THERMAL COMFORT AND ENERGY PERFORMANCE OF PERSONALIZED VENTILATION WITH INDIVIDUAL CONTROL OF PERSONALIZED AIR MOVEMENT IN THE TROPICS

CHEN YIXING

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Summary

Personalized ventilation (PV) was developed to create a healthy and comfortable microenvironment with high energy efficiency. Personalized air, which is conditioned fresh air, is supplied directly to occupants' breathing zone. PV may therefore improve indoor air quality, decrease sick building syndrome symptoms, and improve occupants' thermal comfort. Moreover, PV can provide individual control of the personalized airflow rate, the airflow direction, personalized air movement and/or the personalized air temperature. However, the impacts of individual control of PV system on both thermal comfort and energy performance have yet to be studied in tropical climates.

The objectives of this research are to study the subjects' behavior in operating the controls and the impacts of PV-ICA (PV provided with Individual Control of Airflow rate) system on thermal comfort and energy performance. A subjective study with 46 tropically acclimatized subjects was conducted in a field environment chamber, which was served by a desktop-mounted PV system integrated with an ambient mixing ventilation system. The subjects were provided with individual control of the personalized air movement via control of the personalized flow rate and the direction of personalized air. Human responses to the PV-ICA system were collected through questionnaires.

Moreover, subjects' actions were recorded to evaluate their frequency of changes and preferred airflow rates. The data for energy calculation and simulation were recorded in building automation system (BAS) to evaluate the energy performance of the PV-ICA system. Hence, the results and observations are as follows:

Subjects' behavior in operating the controls under damper opening set-point control strategy was evaluated. The results showed that when the room ambient temperature was 26℃, the frequency of changes was lower and the preferred airflow rates were 17~31% higher compared to a room ambient temperature at 23℃. Moreover, the subjects' preferred airflow rates had a large range, thus indicating that the preferred air movement was over a quite large range.

The thermal comfort results of the PV-ICA system showed that more than 90% of subjects felt that the thermal environment was acceptable while more than 83% of subjects felt that the air movement on face, shoulder and neck were all acceptable. The acceptability of thermal comfort and air movement did not vary significantly with the room ambient temperature, PV temperature or control strategy. Moreover, at a higher room ambient temperature of 26℃, the subjects' whole body and body segments thermal sensations were closer to neutral, which may reduce draft risk around the human body as compared to that at a room ambient temperature of 23℃.

The energy performance results of the PV-ICA system showed that when the room ambient temperature was 23℃, the total cooling load was 9~14% higher than that at a room ambient temperature of 26℃.

In conclusion, it is found that in the tropics, the PV-ICA system can better improve occupants' thermal comfort and enhance energy saving at a room ambient temperature of 26℃ than at 23℃. The best case is that PV-ATD supply personalized air at 20℃ and the room is maintained at 26℃.

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Abbreviations

AFSP	=Airflow Rate Sep Point
AHU	=Air Handling Unit
ATD	=Air Terminal Device
BAS	=Building Automation System
BTM	=Breathing Thermal Manikin
COP	=Coefficient of Performance
DMF	=Desk Mounted Fans
DV	=Displacement Ventilation
DOSP	=Damper Opening Set-Point
ET	=Equivalent Temperature
FEC	=Field Environment Chamber
HVAC	=Heating, Ventilation and Air-Conditioning
IAQ	=Indoor Air Quality
MV	=Mixing Ventilation
PAQ	=Perceived Air Quality
PMV	= Predicted Mean Votes
PV	=Personalized Ventilation
PV-ICA	=PV with Individual Control of Airflow Rate
RH	= Relative Humidity

SBS = Sick Building Syndrome

TAC =Task/Ambient Conditioning

TV =Total Volume

UFAD = Under-Floor Air Distribution

Chapter 1: Introduction

1.1 Background and Motivation

Nowadays, people spend more than 90% of their time indoors (ASHRAE, 2003). The thermal environment and air quality in buildings affect occupants' health, comfort and performance (Tham, 2004; Wargocki et al. 2004a; 2004b; 2005). It is therefore very important to create a healthy and comfortable indoor microenvironment for occupants. The heating, ventilation and air-conditioning (HVAC) systems of buildings today are normally designed to create the above mentioned indoor microenvironment. HVAC system consumes more than 55% of overall building related energy in Singapore (Lee, 2004), while building energy consumption makes up 20%～30% of the total energy consumption (Building Energy-Saving Research Center of Tsinghua University, 2008). Under the global energy crisis, the concepts of "Low Carbon", "Energy Efficiency" and "Environment Friendliness" have become more and more popular and relevant. It is therefore important to design highly energy-efficient HVAC systems

Mixing ventilation (MV) and displacement ventilation (DV) are two main room air distribution systems which are currently used in buildings. In these systems, the clean air is supplied through diffusers that may be located far from the occupants. The supplied air is more or less polluted and warmed by the time it is inhaled by occupants. MV and DV systems aim for total volume (TV) ventilation, which are designed to provide a uniform room environment. Occupants may therefore be exposed to air that is more polluted and warmer than desired.

Large differences exist between occupants with respect to physiological and psychological response, clothing insulation, activity, air temperature and air movement preferences, etc (Melikov, 2004). The thermal insulation of the occupants' clothing may vary from 0.4 clo to 1.2 clo or even more and the metabolic rate may range between 1 met and 2 met due to differences in occupants' physical and mental activities (ASHRAE, 2001). Individual differences in preferred air temperature may be as great as 10 ℃ (Grivel & Gandas, 1991). Occupants' preferences for air movement (air velocity) may differ more than four times in temperate climates (Melikov et al., 1994) and range from 0.3 to 0.9 m/s in tropical climates (Gong et al., 2006). MV and DV do not account for individual differences between occupants and provide only limited or no personal control at all over their microenvironment. It is therefore not surprising that thermal discomfort is often reported by a large percentage of occupants in offices even when the thermal environment complies with the recommendations in the standards. Environmental conditions acceptable for most occupants in rooms may only be achieved by providing each occupant with the possibility to control his/her own preferred microenvironment (Melikov, 2004).

Considering the air quality in occupants' breathing zone and the variety of their preferred microenvironment, personalized ventilation (PV) can be a solution (Melikov, 2004). The concept of PV involves supplying conditioned fresh air close to breathing zone of each occupant without mixing it with re-circulated air. PV air terminal devices (ATD) are typically installed on desktops and occupants may be provided with individual control of the supplied airflow rate, the airflow direction and/or supplied air temperature (Melikov, 2002). Compared with TV ventilation, PV system has the potential to maintain a healthier environment in the occupied zone. PV can improve indoor air quality and occupants' thermal comfort, and decrease sick building syndrome (SBS) symptoms (Melikov, 2004 & 2002; Sekhar et al., 2005; Yang et al, 2008). Moreover, PV in conjunction with MV or DV decreases the risk of airborne transmission of infectious agents and is superior to TV ventilation alone (Cermak, 2006; 2007). Furthermore, Schiavon et al. (2009; 2010) have shown by means of energy simulation that PV has the potential for energy savings in both cold and tropical climates when proper control strategies are applied. Schiavon et al. (2010) shows that, in tropics, reducing the airflow rate due to the higher ventilation effectiveness of the PV system is an effective strategy for reducing the energy consumption. Energy consumption is further reduced by increasing the maximum allowed room air temperature.

Numerous laboratory measurements with either a thermal manikin or human subjects have been conducted to evaluate the ability of PV systems to enhance inhaled air quality and/or thermal comfort. Most of these studies have been conducted in temperate climatic zones. These studies will be reviewed in Chapter 2.

There are however very limited subjective studies of PV system in hot and humid climates (Li et al., 2010; Yang et al., 2010; Sekhar et al., 2005; Skwarczynski et al., 2010), especially with focus on individual control capability and energy performance. As of now, no study on human responses to PV with individual control in tropical climates has been reported in the literature.

This research therefore focuses on this area.

1.2 Research Objectives

This study aims to investigate the performance of PV system with individual control of airflow rate (PV-ICA). The subjects' behaviors in operating the controls and the impacts of the PV-ICA system on thermal comfort and energy performance are studied. The objectives of this study are as follows:

- To study the subjects' behavior such as the frequency of PV airflow rate adjustments and the characteristics of their preferred airflow rate, and consequently, preferred air movement.
- To evaluate human response to the PV-ICA system
- To explore the energy saving potential of the PV-ICA system.

1.3 Outline

This thesis is organized into five chapters in the following sequence:

Chapter one introduces the background and motivation of the research, research objectives and the organization of this thesis. The characteristics of conventional TV ventilation are discussed and the concept and advantages of PV system are introduced.

Chapter two provides the literature review, which consists of the following: (1) general introduction to conventional ventilation systems; (2) the concept of PV system; (3) experiments involving thermal manikin in PV system; (4) studies on human response to PV system; (5) energy performance of PV system; and (6) individual control of PV system. Finally, the knowledge gap identified in this study is discussed.

Chapter three presents the research methodology adopted in this study. The experimental facilities, experimental conditions, subject selection, questionnaire design and experiment design are introduced. The methods of data collection and analysis and the energy model are presented in this chapter as well.

Chapter four discusses the results of the study. First, an evaluation of subjects' behavior such as the frequency of airflow rate adjustments and their preferred airflow rates and air movement is presented. The impacts of the room air temperature, PV supply air temperature and control strategy on subjects' thermal comfort are then analyzed. Finally, the energy performance of the PV-ICA system is presented.

Chapter five covers the conclusions and the research contributions. The limitation of this study and recommendations for further research are presented as well.

Chapter 2: Literature Review

This chapter introduces basic concepts related to conventional ventilation systems and PV. Previous researches in the area of PV are reviewed and the knowledge gap is identified.

2.1 Conventional Ventilation Systems in the Tropics

Tropical zone is a region of the Earth near the Equator. It is limited in latitude by the Tropic of Cancer in the northern hemisphere at 23°26' (23.5°) N and the Tropic of Capricorn in the southern hemisphere at 23°26' (23.5°) S. About 40 percent of the world's human population lives within the tropical zone according to 2008 statistics (Wikipedia, 2010a). The characteristics of tropical climates include warm to hot, moist year-round and abundant rainfall throughout the year.

Singapore’s climate is classified as equatorial, with no true distinct seasons. Due to its geographical location and maritime exposure, its climate is characterized by uniform temperature and pressure, high humidity and abundant rainfall. The temperature hovers around a diurnal range of a minimum of 23℃ and a maximum of 31℃. Relative humidity (RH) has a diurnal range in the high 90% in the early morning to around 60% in the mid-afternoon. (Wikipedia, 2010b)

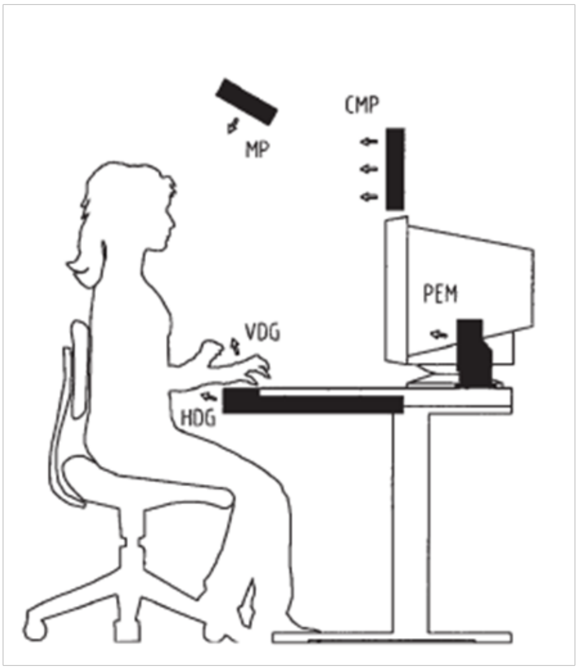
The conventional mechanical ventilation systems of buildings are designed to provide a uniform thermal environment. The two most commonly used air distribution systems in buildings are MV and DV.

MV aims to create a relatively uniform thermal and air quality environment over space. In MV systems, conditioned air is usually supplied and exhausted from ceiling-based air diffusers which may be located far away from the occupants. The conditioned air at a low contaminant concentration is mixed with the room air and may be polluted by the time it is inhaled by the occupants. Occupants in the space might therefore be exposed to mediocre air quality conditions compared to high quality outdoor air.

In DV systems, conditioned air is normally supplied at a very low velocity (less than 0.5 m/s) from a low sidewall or floor diffuser and exhausted at the ceiling level. The supplied air temperature is slightly lower (3~5℃) than the designed room air temperature. Since it is cooler than the room air, the supply air is spread over the floor and then rises as it contacts with heat sources in the occupied space, such as people, computers, etc. Air temperature and contaminant concentrations develop vertically, with cool and less contaminated air at low level and warm and more contaminated air at a higher level of the space. Therefore, DV may create a better indoor environment compared to MV system, especially in rooms with non-passive, heated contaminant sources (Brohus, 1996). However, in real office, diffusers are always far away from occupants and the supplied clean air is more or less polluted by materials and other emissions that are in contact with it. Melikov et al. (2005) did a field survey of occupants' response to the indoor environment in 10 office buildings with DV. About 24% of the occupants in the survey complained that they were bothered by draught, mainly at the lower leg. Almost one half (49%) of the occupants reported that they were bothered by an uncomfortable room temperature. Forty-eight per cent of the occupants were not satisfied with the air quality. Furthermore, the draft at feet caused by the cool supply air and thermal asymmetry due to vertical temperature gradients are prone to incur local thermal discomfort.

2.2 Concept of PV System

PV system, which is also known as desktop task/ambient conditioning (TAC) system, is a relatively new concept in air distribution and is yet to enjoy widespread adoption in office buildings. PV ATDs, as shown in Figure 2-1, are typically installed on desktops. PV systems can supply conditioned cool and fresh air directly to the breathing zone and have the potential to maintain a healthy indoor environment. Unlike conventional TV ventilation systems, the fresh air is less mixed with existing contaminated air before it is inhaled by occupants. Some PV ATDs provide individual control of the supplied airflow rate, the airflow direction and/or the supplied air temperature (Melikov et al., 2002; Melikov, 2004). PV system fulfills the principles of excellent indoor environments suggested by Fanger (2000), which included: 1) better indoor air quality increases productivity and decreases SBS symptoms; 2) unnecessary indoor pollution sources should be avoided; 3) the air should be served cool and dry to the occupants; 4) "personalized air"; and 5) individual control of the thermal environment. These principles of excellence comply with energy efficiency and sustainability. In practice PV are mostly applied together with one kinds of TV, which can be operated either as constant or variable air volume system.



MP: Movable Panel;
CMP: Computer Monitor Panel;
VDG: Vertical Desk Grill;
HDG: Horizontal Desk Grill;
PEM: Personal Environments Module.

Figure 2-1 Examples of some ATDs (Source: Melikov, 2004)

2.3 Experiments with Thermal Manikin in PV Systems

Thermal manikins have been adopted for research and development purposes for more than 60 years. The thermal manikins are widely used for assessing the impact of thermal environments on the human body (Holmér, 2004). In this section, the experiments with thermal manikin to evaluate the performance of PV systems are reviewed.

2.3.1 PV ATD Performance

The supply ATD is an essential part of any PV system. The ATD delivers conditioned outdoor air to end-users and determines the air characteristics in the breathing zone. It plays a major role in the distribution of air around human body and thus, determines occupants' thermal comfort and perceived air quality (PAQ). The performance of PV systems depends to a large extend on the supply ATD. Some studies which compare the performance of different PV ATDs are reviewed.

Melikov et al. (2002) studied the performance of five different ATDs as shown in Figure 2-1. A typical office workplace consisting of a desk with mounted ATDs was simulated in a climate chamber. A breathing thermal manikin was used to simulate a human being. Experiments at room air temperatures of 26℃ and 20℃ and personalized air temperatures of 20℃ supplied from the ATDs were performed. The flow rate of personalized air was changed from around 5 to 23 l/s. Tracer gas was used to identify the amount of personalized air inhaled by the manikin as well as the amount of exhaled air that was re-inhaled. The heat loss from the body segments of the thermal manikin was measured and used to calculate the equivalent temperature (ET) for the whole body as well as segments of the body. An index, personal exposure effectiveness, was used to assess the performance of ATDs with respect to the quality of the air inhaled by the manikin. The personal exposure effectiveness increased with the airflow rate from the ATD to a constant maximum value. A further increase of the airflow rate had no impact on the personal exposure effectiveness. Under both isothermal and non-isothermal conditions the highest personal exposure effectiveness of 0.6 was achieved by a vertical desk grill followed by an ATD designed as a movable panel. The ATDs tested performed differently in regard to the inhaled air temperature used as another air quality indicator, as well as in regard to the ET. The results suggested that PV may significantly decrease the number of occupants dissatisfied with the air quality. The amount of exhaled air re-inhaled by the manikin was rather small with all tested ATDs. The temperature of the inhaled air decreased with the increase of personalized airflow rate. The lowest temperature of the inhaled air was achieved by vertical desk grill. The vertical desk grill provided greatest cooling of the manikin’s head. In practice, this may cause draught discomfort for the occupants.

Sun et al. (2007) examined the performance of a circular perforated panel ATD for a PV system operating under two levels of turbulent intensity. The impact of turbulent intensity on spatial distribution of the cooling effect on the facial region and whole body were studied through experiments carried out in an indoor environment chamber using a breathing thermal manikin and 24 tropically acclimatized subjects. The PV system was adjusted to deliver treated outdoor air over a range of conditions, which were presented blind to the subjects in a balanced order. Over a 15-min exposure, subjects voted their thermal sensation experienced at the facial region and whole body. At each of the conditions, the near body flow field characteristics and heat loss rate on each of the 26 body segments of the manikin were measured. The results indicated that over the range of PV air supply volume studied, by controlling the temperature and velocity of PV air supply at 15 cm from the face, PV air supplied at lower turbulent intensity, when compared against that supplied at higher turbulent intensity, achieved a larger range of velocities at the face, a greater cooling effect on the head region as well as a lower facial thermal sensation, which had potential draft risks (when facial thermal sensation vote <-1).

In this study, a desk-mounted PV ATD as introduced in Section 3.2.2 was used. This PV ATD supplied air at low turbulent intensity and was similar to the ATD designed as a movable panel. The supplied air flow direction can be changed by turning the diffusers horizontally and vertically. However, the location of PV ATD is fixed.

2.3.2 PV in Conjunction with TV

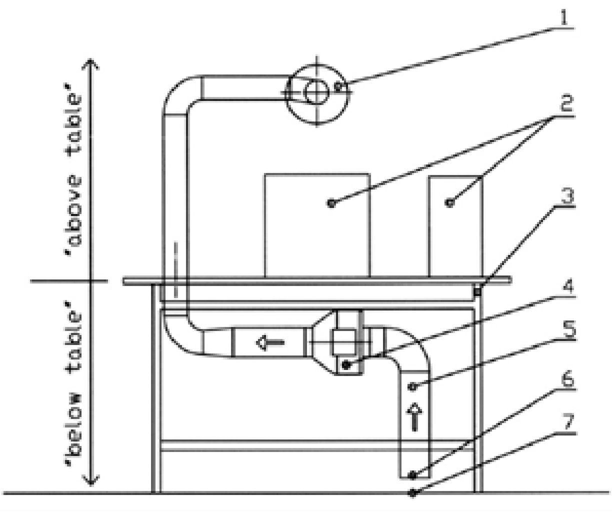
In practice, PV will most often be applied together with TV, which can be operated either as constant or variable air volume system (Melikov, 2004). Some studies which evaluate the performance of PV in conjunction with TV are reviewed in this study.

- **PV in conjunction with MV or DV**

Cermak et al. (2006) evaluated the thermal performance of two types of ATDs for a PV system in conjunction with either a mixing or a displacement TV ventilation system installed in a mock-up of an office. Two occupants were simulated by means of breathing thermal manikins. The results showed that PV always could always improve the inhaled air quality in rooms with MV. In rooms with DV, PV improved the inhaled air quality with regard to pollution emitted from the floor covering. The inhaled air quality related to human-produced contaminants, such as virulent agents associated with exhaled air or bioeffluents, might be improved as well. The improvement depended on the efficiency of the PV system and its ability to promote mixing. The experimental data showed that the design of ATDs for PV and their use had a large impact on the inhaled air quality and thermal comfort of occupants.

- **“Duct less” PV in conjunction with DV**

Halvoňová et al. (2010a; 2010b; 2010c) studied the performance of the novel “ductless” PV in conjunction with DV. The idea behind “ductless” PV was to utilize clean and cool air supplied via DV. The “ductless” PV installed at each desk consisted of an ATD mounted on a movable arm and a small axial fan incorporated in a short duct system (positions 1, 4 and 5 in Figure 2-2). The treated outdoor air supplied to the room near the floor by the DV spread in a relatively thin layer over the floor. The “ductless” PV sucked the clean air direct from this layer at the locations of the desks (position 6 in Figure 2-2) and transported it to the breathing zone of the manikins. The ATD used during the experiments was round movable panel (RMP). The movable arm, to which the RMP was attached, allowed for free positioning of the RMP.



(1) Round moveable panel
(2) Heat sources
(3) Desk
(4) Installed fan
(5) Short duct system
(6) Intake of “ductless” PV
(7) Floor level

Figure 2-2 “Ductless” PV
(Source: Halvoňová et al., 2010a)

Halvoňová et al. (2010c) studied the impact of workstations layout and partitions on the performance of the “ductless” PV in full-scale room experiments in conjunction with DV. The performance of the “ductless” PV system was evaluated under various arrangements of two identical workstations. Two breathing thermal manikins were used to simulate seated occupants. Two tracer gases, one mixed with the air exhaled by one of the manikins and the other generated on the table in front of the same manikin, were used to simulate pollution. When the “ductless” PV system was operational, the inhaled air was as clean as the air inhaled using only the DV alone and even cleaner for some of the layouts studied. The use of “ductless” PV in conjunction with DV substantially decreased the temperature of the inhaled air and increased the body cooling in comparison with use of DV alone. “Ductless” PV also had potential for improving occupants' PAQ and thermal comfort.

Halvoňová et al. (2010a) studied the impact of disturbances due to walking person(s) on the performance of the novel “ductless” PV in conjunction with DV. An office room with two workstations was arranged in a full-scale test room. Two thermal manikins were used as sedentary occupants at the workstations. Two pollution sources, namely exhaled air by one of the manikins and passive pollution on the table in front of the same manikin were simulated. The performance of the ventilation systems was evaluated by the quality of inhaled air and thermal comfort of the seated “occupants”. The walking person(s) caused mixing of the clean and cool air near the floor with the polluted and warmer air at higher levels and disturbed the displacement principle which resulted in a decrease of the inhaled air quality. The performance of the “ductless” PV under the tested conditions was better as opposed to DV alone. In all studied cases the inhaled air temperature with the “ductless” PV remained 2.9-4.5℃ lower than with DV used alone. Thus in practice the “ductless” PV was superior to DV alone in the aspect of perceived quality of inhaled air. The location of a walking person was found to be important. Person(s) walking close to the displacement diffuser can cause greater disturbance.

Halvoňová et al. (2010b) studied the importance of the intake positioning height above the floor level on the performance of “ductless” PV in conjunction with DV with regard to the quality of inhaled air and of the thermal comfort provided. A typical office room with two workstations positioned one behind the other was arranged in a full-scale room. Each workstation consisted of a table with an installed “ductless” PV system, PC, desk lamp and seated breathing thermal manikin. The “ductless” PV system sucked the clean and cool displacement air supplied over the floor at four different heights, i.e. 2, 5, 10 and 20 cm and transported it directly to the breathing level. Moreover, two displacement airflow rates were used with an adjusted supply temperature in order to maintain an exhaust air temperature of 26℃. Two pollution sources, namely air exhaled by one of the manikins and passive pollution on the table in front of the same manikin were simulated by constant dosing of tracer gases. The results showed that the positioning of a "ductless" PV intake height up to 0.2 m above the floor did not significantly influence the quality of inhaled air and thermal comfort. In all studied cases the inhaled air temperature with the "ductless" PV remained 4.9-8℃ lower than with DV alone.

- **PV in conjunction with underfloor ventilation**

Cermak et al. (2007) studied the performance of two PV systems in conjunction with underfloor ventilation, which generated two different airflow patterns in a full-scale test room. Two breathing thermal manikins were used to simulate occupants. The distribution of pollutants associated with exhaled air and floor material emissions was evaluated at various combinations of personalized and underfloor airflow rates. Compared to underfloor ventilation alone, personalized and underfloor ventilation provided excellent protection of seated occupants from any pollution, while the concentration of exhaled air pollution increased in the room. The two types of PV performed differently. Subsequent analyses of airborne infection transmission risk indicated that PV could become a supplement to traditional methods of infection control.

- **Ceiling mounted PV in conjunction with MV**

Yang et al. (2009) studied the interaction of the personalized air supplied from ceiling mounted nozzle (diameter of 0.095 m) with the thermal plume generated by a seated thermal manikin with the body size of an average Scandinavian woman and the impact of the personalized airflow rate on the body cooling. Experiments were performed in a test room with MV under numerous conditions comprising four combinations of room air temperature and personalized air temperature (23.5℃ / 21℃, 23.5℃ / 23.5℃, 26℃ / 23.5℃, 26℃ / 26℃), four airflow rates of the personalized air (4, 8, 12, 16 L/s) and positioning of the manikin directly below the nozzle (1.3m distance between the top of manikin's head and the nozzle). The asymmetric exposure of the body to the personalized flow was studied by moving the manikin 0.2m forward, backward and sideward. The blockage effect of the unheated manikin on the personalized airflow distribution, studied at the case 23.5℃ / 23.5℃, was clearly observed 0.2m above the top of manikin's head where the centerline velocity was reduced to about 85% under all personalized airflow rates. The neutral level, X_{nl}, defined as the distance from the nozzle where the impact of the thermal plume on the velocity distribution in the personalized air flow was observed, increased from 0.8m to 1.1m as the airflow rate increased. The thermal plume was completely destroyed when the personalized airflow rate was large than 16L/s. In comparison with the reference case without personalized air flow, the manikin based ET for the head decreased with the increase of the airflow rate from -1℃ to -6℃ under 23.5℃ / 21℃ case and from -0.5℃ to -4℃ under 26℃ / 26℃ case, which are the two extreme cases among the four cases studied. The personalized air flow was least efficient to cool the body when the manikin was moved forward.

In this research, a desktop mounted PV system integrated with an ambient MV system was examined.

2.3.3 Indoor air quality and thermal comfort

Compared with TV ventilation, PV system has the potential to maintain a healthier environment closer to the occupied zone. Some studies which indicate that PV can improve indoor air quality and occupants' thermal comfort, and decrease SBS symptoms, are reviewed below.

Niu et al. (2007) evaluated a chair-based PV system which potentially was applied in theatres, cinemas, lecture halls, aircrafts, and even offices. Air quality, thermal comfort, and the human response to this ventilation method were investigated by experiments. By comparing eight different ATDs, it was found that inhaled air contained as much as 80% of fresh personalized air when the supply flow rate of less than 3.0 l/s. PAQ improved greatly by serving cool air directly to the breathing zone. Feelings of irritation and local drafts could be eliminated by proper designs. Personalized air at a temperature below that of room air was able to bring “a cool head” and increased thermal comfort in comparison with MV.

Faulkner et al. (1999) investigated two TAC systems with heated thermal manikins seated at desks through laboratory experiments. The personalized air was supplied form desk-mounted air outlets directly to the breathing zone of the thermal manikins. Air change effectiveness was measured with a tracer gas step-up procedure. High values of air change effectiveness (~1.3 to 1.9) and high values of pollutant removal efficiency (~1.2 to 1.6) were measured when these task conditioning systems supplied 100% outdoor air at a flow rate of 7 to 9 Ls^{-1} per occupant. Air change effectiveness was reasonably well correlated with the pollutant removal efficiency. Overall, the experimental data suggest that these TAC systems can be used to improve ventilation and air quality or to save energy while maintaining a typical level of IAQ at the breathing zone.

Bolashikov et al. (2009) developed methods for control of the free convection flow around the human body. The objective was to improve the quality of the inhaled air for occupants at workstations with PV. Two methods of control were developed and explored: passive control, which meant to block the free convection development by modifications in desk design, and active control, which meant by local suction below the desk. The effectiveness of the two methods for enhancing the performance of PV was studied when applied separately and combined, and was compared with the reference case of PV alone. The experiments were performed in a full-scale test room with background MV. A thermal manikin with realistic free convection flow was used. The PV supplied air from front/ above towards the face. All measurements were performed under isothermal conditions at 20℃ and 26℃. The air in the test room was mixed with tracer gas, while personalized air was free of it. Tracer gas concentration measurements were used to identify the effect of controlling the free convection flow on inhaled air quality. The use of both methods improved the performance of PV and made it possible to provide more than 90% of clean air for inhalation at a substantially reduced PV supply flow rate.

Bolashikov et al. (2010) reported on methods for control of the free convective flow around the human body. The objective was to improve inhaled air quality for occupants at workstations with PV. Two PV nozzles were placed sideways at the head level of a seated occupant. Another pair of control nozzles below the PV nozzles, the height of the shoulders, either provided an additional amount of clean PV air or exhausted part of the air from the free convective flow. The effectiveness of the methods for enhancing the quality of the inhaled air was studied in full-scale room experiments. A thermal manikin with a realistic free convective flow was used to resemble an occupant in a state of thermal neutrality at a sedentary activity level. Numerous experiments comprising different combinations of nozzle sizes, supply and exhaust flow rates, and direction of the supplied PV flows and of the control flows, etc., were performed under isothermal conditions at 20℃ and 26℃. The methods of control proved to be efficient and made it possible to increase the amount of clean air into inhalation at reduced personalized flow rate and to reduce the risk of draft.

Zeng et al. (2005) studied the relationship between PAQ and the characteristics of the air jet from a moveable outlet of a PV system using a thermal manikin with breathing function in a climate chamber. The personal exposure effectiveness was based on concentrations of trace gas in the chamber and in the manikin nose. Results indicated that the personal exposure effectiveness was affected more by the distance between the movable outlet and the occupant's breathing zone than by the personalized airflow rate and did not change much for the personalized airflow rate higher than 10 L/s when the distance is fixed.

Nielsen et al. (2007) described an investigation made in a room ventilated by an air distribution system based on a textile terminal. The air distribution in the room was mainly controlled by buoyancy forces from the heat sources, although the flow from the textile terminal could be characterized as a displacement flow with a downward direction in areas of the room where no thermal load was present. The system was extended by a PV system to study the improved protection of people in a room. The investigation involved full-scale experiments with two breathing thermal manikins. One manikin was the source and the other was the target. In general it was found that when the air was supplied from the textile terminal alone, the flow in the room was fully mixed with limited protection of the occupants. Selected locations of supply, return, and heat sources could produce a displacement flow in the room with increased protection of the occupants. It was shown that PV improved the protection of occupants by increasing the personal exposure index.

2.3.4 Studies using computational fluid dynamics (CFD)

CFD is one of the branches of fluid mechanics. It uses numerical methods and algorithms to solve and analyze problems which involve fluid flows. Some studies using CFD are reviewed as follows.

Yang and Sekhar (2007) studied a new approach supplying fresh air directly by utilizing high velocity circular air jet without mixing with recirculated air. Objective measurements and CFD tool were used to evaluate corresponding indoor parameters to verify that it could both supply fresh air into occupied zone effectively and avoid draught rating. It was found that the measured air velocities were within the limits (0.25 m/s) of thermal comfort standards, although they were close to the limits. Higher air change rate could be obtained in breathing zone than that in ambient air in the background area. The predicted results showed unique distributions of airflow characteristics and were in fair agreement with empirical measurements. Different angles of recirculated air diffuser blades, different lengths and directions of protruding fresh air jets and different inlet velocities of fresh air were adopted for comparing the effectiveness and efficiency of this new ventilation strategy numerically.

Tham et al. (2010) examined the performance of a coupled system of desktop PV ATD and desk mounted fans (DMF) in a field environmental chamber. Cooling effect was evaluated using manikin-based ET of each of the 26 body segments of a breathing thermal manikin (BTM) and personal exposure effectiveness was used as an indicator for effectiveness of ventilation. CFD was used to examine the velocity field generated around BTM to provide better understanding of the relationship between air patterns generated and convective cooling effect on each of the body segments. Four different positions of desktop PV ATD were examined. Measurements were conducted at room ambient temperature of 26 ℃ and PV air temperature of 23 ℃ at a flow rate of 10 L/s. The results indicated that coupling of desktop PV ATD and DMF distributes cooling more uniformly across BTM surfaces and therefore had the potential to reduce risk of draft discomfort as compared to usage of desktop PV ATD alone.

Conceição et al. (2010) evaluated the comfort level, namely the thermal comfort, local thermal discomfort and air quality levels, in a classroom with desks equipped with two PV systems, in slightly warm environments. A manikin, a ventilated classroom desk, two indoor climate analyzers, a multi-nodal human thermal comfort numerical model and a CFD numerical model, were used. Each PV system was equipped with two ATDs. One was located above the desk writing area, in front to the trunk area, and another was located below the desk writing area, in front to the legs area. The actual PV system, which promoted an ascendant airflow around the occupant with highest air renovation rate in the respiration area, promoted acceptable thermal comfort conditions and air quality in the respiration area in accordance with the present standards. The draught risk was verified in the head and left leg while the uncomfortable air velocity equivalent frequency was verified in the left arm. The left human body sections presented higher local thermal discomfort levels than the right human body sections, because they were also influenced by the left ATD. In accordance with the obtained results, the classroom desk design, equipped with two PV systems, guaranteed acceptable thermal comfort conditions and promoted good air quality conditions, with acceptable local thermal discomfort conditions and with low energy consumption level.

Gao et al. (2004) studied the micro-environment around human body with and without PV system using a seated computational thermal manikin with geometry of a real human. Two novel evaluation indices, pollutant exposure reduction and personalized air utilization efficiency, were introduced. In the range of the personalized airflow rate from 0.0 to 3.0 l/s, the best inhaled air quality, where maximum PER was 74%, was achieved at the airflow rate of 0.8 l/s in the numerical simulation, whereas in the experiments this occurred at the maximum flow rate 3.0 l/s.

Zhao et al. (2007) studied the dispersion of particles with aerodynamic diameter of 0.5-10 μm in a room with PV system by CFD. After experimental validation, a three-dimensional model was employed to simulate particle dispersion, the airflow rate, and the air temperature around a human body. The cases of PV and MV under different air supply volumes were simulated. For the particles smaller than 2 mm, a strong-enough PV which could disperse the thermal plume was an effective ventilation mode to remove particles. The PV with less air supply volume had no obvious advantage compared to ceiling supply. For the particles bigger than 7.5 mm, PV might not be the best ventilation mode. It might have bigger particle concentration in breathing zone than that of ceiling supply ventilation, and resulted in obvious particle accumulation on the floor.

Khalifa et al. (2009) presented an investigation of the design and performance characteristics of PV systems in combination with general ventilation. The PV systems delivered high quality air to the breathing zone with no more clean air supply than that indicated by ANSI/ASHRAE 62.1-2004. Under these conditions, the energy used for conditioning the clean air would not exceed that of a conventional ventilation system. A novel PV nozzle was introduced which achieved high breathing zone air quality with a small fraction of the clean air as indicated by the ANSI/ASHRAE Standard. Tracer gas experimental results demonstrated the advantages of the novel nozzle comparing to conventional PV nozzles. The results showed that, at a PV clean air supply of only 2.4 l/s, the new nozzle achieved breathing zone ventilation effectiveness close to 7 versus less than 2 for a conventional nozzle delivering the same amount of clean air. Russo et al. (2009) presented a computational analysis of the same concept, which was validated against the experimental results. The CFD model showed excellent agreement with the experimental data and was used to study the effect of nozzle exit boundary conditions such as turbulence intensity and length scale, flow rate and temperature, and manikin temperature on the air quality in the breathing zone of the heated manikin. The results showed that the air quality of the novel PV system was sensitive to the nozzle exit turbulence intensity and flow rate, and insensitive to jet temperature within the 20-26 °C range, and to body temperature within a clo range of 0-1. Russo et al. (2010) used the validated CFD model to calculate the inhalation exposure, expressed as an intake fraction, of a seated person in an office with different contaminant sources, a floor diffuser, and a ceiling vent. These sources included the floor, the walls, a desk, and the human body. First, experimental data was used to determine the correct turbulent Schmidt number for the computational model to predict the transport of the species in an indoor environment. It was found the best fit was produced at a turbulent Schmidt number of ~0.9 when compared to experimental data. Then, the intake fraction was calculated for two representations of the computer simulated person: a computer simulated person with detailed surface geometry, and a simplified computer simulated person with multi-block geometry. It was found that the simplified multi-block geometry was not adequate for predicting intake fraction because it radically changes the flow field of the thermal plume in the breathing zone. Next, the effect of personal ventilation systems on intake fraction was investigated. The results showed that such systems could reduce the intake fraction by an order of magnitude compared with conventional mixing and displacement ventilation systems. Finally, a comparison of intake fraction results was made for a surface body temperature of 32℃ and 28℃. It was found that a 4℃ change in body surface temperature influenced the intake fraction by less than 10%.

