported in Chapter 6 investigated the characteristics of nighttime bedroom cooling load in the subtropics, using a building energy simulation program - EnergyPlus. The weather conditions of, and the typical arrangements of high-rise residential blocks in Hong Kong were used in the simulation study. The simulation results including the cooling load characteristics in bedrooms under three different operating modes of RACs in the summer design day, the breakdown of the total cooling load in a bedroom at nighttime operating mode (NOM), indoor air temperature and mean radiant temperature variation at NOM, and the effects of indoor design air temperature on the cooling load characteristics at NOM were presented. The results of the simulation study showed that the characteristics of nighttime bedroom cooling load differed significantly from those at daytime. The total cooling load peaked at the starting time of a nighttime air conditioning process, when the heat stored inside building envelopes and furniture dominated. A method of sizing RACs for bedroom was proposed based on the known characteristics of nighttime bedroom cooling load.

Chapter 7 reports, first of all, on field monitoring of both indoor overnight CO_2 levels and outdoor ventilation rates in bedrooms employing RACs, so the current situation of ventilation in residential buildings in Hong Kong can be appreciated. This is followed by reporting the results of laboratory experiments where two typical RACs were used to exam the outdoor air ventilation characteristics in bedrooms employing RACs. The results of field studies showed that the outdoor ventilation rates in the studied bedrooms equipped with RACs in residences in Hong Kong could not meet the ventilation requirement specified in ASHRAE Standard 62-2001 even if there was only one occupant in a bedroom. Although the use of a WRAC may provide a higher outdoor ventilation rate than the use of a split-type room air conditioner (SRAC), this may be ascribed to the fact that there was more natural infiltration when a WRAC was used. The ventilation damper currently available in a WRAC did not significantly affect the outdoor ventilation rate. Therefore, such a damper cannot be expected to provide the ventilation rate as required by a ventilation code and its intended function of controlling ventilation was limited. In addition, the air exhausted from indoors to outdoors through the ventilation outlet in a WRAC was what had just been cooled by a cooling coil (evaporator). This was unreasonable, because exhausting just-cooled and dehumidified air was a waste of energy. Therefore, an improved design for a WRAC has been suggested to improve its ventilation control and to save energy. Finally, the outdoor ventilation requirement for bedrooms at nighttime, when occupants were asleep, was discussed. A new ventilation rate of 3.0 L/s per person for sleeping environments has been proposed.

The conclusions of the thesis and the proposed future work are presented in the final chapter.

Chapter 2

Literature Review

2.1 Introduction

The invention of air conditioning cannot be ascribed to a certain date. Its development, as Willis Carrier noted, “is the natural outgrowth of busy, intelligent minds aiming towards improvement” occurring over an extended period of time. Although crude forms of air conditioning appeared hundreds of years ago, its true beginnings as an art and science began at the end of the 19th century [Will 1998].

In the early 1900s, air conditioning was first systematically developed by Willis H. Carrier, who has been recognized as the Father of air conditioning [Wang 1994]. Prior to 1922, air conditioning was primarily used for industrial processes, such as producing candy, gum, cheese, matches, and textile, etc. In 1922 the first comfort air conditioning system was installed in a theater in the United States [Lang 1979]. The earliest fully air conditioned high-rise office building was the Milam Building in San Antonio, Texas, built in 1928 [Pauken 1999]. Today air conditioning is widely used for thermal comfort in various types of buildings.

With increased living standards and expectations for thermal comfort, residential air conditioning is becoming widespread. For example, eleven million RACs were sold in mainland China in 2001. In Hong Kong, the annual total sales of RACs were around 400 thousand in 1999 and 2000, respectively [Zhang 2002]. Given that most residential buildings are normally occupied during non-office hours, i.e. evenings /

nighttimes during weekdays and on whole days at the weekends / holidays, for residential buildings located in the subtropical region, residential air conditioning serves to maintain not only an appropriate indoor thermal environment in daytime, but also an appropriate sleeping thermal environment in bedrooms at nighttime.

The use of residential air conditioning in Hong Kong may be regarded as typical to residential air conditioning in the subtropics. Hong Kong is situated just south of the tropic of cancer, which makes it subtropical. Summers are hot and humid, and may last for over 7 months. The diurnal variations in ambient air temperature are comparatively small: the mean daily range in summer is only 4.95 °C, and the relative humidity remains high at over 70% for most of the time [Lam and Hui 1995]. Table 2.1 shows the monthly mean outdoor weather data in the typical reference year (TRY) (1989) in Hong Kong, as this year has been identified to be a representative weather year in a previous study [Wong and Ngan 1993]. It can be seen that both the outdoor air temperature and relative humidity in Hong Kong remained rather high throughout the reference year, in particular from April to October.

Hong Kong became more affluent over the last few decades, with its GDP increased from HK$ 136.8 billion in 1980 [Census and Statistics Department, Hong Kong 1990] to HK$ 1,267.2 billion in 2000 [Census and Statistics Department, Hong Kong 2001] (HK$ 443.0 billion at constant (1980) market prices). With the increase in household income and consequently the living standard, air conditioning has gradually become an essential provision of living for people in Hong Kong. A survey on household electrical appliances and electricity use in 1993 revealed that the ownership level for air conditioning units was about 90% [Lam 1996]. Another survey showed that the penetration rate (i.e. the average number of units for a specific type of appliance per household) for air conditioners reached 2.2 in 2001 [Leung 2002].

Table 2.1 The monthly mean weather data of a typical year (1989) in Hong Kong

Month	t (dry bulb) °C	t (dew) °C	RH (%)	H (kJ/kg)
Jan	15.7	12.1	79.1	37.9
Feb	16.6	11.7	72.8	38.3
Mar	18.6	13.7	73.1	43.4
Apr	22.0	19.5	85.7	58.2
May	25.1	22.5	85.5	68.9
Jun	27.5	24.3	82.7	76.6
Jul	28.8	24.7	78.6	79.1
Aug	28.9	25.0	79.5	80.2
Sep	28.1	23.6	76.6	75.1
Oct	25.1	19.7	72.0	61.8
Nov	21.5	15.3	67.8	49.1
Dec	17.8	12.8	72.5	41.1

Note:
t (dry bulb) - Dry bulb temperature; t (dew) – Dew point temperature;
RH – Related humidity; H – Enthalpy.

The increased use of air conditioning for maintaining a thermally appropriate sleeping environment in bedrooms has a significant impact on the total amount of electrical energy used in residences. However, the current design standards and practices for air conditioning are in fact fundamentally based on the principle of maintaining thermal comfort in workplaces and are concerned with situations in which people are awake. Therefore, they may not be directly applicable to bedroom air conditioning, during the period of sleep at nighttime. Thus, it becomes necessary to develop appropriate strategies for bedroom air conditioning.

This chapter firstly reviews the electrical energy uses both in residential sector and for residential air conditioning in Hong Kong, to illustrate, as an example, the extent to which the increased residential air conditioning would impact on the total electricity use. This is followed by a detailed review on the precious development work on thermal comfort and on various research approaches and protocols used. Thirdly, reviews and discussions on the two major issues relating to developing bedroom air conditioning strategies, i.e., nighttime cooling load calculation and fresh air supply, are also included. Finally, based on the review, a number of important issues where further extensive research work is required are identified and reported.

2.2 Residential electricity consumption in Hong Kong

2.2.1 The total electricity use in Hong Kong

With the rapid economic growth, the last few decades saw a significant increase in energy, especially electricity, use in Hong Kong. Meanwhile, there has been a growing concern on the increased energy use and its impacts on the environment.

Fig. 2.1 shows the statistics of the electricity uses in residential, commercial, industrial sectors and the total electricity use in Hong Kong from 1985 to 2000 [Census and Statistics Department, Hong Kong 1990a, Census and Statistics Department, Hong Kong 2000]. It can be seen that the total electricity consumption rose from 15,928 GWh in 1985 to 36,299 GWh in 2000, representing a total increase of 128% and an average annual increase rate of 5.3%. Residential buildings consumed 3200 GWh and 8954 GWh of electricity in 1985 and 2000, respectively. This represents a total increase of 180% and an average annual increase rate of 6.6% during the period. It can also be seen that the electricity uses in both residential and commercial sectors and the total electricity use increased steadily from year to year. However, the electricity use in industrial sector slightly increased from 1985, leveled in 1989, then decreased slowly thereafter.

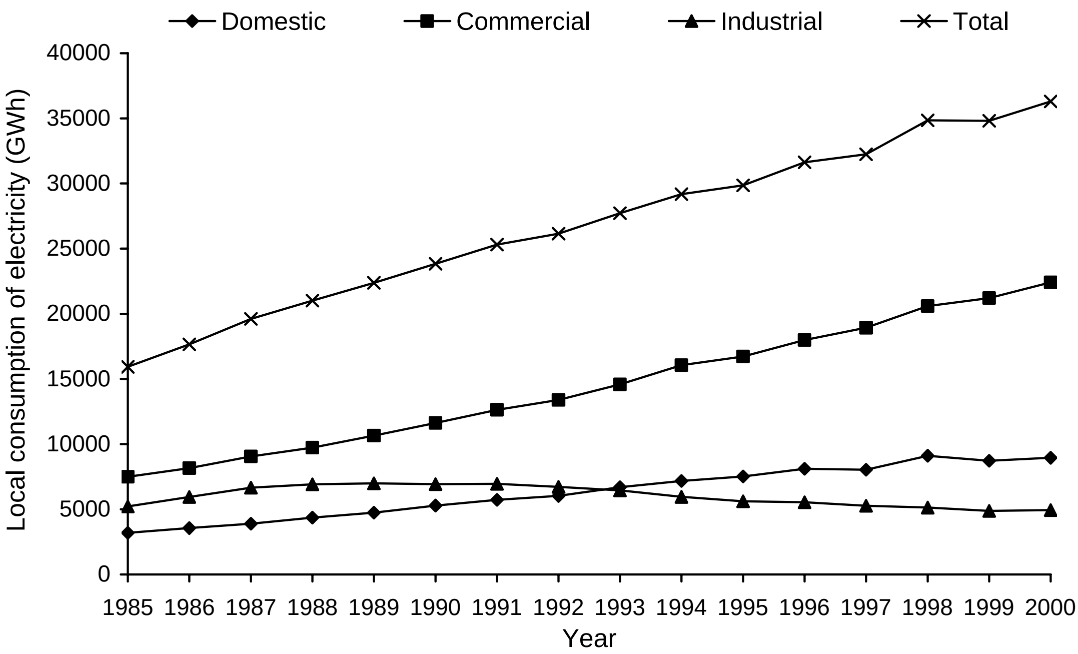

Fig. 2.1 Electricity uses in residential, commercial, industrial sectors and the total electricity consumption in Hong Kong from 1985 to 2000

2.2.2 Electricity use in residential buildings

Fig. 2.2 shows the percentages of residential use in the total electricity consumption in Hong Kong from 1985 to 2000. Residential electricity use accounted for 20-21% from 1985 to 1989. The percentage then increased gradually from 1989 and remained steady at 25-26% between 1994 and 2000. It can be seen that the residential sector was responsible for approximately a quarter of the total electricity use in Hong Kong after 1994.

A study showed that the electricity use for air-conditioning might account for 50-60% of the total electricity use in commercial buildings in Hong Kong [Lam 2000]. However, the situation can be significantly different for residential buildings, due to different natures of the two types of building.

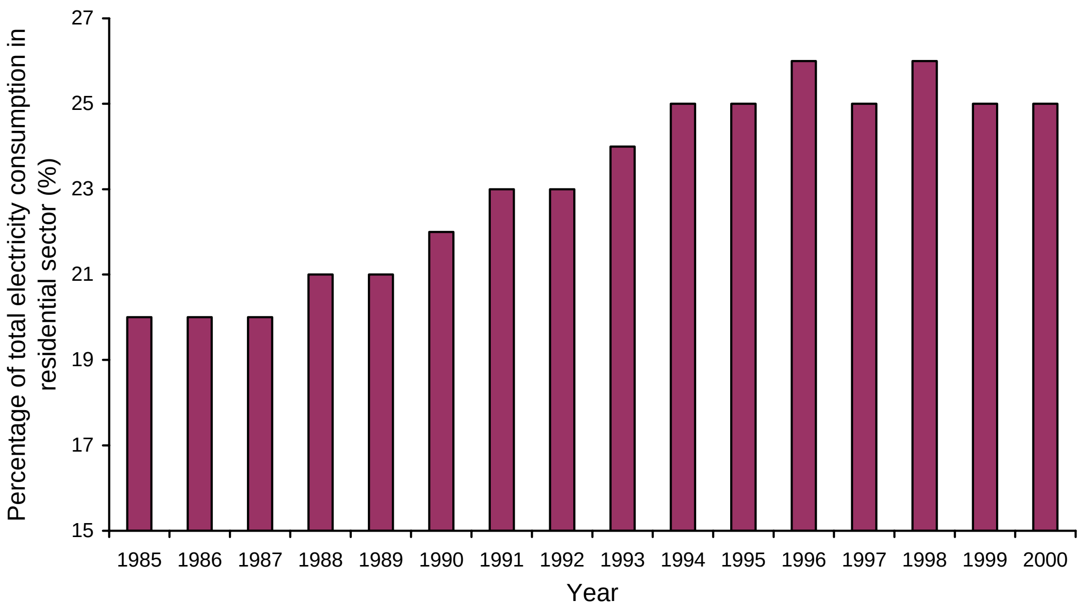

Fig. 2.2 Percentages of residential electricity use in the total electricity consumption in Hong Kong, 1985-2000

Fig. 2.3 shows the normalized monthly electricity use profile for the residential sector in 1998 in Hong Kong [Census and Statistics Department, Hong Kong 1999]. The profile was normalized to account for the average daily electricity use in each month. It is seen that the electricity uses in the months from May to October were clearly higher than those in other months when the electricity uses stayed at below 600 GWh, peaking at over 1200 GWh in August when it was the hottest month in the year. The increase in consumption was mainly due to the increased demand for air conditioning. A survey of user behavior in residential flats in 1993 indicated that most people operated their room air conditioners during the six hot summer months (mid April to mid October) [Lam 1993]. Residential air conditioning consumed 155 GWh of electricity in 1971, representing 14.6% of the total residential electricity use. In 1996, this rose to 2467 GWh, or 30.4% of the total residential electricity use [Lam 2000a].

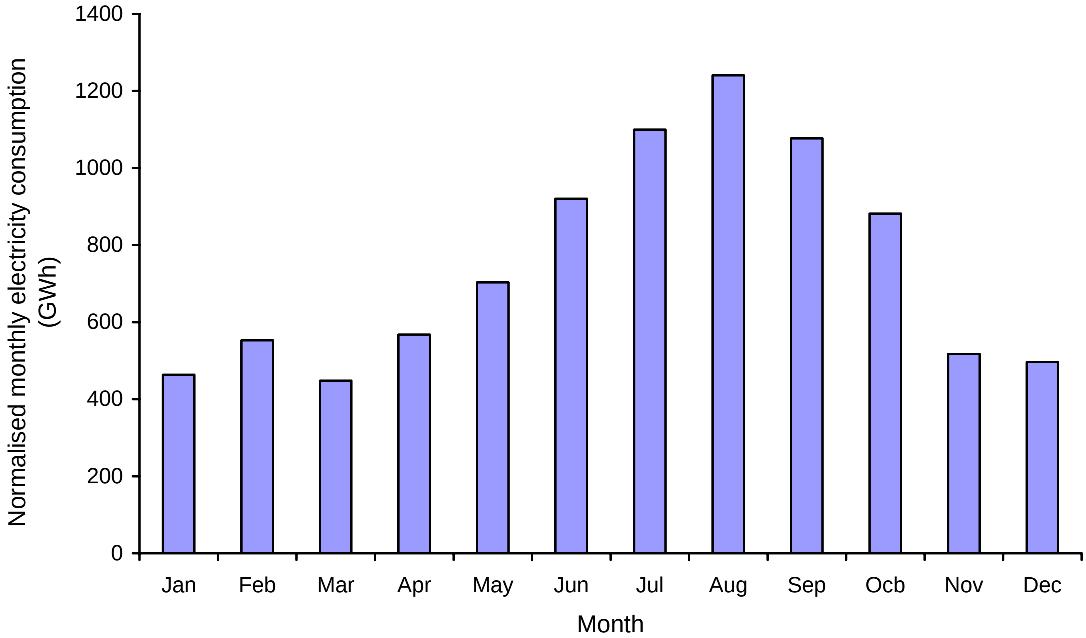

Fig. 2.3 Normalized monthly electricity consumption profile for the residential sector in Hong Kong (1998)

Another study showed that energy uses in communal areas (i.e. lifts, water pumps and communal area lighting) usually accounted for 10-20% of the total residential energy use in Hong Kong [Lam 1996]. End-uses of the household electricity use normally included refrigerators, television sets, videocassette recorders, washing machines and tumble driers, air-conditioners, electrical fans, lightings, etc. Air-conditioning was by far the largest single electricity-consuming end-use, accounting for 36.8% of the total electricity use for a household. Taking the 10-20% of electricity use in communal areas into account, air conditioning would account for 31.3% (85% of the 36.8%) of the total electricity use in residential buildings. Both studies suggested that air conditioning accounted for about one third of the total electricity use in residential buildings in recent years. Consequently, the electricity use for air conditioning in residential buildings accounted for about 8.3% (25% of the 33.3%) of the total electricity use in Hong Kong.

2.2.3 Research related to residential energy use in Hong Kong

Research work on reducing residential energy use in Hong Kong has been reported. Bojic et al. [2002] described the influence of envelope construction on space cooling loads in residential buildings. Lam [1998] performed regression and correlation analysis to investigate the relationships between residential electricity use and various economic and climatic factors in Hong Kong. Leung [2002] focused on the trade-off between air temperature and air-movement for reducing air-conditioning energy use in residential buildings in Hong Kong. Hui [2001] discussed the strategies for low energy building design in densely populated areas, and evaluated the major factors affecting low energy building design. These studies were all related to the energy use in air conditioning systems in residential buildings, however, their emphasis were not on developing strategies for air conditioning in bedroom for maintaining an appropriate sleeping thermal environment, thus contributing to reducing the total energy use in residential buildings.

2.3 Thermal comfort

In developing an appropriate strategy for air conditioning in bedrooms, one fundamental issue is how to judge the thermal comfort in a sleeping environment. This section reviews the previous related work on thermal comfort, most of which are however related to thermal comfort in workplaces in the daytime.

2.3.1 Definition of thermal comfort

American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) defines thermal comfort as “the condition of the mind in which satisfaction is expressed with the thermal environment” [ASHRAE 2001, ANSI/ASHRAE 2004]. It is therefore necessary to know the “mind” of occupants to measure comfort.

2.3.2 Overview of thermal comfort in workplaces

Throughout the last few decades, researchers have been exploring the thermal, physiological and psychological responses of people in their environments in order to develop mathematical models to predict these responses. In 1962 Macpherson defined the following six factors affecting thermal sensation: air temperature, air speed, humidity, mean radiant temperature, metabolic rate, and clothing levels [Goldman 1999, Berglund 1978]. Using the results from experiments with 1296 human subjects in a controlled climate chamber and a steady-state heat transfer model, Fanger developed an empirical comfort equation that incorporated the six factors [Fanger 1970]. It was suggested that the equation could be applicable to across geographical and cultural boundaries, age and sex groups. Fanger related the comfort equation to the “Predicted Mean Vote” (PMV) index, which predicted the mean response of a large group of people according to the ASHRAE thermal sensation scale. There are seven scales in the PMV index: cold (-3), cool (-2), slightly cold (-1), slightly warm (+1), warm (+2), hot (+3) and thermal neutral, 0, which is considered to be the optimum. The PMV was then incorporated into the “Predicted Percentage of Dissatisfied” (PPD) index, which predicted the percentage of the people who felt more than slightly warm or slightly cold (i.e., the percentage of the people who inclined to complain about the environment). Even with PMV=0, indicating that no cooling or heating is needed, about 5% of the people are dissatisfied (PPD equals to 5%). Fanger’s comfort equation described the heat exchange between a human body and its environment, and the relationship between people’s thermal sensation (a psychological response) and the physiological thermal strain on the body due to environmental and personal conditions.

Standards for maintaining comfortable indoor thermal environments have been developed by ASHRAE [ANSI/ASHRAE 2004] and the International Standards Organization (ISO) [ISO 1994]. ASHRAE defined three classes of acceptable thermal environment for general comfort (see Table 2.2) [ANSI/ASHRAE 2004]. Fig. 2.4 shows the comfort zone based on 90% acceptance of thermal conditions or 10% PPD (Comfort Class B), which may be applied to a space where occupants have activity levels with metabolic rates between 1.0 met and 1.3 met and where clothing is worn providing between 0.5 clo and 1.0 clo of thermal insulation.

Table 2.2 Three classes of acceptable thermal environment for general comfort defined by ASHRAE

Comfort Class	PPD (%)	PMV Range
A	< 6	-0.2 < PMV < +0.2
B	< 10	-0.5 < PMV < +0.5
C	< 15	-0.7 < PMV < +0.7

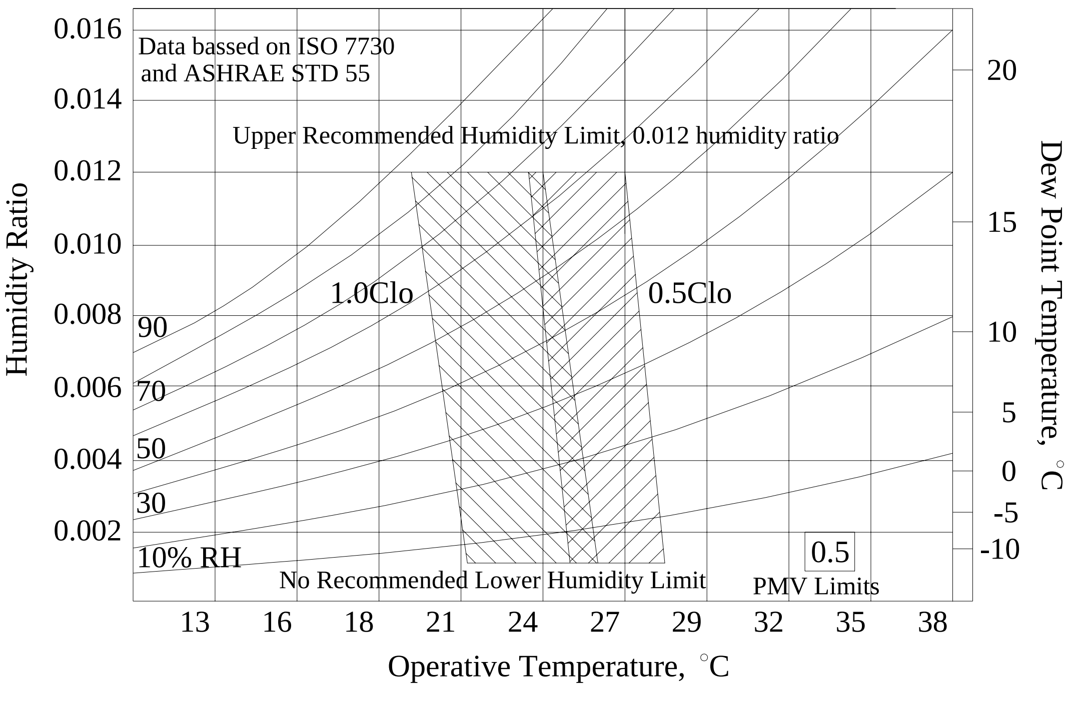

Fig. 2.4 Acceptable ranges of operative temperature and humidity for spaces corresponding to a Class B thermal environment [ANSI/ASHRAE 2004]

Fanger's PMV-PPD model on thermal comfort has been a path breaking contribution to the theory of thermal comfort and to the evaluation of indoor thermal environments in buildings. The model has set the correspondence between the characteristics of individuals, their environments and thermal sensations. However, there have been debates about "standard comfort conditions", because an in-situ building environment is not static, human heat transfer is not steady state, and human comfort perception is highly varied [Holz et al. 1997, Modera 1993]. Clothing is assumed essentially homogeneous in a space, however male workers in summer often have clothing ensembles in the range of 1.0 to 1.2 clo, and females maybe around 0.5 clo [Holz et al. 1997]. Comfort preferences of people in different locations vary in terms of acclimatization to a particular climate; various studies [Schiller 1990, Nicol and Roaf. 1996, Sekhar 1995, Hayter 1996, Chan et al. 1998] have all shown that acclimatization was evident for people in different climates. For example, the neutral temperature and preferred temperature in subtropical Hong Kong were found to be lower than those in other studies in the tropics [Chan et al. 1998]. This holds true not only for different cultures, but within the same culture [Mallick 1996]. Field studies showed that PMV on thermal sensation differed from that in climate chambers. For example, a field study indicated that the difference in neutral temperature reported by field studies and predicted by using a model could be up to 4°C [Schiller 1990]. These differences themselves were not the main object of interest, but they indicated the difficulties at least with the current practice, and possibly with theory [Oseland and Humphreys 1993, Lotens 1988].

Thermal comfort is affected by the six basic variables and an attempt in thermal comfort research over the past years was to find a single parameter that can describe the effects of a complex thermal environment. Many of such a single parameter have been proposed over the years, such as resultant temperature, effective temperature, new effective temperature, PMV, PMV*, P4SR, etc. [McINTYRE 1980]. Each of these indices was developed as an answer to a particular practical situation and was perhaps inappropriate to others. Although there have been efforts in producing universal indices to be applicable to all situations, their resultant complexity limited their usefulness.

2.3.3 Thermal comfort models in workplaces

There are two types of thermal comfort model for workplace. One is heat-balance model of thermal comfort, and the other adaptive model of thermal comfort.

2.3.3.1 Heat balance models

Three basic approaches based on the principle of heat balance have been used to model thermal comfort: physical, physiological, and psychological. ASHRAR's latest research programs attempted to extend its thermal comfort concepts to incorporate effects of noise, odor, and other indoor environment quality (IEQ) parameters, and to link thermal comfort to changes in such factors as productivity.

The example of a physical comfort model is Fanger's PMV-PPD model. The model considered a human body as a heat engine that produced heat internally and excluded the heat at the same rate to maintain its thermal equilibrium. Taffe [1997] proposed a statistical method to improve the PMV-PPD model, while Gagge et al. [1986] developed an improved PMV* index by using ET* (effective temperature) to replace t_o (operative temperature) in Fanger's comfort equation.

A two-node model, the most common physiological model for thermal comfort developed by Gagge et al. [1986] at the J.B. Pierce Foundation Laboratory, Yale University (also known as the Pierce two-node model), was based on the heat balance equation developed by Stolwijk and Hardy [1977], Gagge and Nishi [1977]. Such a comfort model used a two compartment body structure dividing a body into two concentric cylinders – the inner cylinder for the body core whose temperature, t_{re}, is 37°C, and the outer one for the skin layer whose temperature, t_{sk}, is 33°C. The skin wettedness, a rationally derived physiological index defined as the ratio of the actual sweating rate to the maximum rate of sweating that would occur if the skin were completely wet, and skin temperature were incorporated into such a model to indicate the sensation of "comfort and discomfort" caused by perspiration [Gagge et al. 1986].

With the development of Computational Fluid Dynamics (CFD) technique, the use of CFD in thermal comfort research is becoming more and more popular. A combined numerical simulation system was developed for predicting airflow, moisture transport, and thermal radiation for heat release from a human body based on a human thermo-physiological model [Murakami et al. 2000]. Based on Gagge's two compartment body model, Kang et al. developed a 61-node multi-segment human thermoregulation model to evaluate thermal comfort in non-air-conditioned building environments and to predict the body responses to indoor thermal environments [Kang et al. 2001]. Tanabe et al. [2002] proposed a 65-node thermoregulation model combined with a radiation exchange model.

The same six factors used in both physical and physiological thermal models also appear in psychological models. However, heat storage is allowed and the effects of normal physiological temperature regulatory responses (vasoconstriction and vasodilation) to the changes of the four environmental factors (air temperature, air speed, humidity, and mean radiant temperature) are also included in a physiological thermal comfort model. Therefore a physiological model offers greater possibilities to include a full range of working, clothing, and environmental conditions in evaluating thermal comfort.

It is unfortunate that these approaches seem to remain fairly independent of one another. One possible reason may be that thermal comfort is influenced by other factors such as sex, age and culture in addition to the six basic variables. Therefore, thermal comfort is of multiple-discipline nature. It is noted that physical, physiological and psychological models were favored by engineers, physiologists and psychologists respectively. There have been attempts to find ways to combine the three types of thermal comfort models over the last few years [Abdulvahap 1999].

2.3.3.2 Adaptive models

In an adaptive thermal comfort model, the adaptive approach to modeling thermal comfort acknowledges that thermal perception in “real world” setting is also influenced by the complexities of thermal history, culture, technical practices and even the outdoor temperature in addition to the six basic factors. A large number of field surveys showed that in many real situations people would be more tolerant of temperature change than what laboratory studies would suggest. Extensive literatures [Nicol and Roaf 1996, Sekhar 1995, Hayter 1996, Mallick 1996, Hanna 1997, Brager and de Dear 1998, de Dear and Brager 1998, Humphreys and Nicol 1998, McCartney and Nicol 2002, de Dear and Brager 2002, Nicol and Kessler 1998, Nicol and Humphreys 2002 and Nicol 2004] on the topic of thermal adaptation in built environments have been published for the last 30 years. An important premise for an adaptive model is that a person is no longer a passive recipient of a given environment, but instead is an active agent interacting with the person-environment system via multiple feedback loops. Thermal adaptation can be attributed to three different processes - behavioral adjustment, physiological acclimatization and psychological habituation (or expectation). Both climate chamber and field evidences indicated that the behavioral adjustment and expectation have a much great influence on thermal neutral temperature. de Dear and Brager [1998] developed an adaptive regression model that related the indoor neutral temperature to the monthly average outdoor temperature for non-air-conditioned buildings.

A major controversy in the science of thermal comfort is between the heat balance approach and the adaptive approach. However, there were some attempts to combine the best of the two approaches. Fanger and Toftum [2002] pointed out the reasons why a field survey of thermal comfort may be much different from what the PMV model would predict in certain cases such as in buildings without air conditioning in warm climates: low expectations and an estimated high metabolic rate under warm conditions. In order to solve this problem, an expectancy factor was introduced as an extension to the PMV model for use in non-air-conditioned buildings in warm climates, which may be regarded as a combination of a PMV model and an adaptive model. This demonstrated that each type of model would have its own application limit. When one particular type of thermal comfort model is adopted for different applications, some modifications may become necessary.

The extensive previous work in establishing thermal comfort models in workplaces is expected to be useful to developing air conditioning strategies in sleeping environments, as fundamentals such as the concept of thermal comfort and the heat transfer between a human body and its environment remain valid for non-workplace applications.

2.3.4 Research on thermal comfort in sleeping environments

The existing standards that prescribe "ideal" conditions for thermal comfort in workplaces are based on a heat balance model for a human body. They are derived from extensive experiments in climate chambers, conducted primarily with university students in mild-climate regions. Although these standards were initially developed for buildings with centralized heating, ventilation & air conditioning (HVAC) systems, it has been often suggested that they are universally applicable to across all building types, climates, and populations [Taffe 1997]. The question that remains to be asked is then "can the existing thermal comfort standards be applied to sleeping environments in the subtropical regions?"

Strictly speaking, the phase “thermal comfort” does not make too much sense for people during sleep. Based on the definition of thermal comfort, comfort is not a state condition, but rather a state of mind. The questions that need to be addressed include whether a human being can have any “mind” during sleep or not, and whether a human being can express his / her mind during sleep or not.

To understand the relationship between a sleeping human being and his / her sleeping thermal environment, it is necessary to review the objective methods to evaluate the sleeping quality currently used in medical or other related subject areas.

2.3.4.1 Sleep

A human being spends approximately one-third of his / her life in sleep. Sleep is indisputably a basic need, thought its duration varies, often considerably, from person to person. It is, in fact, possible to go without food longer than without sleep [Lavie 1996].

More has been learned about sleep for the past 70 years than in the preceding 6000 years. In this short period of 70 years, researchers have discovered that sleep is a dynamic behavior. Not simply the absence of waking, sleep is a special activity of the brain, controlled by elaborate and precise mechanisms. Sleep is not simply a state of rest, but has its own specific, positive functions [Hobson 1989]. For example, sleep can help people overcome tiredness, and sleep is very important to one’s memory. In recent years, sleep is becoming an increasingly issue of concern. A survey conducted by the Shanghai Traditional Chinese Medicine Hospital showed that more than 15 percent of Shanghai residents were troubled by insomnia [Website_1 2002]. Another survey suggested that nearly half the population of Singapore suffered from insomnia, or at least they had sleep disorder [Website_2 2002]. Although insomnia may be caused by some other factors, a sleeping thermal environment must have a part to play in the quality of sleep. To help people know more about the importance of sleep and the problem of insomnia, the International Foundation for Mental Health and Neuro-Science initiated a World-Wide Sleep and Health Project - International Sleep Day, which is set on March 21 every year.

Modern sleep research is conducted principally in a sleep laboratory, where polysomnographic recordings (including electoencephalogram-EEG, electrooculogram-EOG, and electromyogram-EMG) depicting the course of a sleep are made. These three data sources provide reliable information on the process of falling asleep and the changes that occur during sleep itself: brain waves, eyeball movements, and muscle tonus. Other data such as respiratory movements, air flow through the nose and mouth, blood oxygenation level, heart rate; skin temperature and rectal temperature can also be recorded simultaneously.

The appearance of sleep stages during a night is not a random process but an organized and definitive one. The sleep sequence can best be described with the aid of a hypnogram of sleep recording [Lavie 1996, Brebbia and Altshuler 1965]. Fig. 2.5 shows the hypnogram of a young adult.

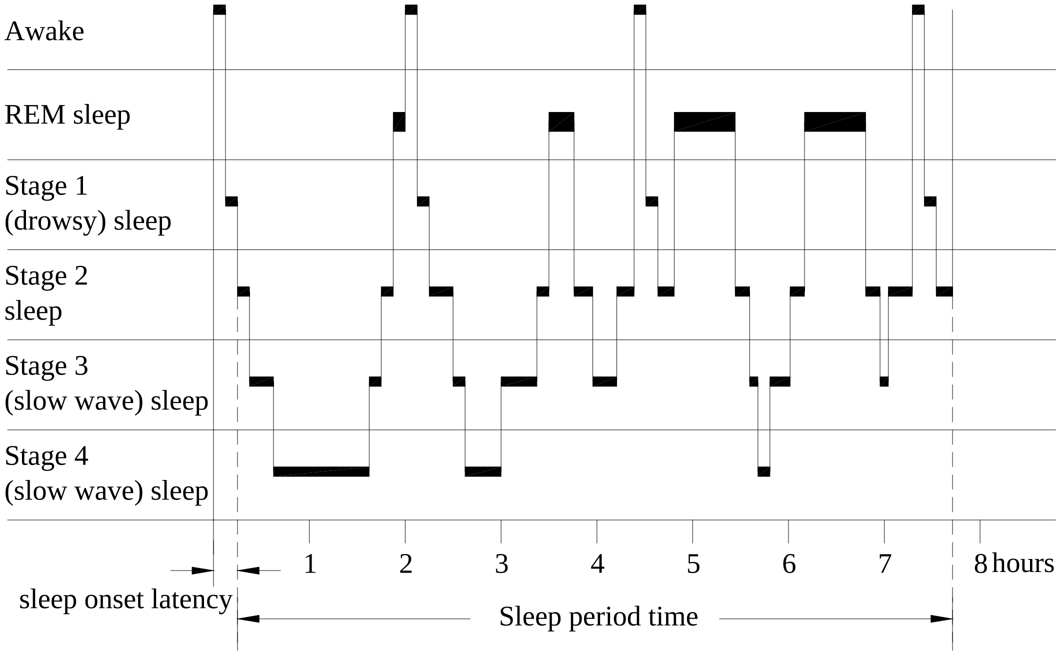

Fig. 2.5 The hypnogram of a young adult [Lavie 1996]

When human subjects pass from arousal to relaxed waking, to drowsiness, to light sleep, and finally to deep sleep, their brain waves change in a characteristic way. The characteristic patterns seen from light sleep to deep sleep have become known as the four stages of sleep; the distinguishing features of the waves in each stage are given in Table 2.3.

Table 2.3 Characteristics of EEG sleep stages

Stage	Frequency (1/s)	Amplitude (μV)	Wave form
1	4-8	50-100	Theta waves
2	8-15	50-150	Spindle waves
3	2-4	100-150	Slow waves plus spindles
4	0.5-2	100-200	Delta waves

Although the progression of EEG waves has been divided into discrete stages, it is actually gradual and continuous. Once Stage 4 is reached, the process reverses itself, and a sleeper goes back through Stage 3, 2 to 1. This clear cyclical process tends to repeat itself, from Stage 1 to Stage 1, at 90- to 100-minute intervals throughout a night [Hobson 1989]. At the end of each sleep cycle, there is a special period, called rapid eye movement (REM) sleep, which can be regarded as the symbol of the start of next sleep cycle or the end of the last cycle. The number of sleep cycles per night depends on the duration of sleep; a younger person's sleep usually has four or five cycles, while an older one has fewer.

2.3.4.2 Relevant research on thermal comfort in sleeping thermal environments

The last 30 years saw extensive research work on thermal comfort, covering many aspects related to thermal comfort. These include establishing models [Fanger 1970, de Dear and Brager 1998], and indices [Gagge et al. 1986], carrying out experiments in climate chambers [Fanger 1970 and Nakano et al. 2002], and field surveys [Schiller 1990, Newsham 1997], establishing thermal comfort standards and evaluation methods [Olesen and Parsons 2002, de Dear and Brager 2002, Doherty and Arens 1988, Evin and Siekierski 2002 and Hoppe 2002], etc. However, almost all related studies were concerned with the situations where people were awake.

Nonetheless, there have been a few scattered research studies on the relationship between a sleeping thermal environment and the quality of sleep. Miyazawa [1994] demonstrated that the range of thermal neutral temperature was about 22 ± 3°C by using a questionnaire survey and measuring the air temperature and humidity in a bedroom as well as the esophageal temperature of five junior college students for a period of 214 days. Miyazawa further showed that when the room air temperature ranged from 11°C to 29°C, the quality of sleep was not influenced remarkably [Miyazawa 1994]. However, the bedroom was naturally ventilated rather than air conditioned. A study designed to examine the effect of sleeping thermal environment on sleep patterns by using EEGs with six men and six women sleeping at night in an experimental chamber where air temperatures were maintained at 10°C, 21.1°C, and 32.2°C respectively was carried out [Rohles 1983]. The results also showed that the air temperature in a sleeping thermal environment did not affect the amount of time spent in various sleep stages. However, each subject slept only one night under the three temperatures respectively and only the air temperature was considered in the experimental study. It is interesting to note that both studies showed that under a rather wide range of air temperature (about 11°C ~30°C), air temperature would not have much effect on the quality of sleep. However Haskell et al. [1981] reported that there were remarkable differences of individual's sensitivity to air temperature during sleep. It was found that the sleep patterns of two subjects were similar when air temperature was maintained at 29°C. However, the patterns differed significantly from each other when air temperature was maintained at 21°C. Although with only two subjects, this study indicated the observations that did not agree well with the results from the previous studies, implying that the individual's sensitivity to sleeping thermal environment may have to be taken into account. Research work on the infant sleeping thermal environment with its main focus on sudden infant death syndrome (SIDS) risk factors has been reported [Mckenna et al. 1993, Nelson and Taylor 2001, Telliez 1997, Schluter et al. 2000]. However the sleeping thermal environment in these studies was a closed incubator, not a normal air-conditioned bedroom, and the research subjects were restricted to infants. In most of these previous studies, the number of human subject samples used was limited to single digits, in contrast to the 1,296 subjects used by Fanger. This implies that using a statistically meaningful number of human subjects to systematically study thermal comfort in sleeping environments is hardly possible.

On sleep research, studies on the influence of air temperature to thermoregulation, metabolism, and the stages of sleep showed that during various sleep stages excluding REM period, a human's physiological response to room temperature changes was similar to that when a human being was awake [Lavie 1996, Candas et al. 1982]. Different researchers carrying out experimental studies on the effects of high and low ambient temperatures on human sleep stages adopted different thermoneutral temperature in sleep in the range of 20 to 32°C [Macpherson 1973, Karacan et al. 1978, Haskell et al. 1981, Candas et al. 1982, Sewitch et al. 1986, Palca 1986, Di Nisi et al. 1989 and Dewasmes et al. 2000]. Table 2.4 lists the details of the different thermoneutral temperature selected in various studies related to sleep. It can be seen that there is a fairly great range of the thermoneutral temperature, indicating that a well-agreed single thermoneutral temperature has not been established. In most of the research studies related to sleep, only ambient air temperature was referred to, but the mean radiant temperature or operative temperature and air velocity were not taken into account. In cases where 20~23°C were selected, the test subjects were covered with bedding, while in others, the subjects may be in naked situation. This indicated that the relationship between the thermoneutral temperature and the insulation of bedding has not yet been established.

Although the thermoneutral temperatures determined by researchers were different because of different experimental conditions (i.e. covered or naked), it can be seen clearly that the determined thermoneutral temperatures (20~22.2°C at covered condition; 28~32°C at naked condition) in sleeping thermal environments are different from the air temperature (24~26°C) normally maintained in workplaces in summer.

Table 2.4 Different thermoneutral temperatures adopted in studies related to sleep

Literature	Thermoneutral temperature (°C)	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 Hong Kong, field studies [Leung 2002] showed that indoor air temperature would be kept in the range of 21 to 24°C during air-conditioning periods in residential buildings. This seems slightly lower for sleeping. Such a low air temperature for a sleeping environment has two disadvantages. The first is obviously more energy consumption for air conditioning, and the second is that a low air temperature means cold stress to human, and cold stress contributes more negative effects to the sleep quality than heat stress [Haskell et al. 1981]. In order to avoid catching cold, people may incline to be covered with a so-called “air-conditioning quilt”. This leads to a question: why should a relative low bedroom air temperature be maintained, while people are covered with a rather thick quilt? The question can only be answered after establishing a relationship between the thermoneutral temperature in a sleeping environment and the factors that will affect the thermoneutral temperature in the region where air conditioning is extensively required during nighttime.

EEGs are useful for evaluating the quality of sleep. However, EEGs are expensive to perform and may therefore only be performed to a limited number of test subjects. In addition to objective experimental studies in sleep laboratories, subjective questionnaire survey is an alternative to measure the quality of sleep. At present, the most commonly used methods measuring subjective mood and feelings after sleep involve the use of adjective check lists (ACLs) or visual analogue rating scales (VASs). One example using ACLs was the St. Mary's Hospital in London sleep questionnaire [Ellis et al. 1981], which was proved to be reliable and acceptable even for patients with psychological disturbance. Fourteen questions were included in the questionnaire with the choices of answer to most of the questions given in a scale of several adjectives. For example, the answer choices to the question "was your sleep" are 1,-very slight; 2,-light; 3,-fairly slight; 4,-light average; 5,-deep average; 6,-deep; 7,-very deep. For VASs, some visual analogue rating scales are given, and a subject is asked to make a mark across a horizontal line at a position, which indicates how he / she feels at the time in relation to two words, such as "alert" and "drowsy" [Herbert et al. 1976]. Factor Analysis of the St. Mary's Hospital sleep questionnaire showed that two factors relating to "sleep latency" and "sleep quality" emerged more clearly than other factors produced, suggesting that the two most important aspects of subjectively perceived sleep were the process of going to sleep and the quality of sleep [Leigh et al. 1988]. Factor Analysis of VASs sleep evaluation questionnaire also showed that different factors would affect different aspects of sleep, such as ease of getting to sleep, the perceived quality of sleep, and ease of awakening from sleep [Herbert et al. 1976, Parrott and Hindmarch 1978].

2.3.4.3 Other issues related to thermal comfort in sleeping environments

A large number of previous studies on the thermal insulation of clothing have been reported [McCullough and Wyon 1983, Olesen et al. 1988, McCullough et al. 1989, Lotens and Havenith 1991, Newsham 1997a, McCullough et al. 1994, Gagge et al. 1941, McCullough 1994]. Based on these studies, a complete full-scale database of the thermal insulation and evaporative resistance of clothing ensembles necessary for the use in various thermal comfort models for daytime was developed, and adopted by relevant standards [ISO 1995, ANSI/ASHRAE 2004]. There were also a limited number of previous studies on the thermal insulation provided by bedding systems, which were carried out using a manikin [McCullough et al. 1987, Madsen and Drengsgard 1981, Mecheels and Umbach 1982]. These studies resulted in a small-scale database of thermal insulation values for a variety of bedding systems. However, most of the reported studies were related to heated environments where the objective was to increase thermal insulation value by a bedding system so as to lower indoor air temperature. Furthermore, the types of beddings studied were in generally based on those primarily used in the USA and Europe, and therefore not applicable to air conditioned sleeping environments in the subtropics.

As most residential buildings adopt discrete air conditioning systems, i.e., RACs, their noise level is an area of concern. As a RAC is normally on / off controlled, the frequent start and shutdown produce noticeable noise in residential buildings. Studies [Griefahn and Gros 1986, Kawada et al. 1997] on the influence of noise to sleep showed that noise slightly influenced the course of sleep in comparison with other variables (age, sex, personality). Small negative effects were found in some physiological reactions from the test subjects, and there was no (complete) habituation to usual acoustical environments.