2.4 Studies on human response to PV system

PV in conjunction with TV ventilation system has the potential to improve occupants' PAQ, thermal comfort, decrease the occurrence of effects such as the SBS symptoms and reduce the risk of transmission of contagion between occupants in comparison with TV ventilation alone (Melikov, 2004). Some studies on human response to PV system are reviewed in this section.

2.4.1 Studies in tropical climates

Li et al. (2010) studied the potential for improving occupants' thermal comfort with PV system combined with under-floor air distribution (UFAD) system through subjective study in tropical climate. Cold draught at feet could be reduced when relatively warm air was supplied by UFAD system and uncomfortable sensation as "warm head" could be reduced by the PV system providing cool and fresh outdoor air at the facial level. The analyses of the results obtained revealed improved acceptability of PAQ and improved thermal sensation with PV-UFAD in comparison with the reference case of

UFAD alone or MV with ceiling supply diffuser. The local thermal sensation at the feet was also improved when warmer UFAD supply air temperature was adopted in the PV-UFAD system.

Yang et al. (2010) studied human responses to a newly developed ceiling-mounted PV system with thirty-two tropically acclimatized subjects in a Field Environmental Chamber. Subjective assessments of thermal environment, air movement, and air quality were performed. Subjects performed normal office work and could choose to be exposed to four different PV airflow rates (4, 8, 12, and 16 L/s), thus offering themselves a reasonable degree of individual control. Room ambient temperatures of 26 and 23.5°C and PV air temperatures of 26, 23.5, and 21°C were employed. The ceiling-mount PV system offered a practical solution to the integration of PV ATDs with the workstations. The results showed that the local and whole body thermal sensations were reduced when PV airflow rates were increased. Moreover, when PV airflow rate was increased or its temperature was reduced, perceived inhaled air temperature was cooler and PAQ and perceived air freshness were improved.

Sekhar et al. (2005) studied the ability of PV systems to enhance thermal comfort, indoor air quality (IAQ) and energy savings under the hot and humid tropical conditions of Singapore. The results indicated that the use of a secondary PV system in conjunction with a primary air-conditioning system not only enhanced thermal comfort and IAQ acceptability but could reduce energy consumption by 15-30%. PV systems could improve ventilation effectiveness at the immediate breathing zone by up to 50% more than that could be obtained with MV alone, and tended to lower the average temperature of inhaled air in the breathing zone by 2-5 ℃, thus enhancing PAQ. PV temperature and PV flow rate were found to be more critical than room ambient temperature for occupants' thermal comfort and the PAQ of inhaled air.

Skwarczynski et al. (2010) studied the effect of facially applied air movement on PAQ at high humidity. Thirty subjects (21 males and 9 females) participated in three, 3-h experiments performed in a climate chamber. The experimental conditions covered three combinations of RH and local air velocity under a constant air temperature of 26℃, which are: 70% RH without air movement; 30% RH without air movement; and 70% RH with air movement under isothermal conditions. PV was used to supply room air from the front toward the upper part of the body (upper chest, head). The subjects could control the flow rate (velocity) of the supplied air in the vicinity of their bodies. The results indicated air flow with elevated velocity applied to the face significantly improved the acceptability of the air quality at the room air temperature of 26℃ and RH of 70%.

2.4.2 Studies in temperate climates

Kaczmarczyk et al. (2004b) compared the responses of 60 human subjects to a PV system with the responses to MV system. The PV system was provided with control of positioning of the ATD and the airflow rate. PAQ, thermal comfort, intensity of SBS symptoms and performance of subjects were studied during 3 h 45 min exposures. In the case of MV alone, the room air temperature was 23°C and 26°C. The PV system supplied outdoor air at 23°C or 20°C or recirculated room air at 23°C when the room temperature was 23°C and outdoor air at 20°C when the room temperature was 26°C. The PV system providing outdoor air improved PAQ and decreased SBS symptoms compared to either MV alone or the PV system which provided re-circulated air. The percentage dissatisfied with air quality, 3 min after initial occupancy, decreased from 22% with MV to 7% with PV system; and from 49% to 20%, at room temperatures 23°C and 26°C, respectively. Over time, these differences in percentage dissatisfied decreased markedly. Headache and decreased ability to think clearly were reported as least intense when the outdoor air was supplied through PV system at 20℃, while the most intense symptoms occurred with MV. PV system increased self-estimated performance.

Kaczmarczyk et al. (2006) studied human response to five different ATDs for a PV system in an experimental office under well-defined conditions. A group of 30 human subjects assessed air quality and rated their thermal comfort and perception of draft. Two temperature levels, 23℃ and 26℃, and two background levels of air quality in the office, high and low, were studied during the experiments. Under all conditions the PV system provided outdoor air at a temperature of 20℃. All PV systems provided individual control of the airflow rate as well as adjustments of the supply air direction. Results showed that all ATDs studied significantly improved PAQ and thermal comfort at the workstation. The greatest improvement was obtained when the pollution level and the temperature in the office were high. The thermal environment created with all systems was assessed as acceptable. Subjects were able to improve thermal comfort with all ATDs studied.

Kaczmarczyk et al. (2010) studied the human response to air movement supplied locally towards the face in a room with an air temperature of 20℃ and a RH of 30%. Thirty-two human subjects were exposed to three conditions: calm environment and facially supplied air flow at 21℃ and at 26℃. The air was supplied with a constant velocity of 0.4 m/s by means of PV towards the face of the subjects. The air flow at 21℃ decreased the subjects' thermal sensation and increased draught discomfort, but slightly improved the PAQ. The air flow at 26℃ decreased the draught discomfort, improved subjects' thermal comfort and only slightly decreased the PAQ. Elevated velocity and temperature of the localized airflow caused an increase of nose dryness intensity and number of eye irritation reports. Results suggest that increasing the temperature of the air locally supplied to the breathing zone by only a few degrees above the room air temperature would improve occupants' thermal comfort and would diminish draught discomfort. Providing individual control was essential in order to avoid discomfort for the most sensitive occupants.

Bauman et al. (1998) studied the impact of installing a TAC system at 42 selected workstations within three San Francisco office buildings occupied by a large financial institution. Field measurements were performed both before and after the TAC system installation to evaluate the impact of the TAC system on occupant satisfaction and thermal comfort. During the follow-up field tests, measurements were repeated under three different room temperature conditions. The result indicated that among the six building assessment categories, installation of the TAC system provided the largest increases in overall occupant satisfaction for thermal quality, acoustical quality, and air quality.

Yang et al. (2003) investigated the impact of fluctuating airflow on human thermal comfort and indoor air quality in a PV system. Three periodically fluctuating airflows with frequencies of 0.1, 0.2 and 0.3 Hz were used as well as airflow with constant velocity. The test room had an air temperature of 28℃ with a personalized air temperature of 25℃ during the test period. The results showed that subjects preferred to select the airflow with a frequency of 0.2 Hz. The subjects had the same level of thermal comfort and PAQ when they were separately exposed to the constant airflow and the preferred fluctuating airflow (0.2 Hz).

Amai et al. (2007) conducted subjective experiments with different task conditioning systems to investigate the effect of three different types of TAC systems on thermal comfort in a climate chamber. The chamber was conditioned at 28℃/50% RH with task systems and 26℃/50% RH without them. Under the condition with the task conditioning systems, the average rating of comfort sensation was between comfortable (0) and slightly uncomfortable (-1). For males it was between -0.5 and -0.7 and for females it was between -0.3 and -0.4. This was equal to or even better than that in condition without task system. It was found that these systems were effective in providing thermal comfort. However, the individual control of task conditioning system contributed to create the preferred environment.

2.5 Energy performance of PV system

Little is known about the energy use of PV system. In this section, some studies related to the energy saving potential of PV system are reviewed.

Schiavon et al. (2010) investigated the energy consumption of a PV system used in an office building, which was located in a hot and humid climate (Singapore) through simulations with the IDA Indoor Climate and Energy software. The results revealed that the use of PV might reduce the energy consumption substantially (up to 51%) compared to MV when the following control strategies were applied: (a) reducing the airflow rate due to the higher ventilation effectiveness of PV; (b) increasing the maximum allowed room air temperature due to PV capacity to control the microclimate; and (c) supplying the outdoor air only when the occupant was at the desk. The results also showed that the strategy to control the supply air temperature did not affect the energy consumption in a hot and humid climate, because the outdoor air which was hot and humidity always need to be cooled before it can be supplied.

Schiavon and Melikov (2009) studied the influence of the personalized supply air temperature control strategy on energy consumption and the energy-saving potentials of a PV system, which was introduced in a high quality Scandinavian office building located in a cold climate, through simulations with IDA Indoor Climate and Energy software. The energy consumption with PV might increase substantially (in the range: 61–268%) compared to MV alone if energy-saving strategies were not applied. The results showed that temperature control of the supplied personalized air had a marked influence on energy consumption and the best supply air temperature control strategy was to provide air constantly at 20 ℃. The most effective way of saving energy with PV was to extend the upper room operative temperature limit to 30 ℃ (saving up to 60% compared to the reference case of MV alone at room temperature of 25.5℃). However, this energy-saving strategy could only be recommended in a working environment where the occupants spent most of their time at their workstation. Reducing the airflow rate did not always imply a reduction of energy consumption. It was not an effective energy-saving strategy to supply the personalized air only when the occupant was at the desk in the cold climate.

Schiavon et al. (2008) studied the potential saving of cooling energy by increasing air speed through simulations with EnergyPlus software. Increasing air speed may offset the impact of increased room air temperature on occupants' comfort, as recommended in the present standards (ASHRAE Standard 55, 2004). Fifty-four cases covering six cities (Helsinki, Berlin, Bordeaux, Rome, Jerusalem and Athens), three indoor environment categories I, II and III (according to standard EN 15251 2007) and three air velocities (<0.2, 0.5 and 0.8 m/s) were simulated. The required cooling/heating energy was calculated assuming a perfectly efficient HVAC system. A cooling energy saving between 17% and 48% and a reduction of the maximum cooling power in the range 10–28% had been obtained. The results revealed that the required power input of the fan was a critical factor for achieving energy saving at elevated room temperature. Under the assumptions of this study, the energy saving might not be achieved by increasing air speed through a series of methods, such as ceiling, standing, tower and desk fans widely used today with a power consumption higher than 20 W.

Seem and Braun (1992) compared the energy use characteristics of systems incorporating personal environmental control with conventional designs through simulations. This study showed up to a 7% savings and a 15% penalty in building lighting and HVAC electrical use could be achieved by using the personal environmental control systems compared to conventional systems.

Pan et al. (2005) evaluated the performance of a personalized air-conditioning system, namely an innovative partition-type fan-coil unit, against a central air-conditioning system, in terms of their thermal comfort provided and cooling energy consumed. For a cooling load given, it was found that the thermal comfort index, predicted mean votes (PMV), resulted from the personalized system was always lower than that from a central system. Also, the PMV-curve of the personalized system responded to the loads faster. The experimental results indicated that the personalized system, as compared to the central system, could shorten the operation time for the same level of thermal comfort required and save up to 45% of the energy consumed by the central system.

2.6 Individual control performance of PV system

Air velocity was identified to be a crucial factor of draft risks, and recommendations on individual control of personalized airflow rate had been suggested by Melikov (2004). The possibility for individual control is one of the most important features of PV. Occupants may be provided with the control of temperature, velocity (flow rate) and/or direction of the personalized flow. Individual control may be assumed to have a psychological impact on occupants. Occupants' complaints decrease and their satisfaction with the local environment increases when they are provided with individual control (Bauman et al., 1998). Individually controlled microenvironment with local heating/cooling of the body has the potential to satisfy more occupants in a space compared to a TV uniform environment that is typically used at present. Some studies related to the individual control performance are reviewed in this section.

Melikov and Knudsen (2007) studied the response of 48 subjects to an individually controlled microenvironment with a room air temperature of 20°C, 22°C and 26°C in cold climates. The individually controlled system comprised PV, an under-desk ATD supplying cool air, a chair with convectively heated backrest, an under-desk radiant heating panel, and a floor-heating panel. The temperature of the air supplied from the PV and the under-desk device were both 20℃. The subjects were provided with control of the flow rate and direction of the personalized air, the under-desk airflow rate, the temperature of the convection flow from the chair, and the surface temperature of the heating panels. The results revealed that the thermal and air quality acceptability was significantly higher with the individually controlled system at all room temperatures compared to the reference condition at a room temperature of 22℃without individually controlled system. Thus, individually controlled system would increase the number of satisfied occupants when applied in practice. The design and control of the individually controlled system, as well as the background air distribution in a room, should be carefully considered in order to obtain the maximum number of occupants who were comfortable with their microenvironment. Watanabe et al. (2010) studied the same individually controlled system used in Melikov and Knudsen (2007) by using a thermal manikin at room temperature 20℃, 22℃ and 26℃ in cold climates. The results showed the whole-body ET ranged from 21.6℃ to 29.7℃ according to the preferred settings of the individually controlled system obtained from subjective experiments. The range of segmental ET, which are needed for preferred local thermal comfort of the body parts, was also wide but different for different body parts.

People learn, exercise successfully and benefit from their control over the flow velocity and direction and positioning of an ATD (Kaczmarczyk et al., 2002, 2004a; Yang et al., 2003; etc.). A tendency to make fewer changes in the positioning over elapsed time had been observed. The freedom of control over direction and flow rate of personalized air was important to lower the risk of draught sensation and to improve occupants' satisfaction. Personalized airflow toward the face was preferred than airflow towards the abdomen, although airflow from the side had been used as well. The preferred flow rate ranged from 3 to 20 L/s (0.2–1.2 m/s). Factors such as ergonomics, appearance, easy control, etc. were also important for subjects' ranking of the performance and acceptance from PV systems of different designs. (Kaczmarczyk, 2003; Kaczmarczyk et al., 2004a)

2.7 Knowledge gap

The review presented above indicates that PV in conjunction with TV can greatly improve indoor air quality, decrease SBS symptoms, and improve occupants' thermal comfort compared with TV alone.

The possibility of individual control is one of the most important features of PV. Occupants may be provided with control of temperature, velocity (flow rate) and/or direction of the personalized flow. Some researches (Kaczmarczyk et al., 2004b; 2006; Melikov et al., 2007) conducted in temperate climates have studied the thermal performance of the PV systems when occupants are provided with individual control. There have been no subjective studies of PV system with individual control of airflow rate in hot and humid climates. Differences may be found in thermal comfort, thermal sensation, air movement perception, acceptability, and preference of tropically acclimatized subjects compared with subjects in temperate climates, due to differences in their physiological acclimatization, clothing, behavior, habituation and expectation. In this study, the thermal comfort of the PV-ICA system in tropical climates is presented.

Little is known about the energy performance of PV system. Schiavon et al. (2008; 2009; 2010) studied the energy performance of the PV system through energy simulation. There have been no experiments under laboratory conditions which focus on the energy performance of PV system in tropical climates. Schiavon et al. (2010) indicated two strategies of energy saving in tropical climates: reducing the airflow rate due to the higher ventilation effectiveness of PV; and increasing the maximum allowed room air temperature due to PV capacity to control the microclimate. However, these two strategies seem unable to be used at the same time. As shown in Figure 2-3, when the ambient room temperature increases, subjects may require more air flow rate to offset the increased temperature without significantly affecting their thermal comfort (Schiavon et al., 2008). In temperate climates, the increase of air flow rate may not directly cause increase of energy consumption, as the outdoor air may have a free cooling effect (Schiavon et al., 2009). However, in tropical climates, the increase of outdoor air flow rate will always bring in extra ventilation load and thus will increase the energy consumption of the HVAC system. In this study, experiments under laboratory conditions with human subjects are conducted to evaluate the energy performance of PV system in tropical climates.

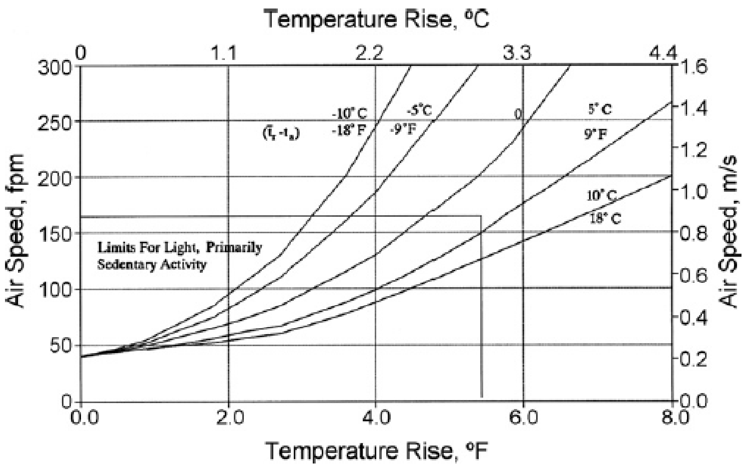


Figure 2-3Air speed required to offset increased temperature (Source: ASHRAE Standard 55, 2004)

A set of experiments were conducted in this research to evaluate the thermal comfort and energy performance of a PV with individual control of air flow rate (PV-ICA) system in tropical climates. The hypotheses of this study are as follows:

- The subjects' preferred airflow rates are quite different.
- Subjects require higher air flow rate when room ambient temperature is warmer.
- PV-ICA system can greatly improve occupants' thermal comfort as well as indoor air quality in the tropical climates context.
- PV-ICA system has significant potential to reduce energy consumption in tropical climates.

Chapter 3: Research methodology

3.1 Research hypothesis

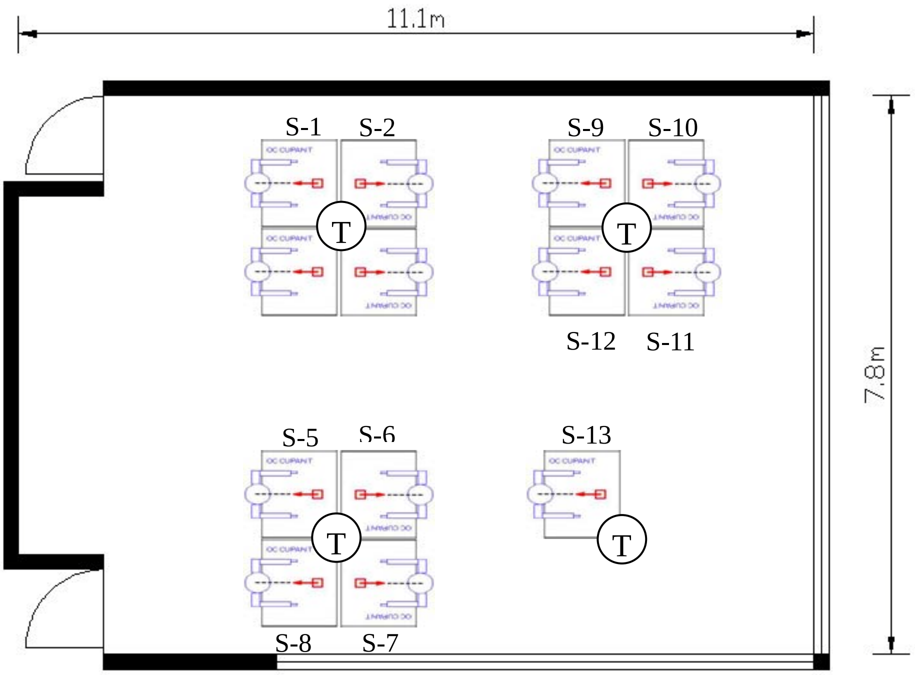
As introduced in Section 1.2, the objectives of this study are to evaluate subjects' preferred airflow rate, human responses to the PV-ICA system, and energy performance of the PV-ICA system. Based on the research objectives, subjective study is adopted in this research. The hypotheses of this study are described as follows:

- The subjects' preferred airflow rates are quite different.
- Subjects require higher air flow rate when room ambient temperature is warmer.
- PV-ICA system can greatly improve occupants' thermal comfort as well as indoor air quality in the tropical climates context.
- PV-ICA system has significant potential for energy savings in tropical climates

3.2 Experimental facilities

3.2.1 Field Environment Chamber

The experiments were conducted in a Field Environment Chamber (FEC), an experimental facility at School of Design and Environment, National University of Singapore. Figure 3-1 shows the layout of the FEC, which has 11.1 m (L) × 7.8 m (W) × 2.6 m (H) clear space. There are 13 workstations in the FEC. Each of the workstation is equipped with a PC and a desk-mounted PV ATD. More details are provided in Section 3.8.



T : Ambient air temperature and RH measurement points

S-1~13: Workstation 1~13.

Figure 3-1 Layout of the field environment chamber

3.2.2 Ventilation system

In this study, a desktop-mounted PV system in conjunction with an ambient MV system is used. Figure 3-2 shows the schematic diagram of this ventilation system. The FEC is served by two dedicated systems: an ambient air handling unit (AHU) with 100% recirculated air that is supplied through ceiling outlets and a PV AHU with 100% outdoor air that is supplied through the PV ATDs.

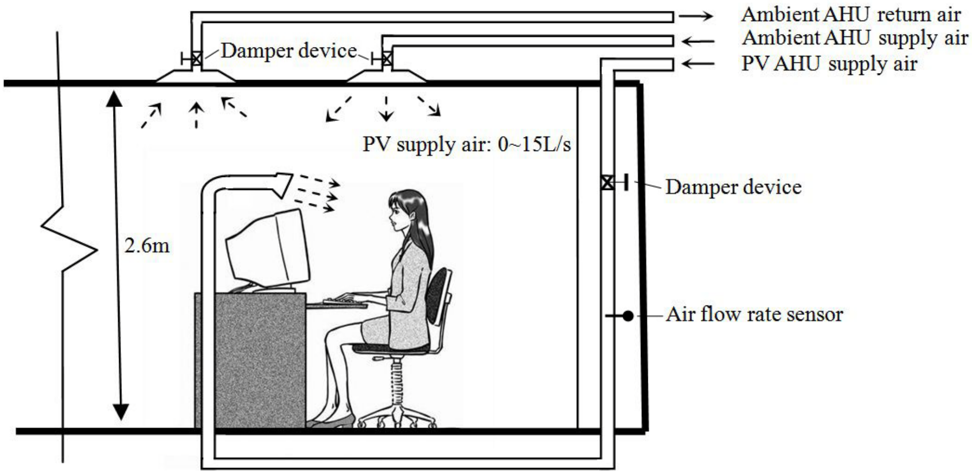


Figure 3-2 Ventilation system in FEC

- **Ambient AHU**

Figure 3-3 shows a schematic of the ambient AHU, to provide mixing ventilation. In this AHU, the exhaust air damper and the outdoor air damper are both closed with 100% recirculated air supplied via the ceiling diffusers.

The MV aims to keep the ambient room air temperature in the chamber uniform. Four ambient room temperature and RH sensors are placed in the chamber close to the ceiling. The measured air temperatures and RH are used to control the airflow rate supplied by the MV system through opening or closing the supply air damper.

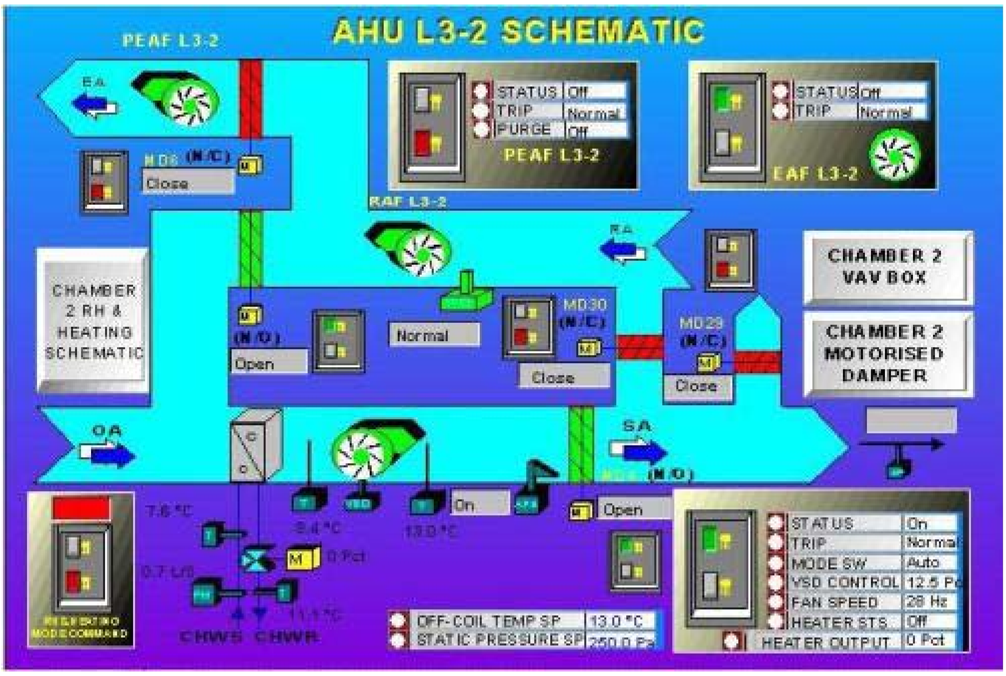


Figure 3-3 AHU for MV system

The parameters measured and recorded in the ambient AHU are as follows:

- Chilled-water supply and return temperature
- Chilled-water flow rate and damper opening
- Off-coil air temperature
- Air temperature after the fan

- Fan speed
- Supply air static pressure
- Set-points of off-coil temperature and static pressure

➢ **PV AHU**

Figure 3-4 shows a schematic of the PV AHU, which is used in the PV system. First, the outdoor fresh air is cooled and dehumidified by one part of a heat pipe and a cooling coil. It is then heated up by another part of the heat pipe and an electric heater. The 100% outdoor air is supplied to the occupants' breathing zone through PV ATDs.

In this study, the temperature of the air supplied from the PV system is designed to be kept constant during each session. The original aim was to keep the PV supply air temperature for all PV ATDs at three discrete values: 20℃, 23℃ and 26℃. However, this is extremely difficult to implement for practical reasons. Although the ducts are insulated there is a certain amount of heat gain via the ducting if personalized air temperature is lower than room temperature. Due to this effect, a decrease in personalized airflow will still result in an increase in the air temperature. Moreover, the air supplied by PV mixes with ambient air, causing the personalized air temperature to change within less than a centimeter from the diffuser outlet. In fact, temperatures measured by a probe inserted into the holes of the diffuser show that temperature starts to change significantly even before air is released from the diffuser. The effect is more pronounced at low air velocities. Therefore, the actual recorded PV supply air temperature, (measured just after the PV supply air fan as shown in Figure 3-4), is kept constant instead of the targeted value. Table 3-1 shows the recorded PV supply air temperature during the experiments. Under these recorded temperatures, the designed PV supply air temperature can be obtained when PV ATDs supply air at maximum flow rate. In this case, when subjects reduce the airflow rate, the PV supply air temperature will be slightly higher than the designed PV supply air temperature.

Table 3-1 Recorded PV supply air temperature

Ambient room temperature	Designed PV supply air temperature	Recorded PV supply air temperature
23℃	20℃	19.5
23℃	23℃	22.8
26℃	20℃	19
26℃	23℃	22
26℃	26℃	25.8

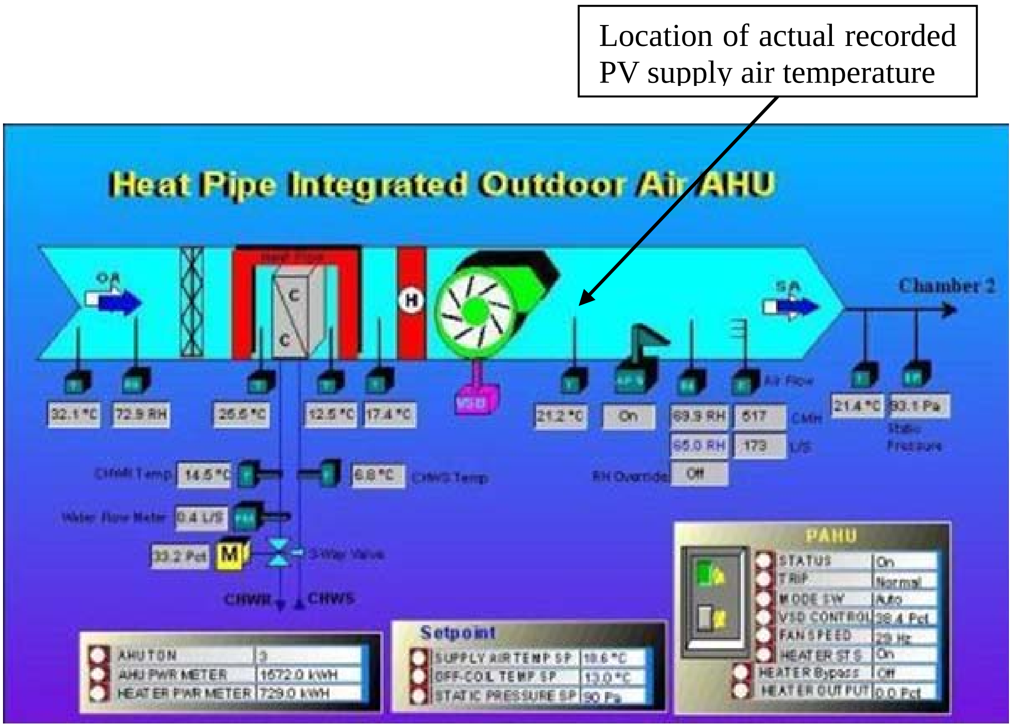


Figure 3-4 AHU for PV system

The parameters measured and recorded in the PV AHU are listed as follows:

- Outdoor air temperature and RH
- Air temperature after the heat pipe and before the cooling coil
- Air temperature after the cooling coil and before the heat pipe
- Air temperature after the heat pipe and before the electric heater
- Air temperature and RH after the supply air fan
- Static pressure of the supply air
- Fan power frequency
- Heater output and heater meter
- Chilled-water supply and return temperature
- Chilled-water flow rate and percentage of damper opening

➢ **PV ATD**

In this FEC, thirteen desk-mounted PV ATDs are installed at locations shown in Figure 3-1. Figure 3-5 shows the desk-mounted PV ATD installed in the FEC workstation. The PV airflow of each workstation can be controlled from 0L/s to approximate 15L/s.

Figure 3-5 Desk-mounted PV ATD in the FEC workstation

The PV ATD provides clean air from outdoor directly to the breathing zone. The PV ATDs used in this study are the same as those in Li Ruixin et al. (2010). Figure 3-6 shows the position and details of PV ATD at the workstation. The PV ATD is 500 mm above the desk and 400 mm away from the desk. The desk is 1m wide, and the PV ATD is placed in the middle of the desk. The PV ATDs are of round shape with a diameter of 0.1m and have perforated front plate and equalizer. The velocity measured near face is 0.3 m/s and 0.7 m/s with the PV airflow rate set at 5 L/s and 10 L/s respectively.

Users can change the direction of flow by turning the diffusers horizontally and vertically.

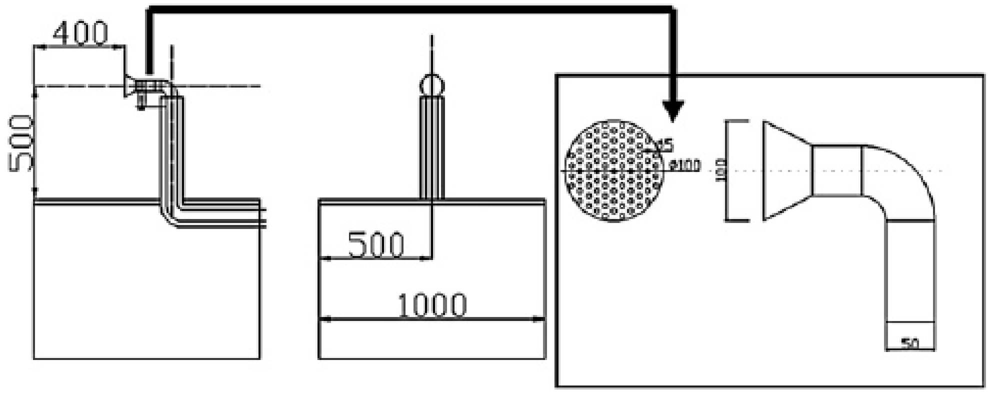


Figure 3-6 Position and details of PV ATD at the workstation
(Source: Li et al., 2010)

Originally, this FEC had 16 workstations with 16 PV ATDs as shown in Li Ruixin et al. (2010). Three workstations were removed before this subjective study started. However, personalized ventilation ducts leading to these workstations are still able to supply fresh air to the FEC.

The PV AHU is a VAV system, which has a minimum air flow rate of approximate 60 L/s. In this study, the subjects were provided with individual control of the airflow rate. There is a possibility that all the subjects might close all the PV ATDs at the same time.

This might cause damage to the PV AHU fan due to the build-up of pressure.

In order to prevent this, the three openings without PV-ATDs are kept open during all the experimental sessions. These three openings can supply air flow rate of about 20~30 L/s each. These openings are also used as ambient fresh air devices to control the ambient CO_2 level.

3.2.3 Individual Control System

- **Airflow rate control system**

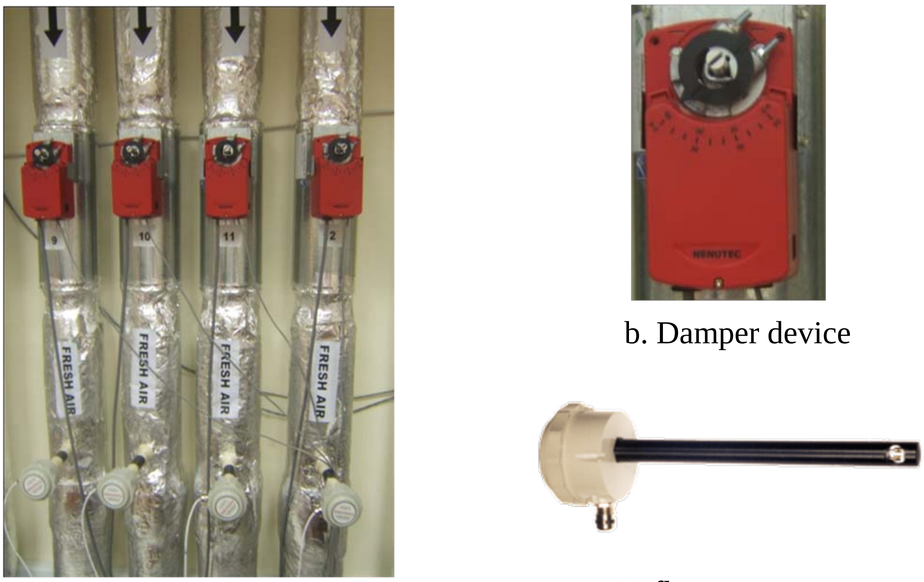
In this study, the airflow rate control system is installed in order to provide subjects with individual control of the airflow rate. Figure 3-7a shows the PV supply air pipe. On the top of the pipe, there is a damper device as shown in Figure 3-7b. At the bottom of the pipe, there is an airflow rate sensor as shown in Figure 3-7c.

Two control strategies are studied, as shown in Figure 3-8. One is damper opening set-point (DOSP) control, and the other is airflow rate set-point (AFSP) control.

The DOSP control means that the damper opening is adjusted directly according to the user's set-point. This set-point controlled by the user is the percentage of damper opening. For example, when the subject adjusts the set-point to 100%, the damper will be fully opened (90 degree). Respectively, when the set point is adjusted at 50%, the damper will be opened 45 degree.

In AFSP control, the air flow rate is maintained constant through closed loop feedback control. In this mode, the user's request to open or close the damper is interpreted as a request to increase or decrease the air flow rate. The AFSP control is achieved through the following procedure:

- When the user requests for a change in the airflow rate, the current airflow rate is measured by the sensor.
- The new air flow rate set-point is then computed by adding a proportional increase or decrease to the current air flow rate, according to the percentage of change demanded by the user.
- The initial estimate of the new damper opening is computed according to the relationship between airflow rate and damper opening that is obtained through calibration.
- The damper opening is adjusted subsequently in the feedback loop. In order to perform the calibration, the maximum airflow rates of all PV ATDs are measured.



a. PV supply air pipe

b. Damper device

c. Airflow rate sensor

Figure 3-7 Airflow rate control system

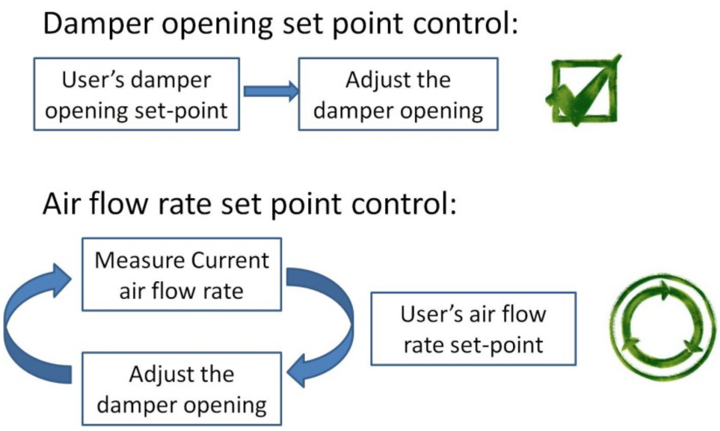


Figure 3-8 Airflow rate control strategies

Figure 3-9 shows the software interface of the airflow rate control system. On the top of the software interface, subjects can identity whether they are *working* or they have *left* the workstations. In case of "leaving", the PV ATDs will be closed automatically unless the CO_2 level in the room is high. In the section of *select damper opening*, subjects can choose the percentage of damper opening for DOSP control or the percentage of maximum airflow rate for AFSP control. If the subjects' preferred settings are not listed, they can input their preferred setting in the textbox on the left of the "Set" button. When the selection is completed, subject can click the "Set" button to set their preferred damper opening or airflow rate.

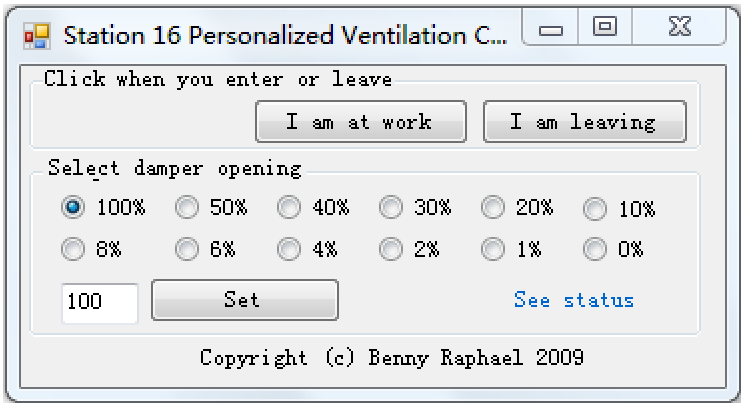


Figure 3-9 Airflow rate control software interface

3.3 Experimental conditions

While the subjective study is being conducted in the FEC, environmental parameters such as ambient air temperature and RH, and the PV supply air temperature are maintained constant during each of the two-hour session. The experimental conditions are shown in Table 3-2. Two control strategies, which are DOSP control and AFSP control, are used during these experiments. The ambient air temperatures in the FEC are set to 23°C or 26°C. The PV supply air temperatures are set to 20°C, 23°C or 26°C.

Table 3-2 Experimental conditions

Case Ref.	Session Number	Control strategies	Ambient Temperature	PV Temperature
D-A23-PV20	1-3	DOSP control	23℃	20℃
D-A23-PV23	4-6	DOSP control	23℃	23℃
D-A26-PV20	7-9	DOSP control	26℃	20℃
D-A26-PV23	10-12	DOSP control	26℃	23℃
D-A26-PV26	13-15	DOSP control	26℃	26℃
A-A23-PV20	16-18	AFSP control	23℃	20℃
A-A23-PV23	19-21	AFSP control	23℃	23℃
A-A26-PV20	22-24	AFSP control	26℃	20℃
A-A26-PV23	25-27	AFSP control	26℃	23℃
A-A26-PV26	28-30	AFSP control	26℃	26℃

However, in each session, there are two parameters that are kept below the maximum value, which are:

- Maximum RH: 60%
- Maximum CO_2 level: 990 ppm

3.4 Subject selection

In this study, thirty-six subjects are planned for each experimental condition. On average, twelve subjects should be involved in each session. In order to prevent unexpected situations (such as several absentees in a session), one more subject is used as backup. A total of thirteen subjects are therefore used in each session in this experimental design. The ideal situation is that every subject participates in all the 10 experimental conditions. However, since all the subjects are students and some of them were not available for the experiment, additional students had to be recruited. Eventually, a total number of 46 tropically acclimatized college-age students participated in the experiments instead of 39 subjects. Six of them attended only 1 or 2 sessions. The other 40 subjects, 14 males and 26 females, participated in more than five sessions in three separate groups, each of which included thirteen subjects. All the subjects are healthy, without chronic disease or allergy. The anthropometric data of the 40 subjects is shown in Table 3-3.

Table 3-3 Anthropometric data of the subjects

Gender	No. of subjects	Age (years)	Weight (kg)	Height (m)
Female	26	21.6±1.3	50.1±4.2	1.61±0.05
Male	14	23.0±2.1	68.9±7.5	1.74±0.05
Average/Overall	40	22.1±1.7	56.7±10.6	1.66±0.08

3.5 Questionnaire design

A computerized questionnaire survey is used in the experiments to obtain the responses from various groups of subjects in 10 series of experimental conditions. There are three parts in the questionnaire, which are explained as follows:

- Part one: "Thermal Sensation and Air Quality", which includes questions related to thermal comfort, thermal sensation, air movement perception, acceptability and preference, indoor air quality and SBS symptoms.

- Part two: “Environmental and other factors”, which includes questions related to noise level, lighting level and the dress of the subjects.
- Part three: “Control of Personalized Ventilation”, which includes questions related to the control of the PV ATD, the frequency of adjusting the PV ATD, and the performance of the PV ATD. The questionnaire is shown in Appendix A.

3.5.1 Part one: thermal sensation and air quality

- **Thermal comfort and thermal sensation**

The acceptability for thermal condition is determined by the divided continuous scale as shown in Figure 3-10. Subjects can indicate their perception by grading the definite degree of acceptability or unacceptability.

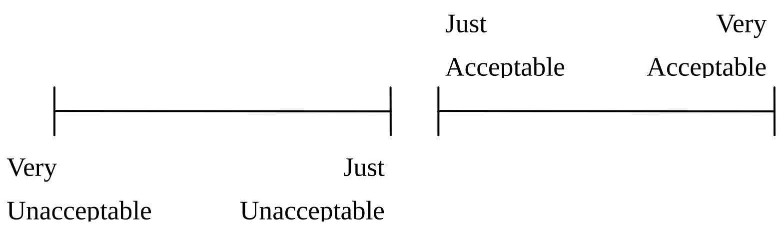


Figure 3-10 Divided continuous visual-analogue scales

The value of very unacceptable is 0; the value of very acceptable is 100; and the values of just acceptable and just unacceptable are both 50. It is not allowed to make between “Just unacceptable” and “Just acceptable”.

ASHRAE 7-point scale (ASHRAE Standard 55, 2004), as illustrated in Table 3-4, is used to evaluate on the subjects' thermal sensation for whole body and thirteen body segments.

Table 3-4 ASHARE's 7-point scale

Cold	Cool	Slightly cool	Neutral	Slightly warm	Warm	Hot
-3	-2	-1	0	+1	+2	+3

Figure 3-11 shows a diagram of a human body with clearly demarcated body segments is used to indicate the thermal sensation. Thermal sensation of the whole-body is indicated as well.

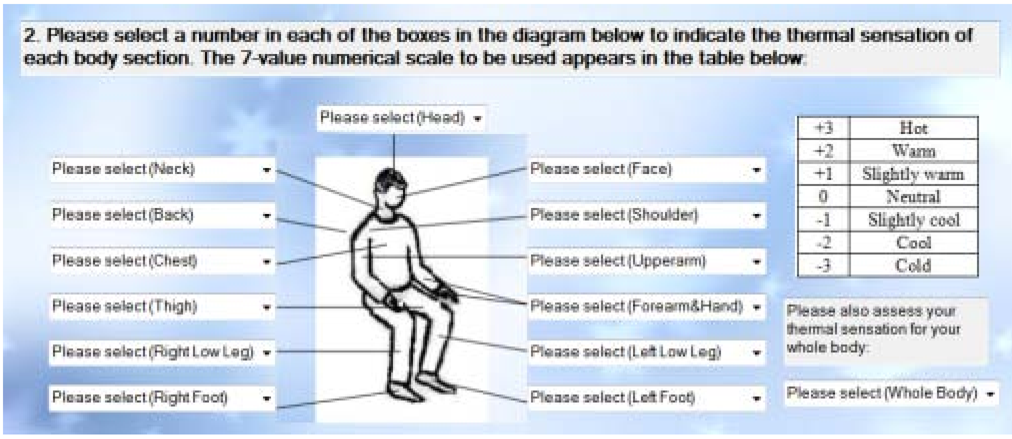


Figure 3-11 Thermal sensation for whole body and thirteen body segments

➢ **Air movement perception, acceptability, and preference**

Subjects are inquired about whether they feel any air movement on their head, face or shoulder separately. If the answer is "Yes", the perception of air movement at that body segment is then accessed, according to the 7-value numerical scale shown in Table 3-5.

Table 3-5 7-point numerical scale for the perception of air movement

Much to breezy	Too breezy	Slightly breezy	Just right	Slightly still	Too still	Much too still
+3	+2	+1	0	-1	-2	-3

Subjects are asked to assess the acceptability of the air movement on top of head, facial part and neck. The divided continuous scale as shown in Figure 3-10 is used. Subjects are asked to indicate the change in air movement preferred on top of head, facial part and neck. Three options, which are *less air movement*, *no change*, and *more air movement*, are provided.

➢ **Indoor air quality**

A few factors related to indoor air quality are evaluated, including inhaled air quality, inhaled air temperature, inhaled air humidity, odour of inhaled air, and freshness of inhaled air. For the factor of inhaled air quality, the divided continuous scale is used. The linear visual analogue scales without intervals are used for the other factors including inhaled air temperature (cold 0 to hot 100), inhaled air humidity (humid 0 to dry 100), odor of inhaled air (no odor 0 to overwhelming odor 100), and freshness of inhaled air (air stuffy 0 to air fresh 100).

- **Sick building syndrome**

The factors related to SBS include the nose dry, lips dry, eyes dry, headache, ability to think, eyes aching, dizzy, tired, sleepy and feeling. The feelings of the environment are all assessed from 0 to 100.

3.5.2 Part two: environmental and other factors

In this part, noise level and lighting level are evaluated and the types of attire are listed for subjects to select. Subjects are not allowed to change their attire during the experiments. In the data analysis, the clothing information is converted into clothing thermal insulation (in Clo) to evaluate the differences among subjects' clothing. Table 3-6 shows some garment insulation values which are used to calculate the clothing thermal insulation.

Table 3-6 Garment Insulation Values
(Source: ASHRAE Handbook, Fundamentals SI, 2005).

Garment Description	$I_{clu,i}$, clo	Garment Description	$I_{clu,i}$, clo
Underwear		Suit jackets and vents (lined)	
Men’s briefs	0.04	Single-breasted (thin)	0.36
Panties	0.03	Single-breasted (thick)	0.44
Bra	0.01	Double-breasted (thin)	0.42
T-shirt	0.08	Double-breasted (thick)	0.48
Full slip	0.16	Sleeveless vest (thin)	0.10
Half slip	0.14	Sleeveless vest (thick)	0.17
Long underwear top	0.20	Sweaters	
Long underwear bottoms	0.15	Sleeveless vest (thin)	0.13
Footwear		Sleeveless vest (thick)	0.22
Ankle-length athletic socks	0.02	Long-sleeved (thin)	0.25
Calf-length socks	0.03	Long-sleeved (thick)	0.36
Knee socks (thick)	0.06	Dresses and skirts [c]	
Panty hose	0.02	Skirt (thin)	0.14
Sandals/thongs	0.02	Skirt (thick)	0.23
Slippers(quilted, pile-lined)	0.03	Long-sleeved shirtdress (thin)	0.33
Boots	0.10	Long-sleeved shirtdress (thick)	0.47
Shirts and Blouses		Short-sleeved shirtdress (thin)	0.29
Sleeveless, scoop-neck blouse	0.12	Sleeveless, scoop neck (thin)	0.23
Short-sleeved, dress shirt	0.19	Sleeveless, scoop neck (thick)	0.27
Long-sleeved, dress shirt	0.25	Sleepwear and Robes	
Long-sleeved, flannel shirt	0.34	Sleeveless, short gown (thin)	0.18
Short-sleeved, knit sport shirt	0.17	Sleeveless, long gown (thick)	0.20
Long-sleeved, sweat shirt	0.34	Short-sleeved hospital gown	0.31
Trousers and Coveralls		Long-sleeved, long gown (thick)	0.46
Short shorts	0.06	Long-sleeved pajamas (thick)	0.57
Walking shorts	0.08	Short-sleeved pajamas (thin)	0.42
Straight trousers (thin)	0.15	Long-sleeved, long wrap robe (thick)	0.69
Straight trousers (thick)	0.24	Long-sleeved, short wrap robe (thick)	0.48
Sweatpants	0.28	Short-sleeved, short robe (thin)	0.34
Overalls	0.30		
Coveralls	0.49		

3.5.3 Part three: control of personalized ventilation

Three questions are listed to assess the general opinions of the PV-ICA system:

1. How easy is it to control the personalized ventilation air terminal device (PV ATD)?

Very Difficult	├──────────┤	Very Easy (100)

2. How frequently did you have to adjust the PV ATD?

Very Often (0)	├──────────┤	Seldom (100)

3. How do you rate the performance of the PV ATD?

Bad (0)	├──────────┤	Good (100)

3.6 Experimental Procedure

It takes two hours for subjects to complete each session. During each session, subjects are allowed to individually regulate the opening of damper in DOSP control or the airflow rate in AFSP control. All changes are automatically recorded in airflow rate control system.

During the first fifteen minutes, subjects get familiar with the room environment by adjusting the flow rate to ensure that the air speed is comfortable to them. When this adjustment is completed, subjects take 90-minute multiple choice tests that evaluate their reading comprehension. Different questions are used in each experiment.

30 minutes after the experiment, subjects are required to fill in a computerized questionnaire feedback.

45 minutes after the commencement of the experiment, half of the participants have to leave the room for a ten-minute period. The other half subjects take a break after 65 minutes. Subjects are required to turn off the PV ATDs when they leave the workstations and to turn on the PV ATDs when they return. When the subjects turn on the PV ATDs, the settings of the PV ATDs will be the same as they leave.

90 five minutes after the commencement of the experiment, CO_2 pumps are used to increase the CO_2 concentration in the room. CO_2 concentration is increased to a maximum of 990 ppm and this increased CO_2 concentration is maintained for five minutes. As indicated in the appendix C of ASHRAE

Standard 62.1-2010 (ASHRAE Standard 62.1, 2010), a steady-state CO_2 concentration in a space no greater than about 700 ppm above outdoor air levels will indicate that a substantial majority of visitors entering a space will be satisfied with respect to human bioeffluents (body odor). Moreover, CO_2 concentrations in acceptable outdoor air typically range from 300 to 500 ppm. A maximum of 990 ppm CO_2 concentration is therefore used in this study. The original objective of pumping CO_2 was to study how subjects react to higher CO_2 concentration levels. However, higher CO_2 levels were not found to have any significant effect on their behavior.

The last fifteen minutes is spent on completing the feedback form.

The various events during the test are summarized in Table 3-7. A schematic diagram of the two-hour session is shown in Figure 3-12.

Table 3-7 Summarization of various events during the test

Time (minutes)	Event
0-15	Users get familiar with the controls
15	Start of test
30-45	Feedback-1
45-55	Half the participants take a break
65-75	Second half of participants take a break
90-95	Increased concentration of CO_2
105	End of test
105-120	Feedback-2

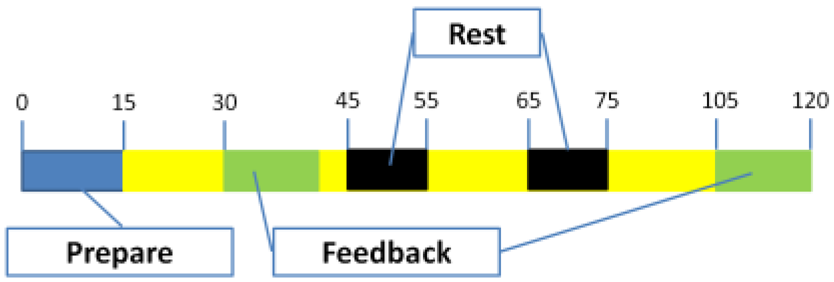


Figure 3-12 Schematic diagram of the two hours

3.7 Method of data collection and analysis

3.7.1 Subjects' behavior

As mentioned in Section 3.6, the subjects are allowed to control the airflow rate individually throughout the experiments. The information of the changes will be recorded in the airflow rate control system.

- **DOSP control**

In DOSP control, the recorded information includes the time, the original damper opening setting, subject's new damper opening setting and the voltage of the airflow rate sensor.

The PV AHU is a variable air volume (VAV) system. The supply air static pressure is kept constant at 180 Pa for all experimental conditions. Before the experiment starts, an objective measurement is adopted to evaluate the performance of the damper devices. Figure 3-13 shows the relationship between airflow rate and percentage of damper opening of work station 11. The relationship is nonlinear. At the beginning, a small opening causes a large increase in the airflow. When the damper opening reaches about 20%, the change in airflow rate is not appreciable because the resistance to the flow is negligible. As the damper opening increases to about 30%, the curve is nearly flat and the slope is close to zero. This curve is plotted by opening only one damper and keeping the dampers of all other PV-ATDs unchanged. Similar results of other work stations are obtained and shown in Figure 3-14. The recorded damper opening data combined with the damper characteristics are used to calculate the subjects' preferred airflow rate.

The recorded control information and the airflow rate calculation results are used to evaluate subjects' behavior such as the frequency of controlling the airflow rate, and the subjects' preferred airflow rate. The results are discussed in Section 4.1.

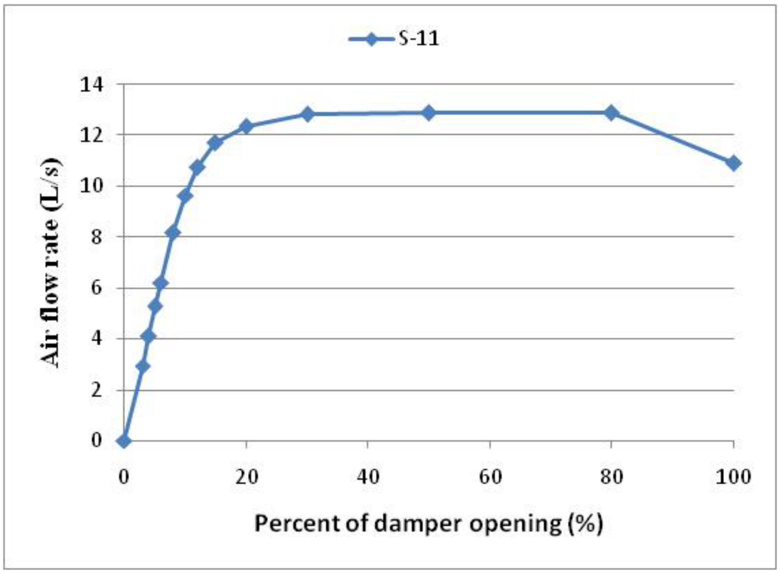


Figure 3-13 Damper characteristics of workstation 11

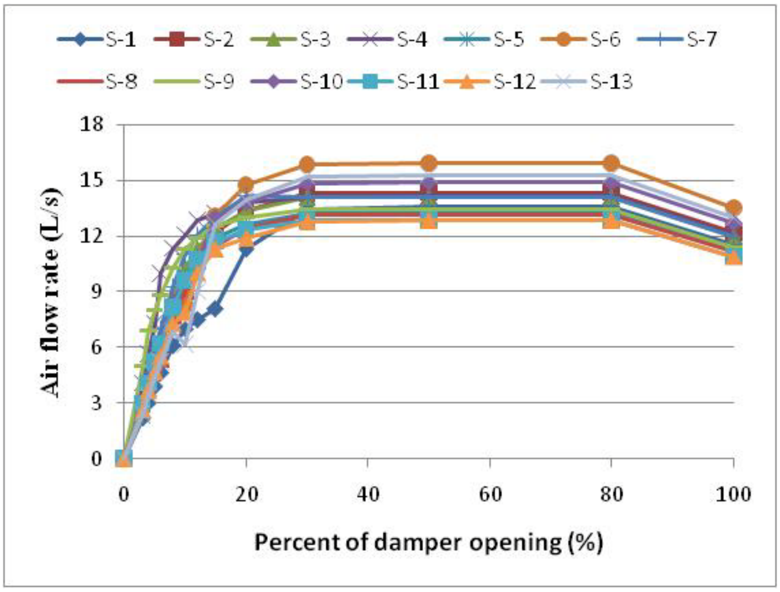


Figure 3-14 Damper characteristic of all workstations

- **AFSP control**

In AFSP control strategy, the recorded information includes the time, the original percentage of airflow rate setting, subject's new percentage of airflow rate setting and the voltage of the airflow rate sensor when subjects change the air flow rate.

Before the experiment starts, an objective measurement is adopted to evaluate the performance of the airflow rate sensors. Figure 3-15 shows the linear relationship between airflow rate through the diffuser and voltage of airflow rate sensor of workstation 11. The equation and R-squared value of the trend line are shown in Figure 3-15 as well. As the minimum voltage of the airflow rate was 0.11 V, it is unable to measure the airflow rate when the airflow rate is lower than 2.5 L/s. This is a limitation of the airflow rate sensors. Similar results shows that most of the airflow rate sensor characteristics of other workstations are linear as shown in Figure 3-16. The linear relationship between air flow rate and the voltage of air flow rate sensors is good for AFSP control.

Although the sensor characteristics of all work stations are obtained by objective measurement, the information obtained is insufficient to calculate the airflow rate. The system records the airflow rate only at the time subjects change the airflow rate. However, as introduced in Section 3.2.3, AFSP control is a loop control. When subject changes the setting, the required airflow rate is achieved gradually by several adjustments. Moreover, the airflow rate sensor has a time delay which is about 1.5 minutes. The measured value of the sensor becomes relative steady 1.5 minutes after the damper opening is changed. This time delay has great impact on the control performance. Each adjustment takes about 2 minutes to process. When subjects change the settings, it takes several minutes until the target airflow rate is reached. During this time, the subjects may change the air flow rate again, causing the air flow rate to be unsteady throughout the experiment. The recorded data could not therefore be used to calculate the subjects' preferred airflow rates. Therefore, data related to user preference in AFSP control is not further discussed in this thesis.

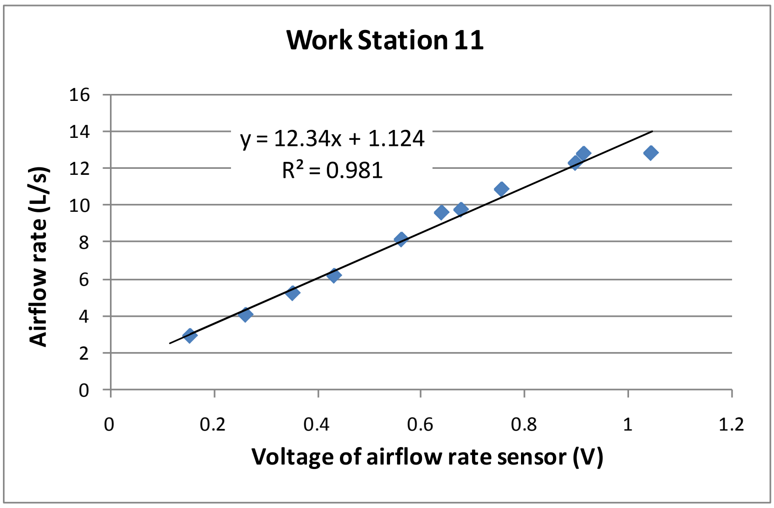


Figure 3-15 Airflow rate sensor characteristic of work station 11

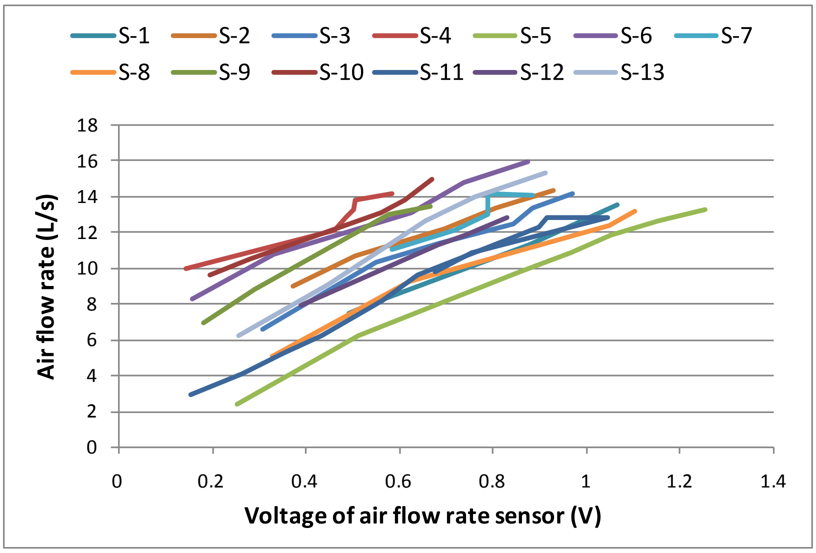


Figure 3-16 Airflow rate sensor characteristic of all work stations

3.7.2 Subjective assessment

Human responses to the PV-ICA system are collected at each session through the questionnaires. The subjects' feedbacks are used to evaluate the thermal performance of the PV-ICA system, including:

- Whole body thermal comfort and its acceptability;
- Whole body and body segments thermal sensation;
- Air movement perception, acceptability and preference;
- Indoor air quality and SBS symptoms;
- Subjects' clothing; and
- The general opinion of the PV-ICA system.

The results of human response to the PV-ICA system are presented in Section 4.2.

3.7.3 Energy consumption of the PV-ICA system

In this section, the method of calculating energy consumption of the PV-ICA system based on the recorded data in BAS is presented. The components of energy consumption of the HVAC system include the energy consumption of fans, and the equivalent energy consumption of the chiller plant.

- **Energy consumption of fans**

There are three types of fans used in this study, which are: two in ambient AHU, a supply-air fan and a return-air fan, and one in PV AHU, a supply-air fan.

In PV AHU, the power frequency and the airflow rate of the fan are recorded in the building automation system (BAS). In ambient AHU, the power frequency of the fans is recorded in the BAS as well. The power meters of the fans are installed to measure the power of the fans. However, the measured data are not automatically recorded in the BAS.

In this case, power curves of the fans obtained by experimental measurement are used to calculate the energy consumption of the fans. During the experimental measurement, the settings of the AHUs are the same as those during the subjective study. When the system is stable, the power and the power frequency of the fans are measured. Figures 3-17 and 3-18 show the power curve and the characteristic of the PV AHU supply air fan. Figures 3-19 and 3-20 show the power curve of ambient AHU return air fan and supply air fan. The energy consumption of the fans is calculated using the recorded fan power frequency and the power curve.

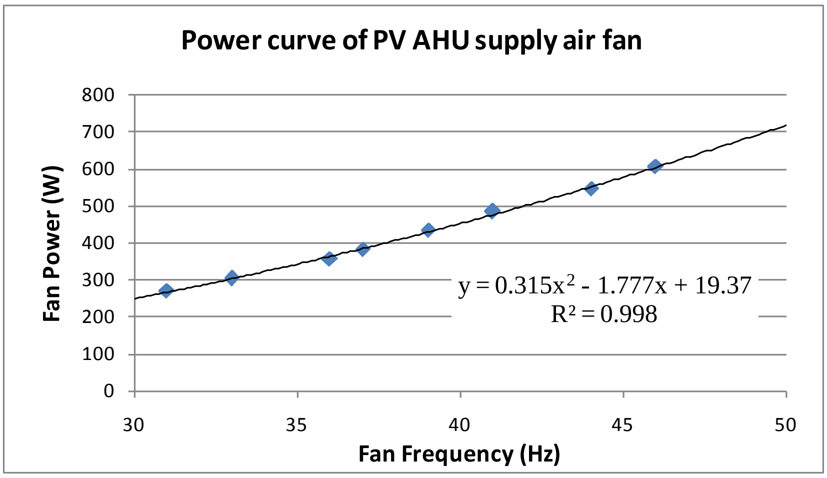


Figure 3-17 Power curve of PV AHU fan

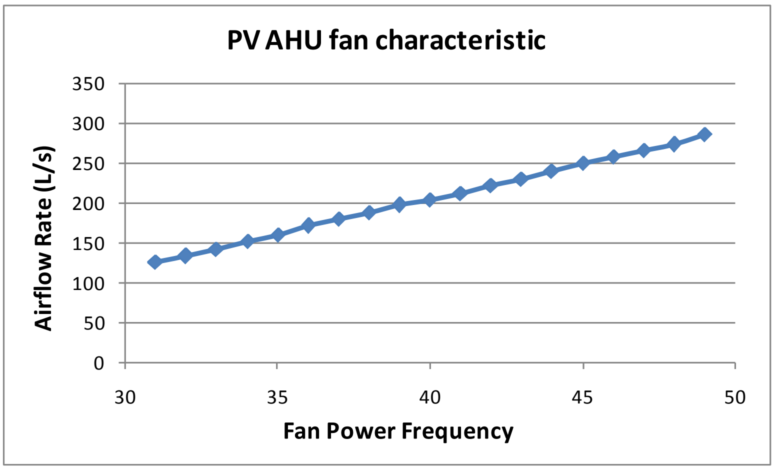


Figure 3-18 PV AHU fan characteristic

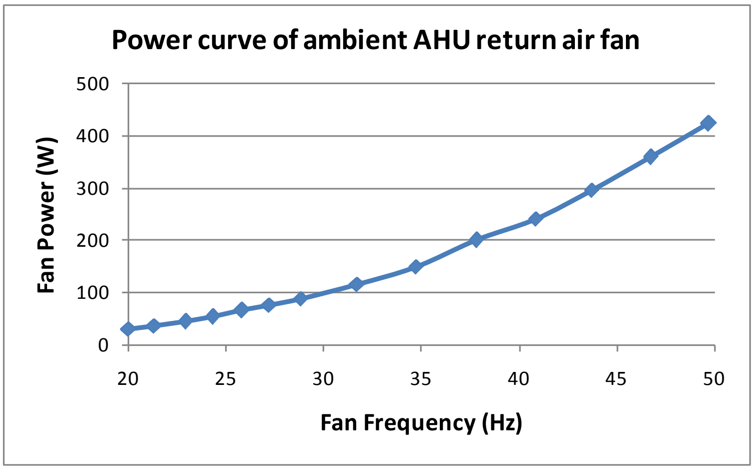


Figure 3-19 Power curve of ambient AHU return air fan

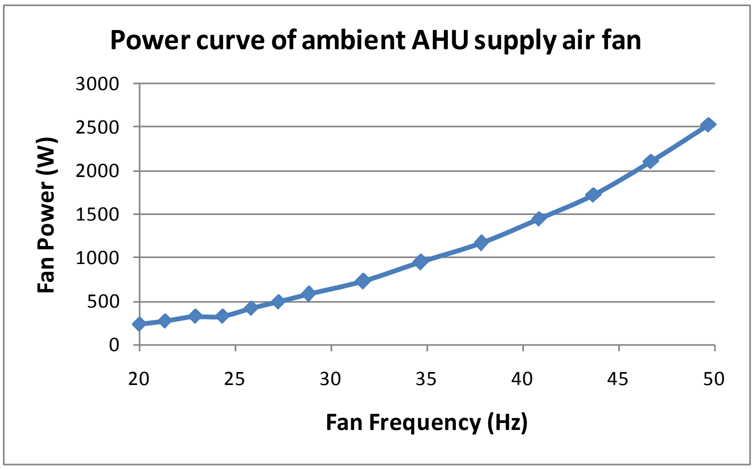


Figure 3-20 Power curve of ambient AHU supply air fan

- **Equivalent energy consumption of the chiller plant**

The following parameters recorded in BAS are used to calculate the cooling load of cooling coil of the PV AHU:

T_{sw-pv} (℃): Chilled-water supply temperature for PV AHU

T_{rw-pv} (℃): Chilled-water return temperature for PV AHU

G_{pv} (L/s): Chilled-water flow rate for PV AHU

The cooling load of cooling coil of PV AHU, Q_{pv} (W), is calculated by Equation 3-1.

$$Q_{pv} = C_p * \rho * G_{pv} * (T_{rw-pv} - T_{sw-pv}) \qquad \text{Equation 3-1}$$

Where:

C_p (J/kg/K) is the specific heat of water at constant pressure

ρ (kg/m^3) is the density of water

The following parameters as recorded in BAS are used to calculate the cooling load of cooling coil of the ambient AHU:

T_{sw-mv} (℃): Chilled-water supply temperature for ambient AHU

T_{rw-mv} (℃): Chilled-water return temperature for ambient AHU

G_{mv} (L/s): Chilled-water flow rate for ambient AHU

The cooling load of cooling coil of ambient AHU, Q_{mv} (W), is calculated by Equation 3-2.

$$Q_{mv} = C_p * \rho * G_{mv} * (T_{rw-mv} - T_{sw-mv}) \text{ ……………………………Equation 3-2}$$

Where:

C_p (J/kg/K) is the specific heat at constant pressure of water

ρ (kg/m^3) is the density of water

The equivalent energy consumption of chiller plant, P (W), is calculated by Equation 3-3.

$$P = \frac{Q}{TCOP} \text{ ………………………………………..…………..Equation 3-3}$$

Where:

Q (W) is the total cooling load

TCOP is total coefficient of performance of the chiller plant, which equals to the cooling water load of the chiller divided by the total energy consumption of the chiller, the pump(s), and the MV AHU fans.

TCOP is affected by many factors, such as chilled water supply temperature, chilled water return temperature, chilled water flow rate, cooling water supply temperature, cooling water flow rate etc. However, in this research, the performance of the chiller plant is not studied. The TCOP is set to 4 in this research.

3.8 Energy model using EnergyPlus

The outdoor air temperature and RH varied from session to session. Outdoor air temperature varied from 26.9℃ to 31.2℃ while the RH varied from 71.7% to 99.7%. The number of subjects in each group varied from 10 to 13. Due to these variations, external and internal heat gains were different for the same controlled temperature settings. To obtain a fair comparison, energy simulation is adopted in this study.

There are three stages in the simulation. In the first stage, the energy model is made based on the experimental conditions. The simulation results are compared with the measurement results to examine the accuracy of the model. In the second stage, some parameters are normalized based on the experimental conditions in order to create a fair comparison. The simulation results are used to evaluate the energy performance of the PV-ICA system. In the third stage, the validated energy model is applied to evaluate the energy saving potential of the PV-ICA system. The impact of ventilation rate on energy consumption of the PV-ICA system is analyzed.

3.8.1 Building locations and weather data

The FEC is located in Singapore. The weather is characterized as hot and humid. In the first stage, the outdoor air dry bulb temperature and RH are measured by BAS and are used as input data, as shown in Figure 3-21and Figure 3-22. In the second and third stages, the ASHRAE IWEC Weather File for Singapore is used to simulate the annual energy consumption.