2.4 Other issues related to nighttime air conditioning

There are two other relevant issues related to maintaining an appropriate thermal environment for sleep in bedrooms in the subtropics. One is bedroom nighttime cooling load calculation, the other fresh air (outdoor air) supply to a bedroom.

2.4.1 Nighttime cooling load calculation

The characteristics of nighttime cooling load in bedrooms in residential buildings are much different from that in commercial buildings in daytime. There will be virtually little solar heat gain at nighttime, little heat gain from lighting and domestic appliances. The heat gain from occupants (including sensible and latent heat gain) during sleep is expected to be lower than that during being awake. The cooling load in a bedroom at nighttime may basically consist of building envelope load due to the temperature difference between indoor and outdoor air; the load due to removing the heat stored in the space's thermal mass, i.e. building envelop and furniture; the load due to heat gain from occupants and finally, the load caused by infiltration and ventilation. Therefore cooling load profiles, and the ratio between the sensible load (or latent load) and the total cooling load at nighttime are obviously different from those at daytime. It is therefore necessary to investigate the characteristic of cooling load in bedrooms during sleeping time, so that relevant strategies for bedroom air conditioning may be developed.

Depending on applications, the cooling load of a building may be expressed in different formats, although their calculation methods can be the same. For sizing HVAC systems in buildings, using so-called "peak load" or "design load", which represents the maximum cooling load under a specified "design condition", is often more than adequate. However, when performing energy analysis of a building, it is more preferred to use the cooling load that the building imposes on building's HVAC systems, where the cooling loads are profiled in a fixed time interval, typically an hour, to form the so-called hourly cooling load. Compared to "peak load", the concept of hourly cooling load is probably more appropriate in studying the strategies for bedroom air conditioning.

Three cooling load calculation methods were adopted by ASHRAE for nonresidential buildings in 1992 [McQuiston and Spitler 1992]: the Transfer Function Method (TFM), the CLTD/SCL/TA method and the TETD/TA method. However, based on the ASHRAE Research Project 875, two new methods for calculating cooling loads in nonresidential buildings were developed in 1998

[Pederson et al. 1998, Spilter and Fisher 1999, Spitler 1997]. One was heat balance method (HB), considered to be the most reliable method for estimating cooling loads for a defined building space; the other radiant time series method (RTS) which was a simplified method directly related to and derived from the HB calculation procedure. These two new methods have been adopted in the new version of ASHRAE Handbook of Fundamentals (2001) to replace the outdated TFM, CLTD/SCL/TA, and TETD/TA methods.

The United Kingdom's Chartered Institution of Building Services Engineers (CIBSE) recommends the use of a different cooling load calculation method, the admittance method. This simplified method is for calculating the effects of fluctuating thermal load and temperature variations. It is interesting to note that cooling load calculation methods adopted in both North America and the UK were compared in a few studies [CIBSE 1998, Rees et al. 2000]. The conclusions were that the approach used in the heat balance method was the least abstracted from physical zone heat transfer processes. The HB method modelled the heat balances explicitly for an exterior surface, an interior surface and zone air, and was therefore the most fundamental and general method. However this method required being computer based.

For residential building cooling load calculation method, the procedures available in the ASHRAE Handbook of Fundamentals and the Air Conditioning Contractors of America (ACCA) "Residential load calculation" manual [ACCA 1986] are based on the results of ASHRAE Research Project 342 [McQuiston 1984, ASHRAE 2001]. However, the method is a simplified CLTD/SCL/TA method [ASHRAE 2001, McQuiston 1984, Xu 1997, Spitler 1996] for approximately calculating cooling load in residential buildings. On the other hand, the residential buildings in North America are normally single-family detached houses or multifamily apartments, more often a house rather than an apartment and a central (within a house) air conditioning system is normally adopted. This is very different from the situation of residential buildings in Hong Kong. Unlike in North America, discrete air conditioners are common and are operated only during occupied periods in residential buildings in Hong Kong. Each bedroom has its own air conditioner, so that each bedroom is an independent zone. Therefore the cooling load of each bedroom needs to be calculated separately. Moreover, ASHRAE suggests that the procedures to calculate heating or cooling load in nonresidential buildings may also be used in residential buildings [ASHRAE 2001].

2.4.2 Fresh air supply

Fresh (outdoor) air, which is closely related to indoor air quality (IAQ), is important for the health of building occupants. IAQ guidelines for offices and public places in Hong Kong have been established recently [Hong Kong Environment Protection Department 2000]. In the guidelines, twelve parameters including air temperature, relative humidity, etc., are included, and three levels of satisfaction specified. Level 1 represents an excellent IAQ condition; Level 2 provides protection to general public while Level 3 is the lowest for occupational safety that all buildings must achieve. Although these guidelines are for office buildings and public places, research work on IAQ in residential buildings is being conducted and the guidelines applicable to residential buildings are expected to be developed in near future. The amount of fresh air supply to an air conditioned space would certainly affect the aforesaid 12 parameters. Increasing the fresh air supply is an effective way to improve IAQ, however, at the expense of consuming more energy in most cases.

Studies on the different outdoor air ventilation strategies and the ventilation characteristics of air-conditioned buildings were reported [Reddy et al. 1998, Sekhar et al. 2002]. However, these were based on office buildings. Little work on fresh air supply in air conditioned residential buildings (particularly in bedrooms) has been reported.

The level of indoor carbon dioxide (CO_2) could be used as an indicator to assess whether an adequate ventilation rate has been supplied to dilute or remove harmful indoor pollutants. It was reported that a CO_2 concentration of more than 1000 ppm was identified in Taiwanese bedrooms during sleeping hours in wintertime [Chao et al. 1998]. Another study indicated that when a bedroom was occupied, with its door and windows closed, the CO_2 level in the bedroom increased steadily over night. The CO_2 concentration level was in excess of 3500 ppm with one person, and in excess of 4500 ppm with two persons, respectively, in heating seasons in Canada [Parent et al. 1998]. This was 2.5 and 3.5 times, respectively, higher than the recommended health guideline for CO_2 level (1000 ppm) by the World Health Organization (WHO) [WHO 1984]. In the meantime, this also exceeded the Canadian exposure guideline for CO_2 level (<3500ppm) for residential indoor air quality. However, the two studies were both for heated environments in winter seasons. A study in Hong Kong showed that the nitrogen oxides (NO_2, NO) levels in residential buildings needed some concerns, and that the sulphur dioxide (SO_2) and ozone (O_3) levels were low in indoor residential environment in Hong Kong by comparing both the indoor and outdoor levels of various air contaminants in 10 non-smoking residential apartments [Chao 2001]. However, the study focused on IAQ during daytime in living rooms, rather than at nighttime in bedrooms. Furthermore, CO_2 concentration was not taken into consideration in the study.

ASHRAE Standard 62-2001 (Ventilation for Acceptable Indoor Air Quality) [ASHRAE 2001a] recommends 15 L/s/room outdoor air flow rate for bedrooms in hotels, motels and resorts, irrespective of the size of a guest room. Considering that most of the bedrooms in hotels and motels are "standard" rooms, i.e. with two single beds in a bedroom, it can be inferred that the outdoor air requirement recommended by ASHRAE is 7.5 L/s per person. In residential buildings, the recommended outdoor air requirement for living areas (including bedroom obviously) is 0.35 air changes per hour but not less than 7.5 L/s per person. It is noted that 8 L/s per person is the minimum outdoor air requirement in ASHRAE Standard 62-2001, which is used in those applications where the density of occupants is rather high and people will not stay for a long time such as the spectator areas in a sport or amusement building, the auditorium in a theater, etc.

2.5 Conclusions

With the rapid economic growth and the continuous increase of people's living standard, the last 20 years saw a remarkable increase in both the penetration rate and the saturation rate of air conditioners in residential buildings in Hong Kong. The energy use for residential air conditioning also correspondingly increased. It can be concluded that in Hong Kong air conditioning is responsible for about one third of the total electricity use in residential buildings, or approximately 8.3% of the total electricity use in Hong Kong. Part of electricity use for residential air conditioning is for maintaining a suitable sleeping thermal environment in bedrooms. In order to reduce energy use while maintaining appropriate sleeping thermal environments, it becomes necessary to develop optimized bedroom air conditioning strategies in Hong Kong.

Extensive research work on thermal comfort was undertaken in the last century, and a number of thermal comfort models and indices have been developed. However, it was revealed that most of the previous work focused on the conditions where people were awake in workplaces during daytime. Nonetheless, there have been a few scattered studies on the relationship between the quality of sleep and a sleeping thermal environment. These studies suggested that the use of bedding could strongly affect thermoneutral temperature. It may be said that in order to reduce the energy use for bedroom air conditioning while maintaining the quality of sleep, the indoor air temperature in a bedroom should not be too low. An indoor air temperature in a bedroom during air conditioning season is expected to be close to 26~27°C under minimal bedding and sleepwear situation.

Considering that bedroom air conditioning is provided most likely during nighttime when solar heat gain does not virtually exist, the nighttime cooling load characteristics and the ratio between the sensible (or latent) load and the total cooling load are expected to be different from those for daytime air conditioning. Therefore the current methods for building cooling load calculation should be reviewed. There are several building cooling load calculation methods such as those adopted by ASHRAE and CIBSE, and many computer software packages for building cooling load calculation are available. Although these methods are predominately developed for non-residential applications, they are reportedly applicable to residential buildings for determining cooling load at nighttime, possibly with some modifications.

As residential air conditioning in Hong Kong is often by discrete systems, i.e. window or split type air conditioners, its organization of fresh air supply will clearly differ from that for centralized air conditioning systems in commercial buildings. However, except for a few isolated case studies on CO_2 levels in heated bedrooms in residences during heating seasons, no previous work on appropriately organizing fresh air supply in air conditioned bedrooms during cooling season can be identified.

The amount of fresh air supplied to a bedroom would affect a sleeping thermal environment and the energy use for bedroom air conditioning. However, current design guidelines and / or standards for fresh air supply are basically for workplace applications. Although 15 L/s per room for lodging buildings and 7.5 L/s per person for residential buildings are recommended by ASHRAE, it is suspicious that this is simply copied from the amount of fresh air supply recommended in workplaces. Therefore a question to be asked is whether this 7.5 L/s per person is justifiable for bedroom air conditioning application. Furthermore, the actual amount of fresh air supplied to a bedroom equipped with a RAC remains problematic. For example, different types of RACs are used in bedrooms in residential buildings in Hong Kong.

Some are of compact type (i.e. window-type) where fresh air can be induced directly to the rooms, while others are of split-type. For a split-type air conditioner, no fresh air can be supplied by the air conditioner itself, outdoor air entering a room is due to infiltration or other independent means of ventilation.

The extensive literature review of air conditioning strategies in bedroom reported in this chapter covers the historic development and the current status of extensive research work on thermal comfort in built environments. Related issues on air conditioning in bedrooms, such as nighttime cooling load calculation and fresh air supply are also reviewed. A number of important areas where further in-depth research work is required on maintaining an appropriate sleeping thermal environment in bedrooms in residential buildings have been identified. These are the expected targets of investigation for the project reported in this thesis.

In order to differentiate the concept of the "thermal comfort" in sleeping environments from that in workplaces during daytime, the "thermal comfort" in sleeping environments in this thesis is defined as "acceptable sleeping quality and thermal neutrality are achieved during sleep and satisfaction with the sleeping thermal environment is expressed after sleep". Furthermore, it is necessary to point out that a sleeping environment involves many different aspects such as thermal, acoustic, odorous and visual environment. This thesis mainly focuses on sleeping thermal environment.

Chapter 3

Proposition

3.1 Background

It is evident from the literature review presented in Chapter 2 that the current practices in air conditioning, as well as the thermal comfort theories on which these practices are based, are primarily concerned with situations in which people are awake in workplaces at daytime. Therefore, these may not be directly applicable to air conditioning for sleeping environments. Given that most bedrooms are occupied during nighttime, bedroom air conditioning would mainly serve to maintain an appropriate sleeping thermal environment.

One fundamental issue in developing air conditioning technology is the theory on thermal comfort, whether for a workplace at daytime or a sleeping environment at nighttime. Currently the theories on thermal comfort in workplaces at daytime are well established. However, research on thermal comfort for sleeping environments at nighttime has been limited and more work concerning the thermal comfort of people during sleep is needed.

In addition, the external conditions for bedroom air conditioning to maintain an appropriate sleeping thermal environment at nighttime are different from those at daytime. For example, there will be virtually no solar heat gain at nighttime. On the other hand, internal conditions will also be different. For example, the heat gain from occupants during sleep is expected to be lower, because people have a lower metabolic rate when asleep than awake. Furthermore, the thermal insulation provided by clothing at daytime, is replaced by that from a bedding system consisting jointly of bed, mattress, bedding, and sleepwear used by people.

Therefore, it is necessary to embark on studying the thermal comfort in sleeping environments, so that a basis for developing the design standards and strategies for bedroom air conditioning may be established. Furthermore, it is also necessary to study the related issues such as the characteristics of nighttime bedroom cooling load and the outdoor air ventilation rate required for sleeping environments.

3.2 Project title

This thesis focuses on three major issues related to developing optimized strategies for bedroom air conditioning in the subtropics: 1) Thermal comfort in sleeping environments; 2) The characteristics of nighttime bedroom cooling load; and 3) Ventilation requirements for sleeping environments. The proposed research project is therefore entitled “Strategies for bedroom air conditioning in the subtropics”.

3.3 Aims and objectives

The literature review presented in Chapter 2 revealed that there have been a large number of reported studies in literature on thermal comfort. However, almost all previous work on thermal comfort focused on the situations where people were awake in workplace during daytime. Little attention has been paid to the thermal comfort in sleeping environments. There is clearly a lack of systematic research work on establishing thermal comfort theories for air conditioned sleeping environments in the subtropics. Therefore, the first objective of this research project is to develop a thermal comfort model for application to sleeping environments in the subtropics, by reference to the existing thermal comfort models for workplaces at daytime.

Given that the external and internal conditions for nighttime air conditioning are different from those for daytime air conditioning, the characteristics of nighttime cooling load in bedrooms are expected to be different from that of daytime cooling load. Therefore, the second objective of this project is to study the characteristics of nighttime cooling load in bedrooms, so that optimized strategies for bedroom air conditioning, e.g. methods of correctly sizing RACs for bedrooms, could be developed.

An appropriate amount of ventilation air should be made available to a nighttime air conditioning process, in order to maintain a suitable sleeping thermal environment, IAQ, and minimize the energy use for bedroom air conditioning. Therefore, the third objective of this project is to establish a suitable outdoor air ventilation rate for sleeping environments in bedrooms, so that the balance between an acceptable IAQ and minimized energy use can be achieved.

3.4 Research methodologies

The current status of sleeping thermal environments and bedroom air conditioning in residential buildings in Hong Kong will be investigated using questionnaire survey and field monitoring of overnight indoor air temperature and relative humidity in a number of bedrooms.

Both analytical and experimental approaches will be applied in studying thermal comfort in sleeping environments. The theoretical analysis of thermal comfort in sleeping environments is based on the heat balances between a sleeping human body and its environment. A thermal comfort model for application to sleeping environments will be developed by introducing appropriate modifications to Fanger's PMV-PPD model so as to make it applicable to sleeping environments. An experimental study using a thermal manikin will be carried out in an environmental chamber to measure the total insulation values of various bedding systems commonly used in the subtropics. The obtained total insulation values of bedding systems will be used to solve the thermal comfort model to be developed for sleeping environments.

The characteristics of nighttime cooling load in bedrooms in the subtropics will be investigated using a building energy simulation program – EnergyPlus, based on the typical layout of high-rise residential blocks in Hong Kong.

The current situation of ventilation in residential buildings in Hong Kong will be investigated through field monitoring of both indoor overnight CO_2 levels and outdoor ventilation rates in bedrooms employing RACs. In addition, the outdoor air ventilation characteristics in bedrooms employing RACs will be studied through laboratory experiments. Tracer gas decay technique will be used to measure the outdoor ventilation rate in both field measurements and laboratory experiments.

3.5 Expected outcomes

The expected outcomes of the research project reported in this thesis include a complete thermal comfort model applicable to sleeping environments in the subtropical regions. The model may be further used as a fundamental basis in establishing necessary thermal comfort criteria for air conditioned sleeping environments. A comfort equation applicable to sleeping environments derived by introducing modifications to Fanger's comfort equation, a small-scale database of the total insulation values provided by various bedding systems commonly used in the subtropics, and the comfort charts established by solving the comfort equation, will be available as parts of the complete comfort model. The database may further be adopted by various standards such as an ISO Standard or an ASHRAE Standard. The outcomes also include guidelines for better design and operation of air conditioning systems for bedrooms such as a method of sizing RACs for bedrooms after knowing the unique characteristics of nighttime cooling load in a bedroom, and the suitable outdoor air ventilation rate for sleeping environments.

Chapter 4

Survey on Sleeping Thermal Environments and Bedroom Air Conditioning in Residential Buildings in Hong Kong

4.1 Introduction

The increased use of residential air conditioning has a significant impact on the total electrical energy use in residences. For example, air-conditioning accounted for 18% of residential electrical energy use in the U.S., and was the primary component of peak load [DOE 1990]. In subtropical Hong Kong, as shown in Section 2.2.3, air-conditioning was by far the largest single electricity end-user, accounting for 31.3% of the total residential electricity use. Given that there has been a growing concern on the increased energy use for residential air conditioning and its impacts on the environment, it is necessary to understand the current situation of bedroom air conditioning and the indoor thermal conditions in order to evaluate the future trend of residential energy consumption for bedroom air conditioning and to provide information to energy system planning and environmental policy-making.

A number of earlier studies [Seligman et al. 1978, McLain et al. 1985, Kempton et al. 1992] suggested that user behavior would significantly affect air-conditioning energy use among similar dwellings. Over the past decades, there have been studies on indoor thermal environment in residential buildings and the use patterns of RAC through questionnaire survey, interviews and field measurements. For example, Yoshino et al. [2002] reported a survey on the actual conditions of residential indoor environments in three cities (Shanghai, Hong Kong and Xian) in China to evaluate thermal comfort and the possibility of energy conservation for space cooling by using questionnaire and field measurements. However, indoor air temperatures were measured mainly using a simple mercury thermometer and recorded only three times (morning, mid-day and evening) daily. Moreover, the monitoring of overnight indoor CO_2 levels was not included in the study. Kempton et al. [1992] studied the operation of RACs to understand how energy consumption and peak power demand were influenced by the needs, perceptions, and behaviors of users through field measurements and interviews. It was found that many non-economic factors would influence the use of RACs. These included health, thermal comfort, safety, folk physiological theories, and folk theories about how RACs function. Li et al. [2002] carried out a study on residential energy consumption through administering questionnaire to 50 households in Shanghai, China and found that there was a close link between building type and residential energy consumption. Through site measurements, interviews and observations, Lutzenhiser [1992] surveyed the use of RACs in 279 California apartments and observed that most users opted for a manual, rather than automatic control strategy. It was found that the choices of control strategies were sometimes guided by the theories of RAC operation and control that were not in agreement with engineering accounts of a machine’s design. However, in these studies, the focuses were only on the control strategies and energy use of RACs, whereas the investigations on indoor thermal environments and IAQ both in individual bedrooms and at nighttime (sleeping period) were not carried out.

This chapter reports on a questionnaire survey on the sleeping thermal environments and bedroom air conditioning, and a field monitoring of overnight indoor air temperature, relative humidity and CO_2 levels in bedrooms in residential buildings in

Hong Kong. The survey focused on human factors such as sleeping habits, the use of bedding and sleepwear while sleeping, preferred indoor air temperature settings in bedrooms by occupants, ventilation control at nighttime with RACs turned on, etc. The results of filed monitoring illustrated the current status of indoor thermal environments and IAQ at nighttime in bedrooms in residential buildings in Hong Kong.

4.2 Methodologies – questionnaire survey and field monitoring

A questionnaire survey to investigate the sleeping habits of local people, user behaviors in using bedroom air conditioning, and the current situation of the use of bedroom air conditioning in residential buildings in Hong Kong was carried out. The following were investigated:

- Use patterns and types of bedroom air conditioning systems used;
- Use of bedding and sleepwear;
- Sleeping thermal environments during nighttime with RACs turned on;
- Ventilation strategies in bedrooms;
- Other issues such as the effects of the noise from operating RACs on sleeping quality.

A questionnaire with 20 questions (details shown in Appendix A) was sent out in three different ways. The first was to distribute the questionnaire to friends, relatives, and classmates of questionnaires' administrators. The second was through an on-line questionnaire survey and the last through target visiting. More than 400 questionnaires were distributed by the first way and 183 respondents replied the on-line questionnaire by email. Most of the data collected by these two ways were from the respondents with the age range of 18-50. In order to minimize bias, target visiting was conducted to obtain data from teenagers and elders. Therefore, one target visiting was carried out, through which 102 questionnaires from elders aged between 50 and 82 years old were collected. Another target visiting was undertaken during a scout training course, through which 71 responses from teenagers aged between 11 and 17 years old were collected.

In addition to the questionnaire survey, field monitoring of overnight indoor air temperature, relatively humidity and indoor CO_2 levels in bedrooms in high-rise residential buildings was also carried out. The monitoring of overnight indoor air temperature and relative humidity helped evaluate the current situation of indoor sleeping thermal environments whilst the overnight indoor CO_2 level monitoring helped appreciate the IAQ in bedrooms given that the indoor CO_2 level has been extensively used as an IAQ indicator [ASTM 1998, Persily 1997].

The questionnaire survey and filed monitoring were carried out between September 2002 and May 2003. A statistical package for social science (SPSS), one of the most popular computer software for social science, was used to analyse the data collected from the questionnaire survey.

4.3 Results

4.3.1 Results of the questionnaire survey

A total of 554 valid questionnaires were returned and the data collected from these questionnaires were analysed using SPSS. Fig. 4.1 shows the breakdown of respondents' age groups. 55.8% of the respondents were male while the other 44.2% female.

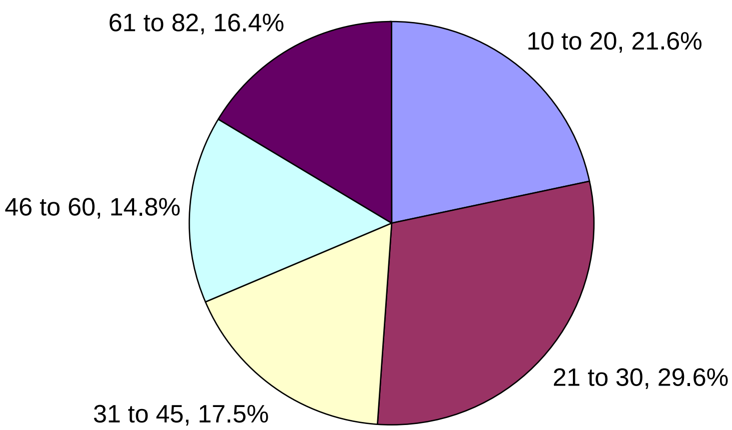

Fig. 4.1 Percentage breakdown of respondents' age group

4.3.1.1 Use patterns and types of bedroom air conditioning systems used

As shown in Fig. 4.2, 68% of the respondents would leave their RACs on throughout the duration of their sleep. The rest would however use their RACs only for certain hours during sleep. Fig. 4.3 shows the percentage breakdown of the number of months when occupants would use their RACs in bedrooms for sleeping in a year. It may be seen that 82.5% of the respondents reported using their RACs during nighttime for more than two months annually.

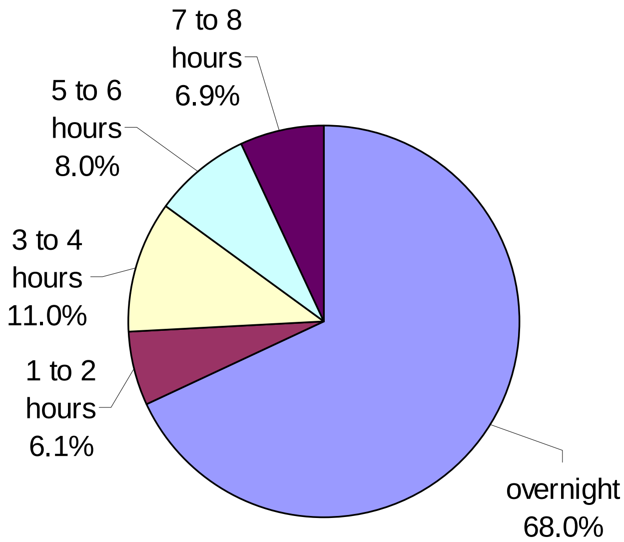

Fig. 4.2 Percentage breakdown of the number of hours with RACs turned on while sleeping at night by respondents

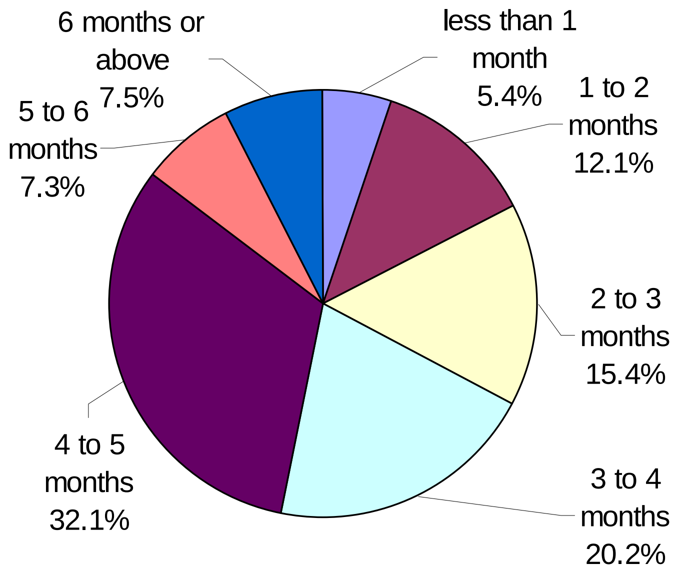

Fig. 4.3 Percentage breakdown of the number of months with RACs turned on while sleeping in a year by respondents

Furthermore, 75% of the respondents used WRACs while the other 25% used SRACs. Other air conditioning systems such as a variable refrigerant volume (VRV) system, a central residential air conditioning system, etc., were not common.

4.3.1.2 Indoor thermal environments

The survey showed that in shared bedrooms, most of the respondents would have the same opinion as their roommates on whether turning RACs on (72%) and on indoor air temperature settings (76%). Fig. 4.4 shows the percentage breakdown of the indoor air temperature settings at nighttime currently used by respondents for bedroom air conditioning in Hong Kong. It can be seen that most of the respondents (>80%) would prefer a relatively low indoor air temperature setting at below 24 °C.

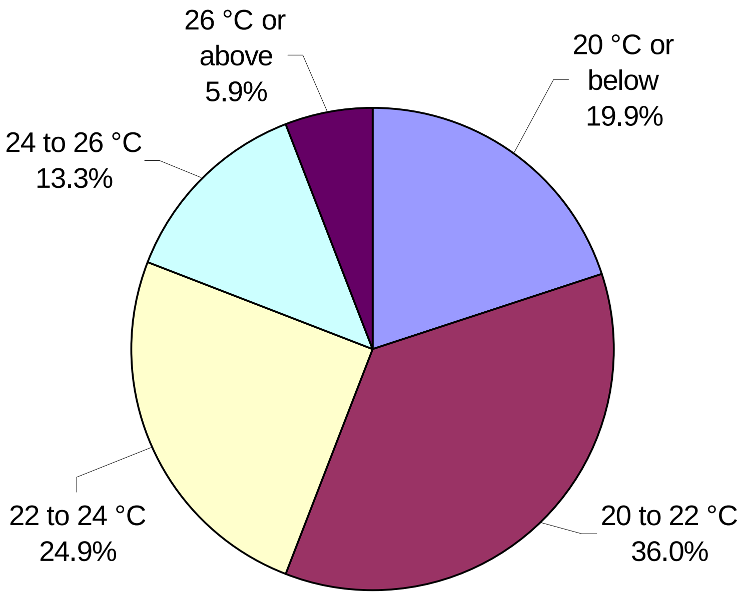

Fig. 4.4 Percentage breakdown of the indoor air temperature settings by respondents

Approximately a quarter of respondents experienced waking up during sleep sometimes because of a high indoor air temperature while another quarter woke up sometimes because of a low indoor air temperature. About 10% of the respondents experienced waking up when indoor air temperature fluctuated. These suggested that approximately 60% of the respondents had experience of waking up during sleep because of thermal discomfort even if their bedroom RACs were turned on.

4.3.1.3 Use of bedding and sleepwear

When the respondents were asleep and their RACs turned on, 47% of the respondents wear sleepwear (full-slip) and 45% wear briefs / panties and half-slips. Half of the respondents covered themselves with quilts, 40% with blankets and the remaining 10% just did not use any bedding. This indicated that most of the respondents would wear some sleepwear and use some bedding during their sleep with RACs turned on.

4.3.1.4 Ventilation and indoor air quality

The majority of the respondents could feel the airflow from RACs when they lied down on bed, but 32% of them did not like the airflow. Over 70% of the respondents felt stuffy because of poor IAQ. However, only 32% experienced waking up during sleep because of poor IAQ.

Regarding the ventilation control for RACs, 66% of the respondents did not know that there was a ventilation switch (bar) in a WRAC (as shown in Fig. 4.7). Over half of the respondents did not know the actual function of the ventilation switch even if they knew its existence in a WRAC. Many misunderstood that the purpose of the ventilation switch in a WRAC was for ventilation only when the WRAC ran at ventilation mode. These respondents did not know that a certain amount of indoor air could be exhausted to outdoors and therefore inducing the same amount of outdoor air to indoors when the WRAC ran at cooling mode. As shown in Fig. 4.5, 68% of the respondents did not use any ventilation control intentionally during their sleep with RACs turned on (this 68% of respondents might probably also include those who did use certain ventilation control without being aware of that they actually did).

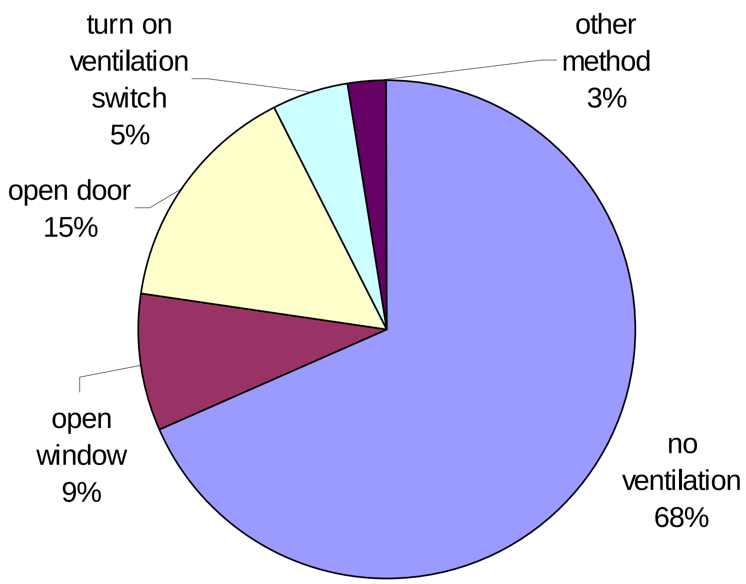

Fig. 4.5 Percentage breakdown of ventilation control strategies used by respondents

4.3.1.5 Other issues

Regarding the effect of the noise from operating RACs on sleeping quality, nearly 70% of respondents felt that the noise level from other sources such as traffic was higher than that from the RACs in their bedrooms. The majority felt that the noise level from a RAC had little influence on their sleeping quality.

On the other hand, the survey results suggested that the percentage breakdown on the preferred indoor air temperature settings by both male and female was almost the same. Furthermore, the survey results also indicated that older people would prefer a relatively higher indoor air temperature.

Generally speaking, 93% of the respondents indicated that using RACs could help improve their sleeping quality.

4.3.2 Results of the field monitoring

Overnight indoor air temperature and relative humidity were continuously monitored at an interval of one minute throughout the period in which the occupants slept with RACs turned on, using temperature and relative humidity sensors with data logging provisions in ten bedrooms in high-rise residential buildings in Hong Kong. The model of the temperature and relative humidity sensor was HOBO H8 Pro RH/Temp, whose measuring ranges and accuracies for air temperature and relative humidity were -30 ~ 50 °C and ±0.2 °C, and 0 ~ 100% and ±3%, respectively. Such a sensor was placed at a height of 1.1 m from floor level in the center of a bedroom. Table 4.1 shows the field monitoring results of overnight mean indoor air temperature and relative humidity in the bedrooms under investigation. The averaged overnight mean indoor air temperature and relative humidity for the studied bedrooms were 23.3 °C, and 54.7%, respectively.

The continuous monitoring of indoor overnight CO_2 levels using CO_2 monitors with data-logging provisions was also carried out in twelve air conditioned bedrooms of various sizes in high-rise residential buildings in Hong Kong. Detailed results of the monitoring of indoor overnight CO_2 levels are reported in Section 7.2.1.2.

Table 4.1 Results of the field monitoring of overnight mean indoor air temperature and relative humidity

Bedroom No.	Mean indoor air temperature (°C)	Mean relative humidity (%)
1	23.2	61.3
2	24.2	56.4
3	23.8	42.3
4	22.6	60.1
5	23.4	46.9
6	22.9	60.8
7	23.8	60.5
8	22.6	61.8
9	23.5	48.0
10	22.9	48.7
Average	**23.3**	**54.7**

4.4 Concluding remarks

The survey results suggested that the majority of the respondents would use their RACs at night when sleeping for more than two months in a year. This may be ascribed to the hot and humid climate and the comparatively small diurnal variations in ambient air temperature in subtropical Hong Kong.

Although the survey results suggested that over 70% of the respondents shared the views with their family members regarding turning on RACs and indoor air temperature settings, a wide range of the adaptation to summer heat (the preferred indoor air temperatures ranging from below 20 °C to over 26 °C) became obvious, as illustrated in Fig. 4.4. This suggested that people might develop distinct individual temperature and thermal comfort preferences. However, over 60% of the respondents would like to maintain an indoor air temperature between 20 and 24 °C, and about 20% of the respondents would prefer even a lower indoor air temperature at below 20 °C.

Normally, in the cases when SRACs are used, the respondents would be able to know the actual indoor air temperature settings from the remote controllers for SRACs, as shown in Fig. 4.6. However, in the cases when WRACs are used, no remote controllers are available and the thermostat knob in a WRAC (as shown in Fig. 4.7) is not labeled with actual temperature. Typically the thermostat knob in a WRAC is marked with an arbitrary 1 to 9 scale, with “1” indicating “Warmer” and “9” “Cooler” or simply an arrow indicating “Cooler” as in the WRAC shown in Fig. 4.7. Therefore, users might not know the actual indoor air temperatures maintained in their bedrooms. Generally speaking, the survey results implied that most respondents would prefer a lower indoor air temperature than the suggested design indoor air temperature (24-25°C, shown in Fig. 2.4). This can be further supported by the results of field monitoring of overnight mean indoor air temperature, averaged at 23.3 °C. This may be partly due to the relatively higher thermal insulation values of sleepwear and bedding used by the respondents during sleep.

Fig. 4.6 The remote controller of a typical SRAC showing indoor air temperature setting

The survey results also showed that nearly half of the respondents would wear full-slip sleepwear and 90% of the respondents would cover themselves with quilts or blankets during sleep with RACs turned on. A question may therefore arise: why should a relatively low bedroom air temperature be maintained, while people are covered with a rather thick quilt at the same time? A lower indoor air temperature means more energy consumption for RACs (see further discussion on this in Section 6.4.4). Moreover, a low air temperature means cold stress to human, which has more negative effects on the sleep quality than heat stress [Haskell et al. 1981]. This may be supported by the survey results that a quarter of respondents experienced waking up during sleep sometimes because of the low indoor air temperature. On the other hand, the fact that around 60% of the respondents had experience of waking up during sleep because of thermal discomfort even if their bedroom RACs were turned on may suggest an opportunity for fundamentally re-designing of RACs' control.

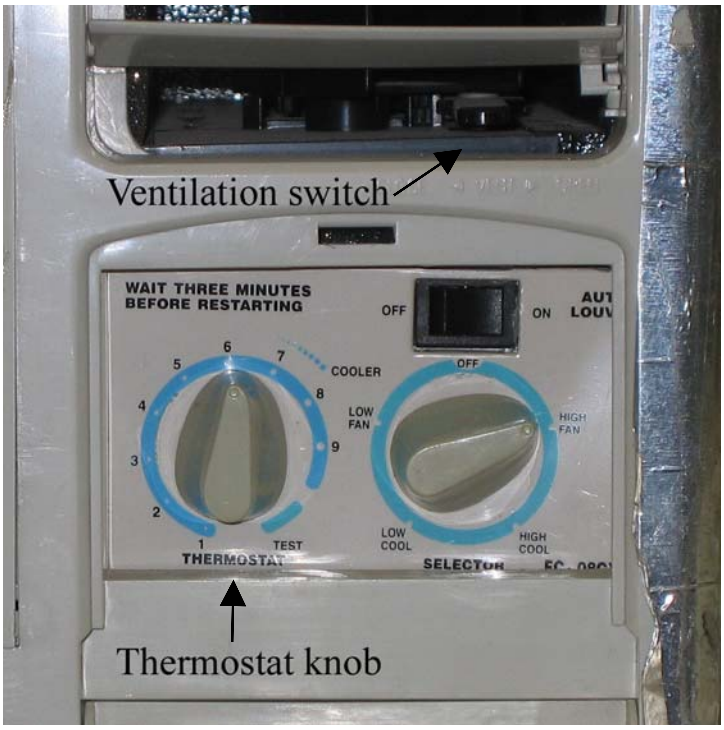

Fig. 4.7 The thermostat knob and ventilation switch in a typical WRAC

The survey results suggested that over 70% of the respondents felt stuffy because of poor IAQ when their RACs were turned on, and that poor IAQ even caused 32% of the respondents to wake up during sleep. This implied that the current situation of IAQ in bedrooms might not be acceptable for most of the surveyed respondents and the ventilation was not adequate to maintain an acceptable IAQ in most of the bedrooms in Hong Kong. The survey results also showed that nearly 70% of the respondents did not use any ventilation control methods (including those who used without being aware of that they actually did) during sleep with RACs turned on. Poor ventilation would therefore result in poor IAQ and make the occupants feel stuffy, hence affecting their sleeping quality. This may also suggest there is an urgent need to review and ultimately redesign the ventilation controls for RACs.

On the other hand, approximately two-third of the respondents did not know that there is a ventilation switch (bar) in a WRAC (as shown in Fig. 4.7) and over half of the respondents did not know the function of a ventilation switch even if they knew its existence in a WRAC. This finding is consistent with the results in a previous study [Kempton et al. 1992], where three-quarters of the surveyed residents did not use their thermostats in WRACs to control cooling, instead, they switched RACs on and off manually. Many were not aware that there was a thermostat in a RAC and even tended to think of the thermostat as an air flow controller, rather than a temperature controller. These surprising mismatches between the control devices provided on WRACs and the ways most people actually use them indicated an urgent need for user education. Furthermore, many people did not have experience with air-conditioning, and their confusions over the functions of the components on a control panel of a WRAC were probably very common. In some cases, differences in use can be traced to residents’ understanding of how RACs should be controlled – ideas often at odds with formal engineering understandings of air-conditioning control and function. But even in cases where RACs were used in an intended fashion, residents’ understandings may also differ from the formal account [Lutzenhiser 1992]. Therefore, a brief but clear user’s manual for RACs would be elementary to help users understand and control these units.

Chapter 5

Thermal Comfort in Sleeping Environments

5.1 Introduction

One of fundamental purposes for HVAC systems is to provide building occupants with comfortable thermal conditions. The definition of thermal comfort (shown in Section 2.3.1) leaves open as to what is meant by condition of mind or satisfaction, but it correctly emphasizes that the judgment of comfort is a cognitive process involving many inputs influenced by physical, physiological, psychological, and other factors. In general, comfort occurs when body temperatures are held within narrow ranges, skin moisture is low, and the physiological effort of regulation is minimized. Comfort also depends on behavioral actions such as altering clothing, altering activity, changing posture or location, changing the thermostat setting, opening a window, complaining, or leaving a space [ASHARE 2001].

Fanger's PMV-PPD model has been widely used and accepted for design and field assessment of thermal comfort [Yang and Su 1997]. Both ISO Standard 7730 and the new version of ASHRAE Standard 55 [ANSI/ASHRAE 2004] include a short computer listing that facilitates the computing of PMV and PPD for a wide range of environmental and clothes resistance parameters. Although the PMV-PPD model is useful only for predicting steady-state comfort responses while a two-node model can be used to predict physiological responses or responses to transient situations, a person sleeping in an air conditioned environment can be considered as being in a steady state and close to thermally neutral. Therefore, Fanger's PMV-PPD model has been adopted as a base model in this study, and is to be modified so that the application of such a model may be extended to sleeping thermal environments.

For instance, the clothing area factor (f_{cl}) included in the PMV-PPD model is meaningless when a body is lying on a bed [McCullough et al. 1987]. Furthermore, it has been recommended in ISO Standard 7730 to use the PMV index when the metabolic rate (M) is between 46 W/m^2 and 232 W/m^2 (0.8 met and 4 met). The metabolic rate of a sleeping person is however 40 W/m^2 (0.7 met), which is slightly out of the above range. On the other hand, it is pointed out in the new version of ASHRAE Standard 55 that when sleeping or resting with a reclining posture, a bed and bedding may provide considerable thermal insulation. It is hardly possible to determine the thermal insulation for most sleeping or resting situations unless a sleeping individual is immobile. Individuals will adjust bedding so as to suit their preferences. Provided that adequate bedding materials are available, the thermal environmental conditions desired for sleeping and resting vary considerably from person to person and cannot be determined by the PMV-PPD methods included in the ASHRAE Standard [ANSI/ASHRAE 2004]. Therefore, it is necessary to modify Fanger's PMV-PPD model for extending its application to sleeping environments.

This chapter reports firstly on a theoretical study on thermal comfort in sleeping environments. A comfort equation applicable to sleeping thermal environments was derived by introducing appropriate modifications to Fanger's comfort equation. This is followed by reporting an experimental study on measuring the total insulation values for a wide range of bedding systems (through different combinations of bed, bedding and its percentage coverage over a human body, and sleepwear) commonly used in the subtropical regions using a thermal manikin. Thirdly, it presents comfort charts which were established by solving the comfort equation, and can be used for determining thermally neutral environmental conditions under a given bedding system. Both PMV and PPD indexes can also be calculated for the purpose of assessing thermal comfort if and when necessary. Finally, a number of issues related to the effect of the total insulation value provided by a bedding system on thermal comfort, the factors influencing the total resistance / insulation of a bedding system, the use of summer quilts and air conditioning culture are discussed.

5.2 Thermal comfort modelling for sleeping environments

Fanger [1970], Hardy [1949], Gagge and Nishi [1977], and Gagge and Hardy [1967] gave quantitative information on calculating the heat exchange between people and their environments. The mathematical descriptions of an energy balance equation and the statements for various terms of the heat exchange used in the heat balance equation are detailed in the ASHRAE Handbook of Fundamentals [ASHRAE 2001].

5.2.1 Energy balance of a human body

Fig. 5.1 shows the thermal interaction between a human body and its environment. The total metabolic rate of work (*M*) produced within the body is the metabolic rate required for the person's activity plus the metabolic rate required for shivering. A portion of the body's energy production may be expended as the external work done by muscles (W). The net heat production in the human body (M-W) is either stored (S), causing the body's temperature to rise, or dissipated to the environment through skin surface (q_{sk}) and respiratory tract (q_{res}). Therefore, the heat balance for a human body is

$$M - W = q_{sk} + q_{res} + S = (C + R + E_{sk}) + (C_{res} + E_{res}) + (S_{sk} + S_{cr}) \qquad (5.1)$$

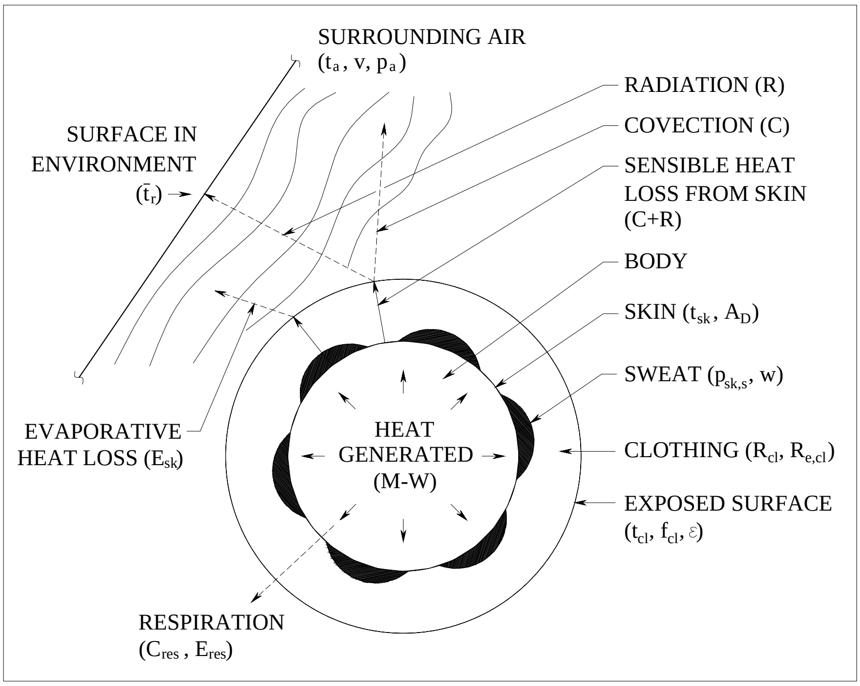

Fig. 5.1 The thermal interaction between a human body and its environment [ASHRAE 2001]

All the terms in Equation (5.1) have the units of power per unit area and refer to the surface area of a nude body. The most useful measure of nude body surface area, originally proposed by DuBois and DuBois [1916], is described by

$$A_D = 0.202 m^{0.425} l^{0.725} \qquad (5.2)$$

The DuBios surface area of an average person of 70 kg weight and 1.73 m height is 1.8 m^2.

A human body can be considered as consisting of two concentric thermal compartments, the skin and the core [Stolwijk and Hardy 1977]. The rate of heat storage in the body can be written separately for each compartment in terms of thermal capacity and change rate of temperature in each compartment, as follows:

$$S_{cr} = \frac{(1-\alpha_{sk})mc_{p,b}}{A_D}\frac{dt_{cr}}{dT} \tag{5.3}$$

$$S_{sk} = \frac{\alpha_{sk}mc_{p,b}}{A_D}\frac{dt_{sk}}{dT} \tag{5.4}$$

5.2.2 Thermal exchanges between a human body and its environment

5.2.2.1 Sensible heat loss from skin

Sensible heat exchange from skin surface to a surrounding environment must pass through clothing. Both convective and radiative heat losses from the outer surface of a clothed body can be expressed in terms of a heat transfer coefficient and the difference between the mean temperature of the outer surface of the clothed body and an appropriate environmental temperature:

$$C = f_{cl}h_c(t_{cl} - t_a) \tag{5.5}$$

$$R = f_{cl} h_r (t_{cl} - \bar{t}_r) \tag{5.6}$$

The coefficients, h_c, and, h_r, are both evaluated at the clothing surface. Equations (5.5) and (5.6) are commonly combined to describe the total sensible heat exchange by these two heat exchange mechanisms in terms of an operative temperature, t_o, and a combined heat transfer coefficient, h:

$$C + R = f_{cl} h (t_{cl} - t_o) \tag{5.7}$$

where

$$t_o = \frac{h_r \bar{t}_r + h_c t_a}{h_r + h_c} \tag{5.8}$$

$$h = h_r + h_c \tag{5.9}$$

Based on Equation (5.8), the operative temperature, t_o, can be defined as the average of the mean radiant and ambient air temperatures, weighted by their respective heat transfer coefficients. The actual transport of sensible heat passing through clothing involves conduction, convection, and radiation. It is usually convenient to combine these into a single thermal resistance of clothing, R_{cl}

$$C + R = (\bar{t}_{sk} - t_{cl}) / R_{cl} \tag{5.10}$$

Combining Equations (5.7) and (5.10) to eliminate t_{cl}:

$$C + R = \frac{\bar{t}_{sk} - t_o}{R_{cl} + 1/(f_{cl} h)} \tag{5.11}$$

5.2.2.2 Evaporative heat loss from skin

Evaporative heat loss from skin, E_{sk}, depends on the amount of moisture on skin and the difference between the water vapor pressure at skin surface and that in the ambient environment:

$$E_{sk} = \frac{w(p_{sk,s} - p_a)}{R_{e,cl} + 1/(f_{cl}h_e)} \tag{5.12}$$

Evaporative heat loss from skin is a combination of the evaporation of sweat secreted due to thermoregulatory control mechanisms, E_{rsw}, and the natural diffusion of water through skin, E_{dif}:

$$E_{sk} = E_{rsw} + E_{dif} \tag{5.13}$$

Theoretically, the maximum possible evaporative heat loss from a skin surface, E_{max}, occurs when the skin surface is completely wet (i.e., the skin wettedness, w, is equal to 1). The skin wettedness is the ratio of the actual evaporative heat loss to the maximum possible evaporative heat loss, E_{max}:

$$w = E_{sk} / E_{\max} \tag{5.14}$$

Skin wettedness is important in determining evaporative heat loss. It ranges from about 0.06 caused by E_{dif} alone (i.e., with no regulatory sweating) for normal conditions, to 1.0 when theoretically a skin surface is totally wet with perspiration, a condition that occurs rarely in practice. For large values of E_{max} or long exposures to low humidity, the value of w may drop to as low as 0.02, since dehydration of outer skin layers alters their diffusive characteristics.

5.2.2.3 Respiratory losses

Respiratory heat loss, q_{res}, is often expressed in terms of sensible heat loss, C_{res}, and latent heat loss, E_{res}. Sensible loss (C_{res}) and latent loss (E_{res}) due to respiration are relatively small compared to the other terms in Equation (5.1) and can be estimated respectively by the following equations [ASHRAE 2001]:

$$C_{res} = 0.0014M(34 - t_a) \tag{5.15}$$

$$E_{res} = 0.0173M(5.87 - p_a) \tag{5.16}$$

5.2.3 Assumptions and modifications adopted for sleeping environments

Equations (5.1) to (5.16) are normally applicable to sedentary or near sedentary physical activity levels, e.g. typical office work. Assumptions and modifications are needed if these equations are to become applicable to sleeping environments.

For a sleeping person in a reclining posture with a specific bedding system which consists of a bed and mattress, bedding and sleepwear, it is assumed that the sleeping person is immobile during the whole period of sleep, therefore,

$$M = 40 \text{ W/m}^2 \tag{5.17}$$

$$W = 0 \text{ W/m}^2 \tag{5.18}$$

For a bedding system rather than clothing, the intrinsic clothing resistance, R_{cl}, in Equation (5.11), cannot be determined because the clothing area factor, f_{cl}, is meaningless when a body is lying on a bed [McCullough, et al. 1987]. Therefore, Equation (5.11) can be rearranged in terms of the total thermal resistance (R_t) provided by a bed, pillow, bedding, sleepwear and the air layer surrounding a human body so that the intrinsic clothing resistance, R_{cl}, and the clothing area factor, f_{cl}, may be substituted by R_t, as follows:

$$R_t = R_{cl} + 1/(hf_{cl}) = R_{cl} + R_a / f_{cl} \tag{5.19}$$

$$C + R = \frac{\bar{t}_{sk} - t_o}{R_t} \tag{5.20}$$

Similarly, Equation (5.12) can be rewritten to become

$$E_{sk} = \frac{w(p_{sk,s} - p_a)}{R_{e,t}} \tag{5.21}$$

According to the Lewis Relation

$$i_m L_R = \frac{R_t}{R_{e,t}} \tag{5.22}$$

Combining Equations (5.21) and (5.22) to eliminate $R_{e,t}$:

$$E_{sk} = \frac{i_m L_R w(p_{sk,s} - p_a)}{R_t} \tag{5.23}$$

where Lewis ratio (L_R) equals approximately to 16.5 K/kPa at typical indoor conditions [ASHRAE 2001].

Since the purpose of the thermoregulatory system in a human body is to maintain an essentially constant internal body temperature, it can be assumed that for long exposure (for periods not less than 15 minutes as specified in ASHRAE Standard 55) to a constant sleeping thermal environment with a constant (*M*-*W*), a heat balance will exist for the human body (steady state condition). In other words, there will be no significant heat storage within the body. Therefore, Equations (5.3) and (5.4) can be changed to:

$$S_{cr} = \frac{(1-\alpha_{sk})mc_{p,b}}{A_D}\frac{dt_{cr}}{dT} = 0 \tag{5.24}$$

$$S_{sk} = \frac{\alpha_{sk}mc_{p,b}}{A_D}\frac{dt_{sk}}{dT} = 0 \tag{5.25}$$

Based on all the assumptions and modifications introduced above for sleeping environments, Equation (5.1) can be rewritten to become:

$$40 = \frac{\bar{t}_{sk} - t_o}{R_t} + \frac{i_m L_R w(p_{sk,s} - p_a)}{R_t} + 0.056(34 - t_a) + 0.0692(5.87 - p_a) \tag{5.26}$$

5.2.4 Conditions for thermal comfort in sleeping environments

A basic condition for thermal comfort in sleeping environments is that thermal neutrality is achieved during sleep. Thermal neutrality for a person is defined as the condition in which the subject would prefer neither warmer nor cooler surroundings. Obviously the first requirement for thermal comfort in sleeping environments is that the heat balance Equation (5.26) be satisfied. However, heat balance alone is not sufficient to achieve thermal comfort. In a wide range of environmental conditions where heat balance can be obtained, thermal comfort may be achieved only within a narrow range of the conditions. The following linear regression equations indicate values of t_{sk} and E_{rsw} that provide thermal comfort, which were proposed as the second and third conditions for optimal thermal comfort by Fanger [1970]:

$$\bar{t}_{sk,req} = 35.7 - 0.0275(M - W) \tag{5.27}$$

$$E_{rsw,req} = 0.42(M - W - 58.15) \tag{5.28}$$

It can be seen from the two equations that in a state of physiological thermal neutrality during sedentary ($M = 58.15\ W/m^2$, $W = 0$), the mean skin temperature is around 34 °C and there is no regulation of body temperature by sweating (i.e., sweating does not occur). The skin temperature necessary for comfort falls, and moderate sweating takes place at a higher activity level. However, in a state of thermal neutrality for a sleeping person whose activity level is lower than sedentary ($M = 40\ W/m^2$, $W = 0$), the mean skin temperature would increase and sweating would either not occur ($E_{rsw,req}$ is meaningless when less than zero). Therefore, the second and third conditions for thermal comfort in a sleeping environment may be changed to:

$$\bar{t}_{sk,req} = 35.7 - 0.0275(M - W) = 34.6 \ (°C) \tag{5.29}$$

$$E_{rsw,req} = 0 \tag{5.30}$$

With no regulatory sweating for normal conditions, the skin wettedness (w) equals to 0.06, caused by E_{dif} alone [Gagge et al. 1986].

$$w = 0.06 \tag{5.31}$$

5.2.5 Comfort equation for sleeping environments

Combining the three conditions (i.e., Equations (5.26), (5.29) and (5.31)) for thermal comfort in a sleeping environment to obtain:

$$40 = \frac{34.6 - t_o}{R_t} + \frac{0.06 i_m L_R (p_{sk,s} - p_a)}{R_t} + 0.056(34 - t_a) + 0.0692(5.87 - p_a) \tag{5.32}$$

In order to solve Equation (5.32), some of its parameters such as the heat transfer coefficients, h_r, and, h_c, and the permeation efficiency, i_m, need to be determined. ASHRAE Handbook of Fundamentals [ASHRAE 2001] provides the necessary data and methods used to calculate these parameters.

The linearized radiative heat transfer coefficients, h_r, can be calculated by

$$h_r = 4\varepsilon\sigma \frac{A_r}{A_D}\left[273.2 + \frac{t_{cl} + \bar{t}_r}{2}\right]^3 \tag{5.33}$$

It is not always possible to solve Equation (5.33) explicitly for the radiant heat transfer coefficient, h_r, since t_{cl} may also be an unknown. However, h_r is nearly constant for typical indoor temperatures, and a value of 4.7 W/(m^2·K) is sufficient for most calculations [ASHRAE 2001].

The convective heat transfer coefficient, h_c, for reclining persons can be calculated by [Colin and Houdas 1967]

$$h_c = 2.7 + 8.7v^{0.67} \quad \text{for } 0.15 < v < 1.5 \tag{5.34}$$

$$h_c = 5.1 \quad \text{for } 0 < v \leq 0.15 \tag{5.35}$$

Quantitative values of h_c are important, not only in estimating convection loss, but in evaluating operative temperature, t_o, and several other environmental indices such as the effective temperature, ET^*.

An experimental study by McCullough et al. [1989] has suggested that the permeation efficiency, i_m, of ensembles worn indoors generally fell in the range of 0.3 to 0.5, and that assuming i_m = 0.38 is reasonably accurate for most applications although i_m for a given clothing ensemble is a function of the environment as well as the clothing properties. Since the properties of bedding are likely to be similar to that of clothing, such a value (i_m = 0.38) can also be adopted for a bedding system.

According to the properties of saturated water / steam, the water vapor partial pressure in saturated air, $p_{sk,s}$, when t_{sk} = *34.6* °C, is:

$$p_{sk,s} = 5.52 \ \text{(kPa)} \tag{5.36}$$

Using Equations (5.8), (5.36) and i_m = 0.38, L_R = 16.5 K/kPa, h_r = 4.7 W/(m^2·K), a comfort equation for sleeping environments, which combines both environmental and personal variables to produce a thermal neutral sensation, may be derived from Equation (5.32) as follows:

$$40 = \frac{1}{R_t}\left[\left(34.6 - \frac{4.7\bar{t}_r + h_c t_a}{4.7 + h_c}\right) + 0.3762(5.52 - p_a)\right] \\ + 0.056(34 - t_a) + 0.0692(5.87 - p_a) \tag{5.37}$$

Obviously the satisfaction of the Comfort Equation (5.37) means that the three comfort conditions are met at the same time since it combines the three equations for thermal comfort in sleeping environments. There are five variables, R_t, $\bar{t}_r$, t_a, p_a, and h_c in Equation (5.37). The convective heat transfer coefficient, h_c, is the function of air velocity, v, therefore, four variables (i.e., $\bar{t}_r$, t_a, p_a, and v) are thermal environmental variables. The total thermal resistance, R_t, is the function of a number of variables such as bedding, sleepwear, bed and mattress, the percentage coverage of body surface area by bedding and bed, air velocity, direction of airflow, and posture, etc. How these factors would influence the total thermal resistance of a bedding system will be discussed in Sections 5.3.5.1 to 5.3.5.4 and 5.5.2.

5.2.6 PMV and PPD for sleeping environments

Fanger [1970] suggested a predicted mean vote (PMV) index which predicted the mean response of a large group of people according to the ASHRAE thermal sensation scale. He related PMV to the imbalance between the actual heat flow from a human body in a given environment and the heat flow required for optimum comfort at a specified activity by the following equation:

$$PMV = [0.303\exp(-0.036M) + 0.028]L = \alpha L \tag{5.38}$$

where L is the thermal load on the body, defined as the difference between internal heat production and heat loss to the environment for a person hypothetically kept at comfort values of t_{sk} and E_{rsw} at the activity level. Thermal load, L, is therefore the difference between the left hand- and right hand- sides of Comfort Equation calculated for the given values of the environmental conditions.

After estimating the PMV using Equation (5.38), the predicted percent dissatisfied (PPD) with a given condition can also be estimated by

$$PPD = 100 - 95\exp[-(0.03353PMV^4 + 0.2179PMV^2)] \tag{5.39}$$

where dissatisfaction is defined as anybody not voting -1, +1, or 0. A PPD of 10% corresponds to the PMV range of ± 0.5, and even with PMV = 0, about 5% of the people are dissatisfied.

Fanger's PMV-PPD model is widely used and accepted for design and field assessment of thermal comfort. However, the sensitivity coefficient, α, used in the PMV Equation (5.38) requires to be experimentally evaluated over a wide range of conditions. The α value in Equation (5.38) was obtained based on the results from experiments with 1296 human subjects with different metabolic rate (≥ 58.15 W/m^2) in controlled climate chambers. For sleeping environments, it is hardly possible to carry out experiments with a statistically meaningful number of human subjects for quantifying the sensitivity coefficient. Furthermore, it is not possible either to conduct a questionnaire survey during the course of sleep. A survey may only be conducted after sleep when however a tested human subject sample may not be able to accurately relate his / her sleep quality to his / her thermal environment. Therefore, it has to be assumed that the sensitivity coefficient, α, obtained by Fanger is also applicable to sleeping environments although the relatively low activity level (sleep) was not included in the experiments. In other words, extrapolation was applied to extend the range of metabolic rate down to 40 W/m^2 (0.7 met when sleeping). Hence, the PMV for a sleeping environment can be calculated by

$$PMV = 0.0998\left\{40 - \frac{1}{R_t}\left[\left(34.6 - \frac{4.7t_r + h_c t_a}{4.7t_r + h_c}\right) + 0.3762(5.52 - p_a)\right]\right\} - 0.0998\left[0.056(34 - t_a) + 0.0692(5.87 - p_a)\right] \tag{5.40}$$

The PPD for a sleeping environment can then be determined by Equation (5.39).

5.3 Measurements of the total resistance / insulation for bedding systems

In order to evaluate various combinations of the environmental variables which will achieve optimal thermal comfort for sleeping persons under steady state conditions [i.e., to solve Comfort Equation (5.37)], the total resistance, R_t, of bedding systems corresponding to those commonly used in the subtropics must be available. However, no database is currently available on the thermal insulation for bedding systems used for air conditioned sleeping environments in the subtropical regions except some documents such as ASHRAE Standard 55 and ISO Standard 9920 giving insulation values for clothing. Therefore, the thermal insulation provided by various bedding systems commonly used in the subtropics must be obtained through extensive laboratory experimental work.

The experimental work was based on a computational thermal manikin (CTM) with geometry of a real human body in an environmental chamber. The purpose of the experimental work was to develop a database of the total insulation values (I_T) of bedding systems commonly used in the subtropics. The database obtained was used as an input to solve the Comfort Equation for sleeping environments, and may further be adopted by various standards such an ISO Standard or an ASHRAE Standard. The measurements and calculations of the total insulation values were carried out in accordance with the corresponding ISO [1995] and ASTM [1999] Standards.

5.3.1 Selection of bedding, sleepwear, bed and mattress

The components of a bedding system include bedding, bed and mattress, and the sleepwear used by occupants during sleep. The results of the questionnaire survey reported in Chapter 4 showed that 92% of the respondents would wear sleepwear (47% used full-slip while the other 45% used half-slip sleepwear) and 90% would cover themselves with quilts (50%) or blankets (40%) during sleep. This helped determine the bedding items and sleepwear type to be used in experiments. Two types of summer quilt (i.e., so called locally air conditioning quilt), a blanket and a multi-purpose quilt as well as a full-slip and a half-slip sleepwear were purchased for use in experimentation. On the other hand, although a variety of mattress types are available, any conventional mattress (i.e., excluding a water bed or a thin cot) will provide considerably more local insulation than bedding. For all practical purposes, it can be assumed that there is no heat loss in steady-state through the body surface in contact with a mattress. However, the firmness of a mattress might affect the amount of body surface area in contact with the mattress, but the resultant variation in insulation is minimal among conventional mattresses [McCullough et al. 1987]. Therefore, only one conventional mattress was purchased for use in experimentation (shown in Fig. 5.2a). Furthermore, it is usually argued that because the area of contact between skin and a mattress is small and the mattress is thick, the heat loss through the mattress can be neglected [Kerslate 1991]. In order to investigate the contribution of bed and mattress to the insulation of a bedding system, a Chinese traditional style bed (called *Zongbang* bed, shown in Fig. 5.2b), was also self-made using palm rope to provide a reference for comparison. This kind of *Zongbang* bed is widely used in the southern provinces in China, in particular in rural areas where air conditioning is not readily available.

Fig. 5.2a A conventional mattress

Fig. 5.2b The *Zongbang* bed

5.3.2 Measurement of the physical properties of textiles

The bed sheet, quilt cover, blanket and sleepwear were laundered once according to the American Association of Textile Chemists and Colorists (AATCC) test method 135 [AATCC 2003] to remove any excess sizing from the fabrics. Weight per unit area of these bedding and sleepwear items was measured according to ASTM D 3776 [ASTM 2002a]. The fabric thickness of each textile item was measured according to ASTM D 1777 [ASTM 2002b] using a carpet thickness gauge meter (measuring range: 0 ~ 25 mm; accuracy: ± 0.01 mm). The detailed measured characteristics of bedding items and sleepwear such as thickness, weight per unit area, fiber content, and fabric structure are shown in Table 5.1. Fig. 5.3 shows the blanket (B), the Summer Quilt 2 (Q2), the Summer Quilt 1 (Q1) and the Multi-purpose Quilt (Q3) from left- to right-hand side, respectively.

Table 5.1 Detailed measured characteristics of bedding items and sleepwear

Bedding item / sleepwear	Detailed description	Weight per unit area (kg/m^2)	Thickness (mm)
Blanket (B)	100% cotton	0.3308	3.03
Summer Quilt 1 (Q1)	Face: 100% cotton Filling: 100% polyester Back: 80% polyester, 20% cotton	0.5612	15.23
Summer Quilt 2 (Q2)	Filling: 100% feathery polyester Cover: Tracted polycotton	0.3096	7.62
Multi-purpose Quilt (Q3)	Filling: 100% polyester Cover: 70% polyester, 30% cotton	0.8390	23.17
Sheet 1	100% cotton	0.1133	0.96
Straw mat 2	100% straw	0.5064	1.45
Full-slip sleepwear (S1)	Long-sleeved, 100% cotton	-	0.46
Half-slip sleepwear (S2)	Short-sleeved, 100% cotton	-	0.83

Fig. 5.3 The Blanket (B), Summer Quilt 2 (Q2), Summer Quilt 1 (Q1) and Multi-purpose quilt (Q3) used in experimentation

5.3.3 Experimental method

Fig. 5.4a Thermal manikin, height 168 cm (modeled after a typical Scandinavian woman)

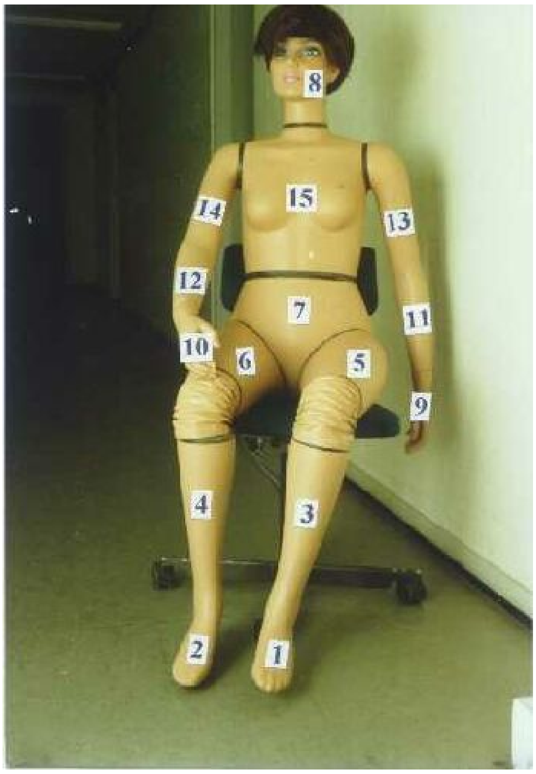

Fig. 5.4b The manikin's independent segments, each with its own heating and computer control system

The Manikin shown in Fig. 5.4a, Alex, is divided into 20 independent segments (Left/Right: foot, low leg, front thigh, back thigh, hand, forearm, upper arm, and pelvis, backside, face, crown, chest and back, as shown in Fig. 5.4b), each with its own temperature sensors, and heating and computer control system to approximately simulate the skin temperature distribution of a human being. In order to correctly simulate the thermal receptors all over the body of a human being, temperature sensing elements are distributed all over the manikin surface. The manikin is heated by the same wiring used for measuring. An individual proportional integrate (PI) controller is used to produce the required mean skin temperature in each body segment of the manikin. The mean skin temperature settings were 31 °C for hands and feet, and 35 °C for other body segments, respectively. Thus a mean skin temperature of 34.6 °C for the whole body (thermal neutral in a sleeping environment as given by Equation (5.29)) was produced.

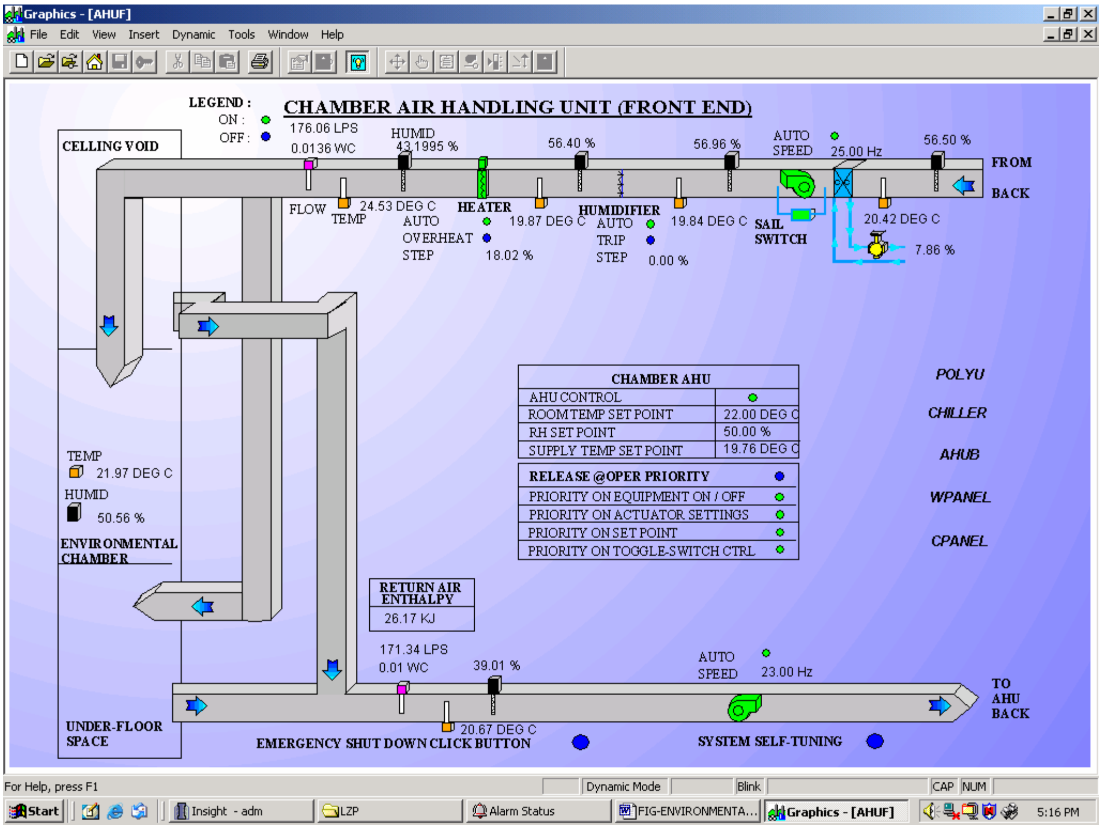

Fig. 5.5 Schematics of the conventional air conditioning system for the environmental chamber

The manikin was placed with a supine position on a single bed with a dimension of 1900 (L) × 920 (W) mm in an environmental chamber. Besides a conventional air conditioning system to control the indoor air temperature and relative humidity, a wall panel cooling / heating system, which can be used to control the wall temperature, is also available in the environmental chamber. There are 10 wall temperature sensors distributed evenly in the wall panels. Figures 5.5 and 5.6 show the schematics of the conventional air conditioning system and the hydronic system for wall panels in the environmental chamber, respectively.

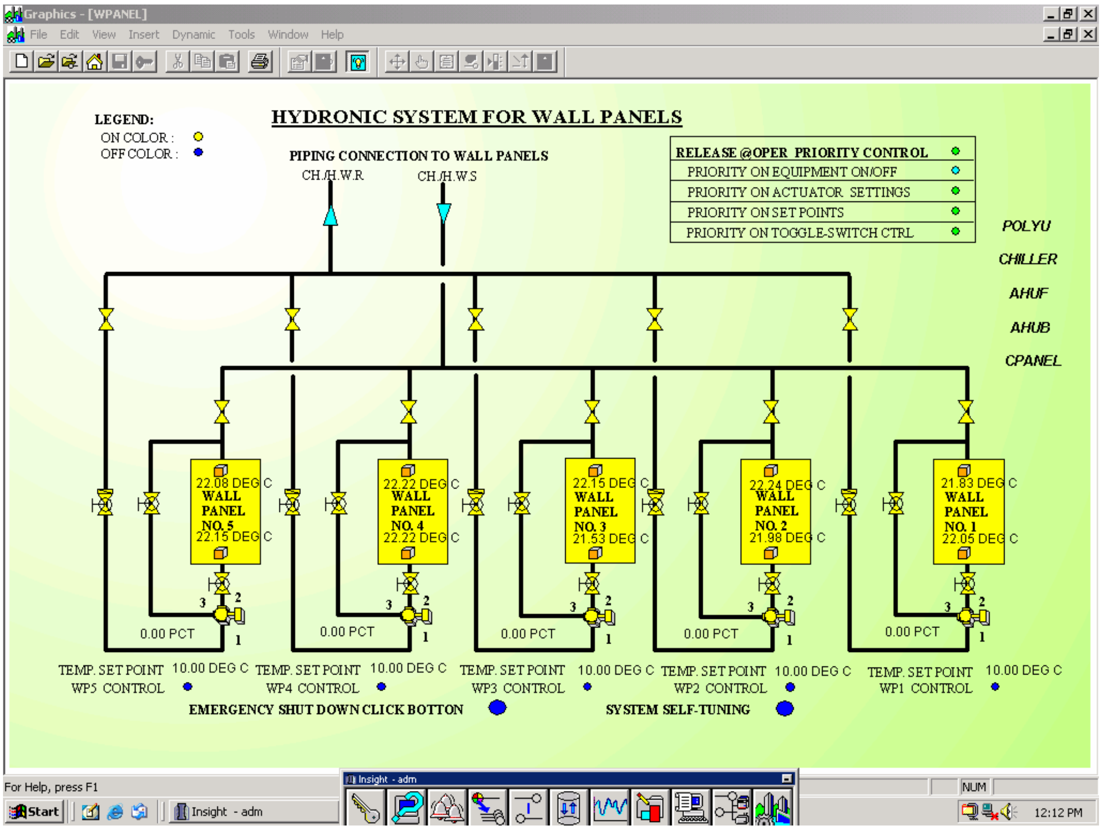

Fig. 5.6 Schematics of the hydronic system for wall panels in the environmental chamber

For each experimental condition, both the indoor air temperature and panel temperature were maintained constant at 22 °C, while the relative humidity inside the chamber at 50%. The air velocity was controlled at not greater than 0.15 m/s. During each test, the power input, the mean skin temperature of each segment, indoor air and panel wall temperature, and indoor relatively humidity were continuously monitored and recorded at an interval of one minute. The mean value of each parameter was calculated based on at least thirty steady-state data. Fig. 5.7

shows an example of one measuring condition. The total thermal resistance (R_T) and insulation (I_T) of a bedding system, including the resistance or insulation from the air layer around the tested manikin, were calculated, respectively, as follows:

$$R_T = \frac{\bar{t}_{sk} - t_o}{H_{sk}} = \frac{\sum(a_i t_{sk,i}) - t_o}{\sum(a_i H_i)} \tag{5.41}$$

$$I_T = KR_T \tag{5.42}$$

where K is unit constant, and equals to 6.45 clo·W/(m^2·°C).

The operative temperature was calculated as the mean of air and mean radiant temperature based on that the difference between mean radiant and air temperature is small (< 4°C):

$$t_o = \frac{t_a + \bar{t}_r}{2} \tag{5.43}$$

$$\bar{t}_r = \sum_{j=1}^{6} t_j F_{P-j} \tag{5.44}$$

A test was also carried out with the nude manikin suspended horizontally in air to determine the air layer insulation, I_a, (fixed at 0.73 clo, exactly the same value as that obtained by McCullough et al. [1987]).

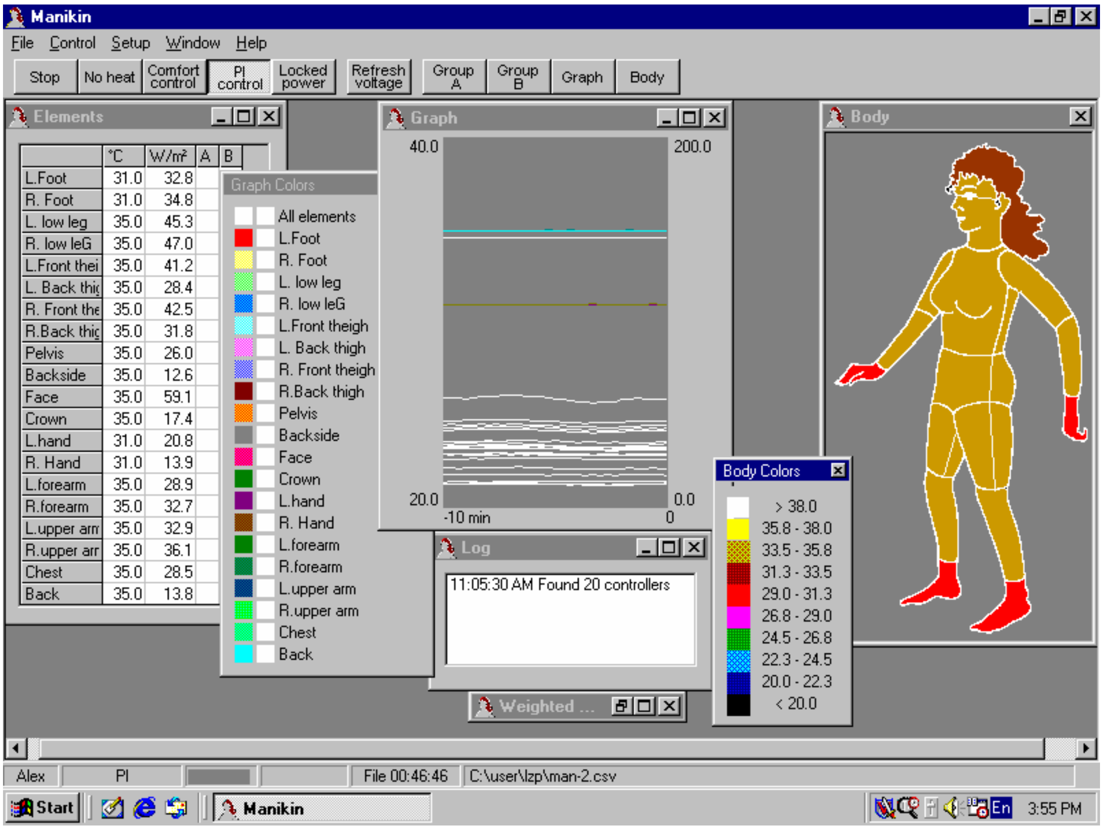

Fig. 5.7 Computer display of an experimental condition

5.3.4 Experimental conditions

The total insulation value of a bedding system may be influenced by various components that make up the bedding system such as bed and mattress, bedding and sleepwear, etc. It is not useful to obtain the thermal insulation of an individual component in isolation, as these components will be used together as part of the bedding system. Therefore the total insulation values for the different combinations of individual component shown in Table 5.2 were measured using the thermal manikin in the environmental chamber. In all experimental conditions, a feather pillow was placed under the manikin's head as shown in Fig. 5.8.

Table 5.2 Detailed combinations of individual components used in experimentation

Experi-mental condition No.	Bed type	
	Bed + conventional mattress + sheet 1 (M1)	*Zongbang* bed + straw mat 2 (M2)
1	Blanket (M1+B)	Blanket (M2+B)
2	Blanket + full-slip sleepwear (M1+B+S1)	Blanket + full-slip sleepwear (M2+B+S1)
3	Blanket + half-slip sleepwear (M1+B+S2)	Blanket + half-slip sleepwear (M2+B+S2)
4	Summer Quilt 1 (M1+Q1)	Summer Quilt 1 (M2+Q1)
5	Summer Quilt 1 + full-slip sleepwear (M1+Q1+S1)	Summer Quilt 1 + full-slip sleepwear (M2+Q1+S1)
6	Summer Quilt 1 + half-slip sleepwear (M1+Q1+S2)	Summer Quilt 1 + half-slip sleepwear (M2+Q1+S2)
7	Summer Quilt 2 (M1+Q2)	Summer Quilt 2 (M2+Q2)
8	Summer Quilt 2 + full-slip sleepwear (M1+Q2+S1)	Summer Quilt 2 + full-slip sleepwear (M2+Q2+S1)
9	Summer Quilt 2 + half-slip sleepwear (M1+Q2+S2)	Summer Quilt 2 + half-slip sleepwear (M2+Q2+S2)
10	Multi-purpose quilt + full-slip sleepwear (M1+Q3+S1)	-

When people use bedding, they seldom cover their entire bodies (at least their heads are exposed). People can change their personal insulation by covering and uncovering parts of their bodies with bedding to achieve thermal comfort. To systematically study this effect, McCullough et al. [1987] developed seventeen different configurations of the body surface coverage by bedding and bed, from the total coverage (100%) to no coverage (23.3% or nude on bed). Eight most commonly-used configurations shown in Fig. 5.9 were selected for experimentation. The figure illustrates the placement of bedding on the manikin, with the percentage coverage (A_C) of body surface area by bedding and bed indicated for each configuration [McCullough et al. 1987]. For each configuration of percentage coverage of body surface area shown in Fig. 5.9, the total insulation values of each combination of bedding, sleepwear, bed and mattress shown in Table 5.2 were measured.

Fig. 5.8 Examples of experimental conditions

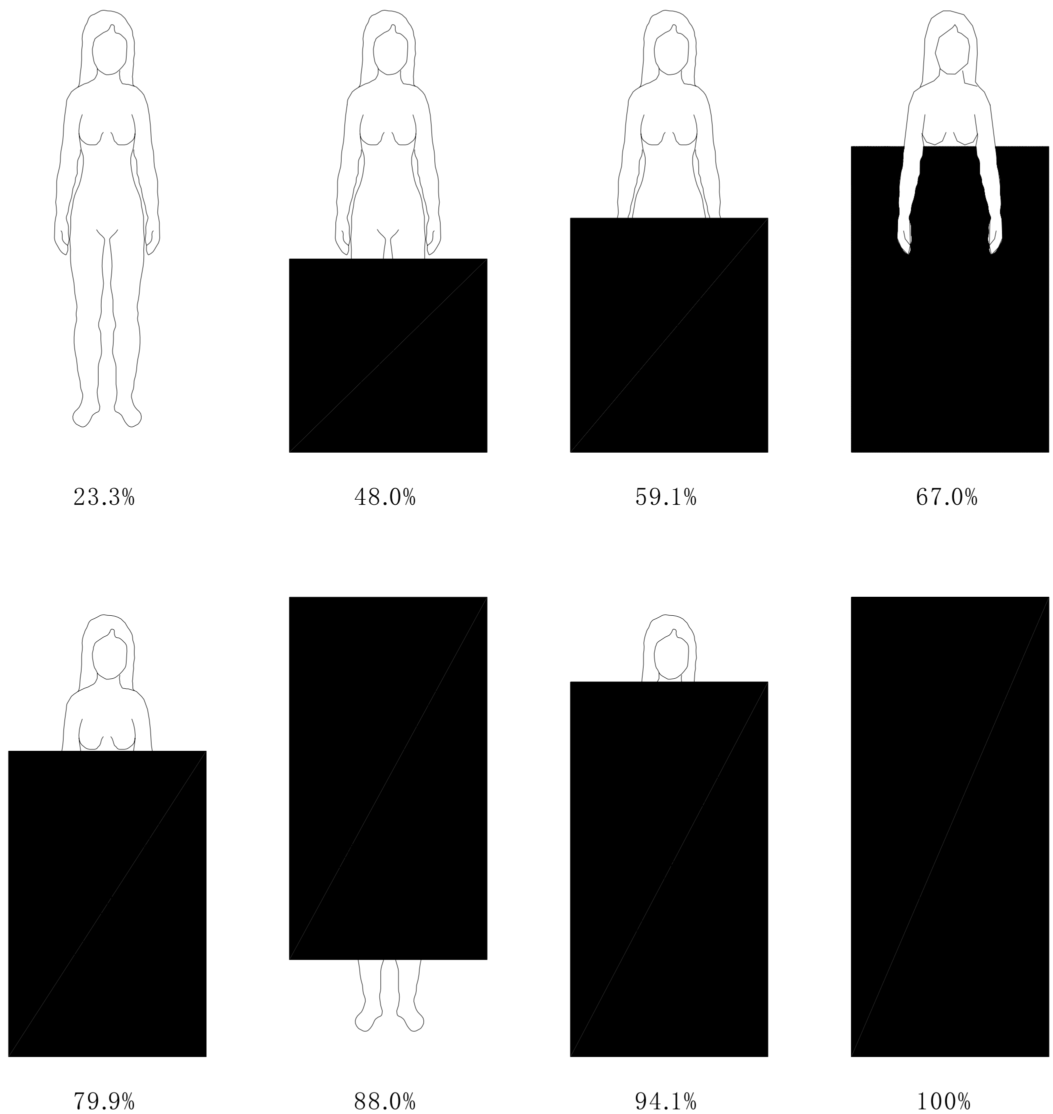

Fig. 5.9 Percentage coverage of body surface area by bedding and bed [McCullough et al. 1987] used in experimentation

5.3.5 Results of measurements

The results of measurements are given in Table 5.3. The total insulation values of the measured bedding systems ranged from 0.90 to 4.89 clo. All of the total insulation values include the insulations contributed by the pillow and the air layer surrounding the manikin.

Table 5.3 The total insulation values (clo) of the measured bedding systems

A_C (%)	23.3	48	59.1	67	79.9	88	94.1	100
M1+Q1+S1	1.57	2.15	2.72	2.88	3.27	3.99	4.56	4.77
M1+Q2+S1		1.84	2.24	2.41	2.81	3.32	3.73	4.06
M1+B+S1		1.82	2.08	2.18	2.22	2.41	2.56	2.65
M1+Q1+S2	1.38	1.65	2.15	2.62	3.18	3.79	4.34	4.60
M1+Q2+S2		1.53	1.93	2.20	2.68	3.26	3.55	3.92
M1+B+S2		1.43	1.76	1.80	2.07	2.36	2.40	2.58
M1+Q1	0.98	1.16	1.43	1.90	2.44	3.68	4.03	4.47
M1+Q2		1.14	1.42	1.69	1.98	2.95	3.03	3.62
M1+B		1.07	1.24	1.45	1.65	1.98	2.11	2.23
M2+Q1+S1	1.31	1.63	1.97	2.32	2.57	3.08	3.32	3.64
M2+Q2+S1		1.61	1.93	2.19	2.42	2.66	2.97	3.12
M2+B+S1		1.55	1.74	1.81	1.92	2.12	2.21	2.31
M2+Q1+S2	1.18	1.51	1.90	2.20	2.55	2.91	3.26	3.67
M2+Q2+S2		1.50	1.73	1.99	2.28	2.66	2.83	3.04
M2+B+S2		1.46	1.62	1.64	1.87	2.02	2.10	2.19
M2+Q1	0.90	1.09	1.35	1.83	2.06	2.67	3.00	3.26
M2+Q2		1.07	1.27	1.58	1.81	2.34	2.50	2.76
M2+B		1.04	1.18	1.30	1.45	1.74	1.84	1.90
M1+Q3+S1	1.57	2.39	2.86	3.08	3.53	4.15	4.66	4.89

5.3.5.1 Effect of the percentage coverage of body surface area by bedding and bed

Fig. 5.10, Fig. 5.11 and Fig. 5.12 illustrate the effect of changing the percentage coverage of body surface area by bedding and bed (A_C) on the total insulation of a bedding system (I_T) at nude (i.e., without sleepwear), full-slip and half-slip sleepwear conditions, respectively. It can be seen from the figures that A_C would affect I_T significantly. I_T would increase with the increase of A_C for all situations. For example, for the situation where the conventional mattress (M1) was used, when 48% of the body surface was covered, the I_T values when using the blanket (M1+B) and the Summer Quilt 1 (M1+Q1) were 1.07 and 1.16 clo, respectively, compared to 2.11 and 4.03 clo, respectively, when 94.1% of the body surface was covered. It can also be observed that when A_C is small, the differences in I_T under different bedding systems are not significant. For example, at conventional mattress condition, when A_C was 48%, the I_T values when using the blanket (M1+B), the Summer Quilt 2 (M1+Q2) and the Summer Quilt 1 (M1+Q1) were 1.07, 1.14 and 1.16 clo, respectively. However, with more and more body surface area covered by bedding, the differences in I_T under different bedding systems become evident. For example, at the same conventional mattress condition when the manikin was totally covered (100%), the I_T values when using the blanket, the Summer Quilt 2 and the Summer Quilt 1 were 2.23, 3.62 and 4.47 clo, respectively. The slopes of the data curves in the figures also indicate that as A_C increases, I_T rises at an increasing rate, in particular for the heavyweight bedding system (e.g., the Summer Quilt 1 (M1+Q1)).