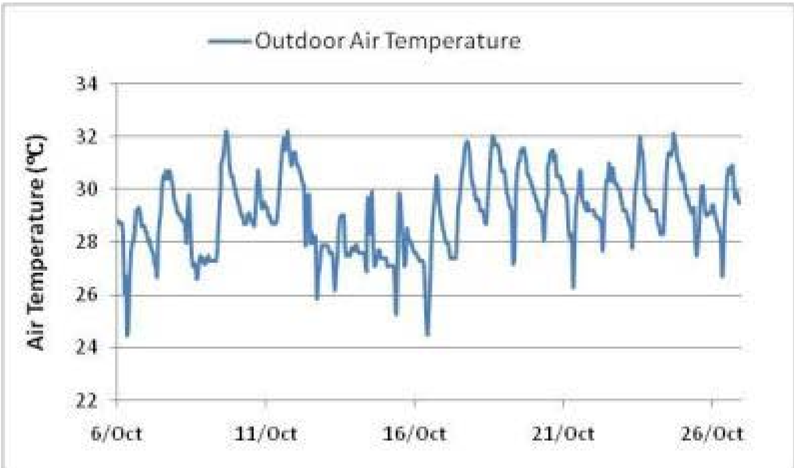


Figure 3-21 Outdoor air temperature during the experiment period

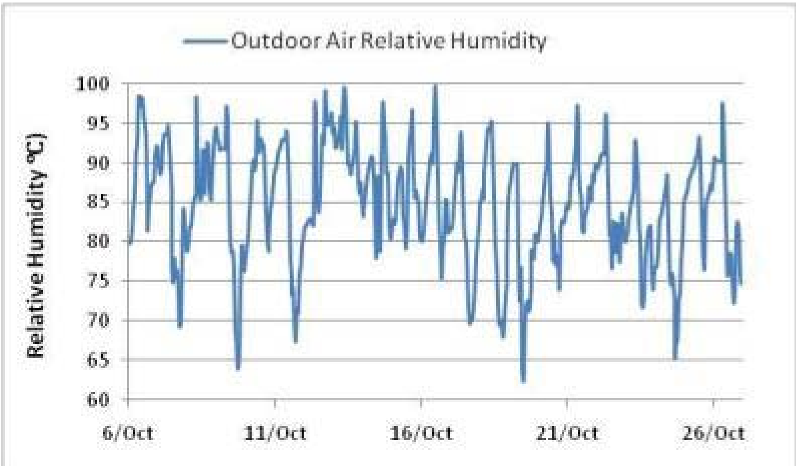
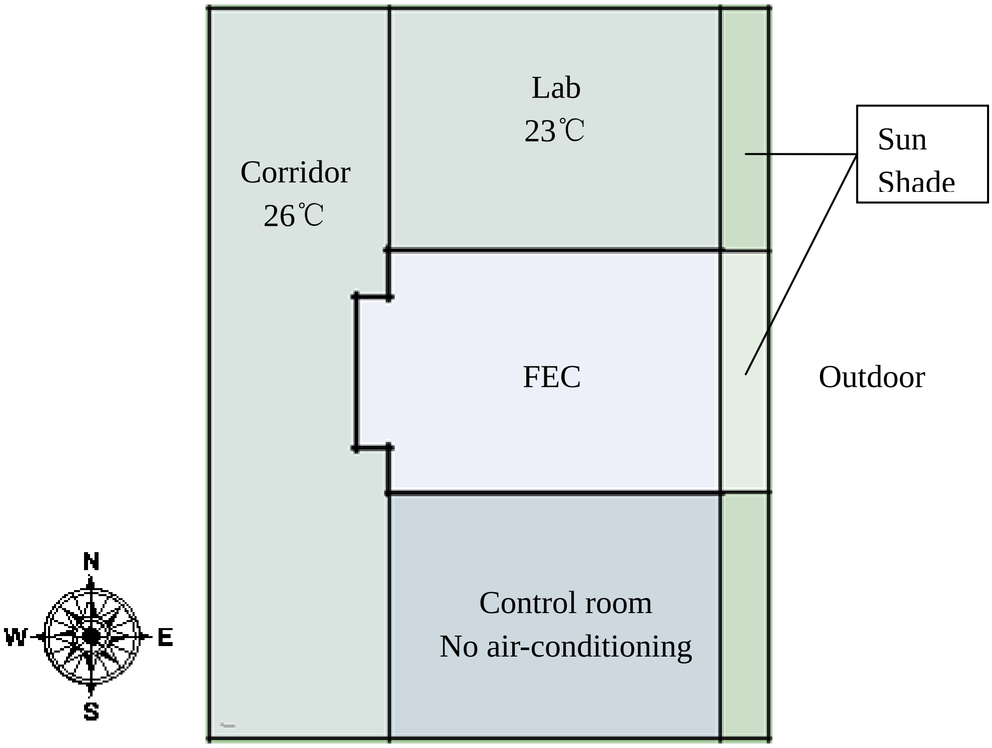
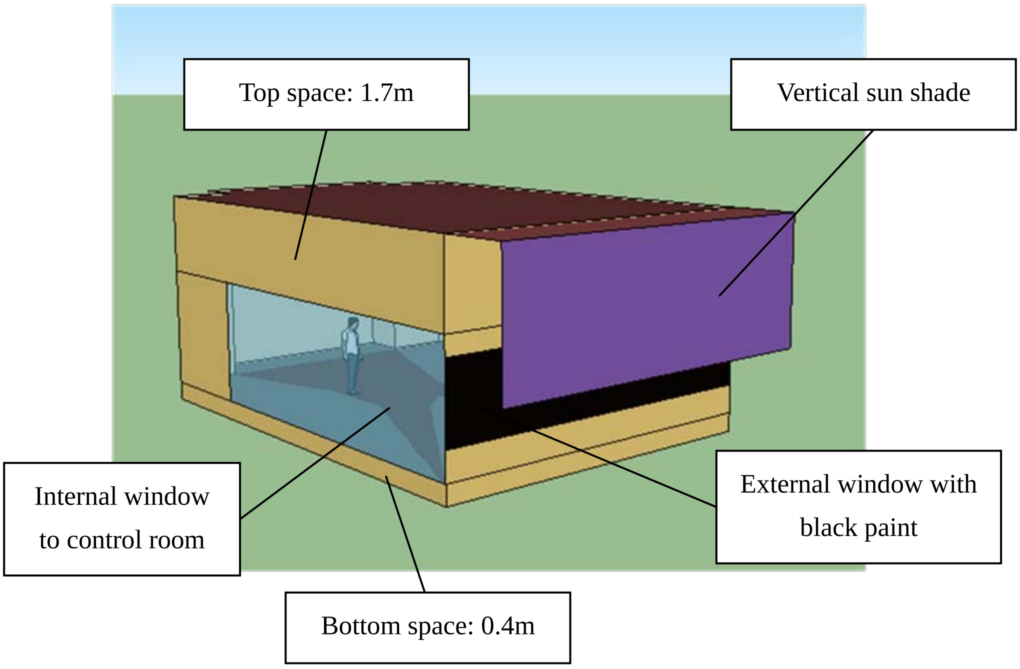


Figure 3-22 Outdoor air RH during the experiment period

3.8.2 Description of the FEC



(a). Plan view



(b). Three-dimensional view

Figure 3-23 Description of the FEC

Figure 3-23 shows the plan view and three-dimensional view of the FEC. The FEC has a floor surface area of 11.1 m × 7.8m as shown in Figure 3-1. The floor-to-ceiling height is 4.7m and the suspended floor to suspended ceiling height is 2.6 m.

The external walls are constructed with 5mm plaster inside (λ =0.814W/K/m, ρ =1600kg/m^3), 150 mm medium-weight concrete (λ =0.667W/K/m, ρ =1500kg/m^3) and 5mm plaster outside; the overall U-value of the external wall is 2.8W/K/m^2 when external and internal heat convections are 20 and 8.7 W/K/m^2.

The internal walls are constructed with 20mm gypsum board (λ =0.198W/K/m, ρ =760kg/m^3) at both sides and 60mm air between the boards (thermal resistance = 0.15m^2.K/W); the overall U-value of the external wall is 1.7 W/K/m^2 when internal heat convection is 8.7 W/K/m^2.

The FEC external window is composed of a 6 mm clear glass (λ =0.9W/K/m) and a black paint; the overall U-value of the external wall is 5.8W/k/m^2 when external and internal heat convections are 20 and 8.7 W/K/m^2. As the window was covered with solar block film, the transmittance and reflectance are both

0.01. The window has a total area of $12m^2$ (1.6m height $\times$7.5m wide, 33% of the envelop area). The window faces east. There is a vertical sun shade.

The other external windows of the nearby rooms and the internal windows between FEC and control room are all composed of 6 mm clear glass (λ =0.9W/K/m). The transmittance is 0.775 and the reflectance is 0.071.

The ceiling and the floor are constructed with 200 mm heavy-weight concrete (λ =1.95W/K/m, ρ =2240kg/m^3).

The suspended ceiling and suspended floor are constructed with 25mm hard wood (λ =0.167W/K/m, ρ =680kg/m^3).

3.8.3 Description of the HVAC system

As described in Section 3.2.2, the FEC is served by two dedicated systems: an ambient air handling unit (AHU) for 100% recirculated air that is supplied through ceiling outlets and a PV AHU for 100% outdoor air that is supplied through the PV ATDs. There is no mechanical exhaust; thus air is exhausted through leaks and holes on the external walls and the suspended ceiling. There is no heater in the ambient AHU. In the PV AHU, an electric heater is used to heat up the personalized air to the target supply-air temperature.

3.8.4 Internal temperature, ventilation and infiltration rate

In this study, two design ambient room temperatures are applied, which are: 23℃ and 26℃. In the first stage, the room temperature and RH measured by BAS as shown in Table 3-8 are used. In the second and third stages, the room temperatures are normalized to the design room temperature, which is either 23℃ or 26℃. The room RH is set to 50% for all experimental conditions.

Table 3-8 Room temperature and RH of all experimental conditions

Case Ref.		FEC RH	FEC Temp.	Case Ref.		FEC RH	FEC Temp.
D-A23-PV20	A	52	22.7	A-A23-PV20	A	56.1	22.9
D-A23-PV20	B	49.2	23.1	A-A23-PV20	B	59.5	22.1
D-A23-PV20	C	49.2	23	A-A23-PV20	C	56.8	22.6
D-A23-PV23	A	50.6	23.3	A-A23-PV23	A	58.2	22.7
D-A23-PV23	B	51.4	23	A-A23-PV23	B	56.9	23.1
D-A23-PV23	C	55.8	23.3	A-A23-PV23	C	58.5	22.9
D-A26-PV20	A	44.3	26	A-A26-PV20	A	48.1	25.7
D-A26-PV20	B	44.3	25.8	A-A26-PV20	B	45.6	26.7
D-A26-PV20	C	44.8	25.8	A-A26-PV20	C	49.1	25.6
D-A26-PV23	A	45.8	26.1	A-A26-PV23	A	49.1	26
D-A26-PV23	B	44.4	26	A-A26-PV23	B	47.5	26.1
D-A26-PV23	C	46.3	25.8	A-A26-PV23	C	47.4	26.5
D-A26-PV26	A	47.7	26.7	A-A26-PV26	A	49.2	26.5
D-A26-PV26	B	44.1	26.8	A-A26-PV26	B	49.6	26.3
D-A26-PV26	C	51.9	24.5	A-A26-PV26	C	49.4	26.5

The ambient AHU uses 100% recirculated air while the PV AHU uses 100% outdoor air. So the ventilation rate equals to the airflow rate of PV AHU, which is measured by the BAS. In the first stage, the ventilation rates are equal to recorded PV AHU supply airflow rates, which include the personalized air for three openings and thirteen PV ATDs, even when some workstations are non-occupied. In the second stage, the three openings are closed and the thirteen workstations are all occupied. In each experimental condition, the air flow rate of each ATD is equal to the average value of subjects' preferred airflow rate between 45th and 120th minute. In the third stage, the impacts of ventilation rate on energy consumption of the PV-ICA system are analyzed. The details of the ventilation rates are introduced in Section 4.3.3.

As there is no mechanical exhaust, the pressure of FEC may be slightly higher than that of outdoor air. So in this study, the infiltration rate is ignored and set to 0 for both simulation stages.

3.8.5 Internal heat gains and occupancy

There are twenty-one 54 W and two 36 W fluorescent lighting fixtures, which are used in the FEC to mock up a typical open plan office. The total power of lightings is 1206 W. The lightings are all on during the two-hour experiment.

There are 14 desktop computers and 1 laptop in the FEC. Among them, 13 desktop computers are placed on the 13 workstations and 1 desktop computer and the laptop are used as servers to control and collect data of the airflow rate control system. Each desktop computer is 145 W and the laptop is 60 W. The total power of equipments is 2090 W. The computers and laptop are all on during the experiment.

In the first stage, the number of subjects in each session is shown in Table 3-9. The number ranges from 10 to 13. The subjects are asked to do a reading test; the activity level is 108 W/person (ASHRAE, 2005). The schedule of subjects is introduced in Section 3.6. In the second and third stages, the numbers of occupants are normalized to 13 for all experimental conditions.

Table 3-9 Number of subjects in each session

Case Ref	Group	Number of subjects	Case Ref	Group	Number of subjects
D-A23-PV20	A	13	A-A23-PV20	A	13
D-A23-PV20	B	11	A-A23-PV20	B	11
D-A23-PV20	C	13	A-A23-PV20	C	13
D-A23-PV23	A	13	A-A23-PV23	A	10
D-A23-PV23	B	13	A-A23-PV23	B	13
D-A23-PV23	C	13	A-A23-PV23	C	13
D-A26-PV20	A	13	A-A26-PV20	A	13
D-A26-PV20	B	12	A-A26-PV20	B	12
D-A26-PV20	C	10	A-A26-PV20	C	13
D-A26-PV23	A	13	A-A26-PV23	A	12
D-A26-PV23	B	12	A-A26-PV23	B	12
D-A26-PV23	C	12	A-A26-PV23	C	12
D-A26-PV26	A	12	A-A26-PV26	A	13
D-A26-PV26	B	11	A-A26-PV26	B	11
D-A26-PV26	C	13	A-A26-PV26	C	13

3.8.6 Simulation software

In this research, EnergyPlus V5.0 is used to simulate the cooling load of the FEC.

3.8.7 Simulation cases

In the first stage, the EnergyPlus model of the FEC is made based on the descriptions above. The model is used to simulate the cooling load of all the 30 experimental sessions. As discussed in Section 4.1, the subjects have made frequent changes during the first 15 minutes and the subjects are asked to leave the room in two groups between the 45th minute and the 75th minute. In this case, two periods are simulated for each experimental session, which are: first, from 15th to 45th minute; and second, from 75th to 105th minute. The simulation results are compared with the measurement data to examine the accuracy of the energy model.

In the second stage, some parameters, which are normalized in order to create a fair comparison, are summarized as follows:

- ASHRAE IWEC Weather File for Singapore is used for simulating the annual energy consumption
- The room temperatures are normalized to the design room temperature, which is either 23℃ or 26℃.
- The three ambient openings, without ATDs, are all closed. The ventilation rates are normalized to supply personalized air to 13 PV ATDs only. The air flow rate for each workstation of each session is set to the average value of the subjects' preferred airflow rate between 45th and 120th minute.
- The numbers of occupants are normalized to 13 for all experimental conditions.
- The simulation period is normalized to the whole year. During the subjective experiment, there are two sessions per working day. The first

one is from 10 am to 12 pm while the other one is from 4 pm to 6 pm. Therefore, in the second stage of simulation, the occupants' presence in the room is set to four hours per working day, which include 10 am to 12 pm and 4 pm to 6 pm.

As the normalization requires the information of subjects' preferred air flow rates, only the DOSP control cases are simulated in this study. After the normalization, the energy model is used for simulating the five DOSP control experimental conditions.

In the third stage, the impacts of ventilation rate on the energy consumption of the PV-ICA system are analyzed.

Chapter 4: Results and discussion

In this chapter, analysis of subjects' behavior in operating the controls, subjective assessment and energy performance of the PV-ICA system are presented.

4.1 Subjects' behavior in operating the controls

In this study, the subjects were provided with individual control of personalized airflow rate that results in variations in air movement. Subjects' frequency of changes and their preferred airflow rates are discussed in this section.

4.1.1 Frequency of changes to settings

As introduced in Section 3.2, two control strategies are applied in this research: DOSP control and AFSP control. Subjects' behavior in operating the controls under these two control strategies is discussed.

- **DOSP control**

Figure 4-1 shows the average number of times the subjects change the airflow rate during the two-hour experimental session in DOSP control cases. The average values are 7.5 at room ambient temperature of 23℃ and 3.9 at room ambient temperature of 26℃. It is clearly shown that the average number of times that subjects change the airflow rate at room ambient temperature of 26℃ is much lower than that at room ambient temperature of 23℃.

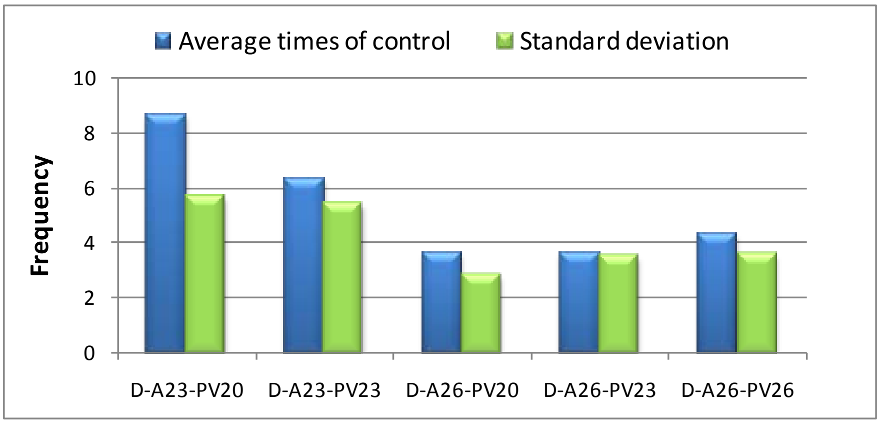


Figure 4-1 Frequency of operation of changes in DOSP control cases

Figure 4-2 shows the time distribution of the operation of changes in DOSP control cases. Figure 4-3 shows the time distribution of the average number of times the changes are operated in all the DOSP control cases. The result indicates that subjects change the settings more often during the first 30 minutes. About 56% of the operations are performed during the first 30 minutes. The flow rate of the last one hour and 15 minutes is relatively steady. The airflow rate characteristics are discussed in Section 4.1.2.

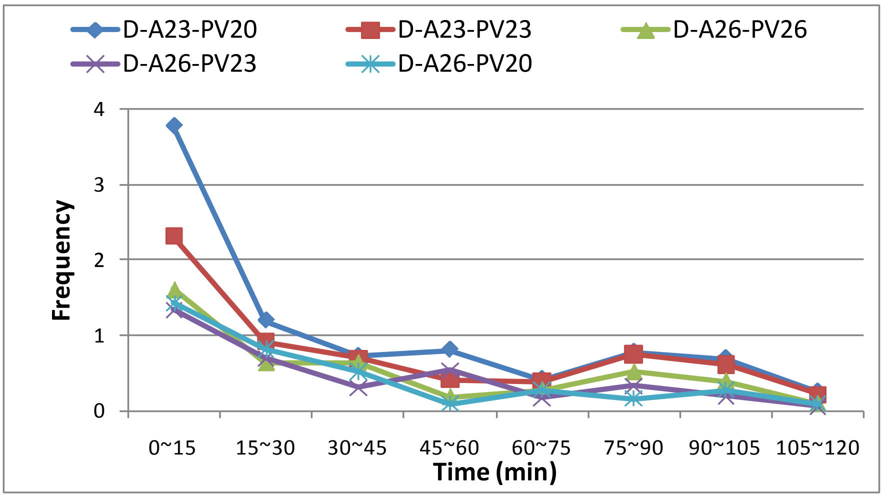


Figure 4-2 Time distribution of the operation of changes in DOSP control cases

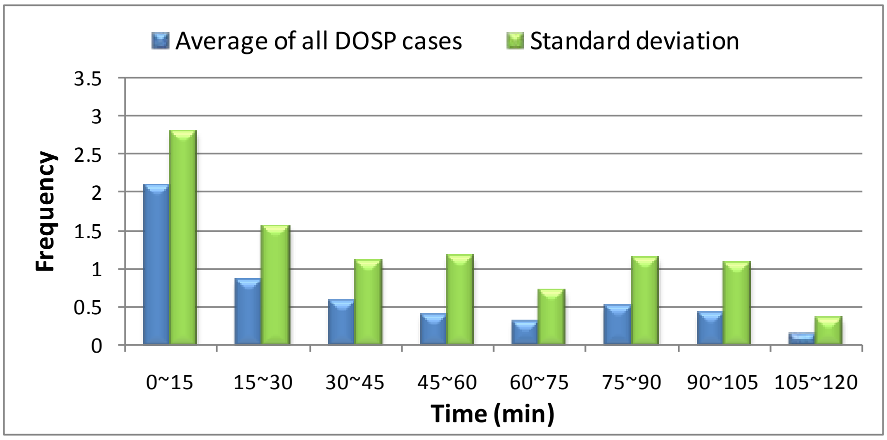


Figure 4-3 Time distribution of average number of times the changes are operated in all the DOSP control cases

Figure 4-4 shows the number of times that subjects increase and decrease the airflow rate in DOSP control cases. The average number of times subjects changed the airflow in all DOSP control cases are 5.3, with 1.8 times for the cases involving airflow rate increase and 3.5 times for those with airflow rate decrease. The number of times of the air flow rate is decreased is higher than when it is increased, especially when room ambient temperature is 23℃. Figure 4-5 shows the time distribution of the average number of times of "increase" and "decrease" in all the DOSP control cases. The result indicates that subjects seem to decrease the airflow rate rather than increase it throughout the experiment. However, this result only identifies the nature of changes to the settings. The airflow rates characteristics are discussed in Section 4.1.2 to evaluate whether subjects' preferred airflow rates reduce or not throughout the two hours duration.

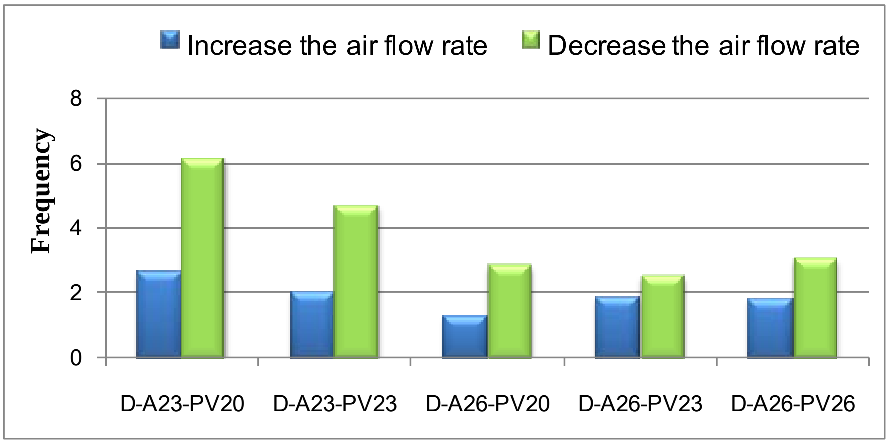


Figure 4-4 Number of times that subjects increase and decrease the airflow rate in DOSP control cases

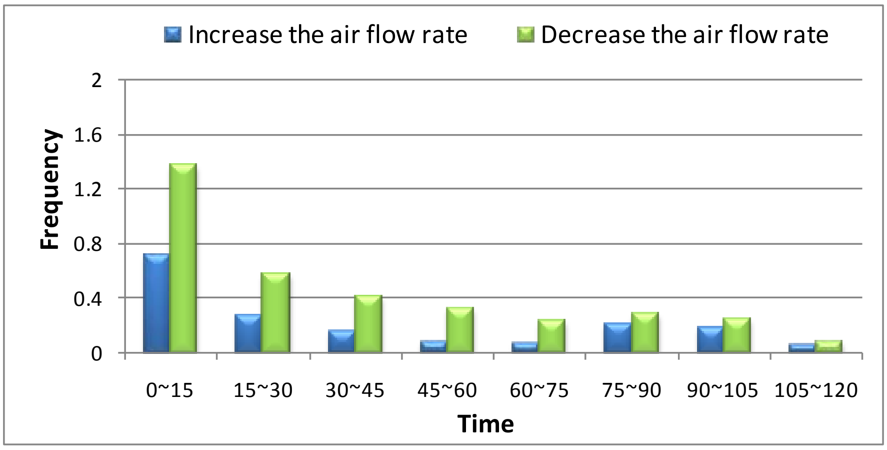


Figure 4-5 Time distribution of average number of times of increase and decrease in all the DOSP control cases

- **AFSP control**

Figure 4-6 shows the average number of times the subjects change the airflow rate during the two-hour experimental session in AFSP control cases. The average values range from 3.5 to 7.3. Unlike DOSP control, the average number of times does not vary significantly with the room ambient temperature or PV supply air temperature in AFSP control strategy.

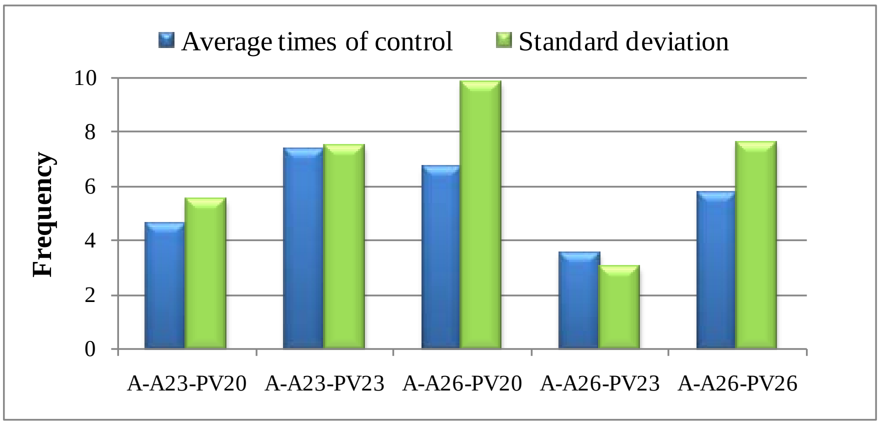


Figure 4-6 Frequency of operation of controls in AFSP control cases

Figure 4-7 shows the time distribution of the operation of changes in AFSP control cases. Figure 4-8 shows the time distribution of the average number of times the changes are operated in all the AFSP control cases. The result indicates that subjects change settings more often during the first 30 minutes. About 52% of the operations are performed during the first 30 minutes. The flow rate of the last one hour and 15 minutes is relatively steady.

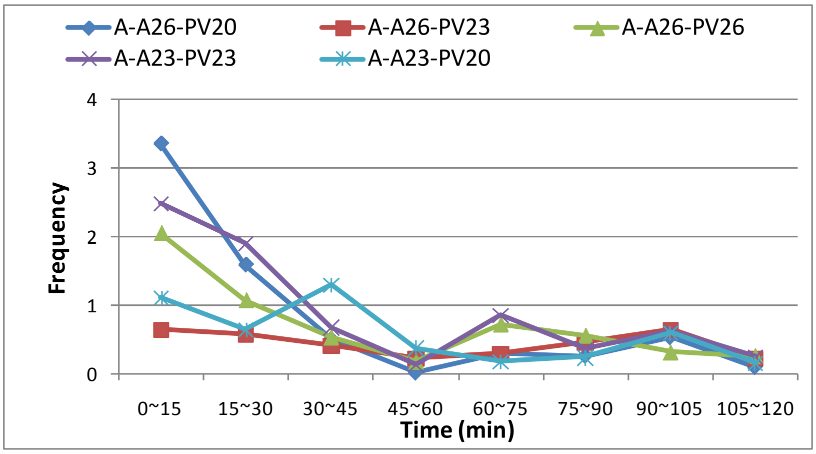


Figure 4-7 Time distribution of the operation of changes in AFSP control cases

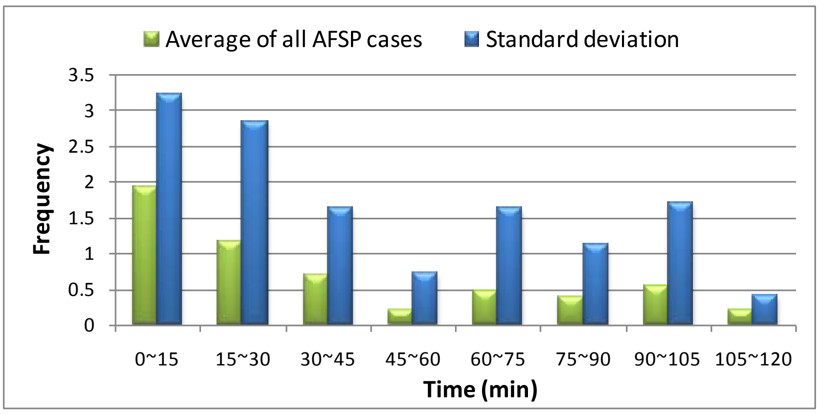


Figure 4-8 Time distribution of the average number of times the changes are operated in all the AFSP control cases

Figure 4-9 shows the number of times the subjects increase and decrease the airflow rate in AFSP control cases. The average number of times subjects changed the air flow in all the AFSP control cases are 5.6, with 1.8 times for the cases involving air flow rate increase and 3.8 times for those with air flow decrease. The frequency of "decrease" is higher than that of "increase", especially when room ambient temperature is 23℃. Figure 4-10 shows the time distribution of the average number of times of "increase" and "decrease" in all the AFSP control cases. The result indicates that subjects seem to decrease the airflow rate rather than increase it throughout the experiment.

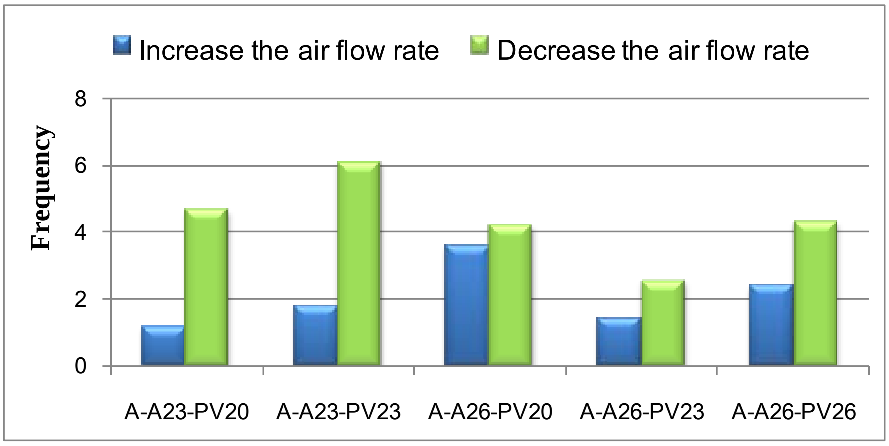


Figure 4-9 Number of times that subjects increase and decrease the airflow rate in AFSP control cases

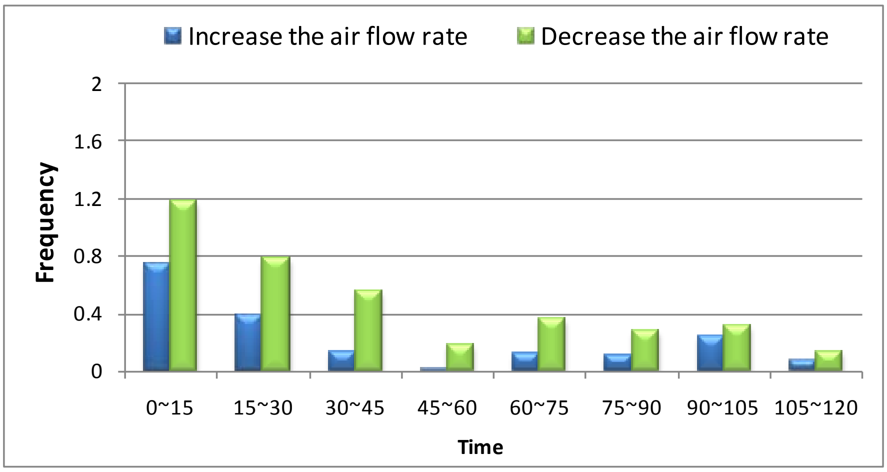


Figure 4-10 Time distribution of average number of times of increase and decrease in all the AFSP control cases

4.1.2 Subjects' preferred airflow rate

In this section, only the subjects' preferred airflow rate of DOSP control was discussed.

As discussed in Section 4.1, Figures 4-2 and 4-3 show that the number of times subjects controlled the airflow rate are relatively low after 45 minutes, so the airflow rate may be considered to be relatively steady. Figure 4-11 shows the average and standard deviation of subjects' preferred airflow rate of DOSP control cases after 45 minutes. When the room ambient temperature is 26℃, the average value of subjects' preferred airflow rate is 9.51 L/s/person. It is changed to 7.46 L/s/person when the room ambient temperature is 23℃. When PV supply air temperature is 20℃, the subjects' preferred airflow rate at room ambient temperature of 26℃ is 16.8% higher than that at room ambient temperature of 23℃. When PV supply air temperature is 23℃, the subjects' preferred airflow rate at room ambient temperature of 26℃ is 31.2% higher than that at room ambient temperature of 23℃. It is clearly shown that when room ambient temperature increases, the subjects' preferred airflow rates increase. The relationship between the PV supply air temperature and the subjects' preferred airflow rates seems to be not clear. The reasons, why subjects' preferred airflow rates increase with the ambient room temperature, may be as follows:

- Subjects need more air movement to offset the impacts of increased temperature on thermal comfort and indoor air quality;
- Subjects need more conditioned air to cool down the body;
- And subjects need more fresh air to improve PAQ, although not substantiated through experiments in this study.

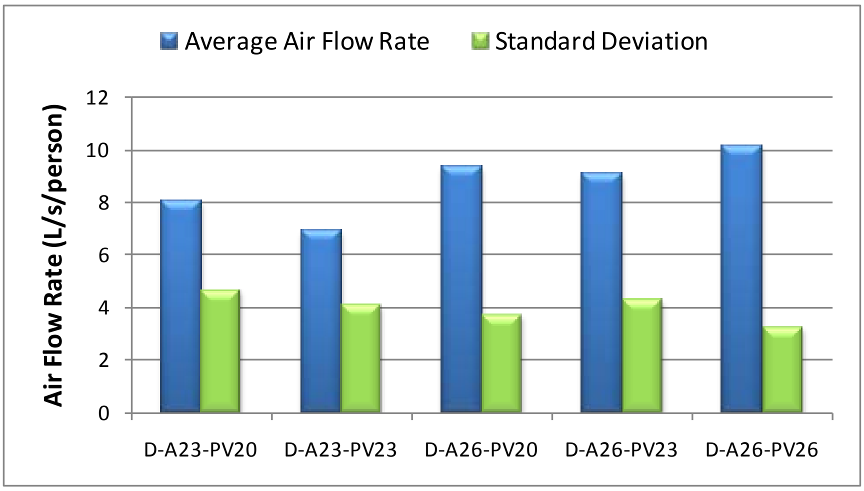


Figure 4-11 Subjects' preferred airflow rate of DOSP control after 45 minutes

As discussed in Section 4.1, subjects seem to decrease the airflow rate rather than increase it throughout the experiment, especially when room ambient temperature is 23℃. Figure 4-12 shows the time distribution of subjects' preferred airflow rate of DOSP control at room ambient temperature of 23℃. When PV supply air temperature is 20℃, the average airflow rate is 10.0 L/s/person during the first 15 minutes and then reduces to 7.0 L/s/person (30% decrease) during the last 15 minutes. The airflow rates seem to be relatively steady after 90 minutes. When PV supply air temperature is 23℃, the average airflow rate is 9.1 L/s/person during the first 15 minutes and then reduces to 6.1 L/s/person (33% decrease) during the last 15 minutes. The airflow rates seem to be relatively steady after 60 minutes. The results show that subjects' preferred airflow rate clearly reduce during the first one hour and then slightly reduce during the last hour. The reason may be that the subjects are exposed to higher air temperature (e.g. outdoor air) before they come in the FEC, therefore, the subjects need more airflow to cool down the body at the beginning and after exposing themselves sometime in the cool environment, they may feel that the room temperature (23℃) is too cool so that they reduce the airflow rate. Subjects' preference for air movement is changing with time, which indicates that individual control is helpful in these conditions.