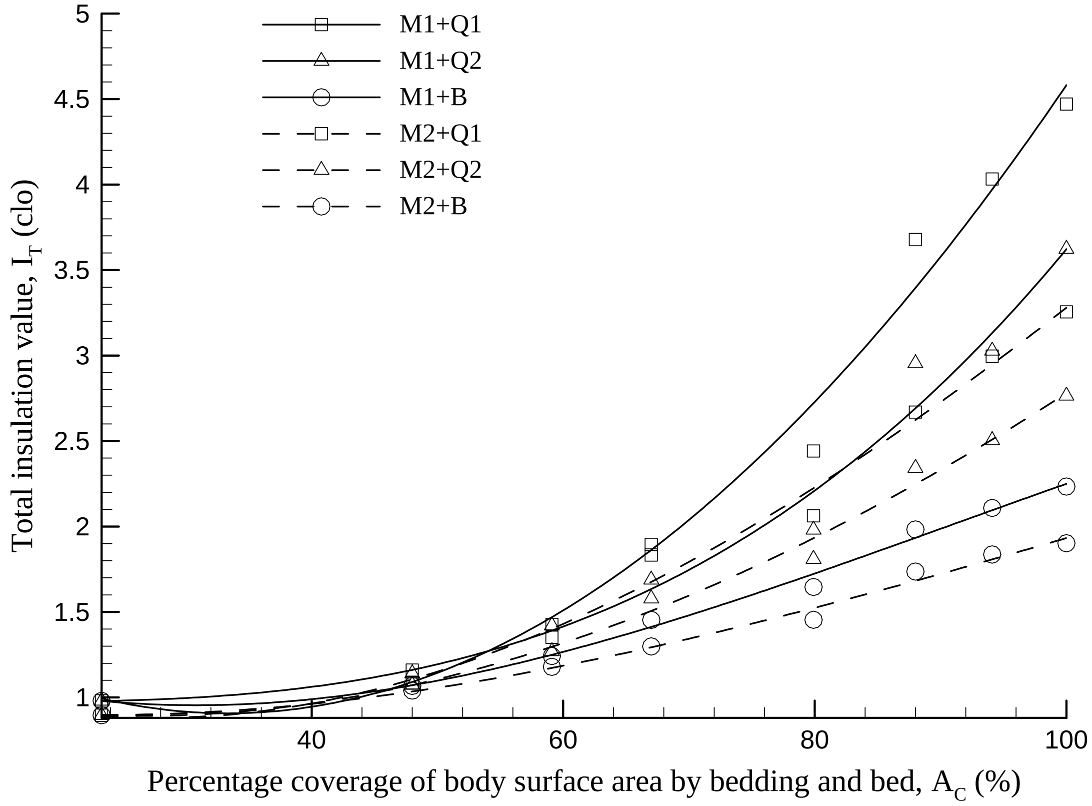

Fig. 5.10 The measured total insulation values provided by different bedding systems at different percentage coverage of body surface area by bedding and bed (nude condition)

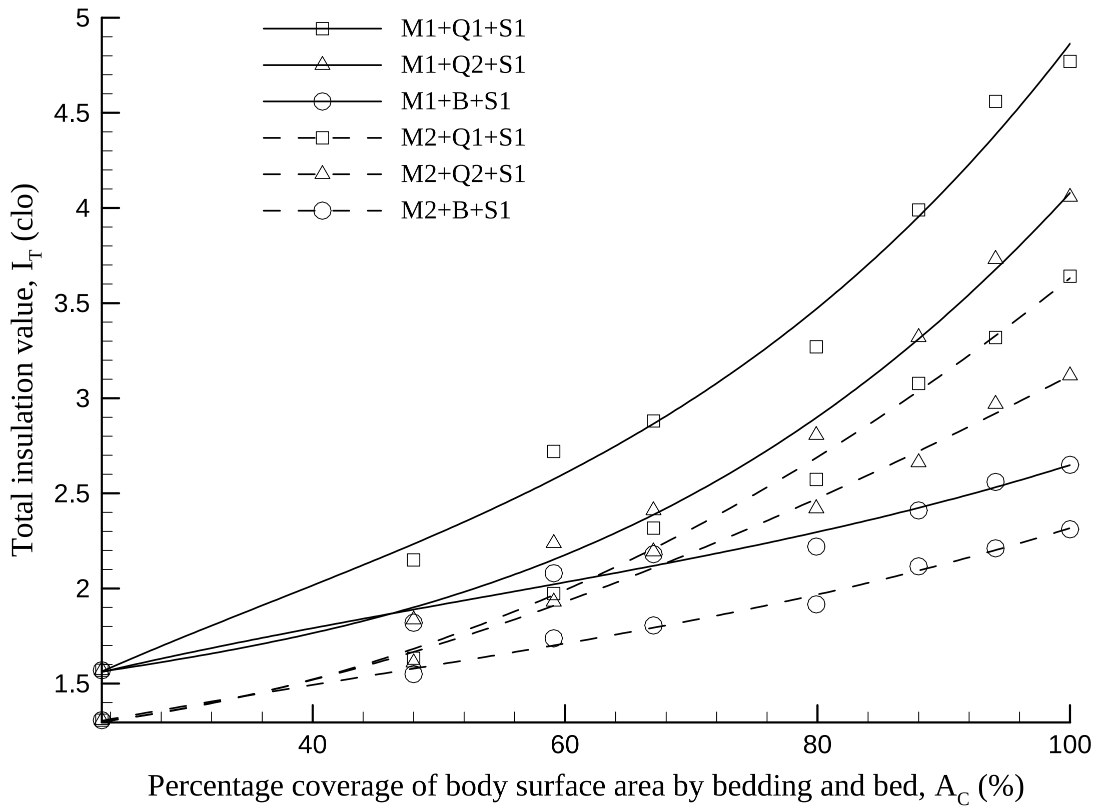

Fig. 5.11 The measured total insulation values provided by different bedding systems at different percentage coverage of body surface area by bedding and bed (full-slip sleepwear condition)

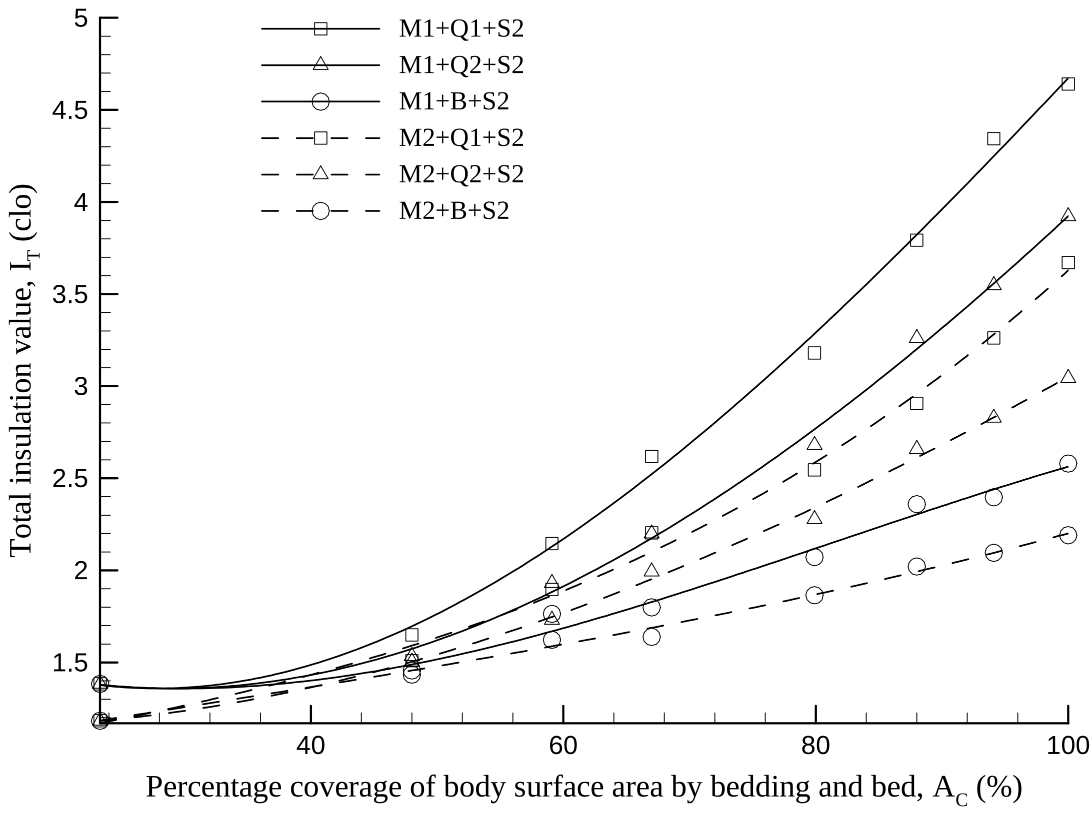

Fig. 5.12 The measured total insulation values provided by different bedding systems at different percentage coverage of body surface area by bedding and bed (half-slip sleepwear condition)

5.3.5.2 Effect of bedding

It can also be seen from Fig. 5.10, Fig. 5.11 and Fig. 5.12 that comparing the I_T values when using the Summer Quilts 1 and 2, both of which were made up of polyester, the thicker (or heavier) a summer quilt was, the higher the I_T value would be resulted in. This was understandable and consistent with common understanding. However, comparing the I_T values when using the blanket (made up of cotton) and the Summer Quilt 2, although the weight per unit area of the blanket was higher than that of the Summer Quilt 2, its thickness was less than that of the Summer Quilt 2 and, the measured I_T values when using the blanket were less than those when using the Summer Quilt 2. This implied that a bedding material had a limited influence on insulation, and that the insulation provided by a bedding component was mainly influenced by its thickness rather than its weight per unit area. This observation agreed well with the statement concerning the thermal insulation of clothing in ISO Standard 9920 [ISO 1995] - "The type of material, however, has a limited influence on the thermal insulation. Instead the insulation is mainly influenced by the thickness and body area covered".

On the other hand, it should be noted that the difference in I_T using different bedding would also depend on A_C and the type of sleepwear. For example, it can be observed in Fig. 5.10 that at the nude condition, the bedding did not significantly affect I_T when A_C was less than 60%. However, the differences in I_T using different bedding increased when A_C was greater than 60%. For the sleepwear conditions shown in Fig. 5.11 and Fig. 5.12, the differences in I_T using different bedding started to become noticeable when A_C was greater than 50%.

Fig. 5.13 shows the comparison of the measured total insulation values when using the three different quilts under the conventional mattress and full-slip sleepwear condition in order to evaluate their effects on the total insulation. The Summer Quilt 1 and the relatively thinner Summer Quilt 2 were those commonly used in summer whilst the Multi-purpose Quilt 3 was a typical one commonly used in spring and autumn in Hong Kong (even in winter considering the subtropical weather condition). It can be observed from Fig. 5.13 that although the I_T values when using the Summer

Quilt 1 were lower than those when using the Multi-purpose Quilt 3, the differences in the I_T values between using the Summer Quilt 1 and the Multi-purpose Quilt 3 ranged from 2% to 10%, which did not appear significantly large. On the other hand, the differences in the I_T values between using the Summer Quilt 1 and the Summer Quilt 2 ranged from 14% to 18%, due to different thicknesses (weights) of bedding.

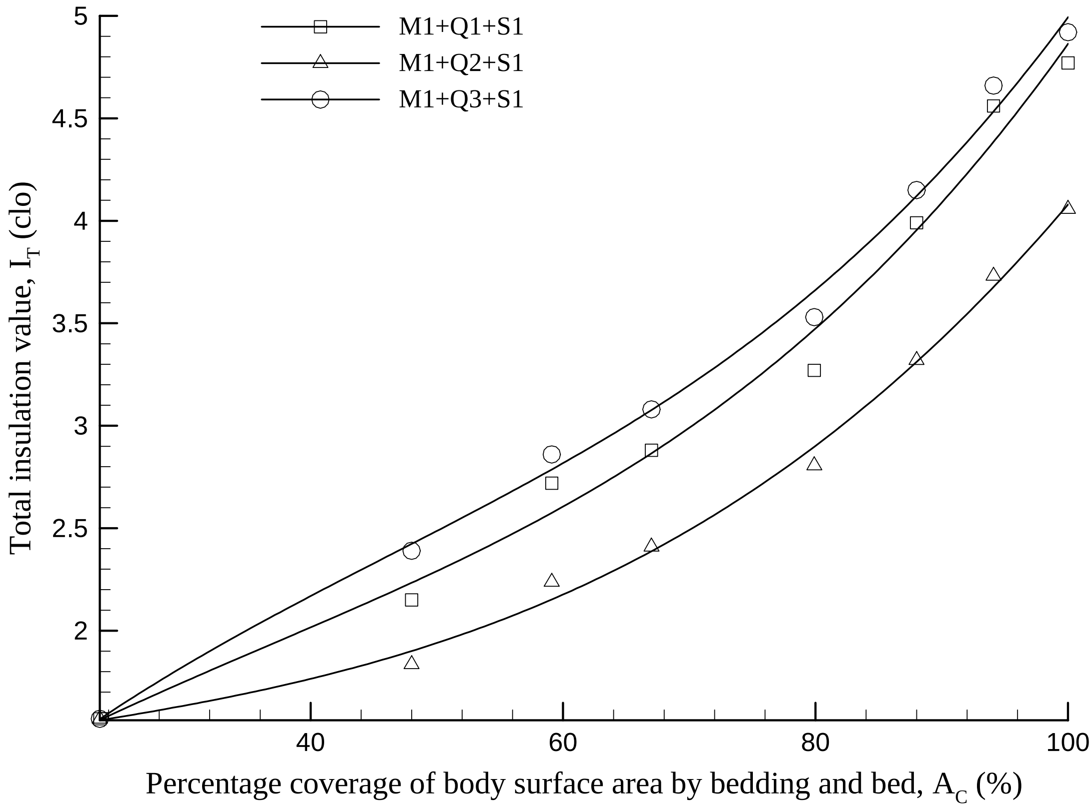

Fig. 5.13 The measured total insulation values when using the three different quilts under the conventional mattress and full-slip sleepwear condition

5.3.5.3 Effect of sleepwear

Fig. 5.14 and Fig. 5.15 illustrate the comparison of the measured total insulation values resulted from different combinations of sleepwear and bedding when using the conventional mattress and *Zongbang* bed, respectively. It can be observed from the figures that for both conditions, the insulation values rose at a higher increasing rate when there was no sleepwear. This was particularly true when A_C was over 60%. It can also be seen from the two figures that the contribution of sleepwear to the total insulation would also depend on A_C and bedding. If A_C was high, the absolute increase in insulation value due to the addition of sleepwear would be smaller than if A_C was low. For example, in using the conventional mattress and the blanket, at 94.1% A_C, when a half-slip sleepwear was added to the nude manikin, I_T value increased from 2.11 (M1+B) to 2.40 clo (M1+B+S2) – a change of 0.29 clo, or 13.7%. For the same conventional mattress and the blanket, however, at 59.1% A_C, when the same sleepwear was added to the nude manikin, I_T value increased from 1.24 to 1.76 clo – a change of 0.52 clo, or 41.9%. On the other hand, if the insulation contributed by bedding was high, the absolute increase in I_T due to the addition of sleepwear would be smaller than that when the insulation contributed by bedding was low. For example, in using the conventional mattress and the Summer Quilt 1, at 94.1% A_C, when a full-slip sleepwear was added to the nude manikin, I_T value increased from 4.03 (M1+Q1) to 4.56 clo (M1+Q1+S1) – a change of 0.53 clo, or 13.2%. However, in using the conventional mattress and the Summer Quilt 2, at 94.1% A_C, when the same sleepwear was added to the nude manikin, I_T value increased from 3.03 (M1+Q2) to 3.73 clo (M1+Q2+S1) – a change of 0.70 clo, or 23.1%.

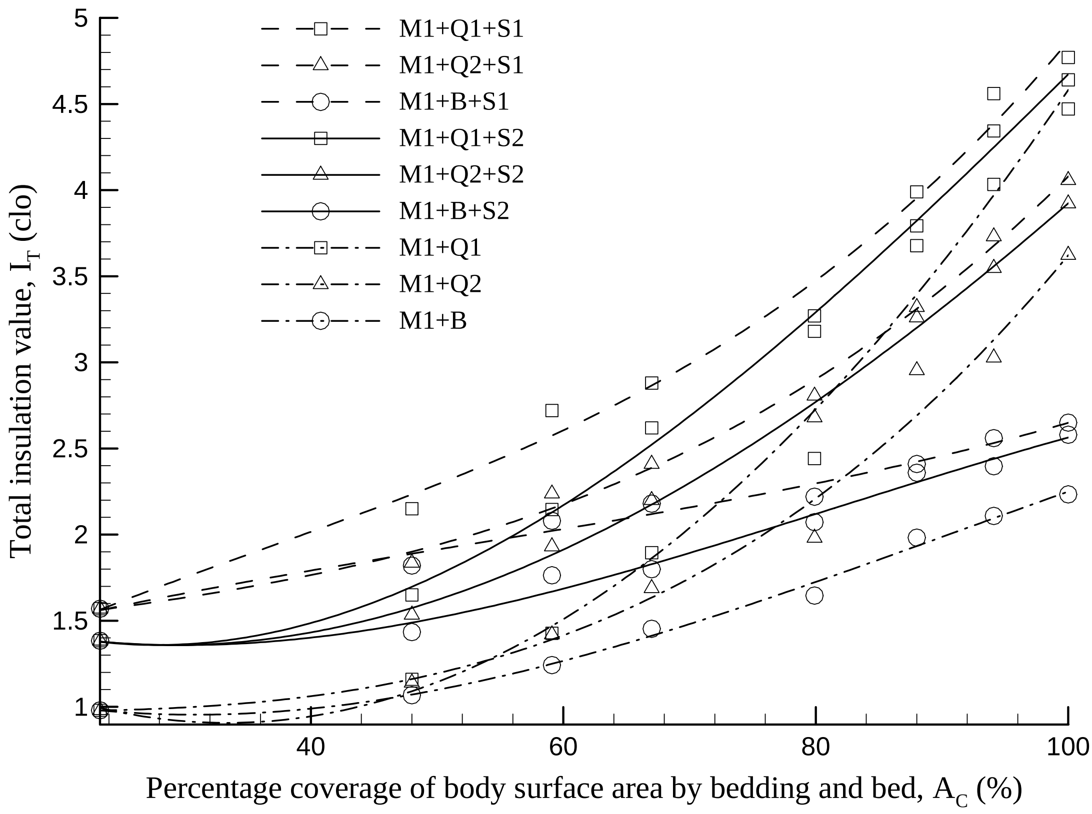

Fig. 5.14 The measured total insulation values provided by different combinations of sleepwear and bedding (conventional mattress condition)

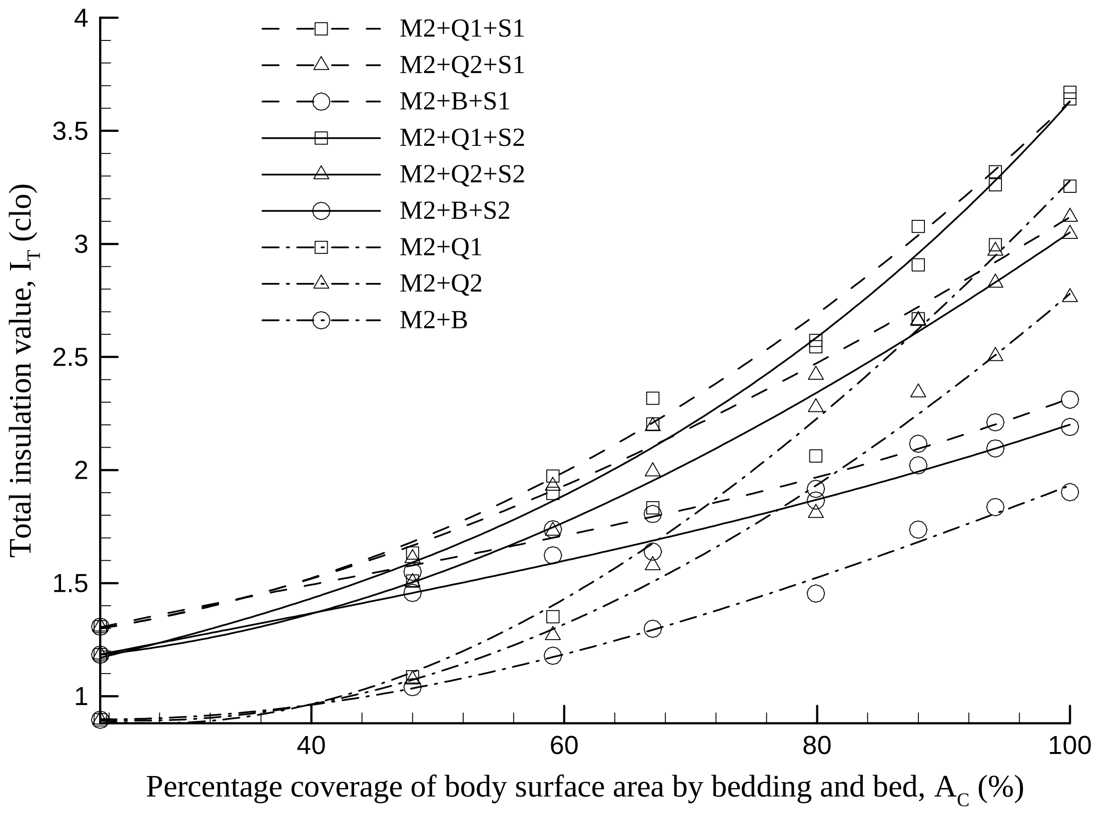

Fig. 5.15 The measured total insulation values provided by different combinations of sleepwear and bedding (*Zongbang* bed condition)

5.3.5.4 Effect of bed and mattress

Fig. 5.16 shows the comparison of the measured total insulation values for both the conventional mattress and *Zongbang* bed under nude, full-slip and half-slip sleepwear conditions when A_C was 23.3% (i.e., without bedding). It can be seen from the figure that the use of *Zongbang* bed may provide less insulation than the use of conventional mattress at all conditions (i.e., nude, full-slip and half-slip sleepwear). The differences in I_T between using the conventional mattress and *Zongbang* bed under nude, full-slip and half-slip conditions were 0.08, 0.19 and 0.13 clo, or 8%, 12% and 10% in terms of percentage differences, respectively.

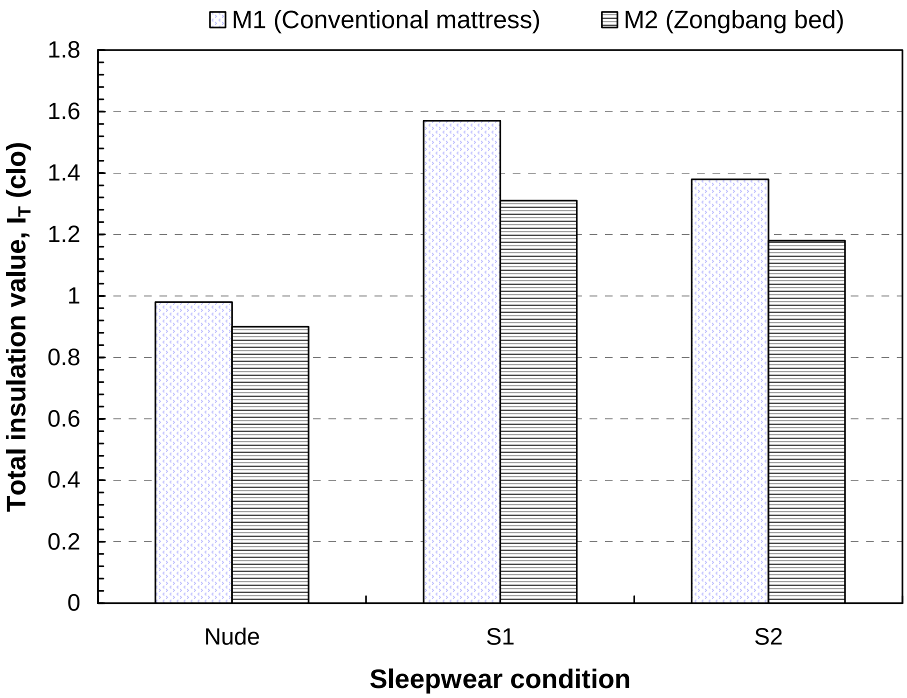

Fig. 5.16 Comparison of the measured total insulation values provided by using the conventional mattress and *Zongbang* bed under different sleepwear conditions (with A_C = 23.3%)

Concerning the situations where bedding were used, it can be seen from the Fig. 5.10, 5.11 and 5.12 that similar to situations for the effects of bedding and sleepwear, the contribution of bed and mattress to I_T would also depend on A_C, bedding and sleepwear. It can be observed from these figures that the differences in I_T between using the conventional mattress and *Zongbang* bed would increase with the increase of A_C. For example, when the Summer Quilt 2 was used, at 94.1% A_C, I_T value decreased from 3.03 clo when using the conventional mattress (M1+Q2), to 2.50 clo when using the *Zongbang* bed (M2+Q2) – a change of 0.53 clo, or 17.5%. For the same Summer Quilt 2, at 48% A_C, however, I_T value decreased from 1.16 clo when using the conventional mattress, to 1.07 clo when using the *Zongbang* bed – a change of 0.09 clo, or 7.8%. On the other hand, the higher the insulation contributed by bedding and sleepwear, the larger the absolute decrease in I_T due to the use of *Zongbang* bed and vice verse. For example, when the full-slip sleepwear and the Summer Quilt 1 were used, at 67.0% A_C, I_T value decreased from 2.88 clo when using the conventional mattress (M1+Q1+S1), to 2.32 clo when using the *Zongbang* bed (M2+Q1+S1) – a change of 0.56 clo, or 19.4%. However, when the half-slip sleepwear and the blanket were used, at 67.0% A_C, I_T value decreased from 1.80 clo when using the conventional mattress (M1+B+S2), to 1.64 clo when using the *Zongbang* bed (M2+B+S2) – a change of only 0.16 clo, or 8.9%.

5.4 Solving Comfort Equation and establishing comfort charts

There are a large number of combinations of variables (i.e., environmental variables such as air temperature, air velocity, and the total resistance value of a bedding system) that may satisfy the thermal Comfort Equation for sleeping environments. A FORTRAN program (details shown in Appendix B) has been developed and run in the platform of Microsoft Fortran PowerStation to solve the Comfort Equation (5.37) for various combinations of variables. Comfort charts (Fig. 5.17 ~ Fig. 5.18) have been established, and can be used for determining thermally neutral environmental conditions under a given bedding system. Furthermore, if and when necessary, both PMV and PPD indexes can also be calculated for the purpose of assessing thermal comfort.

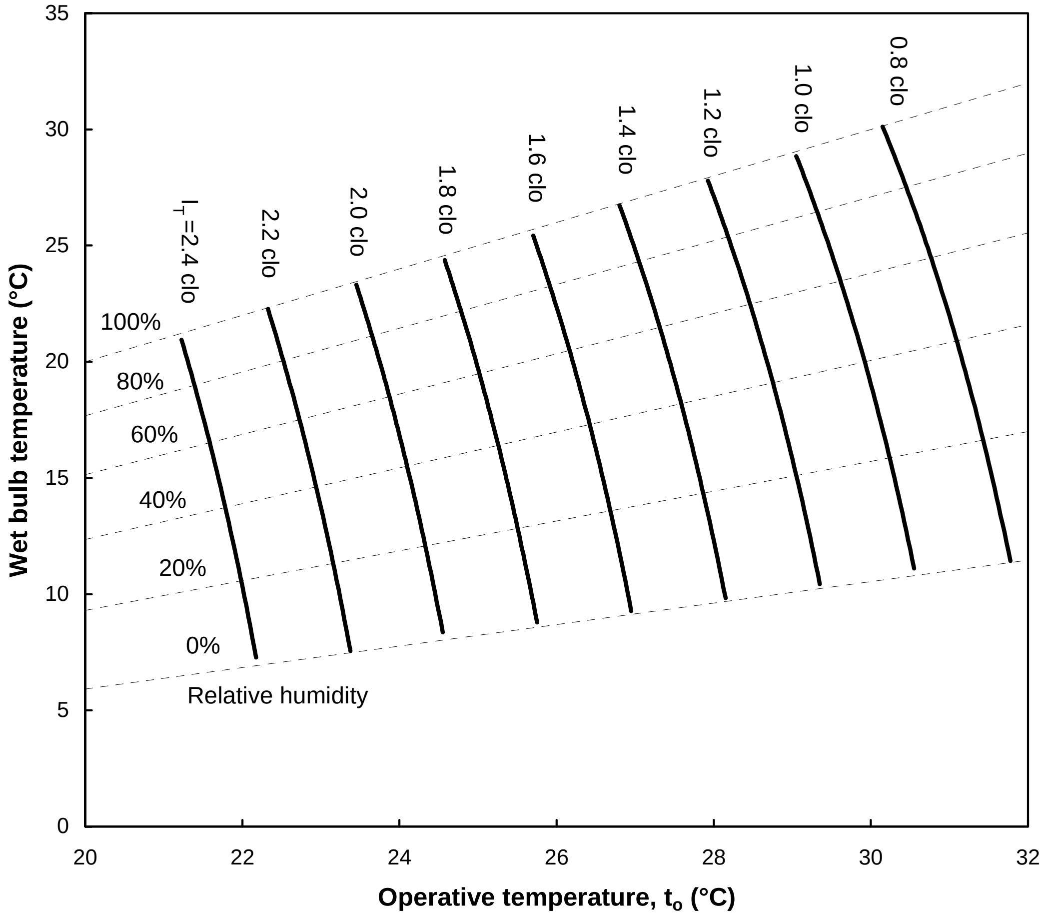

Fig. 5.17 Comfort lines (operative temperature vs. wet bulb temperature) with an air velocity not greater than 0.15 m/s

Fig. 5.17 illustrates the comfort lines showing the combinations of operative temperature, wet bulb temperature and the total insulation value, under which thermal neutrality can be achieved. It can be seen from the figure that the influence of relative humidity on the thermal comfort of a sleeping person is relatively small. A change from absolutely dry air (RH = 0%) to saturated air (RH = 100%) can be compensated by only a 0.95 ~ 1.63 °C (at the range of 2.4 ~ 0.8 clo total insulation values) decrease of operative temperature. The higher the total insulation value, the less the decrease of operative temperature. For example, when the total insulation value is 1.0 clo, a 1.5 °C reduction of operative temperature will adequately compensate the change of relative humidity from 0% to 100%, compared to 1.1 °C reduction of operative temperature at the total insulation of 2.0 clo.

It should be noted that the comfort lines in Fig. 5.17 include the entire humidity range from 0% to 100%. Although there is no reason, from a thermal comfort point of view, to avoid extreme relative humidity, there exist other reasons for avoiding the extremes. For example, it is recommended in ASHRAE Standard 62 [ASHRAE 2001a] that the relative humidity in habitable spaces preferably be maintained between 30% and 60% to minimize the growth of allergenic or pathogenic organisms. Similar to the ASHRAE Standard, ISO Standard 7730 [ISO 1994] also recommends that the relative humidity be kept between 30% and 70%. The limits are set to decrease the risk of unpleasantly wet or dry skin, eye irritation, static electricity, microbial growth and respiratory diseases.

On the other hand, the total insulation value provided by a bedding system would significantly influence the thermal neutral temperature for sleeping persons. This can be more clearly illustrated in Fig. 5.18, which shows the relationships between the thermal neutral temperature and the total thermal insulation of a bedding system under different indoor relative humidity levels. It can be seen in Fig. 5.18, for example, at 50% relative humidity when the total insulation value increases from 1.0 clo to 2.0 clo, the thermal neutral temperature will decrease from 29.2 °C to 23.4 °C. It can also be seen from Fig. 5.18 that a linear relationship between operative temperature and the total insulation value is demonstrated. Fig. 5.18 also provides answers to the question which is related to the thermoneutral temperature during sleep and raised in Chapter 2. In other words, the relationship between the thermoneutral temperature and the total insulation value provided by a bedding system has been established. For example, the thermoneutral temperature would be 29.3 °C at 50% relative humidity for a naked sleeping person with the total insulation value of a bedding system being 0.98 clo. This agrees well with the range of 28 ~ 32 °C for thermoneutral temperature at naked conditions in the various earlier studies related to sleep (see Section 2.3.4.2).

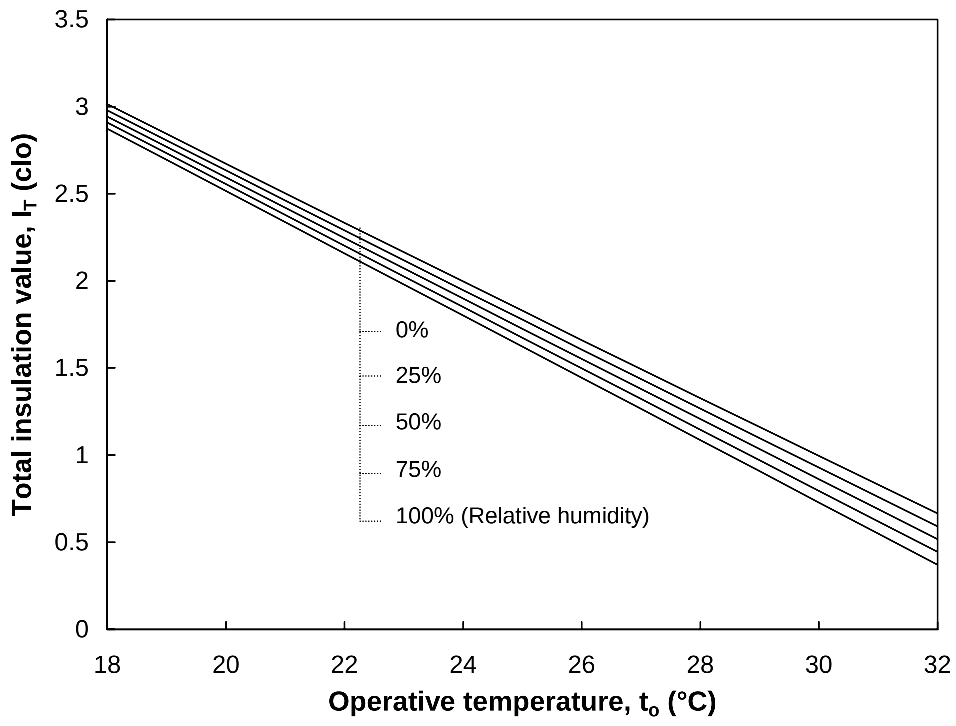

Fig. 5.18 Relationship between operative temperature and the total insulation value with an air velocity not greater than 0.15 m/s

5.5 Discussions

5.5.1 The effect of the total insulation of a bedding system on thermal comfort

By solving the Comfort Equation (5.37), it can be seen that the total thermal insulation value of a bedding system significantly affects the thermal neutral temperature for sleeping environments and is therefore an important variable in the Comfort Equation. The slope of the comfort line at 50% relative humidity in Fig. 5.18 is -0.173 clo/°C, indicating that a decrease of only 0.173 clo in I_T would result in an increase of 1 °C of optimum operative temperature in order to maintain the thermal neutrality for sleeping. Therefore, during sleeping when the metabolic rate is 0.7 met, the effect of changing I_T on the optimum operative temperature is approximately 5.8 °C per clo. This is similar to the effect of changing clothing's insulation on the optimum operative temperature for sedentary activity (6.0 °C per clo).

On the other hand, the results of experiments on measuring the total insulation values provided by different bedding systems have suggested that the total insulation values of the measured bedding systems ranged from 0.90 to 4.89 clo. The lowest, 0.90 clo, occurred at naked condition using the *Zongbang* bed without any bedding, and the highest, 4.89 clo, at full-slip sleepwear condition using the conventional mattress, with the manikin 100% covered with the Multi-purpose quilt, respectively. The thermal neutral temperatures at 50% relative humidity for sleeping environments were 29.8 °C at 0.90 clo and 6.7 °C at 4.89 clo. This suggested that the differences in the total insulation would be very large if different bedding systems were used.

Therefore, the bedding system used by a sleeping person has a substantial impact on thermal comfort.

5.5.2 The factors influencing the total resistance / insulation of a bedding system

There are a number of factors that would influence the total resistance / insulation of a bedding system. The effects of the percentage coverage of body surface area by bedding and bed, and the components of a bedding system such as bedding, sleepwear, bed and mattress on the total resistance / insulation of a bedding system are discussed in Sections 5.3.5.1 to 5.3.5.4. It can also be noted that it is not useful to measure the thermal resistance from an individual component of a bedding system in isolation, as these components will be used together to form a bedding system and the contribution of one component to R_T or I_T will depend on A_C and the presence of other components making up the bedding system.

In addition, there are other factors that would influence the total insulation of a bedding system, such as air velocity, direction of airflow, posture and body figures of occupants, etc.

As the total insulation of a bedding system includes the surface insulation of the air layer, the air velocity, v, not only influences the value of convective heat transfer coefficients, h_c, (illustrated in Equations (5.34) and (5.35)) and, hence the operative temperature, t_o, but also the total insulation of a bedding system through air layer insulation although the effect might be much smaller than that from other factors.

When the air velocity increases, the total insulation value of a bedding system would decrease since the increased air velocity would enhance the convective heat transfer between a covered human body and its environment.

The direction of airflow, whether air flow is parallel to, or perpendicular to the major axis of a human body, may also influence the convective heat exchange between a human body and its environment. A previous experimental study [Colin and Houdas 1967] showed that the differences in the convective heat exchange between the two different directions of airflow could be observed. This is ascribed to the fact that the effective convection area of a human body is smaller when air flow is perpendicular to the major axis of a human body than that when air flow is parallel to the major axis of a human body. Considering the diminution of effective area when air flow direction is perpendicular to the major axis of a human body, Colin and Houdas [1967] obtained different formulas to calculate the convective heat exchange between a human body and its environment.

Therefore, both the air velocity and the direction of airflow have an impact on the total insulation of a bedding system. However, their effects were not considered in this project since in practice the air velocity is limited to not greater than 0.15 m/s for air-conditioned sleeping environments. There are two reasons for this limitation. The first is that a higher velocity may cause draft (a local convective cooling) and local thermal discomfort since people are more thermally sensitive and consequently the risk of local discomfort is higher during sleep with lower metabolic rates and / or with less insulation. The second is that even for daytime applications, the precise relationship between increased air speed and improved comfort is not yet to be established although an elevated air velocity may be used to increase the maximum temperature for acceptable comfort if an affected occupant is able to control the air velocity. However, an occupant is obviously not able to control the air velocity when sleeping.

Posture and body figures of occupants may also influence the total insulation of a bedding system. The actual transport of sensible heat through a bedding system involves conduction, convection, and radiation. With different postures such as supine and side-reclining, the effective conduction, convection and radiation areas of a human body will be different. Individuals with different body shape (thin or fat) may also result in different effective areas for conduction, convection and radiation heat transfer even if they are in same posture.

Furthermore, turning during sleep may also influence the total insulation of a bedding system since it will change the posture and the percentage coverage of the surface area by bedding and bed. In this study, it was assumed that people remained immobile during sleep. Limitations of this study included therefore the use of a rigid manikin to represent all human body types and the use of only one body position (i.e., supine); both could alter body contact with the bed / mattress, and consequently, heat transfer area. A person’s body tissue will “spread out” when reclined, whereas a rigid manikin will not. Consequently, the total resistance of bedding systems as measured by using a manikin is probably on the lower side of actual values when a human subject of the same height and weight as that of the manikin is used.

5.5.3 The usefulness of a summer quilt

A summer quilt, locally so-called air conditioning quilt, is expected to provide a lower thermal insulation than other types of bedding so that a relatively high indoor air temperature could be used in a bedroom. However, the results of the measurement suggested that the differences in the total thermal insulation between using the Summer Quilt 2 and the Multi-purpose Quilt 3 did not appear significantly large at only 2% to 10%. On the other hand, there existed noticeable differences (14% - 18%) in the total thermal insulation between using the Summer Quilt 1 and the Summer Quilt 2 because of their different thicknesses. The total insulation values when using both the Summer Quilt 1 and the Summer Quilt 2 were larger than that using the blanket. This implies that the use of a summer quilt cannot help lower insulation significantly, and therefore, its intended function is limited.

5.5.4 Air conditioning culture

Individuals may adjust their bedding so as to suit individual preferences when sleeping. A person’s dressing / bedding code may further be affected by the local custom and culture in addition to personal preference. However, the culture and habits of dressing codes in Hong Kong have led to the use of a relatively low indoor air temperature for daytime air conditioning. The Cantonese translation of the term of air-conditioning is literally wrong meaning “Cold Air” when air-conditioning was first introduced to Hong Kong. Since then, the notion that an air-conditioned space has to be “cold” has been deeply implanted into the minds of occupants. Even when energy conservation became a design issue due to the energy crisis in 1973, the call for a lower indoor air temperature still prevailed. Dressing / bedding (i.e., suit and mattress) codes with a relatively high clo value (high insulation) in summer has consequently become a culture in offices and residences. This may partly help explain the results of the questionnaire survey, which showed that most of the local people would prefer a relatively low indoor air temperature at below 24 °C, and about 20% of the respondents would prefer even a lower indoor air temperature at below 20 °C. On the other hand, they would wear suits rather than shirts / shorts in offices, and use mattresses, wear sleepwear and cover themselves with quilts or blankets during sleep.

The sensation of a human being is the “trigger” for him / her to change his / her own preference to suit the environment, or to change the environment to suit him / her. If a relatively low indoor air temperature is pre-maintained in a sleeping environment, an occupant will therefore use a bedding system of a higher insulation value to suit the environment. However, this kind of air-conditioning culture contradicts to the principle of energy conservation. From the energy conservation view of point, people should use as less sleepwear and bedding (or cover as less body area by bedding) as possible to lower the total thermal insulation of a bedding system. Consequently, a higher indoor air temperature in bedrooms at night may be maintained for thermal comfort. Therefore, it is necessary to encourage people to change their habits of using sleepwear and bedding (i.e., change the local air-conditioning culture) in order to save energy and their costs for maintaining a suitable sleeping thermal environment could also be reduced considerably.

5.6 Conclusions

One fundamental issue in developing air conditioning technology is the theory on thermal comfort. Currently the theories on thermal comfort in workplaces at daytime are well established. However, research on thermal comfort for sleeping environments at nighttime is limited. A theoretical study on thermal comfort for sleeping environments has been conducted and reported in this chapter. The results of the study may be potentially used as the basis for establishing criteria on thermal comfort for sleeping environments.

A comfort equation applicable to sleeping thermal environments has been derived by introducing appropriate assumptions and modifications to Fanger's comfort equation. The comfort equation for sleeping environments contains five variables: air temperature, mean radiant temperature, water vapor pressure in ambient air, air velocity and the total resistance / insulation provided by a bedding system.

Based on the derived comfort equation for sleeping environments, the relationship between the thermoneutral temperature and the total insulation value provided by a bedding system has been established. The comfort charts with different comfort lines have also been developed, and can be used for determining thermally neutral environmental conditions under a given bedding system. Furthermore, both PMV and PPD indexes can also be calculated for the purpose of assessing thermal comfort, if and when necessary.

The total insulation (clo) values of various bedding systems commonly used in the subtropical Hong Kong have been measured using a thermal manikin in an environmental chamber. The total insulation values of the measured bedding systems varied greatly from 0.90 to 4.89 clo, depending upon bedding, the type of sleepwear, and percentage coverage of body surface area by bedding and bed. A small-scale database of the total insulation values provided by bedding systems commonly used in the subtropics has been developed.

The use of a Chinese traditional style bed - *Zongbang* bed can provide less insulation than the use of the conventional mattress commonly used in Hong Kong. One the other hand, the use of so-called air conditioning quilt (summer quilt) cannot help lower the total insulation significantly.

The effect of changing the total thermal insulation of a bedding system on the optimum operative temperature is approximately 5.8 °C per clo, indicating that the total insulation value provided by a bedding system significantly affects the thermal neutral temperature for sleeping persons.

Locally, the air conditioning culture of having dressing / bedding with a relatively high clo value (higher insulation) in summer, and at the same time, maintaining a relatively low indoor air temperature, should be reviewed and changed. People should use as less sleepwear and bedding (or cover as less body surface area by bedding) as possible to lower the total insulation of a bedding system. This would result in a relatively higher indoor air temperature maintained in bedrooms without losing thermal comfort at night, and consequently, reduced energy use for air conditioning for sleeping environments.

Having established the thermal comfort model for sleeping environments, it is necessary to investigate two other issues closely related to nighttime air conditioning for sleeping environments: nighttime space cooling load characteristics and fresh air requirement. These are presented in Chapter 6 and 7, respectively.

Chapter 6

The Characteristics of Nighttime Bedroom Cooling Load

6.1 Introduction

As mentioned in Section 2.4.1, a number of cooling load calculation methods such as the Transfer Function Method (TFM), the CLTD/SCL/TA method, the TETD/TA method, heat balance method (HB) and the radiant time series method (RTS) have been developed by ASHRAE. CIBSE, on the other hand, recommended the use of a different cooling load calculation method, the admittance method. All of these methods are applicable to non-residential buildings.

Based on the development of the above-mentioned load calculation methods, many computer programs for calculating building thermal load and energy use have been developed. BLAST (Building Loads Analysis and System Thermodynamics) and DOE-2, both developed and released in the late 1970s and the early 1980s as energy and load simulation tools, are two of the most widely used software packages. Both programs had their merits, and shortcomings, their supporters and detractors, and solid user bases both within the USA and outside [EnergyPlus Archive 2003]. However, EnergyPlus, a new generation of building energy simulation computer program, has been developed by the Department of Energy (DOE) in cooperation with the U.S. Army Construction Engineering Research Laboratory, the University of Illinois, Oklahoma State University, the Department of Energy's Lawrence Berkeley National Laboratory, the Florida Solar Energy Center, and GARD Analytics. Although these software packages were predominately developed for nonresidential applications, they are reported applicable to residential buildings for determining cooling load at nighttime. Therefore, EnergyPlus has been used in this project.

Over the last twenty years, studies on the cooling load characteristics and energy use for air conditioning in different type of residential buildings have been reported. Sullivan et al. [1994] investigated the effects of changing window U-value on residential cooling loads for a prototype single-story house in the U.S. using DOE-2.1D. Andersson et al. [1985] investigated the impact of building orientation on residential heating and cooling loads in the U.S. using BLAST. Douglas et al. [1995] developed a mathematical model to predict summer residential cooling loads. Shariah et al. [1997] suggested that wall insulation would cause a negative impact on the average monthly cooling load in Jordan. For residential buildings in subtropical Hong Kong, Bojic et al. [2001, 2002] investigated the influence of thermal insulation of building envelope on the cooling load in a high-rise residential building by employing an in-house simulation package. It was found that providing thermal insulation to external walls of residential buildings in hot climate regions would not lead to a significant cooling load reduction. Lam [2000a] used DOE-2.1E to investigate the thermal and energy performance in a 25-story residential block with eight units per floor, and found that the building envelope load was dominated in residential buildings. This type of cooling load was influenced by orientation, thermal insulation, window area, type of glazing, external shading and color of external surface. However, in these reported studies the cooling load characteristics both in individual bedrooms and at nighttime were not investigated.

It is extremely important to specify a RAC with an appropriate cooling capacity. A RAC having more cooling capacity than necessary will be inefficient and expensive to operate. On the other hand, a RAC having less capacity than required will not perform satisfactorily. There exists also a close relationship between the cooling capacity of a RAC and its dehumidifying ability. In general, the capacity of RAC is determined based on a detailed calculation of cooling load using established methodologies. However, RACs are sized often by rules of thumb that have accumulated over the years [Lao and Deng 2001]. These rules of thumb are based on the application and practice of air conditioning technology at daytime operating period. For example, it was recommended that the cooling load in a room (for both a living room and a bedroom) be estimated simply based on the floor area of the room, the areas of external windows facing different orientations, and the number of occupants [Consumer Council 1988]. Consumers were also recommended by estimating the cooling load in a room based on a suggested W/m^2, taking "room size" as an only affecting factor [Consumer Council 1988]. These would lead to potentially oversizing the capacity of a RAC employed in a bedroom in most cases. In Hong Kong, it is a standard provision for local property developers to supply RACs in high-rise residential blocks and the sizes of the RACs for the bedrooms with the same floor areas are normally the same irrespective of their orientations. A number of questions would thus arise: Is it reasonable to size bedrooms' RACs whose operations are most likely at nighttime, based on the cooling load at day time operating mode and without considering bedrooms' orientation? What is the difference between the cooling load characteristics in bedrooms at nighttime and that at daytime? Unfortunately, there has been little research work reported in open literature to address these questions.

This chapter reports on the characteristics of nighttime bedroom cooling load in the subtropics, using EnergyPlus. The weather conditions of Hong Kong and the typical arrangements of Hong Kong’s high-rise residential blocks were used in this simulation study as an illustrative example for the characteristics of nighttime bedroom cooling load in the subtropics. Firstly, EnergyPlus, the simulation software, is briefly introduced. This is followed by detailed descriptions of both a model building which served as a platform to perform the simulation study and a number of assumptions made in the simulation. Thirdly, the simulation results on the cooling load characteristics in bedrooms under three different operating modes of RACs in the summer design day, the breakdown of the total cooling load in bedrooms at nighttime operating mode, indoor air temperature and mean radiant temperature variations at NOM, and the effects of indoor design air temperature on the cooling load characteristics at NOM are presented. Finally, the differences in the cooling load characteristics among three different operating modes and the issues related to the sizing of RACs are discussed.

6.2 The simulation software - EnegyPlus

EnergyPlus, which (Version 1.0) was first released on April 12 2001, is a building energy simulation program for modeling building heating, cooling, lighting, ventilating, and other energy flows. Based on a user’s description of a building from the perspective of the building’s physical make-up, associated mechanical systems, etc., EnergyPlus calculates the heating and cooling loads necessary to maintain the indoor thermal control setpoints, the conditions throughout a secondary HVAC system, the coil loads, and the energy consumption of primary plant equipment [EnergyPlus Archive 2003].

EnergyPlus has its roots in both BLAST and DOE-2 programs. It builds on the most popular features and capabilities of the BLAST and the DOE-2, but also includes many additional innovative simulation capabilities such as time steps of less than an hour, modular systems and plants integrated with heat balance-based zone simulation, multi-zone air flow, thermal comfort, and photovoltaic systems [Website_3 2003]. EnergyPlus allows users to calculate the impacts of different heating, cooling and ventilating equipment and various types of lighting installations and windows to maximize building energy use efficiency and occupant comfort. Users can simulate the effect of window blinds, electrochromic window glazings, and complex daylighting systems, which are the features not seen in earlier DOE software [Website_4 2003]. The current version of EnergyPlus (Version 1.2.1) provides an integrated, simultaneous solution where building responses and primary and secondary systems are tightly coupled (iteration performed when necessary). It employs the heat balance based solution technique for determining building thermal loads, allowing simultaneous calculations of radiant and convective effects at both in the interior and exterior surfaces during each time step of simulation. It combines heat and mass transfer models that account for moisture adsorption / desorption either as a layer-by-layer integration into the conduction transfer functions or as an effective moisture penetration depth model. Furthermore, it links to other simulation environments such as WINDOW5, COMIS (an air flow model), TRANSYS and SPARK to allow more detailed analysis of other components in building environmental systems.

The source code of EnergyPlus is currently available in public domain and open for public inspection, revision, etc. EnergyPlus has an improved structure to define a well organized modular concept that would facilitate adding features and links to other programs. The key benefit of modularity is that new modules can be developed concurrently without interfering with other modules under development and with only a limited knowledge of the entire program structure. FORTRAN90, which is a modern, modular language with good compilers on many platforms and allows C-like data structure and mixed language modules, has been used for the initial release of EnergyPlus.

However, EnergyPlus is a stand-alone simulation program without a 'user friendly' graphical interface. It reads input and writes output as simple ASCII text files. Furthermore, it is currently not a life cycle cost analysis tool.

6.3 Description of a model building and assumptions used in the simulation

EnergyPlus has been used to investigate the thermal load characteristics in a hypothetic 30-story residential block, which was modeled after those widely used in Hong Kong. A typical floor plan of the residential block under study is shown in Fig. 6.1. In each floor, eight apartments were arranged symmetrically, and there was a common lift lobby / staircase.

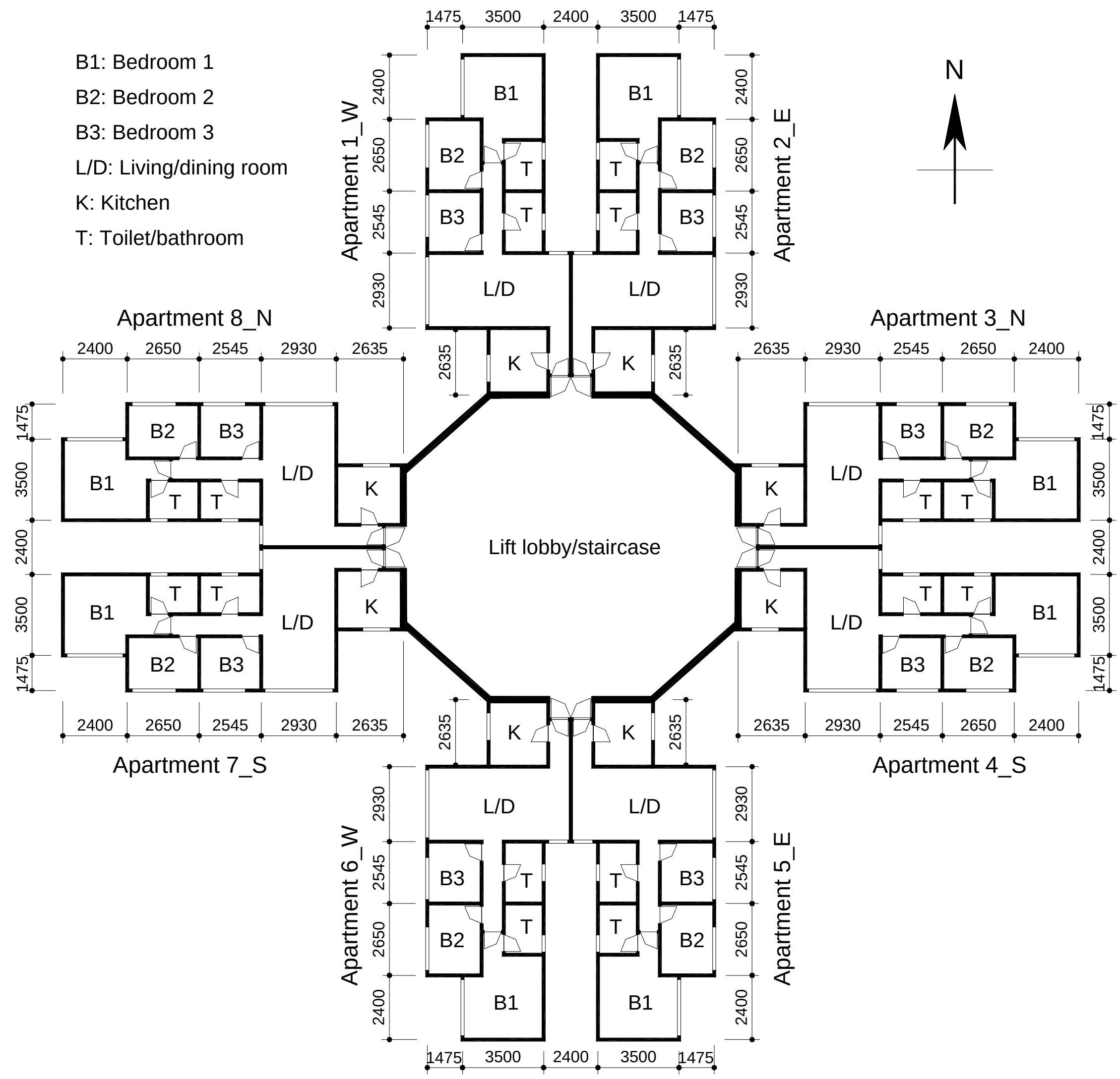

Fig. 6.1 Typical floor plan of a hypothetic high-rise model residential building

6.3.1 Occupancy pattern

In each apartment, there were three bedrooms, one living / dining room, one kitchen and two bathrooms. Each apartment was occupied by a four-person family, which consisted of two working adults and two school children. The master bedroom (i.e., bedroom 1) was occupied by the two working adults, and bedroom 2 and bedroom 3 each by one school child, respectively. Since there existed the partitions (walls and doors) between the bedrooms and the living / dining room, heat and moisture flux would occur in these partitions. Therefore, although in each apartment, the thermal loads in bedrooms were the principal subjects of investigation, the thermal conditions in the living / dining room were also considered.

6.3.2 Building envelope

The construction details of an apartment are the necessary inputs to EnergyPlus. These are detailed in Table 6.1a and Table 6.1b respectively.

Table 6.1a Details of building elements (1)

Building element	Layer 1 (facing outdoors)	Layer 2	Layer 3 (facing indoors)
External wall	13mm Cement / sand plaster	100mm Concrete	13mm Gypsum plaster
Internal partitions	13mm Gypsum plaster	100mm Concrete	13mm Gypsum plaster
Flooring	5 mm Vinyl tiles	25mm Screed	100mm Concrete

Table 6.1b Details of building elements (2)

Building element	Description
Door (internal)	Each composed of two sheets of 6 mm thick plywood, separated by a 38 mm air cavity.
Window	Each composed of one sheet of 5 mm thick glass (i.e., single-layer window), a generic interior window shade and an external attached shading concrete slab of 500 mm wide.
Furniture	Each composed of one sheet of 18 mm thick wood with 6, 3, 3 m^2 of surface area in bedroom 1, 2 and 3, respectively.

Table 6.2 lists the physical properties of materials used in building elements. The hypothetical residential block was not shaded by other buildings in its close vicinity.

Table 6.2 Physical properties of building elements

Materials	Specific heat capacity (J/kg K)	Density (kg/m^3)	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

6.3.3 Operating patterns of air conditioners

In each apartment, each bedroom was equipped with an individual WRAC while the living / dining room was equipped with a SRAC. However, no air-conditioning was provided in the toilet / bathrooms and the kitchen. In order to better understand the cooling load characteristics in bedrooms at nighttime and their differences from that at daytime, simulations were performed under three operating modes of RACs. The first was nighttime operating mode (NOM), i.e., RACs operated daily from 10 p.m. to 7 a.m. in the following day. The second was daytime operating mode (DOM), i.e., RACs operated daily from 8:00 to 18:00. The last was day-and-night operating mode (DNM), i.e., RACs operated 24 hours a day continuously. Although DNM itself was not common for most RACs used in bedrooms in the subtropical regions, its resulted cooling load characteristics in a bedroom could be used as a basis for comparison with those at DOM and NOM. The operating details of RACs under the three operating modes are shown in Table 6.3.

Table 6.3 Operating details of RACs under the three different operating modes

Mode	Time	WRACs in bedrooms	SRAC in living / dinging room
NOM	00:00-7:00	on	off
	7:00-18:00	off	off
	18:00-22:00	off	on
	22:00-00:00	on	off
DOM	00:00-8:00	off	off
	8:00-18:00	on	on
	18:00-00:00	off	off
DNM	00:00-8:00	on	off
	8:00-18:00	on	on
	18:00-00:00	on	off

The operation mode of the RACs (including both WRACs and SRAC) was set at "continuous fan cycling compressor", which meant that the fan in a RAC ran continuously while the compressor on-off cycled to meet space cooling load. This operation mode is commonly used for RACs.

6.3.4 Internal heat gains and ventilation

The activity levels of occupants were assumed to be 66 W/m^2 per person (1.1 met for resting) when they awoke and 40 W/m^2 per person (0.7 met for sleeping) when they slept [ANSI/ASHRAE 2004]. Detailed internal heat gain from lights, electric appliance, and occupants under the three operating modes are summarized in Table 6.4.

Table 6.4 Hourly internal heat gains from lights, electric appliance and people under the different operating modes

Opera -ting mode	Time	Lighting load* (W/m^2)		Electric appliance load* (W/m^2)		Activity level (W/m^2 per person)	
		Bedrooms	Living /dining room	Bedrooms	Living /dining room	Bedrooms	Living /dining room
NOM	00:00-6:00	0	0	0	1.36	40	0
	6:00-7:00	8.165	9.865	23.53	1.36	40	0
	7:00-8:00	0	0	0	1.36	0	66
	8:00-18:00	0	0	0	1.36	0	0
	18:00-20:00	0	19.73	23.53	1.36	0	66
	20:00-21:00	16.33	19.73	23.53	27.22	0	66
	21:00-22:00	16.33	19.73	23.53	1.36	66	66
	22:00-23:00	16.33	0	23.53	1.36	66	0
	23:00-00:00	0	0	0	1.36	40	0
DOM	00:00-6:00	0	0	0	1.36	40	0
	6:00-7:00	8.165	9.865	23.53	1.36	40	0
	7:00-18:00	16.33	19.73	23.53	1.36	66	66
	18:00-23:00	0	0	0	1.36	0	0
	23:00-0:00	0	0	0	1.36	40	0
DNM	00:00-6:00	0	0	0	1.36	40	0
	6:00-7:00	8.165	9.865	23.53	1.36	40	0
	7:00-18:00	16.33	19.73	23.53	1.36	66	66
	18:00-23:00	16.33	0	23.53	1.36	66	0
	23:00-00:00	0	0	0	1.36	40	0

* From Bojic et al. [2002].

The ventilation rate for the bedrooms and the living / dining room was 7.5 L/s per person as required by ASHRAE Standard 62-2001 when the RACs were operated.

However, when the rooms were not air-conditioned, they were naturally ventilated with windows opened, at a ventilation rate of 12 air changes per hour (ACH).

6.3.5 Indoor air design temperature and meteorological data

ASHRAE [ANSI/ASHRAE 2004] recommends a comfort zone based on 90% acceptance of thermal conditions or 10% PPD (Comfort Class B) as shown in Fig. 2.4. The operative temperature is around 25 °C at a relative humidity of 60% in this comfort zone. Since the operative temperature is approximately the mean value of air temperature and mean radiant temperature in most practical cases [ANSI/ASHRAE 2004] and indoor air temperature is normally lower than the mean radiant temperature (this will be detailed in Section 6.4.3), the indoor air temperature should be below 25 °C. Furthermore, in Hong Kong, it is recommended to use 22 °C / 50% as the default indoor conditions for bedrooms in residential flats [HK-BEAM 2003]. The results of the survey (reported in Section 4.3) also showed that the indoor air temperature setting during nighttime generally ranged from 20 to 24 °C (dry-bulb). Therefore, 23°C was selected as the indoor air temperature setpoint required by EnergyPlus. Only indoor air temperature was controlled by the thermostat in a RAC, with indoor relative humidity left uncontrolled.

Using the outdoor cooling design conditions corresponding to annual percentile value of 0.4, the maximum dry-bulb (DB) temperature, the mean wet-bulb (MWB) temperature and the daily temperature range for the summer design day for Hong Kong were 33.2 °C, 26.1 °C, and 4.5 °C, respectively [ASHRAE 2001]. EnergyPlus sinusoidally distributed the 4.5 °C range over 24 hours in the design day, as shown in Fig. 6.2.

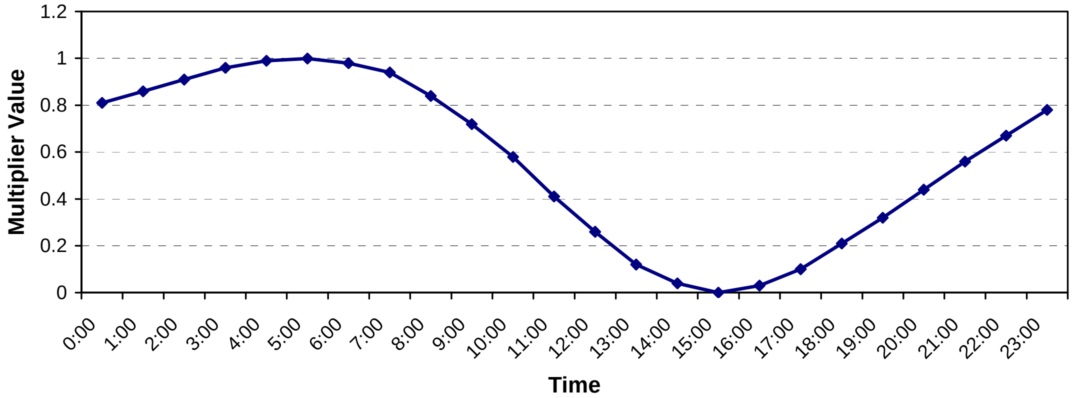

Fig. 6.2 Daily range multiplier for the design day

Outdoor air temperature at any hour in the design day may then be determined by the following equation:

$$t_{current} = t_{Max} - t_{range} \cdot t_{multiplier} \tag{6.1}$$

In addition to the parameters for the summer design day, a weather database has been compiled using the measured hourly data for dry bulb air temperature, dew point temperature, solar radiation, wind speed and direction, cloud cover, etc., for the year 1989, which was considered as the typical reference year (TRY) for Hong Kong.

6.4 Simulation results

EnergyPlus has been used to perform the computations of cooling loads and the corresponding energy requirements for bedrooms at a time step of 10 minutes in the summer design day and the period from 1 May to 31 October to investigate the space cooling load characteristics and the energy use for air conditioning under the three different operating modes. In addition, the cooling load characteristics in the "summer design day" under the three different operating modes were also investigated for sizing RACs. The solution algorithm was based on the moisture transfer function (MTF) module, taking into account moisture storage and diffusion in the construction elements and therefore was a simultaneous heat and mass transfer solution with vapor adsorption taking place within the building constructions.

6.4.1 Cooling load characteristics under the three different operating modes at the summer design day

6.4.1.1 Cooling load characteristics at NOM

The cooling load characteristics in each bedroom in each apartment facing four different orientations at the summer design day at NOM were studied. Table 6.5 shows the peak sensible and the peak total cooling load (normalized by per unit floor area, i.e., W/m^2) for each bedroom, all occurring at 10:00 pm when the WRAC in each bedroom started operation. This is different from a daytime air conditioning process where the total cooling load usually peaks in the afternoon when both outdoor air temperature and solar radiation reach their maximum. However, in a nighttime air conditioning process, the outdoor air temperature gradually goes down, and there is no solar heat gain, but with substantial heat stored in a space's thermal mass. Therefore, at the beginning of a nighttime air conditioning process, the amount of heat and moisture to be removed in a space is at its maximum value.

Table 6.5 Peak sensible and peak total cooling load (W/m^2) for each bedroom at NOM

Appart-ment No.	Orienta-tion	Bedroom 1		Bedroom 2		Bedroom 3	
		Sensible	Total	Sensible	Total	Sensible	Total
3 and 8	N	172	241	159	203	159	204
2 and 5	E	**184**	**256**	141	181	157	202
4 and 7	S	154	218	131	169	147	189
1 and 6	W	159	224	**167**	**212**	**189**	**240**

It can be seen that both the highest peak sensible and the highest peak total cooling load for all bedroom 1s occurred in the two facing east, and for all bedroom 2s and 3s, in the two facing west. For bedroom 3, there was only one external wall, as compared to bedroom 1 and 2, where there were more than one external wall facing different orientations. To simplify the discussion while maintaining essential representation, the detailed cooling load characteristics for the bedroom 3s were investigated. Furthermore, load characteristics for the bedroom 3s facing west (i.e., in the apartment 1 and 6) were taken as the basis for comparison, because the bedroom 3s facing west were subjected to the highest peak total cooling load.

Fig. 6.3 shows the hourly profiles of the total cooling load for all bedroom 3s facing four orientations in the summer design day at NOM. It can be observed that all the total cooling loads peaked at 22:00 (i.e., the starting time of WRACs' operation), decreased rather quickly over the next two hours (i.e., between 22:00 and 0:00), and then decreased relatively slowly between 0:00 and 6:00. After 6:00, they increased due to the resumption of internal space heat gain from lights and electric appliances and outdoor air temperature going up slightly from its lowest value at that time. It can also be observed that the percentage differences between the hourly total cooling loads for the bedroom 3s facing west and those for bedroom 3s facing other orientations within the operating period ranged from 6.6% to 25.3%. As shown in Table 6.5, the peak total cooling loads for the bedroom 3s facing east, north, and south were 16%, 15% and 22%, respectively, less than that for the bedroom 3s facing west.

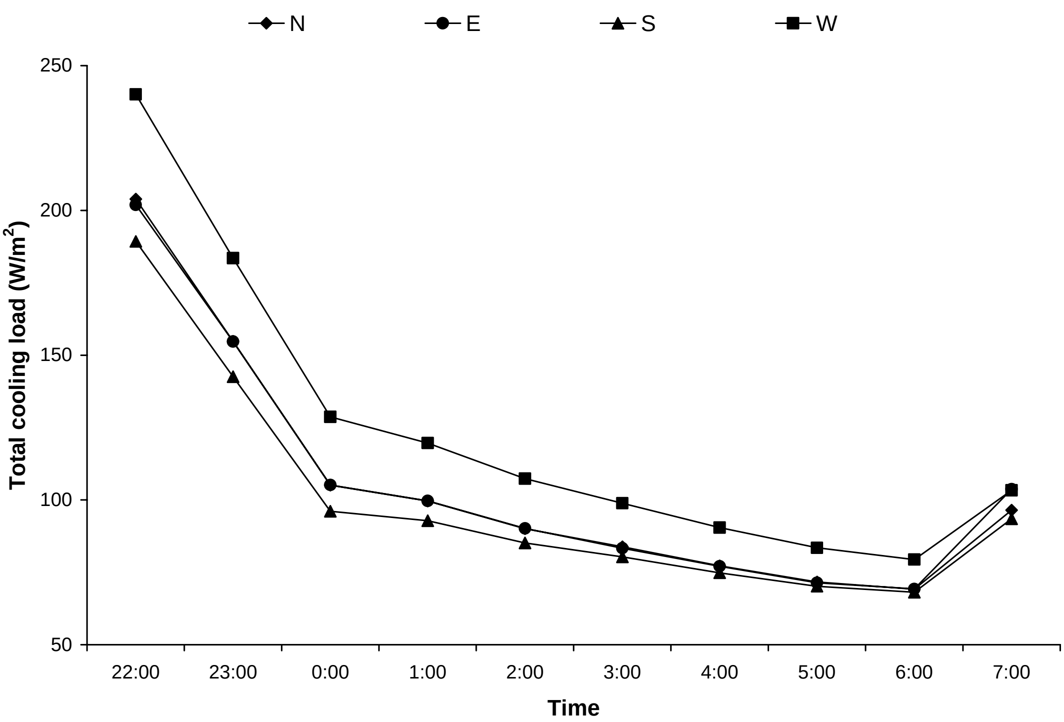

Fig. 6.3 Hourly profiles of the total cooling load for the bedroom 3s facing four different orientations in the summer design day at NOM

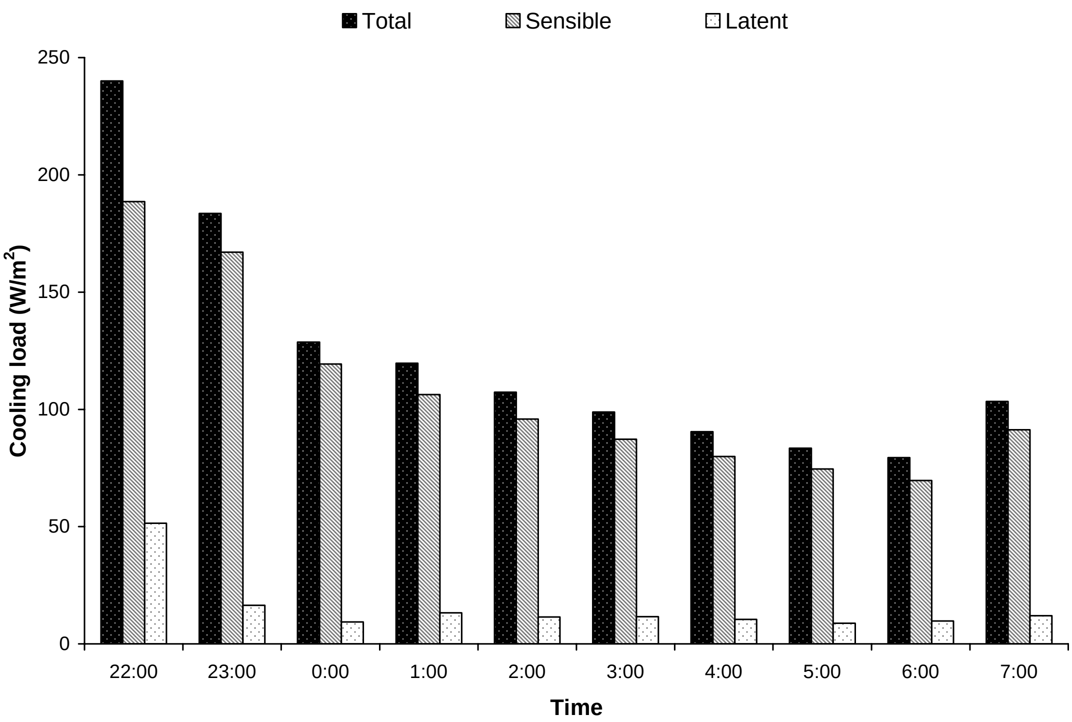

Fig. 6.4 Hourly profiles of cooling loads for the bedroom 3s facing west in the summer design day at NOM

Fig. 6.4 shows the hourly profiles of the total cooling load, sensible cooling load and latent cooling load for the bedroom 3s facing west in the summer design day at NOM. It can be seen that the variation pattern of sensible cooling load was similar to that of the total cooling load. However, latent cooling load also peaked at 22:00 when the WRAC starts operation, and decreased significantly in the next hour and then stayed relatively steady between 0:00 and 7:00.

6.4.1.2 Cooling load characteristics at DOM

Fig. 6.5 shows the hourly profiles of the total cooling load for all bedroom 3s facing four orientations in the summer design day at DOM. It can be seen that the orientation influenced on the total cooling load more at DOM than at NOM. The total cooling load for the bedroom 3s facing east decreased gradually from 8:00 to 18:00. However, the total cooling load for the bedroom 3s facing west decreased during the first hour, increased slightly from 9:00 to 14:00 and then significantly after 14:00 and leveled at 17:00. Those for the bedroom 3s facing north and south stayed relatively constant during the same operating period. It can be observed that the percentage differences between the hourly total cooling loads for the bedroom 3s facing west and those for bedroom 3s facing other orientations ranged from -57% to 44% during the daytime operating period.

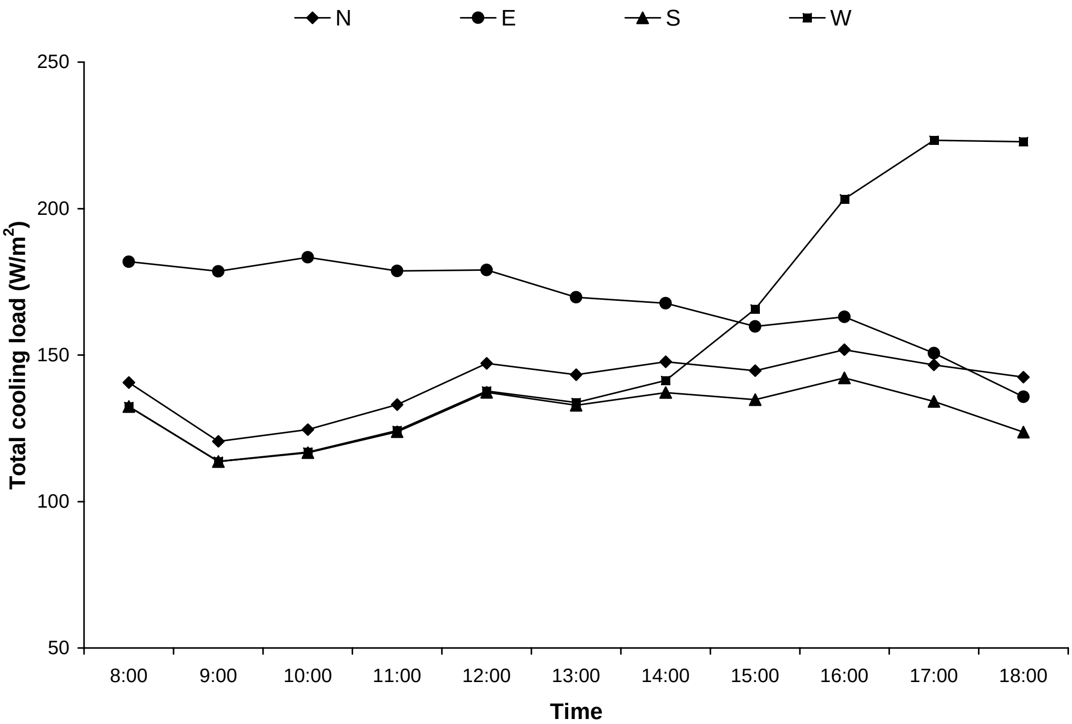

Fig. 6.5 Hourly profiles of the total cooling load for the bedroom 3s facing four different orientations in the summer design day at DOM

The peak total cooling loads for the bedroom 3s facing west, east, north and south were 223 W/m^2, 183 W/m^2, 152 W/m^2 and 142 W/m^2, respectively. The peak total cooling load for the bedroom 3s facing east, north, and south were 15%, 32% and 36%, respectively, less than that for the bedroom 3s facing west. Unlike the peak total cooling load at NOM for all bedroom 3s occurring at the starting time of WRACs' operation, the peak total cooling load at DOM for the bedroom 3s facing north, east, south and west occurred at 16:00, 10:00, 16:00 and 17:00, respectively.

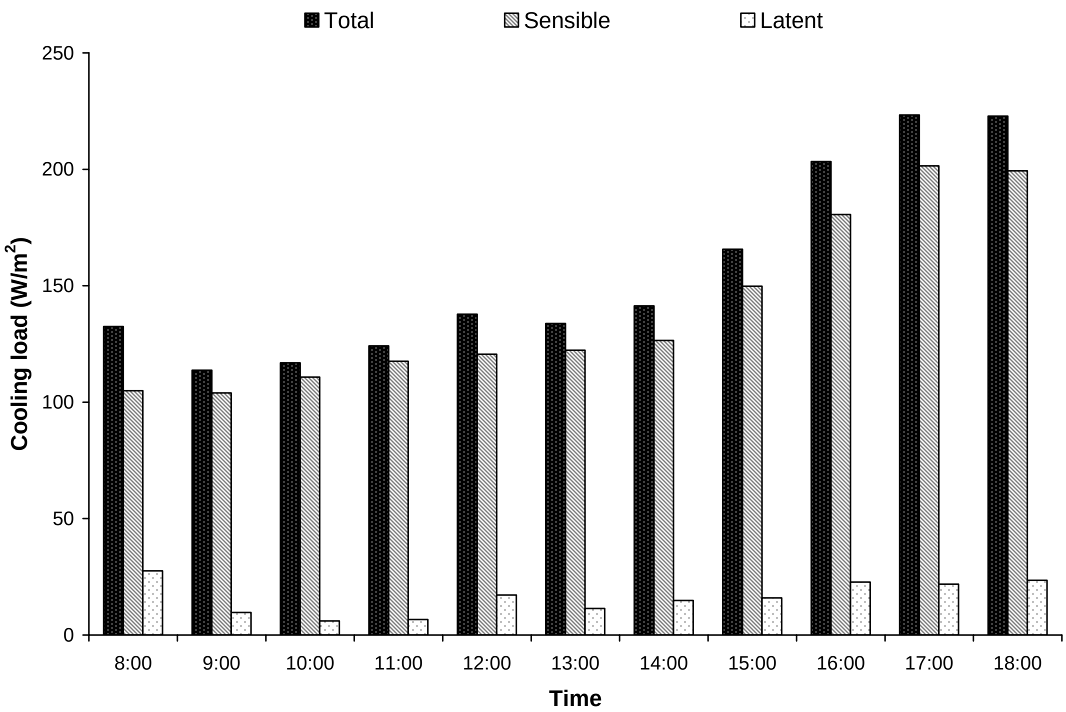

Fig. 6.6 Hourly profiles of cooling loads for the bedroom 3s facing west in the summer design day at DOM

Fig. 6.6 shows the hourly profiles of the total cooling load, sensible cooling load and latent cooling load for the bedroom 3s facing west in the summer design day at DOM. It can be seen that sensible cooling load dominated the total cooling load, with its variation pattern being similar to that of the total cooling load. Latent cooling load peaked at 8:00 when the WRAC started operation, stayed steady at a relatively low level between 9:00 and 15:00, and then slightly increased after 15:00 and leveled between 16:00 and 18:00.

6.4.1.3 Cooling load characteristics at DNM

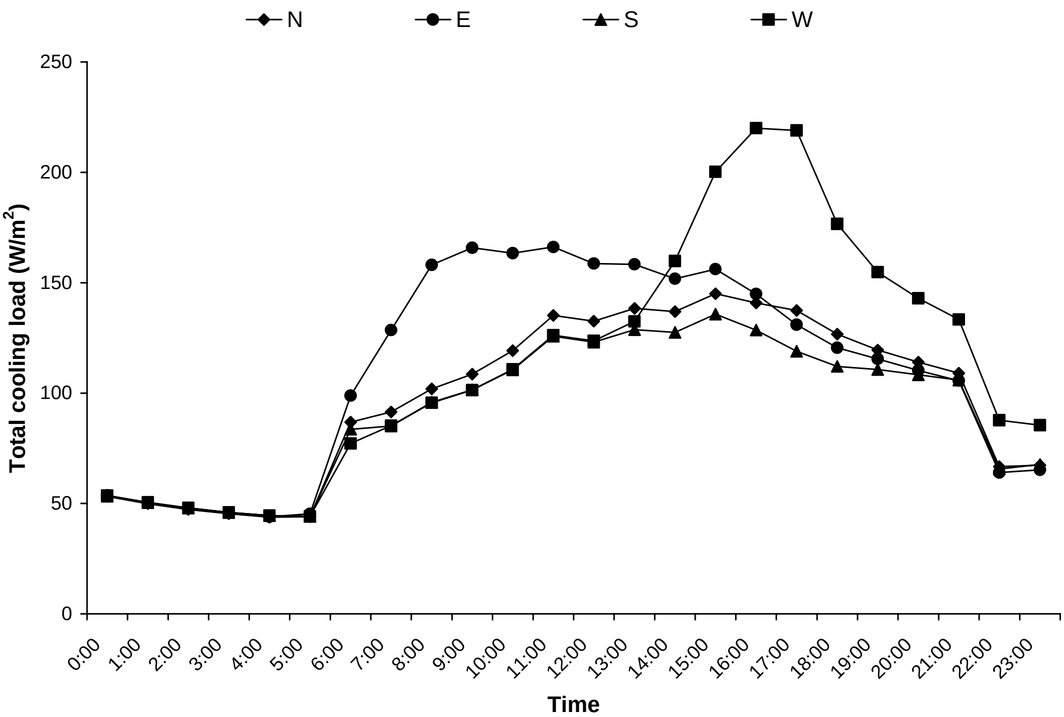

Fig. 6.7 Hourly profiles of the total cooling load for the bedroom 3s facing four different orientations in the summer design day at DNM

Fig. 6.7 shows the hourly profiles of the total cooling load for all bedroom 3s in the summer design day at DNM. It can be observed that between 00:00 to 6:00, the curves of the total cooling load for all bedroom 3s overlapped one another. Starting from 7:00, these curves started to deviate from one another. The peak total cooling loads for the bedroom 3s facing west, east, north and south were 220 W/m^2, 166 W/m^2, 145 W/m^2, 136 W/m^2, occurring at 16:00, 11:00, 15:00 and 15:00, respectively. The peak total cooling load for the bedroom 3s facing east, north and south were 25%, 34% and 38%, respectively, less than that for the bedroom 3s facing west. It can also be observed that the percentage differences between the hourly total cooling loads for the bedroom 3s facing west and those for bedroom 3s facing other orientations within the 24 hour operating period ranged from -65% to 45% at DNM.

Fig. 6.8 shows the hourly profiles of the total cooling load, sensible cooling load and latent cooling load for the bedroom 3s facing west in the summer design day at DNM. Similar to the case at DOM, at DNM, sensible cooling load dominated the total cooling load, with a similar variation pattern to that of the total cooling load.

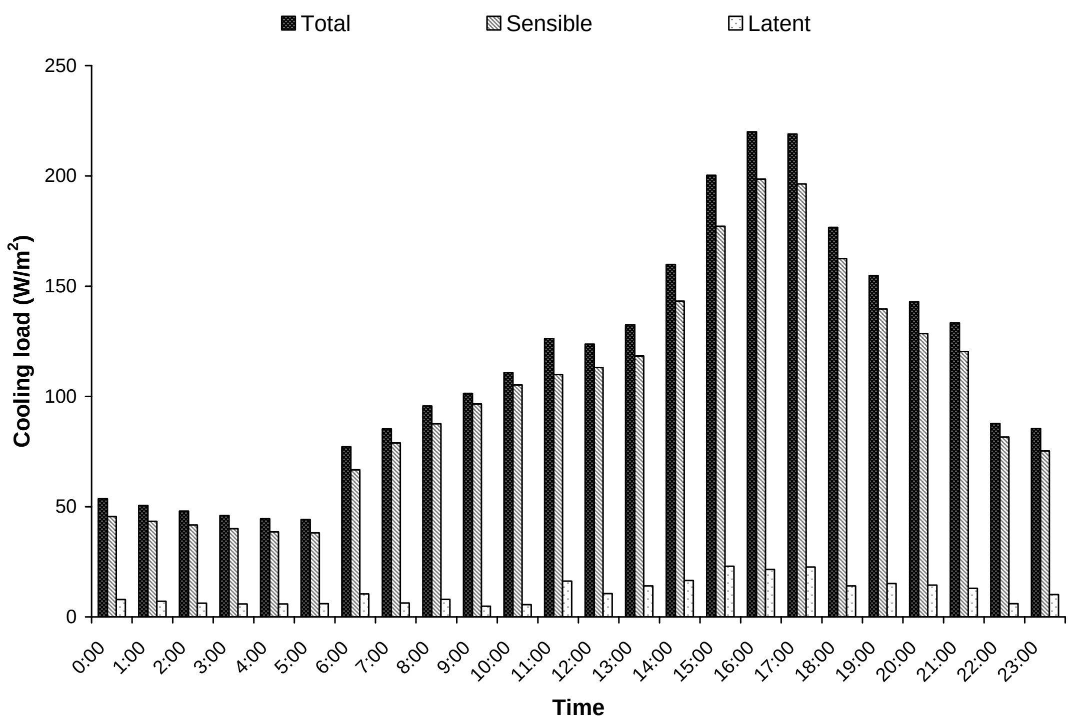

Fig. 6.8 Hourly profiles of cooling loads for the bedroom 3s facing west in the summer design day at DNM

6.4.2 Breakdown of the total cooling load in bedrooms at NOM

Since the peak total cooling load in the bedroom 3s facing west was the highest, its detailed characteristics in the summer design day were representative and therefore used here as an example to illustrate the cooling load breakdown in bedrooms at NOM. On the other hand, considering that there was only one occupant in bedroom 3s, the cooling load characteristics in the summer design day in bedroom 1s facing east whose peak total cooling load was the highest with two occupants were also studied, so that the effect of number of occupants on the total cooling load characteristics may be accounted for.

Fig. 6.9a and Fig. 6.9b illustrate the breakdown of the total cooling load for the bedroom 3s facing west and the bedroom 1s facing east (i.e., in the apartment 2 and 5) at 22:00 (i.e., the starting time of WRAC's operation) when the total cooling load peaked at NOM. It can be seen that the percentage breakdown of the total cooling load at that time for bedroom 3s was similar to that for bedroom 1s. For both bedroom 1s and 3s, the building envelope load dominated, followed by electric appliance load and the lighting load, as the second and third largest component, respectively.

However, the percentage breakdown of the sum of the hourly total cooling load for the whole duration of NOM (i.e., from 22:00 to 7:00), shown in Fig. 6.10a and Fig. 6.10b, were different from those of the peak total cooling load at the starting time of 22:00. Although the building envelope load still dominated, the ventilation load became the second largest component, followed by occupancy load, electric appliance load, and lighting load, respectively.

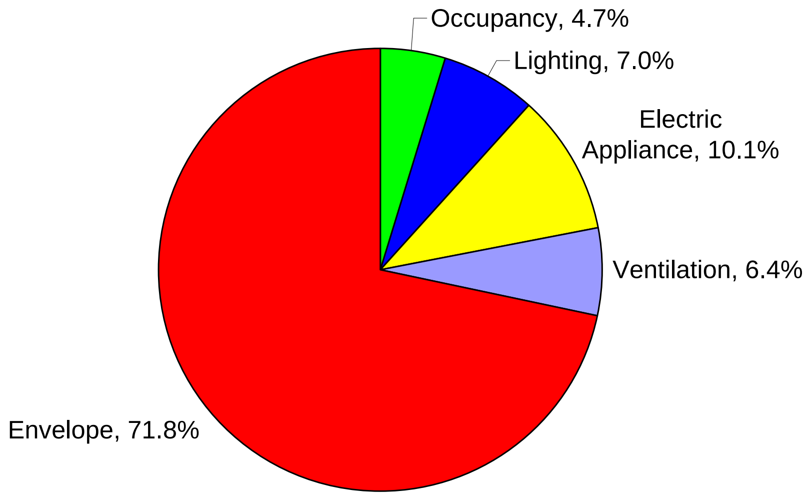

Fig. 6.9a Percentage breakdown of the total cooling load for the bedroom 3s facing west at the starting time of WRAC's operation (22:00)

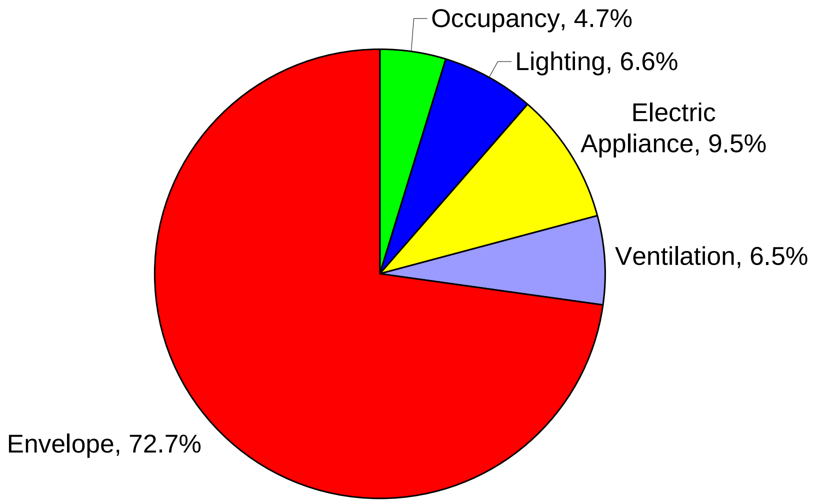

Fig. 6.9b Percentage breakdown of the total cooling load for the bedroom 1s facing east at the starting time of WRAC's operation (22:00)

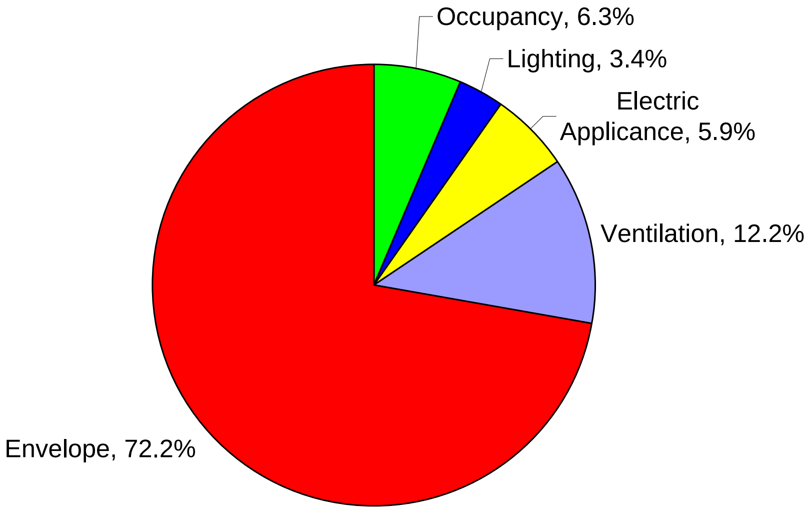

Fig. 6.10a Percentage breakdown of the sum of the hourly total cooling load for the bedroom 3s facing west for the whole duration of NOM (from 22:00 to 7:00)

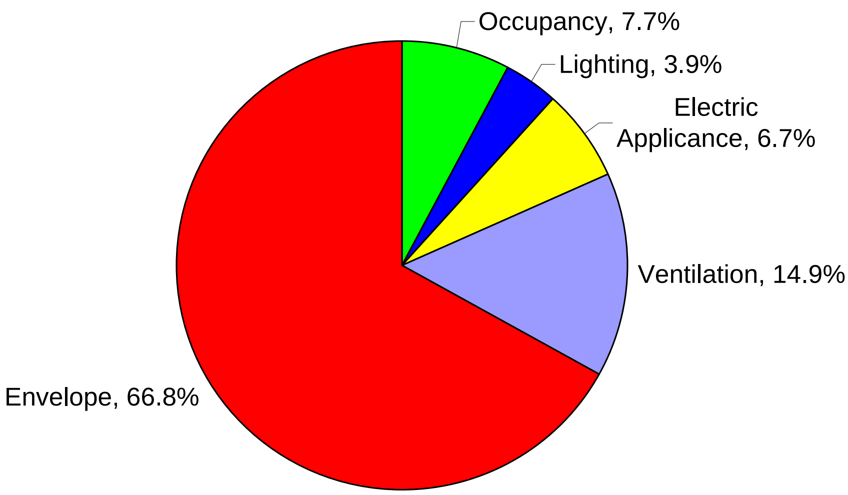

Fig. 6.10b Percentage breakdown of the sum of the hourly total cooling load for the bedroom 1s facing east for the whole duration of NOM (from 22:00 to 7:00)

Comparing the breakdowns of the total cooling load for bedroom 1s to those for bedroom 3s, both ventilation load and occupancy load for bedroom 1s were higher than those for bedroom 3s. This was obvious since there were more occupants in bedroom 1s (double occupancy) than in bedroom 3s (single occupancy), requiring a higher outdoor ventilation rate.