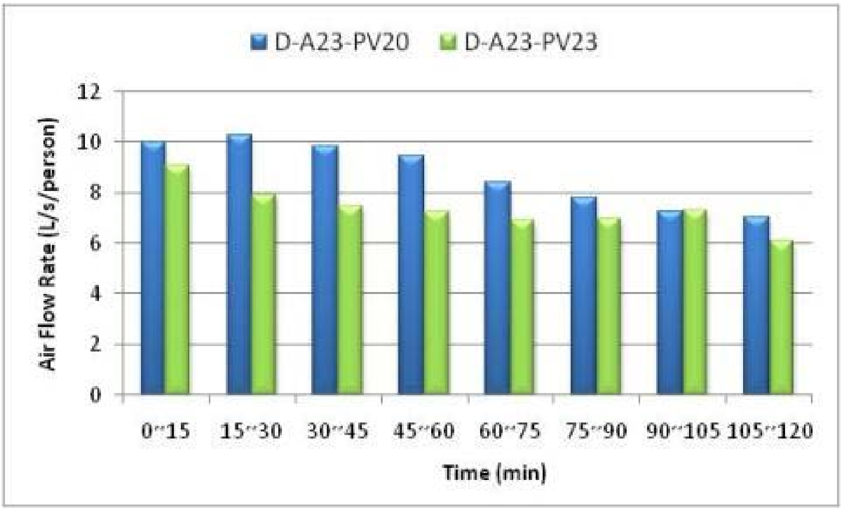


Figure 4-12 Time distribution of subjects' preferred airflow rate of DOSP control when room ambient temperature is 23℃

Figure 4-13 shows the time distribution of subjects' preferred airflow rate of DOSP control when room ambient temperature is 26℃. When PV supply air temperature is 20℃, the average airflow rate is 10.6 L/s/person during the first 15 minutes and then reduces to 8.7 L/s/person (18% decrease) during the last 15 minutes. The airflow rates seem to be relatively steady after 75 minutes. When PV supply air temperature is 23℃, the average airflow rate is 10.4 L/s/person during the first 15 minutes and then slightly reduces to 9.1 L/s/person (12% decrease) during the last 15 minutes. The airflow rates seem to be relatively steady after 60 minutes. When PV supply air temperature is 26℃, the airflow rate seems to be relatively steady. The results indicate that subjects' preferred airflow rate only slightly reduce throughout the experiment.

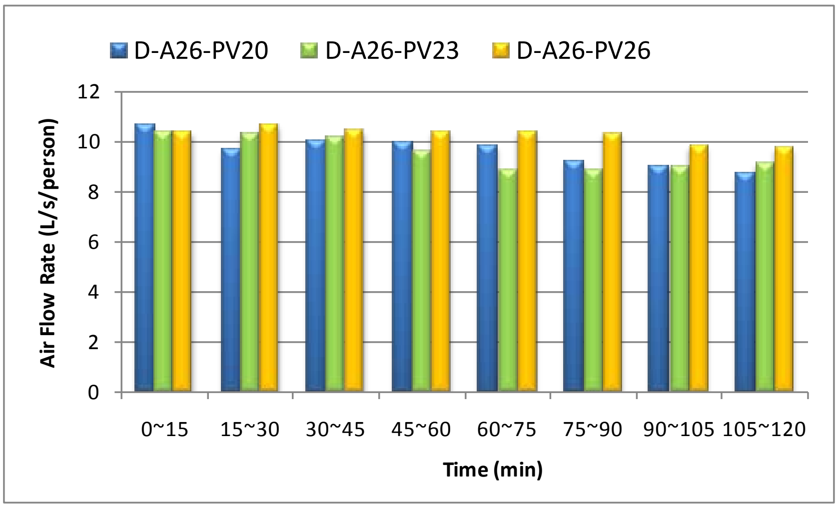


Figure 4-13 Time distribution of subjects' preferred airflow rate of DOSP control when room ambient temperature is 26℃

Figure 4-14 shows the subjects' preferred airflow rate in D-A23-PV20 case. The airflow rates range from 0.2 to 14.7 L/s/person. The similar results are found in other conditions with DOSP control as shown in Figure 4-15. The results show that subject' preferred airflow rate are quite different, which indicate that individual control of airflow rate may be helpful to fully meet occupants' requirements.

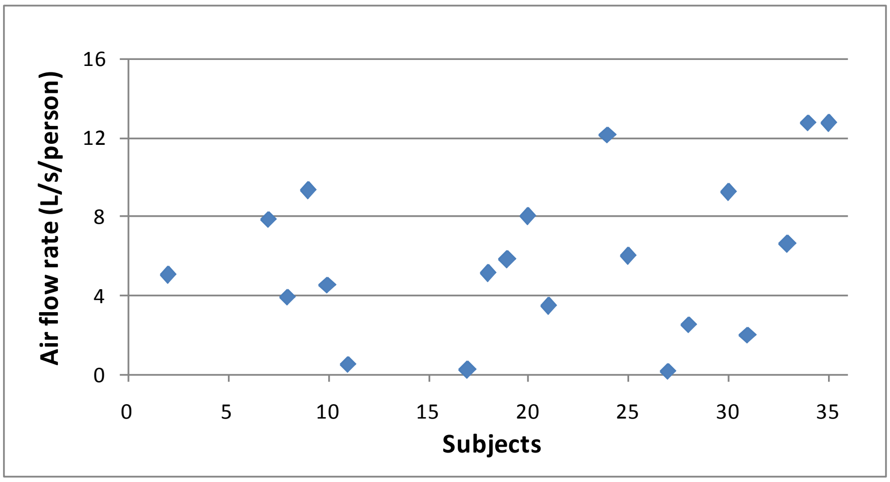


Figure 4-14 Subjects' preferred airflow rate of D-A23-PV20 case

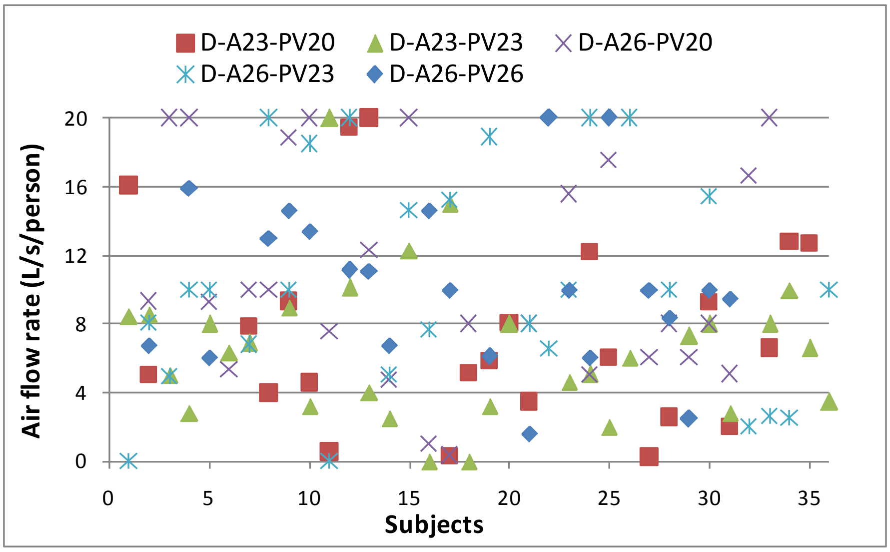


Figure 4-15 Subjects' preferred airflow of all DOSP control cases

4.1.3 Comparison with similar studies

In the subjective study of Yang et al. (2010), which was conducted in the same FEC as described in Section 3.2.1, it was found that under the same PV air temperature, the subjects' preferred air movement increased when the room air temperature increased. The results from this study confirm this conclusion. However, some other observations from the earlier study are not supported by the present study. Yang et al. (2010) found that under the same room air temperature, the subjects' preferences of air movement increased when the PV air temperature increased. However, in this research, there is no clear relationship between the PV air temperature and the subjects' preferred airflow rates under the same ambient air temperature. The reason for this discrepancy might be because the PV-ATDs used in this study are closer to the occupants and they provide better ventilation effectiveness diminishing the effects of increased air temperature.

4.2 Subject assessment

Human responses of the PV-ICA system were collected through questionnaires. The impact of the room air temperature, PV supply air temperature and control strategy on subjects' thermal comfort and thermal sensation; air movement, perception, acceptability and preference; subjects' clothing and the general concept of this PV-ICA system is analyzed and discussed in this section. The results related to indoor air quality and sick building syndromes are shown in Appendix B.

4.2.1 Thermal comfort and thermal sensation

- **Thermal comfort acceptability**

Subjects' thermal comfort acceptability of the PV-ICA system is shown in Figure 4-16. The thermal comfort acceptability varies from 68.9 to 73.6, and has an average value of 70.8 and standard deviations of about 15. The acceptability of thermal comfort conditions does not vary significantly with the room ambient temperature, PV temperature or control strategy. About 91.8% of the subjects feel the environment conditions are acceptable. Subjects have control over the airflow through the PV ATDs and the perceived thermal discomfort may be overcome by adjusting the airflow rate. This result indicates that PV-ICA system can create the environmental conditions which are acceptable for most occupants in room.

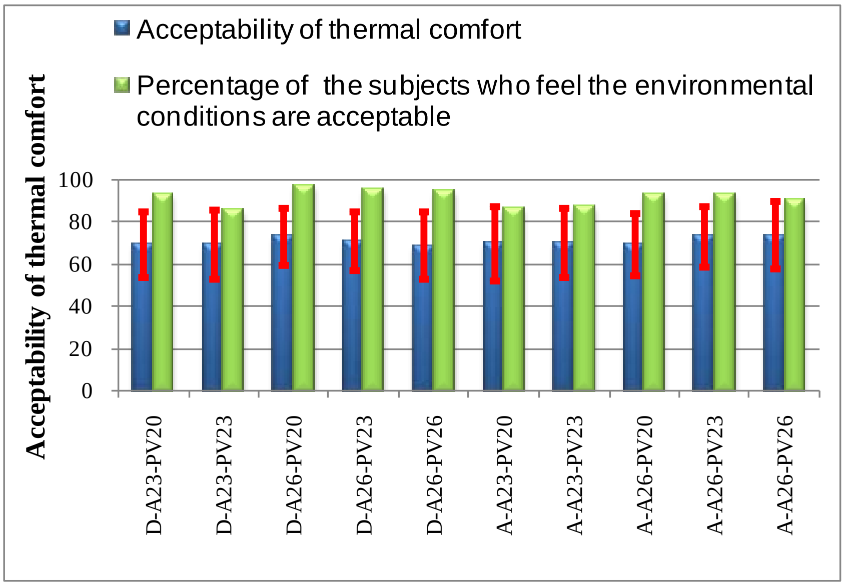


Figure 4-16 Subjects' thermal comfort acceptability

- **Whole body thermal sensation**

Figure 4-17 shows the whole body thermal sensation of subjects at 10 experimental conditions. When room ambient temperature is 26℃, the whole body thermal sensations are between -0.61 and -0.35, while when room ambient temperature is 23℃, the whole body thermal sensations are between -1.09 and -1.38. Figure 4-18 shows the percentage of subjects who reported whole body thermal sensation between slightly cool (-1) and slightly warm

(+1). When room ambient temperature is 26℃, the average value is about 91% and the value is only 65% when room ambient temperature is 23℃. The results show that the subjects may feel draughty when room ambient temperature is 23℃. When room ambient temperature is 26℃, the thermal conditions are closer to neutral as compared to those when room ambient temperature is 23℃, which indicate that whole thermal sensations are largely improved when room ambient temperature is 26℃. This result is significant as higher ambient room temperature may have the potential of energy savings.

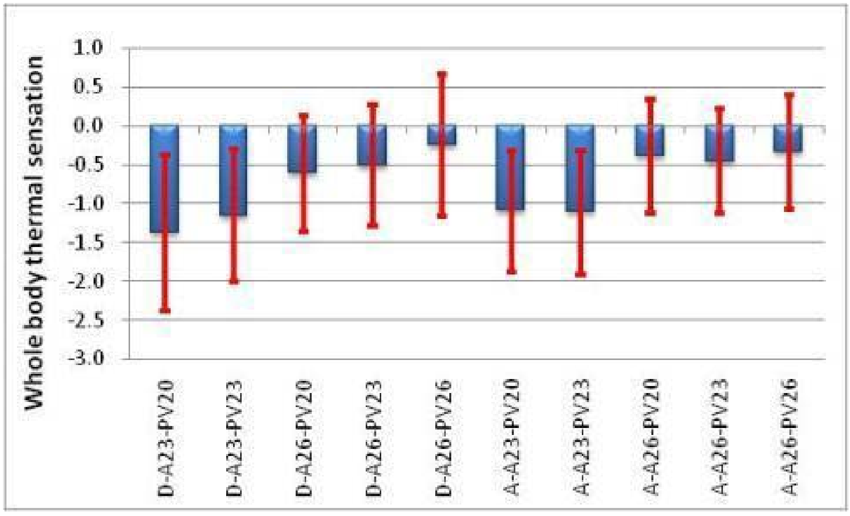


Figure 4-17 Whole body thermal sensation

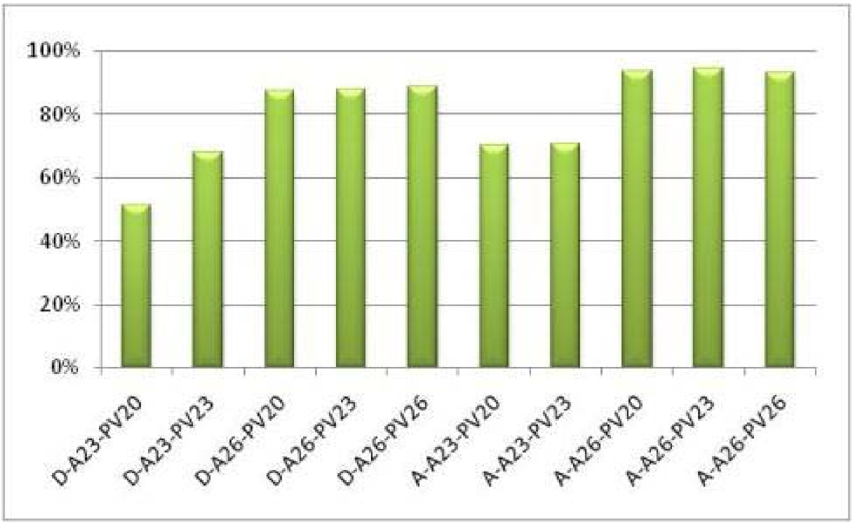


Figure 4-18 Percent of the thermal sensation between slightly cool (-1) and slightly warm (+1)

- **Local thermal sensation**

The impact of the room ambient temperature and PV supply air temperature on different segments of subjects are analyzed to show the possible reasons of local thermal discomfort. Figures 4-19 and 4-20 present the local thermal sensation of subjects with DOSP control and AFSP control. When room ambient temperature is 26℃, the thermal sensations of each body segment are closer to neutral compared with those when room ambient temperature is 23℃. When room ambient temperature is 26℃, it is clearly shown that the thermal sensation of face is slightly lower than that of other body segments. This may be the result of directly supplying conditioned air to the breathing zone. The air movement perception, acceptability and performance of the face, hand and shoulder are discussed in Section 4.2.2. When room ambient temperature is

23℃, the thermal sensation of head, face, shoulder, neck, arm, hand and foot are all lower than 0.5. As discussed in Section 4.1.2, when room ambient temperature is 23℃, the subjects' preferred airflow rates are lower than those when room ambient temperature is 26℃. Although the airflow rates are lower, the subjects still feel cool when room ambient temperature is 23℃. These results indicate that thermal sensations are largely improved when room ambient temperature is 26℃ as well.

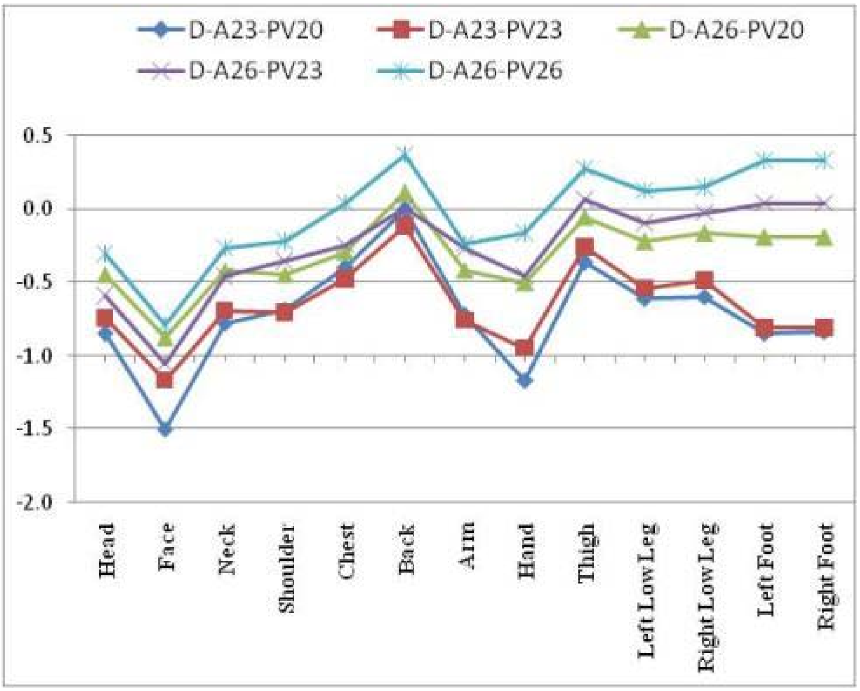


Figure 4-19 Local thermal sensation of DOSP control

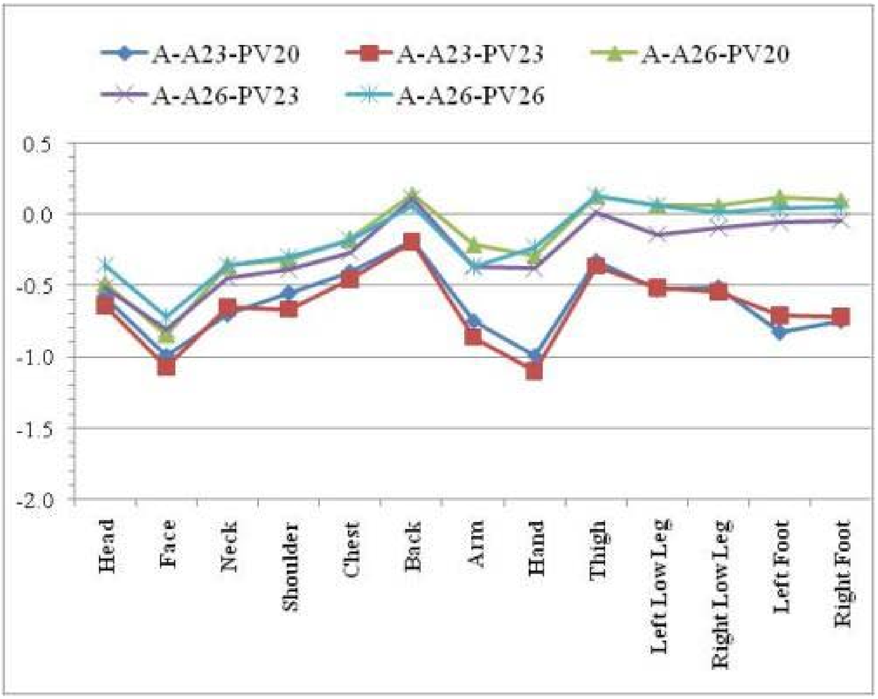


Figure 4-20 Local thermal sensation in AFSP control

Figure 4-21 shows the relationship of whole body thermal sensation and whole body thermal comfort acceptability of all experimental conditions. Figure 4-22 shows the distribution of the subjects' report of whole body thermal sensation. About 95% of the subjects reported that the whole body thermal sensation was neutral (0), slightly cool (-1) or cool (-2). As shown in Figure 4-21, the average value of thermal comfort acceptability is 67 when whole body thermal sensation is cool (-2). Although some subjects reported that the whole body thermal sensation is cool (-2), they felt that the thermal environment is acceptable. Although both the whole body thermal sensation and body segments thermal sensation are slightly lower when room ambient temperature is 23℃, the slightly lower thermal sensation does not appear to cause thermal discomfort. This result indicates that PV-ICA system may extend the subjects' comfort zone.

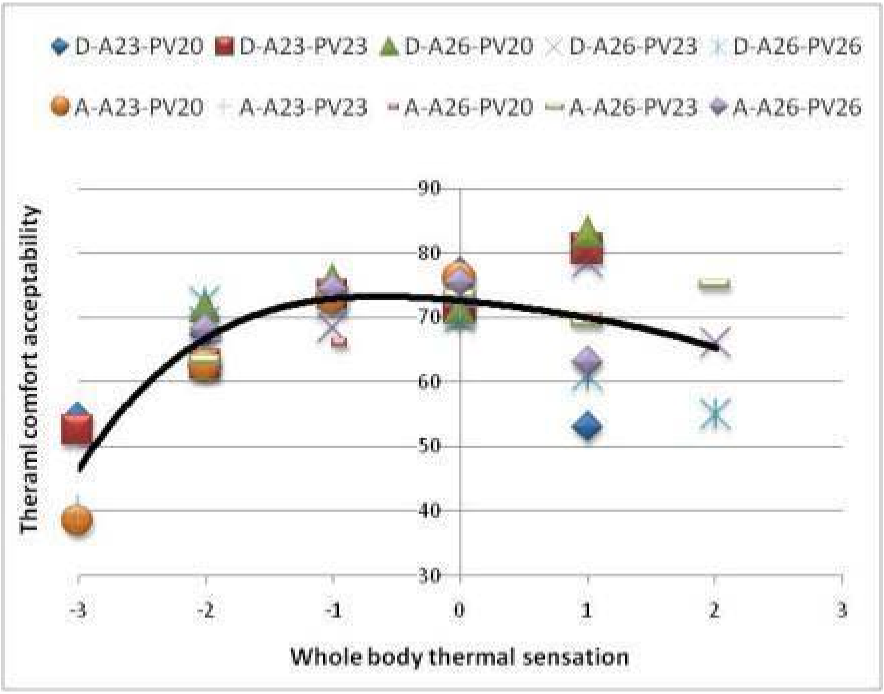


Figure 4-21 Relationship of whole body thermal sensation and whole body thermal comfort acceptability

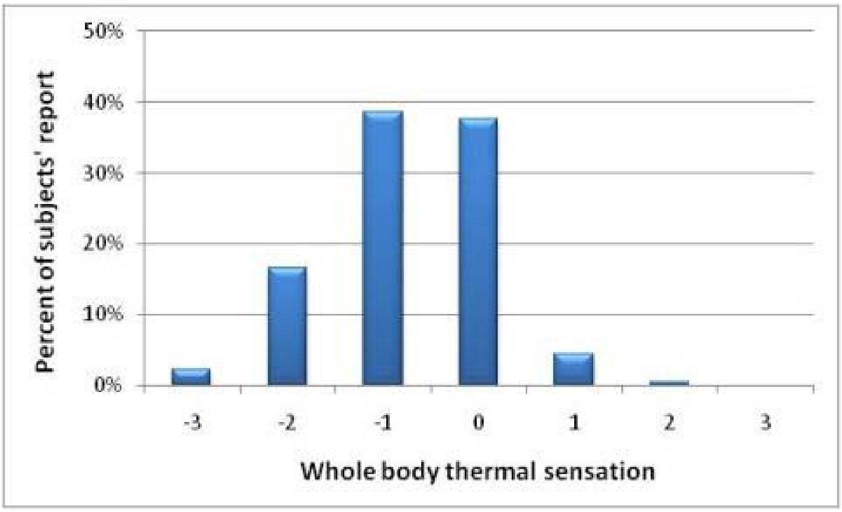


Figure 4-22 Distribution of subjects' report of whole body thermal sensation

4.2.2 Air movement perception, acceptability and preference

➢ **Air movement perception**

Figure 4-23 shows the percentage of subjects who feel the air movement on their head, face and shoulder. 78.6% of subjects feel the air movement on their face. In A-A23-PV20, A-A23-PV23, and A-A26-PV23 conditions, the percentage of subjects who feel the air movement on their face is slightly lower than the other conditions. The percentage of subjects who feel the air movement on head varies from 40.5% to 54.2% and has an average of 46.6%. The percentage of subjects who feel air movement on shoulder is 34.4%. The similar results of all the experimental conditions indicate that subjects' perception of air movement do not vary significantly with the room ambient temperature, PV temperature or control strategy.

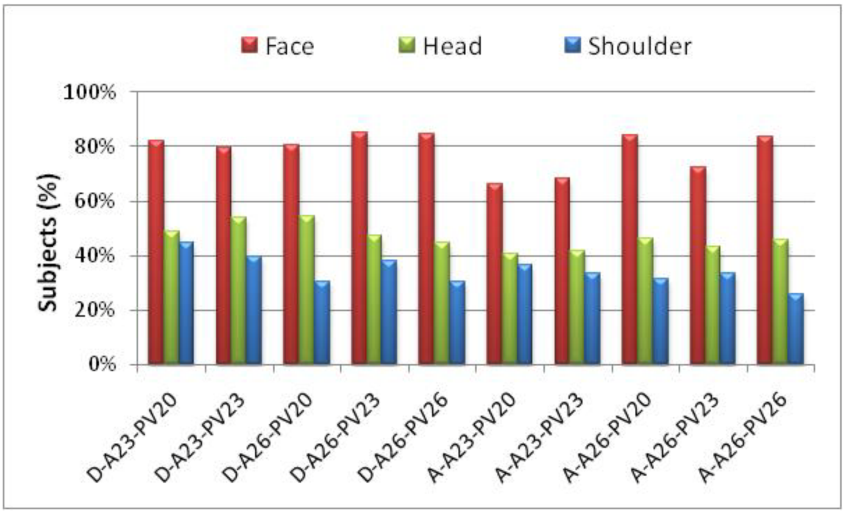


Figure 4-23 Percentage of subjects who feel the air movement on face head and shoulder

Figure 4-24 shows the perception of air movement on face, head and shoulder. Table 3-5 shows the 7-point numerical scale for the perception of air movement, where -3 means the air movement is much too still and +3 means the air movement is much too breezy. Figure 4-25 shows the distribution of the subjects' perception of air movement on face, head and neck. About 85% of those subjects who feel air movement report that the perception of air movement is just right (0) or slightly breezy (+1). The average perception of air movement of all experimental conditions is 0.54 on face, 0.33 on head and 0.23 on shoulder. In DOSP control, when PV supply air temperature is lower, the perception of air movement is higher.

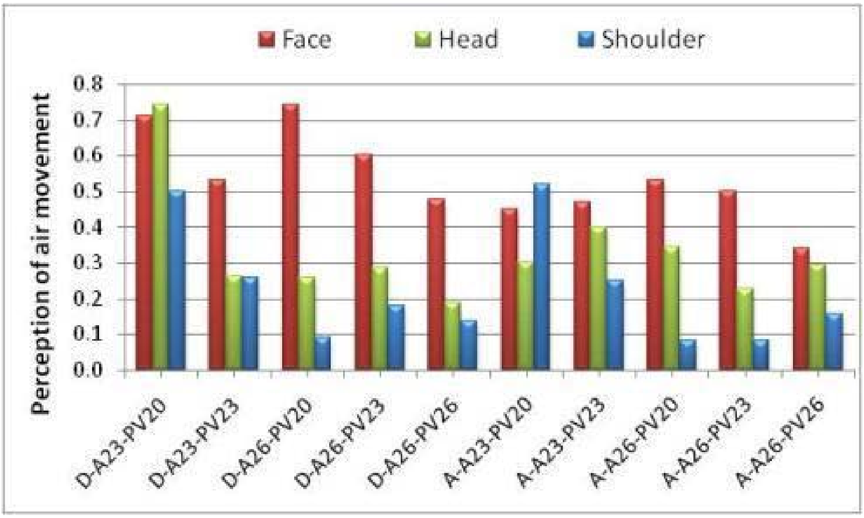


Figure 4-24 Perception of air movement on face, head and neck

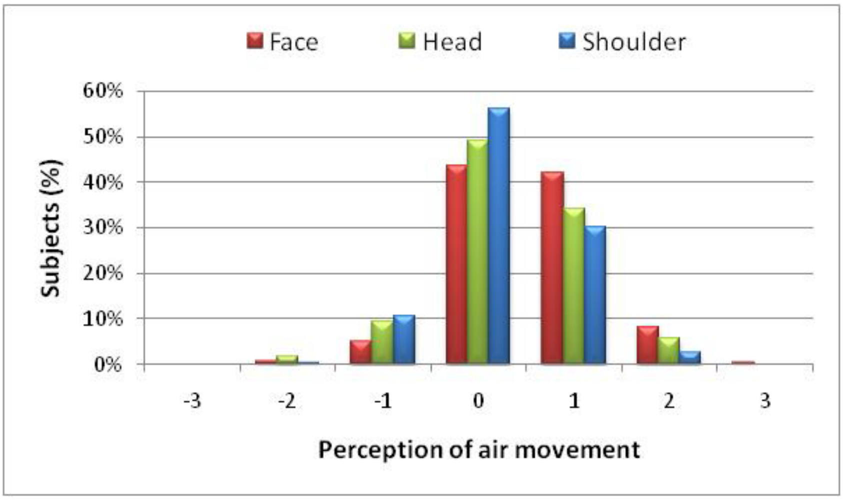


Figure 4-25 Distribution of subjects' perception of air movement on face, head and neck

➢ **Air movement acceptability**

Figure 4-26 shows the subjects' acceptability of air movement on facial region. The values vary from 64.5 to 74.7 with an average of 68.7. About 83.1% of subjects feel the air movement is acceptable. The acceptability of air movement does not vary significantly with the room ambient temperature, PV supply air temperature or control strategy. The similar results of the subjects' acceptability of air movement on the top of head and neck are found as shown in Figures 4-27 and 4-28.

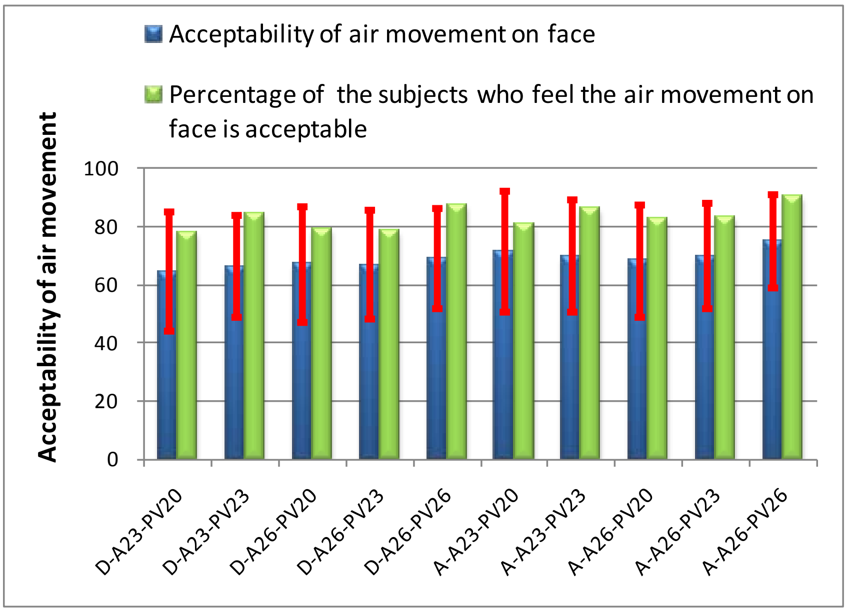


Figure 4-26 Subjects' acceptability of air movement on facial part

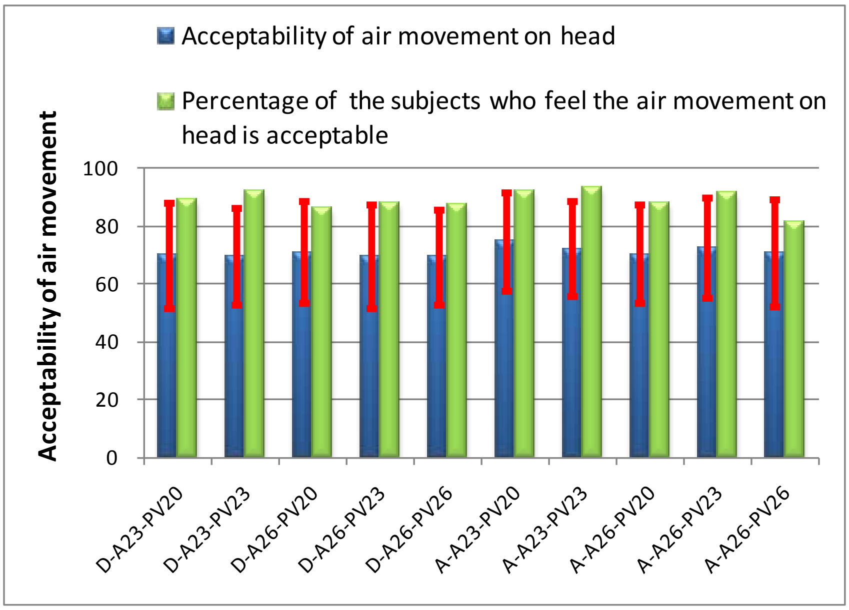


Figure 4-27 Subjects' acceptability of air movement on the top of head

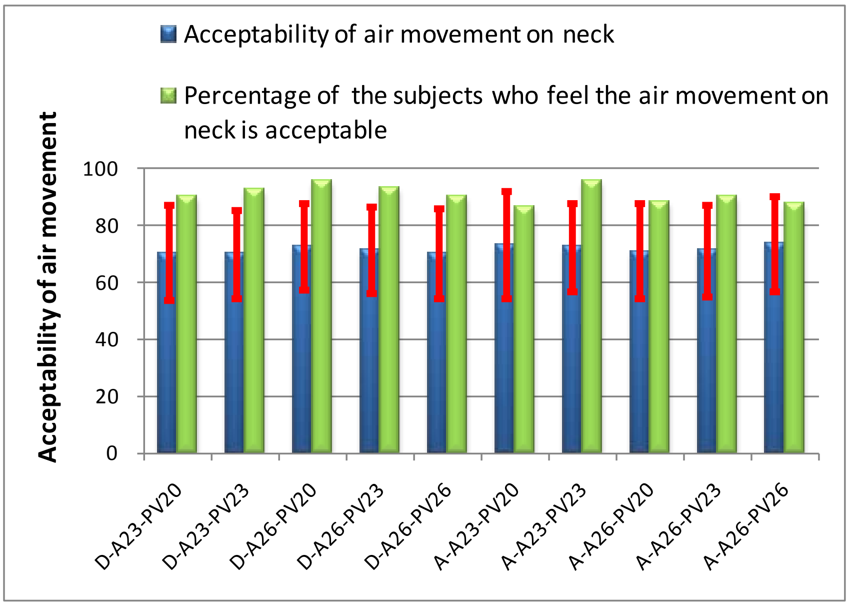


Figure 4-28 Subjects' acceptability of air movement on neck

- **Air movement preference**

Figure 4-29 shows the preference for air movement on the facial region. About 65% of subjects prefer no change of the air movement, 30% of subjects prefer less air movement and only 5% of the subjects prefer more air movement. Figure 4-30 shows the preference for air movement on the top of head. About 70% of subjects prefer no change of the air movement, 10% of subjects prefer less air movement and 20% of the subjects prefer more air movement. Figure 4-31 shows the preference for air movement on neck. About 77% of subjects prefer no change of the air movement, 8% of subjects prefer less air movement and 15% of the subjects prefer more air movement. Figure 4-32 shows the relationship between preference for air movement and acceptability of air movement on head, face and neck. When subjects prefer less air movement, their acceptability of air movement is only about 50. The result indicates the higher air movement on the face may be the reason of low acceptability of air movement. New PV ATD should be designed to overcome this disadvantage.