6.4.3 Indoor air temperature and mean radiant temperature variation at NOM

A detailed analysis of the weather data of TRY for Hong Kong using EnergyPlus revealed that the extreme summer week period (a parameter defined within EnergyPlus) when outdoor air temperature was nearest the annual maximum temperature occurred between 1 July and 7 July, with the maximum temperature of 33.7 °C and a variation range of 3.94 °C. Because the operative temperature in bedrooms determined by indoor dry-bulb air temperature and mean radiant temperature at nighttime are useful for discussion on thermal comfort, the variation of indoor air temperature and mean radiant temperature in the bedroom 3s facing west during this extreme summer week have been investigated and included here as an example to illustrate the variations of indoor air temperature and mean radiant temperature at NOM.

Fig. 6.11 shows the variation curves of indoor air temperature, mean radiant temperature and outdoor air temperature for the bedroom 3s facing west, from 0:00 on 2 July to 24:00 on 4 July (three days' data presented here as an example) at NOM. It can be observed from this diagram that similar to the changes of outdoor air temperature, indoor air temperature and mean radiant temperature cyclically varied. When the WRAC was switched off at 8:00 on 2 July, the heat gains resulted from infiltration and temperature difference between indoor air and outdoor air would cause a sudden increase in indoor air temperature. Between 8:00 and 16:00 when the WRAC was not operated, the indoor air temperature rose further due to the heat gains from natural ventilation, solar radiation and the indoor and outdoor air temperature difference. After 16:00, the outdoor air temperature began to drop gradually. However, owing to the thermal storage effect of building envelope and internal furniture, the heat stored in these thermal masses released to indoors and therefore caused the indoor air temperature to rise continuously, peaking at around 18:00. Indoor air temperature then dropped slightly due to the smaller difference between indoor and outdoor air temperature and reduced solar radiation intensity. At 22:00 when the WRAC was switched on, the supply of cold and dehumidified air from the WRAC caused a sudden drop in space air temperature. During the period of WRAC's operation (i.e., from 22:00 to 7:00), the indoor air temperature stayed steady at 23 °C, the setpoint of the WRAC's thermostat. These completed a daily variation cycle.

It can also be seen from Fig. 6.11 that during the period of WRAC's operation from 22:00 to 7:00, the mean radiant temperature dropped gradually from 30.6 °C to 26.3 °C, but was always higher than indoor air temperature. This may affect the value of the operative temperature and therefore influence the thermal comfort of occupants.

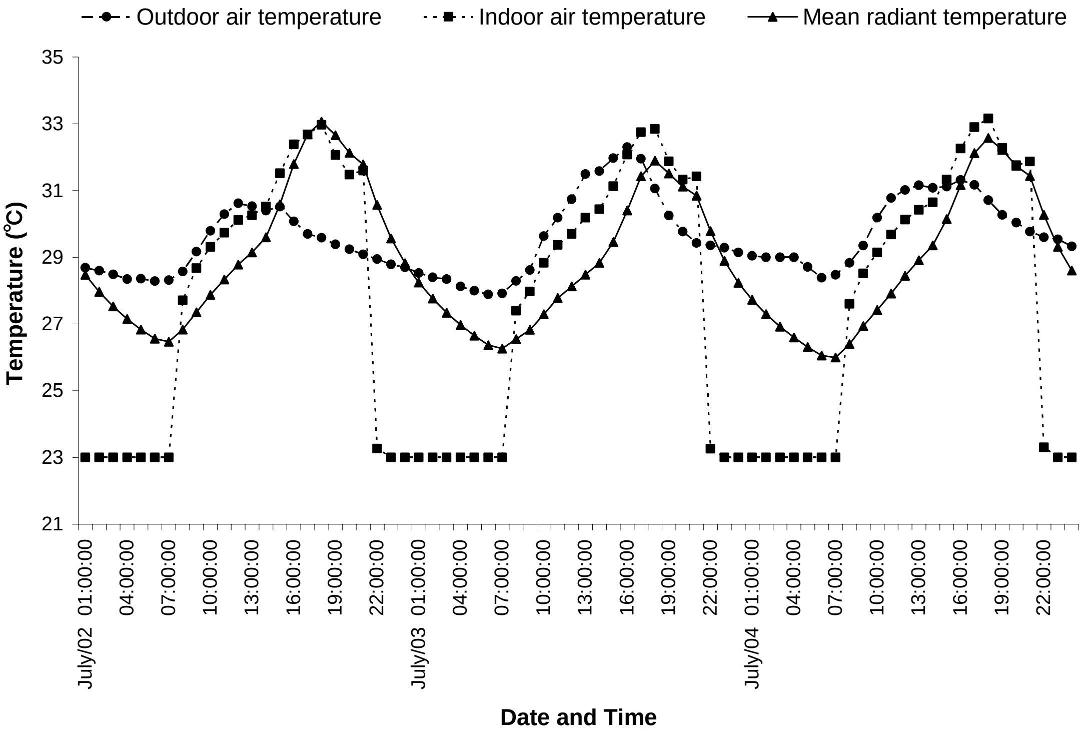

Fig. 6.11 Temperature variations (from 0:00 2 July to 24:00 4 July) at NOM

6.4.4 The effect of indoor design air temperature on the cooling load characteristics and the yearly total cooling energy for a bedroom at NOM

Different indoor design air temperature would result in different space cooling load characteristics and the yearly total cooling energy (an output variable from EnergyPlus). It is therefore necessary to evaluate the effect of indoor design air temperature on the resulted cooling load characteristics and consequently the required cooling equipment capacity for a bedroom at NOM.

For the purpose of clear illustrations, the following parameters were defined:

1) The yearly total cooling energy ($TCE_{j,t}$) for bedroom j at an indoor design air temperature t, which is the sum of the hourly total cooling energy for a whole year, is calculated by:

$$TCE_{j,t} = \sum_{i=1}^{1840} CLQ_{i,j,t} \times 1 \tag{6.2}$$

The yearly total cooling energy is actually an integral of the hourly total cooling load over the whole period of RAC's operating (i.e., 1840 hours) for one year. It may then determine the yearly total electricity consumption for bedroom air conditioning and the yearly CO_2 production due to the generation of this amount of electricity in a power plant. Therefore, the yearly total cooling energy can also be used, to a certain extent, to determine the environmental performance of bedroom air conditioning.

2) Rate of variation in the yearly total cooling energy ($TRV_{j,t}$) for bedroom j at an indoor design air temperature t, which is defined by:

$$TRV_{j,t} = \frac{TCE_{j,t} - TCE_{j,t+1}}{TCE_{j,t}} \times 100 \tag{6.3}$$

3) The maximum total cooling load ($MCLQ_{j,t}$) for bedroom j at an indoor design air temperature t, which is the maximum value of hourly total cooling load in the summer design day, is calculated by:

$$MCLQ_{j,t} = MAX\left(CLQ_{i',j,t}\right) \tag{6.4}$$

In addition to the yearly total cooling energy, this parameter is also essential to evaluate the environmental performance for bedroom air conditioning, as it would determine the sizes of not only RACs but also a power plant that generates the amount of electricity consumed by the RACs.

4) Rate of variation in the maximum total cooling load ($MRV_{j,t}$) for bedroom j at an indoor design air temperature t, which is defined by:

$$MRV_{j,t} = \frac{MCLQ_{j,t} - MCLQ_{j,t+1}}{MCLQ_{j,t}} \times 100 \tag{6.5}$$

Fig. 6.12 shows the yearly total cooling energy and their corresponding rates of variation for the three bedrooms facing west (in the apartment 1 and 6) at different indoor design air temperatures at NOM. It can be seen that all the yearly total cooling energy for the three bedrooms decreased with the increase of indoor design air temperature. Near-linear relationships between the yearly total cooling energy and indoor design air temperature were demonstrated, in particular, at an indoor design air temperature range of 21 °C to 25 °C. Within the temperature range of 21 °C to 26 °C, the rates of variation in the yearly total cooling energy were 14.3% to 28.4% for bedroom 1s, 14.0% to 23.3% for bedroom 2s, and 13.2% to 22.4% for bedroom 3s, respectively. This indicated that indoor design air temperature significantly affected the yearly total cooling energy. The higher the indoor design air temperature was, the less the yearly total cooling energy and the less the energy consumption for the WRACs.

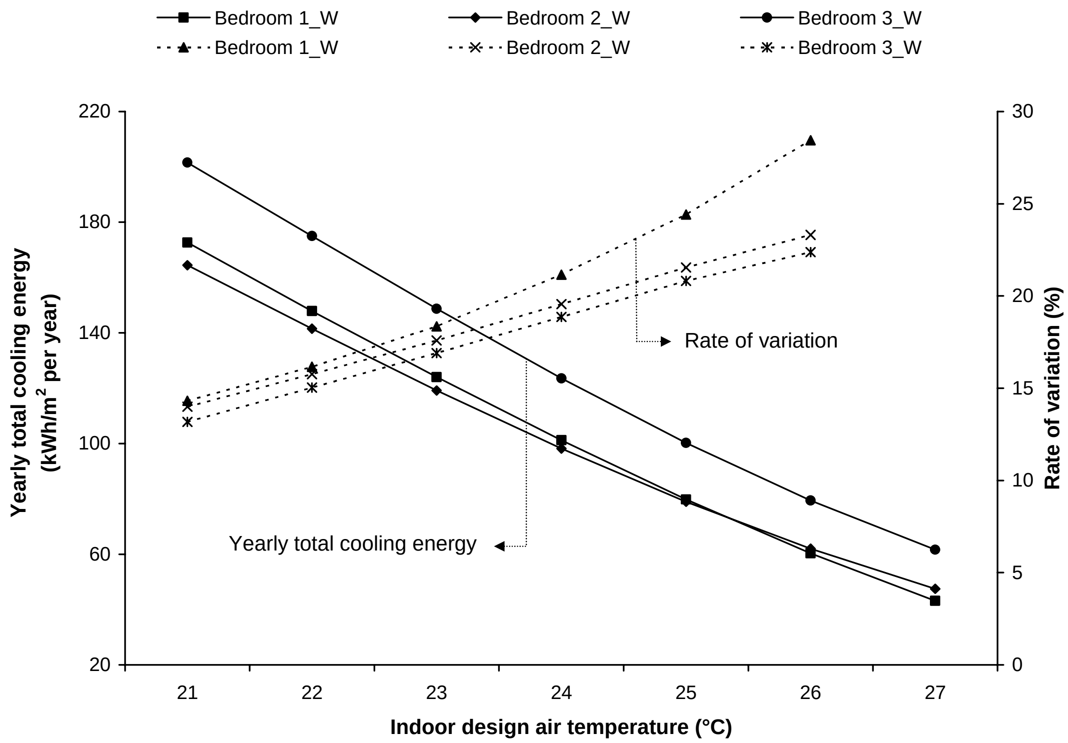

Fig. 6.12 The yearly total cooling energy and their rates of variation at different indoor design air temperatures at NOM

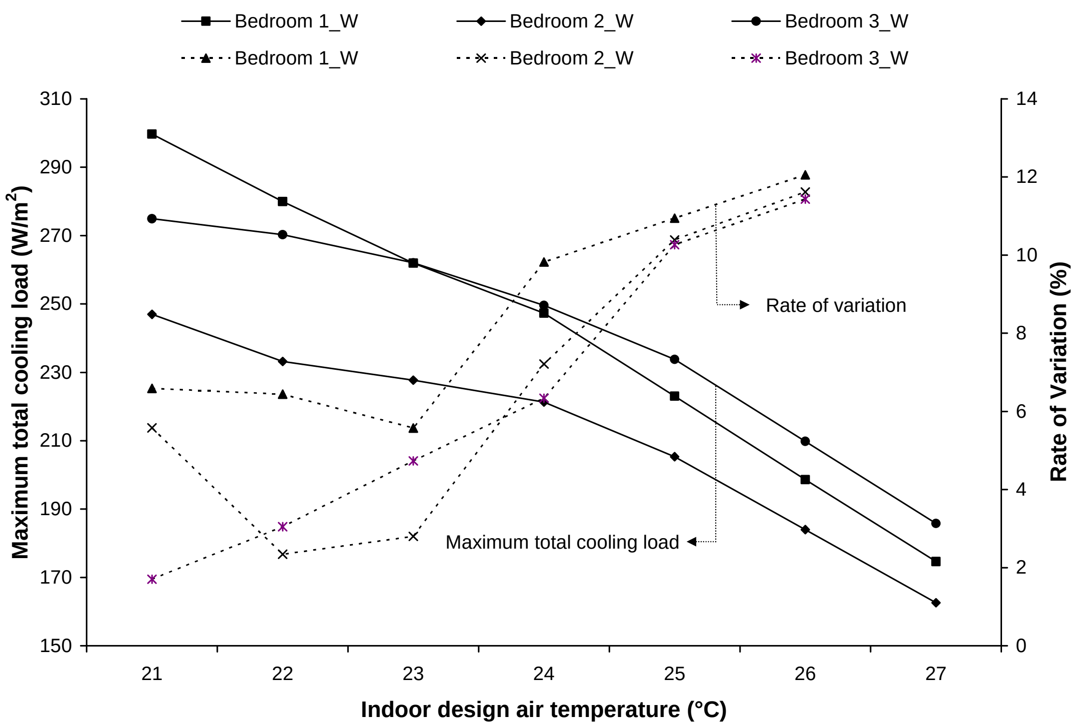

Fig. 6.13 The maximum total cooling loads and their rates of variation at different indoor design air temperatures at NOM

Fig. 6.13 shows the maximum total cooling load and their corresponding rates of variation for the three bedrooms facing west (in the apartment 1 and 6) at different indoor design air temperatures at NOM. All the maximum total cooling loads decreased with the increase of indoor design air temperature. It can be seen that the maximum total cooling loads decreased faster over the range of indoor design air temperature from 24 to 27 °C than those from 21 to 23 °C. Non-linear relationships between the maximum total cooling load and indoor design air temperature were demonstrated.

6.5 Discussions

6.5.1 The differences between the cooling load characteristics at NOM and that at DNM

RACs employed in bedrooms in residential buildings might be operated at any one of the three different operating modes. However, in most residential buildings in the subtropical regions, RACs in bedrooms are often shut down during day time, or unoccupied periods (i.e., RACs operate at NOM). Using the simulated cooling load characteristics for the bedroom 3s facing west at NOM, DOM and DNM, it can be seen that there were three distinctive periods (i.e., shutdown period, cool-down period and conditioning period) having different cooling load characteristics in a bedroom at NOM in a 24-hour cycle, as shown in Fig. 6.14.

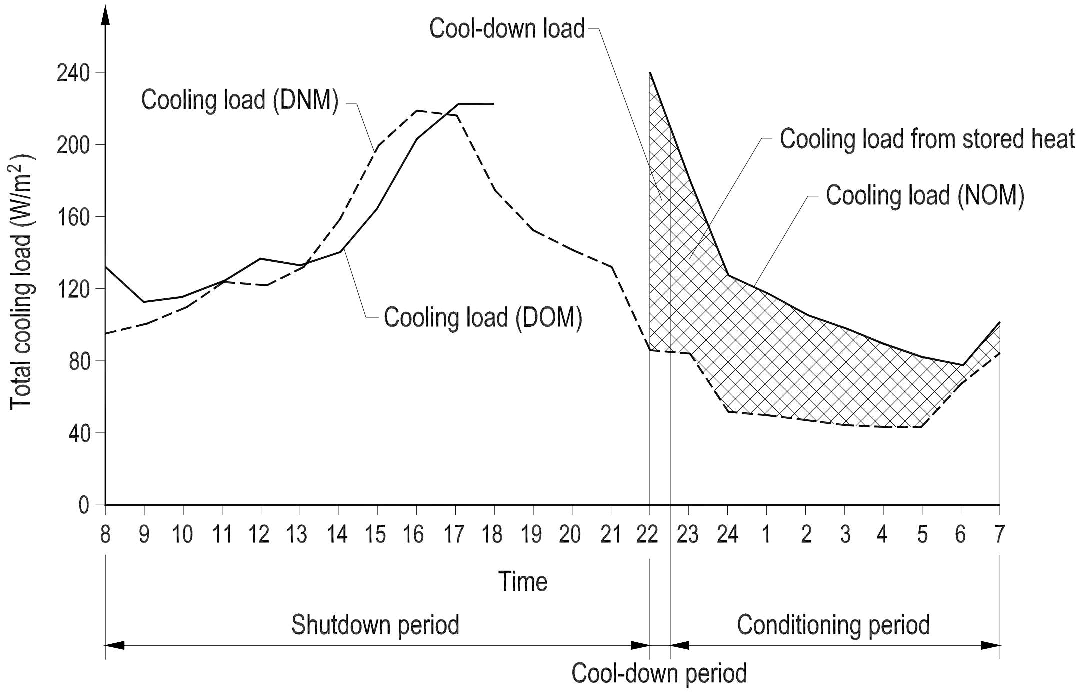

Fig. 6.14 Three periods having different cooling load characteristics in a bedroom at NOM in a 24-hour cycle

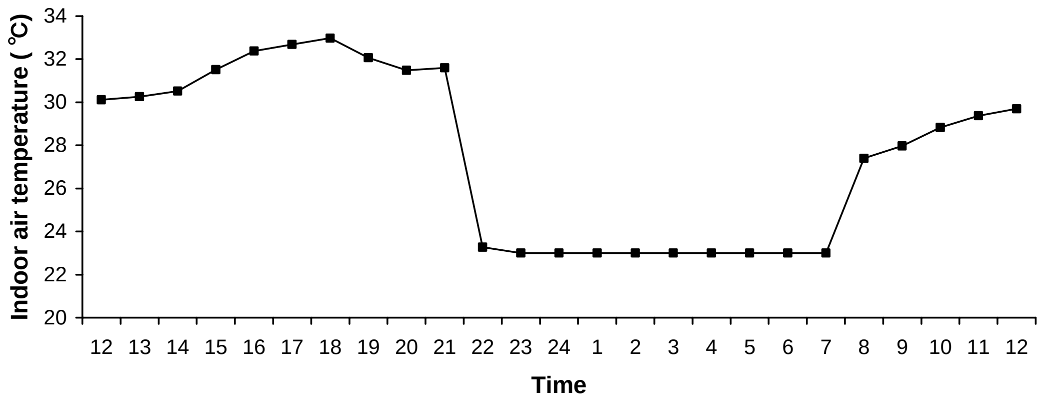

Fig. 6.15 Indoor air temperature variation in a bedroom at NOM in a 24-hour cycle

Under NOM, shutdown period commenced when a RAC was switched off in the morning and ended when the RAC was switched on again in the late evening. When the RAC was turned off, indoor air temperature would increase rapidly (as shown in Fig. 6.15) due to the heat gain from infiltration air, indoor-outdoor air temperature difference, and solar heat gain if any. After that, indoor air temperature would rise further and peak in later afternoon, and then dropped slightly, when outdoor air temperature started to drop. At the same time, the indoor air moisture content also increased due to the moisture infiltration and transfer. Cool-down period commenced when the RAC began to operate and ended when the indoor temperature setting arrived. When the RAC was turned on in the late evening, the supply of cold and dehumidified air caused a sudden drop in indoor air temperature and moisture content. Both heat and moisture were transferred from building envelope to indoor air because of comparatively higher temperature and moisture stored in building envelope. These heat and moisture transfers formed the cool-down cooling load. The length of this cool-down period depended on the tightness of a bedroom, the physical properties of building envelope such as moisture absorption rate, the differences of the temperature and moisture content of indoor air between the shut-down and cool-down periods, and the cooling capacity of a RAC. Conditioning period commenced when cool-down period ended, and ended when the RAC was shut down. In this period, the indoor air temperature was maintained around its setpoint.

Fig. 6.14 also illustrates that at NOM, the cooling load due to the heat stored inside building envelope and furniture (the hatched area in Fig. 6.14), which was accumulated over the shutdown period, dominated the total cooling load, in particular at the beginning period of the RAC's operation.

6.5.2 The differences between the cooling load characteristics at NOM and that at DOM

It can be observed from Fig. 6.14 that the peak total cooling load for the bedroom 3s facing west at NOM, DOM and DNM were 240 W/m^2, 223 W/m^2, and 219 W/m^2, respectively. The peak total cooling load at NOM was 7.7% greater than that at DOM. It can also be seen that the variation of the total cooling load at DOM was rather similar to that at DNM during the same operating period (i.e., from 8:00 to 18:00). This implied that at DOM, the cooling load from stored heat, which was accumulated over nighttime, could be neglected.

On the other hand, the simulated results (shown in Section 6.4.1) suggested that the cooling load characteristics in bedrooms at NOM were significantly different from those at DOM. The total cooling load at NOM peaked at the starting time of an overnight air conditioning process, decreased rather rapidly in the next two hours and then decreased slowly. The maximum total cooling loads for the bedroom 3s facing four different orientations were 2.8 to 3.0 times of their corresponding minimum total cooling load, with an average value of 2.9 times, indicating that the total cooling load at NOM would significantly fluctuate. This can be ascribed to the large amount of heat stored inside building envelope and furniture, which was accumulated over the daytime period when the RAC was not operated. When the RAC was switched on in the late evening, the RAC needed to firstly remove the stored heat, and a large cool-down load was therefore resulted in. The amount of heat stored depended mainly on the mass of building envelope (whether it is heavy, medium, or light); the duration of the operating period of the RAC within a 24-hour cycle; and the characteristics of heat gain, i.e., whether radiant heat or convective heat dominated.

6.5.3 Sizing of RACs in bedrooms

A long-time HVAC industry wise man, John Malloy, believed that equipment sizing and selection (application) is an art form. He suggested that the accepted design criteria, the mathematics and all of the procedures and then “one’s head” should be used [Kurtz 2003]. However, most architects, engineers and air-conditioning contractors do not like to be sued for providing systems with inadequate capacity, which is understandable. Too often, “safety factors” are piled into load computations, and “the worst case” assumptions are made as to simultaneous occurrence of exterior load peaks and interior load peaks [Kohloss 1981]. Therefore, oversizing of a RAC is often seen. An oversizing RAC in a bedroom will result in a wide indoor air temperature swing and elevated humidity levels due to frequent cycling of compressor along with potentially high operating costs, poor system performance and dissatisfied occupants.

Furthermore, oversizing would lead to the problem of under-dehumidification. In a hot, humid climate like Hong Kong, a RAC is in a race with not only sensible heat but also water vapor. Sensible heat is easily beaten. However, unless water vapor, or latent heat, also is beaten, the whole race is lost [Kurtz 2003]. Many previous studies [Henderson 1990, Henderson and Rengarajan 1996, Henderson 1998, Doty 2001, Harriman III and Judge 2002] on the dehumidification performance of direct expansion (DX) air conditioning equipment showed that the dehumidification ability of an air conditioner was strongly affected by part-load, or cycling operation. This was particularly true in applications where a supply fan ran continuously (i.e., at a constant fan mode). Fig. 6.16 [Henderson 1998] shows the dehumidification performance of an air conditioner for an on-off cyclic operation at its rated condition. The compressor in a typical thermostatically controlled, constant-volume DX cooling system only operates for a certain fraction of each hour (i.e., on-off cycling operation). When the compressor is operated at its on cycle, the cooling coil dehumidifies along with removing the sensible heat from a space. As soon as the thermostat is satisfied, the compressor shuts off, the coil stops dehumidifying, and all or part of moisture remaining on the surface of the coil can re-evaporate into the air stream. Therefore, moisture removed by the coil during the on-period is offset by moisture evaporated back into the air stream during the off-period. The net effect is that the performance of moisture removal by the coil is degraded. This degradation of dehumidification may occur at the design condition, but is particularly serious at part-load conditions.

The proper application, selection, sizing, and operation of a RAC are the keys to controlling humidity levels. This requires that the RAC meet the requirement of both sensible and latent loads, not only at the design conditions (full load), but also over a wide range of off-design conditions (part loads) [Amrane et al. 2003]. An oversized RAC often results in an excessive space humidity level and subsequent mold and mildew damage, in particular at part load operating conditions which occur for the majority of a RAC's operating hours. Unfortunately there exist misimpressions that extra cooling capacity somehow removes moisture among designers and developers who incline to add cooling capacity to oversize a RAC. However, extra cooling capacity does not help remove moisture when a RAC is controlled by thermostat, as shown in Fig. 6.16. Undersizing the cooling equipment for the peak sensible load would be a slightly-better strategy to moderate high humidity. The compressor would run longer to remove the sensible load, enhancing dehumidification at the same time [Harriman III and Judge 2002].

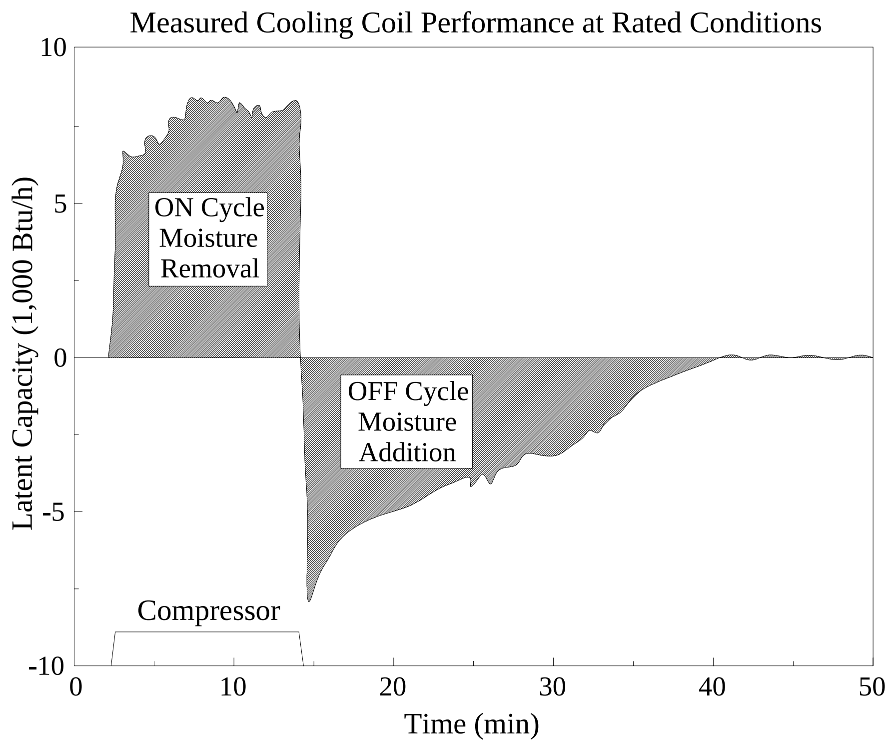

Fig. 6.16 After the compressor shuts off, moisture condensed on the cooling coil re-evaporates [Henderson 1998]

Currently the sizing of RACs in bedrooms is primarily based on the peak total cooling load at DOM. The simulation results suggested that the percentage differences between the hourly total cooling loads for the bedroom 3s facing west and those for bedroom 3s facing other orientations ranged from -57% to 44% while the differences between the peak total cooling load for the bedroom 3s facing west and that for those facing other orientations ranged from 15% to 36% at DOM. Both were greater than the corresponding figures of 6.6% to 25% and 16% to 22% at NOM. This illustrates that the impact of orientation on the total cooling load is more important at DOM than that at NOM. Therefore, sizing the RACs for bedrooms based on the peak total cooling load at DOM without taking orientation into account would potentially oversize bedroom RACs. Given that RACs installed in bedrooms are most likely used at nighttime to maintain an appropriate sleeping thermal environment, i.e., operated at NOM, it is reasonable to size the RACs for bedrooms based on the total cooling load characteristics at NOM. However, based on the cooling load characteristics at NOM, the maximum total cooling load occurred at the beginning of a night air conditioning process, and the load reduced significantly in the next two hours after a RAC started to operation. This special load characteristic would provide some useful insights to avoid oversizing.

The amount of heat stored inside building envelope and furniture would significantly affect the size of RACs to be installed. This is particular true when sizing the RACs for bedrooms based on the cooling load characteristics at NOM since the cool-down load is always the maximum cooling load and the heat stored inside building envelope and furniture dominates the total cooling load at NOM as shown in Fig. 6.14. When sizing a RAC, it can be expected that the room temperature setting is achieved within a reasonable duration of time (i.e., the length of a cool-down period). If this is not the case, the time to achieve a comfort condition may be too long (the RAC is undersized), or too short (the RAC is oversized). For residential buildings, bedrooms are occupied most likely during sleeping time. Therefore, it is reasonable to lengthen the cool-down period in order to reduce the size of RACs for bedrooms. A relatively smaller rated cooling capacity could be offset by a relatively longer cool-down time. This should be acceptable for bedroom air conditioning since occupants might purposely turn on their RACs in advance.

Supposing that the RACs in bedrooms are operated at a new suggested NOM, in which the RACs are switched on one hour earlier than at normal NOM, Fig. 6.17 shows the comparison of the total cooling load for the bedrooms facing west between the new suggested NOM and the normal NOM. At this new suggested NOM, the RACs are switched on one hour in advance to remove the heat stored inside building envelope and furniture although the bedrooms themselves are not occupied during this one hour period. Therefore, the maximum total cooling load in bedroom 1s and 3s can be reduced by 24% and 20% respectively (ΔCLQ as shown in Fig. 6.17), which means that the cooling capacities of the RACs installed in the bedrooms might also be reduced by 20%. If the RACs are switched on two hours earlier than at normal NOM, the maximum total cooling load in bedroom 1s and 3s could be further reduced by 30% and 25%, respectively.

There is one more reason to support downsizing RACs employed in bedrooms. The average maximum total cooling load is 2.9 times of the average minimum total cooling load for the bedroom 3s facing the four orientations at NOM. However, at DOM, the average maximum is only 1.4 times of the average minimum. This implies that the variation of the total cooling load at NOM is larger than that at DOM. If the RACs are already oversized at the beginning of NOM, the problem of oversizing will become more serious during the latter part of NOM. Moreover, the RACs are in fact operated over a wide range of part-load conditions. Even with a correctly sized RAC at the design condition, the part-load problem associated with the latent load remains, although not to the same extent.

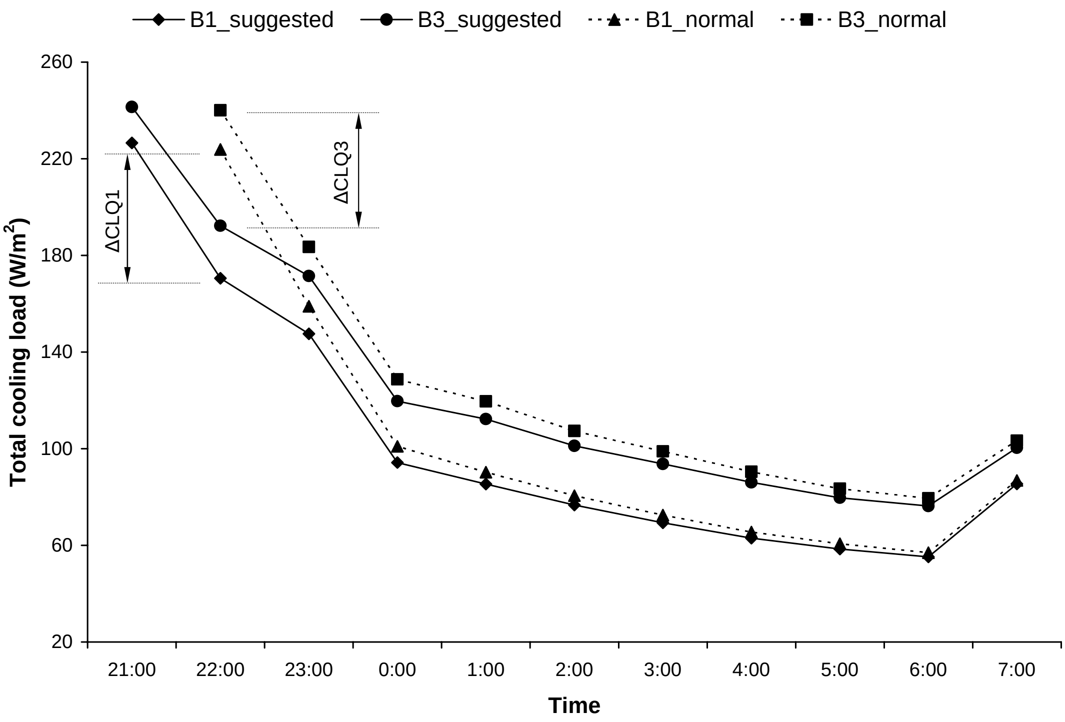

Fig. 6.17 Comparison of total cooling load between the suggested and normal NOM

Therefore, given the special features of bedroom cooling load characteristics at NOM, RACs' sizing should be preferably be based on the load in the later part of an overnight air conditioning process at NOM. This will lead to a minimum number of off cycles, and consequently minimized indoor air temperature and humidity swing, resulting in a high level of thermal comfort and operating energy efficiency. The inadequacy of cooling capacity during the first few hours' operation of a RAC may be compensated by the earlier starting of the RAC which would therefore run longer to remove the stored energy inside building thermal masses.

6.6 Conclusions

The cooling load characteristics in bedrooms at nighttime have been investigated using EnergyPlus and reported in this chapter. It has been demonstrated that the cooling load characteristics in bedrooms at NOM are significantly different from those at DOM. The total cooling load at NOM peaks at the starting time of a nighttime air conditioning process, decreases rather rapidly in the next two hours and then decreases slowly. At NOM, the cooling load due to the heat stored inside building envelope and furniture, which is accumulated over the non- air conditioning (shutdown) period, dominates the total cooling load, in particular at the earlier hours of a nighttime air conditioning process. For the whole duration of NOM, building envelope load dominates the total cooling load, followed by ventilation load, occupancy load, electric equipment load, and lighting load, respectively.

Orientation influences on the total cooling load less at NOM than at DOM. The differences between the hourly total cooling loads for the bedroom 3s facing west and those for the bedroom 3s facing other orientations at NOM range from 6.6% to 25.3%, compared to -57% to 44% at DOM.

Indoor design air temperature significantly affects the yearly total cooling energy. Both the yearly total cooling energy and the maximum total cooling load decrease with the increase of indoor design air temperature. A higher indoor design air temperature would result in a small cooling load, thus less energy consumption and better environmental performance for air conditioning in bedrooms.

The sizing of RACs for bedrooms should be based on the cooling characteristics at NOM given that RACs installed in bedrooms are most likely used at nighttime to maintain an appropriate sleeping thermal environment (i.e., operated at NOM). Moreover, given the special features of bedroom cooling load characteristics at NOM, it is reasonable to size the RACs for bedrooms preferably based on the load at the later hours of an overnight air conditioning process, rather than the peak load at the beginning of the nighttime air conditioning process, with the smaller RACs so sized starting operation adequately before occupancy. A relatively smaller rated cooling capacity of a RAC for a bedroom could be offset by a relatively longer cool-down time. This will lead to a minimum number of off cycles, and consequently a minimized indoor air temperature and humidity swing, resulting in a high level of thermal comfort and operating energy efficiency.

Although the simulation study reported in this chapter is based on the weather conditions and the typical arrangements of high-rise residential blocks in Hong Kong, it is expected the results obtained can be indicative for applications in other tropical and subtropical regions.

Chapter 7

The Outdoor Air Ventilation Rate in Bedrooms

7.1 Introduction

It has been known that outdoor air ventilation significantly influences not only the energy consumed by air conditioning, but also IAQ. Experimental studies [Berg-Munch et al. 1986; Cain et al. 1983; Fanger and Berg-Munch 1983; Fanger 1988; Rasmussen et al. 1985] showed that an outdoor air ventilation rate of 7.5 L/s per person would control human body odor such that approximately 80% of unadapted persons (visitors) would find the odor acceptable. These studies also showed that the same level of body odor acceptability would occur at an indoor CO_2 concentration of 700 ppm above that of the level outdoors. Minimum rates of outdoor air ventilation have been specified in building codes and IAQ standards (see Section 2.4.2). Increasing the outdoor air ventilation rate is an effective way to improve IAQ; however, in most cases it results in consuming more energy.

Two types of RAC are commonly used in high-rise residential buildings in Hong Kong; i.e., WRAC and SRAC. In the 1980s, it was standard for local property developers to supply WRACs in high-rise residences. However, towards the end of the 1990s, there came a trend for property developers to provide SRACs in high-rise apartment buildings in lieu of WRACs [Chow and Lin 2001]. The advantages of using a SRAC include the fact that they are quieter, offer greater flexibility in multi-room services (i.e., two to four indoor units can connect to one outdoor unit) [Lang 1995], and permit greater integrity of the external facades. In most cases, however, the disadvantages include a poorer IAQ due to inadequate ventilation, and higher equipment and installation costs.

In a typical WRAC, a ventilation air outlet complete with a damper is available. It is supposed to control a certain amount of indoor air exhausted to the outdoors. The same amount of outdoor air will then infiltrate through gaps in building envelope, such as poorly sealed doors and windows. In other words, ventilation may be controlled with a WRAC. For a SRAC, however, no provision for ventilation is available. This raises a number of questions: can outdoor ventilation rates in residential buildings employing RACs comply with ASHRAE Standard 62? What is the outdoor airflow rate entering indoors because of the ventilation provision available in a WRAC and is it adequate for an acceptable IAQ for nighttime air conditioning? To what extent could a WRAC provide a better IAQ than a SRAC because of better-controlled ventilation? Unfortunately, there has been little research work reported on outdoor air ventilation rates in rooms employing RACs to answer these questions.

On the other hand, studies on different outdoor air ventilation strategies and the ventilation characteristics in air-conditioned non-residential buildings were reported [Reddy et al. 1998, Sekhar et al. 2002]. One study compared the energy costs of various ventilation strategies for a wide variety of climates and housing types in the United States [Wray et al. 2000]. However, the study was for low-rise residential buildings (i.e., up to three stories) covered by ASHRAE Standard 62.2P [ASHRAE 1999]. These types of residential buildings are very different from the high-rise residential apartment buildings in Hong Kong.

This chapter reports firstly on field studies of monitoring both indoor overnight CO_2 levels and outdoor ventilation rates in bedrooms employing RACs, so that the current situation of ventilation in actual high-rise residential buildings in Hong Kong can be appreciated. This is followed by reporting the results of laboratory experiments on outdoor air ventilation characteristics in bedrooms employing RACs. Thirdly, discussions on the field studies and the laboratory experiments are presented and an improved design for a WRAC is suggested. Finally, the outdoor ventilation requirements for a sleeping environment in high-rise residential buildings are discussed.

7.2 Field studies

7.2.1 Survey on indoor overnight CO_2 levels in bedrooms equipped with RACs in Hong Kong

Outdoor air ventilation rate affects indoor contaminant levels and is therefore closely related to IAQ. There exist a number of approaches to evaluate building ventilation and IAQ. One commonly used approach is to measure and analyze the indoor CO_2 level, which has been referred to as an IAQ indicator [Persily 1993, ASTM 1998].

7.2.1.1 Backgrounds of the surveyed bedrooms

Continuous monitoring of indoor overnight CO_2 levels using CO_2 monitors with data-logging provisions was carried out in twelve bedrooms of various sizes in high-rise residential buildings in Hong Kong. Three of the rooms were equipped with SRACs and all of the others with WRACs. There was an exhaust air outlet complete with a damper in all of the WRACs except for the two installed in Rooms 11 and 12. For those bedrooms equipped with WRACs with a ventilation damper, monitoring was conducted under two positions of the damper, i.e., the ventilation damper was either opened or closed in order to evaluate its effectiveness in controlling ventilation. Other conditions such as fan-speed, indoor air temperature setting, number of occupants, etc., were identical for the two tested damper positions. Indoor CO_2 levels were continuously monitored at an interval of one minute throughout the period during which occupants slept. Details of the bedrooms and monitoring conditions are given in Table 7.1.

The model of CO_2 monitors used in the field studies is Telaire 1050, and their detailed descriptions are given in Section 7.2.2.3.

Table 7.1 Details of the bedrooms and monitoring conditions

Room No.	Date	Sleeping Time	RAC Type	Condition of Ventilation Damper	Net floor area (m^2)	No. of occupants	Capacity (horse power)	Low/High Cool	House Type	Building Year of construction	Location
1	16/10	11:45-06:00	Split		5.8	2	1	Low	Public	2002	3/F Cheung Chau
2	19/10	00:30-08:00	Split		7.4	2	1		Public	1983-86	25/F Chai Wah
3	10/10	02:00-08:30	Split		7.4	2	1	Low	Private	1998	29/F Tai Kok Tsui
4	15/10 18/10	01:00-06:30 00:30-06:30	Window	Closed Opened	4.8	2	1	Low	Public	1980s	14/F Tuen Mun
5	25/10 27/10	01:00-07:00 00:30-07:30	Window	Opened Closed	7.8	2	3/4	Low	Public	2000	30/F Ho Man Tin
6	27/10 28/10	0:00-08:30 0:00-07:30	Window	Closed Opened	6.0	2	3/4	Low	Public	1990s	4/F Tai Wo
7	27/9 28/9	22:55-06:00 23:55-07:00	Window	Closed Opened	5.4	1	1		Public	2002	3/F Cheung Chau
8	13/10 14/10	00:30-06:30 00:30-07:00	Window	Closed Opened	5.6	2	1	Low	Private	1990s	26/F Fanling
9	12/10 14/10	21:25-08:10 02:30-12:00	Window	Closed Opened	6.0	1	1	High	Private	1990	13/F Fanling
10	11/10 21/10	02:30-08:00 02:30-09:30	Window	Opened Closed	5.6	1	1	High	Private	1997	7/F Tseung Kwan
11	5/10	23:15-07:10	Window		8.4	2	1	Low	Private	1992	3/F Fanling
12	4/10	00:45-08:45	Window		6.5	2	1	Low	Private	1987	11/F Mong Kok

7.2.1.2 Results

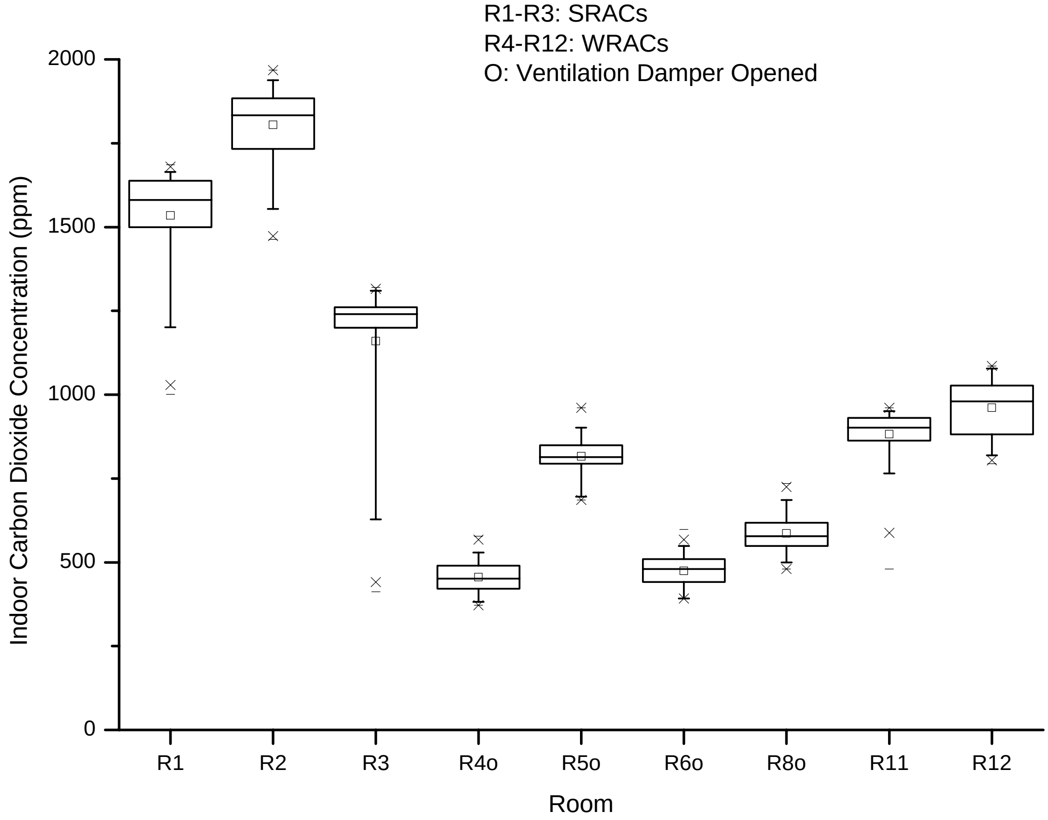

Fig. 7.1 Comparison of the measured overnight CO_2 levels in bedrooms equipped with SRACs with those equipped with WRACs (with ventilation dampers opened), with two occupants

Fig. 7.1 is the statistical box chart for comparing the measured overnight CO_2 levels in different rooms equipped with either SRACs or WRACs. The boxes are determined by the 25th and 75th percentiles, while the whiskers by the 5th and 95th percentiles. It can be seen from Fig. 7.1 that the indoor CO_2 levels in the bedrooms equipped with SRACs were higher than those equipped with WRACs (with ventilation dampers opened) with the same number of occupants. The average CO_2 levels in the measured bedrooms equipped with SRACs ranged from 1160 to 1805 ppm, with a mean average CO_2 level of 1500 ppm. This compared to 456 to 1048 ppm with a mean of 720 ppm for the measured bedrooms using WRACs, with the same number (2) of occupants.