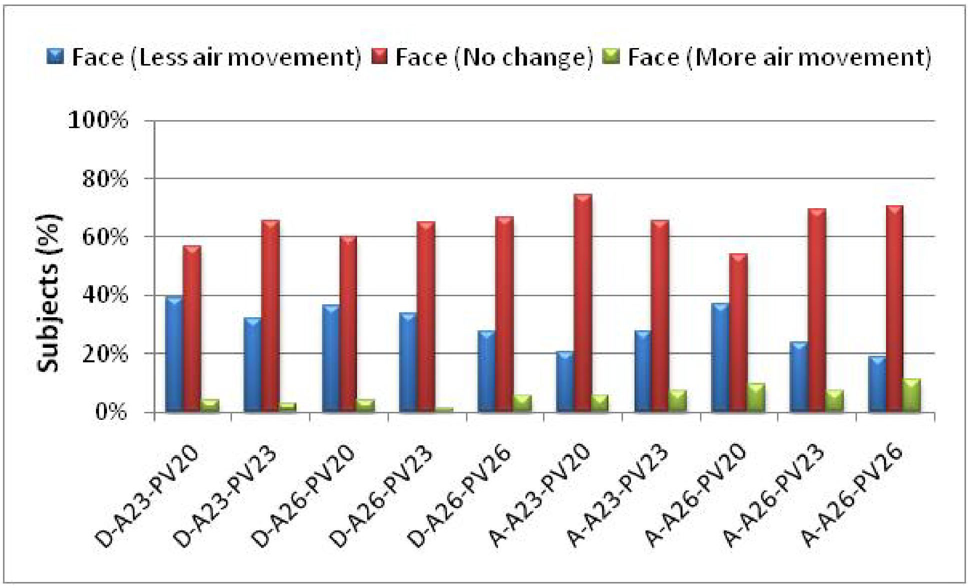


Figure 4-29 Preference for air movement on the facial part

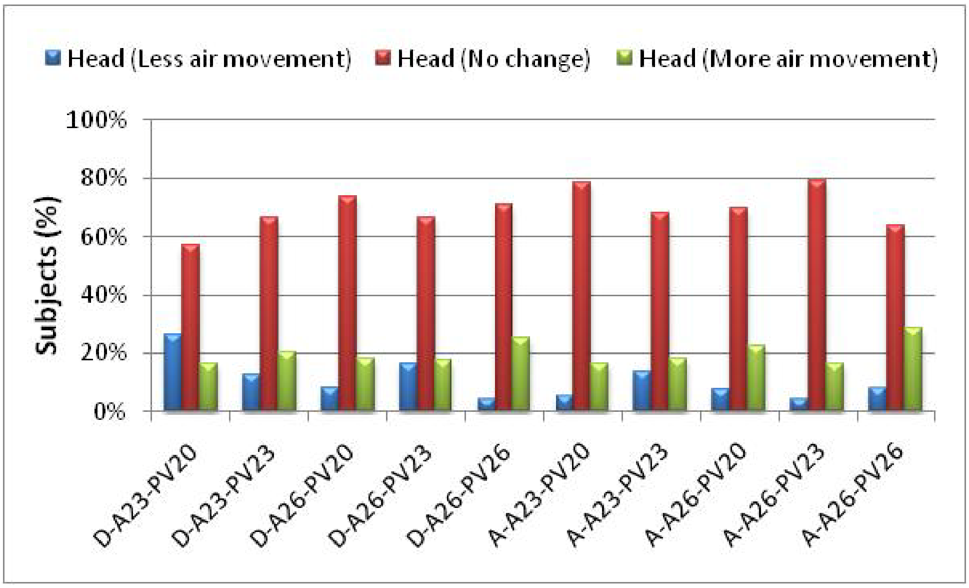


Figure 4-30 Preference for air movement on the top of head

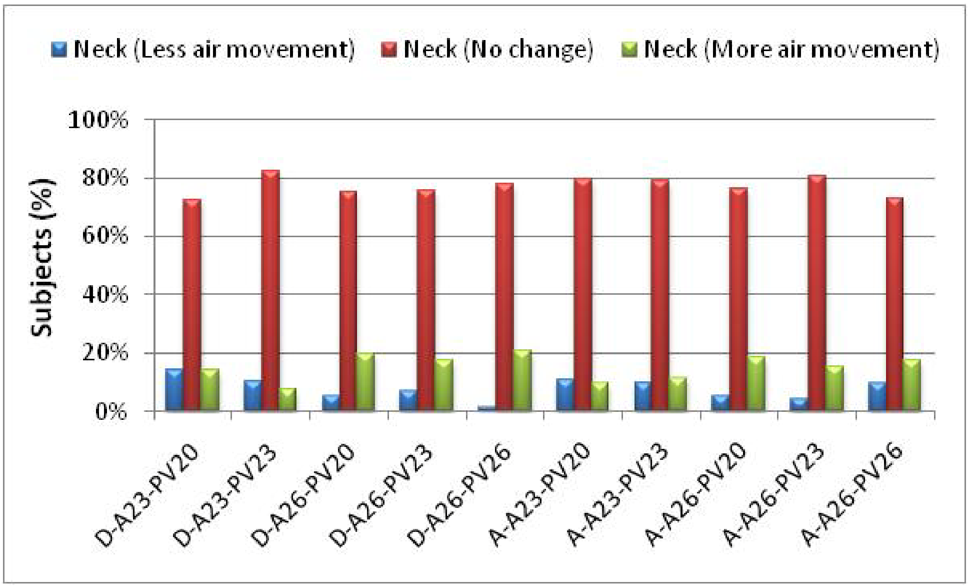


Figure 4-31 Preference for air movement on neck

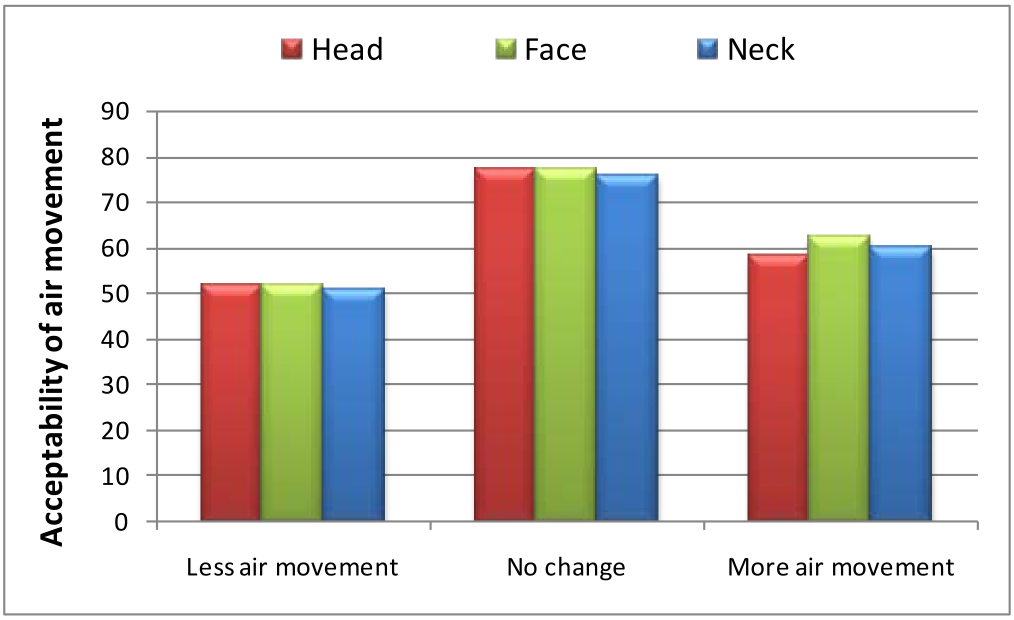


Figure 4-32 Relationship between preference for air movement and acceptability of air movement on head, face and neck

4.2.3 Subjects' clothing

In this study, the subjects are not allowed to change their clothing during the experiments. Figure 4-33 shows the insulation for entire ensemble of all experimental conditions. The average value is about 0.51 clo. About 88% of subjects wore short-sleeved shirts or blouse. About 36% of subjects wore short trouser; about 25% of subjects wore normal trouser; and about 25% of subjects wore Bermudas. Figure 4-34 shows the subjects' insulation for entire ensemble of D-A23-PV20 case. The insulation varies from 0.38 to 0.95. The similar results were found in other experimental conditions which are shown in Appendix B.

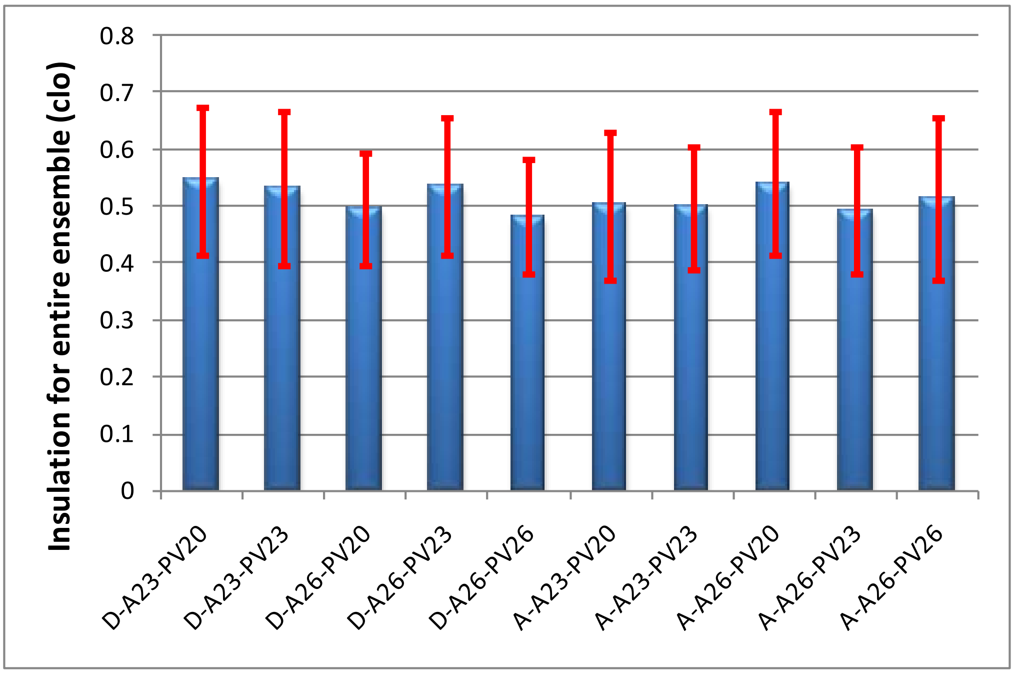


Figure 4-33 Insulation for entire ensemble of all experimental conditions

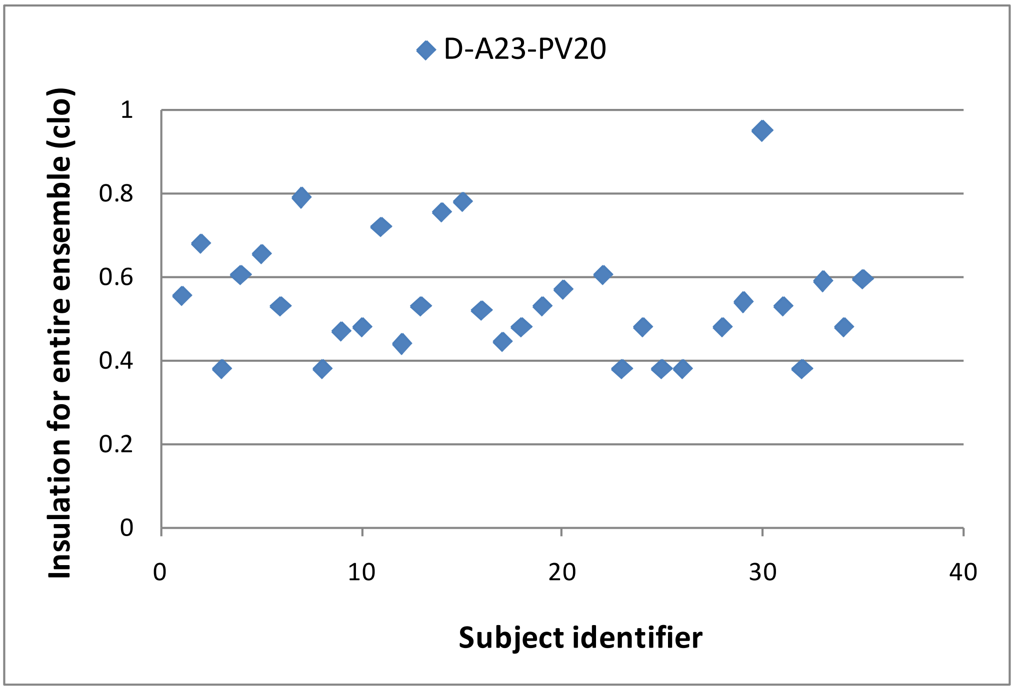


Figure 4-34 Subjects' insulation for entire ensemble of D-A23-PV20 case

4.2.4 General impression of the PV system

As introduced in Section 3.4 questionnaire design, the three questions related to general impression of the PV-ICA system are analyzed and discussed in this section.

Figure 4-35 shows the average values of human response to the performance of the PV ATD of each experimental condition. The average values range from 74.5 to 94.5, which means that the performance of this PV ATD is reasonably good.

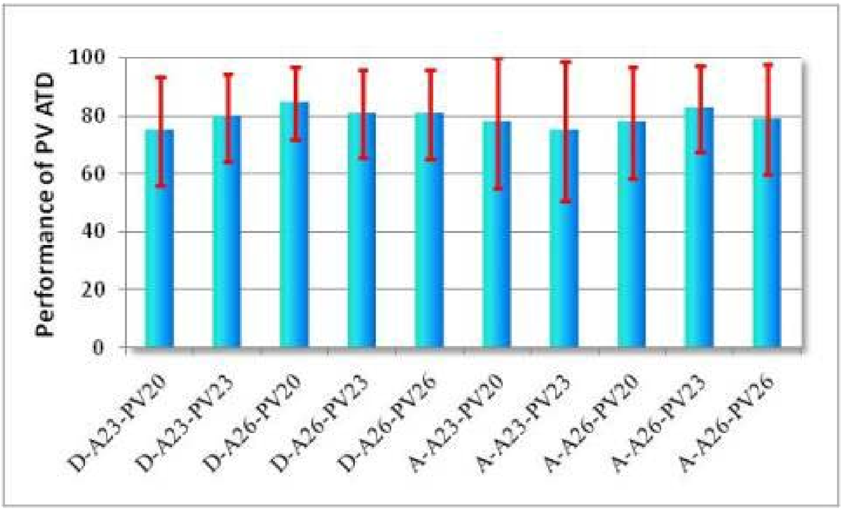


Figure 4-35 Performance of PV ATD

Figure 4-36 shows the average values of human response to the frequency to adjust the PV ATD of each experimental condition. The average values range from 70.4 to 81.7 and the average of the standard deviations is 21.5. The result indicates that frequent interventions are not necessary; however, few interventions are needed.

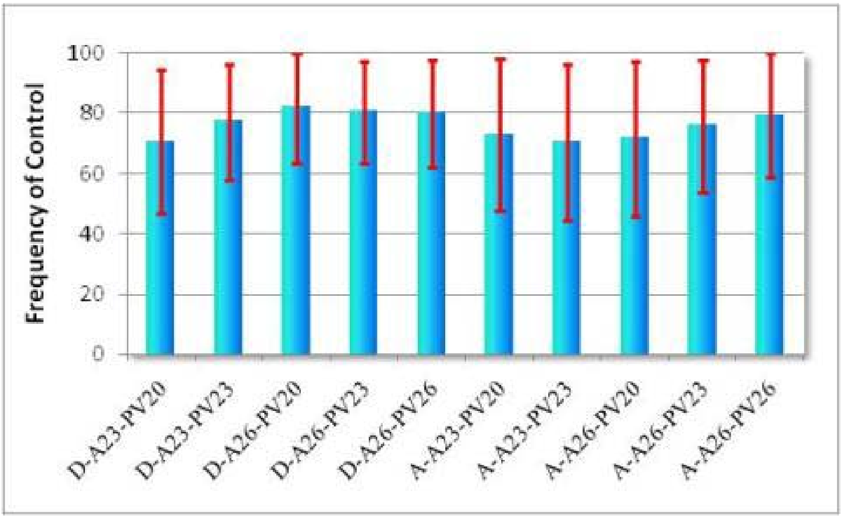


Figure 4-36 Frequency to adjust the PV ATD

Figure 4-37 shows the average values of human response to the difficulty level of controlling the PV ATD of each experimental condition. The average values range from 75.6 to 83.4 and the average of the standard deviations is 14. The high average values and low standard deviations indicate that it is easy to control the PV ATD through the software interface and conventional mechanical control is not necessary.

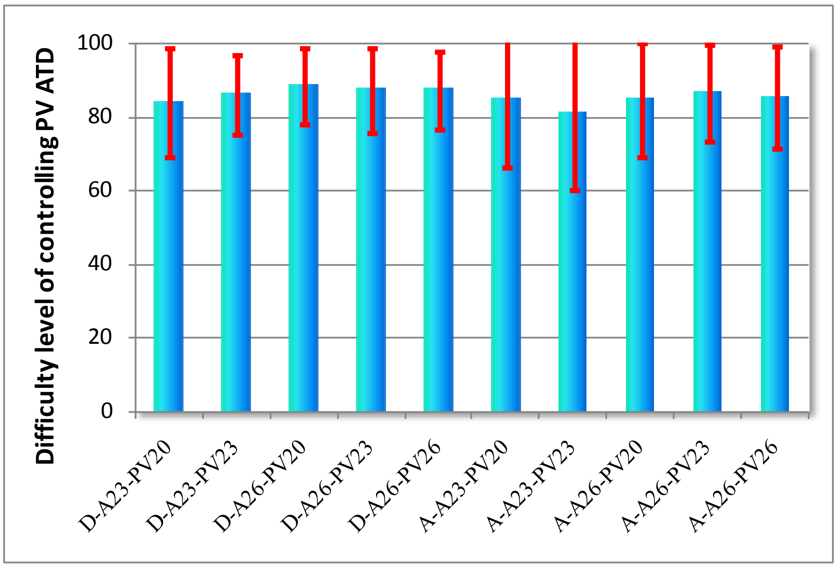


Figure 4-37 Difficulty level of controlling the PV ATD

4.3 Energy performance

4.3.1 Comparison of energy simulation results with measurement data

There are two main components of the cooling load: FEC room cooling load and ventilation cooling load. The fan power and heater power also contribute to the cooling load. These loads are simulated and the results are compared with the measured cooling-water load from the PV AHU and the ambient AHU. The objective of this exercise is to identify the accuracy of the model and make sure that the energy simulation results can represent the energy consumption of the HVAC systems in FEC, so that the energy model can be used for further analysis.

- **FEC cooling load simulation results**

In the first stage of the simulation, the cooling loads of the FEC of all the experimental conditions were simulated using EnergyPlus. Table 4-1 shows the general information of the simulation conditions, which include the experimental conditions, date, the room air temperature and RH, outdoor air temperature and RH.

Table 4-1 Room and outdoor air conditions of all simulation conditions

Case Ref	Date	Room Temp.	Room RH	Outdoor Temp.	Outdoor RH
D-A23-PV20-1	Oct-6, AM	22.8	53.0	26.9	97.5
D-A23-PV20-2	Oct-6, PM	23.2	51.5	29.2	83.7
D-A23-PV20-3	Oct-7, AM	23.1	52.1	29.1	86.6
D-A23-PV23-1	Oct-7, PM	23.4	51.1	30.5	74.2
D-A23-PV23-2	Oct-8, AM	23.1	52.2	28.9	87.3
D-A23-PV23-3	Oct-8, PM	23.3	51.4	26.9	90.9
D-A26-PV20-1	Oct-14, AM	26.1	44.7	28.7	83.1
D-A26-PV20-2	Oct-14, PM	25.9	44.8	27.3	94.7
D-A26-PV20-3	Oct-15, AM	26.0	44.6	28.4	81.3
D-A26-PV23-1	Oct-12, PM	26.1	44.6	27.2	95.6
D-A26-PV23-2	Oct-13, AM	26.2	44.4	28.0	95.3
D-A26-PV23-3	Oct-13, PM	26.0	44.6	27.9	91.7
D-A26-PV26-1	Oct-9, AM	26.8	43.2	29.8	84.3
D-A26-PV26-2	Oct-9, PM	26.7	43.2	31.9	66.0
D-A26-PV26-3	Oct-12, AM	25.8	45.4	29.3	86.6
A-A23-PV20-1	Oct-23, PM	22.9	52.6	31.0	78.3
A-A23-PV20-2	Oct-26, AM	22.2	54.7	29.3	84.9
A-A23-PV20-3	Oct-26, PM	22.7	53.1	30.7	74.6
A-A23-PV23-1	Oct-22, AM	22.7	53.1	29.9	84.9
A-A23-PV23-2	Oct-22, PM	23.1	52.1	30.5	80.8
A-A23-PV23-3	Oct-23, AM	22.9	52.7	29.8	85.3
A-A26-PV20-1	Oct-15, PM	26.1	44.5	27.6	94.7
A-A26-PV20-2	Oct-16, AM	26.6	44.0	25.2	96.8
A-A26-PV20-3	Oct-16, PM	25.9	45.0	30.0	78.8
A-A26-PV23-1	Oct-19, AM	26.0	44.8	29.9	69.2
A-A26-PV23-2	Oct-19, PM	26.1	44.4	31.4	72.3
A-A26-PV23-3	Oct-20, AM	26.5	43.6	29.8	84.4
A-A26-PV26-1	Oct-20, PM	26.6	43.2	31.3	77.2
A-A26-PV26-2	Oct-21, AM	26.3	43.9	29.3	88.7
A-A26-PV26-3	Oct-21, PM	26.5	43.4	29.5	84.6

Figure 4-38 and 4-39 show the room cooling load components of FEC at room temperature 23℃ and 26℃. The results indicate that the internal heat gain (lighting and equipment) and the occupancy load are almost the same for all conditions. The surface convection which includes the convection load of the walls and windows varies among the experiment conditions as both room air and outdoor air conditions are varied. The total room cooling loads of FEC at room temperature of 23℃ range from 6.8kW to 7.7kW as shown in Figure 4-38; the values range from 5.0Kw to 6.7kW when room temperature is 26℃ as shown in Figure 4-39.

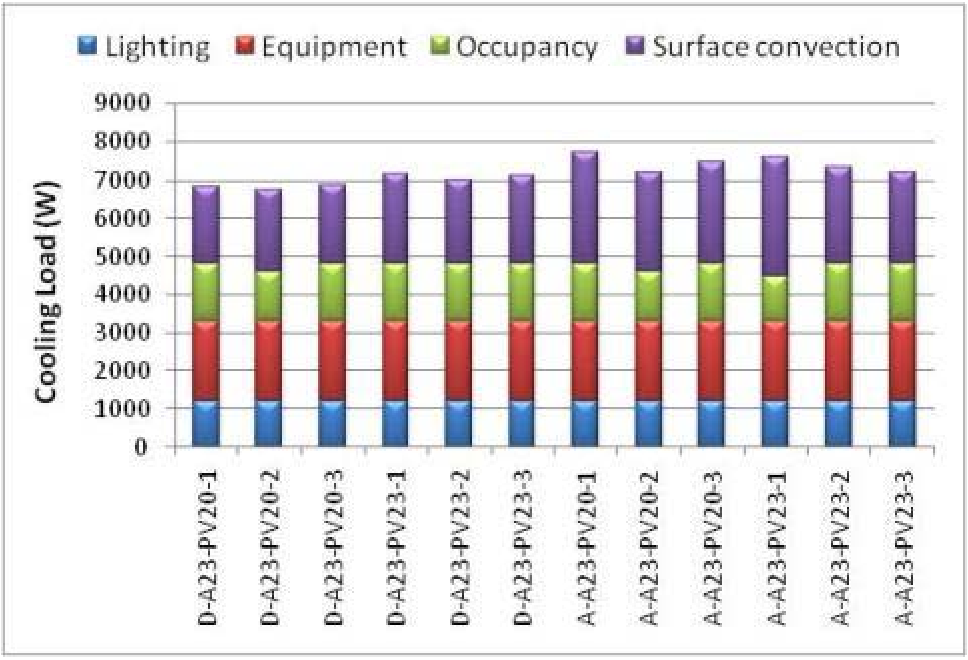


Figure 4-38 Cooling load components of FEC at room temperature 23 ℃

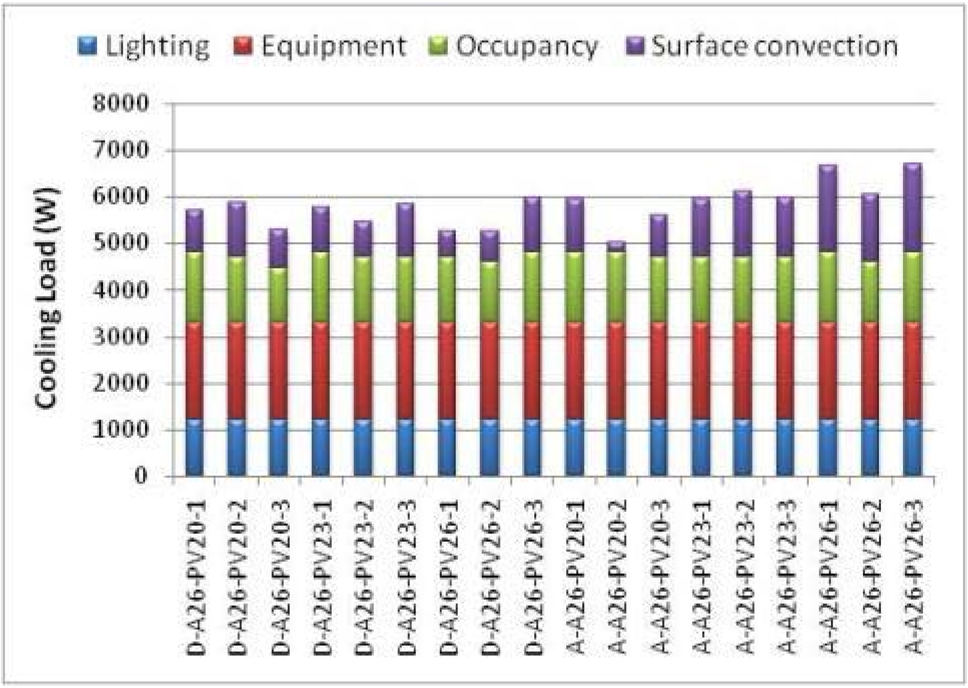


Figure 4-39 Room cooling load components of FEC at room temperature 26 ℃

Figures 4-40 and 4-41 show the ventilation cooling load and load from fans and heater at room temperature 23℃ and 26℃. The ventilation cooling loads depend on the airflow rate, outdoor air condition and room air condition. When room ambient temperature is 23℃, the sum of ventilation cooling load and loads from fans and heater vary from 7.6 to 12.0kW with an average value of 9.4kW. When room ambient temperature is 26℃, the values vary from 6.0 to 12.0kW with an average value of 9.2kW.

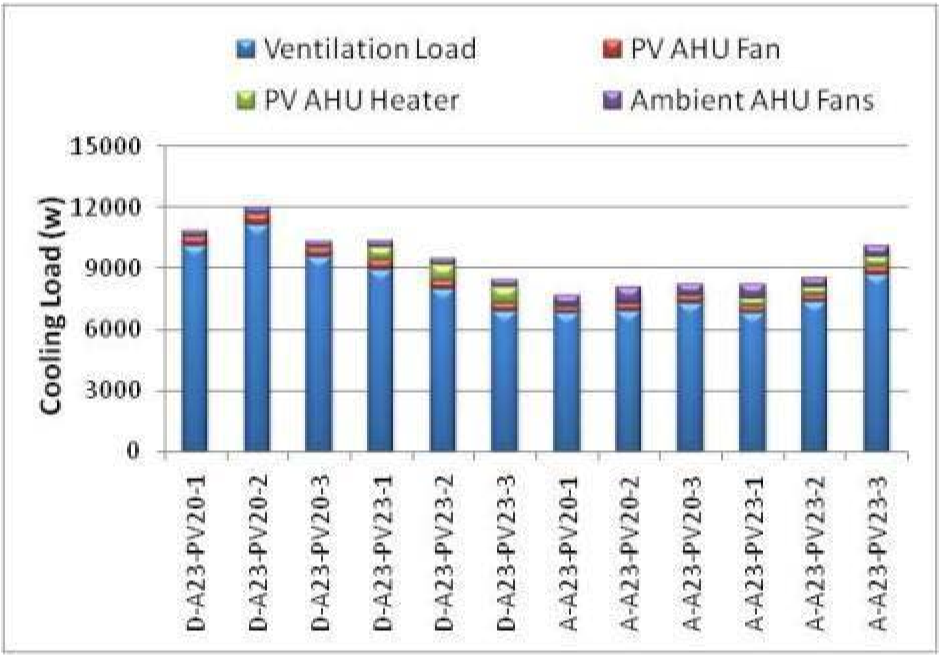


Figure 4-40 Ventilation cooling load components at room temperature 23 ℃

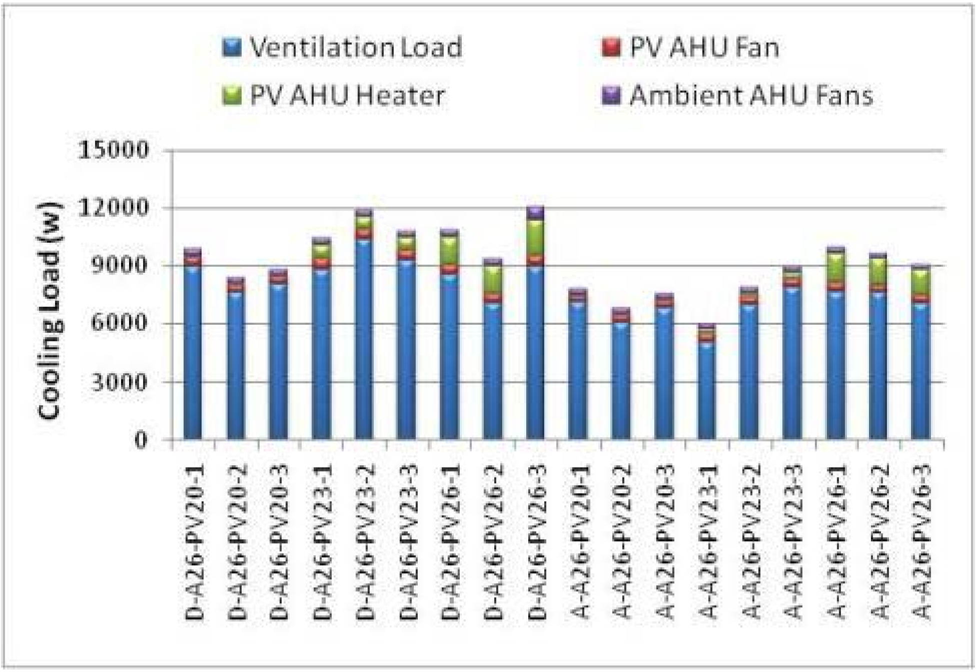


Figure 4-41 Ventilation cooling load components at room temperature 26 ℃

Figures 4-42 and 4-43 show the total cooling load of the FEC at room temperature 23℃ and 26℃. When room ambient temperature is 23℃, the total cooling loads vary from 15.3kW to 18.8kW and have an average value of 16.6kW. When room ambient temperature is 26℃, the total cooling loads vary from 11.8kW to 18.0kW and have an average value of 15.0kW.

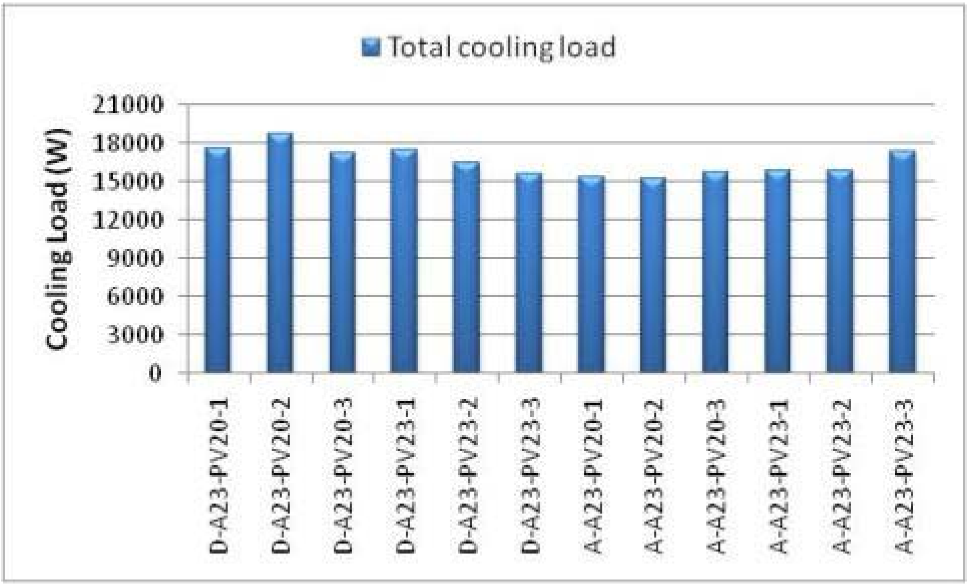


Figure 4-42 Total cooling load at room temperature 23 ℃

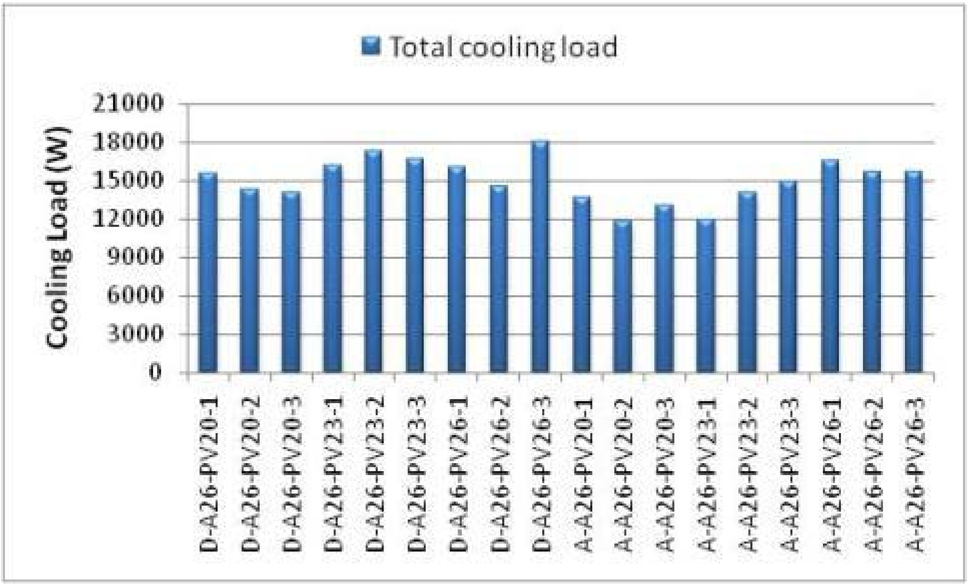


Figure 4-43 Total cooling load at room temperature 26 ℃

- **Cooling capacity of PV AHU and ambient AHU**

Figures 4-44 and 4-45 show the water-side cooling capacity of PV AHU and ambient AHU at room temperature 23℃ and 26℃ respectively. When room ambient temperature is 23℃, the total cooling capacities of PV AHU and ambient AHU vary from 15.8 to 19.4kW with an average value of 18.0kW. When room ambient temperature is 26℃, the total cooling capacities of PV AHU and ambient AHU vary from 12.3 to 19.2kW with an average value of 15.5kW.