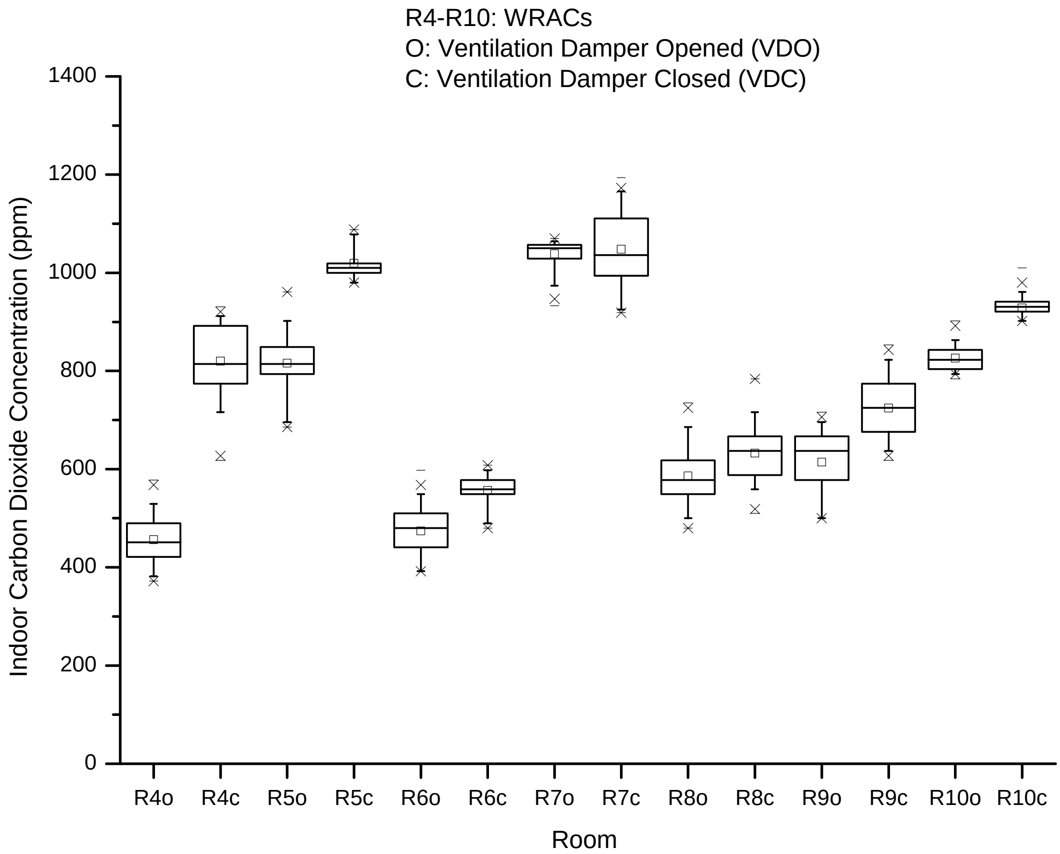

Fig. 7.2 Comparison of the measured overnight CO_2 levels in bedrooms equipped with WRACs between the VDO and VDC condition

Fig. 7.2 is the statistical box chart comparing the overnight CO_2 levels measured in bedrooms using the same WRACs, in conditions where the ventilation dampers were opened (VDO) and where the ventilation dampers were closed (VDC). The boxes are also determined by the 25th and 75th percentiles, while the whiskers by the 5th and 95th percentiles. The figure shows that the indoor overnight CO_2 levels in the same bedrooms in the VDO condition were lower than those in the VDC condition. The average CO_2 levels in the VDO condition ranged from 456 to 1038 ppm, with a mean average CO_2 level of 687 ppm; compared to 632 to 1048 ppm, with a mean of 818 ppm in the VDC condition.

7.2.2 Field studies on outdoor ventilation rates in bedrooms equipped with RACs

The survey results of the indoor overnight CO_2 levels in the bedrooms indirectly suggested that outdoor ventilation rates were different when using different types of RAC, and also under the two different conditions of VDO and VDC for the same WRACs. In order to study the outdoor air ventilation characteristics in rooms employing RACs, field studies on outdoor ventilation rates in bedrooms equipped with RACs based on tracer gas decay technique were carried out. These, together with field survey results of overnight CO_2 levels, would demonstrate more clearly the current situation of ventilation in actual high-rise residential buildings employing RACs.

7.2.2.1 Methodology

The test method for determining air change rates in a single zone by means of the tracer gas decay technique is based on ASTM Standard E741 [ASTM 2001]. It applies to single zone spaces, defined in the Standard as a space or number of spaces wherein the tracer gas concentration can be maintained at a uniform level. The total amount of outdoor air entering a single zone space so tested includes both uncontrolled infiltration through leaks and other openings in building envelope and controlled outdoor air intake through mechanical ventilation.

To determine an average air change rate, firstly a small amount of tracer gas was uniformly introduced into the zone, and afterwards its concentration was continuously monitored over a period of time. The average air change rate for that period can then be determined using the regression method.

Plotting the monitored data on axes of the logarithm of concentration, *lnC(T)*, against time, *T*, as illustrated in Fig. 7.3, and with the assumption of constant air change, the following linear relationship holds:

$$\ln C(T) = -AT + \ln C(0) \qquad (7.1)$$

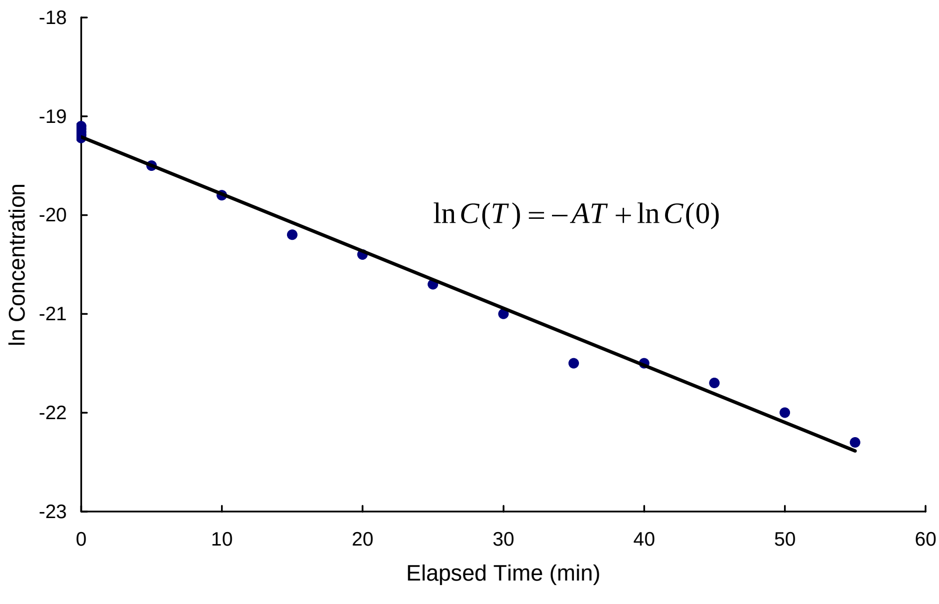

Fig. 7.3 The logarithm of measured concentrations and estimated concentrations of a tracer gas over time

Confidence in A as a constant value can be established by the following statistical procedure:

The estimated standard error of A can be determined by:

$$ESE_A = \frac{s}{[\sum (T_i - \bar{T})^2]^{1/2}} \tag{7.2}$$

$$s^2 = \frac{\sum (Y_i - \hat{Y}_i)^2}{(n-2)} \tag{7.3}$$

The confidence limits on A for the number of testing points (n) and a desired probability of 100(1- α)% were calculated, using a t-distribution table, by:

$$CONF_A(T) = A \pm ESE_A \times t(n-2, 1-\alpha) \tag{7.4}$$

Table 7.2 Minimum durations between the initial and final samples

Air Change Rate (h^{-1})	Minimum Duration of Test (h)
0.25	4
0.5	2
1	1
2	0.5
4	0.25

This tracer gas decay technique is extremely sensitive to zero drift and tends to overweight the contribution of the lowest concentration values to the fitted linear model. Therefore, the sampling time periods were selected carefully to avoid such an overweighting. On the other hand, the minimum sampling durations shown in Table 7.2 were satisfied [ASTM 2001]. The minimum sampling durations are based on a 10% uncertainty at a 95% confidence level in determining air change rates.

7.2.2.2 Issues related to CO_2 tracer gas

There are different kinds of tracer gases that may be used to determine air change such as hydrogen, helium, carbon monoxide, carbon dioxide, sulfur hexafluoride, nitrous oxide, etc. CO_2 was used here because it is inexpensive, relatively safe, non-reactive and measurable. However, applying the tracer gas decay technique to occupant-generated CO_2 involves some considerations not explicitly covered in the ASTM Standard. For example, the decay technique is based on the assumption that there is no source of tracer gas in a building, which, in the case of CO_2, means that the building is unoccupied [Persily 1997]. Therefore, after CO_2 is introduced into a room and during the morning period of CO_2 concentration, the room will have to be kept unoccupied. Another important assumption for the tracer gas decay technique described in ASTM E741 is that the outdoor tracer gas concentration is zero, which is not the case with CO_2 either. However, if the outdoor CO_2 concentration remains constant during the period of decay measurement, by substituting the difference between the indoor and the outdoor concentrations for the indoor concentration, the tracer gas decay technique described above can still be used [ASTM 1998, Sherman 1990]. Therefore, in addition to using an indoor CO_2 monitor, a second CO_2 monitor was also used for measuring the outdoor CO_2 concentration. In order to ensure better measuring accuracy, the outdoor CO_2 concentration was measured before, during, and after the indoor CO_2 concentration monitoring.

7.2.2.3 Instrumentation

The following measuring instruments were used:

- Anemometer

  Model: TSI 8386A VelociCalc Plus
  Range: 0~50m/s
  Accuracy: ±3% of reading or ±0.015m/s
  Function: Measuring outdoor wind velocity

- CO_2 monitors with data logger

  Model: Telaire 1050
  Range: 0~1999ppm
  Accuracy: ±5% of reading or ±50ppm, whichever is greater
  Function: Measuring indoor and outdoor CO_2 concentration

- Temperature and relative humidity sensors with data logger

  Model: HOBO H8 Pro RH/Temp
  Range: -30~50℃; 0~100%
  Accuracy: ±0.2℃; ±3%
  Function: Measuring indoor and outdoor air temperature and relative humidity

All the instruments were calibrated before use.

7.2.2.4 Backgrounds of the studied bedrooms

Field studies on outdoor ventilation rates based on the trace gas decay technique in rooms using RACs were carried out in five bedrooms in high-rise residential buildings in Hong Kong, two equipped with SRACs and the other three with WRACs. For the bedrooms equipped with WRACs, experiments were carried out under two conditions; i.e., VDO and VDC. Detailed background information of the bedrooms is given in Table 7.3.

Table 7.3 Detailed backgrounds of the studied bedrooms

Room	RAC Type	Net Floor Area (m^2)	Net Volume (m^3)	Floor/Total floors	Housing Type	Year of Constru-tion
1	Split	5.57	10.55	3th/5	Public	2002
2	Split	6.62	15.90	29th/41	Private	1998
3	Window	4.21	10.23	3th/5	Public	2002
4	Window	6.70	11.58	6th/16	Private	1980s
5	Window	6.02	12.01	6th/16	Private	1980s

7.2.2.5 Results

The results of field studies on outdoor ventilation rates in rooms using RACs are shown in Fig. 7.4. It can be seen that the outdoor ventilation rates in the studied bedrooms equipped with SRACs were lower than in those equipped with WRACs. The outdoor air flow rates in rooms with SRACs ranged from 1.4 to 2.2 L/s, with a mean value of 1.8 L/s; compared to 2.8 to 4.4 L/s with a mean of 3.4 L/s for WRACs in the VDO condition. Meanwhile, the outdoor air flow rates for WRACs in the VDC condition ranged from 1.7 to 4.1 L/s, with a mean value of 2.7 L/s. This suggested that for the same WRACs, the outdoor ventilation rate in the VDC condition was on average 80% of that in the VDO condition. These agreed well with the survey results of the field monitoring of indoor overnight CO_2 levels in bedrooms equipped with RACs. However, the outdoor ventilation rates in the measured bedrooms, equipped with either SRACs or WRACs, cannot meet the ventilation requirement in ASHRAE Standard 62, i.e., 7.5 L/s per person even if there is only one occupant in a bedroom.

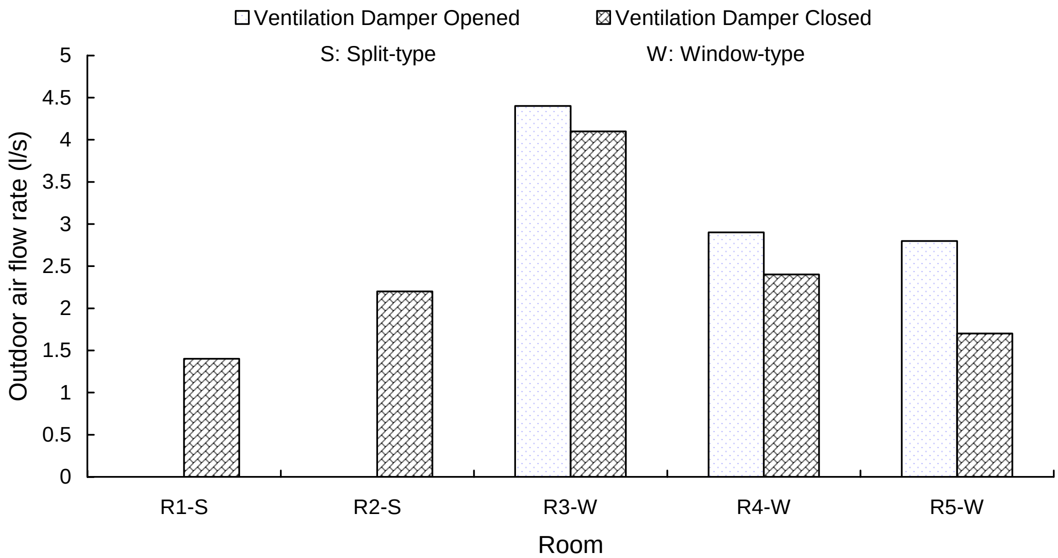

Fig. 7.4 Outdoor ventilation rates in the measured bedrooms
in actual residential buildings in Hong Kong

7.3 Laboratory experiments on outdoor ventilation rates with RACs

The results of field studies may provide both a better understanding of the current ventilation situation in residences using RACs and indicative answers to the questions raised in the Introduction of this chapter. They also suggested that further laboratory-based experiments were desirable in order to eliminate a number of factors that may affect the actual ventilation rate in field studies, such as the tightness of the building envelope, the pressure differences induced by wind and temperature, the operation of mechanical systems, etc. Therefore, laboratory experiments with two typical RACs based on the tracer gas decay technique were also carried out.

7.3.1 Experimental setup

An experimental setup whose schematics are shown in Fig. 7.5 was established. A wooden enclosure, with a dimension of 1730(W) × 1220(L) × 1830(H) mm, was built and equipped with a typical WRAC and a typical SRAC. Their rated cooling capacities and circulation air flow rates were 2 kW, 0.1 m^3/s, and 6.4 kW, 0.3 m^3/s, respectively. In order to investigate the effects of outdoor wind velocity on the ventilation rate with WRACs, an axial fan with a speed controller and corresponding air duct were installed to simulate different outdoor wind velocities. Furthermore, an oscillating fan was placed inside the enclosure to make the indoor tracer gas concentration more uniform. Two CO_2 monitors with date logging device were used to measure the indoor and outdoor CO_2 concentrations, respectively, at an interval of one minute. During the period in which CO_2 concentrations were measured, the simulated wind velocity, and the indoor and outdoor air temperature and relative humidity were also monitored. In order to eliminate the effect of differences in indoor and outdoor air temperature on ventilation rate, the RACs were operated at the fan-only mode; i.e., no cooling or heating was permitted during the course of all of the laboratory experiments.

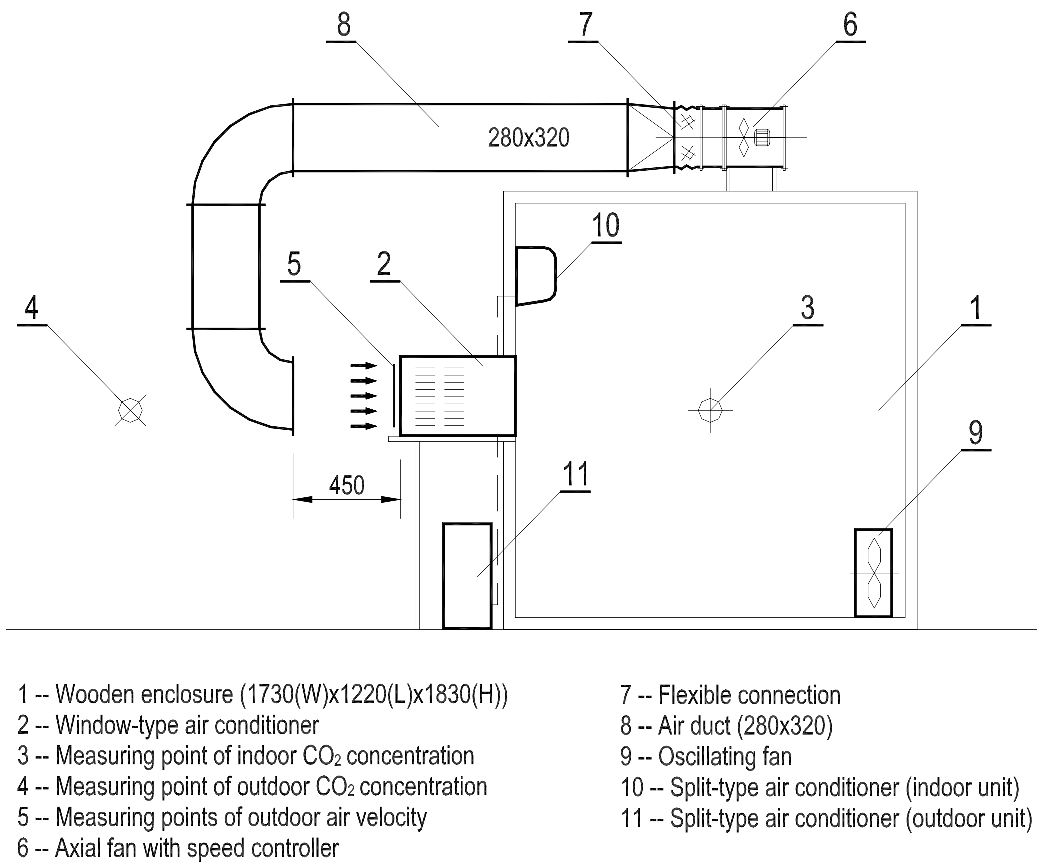

Fig. 7.5 Schematics of the laboratory experimental setup

7.3.2 Experimental conditions

For the WRAC, two fan-speeds could be set to obtain different air circulation flow rates and a ventilation damper was available. However, for the SRAC, three fan-speeds could be set and no provision for ventilation was available. The average air velocity at the outer surface of the WRAC (i.e., the simulated wind velocity), could be adjusted at 0.8, 1.3, 1.8 and 2.2 m/s, respectively, using the axial fan speed controller.

The laboratory experiments were carried out under the fifteen different experimental conditions shown in Table 7.4. For each condition, the air change rate was obtained using the regression method, and the outdoor ventilation rate was then obtained with the known enclosure volume.

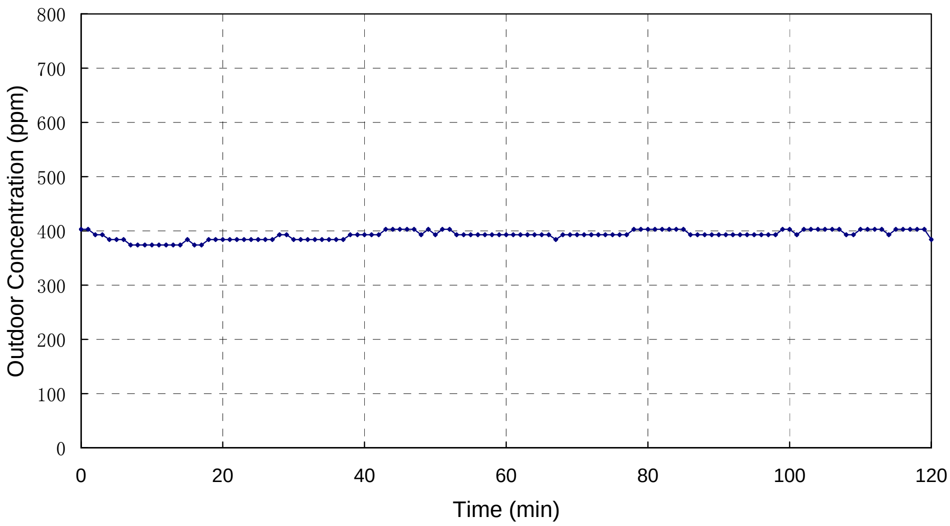

Fig. 7.6 Outdoor CO_2 concentration variation curve (Condition 1)

As an example, for Condition 1, Fig. 7.6 and Fig. 7.7 show the outdoor CO_2 concentration variation curve and the indoor CO_2 concentration decay curve, respectively.

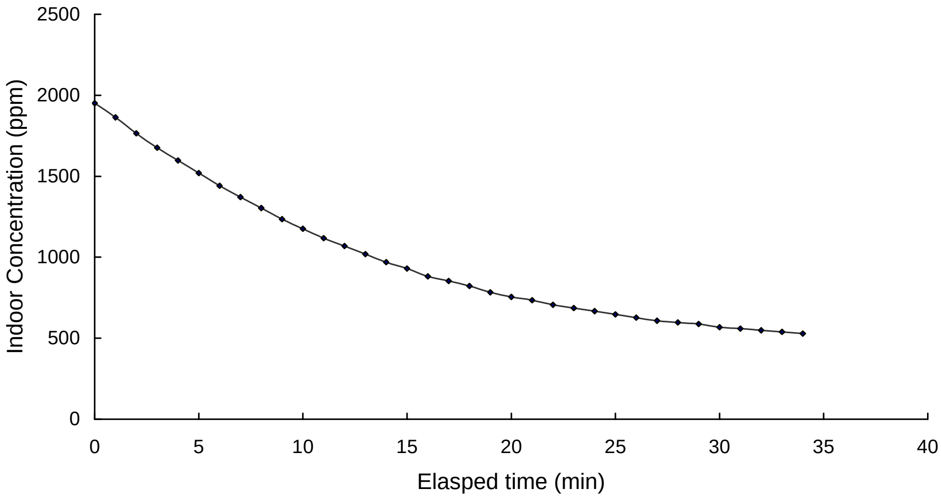

Fig. 7.7 Indoor CO_2 concentration decay curve (Condition 1)

Fig. 7.8 shows the logarithm of the measured concentrations and the estimated concentrations over time. It can be seen from Fig. 7.6 that the outdoor CO_2 concentration stayed relatively constant over the period of monitoring, so the tracer gas decay technique described in Section 7.2.2.1 could be used by substituting the indoor CO_2 concentration with the difference between the indoor and the outdoor CO_2 concentrations. It can also be observed from Fig. 7.8 that the air change rate was almost constant during the measurement because of high coefficient of determination (R^2 =0.9995).

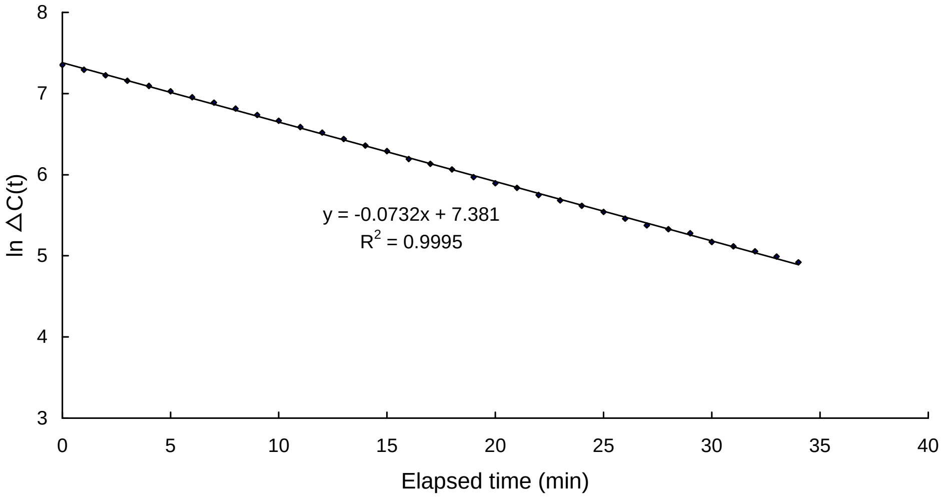

Fig. 7.8 The logarithm of the measured concentrations and estimated concentrations over time (Condition 1)

7.3.3 Results

The results of the laboratory experiments on outdoor ventilation rates with the two RACs are shown in Table 7.4.

Table 7.4 The results of laboratory experiments on outdoor ventilation rates

Condition	Type of RAC	Outside Air Velocity (m/s)	Fan speed	Ventilation damper	Air change rate (h^{-1})	Probability of 95% for ACR	Total outdoor airflow (L/s)
1	Window	0	High	Opened	4.39	4.39±0.10	4.71
2	Window	0	High	Closed	4.29	4.29±0.10	4.60
3	Window	0	Low	Opened	3.73	3.73±0.10	4.00
4	Window	0	Low	Closed	3.53	3.53±0.04	3.78
5	Window	2.2	High	Opened	5.29	5.29±0.10	5.67
6	Window	1.8	High	Opened	5.17	5.17±0.10	5.54
7	Window	1.3	High	Opened	4.90	4.90±0.04	5.26
8	Window	0.8	High	Opened	4.66	4.66±0.10	5.00
9	Window	2.2	High	Closed	5.17	5.17±0.10	5.55
10	Window	1.8	High	Closed	5.13	5.13±0.10	5.50
11	Window	1.3	High	Closed	4.91	4.91±0.10	5.27
12	Window	0.8	High	Closed	4.60	4.60±0.04	4.93
13	Split	0	High		1.97	1.97±0.04	2.11
14	Split	0	Medium		1.91	1.91±0.04	2.05
15	Split	0	Low		1.90	1.90±0.04	2.03

The following may be observed from the results of the laboratory experiments:

- As can be seen from Fig. 7.9 which was produced using the data for Conditions 1-4 and 13-15, the use of a WRAC may provide a higher outdoor ventilation rate than the use of a SRAC.

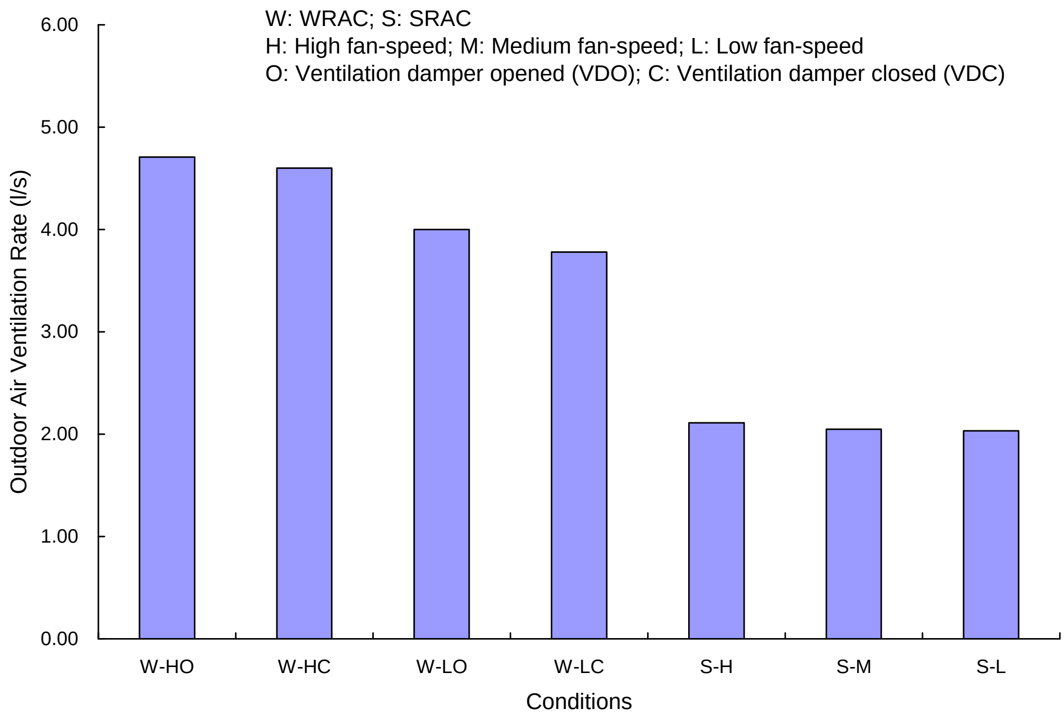

Fig. 7.9 Comparison of outdoor ventilation rates using a WRAC and a SRAC

- Fig. 7.9 also shows that outdoor ventilation rate for a WRAC may slightly increase at a higher fan-speed. However, different fan-speeds had little effect on the outdoor ventilation rate for a SRAC.

- Outdoor ventilation rates in all conditions were lower than the ventilation requirement in ASHRAE Standard 62 [ASHRAE 2001a], i.e., 0.35 ACH but not less than 7.5 L/s per person.

- Fig. 7.10, which was produced using the data for Conditions 1-2 and 5-12, shows the relationship between outdoor ventilation rate and outdoor air velocity for the WRAC. For both the VDO and VDC conditions, linear relationships between outdoor air velocity and outdoor ventilation rate were demonstrated for the

WRAC. A higher wind velocity would result in a higher ventilation rate. It should be noted that the wind direction in these experiments was normal to the outside surface of the experimental WRAC, as shown in Fig. 7.5.

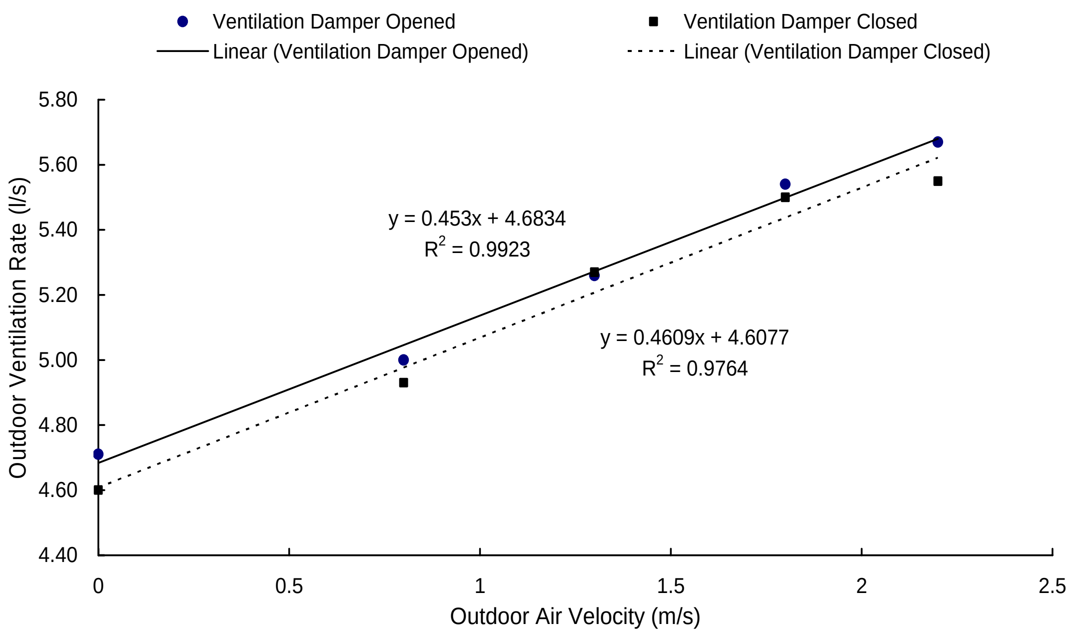

Fig. 7.10 The relationship of outdoor ventilation rate with outdoor air velocity for the WRAC

- The opening or closing of the ventilation damper in the experimental WRAC did not appear to significantly affect the outdoor ventilation rate. As can be seen in Table 7.4, at zero wind velocity and a high fan-speed, the total outdoor ventilation was 4.71 L/s for the VDO condition, compared to 4.60 L/s for the VDC condition. Similarly, at a low fan-speed, the outdoor ventilation rate was 4.00 L/s for the VDO condition, compared to 3.78 L/s for the VDC condition. Therefore, the usefulness of the ventilation damper in the experimental WRAC for controlling outdoor air ventilation was questionable.

7.4. Discussions

7.4.1 Ventilation in residential buildings employing RACs

The results of the field survey indicated that the indoor overnight CO_2 levels in the surveyed bedrooms using WRACs were lower than those using SRACs. They also suggested that for the same WRACs, the indoor CO_2 levels were lower when the ventilation dampers were opened than when they were closed. Furthermore, both the results of the field studies and the laboratory experiments suggest that WRACs may provide a higher outdoor ventilation rate than SRACs. For the same WRAC, the outdoor air flow rate in the VDO condition was higher than that in the VDC condition. Therefore, the results of the laboratory experiments agreed well with those from the field survey. These are understandable and consistent with common understanding, since a WRAC has been perceived to be able to control ventilation with its ventilation damper.

The results of field studies and laboratory experiments suggest that a certain amount of ventilation air can still be found in a room where a SRAC has been used, although a SRAC itself has no provision for controlling ventilation. This may be due to natural infiltration through leaks and other openings in the room. The results of field studies also suggest that the outdoor ventilation rates in the rooms equipped with WRACs in the VDC condition are higher than those using SRACs. This may be explained by the fact that the installation of a WRAC would inevitably produce gaps between the unit and wall, and that the WRAC itself may have some leakages. In addition, WRACs have to be installed in a room with an external wall, in which windows are normally available, compared to SRACs, which can be installed in a room without external walls. Furthermore, high-rise residential buildings equipped with WRACs in Hong Kong are normally old and publicly owned; and therefore may be less airtight than those equipped with SRACs, which are normally newer and privately owned. Therefore, with more air infiltration passages, more outdoor air can infiltrate indoors for WRACs.

On the other hand, the differences in outdoor ventilation rate between the VDO and VDC conditions for the same WRAC were not significantly large. The results of the field studies suggested that the outdoor air flow rate in the VDC condition decreased to 80% of that in the VDO condition for the same WRAC. The results of the laboratory experiments suggested even higher percentage values, with a minimum of 94%. This implies that natural infiltration, rather than the "ventilation function" available in a WRAC, dictates the outdoor ventilation rate in a room equipped with WRAC. This argument can be further substantiated by the results of laboratory experiments on the relationship between outdoor air velocity and outdoor ventilation rate for the WRAC. Since the wind direction was normal to the outside surface of the experimental WRAC, a high wind velocity would cause a reduction in the exhaust air flow rate from indoors, and therefore effectively decreased the ventilation rate. However, the experimental evidence suggested that a higher wind velocity would result in a higher ventilation rate in both the VDO and VDC conditions. This may be because a higher wind velocity means a higher pressure difference between the indoor and outdoor environments, therefore resulting in more natural infiltration through the various air passages of the enclosure.

In order to understand why the ventilation damper of the experimental WRAC is not significantly useful in controlling the outdoor ventilation rate, the construction details of the WRACs currently available in the market have been examined. Fig. 7.11 shows schematically the top-view of a typical WRAC. Most WRACs have an exhaust air outlet, complete with a damper located in either position A or position A'.

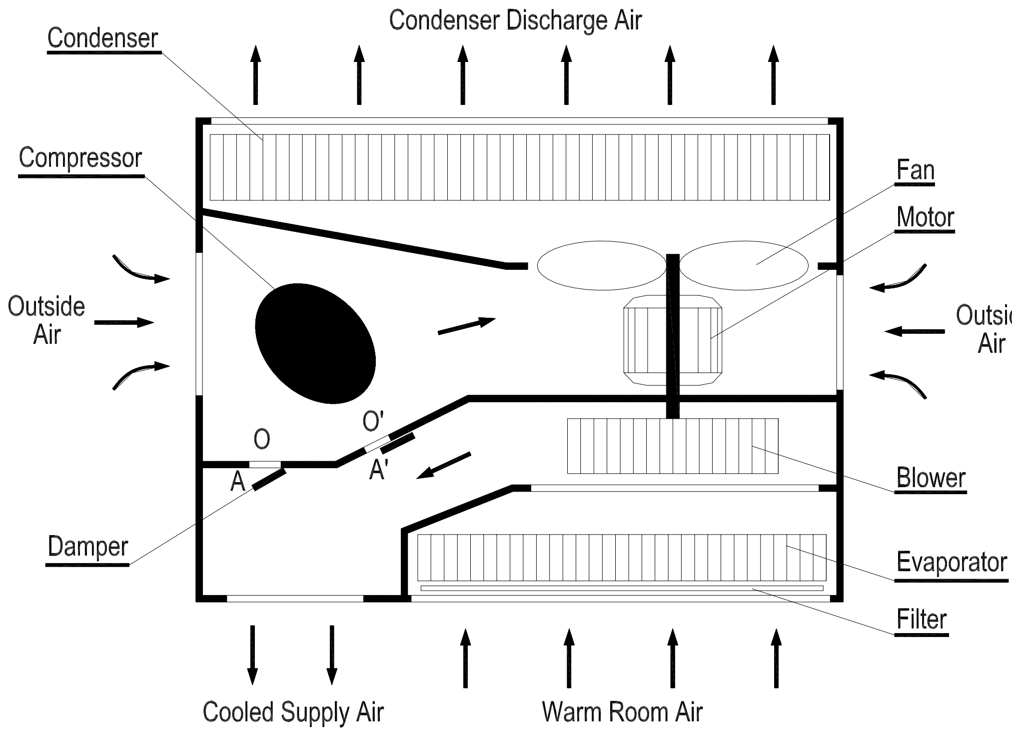

Fig. 7.11 Schematic top-view of a typical WRAC

The air flow rate exhausted from indoors to outdoors through the outlet can be determined by:

$$Q = C_Q (\Delta P_{AO})^n \tag{7.5}$$

Therefore, the amount of exhaust air flowing through the outlet depends on the static pressure difference between A (A') point and O (O') point, and on air flow coefficient of the air flow passage. To increase exhaust airflow rate and consequently the ventilation airflow rate, one method is to increase ΔP_{AO}, and the other to increase C_Q value. However, in the current design of a typical WRAC, the residual pressure of indoor air blower (i.e., the static pressure at A (A') point) cannot be too high if the indoor noise level caused by using a RAC is to be tolerable. This helps explain why the status of a ventilation damper, whether opened or closed, did not affect outdoor ventilation rate significantly, as in the case of the experimental WRAC. In addition, the air exhausted from indoors to outdoor through a ventilation outlet is air that has just exited from a cooling coil (evaporator). This is unreasonable, because exhausting just-cooled and dehumidified air is a waste of energy. Therefore, it is necessary to improve the current design of WRACs, with respect to the location of ventilation air outlet. An improved WRAC design is therefore suggested, as shown in Fig. 7.12.

The location of ventilation air outlet complete with a damper is altered from A (A') to B. Point B is located on the condenser fan suction side. Hence, a certain amount of indoor air can be exhausted. With this improved design, indoor air rather than just-cooled supply air is exhausted. However, with this improved design the physical size of a WRAC might slightly increase because of the presence of a separate exhausting air duct in the unit. It will be desirable to experimentally test the ventilation performance for such an improved design WRAC in the future when its prototype is available.

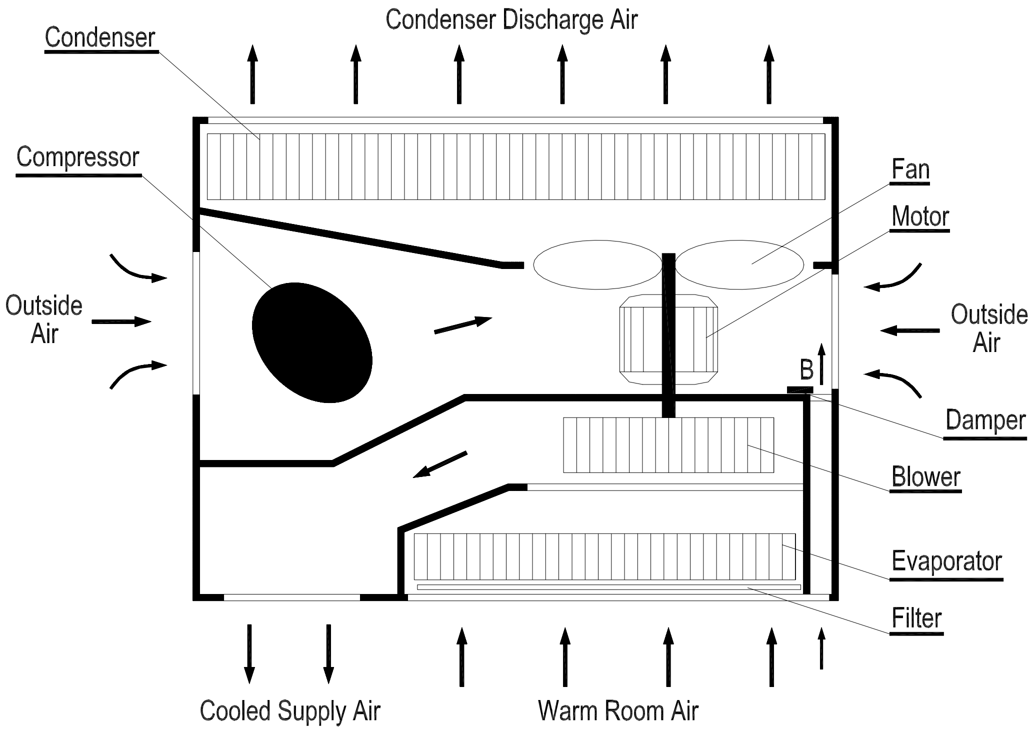

Fig. 7.12 Schematic top-view of an improved WRAC

7.4.2 Ventilation in bedrooms employing RACs at nighttime

Indoor CO_2 level has been used as an indicator of whether an adequate ventilation rate has been obtained for diluting or removing harmful indoor pollutants. However, CO_2 at typical concentrations is not generally considered by health authorities as a health hazard. The time-weighted average threshold limit (based on an 8-hour exposure and a 40-hour work week) and the short-term exposure limit (a 15-minute exposure) for CO_2 are 5,000 ppm and 30,000 ppm, respectively [ACGIH 1995]. The 1000 ppm limit, or the limit of 700 ppm difference between the indoor and outdoor CO_2 concentration specified in ASHRAE Standard 62 is based on its association with human body odor, not with any health or comfort effects of CO_2 itself. Moreover, the recommended outdoor ventilation requirement (i.e., 7.5 L/s per person specified in the Standard) is based on the rate of CO_2 generated per person at an activity level of 1.2 Met. For a sleeping person whose metabolic rate is 0.7 Met [ANSI/ASHRAE 2004], the CO_2 generated per person will correspondingly decrease to 58% of that generated by a sedentary person (1.2 Met), given that the generation of CO_2 has a linear relationship with physical activity [ASHRAE 2001a], shown in Fig. 7.13. On the other hand, people normally adapt quickly to bioeffluents. For adapted persons (occupants), the ventilation rate per person to establish the same acceptance is approximately one third of that for unadapted persons (visitors); in other words, the corresponding difference in indoor and outdoor CO_2 concentration for occupants can be three times higher [Berg-Munch et al. 1986]. Therefore, in bedrooms during nighttime, when the occupants are asleep and there are unlikely visitors, people will generate less CO_2 and bioeffluents. Thus, it can be argued that outdoor ventilation rates might be decreased to a fraction of the normal recommended value.