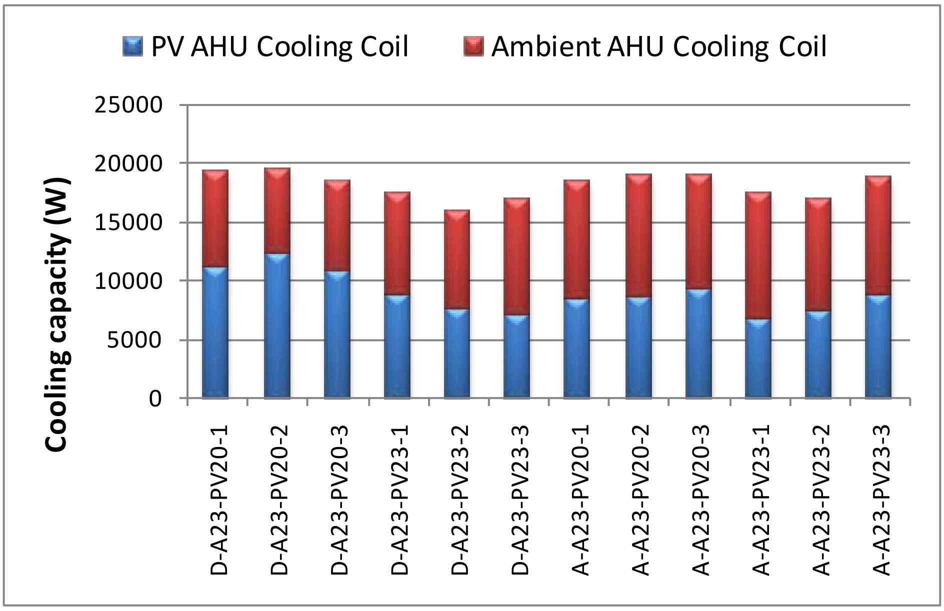


Figure 4-44 Cooling capacity of AHUs at room temperature 23 ℃

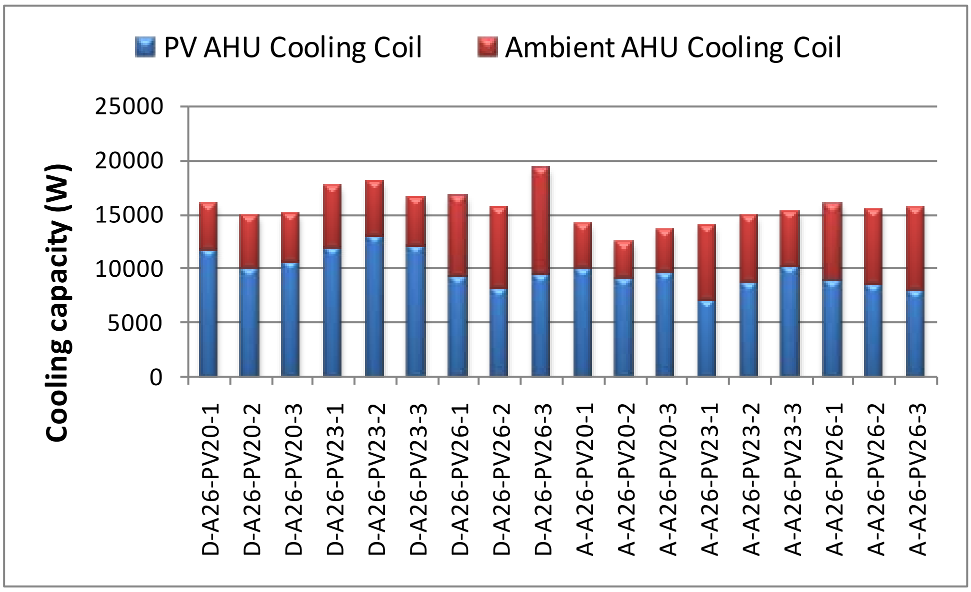


Figure 4-45 Cooling capacity of AHUs at room temperature 26℃

- **Comparison of simulation results with measurement results**

Figures 4-46 and 4-48 show the cooling load comparison at room temperature 23℃ and 26℃. Figures 4-47 and 4-49 show the difference between the simulation and measurement results. The difference is calculated by Equation 4-1.

$$Diff = \frac{Q_{Simulation} - Q_{Measurement}}{Q_{Measurement}} \times 100\% \quad \text{.............................Equation 4-1}$$

Where:

- $Q_{Simulation}$ is the simulated cooling load
- $Q_{Measurement}$ is the measured cooling load

When room ambient temperature is 23℃, the differences between simulation and measurement results vary from -19.4% to 4.3% with an average value of -8.0%. When room ambient temperature is 26℃, the differences between simulation and measurement results vary from -14.3% to 4.7% with an average value of -3.4%.

In A-A23-PV20-1, A-A23-PV20-2 and A-A23-PV20-3 sessions, the differences between simulation results and measurement result are quite large. The reason may be that during these sessions, the chilled water return temperature is quite high and the water flow rate of the cooling coil is quite low, which is about 0.2~0.4 L/s. As the minimum unit (precision) of the water flow rate sensors is 0.1 L/s, these low water flow rates may lead to about 12.5~25.0% of error.

The comparison indicates that the measured cooling loads are slightly higher than the simulated cooling loads. In this study, the transportation loss is not considered. This energy model is used to evaluate the energy performance of the PV-ICA system.

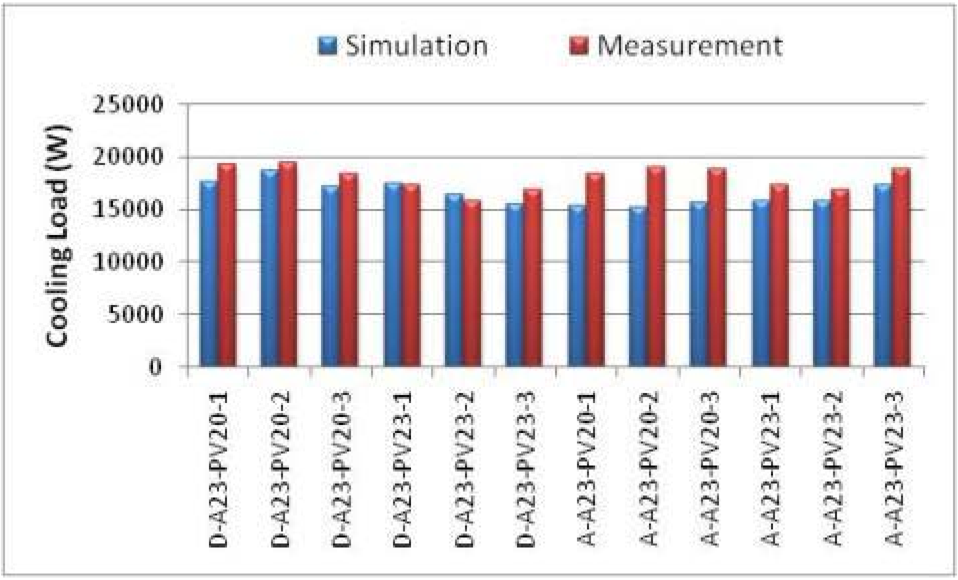


Figure 4-46 Cooling load comparison at room temperature 23 ℃

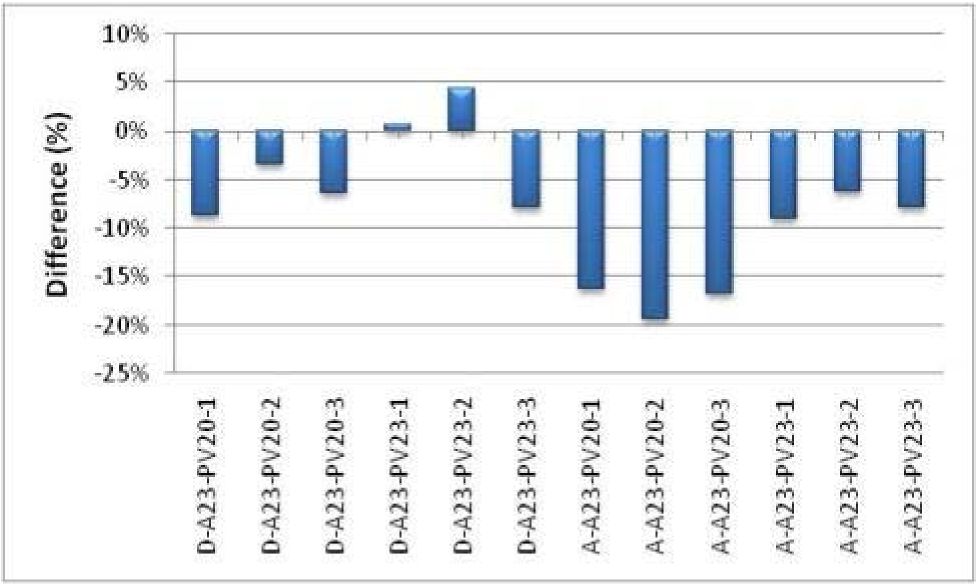


Figure 4-47 Difference between simulation and measurement results at room temperature 23 ℃

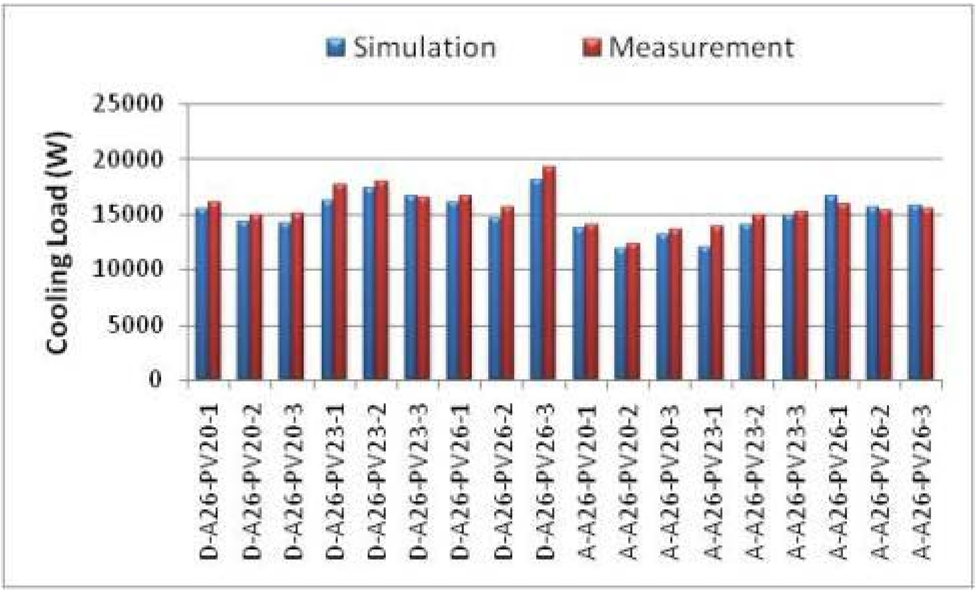


Figure 4-48 Cooling load comparison at room temperature 26℃

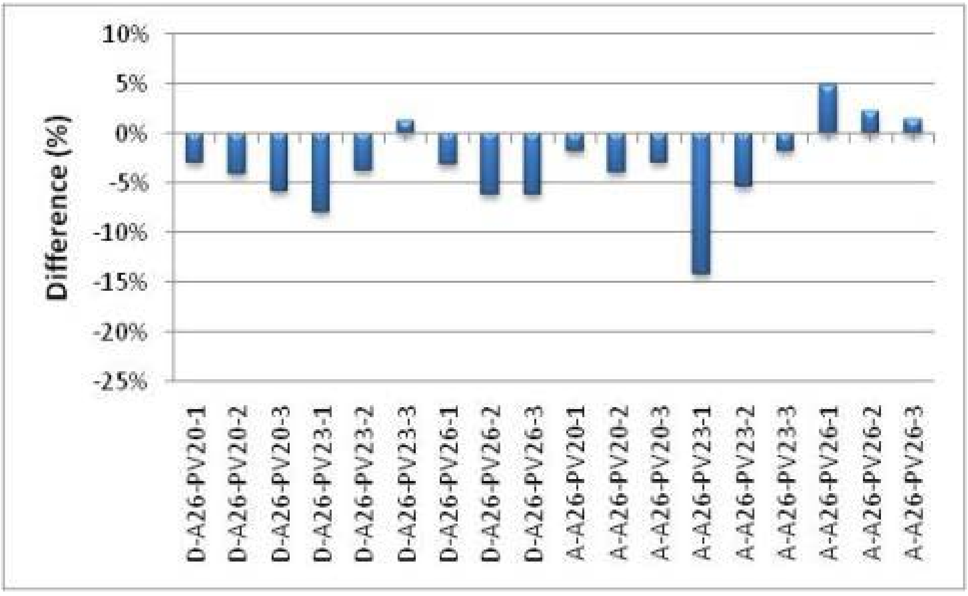


Figure 4-49 Difference between simulation and measurement results at room temperature 26℃

4.3.2 Energy savings

➢ **Total cooling load**

Figure 4-50 shows the FEC cooling load simulation results of DOSP control cases. The occupancy, lighting and equipment load are the same for all conditions. The surface convection at room ambient temperature of 26℃ is 3.4 GJ (14%) smaller than that at room ambient temperature of 23℃. Figure 4-51 shows the ventilation load. The ventilation loads do not vary too much because when room ambient temperature is 26℃, the subjects' preferred airflow rate is higher than that at room ambient temperature of 23℃.

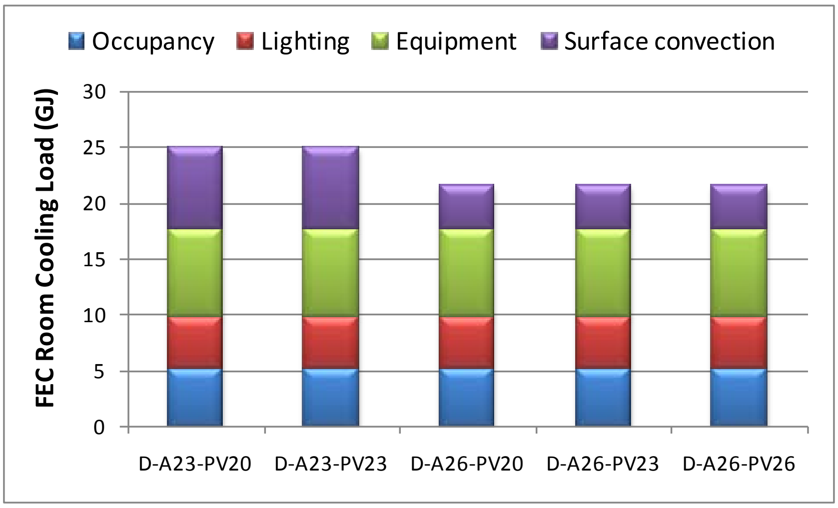


Figure 4-50 FEC room cooling load simulation results of DOSP control cases

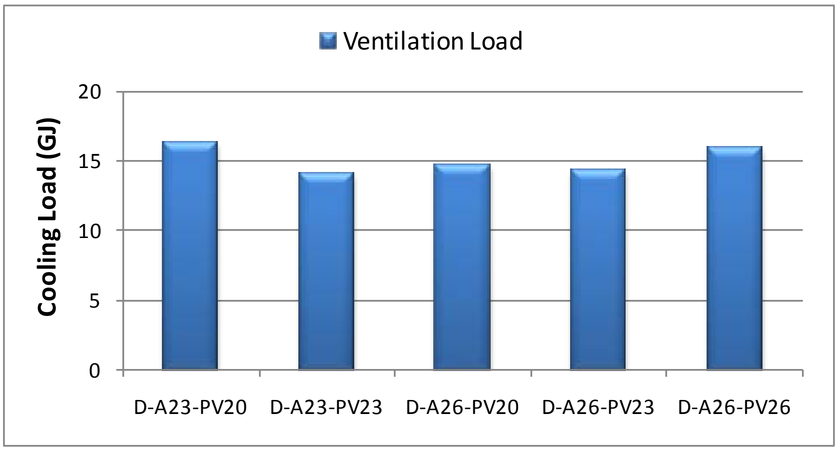


Figure 4-51Ventilation cooling load simulation results of DOSP control cases

Figures 4-52 and 4-53 show the total cooling load simulation results of DOSP control. The total cooling load of D-A23-PV20 is 13.8% higher than that of D-A26-PV20. The total cooling load of D-A23-PV23 is 8.9% higher than that of D-A26-23. The PV supply air temperature does not significantly affect the total cooling load.

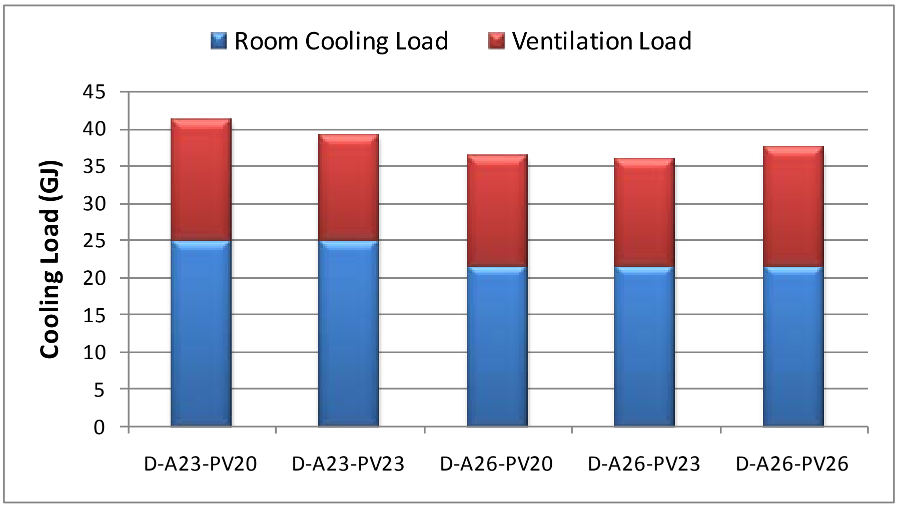


Figure 4-52 Total Cooling load simulation results of DOSP control cases

- **Energy performance of the PV-ICA system**

In this study, the performance of the chiller plant is not studied. The TCOP is set to 4 for all experimental conditions. The energy consumption of the PV AHU fan and the PV AHU heater are included in this section. The energy consumption of the HVAC system is shown in Figure 4-53. It is clearly shown that the energy consumption of the heater increases when PV supply air temperature increases in the PV-ICA system. As there is a heat pipe used in the PV AHU, the heater is not required when PV supply air temperature is 20℃. Among the 5 conditions, the energy consumption of D-A26-PV20 is the lowest. Under the same PV supply air temperature at 20℃, the energy consumption at room temperature of 23℃ is 11% higher than that at room temperature of 26℃. However, under the same PV supply air temperature at 23℃, the energy consumptions at room temperature of 23℃ and 26℃ are almost the same. The result indicates that in view of energy saving potential, the best case is that PV-ATD supply personalized air at 20℃ and the room is maintained at 26℃.Heater consumed about 10% of the total energy at room temperature of 23℃ and more than 20% at room temperature of 26℃. A new design of PV AHU should be developed to be able to supply PV air at high temperature (such as 26℃) without the need for reheating.

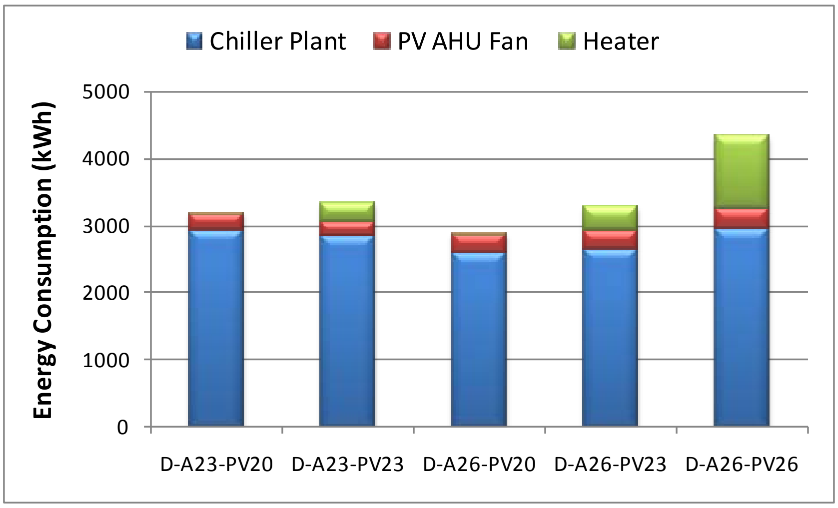


Figure 4-53 HVAC energy simulation results of DOSP control

4.3.3 Energy saving potential of the PV-ICA system

As discussed in Section 4.1, there may be several reasons why subjects increase the air movement when the room ambient temperature increases. Subjects may just need to increase the air movement to enhance the thermal comfort. It may, therefore, not be necessary to supply 100% fresh air in view of energy conservation. As shown in Table 4-2, the minimum PV airflow rate is 6.92 L/s/person found in D-A23-PV23. The amended personalized airflow rates are shown in Table 4-2. The fresh air flow rates are set to 7L/s/person for all cases and additional recirculated air is applied in order to get the same airflow rates as the original ones.

Table 4-2 Amended personalized airflow rates

Case Ref.	Original fresh airflow rate (L/s/person)	Amended airflow rate(L/s/person)	
		Fresh air	Recirculated air
D-A23-PV20	8.00	7.00	1.00
D-A23-PV23	6.92	7.00	0
D-A26-PV20	9.34	7.00	2.34
D-A26-PV23	9.08	7.00	2.08
D-A26-PV26	10.11	7.00	3.11

The ventilation loads of the amended cases are shown in Figure 4-54. The total cooling loads of the amended cases are shown in Figure 4-55. As the ventilation rates are amended to the same level, both the ventilation loads and the total cooling load do not vary with the PV supply air temperature. The ventilation load at room air temperature of 23℃ is 29.0% higher than that at room air temperature of 26℃.The total cooling load at room air temperature of 23℃ is 20.4% higher than that at room air temperature of 26℃.

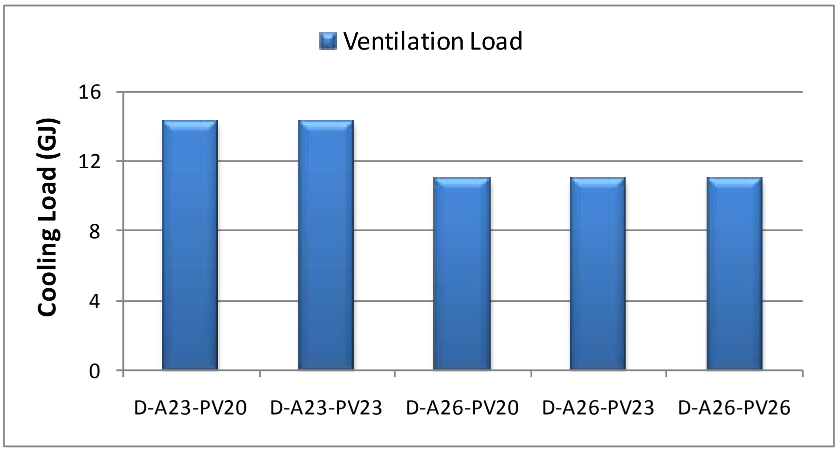


Figure 4-54 Ventilation cooling load of amended cases

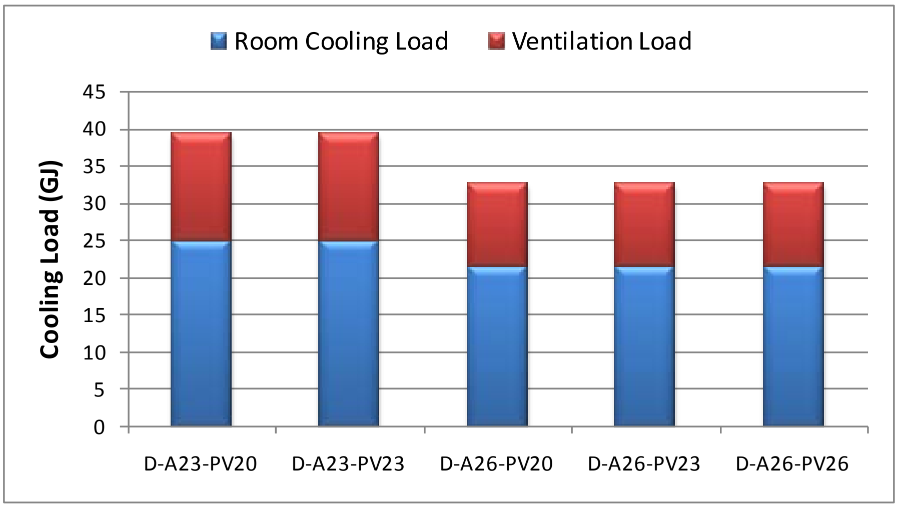


Figure 4-55 Total cooling load of amended cases

The energy consumption of the PV-ICA system of the amended cases is shown in Figure 4-56. Under the same PV supply air temperature of 20℃, the energy consumption of the PV-ICA system at room air temperature of 23℃ is 15.7% higher than that at room air temperature of 26℃.Under the same PV supply air temperature of 23℃, the energy consumption of the PV-ICA system at room air temperature of 23℃ is 13.1% higher than that at room air temperature of 26℃.

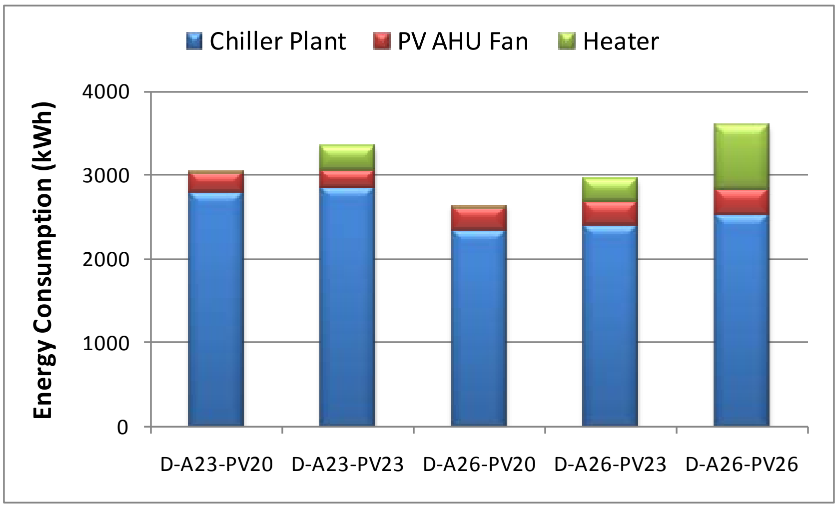


Figure 4-56 HVAC energy consumption of amended cases

4.3.4 Summary of energy performance of the PV-ICA system

Table 4-3 summarizes the results of all the simulated cases. The results show that the best case is when the PV-ATD supplies personalized air at 20℃ and the room is maintained at 26℃.

Table 4-3 Summary of energy performance of the PV-ICA system

Simulation Cases	Case Ref.	Chiller Plant (kWh)	PV AHU Fan (kWh)	Heater (kWh)	Total Energy (kWh)
100% fresh air	D-A23-PV20	2925	250	5	3180
	D-A23-PV23	2833	222	275	3330
	D-A26-PV20	2589	289	5	2883
	D-A26-PV23	2647	281	358	3286
	D-A26-PV26	2949	312	1086	4348
Amended fresh air	D-A23-PV20	2784	250	4	3038
	D-A23-PV23	2845	222	278	3345
	D-A26-PV20	2333	289	4	2626
	D-A26-PV23	2400	281	278	2959
	D-A26-PV26	2528	312	760	3600

Chapter 5: Conclusions

5.1 Achievement of research objectives

The thermal comfort and energy performance of the PV-ICA system is studied through both subjective assessment and objective experiments. The objective of the research and the results obtained are summarized as follows:

- **First objective**

The first objective is to study aspects of subjects' behavior such as the frequency of adjusting PV airflow rate and their preferred airflow rate.

The results of subjects' control behavior are shown as follows:

(1) The average number of times subjects changed the airflow in all experimental conditions is 5.4, with 3.6 times for air flow decrease case and 1.8 times for air flow increase case. More than 50% of operations were performed during the first 30 minutes.

(2) In DOSP control, subjects' preferred airflow rates at room ambient temperature of 26℃ are 17~31% higher compared to the room ambient temperature at 23℃.

(3) In DOSP control, when the room ambient temperature is 23℃, subjects' preferred airflow rate clearly reduced during the first one hour and then reduced moderately during the last hour. When the room ambient

temperature is 26℃, subjects' preferred airflow rates reduced slightly during the first one hour and then seem to keep relatively steady.

(4) Subject' preferred airflow rates are quite different, which range from 0.2 to 14.7 L/s/person.

The clear patterns that have been observed in the operations of controls show that subjects need to adjust the air flow with time. They need more air flow at the beginning and after exposing themselves in the cool environment, their requirement for airflow reduces. It is clear that supplying a constant steady air flow cannot meet most of the subjects' requirement of air movement

- **Second objective**

The second objective is to evaluate human response to the PV-ICA system, with regard to local and whole body thermal sensation and its acceptability; thermal comfort; air movement perception, acceptability and preference; etc.

The results of thermal comfort of the PV-ICA system are shown as follows:

(1) More than 90% of subjects feel the thermal environment is acceptable while more than 83% of subjects felt that the air movement on face, shoulder and neck were all acceptable. The acceptability of thermal comfort and air movement does not vary significantly with the room

ambient temperature, PV temperature or control strategy.

(2) At room ambient temperature of 23℃, the subjects' whole body thermal sensation and thermal sensations of hand and face are lower than slightly cool (-1), which may cause draft.

(3) At higher room ambient temperature of 26℃, the subjects' whole body and body segments thermal sensations are closer to neutral as compared to that at room ambient temperature of 23℃, which may reduce the draft risks.

(4) About 80% of subjects feel air movement on their face and 30% of the subjects prefer less air movement on face. About 40% of subjects feel air movement on their head and shoulder.

The vast differences in the preferred air flow and thermal sensation reiterate the need for individual control in PV systems

- **Third objective**

The third objective is to evaluate the energy saving potential of the PV-ICA system.

The results of energy performance of the PV-ICA system are shown as follows:

(1) The FEC room cooling load is about 6.4 kW (76 W/m^2). The ventilation load is about 7.9kW (95 W/m^2). The total cooling load is about 15.6kW (187 W/m^2).

(2) Except for 4 sessions (A-A23-PV20-1, A-A23-PV20-2, A-A23-PV20-3 and A-A26-PV23-1), the differences between simulation results and measurements are less than 9% with an average value of 3.5%.

(3) When room ambient temperature is 26℃, the FEC room cooling load is 14% lower than that at room ambient temperature of 23℃.The ventilation loads does not vary too much. When room ambient temperature is 23℃, the total cooling load are 9~14% higher than that at room ambient temperature of 26℃. Under the same PV air temperature at 20℃, the energy consumption of the PV-ICA system at room air temperature of 23℃ is 11% higher than that at room air temperature of 26℃.

(4) When the fresh air flow rate are amended to 7L/s/person for all cases, the ventilation load at room temperature of 23% is 29% higher than that at room ambient temperature of 26℃. When room ambient temperature is 23℃, the total cooling load of amended cases is 20.4% higher than that at room ambient temperature of 26℃. The energy consumption of the PV-ICA system at room air temperature of 23℃ is 13~16% higher than that at room air temperature of 26℃.

(5) Heater consumed about 10% of the total energy at room temperature of 23℃ and more than 20% at room temperature of 26℃.

In conclusion, it is found that in the tropics, the PV-ICA system can better improve occupants' thermal comfort and enhance energy saving at a room ambient temperature of 26℃ than at 23℃. The best case is that PV-ATD supply personalized air at 20℃ and the room is maintained at 26℃.

5.2 Limitations

The first limitation was the fact that the subjects involved in this study were undergraduates with an average age of 22 (ranging from 20 to 25). However, the occupants in a real office may have a wider range of age and they might have different preferences when using the PV-ICA system. The results would be a better representation of an office environment if more people from different backgrounds and ages had participated in the subjective study.

The second limitation is the damper devices. The damper characteristics are non-linear, which bring considerable difficulty in the control. Dampers with linear characteristics might have improved the results, especially in AFSP control mode.

The third limitation is the time delay of the air flow sensors. The time delay of the air flow rate sensors significantly affects the AFSP control performance.

5.3 Recommendations for future work

With the conclusions drawn from this study, future research can be pursued in the following directions:

1) As discussed in Section 4.1, there may be several reasons why subjects' preferred airflow rates increase with the ambient room temperature. Further studies should be conducted to identify the primary reasons. A new PV ATD, which can achieve higher air movement, should be design to identify whether increasing air movement is the primary reason why subjects increase the airflow rate.
2) The individual control performance and energy performance of PV system in conjunction with DV or under-floor air distribution system should be evaluated.
3) The temperature distribution of the room with PV-ICA system should be studied using CFD modeling.
4) Heater consumed about 10% of the total energy at room temperature of 23℃ and more than 20% at room temperature of 26℃. A new design of PV AHU should be developed in order to supply PV air at higher temperatures (such as 26℃) without the need for reheating.

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Appendix A: Questionnaires

Questionnaire to Assess the Performance of

Personalized Ventilation Control

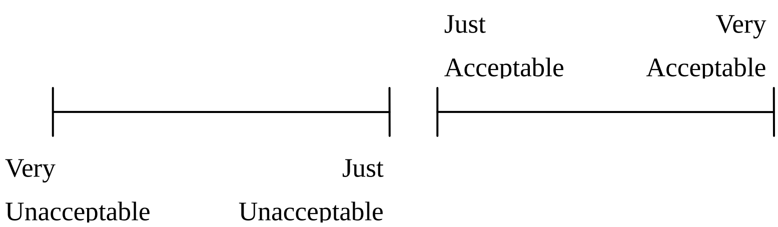
The purpose of this survey is to collect your feedback on the performance of the personalized ventilation system. For details, please refer to the participant information sheet and the consent form which were distributed to you earlier.

Session Number: **Date**: **Time**:

Your Name:

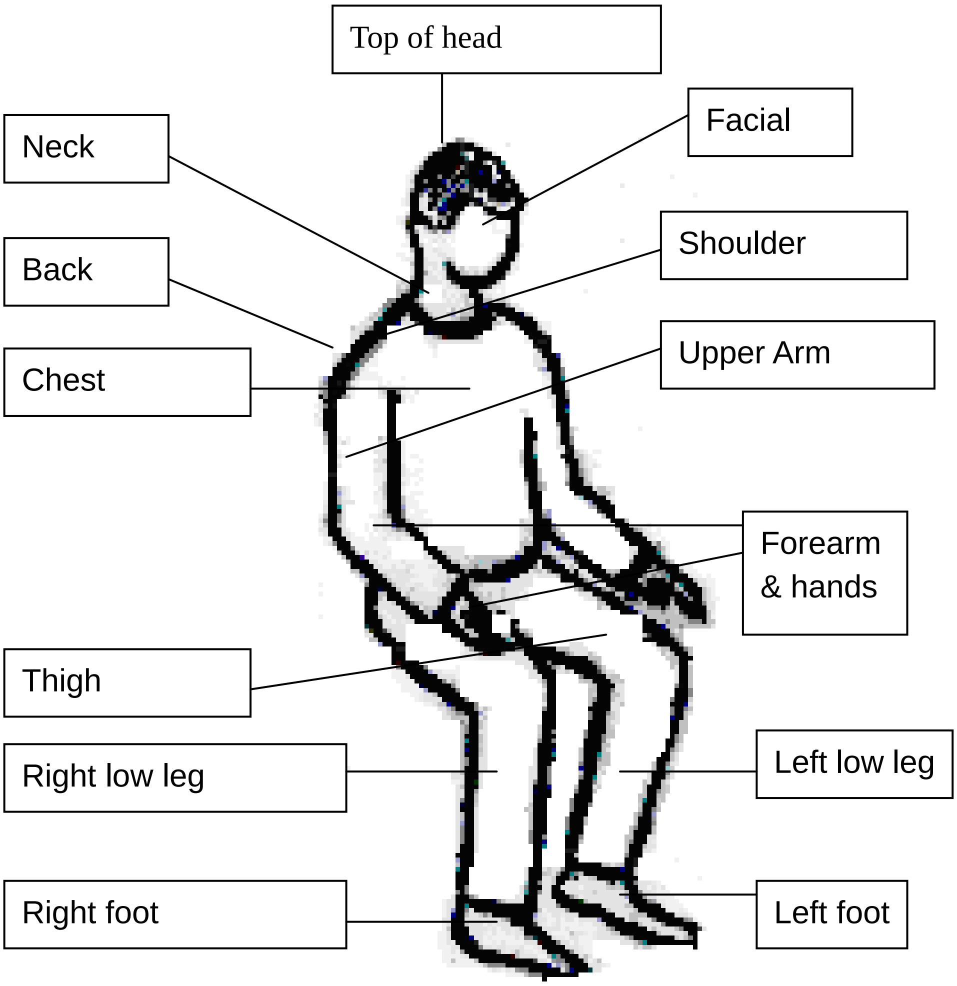
Part 1. Thermal Sensation and air quality

1. Please assess the acceptability of your thermal comfort condition in the following continuous scale. Please do NOT mark between “Just Unacceptable” and “Just acceptable”:



2. Please select a number in each of the 13 boxes in the diagram below to indicate the thermal sensation of each part of your body. The 7-value numerical scale to be used appears in the table below:

+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold



Please provide your assessment of thermal sensation for your whole body:

3. Air Movement Perception

Do you feel air movement on your head, face or shoulder: yes no

If “yes” please indicate your perception of air movement according to the 7-value numerical scale shown in the table below:

+3	much too breezy
+2	too breezy
+1	slightly breezy
0	just right
-1	slightly still
-2	too still
-3	much too still

4. Please assess the acceptability of the air movement in the following continuous scale. Please do NOT mark between “Just Unacceptable” and “Just acceptable”:

a1. Please assess the acceptability of the air movement on the top of head

Just Acceptable   Very Acceptable

Very Unacceptable   Just Unacceptable

a2. Please indicate the change in the air movement preferred on the top of head?