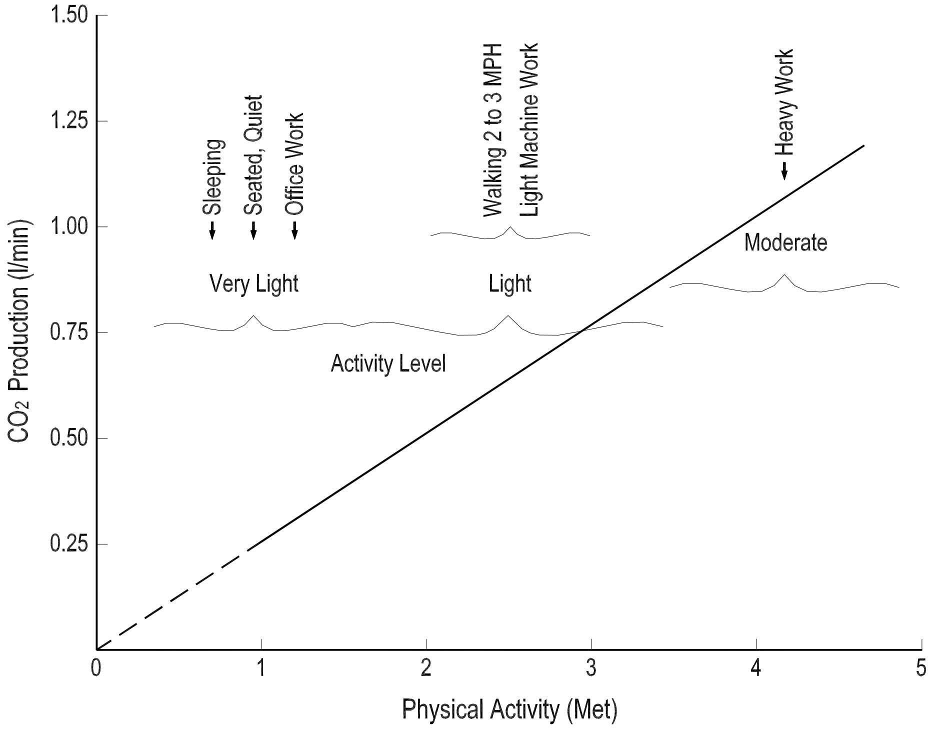

Fig. 7.13 The relationship between CO_2 generation and physical activity [ASHRAE 2001a]

A simple mass balance equation giving the outdoor air flow needed to maintain the steady-state CO_2 concentration within a space below a given limit is as follows:

$$V_o = \frac{N}{C_s - C_o} \tag{7.6}$$

For a sedentary person whose activity level is 1.2 Met, V_o, N and (C_s-C_o) are 7.5 L/s per person, 0.31 L/min, 700 ppm, respectively [ASHRAE 2001a]. Therefore, the outdoor ventilation rate for a sleeping person (0.7 Met) may be decreased to a threshold, V_o'

$$V_o' = V_o \times \frac{N'(C_s - C_o)}{N(C_s' - C_o')} = 7.5 \times \frac{0.7 \times 0.31 \times 700}{1.2 \times 0.31 \times 700 \times 3} = 1.5 \text{ (L/s per person)} \tag{7.7}$$

This means that the threshold of outdoor ventilation rate requirement for a sleeping person in a bedroom may be decreased to only one fifth of that for a sedentary person in a workplace. This threshold value (i.e., 1.5 L/s per person) is based on three assumptions. The first is that the CO_2 generation rate for a sleeping person is a fraction (0.7/1.2, or 58%) of that for a sedentary person in an office. The second is that the ventilation rate per person to establish the same acceptance for occupants is approximately one third of that for visitors [Berg-Munch et al. 1986]. The last is that there are unlikely visitors in a bedroom at nighttime. If only the first assumption is taken into consideration, the ventilation rate for a sleeping person can be decreased to 4.4 L/s per person (7.5×0.58). Therefore, the ventilation rate for a sleeping person may be reduced to somewhere between 1.5 and 4.4 L/s per person. It is suggested, therefore, that the mean of the two values (i.e., 3.0 L/s per person) be used as the proposed new ventilation rate for sleeping environments.

On the other hand, the current situation of ventilation for bedrooms employing RACs in residential buildings in Hong Kong inclines to support this argument. The results of field studies suggested that the mean outdoor air ventilation rates for the measured bedrooms with SRACs and WRACs were 1.8 L/s and 3.4 L/s, respectively. On the other hand, the survey results suggested that the mean CO_2 levels for the monitored bedrooms employing SRACs and WRACs were 1500 and 720 ppm, respectively, with two occupants at nighttime.

It should be also pointed out that although CO_2 concentrations can be an appropriate means of characterizing the acceptability of a space in terms of body odor, they do not provide information on the concentrations of contaminants from other pollutant sources such as building and insulation materials, furniture, fabrics and furnishings, glues, cleaning products and other consumer products. Therefore, when there are some other indoor pollutants other than bioeffluents (e.g., tobacco smoke odor), the proposed outdoor ventilation rate of 3 L/s per person will not become applicable.

7.5 Conclusions

The indoor overnight CO_2 levels were lower than 1000 ppm in most of the surveyed bedrooms in high-rise residential buildings employing WRACs in Hong Kong. However, the CO_2 levels were higher than 1000 ppm in those bedrooms using SRACs.

The use of WRAC may provide a higher outdoor ventilation rate than the use of SRAC. However, this may be ascribed to natural infiltration through leaks of the envelope of a room, since there may be more air infiltration passages in a room with a WRAC than in that with a SRAC.

For WRACs, the ventilation provision currently available (i.e., the ventilation outlet complete with a damper) does not significantly affect outdoor ventilation rate. Therefore, such a damper cannot be expected to provide the ventilation rate as required by a ventilation code, and its intended function of controlling ventilation is limited. In addition, the air exhausted from indoors to outdoors through the ventilation outlet is the air that has just exited from a cooling coil (evaporator). This is unreasonable, because exhausting just-cooled and dehumidified air is a waste of energy. Therefore, an improved design for WRAC has been suggested.

Outdoor ventilation rates in bedrooms equipped with either SRACs or WRACs, in high-rise residential buildings in Hong Kong cannot comply with the ventilation requirement in ASHRAE Standard 62 (i.e., 0.35 ACH but not less than 7.5 L/s per person) even if there is only one occupant in a bedroom, let alone where there are more than one. For a sleeping environment during nighttime, when occupants are asleep and unlikely there are visitors, people will generate less CO_2 and bioeffluents. It can then be argued that outdoor ventilation rate might be decreased to a fraction of what is needed for an office worker. A new ventilation rate of 3.0 L/s per person for a sleeping environment in residential buildings is proposed. This new rate may also be applied to hotel guest rooms during nighttime.

For the situation where the other common activities (reading, dressing, etc.) in bedrooms might also need to be taken into consideration, ventilation rate could be arranged in two different modes (i.e., daytime mode at 7.5 L/s per person and sleeping mode at 3.0 L/s per person).

Chapter 8

Conclusions and Future Work

8.1 Conclusions

The results of the questionnaire survey suggested that over 80% of the respondents would prefer a relatively low indoor air temperature at below 24 °C. On the other hand, 92% of the respondents would wear sleepwear and 90% would cover themselves with quilts or blankets during sleep. The survey results also suggested that approximately 60% of the respondents experienced waking up during sleep because of thermal discomfort, even if the RACs in their bedrooms were turned on. The current IAQ situation in bedrooms in Hong Kong might not be acceptable for most of the surveyed. Over 70% felt stuffy because of poor IAQ with their RACs turned on at night. This may be ascribed to the fact that nearly 70% of the respondents did not use any ventilation methods during sleep when using RACs. Around two-third of the respondents did not know that there was a ventilation switch (bar) in a WRAC and over half of the respondents did not know the function of such a ventilation switch even if they knew its existence in a WRAC. This lack of knowledge of ventilation control devices provided on WRACs indicated an urgent need for user education.

A comfort equation applicable to sleeping thermal environments was derived by introducing appropriate assumptions and modifications to Fanger's comfort equation. The relationship between the thermoneutral temperature and the total insulation value provided by a bedding system has been established by solving the comfort equation. Comfort charts have also been established, and can be used for determining thermally neutral environmental conditions under a given bedding system. The total resistance / insulation of a bedding system which is influenced by a number of factors such as the components of the bedding system (i.e., bedding, sleepwear, bed and mattress), the percentage coverage of surface area by bedding and bed, etc., has a significant impact on thermal comfort in sleeping environments. A small-scale database of the total insulation values provided by the bedding systems commonly used in the subtropics has been developed through experiments using a thermal manikin. The measured total insulation (clo) values of various bedding systems varied greatly ranging from 0.90 to 4.89 clo. The results of the experiments also suggested that the use of a Chinese traditional style bed - *Zongbang* bed, could provide less insulation than the use of the conventional mattress commonly used in Hong Kong. One the other hand, the use of locally so-called air conditioning quilt (summer quilt) cannot help to lower the total insulation significantly. The local air conditioning culture of having dressing / bedding with a relatively high clo value (higher insulation) in summer, and at the same time, maintaining a relatively low indoor air temperature, should be reviewed and changed. People should be encouraged to use as less sleepwear and bedding (or cover as less body surface area by bedding) as possible to lower the total insulation of a bedding system so that the energy use for air conditioning for sleeping environments could be reduced.

The cooling load characteristics in bedrooms at NOM are significantly different from those at DOM. The total cooling load at NOM peaks at the starting time of a nighttime air conditioning process and decreases rather rapidly in the next two hours after starting. At NOM, the cooling load due to the heat stored inside building envelopes and furniture, which is accumulated over non- air conditioning period, dominates the total cooling load, in particular at the earlier hours of a nighttime air conditioning process. A relatively smaller rated cooling capacity of a RAC for a bedroom could be offset by a relatively longer cool-down time, which should be acceptable for bedroom air conditioning. Therefore, the sizing of RACs in bedrooms should not be based on the peak load which occurs at the beginning of a nighttime air conditioning process, but preferably on the cooling load at the later hours of a nighttime air conditioning process, with the smaller RACs so sized starting operation adequately before occupancy. This will lead to a minimum number of off cycles, and consequently a minimized indoor air temperature and humidity swing, resulting in a high level of thermal comfort and operating energy efficiency.

The use of a WRAC may provide a higher outdoor ventilation rate than the use of a SRAC. However, this may be ascribed to natural infiltration through leaks of the envelope of a room, since there may be more air infiltration passages in a room with a WRAC than in that with a SRAC. For WRACs, the ventilation provision currently available does not significantly affect outdoor ventilation rate. Therefore, such a damper cannot be expected to provide the ventilation rate as required by a ventilation code, and its intended function of controlling ventilation is limited. In addition, the air exhausted from indoors to outdoors through the ventilation outlet is the air that has just been cooled by a cooling coil (evaporator). This is unreasonable, because exhausting just-cooled and dehumidified air is a waste of energy. Therefore, an improved design for a WRAC has been proposed to improve ventilation control and to save energy. On the other hand, outdoor ventilation rates in bedrooms equipped with either SRACs or WRACs, in high-rise residential buildings in Hong Kong cannot comply with the current ASHRAE ventilation requirement even if there is only one occupant in a bedroom, let alone where there are more than one. For a sleeping environment during nighttime, when the occupants are asleep and there are unlikely visitors, people will generate less CO_2 and bioeffluents. It can then be argued that outdoor ventilation rate might be decreased to a fraction of what is needed for an office worker. A new ventilation rate of 3.0 L/s per person for sleeping environments is proposed. This new rate may be applied to not only bedrooms in residences but also hotel guest rooms during nighttime.

The outcomes of the project reported in this thesis would provide a fundamental basis in establishing necessary thermal comfort criteria for air conditioned sleeping environments, and in developing the strategies for better design and operation of air conditioning systems for bedrooms. This would in turn help achieve better thermal comfort and improved IAQ for occupants, and reduced the energy use for bedroom air conditioning. The long-term significance of the project is that it will help people to better understand thermal comfort and their actual air conditioning needs, leading to a reduction in the energy use for air-conditioning sleeping environments, in Hong Kong and other tropical and subtropical regions.

8.2 Proposed future work

The thermal comfort model applicable to sleeping environments, which is reported in Chapter 5, was developed based on Fanger's PMV-PPD model; the later used the results from experiments with 1296 human subjects in controlled climate chambers.

However, these experiments only included those samples with an activity level of not less than 58.15 W/m^2 but did not cover the activity level of sleep (40 W/m^2). Although it is very difficult, time consuming and costly to carry out experiments with a statistically meaningful number of human subjects, considerations should be given to carry out experiments in future using a limited number of human subject samples. These future experiments shall include both overnight monitoring of the environmental parameters and the relevant parameters related to sleep quality such as EEG, EOG, EMG, skin temperature and rectal temperature, etc., during the course of sleep and questionnaire surveys after sleep. The future experimental work with human subjects will help determine the exact value of t_{sk} when thermal comfort during sleep is achieved (i.e., the second condition for optimal thermal comfort). In addition, it will also help exactly quantify the sensitivity coefficient, α, used in Equation (5.38) for calculating PMV for sleeping environments.

The project reported in this thesis has opened up a new area of application of air conditioning technology, and also brought in new challenges to air conditioning professionals. Much future work in this specific research area should be initiated and undertaken.

For example, in addition to RACs, many central residential air conditioning systems may also be used in bedrooms. It would be beneficial to compare different types of air condition systems, under the known nighttime bedroom cooling load characteristics and thermal comfort conditions for bedrooms and therefore optimized air conditioning systems used in bedrooms in residential buildings in the subtropics may be specified. On the other hand, it would be also necessary to develop optimized control strategies for direct-expansion (DX) residential air conditioners which are commonly used in bedrooms currently, with regards to their capacity control using variable speed drive technology under the known characteristics of nighttime bedding cooling load.

Given the recommendations that occupants should be encouraged to wear less or use less bedding in bedrooms, so that a higher indoor air temperature may be adopted. However, bedrooms are relatively small compared to commercial spaces such as offices, the temperature and velocity of supply air and its locations may cause such problems as cold draft if these are not properly optimized. The problem of cold draft will deteriorate during sleep if people wear or cover less or if they are too close to supply air flow Therefore, it would be necessary to optimize the three parameters of air supply, locations of supply-return louvers, supply air temperature and velocity, in a bedroom air conditioning system under various bedding systems using CFD technique.

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Appendix A

Survey Questionnaire Form

Survey on thermal sleeping environment and bedroom air conditioning in Hong Kong

Today's date:

Age: ______

Sex: ☐ Male ☐ Female (please tick in the applicable ☐)

1. Do you often turn your air conditioner on at nigh when you sleep in summer?

   ☐ Yes ☐ No

   If yes, do you leave the air conditioner on throughout the duration of your sleep?

   ☐ Yes ☐ No, just ______ hours

2. What kind of air conditioner do you use in your bedroom?

   ☐ Separate window-type ☐ Split-type ☐ Others

3. How many months do you use your air conditioner in your bedroom for sleeping in one year?

   ☐ Less than one month
   ☐ One to two months
   ☐ Two to three months
   ☐ Three to four months
   ☐ Four to five months
   ☐ Five to six months
   ☐ Six to seven months
   ☐ More than seven months

4. Do you have the same opinion as others sharing a bedroom in your family on turning the air conditioner on?

   ☐ Yes ☐ No

5. Do you have the same opinion as others sharing a bedroom in your family on the

indoor air temperature maintained in the bedroom?

☐ Yes ☐ No

6. What is the bedroom air temperature you would like to maintain when you sleep in summer?

☐ Lower than 20℃

☐ 20 to 22℃

☐ 22 to 24℃

☐ 24 to 26℃

☐ 26 to 28℃

☐ 28 to 30℃

☐ 30 to 32℃

☐ Higher than 32℃

7. Do you have any experience of waking up at mid-night because of thermal discomfort when your bedroom air conditioner is turned on?

☐ No

☐ Yes, some times because the bedroom air temperature maintained was high

☐ Yes, some times because the bedroom air temperature maintained was low

☐ Yes, very often because sometimes feel hot and sometimes feel cold

8. What kind of clothing do you wear during sleep with the air conditioner turned on?

☐ Sleepwear (Full slip)

☐ Men's briefs/panties and half slip

☐ Men's briefs/panties

☐ Others, such as naked

9. Do you use any bedcovers during sleep with the air conditioner turned on?

☐ No

☐ Yes, cover with quilt (_____ kg)

☐ Yes, cover with blanket

☐ Yes, cover with something else, such as ______

10. Can you feel the air flow from the air conditioner when you lie down on your bed?

☐ No

☐ Yes, I can feel low air flow

☐ Yes, I can feel medium air flow

☐ Yes, I can feel high air flow

11. Do you like the air flow from the air conditioner when you lie down on your bed?

☐ No

☐ Yes

☐ Doesn't matter

12. Do you feel any stuffy because of poor indoor air quality when the air conditioner is turned on in your bedroom?

☐ No ☐ Yes

13. Do you have any experience of waking up at mid-night because of poor indoor air quality during your sleep with the air conditioner turned on in your bedroom?

☐ No

☐ Yes, sometimes

☐ Yes, very often

14. Do you know there is a ventilation switch (bar) in a separate window-type air conditioner?

☐ Yes ☐ No

15. Do you open the ventilation switch (bar) during your sleep with a separate window-type air conditioner turned on?

☐ No

☐ Yes

☐ Don't know, never know the purpose of the ventilation switch (bar)

16. Do you know the purpose of the ventilation switch (bar) in a separate window-type air conditioner?

☐ No

☐ Yes, induces a little fresh air from outdoor when running at cooling model

☐ Yes, induces a lot of fresh air from outdoor when running at cooling model

☐ Yes, ventilates when running at ventilating model

17. Do you use any ventilation method during your sleep with the air conditioner turned on?

☐ No

☐ Yes, open the window a little

☐ Yes, open the door a little

☐ Yes, turn on the ventilation switch (bar)

☐ Yes, turn on the ventilator

☐ Yes, other method, such as ______

18. Do you think that the bedroom air conditioning could help improve your sleeping quality?

☐ No

☐ Yes, a little help

☐ Yes, some help

☐ Yes, much help

19. Do you think the noise of an air conditioner has any influence on your sleeping quality during your sleep with the air conditioner turning on?

☐ Yes, much influence

☐ Yes, some influence

☐ Yes, a little influence

☐ No influence at all

20. Which noise is higher during your sleep, the noise of your bedroom air conditioner or any other noise such as traffic?

☐ Noise of my bedroom air conditioner

☐ Noise of other sources

Thank you very much for your help!

Appendix B

Program Listing

```
!
! Program for solving the thermal comfort equation for sleeping
environments
! And calculating the PMV and PPD value for a given environmental
condition
!
      Program Thermal Equation for Sleeping Environment
      Integer n
      print *,'This program used to deal with the following relations'
      print *, '----------------------------------------------------'
      print *, '          1--relation of Rt and ta'
      print *, '          2--relation of twb and ta'
      print *, '          3--relation of tr and ta'
      print *, '          4--Others:twb for given ta and RH'
      print *, '          5--Calculating PMV and PPD'
      print *, '----------------------------------------------------'
      print *,'please input the number of calculation you want to do'
      read *,n
      print *, '----------------------------------------------------'
      IF (n .EQ. 1) Then
         call Rt_ta_1
      Else if (n .EQ. 2) Then
              call TWB_ta_2
             else if (n .EQ. 3) Then
                  call Tr_ta_3
               else if (n .EQ. 4) Then
                      call twb_taRH_4
                    else
                        call PMV_PPD_5

      End IF
      End
*
*
      Subroutine Rt_ta_1

* This subroutine deals with the relation of Rt and ta (tr=ta)

      implicit none
      Integer I
      real Rt(40),ta(40),pa(40),Rtclo(40),Pqb(40),a2(40),a3(40)
      real hc, v, RH,FPWS
```

```
      print *,'Subroutine used to calculate the realtion of Rt and ta'
      print *,'please input the values of varaibles v and RH'
      print *,'please input value of v=     m/s'
      read *,v
      print *,'please input value of RH=      %'
      read *,RH

* Calculating the convective heat transfer coefficient

      IF (v .GE. 1.5) then
         print *, 'The value of v is out of range'
         goto 100
      Else
          if (v .LE. 0.15) then
            hc=5.1
          else
            hc=2.7+8.7*v**0.67
          end if
      End IF

* Given values to different air temperature (C)

      ta(1)=-5.0
     do 10,I=2,40
         ta(I)=ta(I-1)+1.0
  10  continue

* Calculating the values of pa

      do 20,I=1,40
         Pqb(I)=FPWS(ta(I))/1000.
         pa(I)=Pqb(I)*RH/100.
  20  continue

* Calculating the values of Rt

      do 30,I=1,40
         a2(I)=0.0692*(5.87-pa(I))
         a3(I)=0.3762*(5.52-pa(I))
         Rt(I)=((34-ta(I))+a3(I))/(40-0.056*(34-ta(I))-a2(I))
         Rtclo(I)=Rt(I)/0.155
  30  continue
      print *,'The following is the results:'
      write (*,*)'Conditions: v=',v,'   (m/s)'
      write (*,*)'            RH=',RH,'   (%)'
      print *, '---------------------------------------------'
      print *, '     ta (C)     pa (kpa)      Rt (clo)'
      print *, '---------------------------------------------'
```

```
      do 40,I=1,40
         print '(F12.2,F13.6,F13.4)',ta(I),pa(I),Rtclo(I)
  40  continue
      open (3,file='data-1.txt')
      do 50 I=1,40
        write (3,'(2F13.4)')ta(I),Rtclo(I)
  50  continue
 100  End
*
*
      Subroutine TWB_ta_2

* This subroutine deals with the relation of twb and ta (tr=ta)

      Integer I
      real ta(600), pa(600), RH(600),Pqb(600),W(600),TWB(600)
      real a4(600), a6(600)
      real hc, v, Rtclo, Rt, a2, a3, FPWS, FTWB, FWPHI, Patm
      print *,'Subroutine used to calculate the ralation of RH and ta'
      print *,'please input the values of varaibles v and Rt'
      print *,'please input value of v =      m/s'
      read *,v
      print *,'please input value of Rt =       clo'
      read *,Rtclo

* Calculating the convective heat transfer coefficient

      IF (v .GE. 1.5) then
         print *, 'The value of v is out of range'
         goto 100
      Else
          if (v .LE. 0.15) then
            hc=5.1
          else
            hc=2.7+8.7*v**0.67
          end if
      End IF

* Given values to different air temperature (C)

      ta(1)=18.0
      do 10,I=2,600
         ta(I)=ta(I-1)+0.025
  10  continue

* Calculating the values of Pqb

      do 20,I=1,600
```

```
         Pqb(I)=FPWS(ta(I))/1000.
  20   continue

* Calculating the values of RH

       Rt=Rtclo*0.155
       a2=0.3762/Rt+0.0692
       a3=0.3762/Rt*5.52+0.0692*5.87-40.
       do 30,I=1,600
          a4(I)=1./Rt*(34.6-ta(I))+0.056*(34-ta(I))
          a6(I)=a4(I)+a3
          pa(I)=a6(I)/a2
          RH(I)=pa(I)/Pqb(I)*100.
  30   continue

* Calculating Wet bulb temperature based on ta and RH

       Patm=101325.
       do 35,I=1,600
          if (RH(I) .GE. 0.0 .AND. RH(I) .LE.100.) then
             W(I)=FWPHI(ta(I),RH(I),patm)
             TWB(I)=FTWB(ta(I),W(I),patm)
          else
             goto 35
          end if
  35   continue

       print *,'The following is the results:'
       write (*,*)'Conditions: v=',v,'   (m/s)'
       write (*,*)'            Rt=',Rtclo,'   (clo)'
       print *, '---------------------------------------------------'
       print *, '     ta (C)     pa (kpa)      RH (%)      twb(C)'
       print *, '---------------------------------------------------'
       do 40,I=1,600
          print '(F12.2,F13.6,F13.4,F11.2)',ta(I),pa(I),RH(I),TWB(I)
  40   continue
       open (3,file='data-2.txt')
       do 50 I=1,600
         write (3,'(4F13.4)')ta(I),pa(I),RH(I),TWB(I)
  50   continue
 100   End
*
*
       Subroutine twb_taRH_4

* This subroutine calculating twb at given ta and RH

       Integer I,J
```

```
      real ta(600),TWB(600),W(600)
      real RH,FWPHI,FTWB
      ta(1)=18.0
      open (3,file='data-10.txt')
     do 10,I=2,600
         ta(I)=ta(I-1)+0.025
 10   continue
      Patm=101325.
      RH=0.0
      do 20,J=1,6
         do 30,I=1,600
            W(I)=FWPHI(ta(I),RH,patm)
            TWB(I)=FTWB(ta(I),W(I),patm)
           write (3,'(F13.4)')TWB(I)

 30      continue
         RH=RH+20.
 20  continue
      End
*
*
      Subroutine Tr_ta_3

* This subroutine deals with the relation of tr and ta

      implicit none
      Integer I
      real ta(300), pa(300), tr(300)
      real a1(300), a2(300)
      real hc, v, RH, Rt, Rtclo, FPWS
      print *,'Subroutine used to calculate the realtion of tr and ta'
      print *,'please input the values of varaibles v and Rt'
      print *,'please input value of v=     m/s'
      read *,v
      print *,'please input value of Rtclo=      clo'
      read *,Rtclo

* Calculating the convective heat transfer coefficient

      IF (v .GT. 1.5) then
         print *, 'The value of v is out of range'
         goto 100
      Else
          if (v .LE. 0.15) then
            hc=5.1
          else
            hc=2.7+8.7*v**0.67
          end if
```

```
      End IF

* Given values to different air temperature (C)

      ta(1)=10.0
     do 10,I=2,300
         ta(I)=ta(I-1)+0.1
  10  continue

* Calculating the values of pa (kpa)

      RH=0.5
      do 20,I=1,300
         pa(I)=FPWS(ta(I))*RH/1000.
  20  continue

* Calculating the values of tr

      Rt=Rtclo*0.155
      do 30,I=1,300
         a1(I)=Rt*(40-0.056*(34-ta(I))-0.0692*(5.87*pa(I)))
         a2(I)=34.6-(a1(I)-0.3762*(5.52-pa(I)))
         tr(I)=(a2(I)*(4.7+hc)-hc*ta(I))/4.7
  30  continue
      print *,'The following is the results:'
      write (*,*)'Conditions: v=',v,'   (m/s)'
      write (*,*)'        Rtclo=',Rtclo,'   (clo)'
      print *, '---------------------------------------------'
      print *, '     ta (C)     pa (kpa)      tr (C)'
      print *, '---------------------------------------------'
      do 40,I=1,300
         print '(F12.2,F13.6,F13.4)',ta(I),pa(I),tr(I)
  40  continue
      open (3,file='data-3.txt')
      do 50 I=1,300
        write (3,'(2F13.4)')ta(I),tr(I)
  50  continue
 100  End
*
*
      Subroutine PMV_PPD_5

* This subroutine calculating the PMV and PPD value for a
* given environmental condition for sleeping enviornment

      implicit none
      real hc, v, RH, pa, ta, tr, to, Rtclo, Rt, FPWS
      real PMV, PPD, a1, Pqb, afa
```

```
      print *,'Subroutine used to calculate PMV and PPD'
      print *,'please input the varaibles of the environment'
      print *,'please input value of v=     m/s'
      read *,v
      print *,'please input ambient air temperature ta=     C'
      read *,ta
      print *,'please input value of RH=      %'
      read *,RH
      print *,'please input mean radiant temperature tr=     C'
      read *,tr
      print *,'please input the total insulation value Rt=     clo'
      read *,Rtclo

* Calculating the convective heat transfer coefficient

      IF (v .GE. 1.5) then
         print *, 'The value of v is out of range'
         goto 100
      Else
          if (v .LE. 0.15) then
            hc=5.1
          else
            hc=2.7+8.7*v**0.67
          end if
      End IF

* Calculating the values of pa

      Pqb=FPWS(ta)/1000.
      pa=Pqb*RH/100.

* Calculating the value of PMV and PPD

      to=(4.7*tr+hc*ta)/(4.7+hc)
      afa=0.303*(exp(-0.036*40)+0.028)
      a1=(34.6-to)+0.3762*(5.52-pa)
      Rt=0.155*Rtclo

      PMV=afa*(40-a1/Rt-0.056*(34-ta)-0.0692*(5.87-pa))
      PPD=100.-95.*exp(-(0.03353*PMV**4+0.2179*PMV**2))

      print *,'The following is the results:'
      write (*,*)'Conditions: v=',v,'    (m/s)'
      write (*,*)'            ta=',ta,'    (C)'
      write (*,*)'            RH=',RH,'    (%)'
      write (*,*)'            tr=',tr,'    (C)'
      write (*,*)'            Rt=',Rtclo,'    (clo)'
      print *, '---------------------------------------'
```

```
      print *, '       to         PMV         PPD'
      print *, '------------------------------------'
      print '(3F12.4)',to,PMV,PPD
 100  End

C*********************************************************************
C* PSYCHROMETRIC FUNCTIONS AS A FUNCTION OF THE BAROMETRIC PRESSURE - *
C* BASED OF THE ASHRAE FUNDAMENTALS HANDBOOK -  CHAPTER 5             *
C*                                                                    *
C* UNITS : KJ, KG, PA, DEGREE C                                       *
C*                                                                    *
C*     FUNCTION NAME        RETURNED VALUE                            *
C* ___________________________________________________________________*
C*  1. FPWS(TDB)            PWS  = SATURATED VAPOUR PRESSURE(PA)      *
C*  2. FTDEW(W,PATM)        TDEW = DEW POINT TEMP. OR SATUR. TEMP.(C) *
C*  3. FPWW(W,PATM)         PW   = VAPOR PRESSURE(PA)                 *
C*  4. FWPW(PW,PATM)        W    = HUMIDITY RATIO(KG MOIST./KG DRY AIR*
C*  5. FWPHI(TDB,PHI,PATM)  W    = HUMIDITY RATIO(KG MOIST./KG DRY AIR*
C*  6. FWTWB(TDB,TWB,PATM)  W    = HUMIDITY RATIO(KG MOIST./KG DRY AIR*
C*  7. FWHA(TDB,HA)         W    = HUMIDITY RATIO(KG MOIST./KG DRY AIR*
C*  8. FTDB(W,HA)           TDB  = AIR DRY BULB TEMPERATURE(C)        *
C*  9. FPHI(TDB,W,PATM)     PHI  = RELATIVE HUMIDITY(%)               *
C* 10. FHAIR(TDB,W)         HAIR = ENTHALPY OF MOIST AIR(KJ/KG DRY AIR*
C* 11. FHSAT(TSAT,PATM)     HSAT = SATURATED ENTHALPY(KJ/KG DRY AIR)  *
C* 12. FTSAT(HS,PATM)       TSAT = SATURATED TEMPERATURE(C)           *
C* 13. FTWB(TDB,W,PATM)      TWB  = WET BULB TEMPERATURE(C)           *
C* 14. FTDB2(TWB,PHI,PATM)                                            *
C*                                                                    *
C* ___________________________________________________________________*
C* THE SPECIFIC HEATS ARE NOT VARYING WITH THE TEMPERATURE            *
C* CPA = 1.006 KJ/KG*K,  SPECIFIC HEAT OF DRY AIR                     *
C* CPG = 1.805 KJ/KG*K,  SPECIFIC HEAT OF WATER VAPOR                 *
C* HFG = 2501. KJ/KG,    LATENT HEAT OF VAPORISATION                  *
C* CPW = 4.187 KJ/KG*K,  SPECIFIC HEAT OF WATER                       *
C*********************************************************************
C
C*********************
C* LIST OF VARIABLES *
C*********************

C HA   = H, AIR ENTHALPY (KJ/KG DRY AIR)
C HSAT = HS, AIR SATURATION ENTHAPY (KJ/KG DRY AIR)
C PATM = ATMOSPHERIC PRESSURE (PA)
C PHI  = RELATIVE HUMIDITY (%)
C PS   = PWS, WATER VAPOR SATURATION PRESSURE (PA)
C PW   = PARTIAL PRESSURE OF WATER VAPOR (PA)
C TDB  = AIR DRY BULB TEMPERATURE (C)
C TDEW = DEW POINT TEMPERATURE (C)
```

```
C TSAT = TS, SATURATION TEMPERATURE (C)
C TWB  = AIR WET BULB TEMPERATURE (C)
C W    = HUMIDITY RATIO (KG MOISTURE/KG DRY AIR)
C WS   = HUMIDITY RATIO AT SATURATION CONDITIONS (KG MOISTURE/KG DRY AIR)

C*****************************************************************
C*1. SATURATION PRESSURE OVER LIQUID WATER FOR THE TEMPERATURE  *
C* RANGE OF 0 DEG C TO 200 DEG C                                *
C* REFERENCE: ASHRAE FUNDAMENTALS HANDBOOK - CHAPTER 5          *
C*****************************************************************

      FUNCTION FPWS(TDB)

      DATA C1/-5800.2206/,C2/1.3914993/,C3/-0.048640239/,
     +C4/0.41764768E-4/,C5/-0.14452093E-7/,C6/6.5459673/

      TD=TDB+273.15

      FPWS=EXP(C1/TD+C2+C3*TD+C4*TD**2.+C5*TD**3.+C6*ALOG(TD))
      RETURN
      END

C*********************************************************************
C* 2. DEW POINT TEMPERATURE OR SATURATION TEMPERATURE AS A FUNCTION *
C* OF THE HUMIDITY RATIO                                            *
C* RANGE OF 0 DEG C TO 70 DEG C                                     *
C* REFERENCE: ASHRAE FUNDAMENTALS HANDBOOK - CHAPTER 5              *
C*********************************************************************

      FUNCTION FTDEW(W,PATM)
      DATA C0/-35.957/,C1/-1.8726/,C2/1.1689/
      PS=FPWW(W,PATM)

      IF (PS.LT.1.E-3) THEN
      FTDEW=0.
      RETURN
      ENDIF

      ALPHA=ALOG(PS)
      FTDEW=C0+C1*ALPHA+C2*ALPHA**2
      RETURN
      END

C***********************************************************************
C* 3. PARTIAL PRESSURE OF WATER VAPOR AS A FUNCTION OF THE HUMIDITY   *
C* RATIO AND OF THE BAROMETRIC PRESSURE                               *
C***********************************************************************
```

```
      FUNCTION FPWW(W,PATM)

      FPWW=PATM*W/(0.62198+W)

      RETURN
      END

C***********************************************************************
C* 4. HUMIDITY RATIO AS A FUNCTION OF THE VAPOR PRESSURE AND THE      *
C* BAROMETRIC PRESSURE                                                 *
C***********************************************************************

      FUNCTION FWPW(PW,PATM)

      FWPW=0.62198*PW/(PATM-PW)
      RETURN
      END

C***********************************************************************
C* 5. HUMIDITY RATIO AS A FUNCTION OF THE AIR DRY BULB TEMPERATURE AND*
C* THE RELATIVE HUMIDITY                                               *
C***********************************************************************

      FUNCTION FWPHI(TDB,PHI,PATM)
      PS=FPWS(TDB)
      PW=0.01*PHI*PS
      FWPHI=FWPW(PW,PATM)
      RETURN
      END

C**********************************************************************
C* 6. HUMIDITY RATIO AS A FUNCTION OF THE AIR DRY BULB TEMPERATURE   *
C* AND OF THE AIR WET BULB TEMPERATURE                                *
C**********************************************************************

      FUNCTION FWTWB(TDB,TWB,PATM)
      DATA CPA/1.006/,CPG/1.805/,HFG/2501./,CPW/4.187/

      PSTWB=FPWS(TWB)
      WS=FWPW(PSTWB,PATM)
      FWTWB=(WS*(HFG+(CPG-CPW)*TWB)-CPA*(TDB-TWB))/(HFG+CPG*TDB-CPW*TWB)

      RETURN
      END

C *********************************************************************
C * 7. HUMIDITY RATIO AS A FUNCTION OF THE AIR DRY BULB TEMPERATURE   *
C * AND OF THE AIR ENTHAPY                                             *
```

```
C ********************************************************************

      FUNCTION FWHA(TDB,HA)
      DATA CPA/1.006/,CPG/1.805/,HFG/2501./

      FWHA=(HA-CPA*TDB)/(CPG*TDB+HFG)

      RETURN
      END

C ********************************************************************
C * 8. AIR DRY BULB TEMPERATURE AS A FUNCTION OF THE HUMIDITY RATIO   *
C * AND OF THE AIR ENTHAPY                                           *
C ********************************************************************

      FUNCTION FTDB(W,HA)
      DATA CPA/1.006/,CPG/1.805/,HFG/2501./

      FTDB=(HA-HFG*W)/(CPA+CPG*W)

      RETURN
      END

C*********************************************************************
C* 9. RELATIVE HUMIDITY AS A FUNCTION OF THE DRY AIR TEMPERATURE AND *
C* OF THE HUMIDITY RATIO                                             *
C*********************************************************************

      FUNCTION FPHI(TDB,W,PATM)
      PW=FPWW(W,PATM)
      PS=FPWS(TDB)

      IF (PS.EQ.0.) THEN
      FPHI=0.
      ENDIF

      FPHI=100*PW/PS
      RETURN
      END

C ****************************
C * 10. ENTHAPY OF MOIST AIR *
C ****************************

      FUNCTION FHAIR(TDB,W)
      DATA CPA/1.006/,CPG/1.805/,HFG/2501./

      FHAIR=CPA*TDB+W*(CPG*TDB+HFG)
```

```
      RETURN
      END

C ***********************************************************************
C * 11. AIR SATURATION ENTHALPY AS FUNCTION OF SATURATION TEMPERATURE *
C ***********************************************************************

      FUNCTION FHSAT(TSAT,PATM)
      PS=FPWS(TSAT)
      WS=FWPW(PS,PATM)
      FHSAT=FHAIR(TSAT,WS)
      RETURN
      END

C ************************************************************************
C * 12. SATURATION TEMPERATURE AS A FUNCTION OF THE AIR SATURATION     *
C * ENTHALPY                                                            *
C *                                                                     *
C * DETERMINATION OF TSAT BY THE SECANT METHOD                          *
C ************************************************************************

      FUNCTION FTSAT(HS,PATM)

C ******************
C * INITIAL VALUES *
C ******************************************************************
C * FIRST VALUE : FIT ON THE ASHRAE VALUES AT BAROMETRIC PRESSURE *
C ******************************************************************

      DATA C0/-6.0055/,C1/0.68510/,C2/-0.0056978/,C3/3.5344E-5/,
     + C4/-1.2891E-7/,C5/2.0165E-10/

      TS1=C0+HS*(C1+HS*(C2+HS*(C3+HS*(C4+HS*C5))))

      HS1=FHSAT(TS1,PATM)
      DELTH1=HS1-HS
      IF(ABS(DELTH1).LT.1.E-3) THEN
      FTSAT=TS1
      RETURN
      ENDIF

      TS2=TS1-5.

C ******************************
C * STARTING OF THE ITERATIONS *
C ******************************
```

```
      DO 10 I=1,50
      HS2=FHSAT(TS2,PATM)
      DELTH2=HS2-HS

      IF(ABS(DELTH2).LT.1.E-3) GOTO 20
c
       IF(ABS(DELTH2-DELTH1).GT.1E-5) THEN
         TS=TS1-DELTH1*(TS2-TS1)/(DELTH2-DELTH1)
       ENDIF
c
      TS1=TS2
      HS1=HS2
      DELTH1=DELTH2
      TS2=TS
10    CONTINUE

C *********************************************************
C * IF NO CONVERGENCE AFTER 50 ITERATIONS : ERROR MESSAGE *
C *********************************************************

C     NO CONVERGENCE IN THE FUNCTION FHSAT(TS,PATM), HS(GIVEN)
C     USE LAST CALCULATED VALUES

20    FTSAT=TS2
      RETURN
      END

C **********************************************************************
C * 13. AIR WET BULB TEMPERATURES AS A FUNCTION OF THE AIR DRY BULB    *
C * TEMPERATURE,THE HUMIDITY RATIO AND THE BAROMETRIC PRESSURE         *
C *                                                                    *
C * DETERMINATION OF TWB BY THE SECANT METHOD                          *
C **********************************************************************

      FUNCTION FTWB(TDB,W,PATM)
      DATA CPW/4.187/

C *********************************************************************
C * INITIAL VALUES - TWB2 = SATURATION TEMPERATURE CORRESPONDING TO H *
C *                  TWB1 = TWB2 - 1                                  *
C *********************************************************************

      H=FHAIR(TDB,W)

      TWB1=FTSAT(H,PATM)-5.
      PWS1=FPWS(TWB1)
      WS1=FWPW(PWS1,PATM)
      H1=FHAIR(TWB1,WS1)-CPW*TWB1*(WS1-W)
```

```
      DELTH1=H1-H

      TWB2=TWB1+5.

C ******************************
C * STARTING OF THE ITERATIONS *
C ******************************

      DO 10 I=1,50
      PWS2=FPWS(TWB2)
      WS2=FWPW(PWS2,PATM)
      H2=FHAIR(TWB2,WS2)-CPW*TWB2*(WS2-W)
      DELTH2=H2-H

      IF(ABS(DELTH2).LT.1.E-3) GOTO 20

      IF(ABS(DELTH2-DELTH1).GT.1E-50) THEN
       TWB=TWB1-DELTH1*(TWB2-TWB1)/(DELTH2-DELTH1)
      ELSE
       TWB=TWB1
      ENDIF
c
      TWB1=TWB2
      H1=H2
      DELTH1=DELTH2
      TWB2=TWB
10    CONTINUE

C *********************************************************
C * IF NO CONVERGENCE AFTER 50 ITERATIONS : ERROR MESSAGE *
C *********************************************************

C     NO CONVERGENCE IN THE FUNCTION FTWB(TDB,W,PATM),  H(TDB,W)
C     USE LAST CALCULATED VALUES

20    FTWB=TWB2
      RETURN
      END

C ***********************************************************************
C * 14. AIR DRY BULB TEMPERATURES AS A FUNCTION OF THE AIR WEB BULB    *
C * TEMPERATURE,RELATIVE HUMIDITY AND THE BAROMETRIC PRESSURE          *
C *                                                                     *
C * DETERMINATION OF TDB BY THE SECANT METHOD                          *
C ***********************************************************************

      FUNCTION FTDB2(TWB,PHI,PATM)
```

```
C ********************************************************************
C * INITIAL VALUES - TDB1 = TWB                                      *
C *                 TDB2 = TDB2+0.1                                  *
C ********************************************************************

      TDB1=TWB
      A=0.00001*(65+6.75/5.)
      Pqbs=FPWS(TWB)

C ******************************
C * STARTING OF THE ITERATIONS *
C ******************************

      do 20 i=1,400
      Pqb=FPWS(TDB1)
      Pq=Pqbs-A*(TDB1-TWB)*PATM
      H=Pq/Pqb*100.
      DELTH=H-PHI
      if (ABS(DELTH) .LT. 0.2) then
         goto 30
      else
         TDB1=TDB1+0.1
      end if
   20 continue

C *********************************************************
C * IF NO CONVERGENCE AFTER 400 ITERATIONS : ERROR MESSAGE *
C *********************************************************

C    NO CONVERGENCE IN THE FUNCTION FTDB(TWB,PHI,PATM)
C    USE LAST CALCULATED VALUES

   30 FTDB2=TDB1
      return
      end
```

Appendix C

Publications Arising from the Thesis

I. Journal papers

- Zhongping Lin and Shiming Deng. The outdoor air ventilation rate in high-rise residences employing room air conditioners. *Building and Environment*, Vol. 38, No. 12, pp. 1389-1399. 2003 (based on Chapter 7)
- Zhongping Lin and Shiming Deng. A study on the characteristics of nighttime bedroom cooling load in tropics and subtropics. *Building and Environment*, Vol. 39, No. 9, pp. 1101-1114. 2004 (based on Chapter 6)

II. Manuscripts

- Zhongping Lin and Shiming Deng. Thermal comfort in sleeping environments in the subtropics. To be submitted to *Indoor Air* (based on Chapter 5)
- Zhongping Lin and Shiming Deng. Sizing room air conditioners used in sleeping environments in the subtropics. Submitted to *ASHRAE Journal* (based on Chapter 6)

III. Conference papers

- Zhongping Lin and Shiming Deng. Sleeping thermal environment and bedroom air conditioning in residential buildings in subtropical areas. *9th International Conference on Air Distribution in Rooms (RoomVent2004)*, Coimbra, Portugal, September 5 – 8, 2004
- Zhongping Lin and Shiming Deng. A questionnaire survey on sleeping thermal environment and bedroom air conditioning in high-rise residences in Hong Kong. *The 3rd International Workshop on Energy and Environment of Residential Buildings*, Xian, China, November 21 – 24, 2004
- Zhongping Lin and Shiming Deng. A study on the applications of air conditioners in bedroom in subtropical areas. *2004' Chinese National Conference on Refrigerating & Air Conditioning*, Lanzhou, China, August 10 – 14, 2004 [in Chinese]