☐ Less air movement   ☐ No change   ☐ More air movement b1. Please assess the acceptability of the air movement on the facial part

Just Acceptable — Very Acceptable

Very Unacceptable — Just Unacceptable

b2. Please indicate the change in the air movement preferred on the facial part?

☐ Less air movement   ☐ No change   ☐ More air movement

c1. Please assess the acceptability of the air movement on neck

Just Acceptable — Very Acceptable

Very Unacceptable — Just Unacceptable

c2. Please indicate the change in the air movement preferred on neck?

☐ Less air movement   ☐ No change   ☐ More air movement

5. Please assess the inhaled air quality (do not mark between “Just Unacceptable” and “Just acceptable”):

Just Acceptable | Very Acceptable

Very Unacceptable | Just Unacceptable

6. Please assess the inhaled air temperature:

Cold | Hot

7. Please assess the inhaled air humidity:

Humid | Dry

8. Please assess the odour of inhaled air:

No Odour | Overwhelming Odour

9. Please assess the freshness of inhaled air:

Air Stuffy | Air Fresh

10. Please indicate in the following continuous scale if you notice any of the following symptoms:

Nose Drv	|———————|	Nose Not Drv
Lips Dry	|———————|	Lips Not Dry
Eyes Dry	|———————|	Eyes Not Dry
Server Headache	|———————|	No Headache
Difficult To Think	|———————|	Head Clear
Eyes Aching	|———————|	Eyes Not Aching
Dizzy	|———————|	Not Dizzy
Tired	|———————|	Not Tired
Sleepy	|———————|	Not Sleepy
Feeling Bad	|———————|	Feeling Good

Part 2: Environmental and other factors

1. Evaluation of noise level:

Dissatisfied |——————————————————| Satisfied

2. Evaluation of lighting level:

Dissatisfied |——————————————————| Satisfied

3. Please tick which of the following dress you are wearing:

Shirts/Blouses	☐ short sleeves ☐ Long sleeves light weight shirt	☐ Long sleeves normal shirt ☐ Long sleeves flannel shirt	☐ Tube top ☐ Long sleeves light weight blouse	☐ Long sleeves turtleneck blouse

Trousers	☐ shorts ☐ exercise shorts	☐ Light weight trousers ☐ Normal trousers	☐ Bermudas ☐ Overalls

Skirts, Dresses	☐ Light skirt, 15cm above knees ☐ Knee-length heavy skirt	☐ Light skirt, 15cm below knee ☐ Long sleeves winter dress	☐ Sleeves light dress

Jacket	☐ Vest	☐ Light summer jacket ☐ Jacket	☐ High insulative fibre-pelt vest ☐ High insulative fibre-pelt jacket

Socks	☐ Socks	☐ Thick ankle socks ☐ Thick long socks	☐ High insulative fibre-pelt vest ☐ High insulative fibre-pelt jacket

Shoes	☐ Sandals, mules	☐ Thin soled shoes ☐ Thick soled shoes	☐ Ankle boot ☐ Boots

If you wear something that you can't find proper description from above, please write here.

Part 3: Control of personalized ventilation

1. How easy is it to control the personalized ventilation air terminal device (PV ATD)?

Very Difficult ├──────────┤ Very Easy

2. How frequently did you have to adjust the PV ATD?

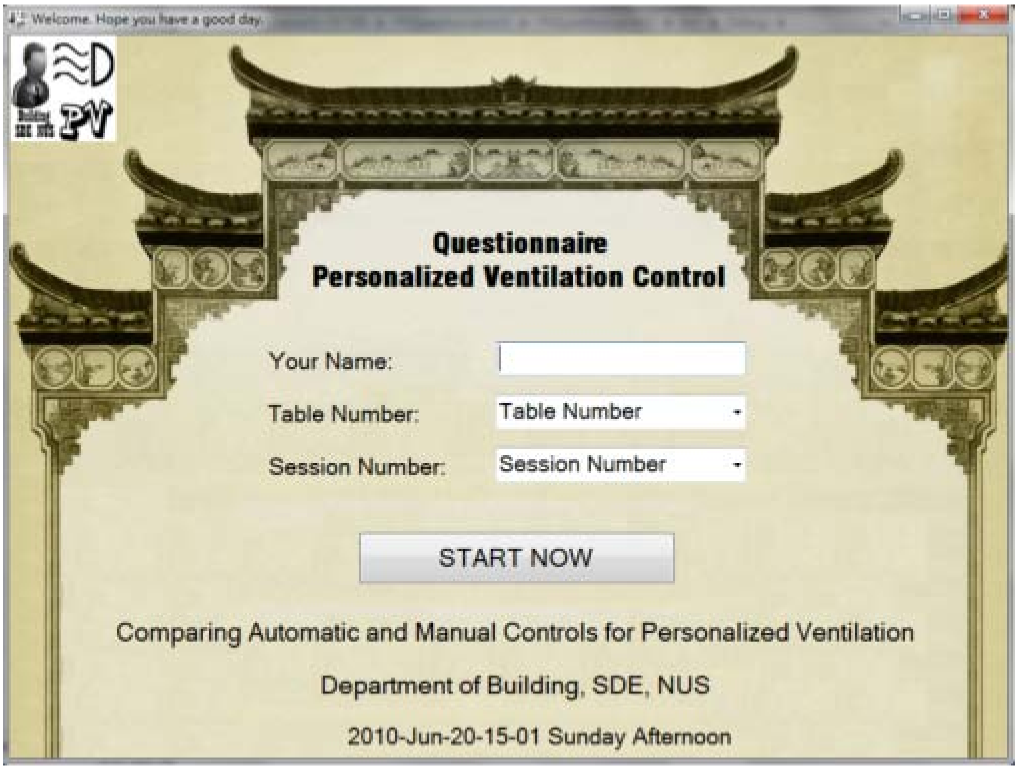
Very Often ├──────────┤ Seldom

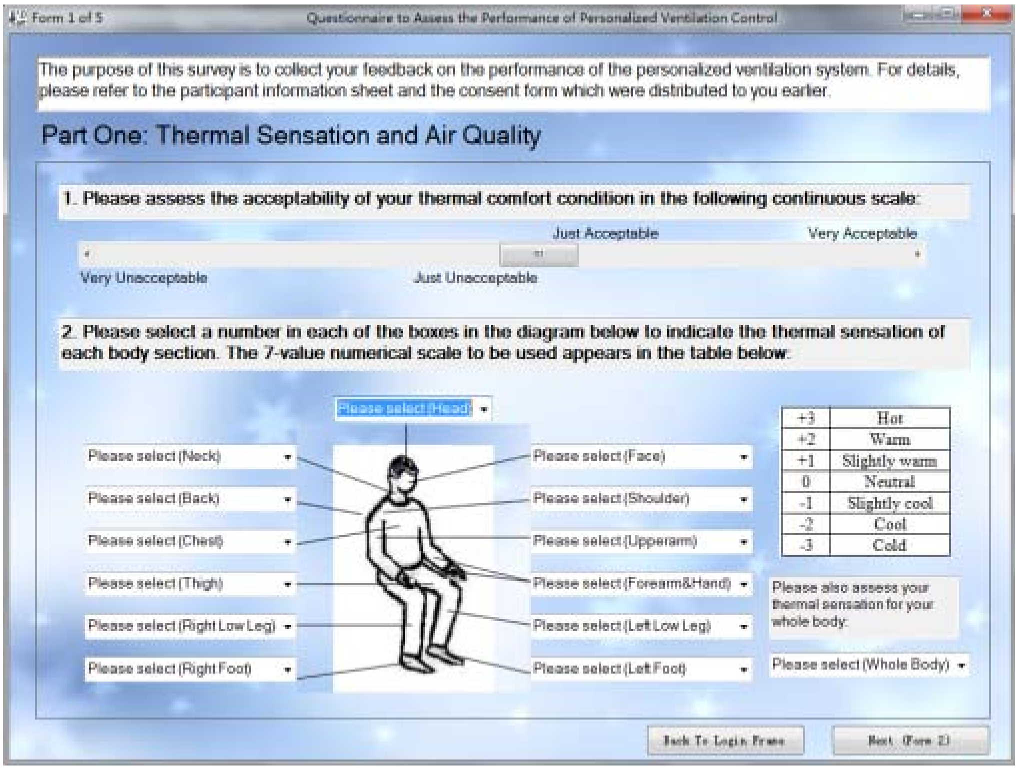
3. How do you rate the performance of the PV ATD?

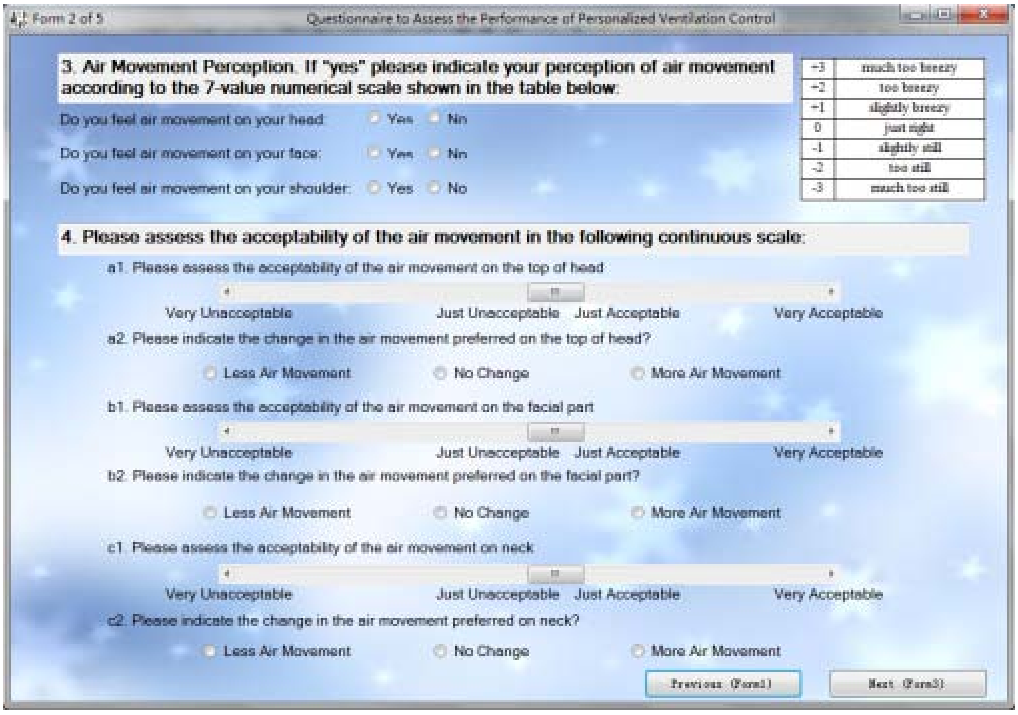
Bad ├──────────┤ Good

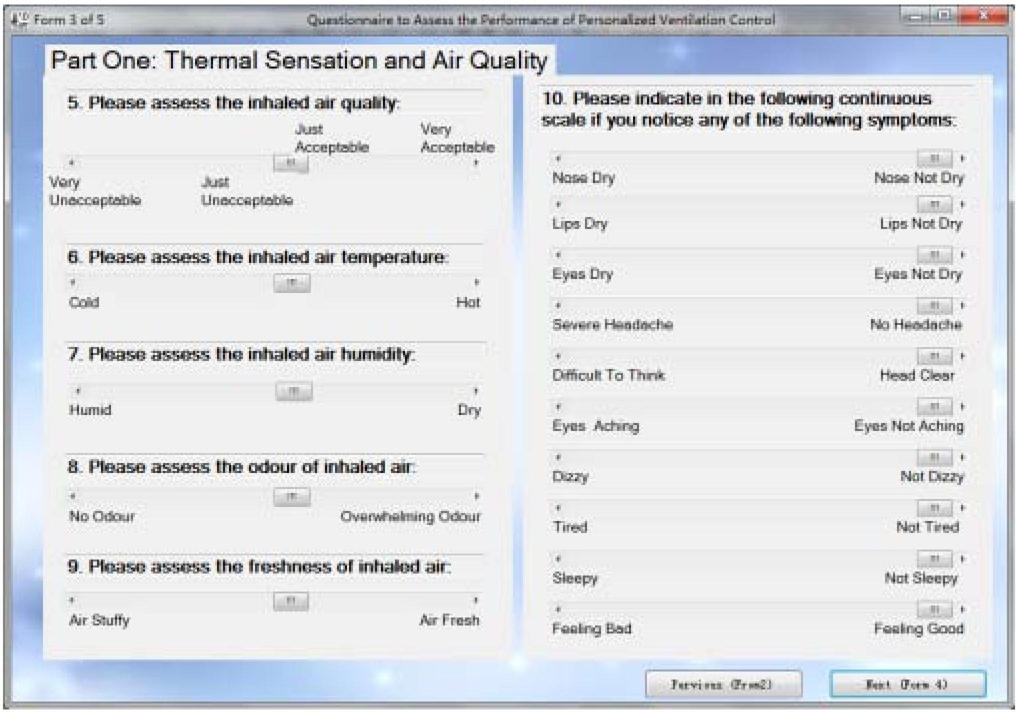
4. Please describe any problems you faced with the PV ATD.

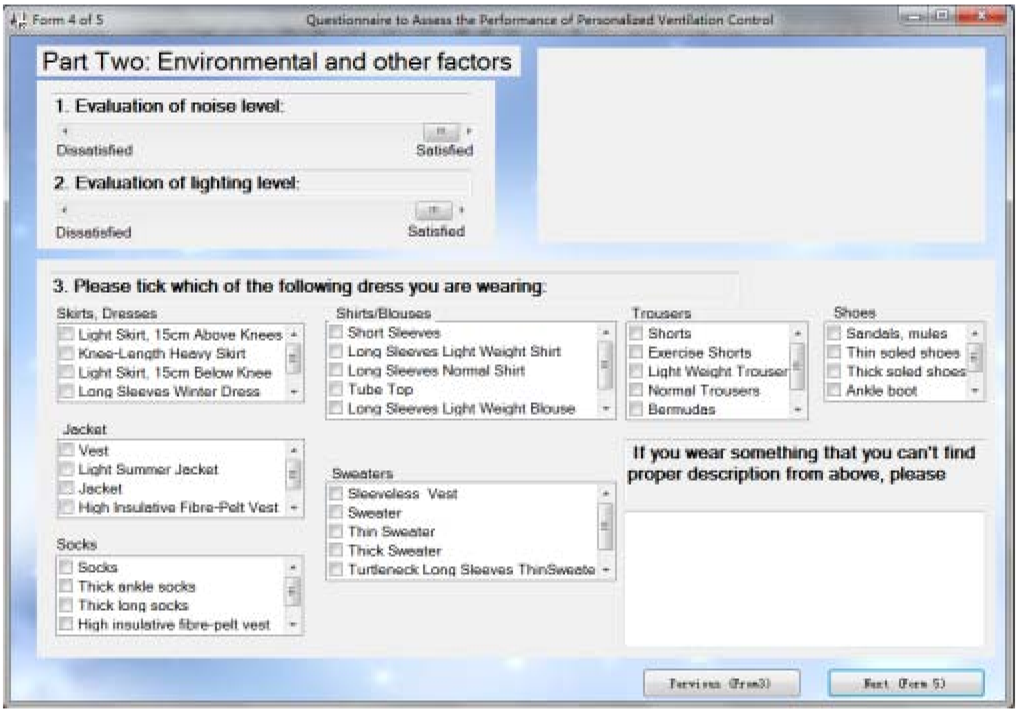
The software interface of the questionnaires

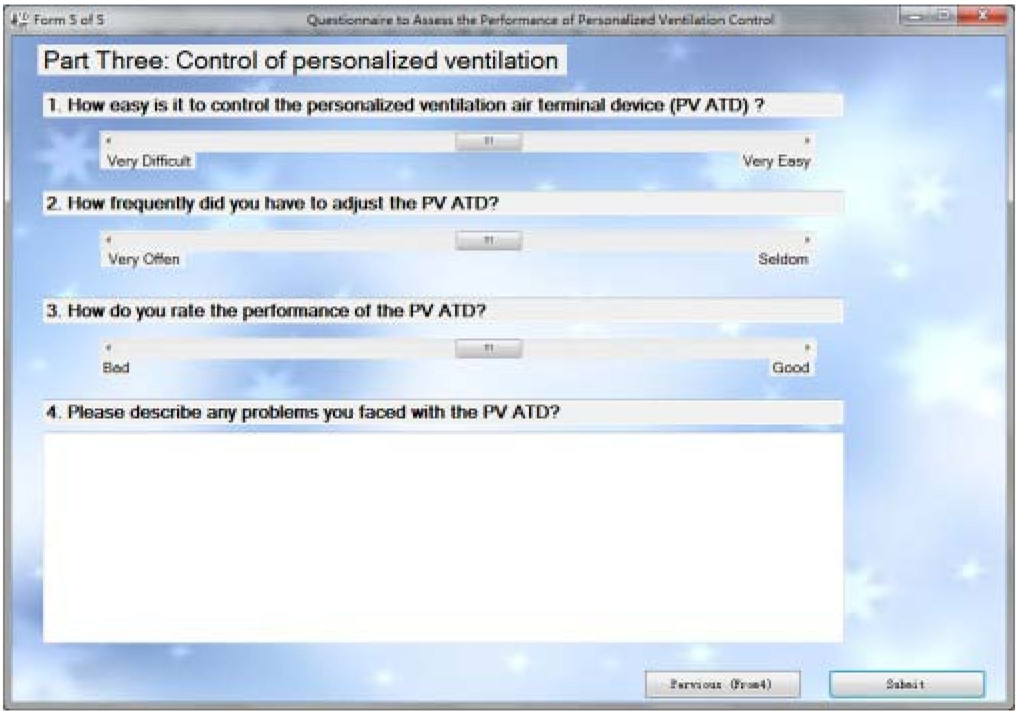





Form 2 of 5
Questionnaire to Assess the Performance of Personalized Ventilation Control
3. Air Movement Perception. If "yes" please indicate your perception of air movement according to the 7-value numerical scale shown in the table below:
Do you feel air movement on your head: Yes No
Do you feel air movement on your face: Yes No
Do you feel air movement on your shoulder: Yes No
+3 much too breezy
+2 too breezy
+1 slightly breezy
0 just right
-1 slightly still
-2 too still
-3 much too still
4. Please assess the acceptability of the air movement in the following continuous scale:
a1. Please assess the acceptability of the air movement on the top of head
Very Unacceptable Just Unacceptable Just Acceptable Very Acceptable
a2. Please indicate the change in the air movement preferred on the top of head?
Less Air Movement No Change More Air Movement
b1. Please assess the acceptability of the air movement on the facial part
Very Unacceptable Just Unacceptable Just Acceptable Very Acceptable
b2. Please indicate the change in the air movement preferred on the facial part?
Less Air Movement No Change More Air Movement
c1. Please assess the acceptability of the air movement on neck
Very Unacceptable Just Unacceptable Just Acceptable Very Acceptable
c2. Please indicate the change in the air movement preferred on neck?
Less Air Movement No Change More Air Movement
Previous (Form1)
Next (Form3)

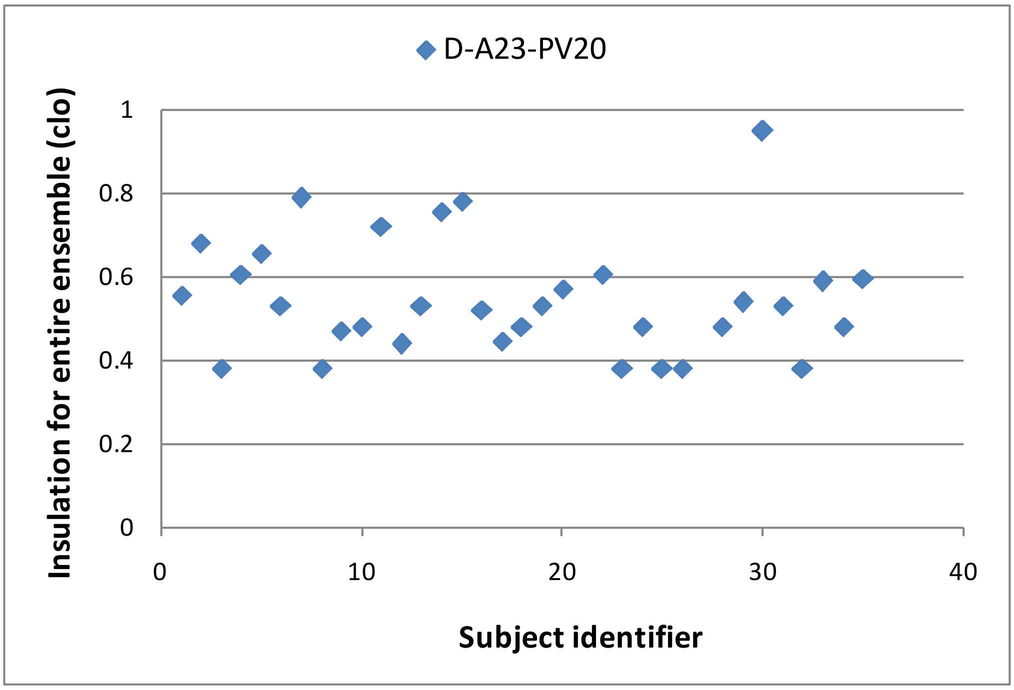

Form 3 of 5
Questionnaire to Assess the Performance of Personalized Ventilation Control
Part One: Thermal Sensation and Air Quality
5. Please assess the inhaled air quality:
Very Unacceptable Just Unacceptable Just Acceptable Very Acceptable
6. Please assess the inhaled air temperature:
Cold Hot
7. Please assess the inhaled air humidity:
Humid Dry
8. Please assess the odour of inhaled air:
No Odour Overwhelming Odour
9. Please assess the freshness of inhaled air:
Air Stuffy Air Fresh
10. Please indicate in the following continuous scale if you notice any of the following symptoms:
Nose Dry Nose Not Dry
Lips Dry Lips Not Dry
Eyes Dry Eyes Not Dry
Severe Headache No Headache
Difficult To Think Head Clear
Eyes Aching Eyes Not Aching
Dizzy Not Dizzy
Tired Not Tired
Sleepy Not Sleepy
Feeling Bad Feeling Good
Previous (Form2)
Next (Form 4)

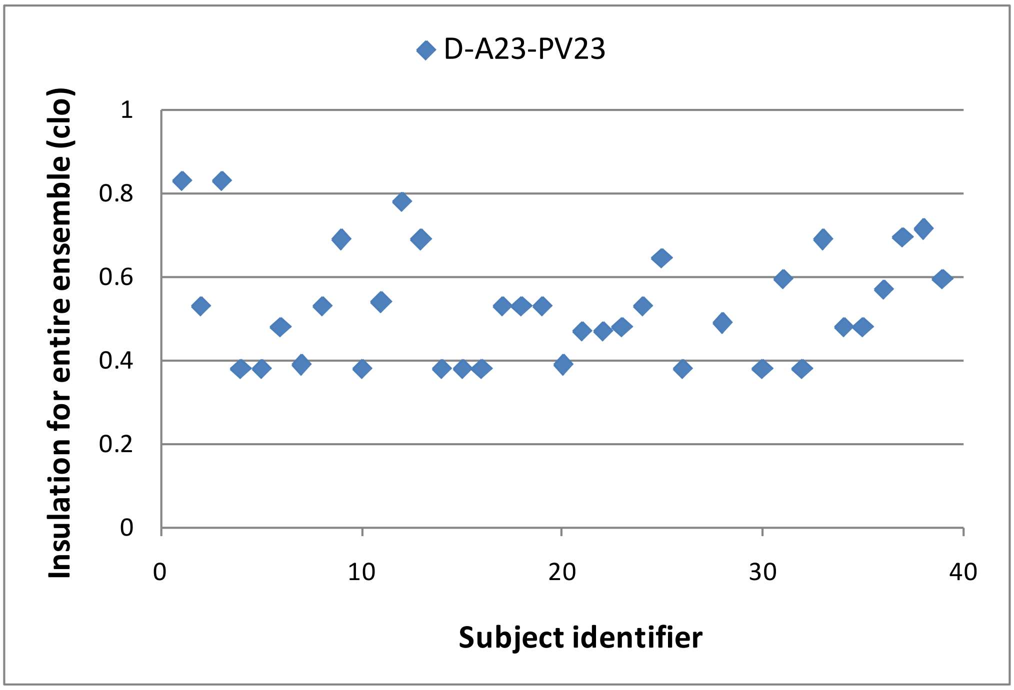

Form 4 of 5
Questionnaire to Assess the Performance of Personalized Ventilation Control
Part Two: Environmental and other factors
1. Evaluation of noise level:
Dissatisfied
Satisfied
2. Evaluation of lighting level:
Dissatisfied
Satisfied
3. Please tick which of the following dress you are wearing:
Skirts, Dresses
Light Skirt, 15cm Above Knees
Knee-Length Heavy Skirt
Light Skirt, 15cm Below Knee
Long Sleeves Winter Dress
Shirts/Blouses
Short Sleeves
Long Sleeves Light Weight Shirt
Long Sleeves Normal Shirt
Tube Top
Long Sleeves Light Weight Blouse
Trousers
Shorts
Exercise Shorts
Light Weight Trouser
Normal Trousers
Bermudas
Shoes
Sandals, mules
Thin soled shoes
Thick soled shoes
Ankle boot
Jacket
Vest
Light Summer Jacket
Jacket
High Insulative Fibre-Pelt Vest
Sweaters
Sleeveless Vest
Sweater
Thin Sweater
Thick Sweater
Turtleneck Long Sleeves ThinSweate
Socks
Socks
Thick ankle socks
Thick long socks
High insulative fibre-pelt vest
If you wear something that you can't find proper description from above, please
Next (Form 5)

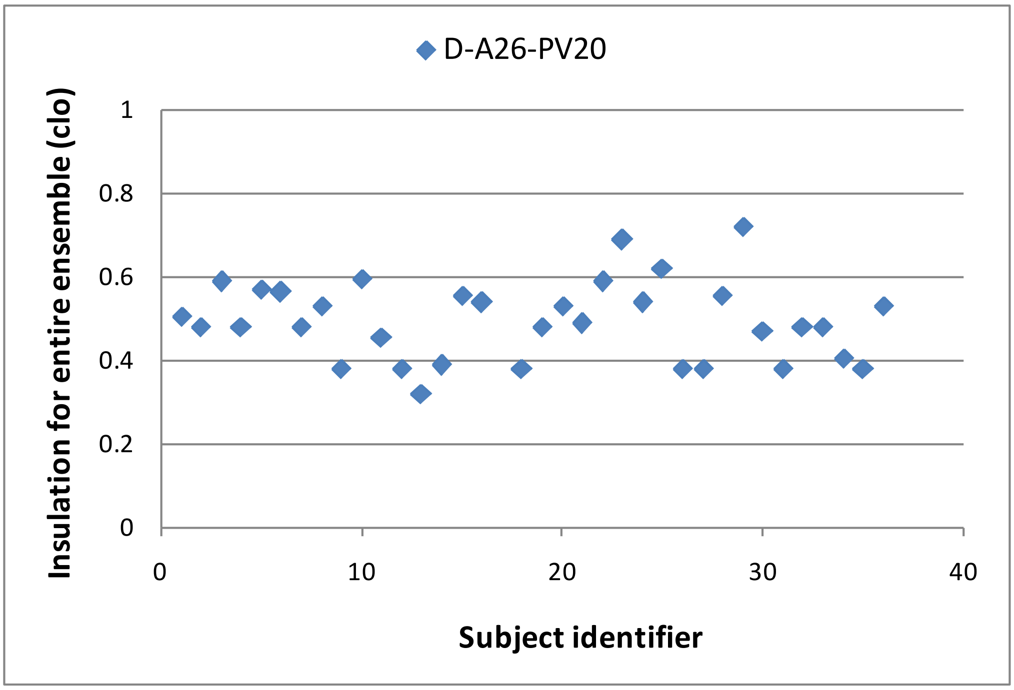

Form 5 of 5
Questionnaire to Assess the Performance of Personalized Ventilation Control
Part Three: Control of personalized ventilation
1. How easy is it to control the personalized ventilation air terminal device (PV ATD) ?
Very Difficult
Very Easy
2. How frequently did you have to adjust the PV ATD?
Very Often
Seldom
3. How do you rate the performance of the PV ATD?
Bad
Good
4. Please describe any problems you faced with the PV ATD?
Submit

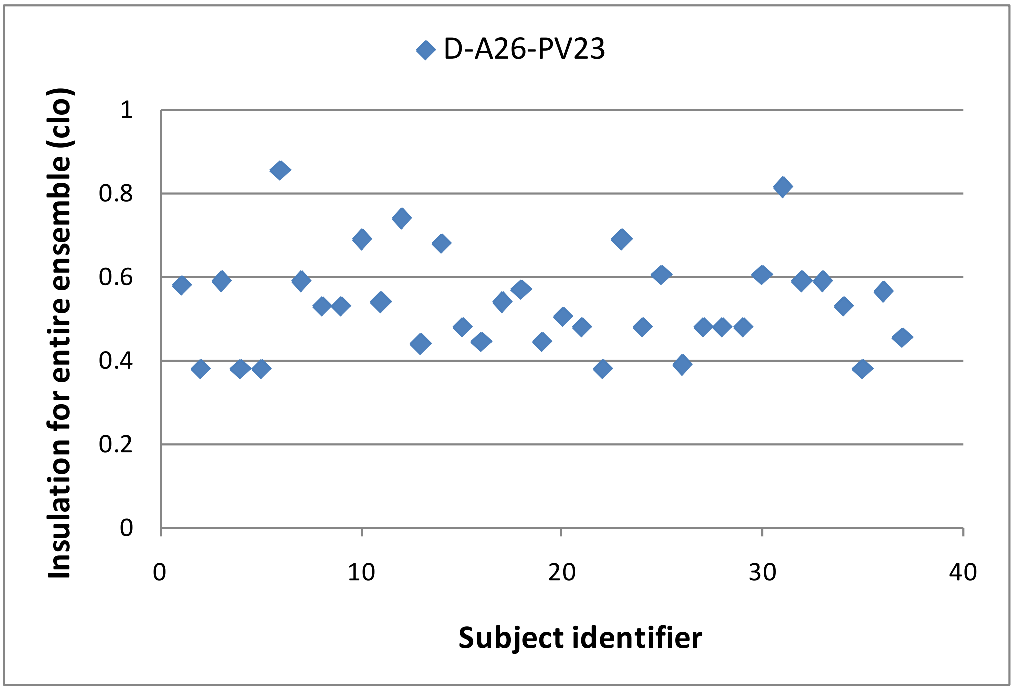
Appendix B: Subjective experiment results

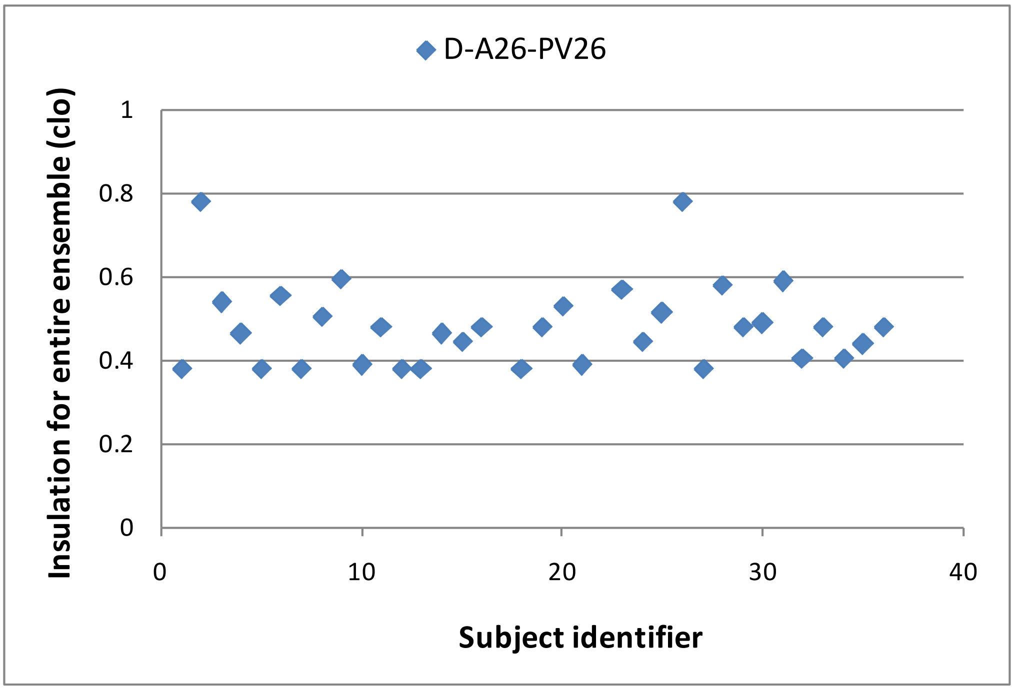
The results of insulation for entire ensemble of all experimental conditions

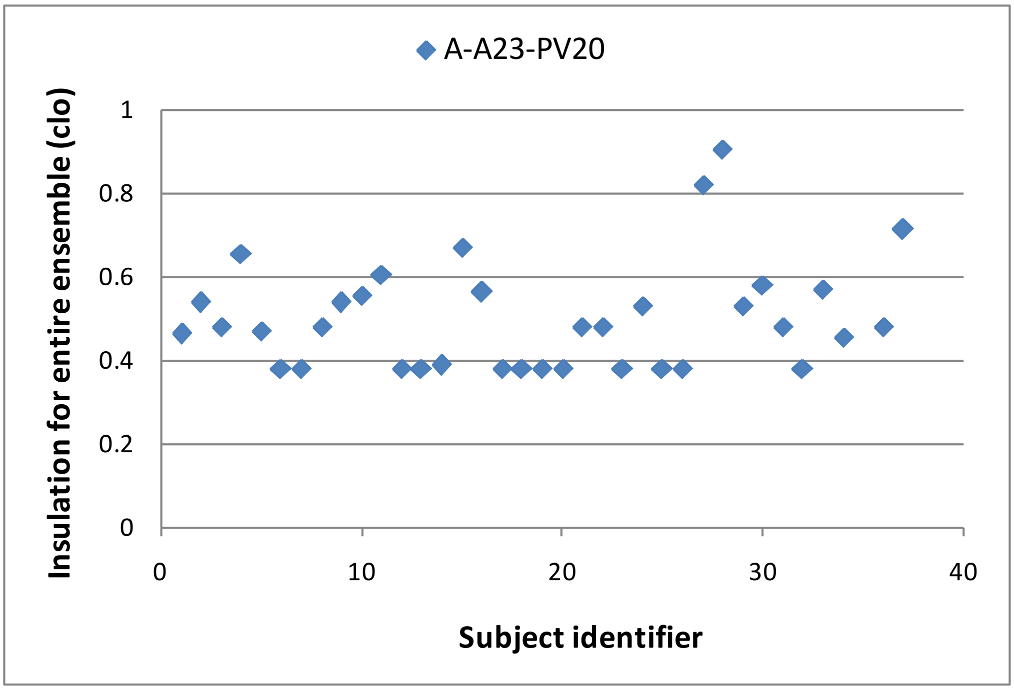


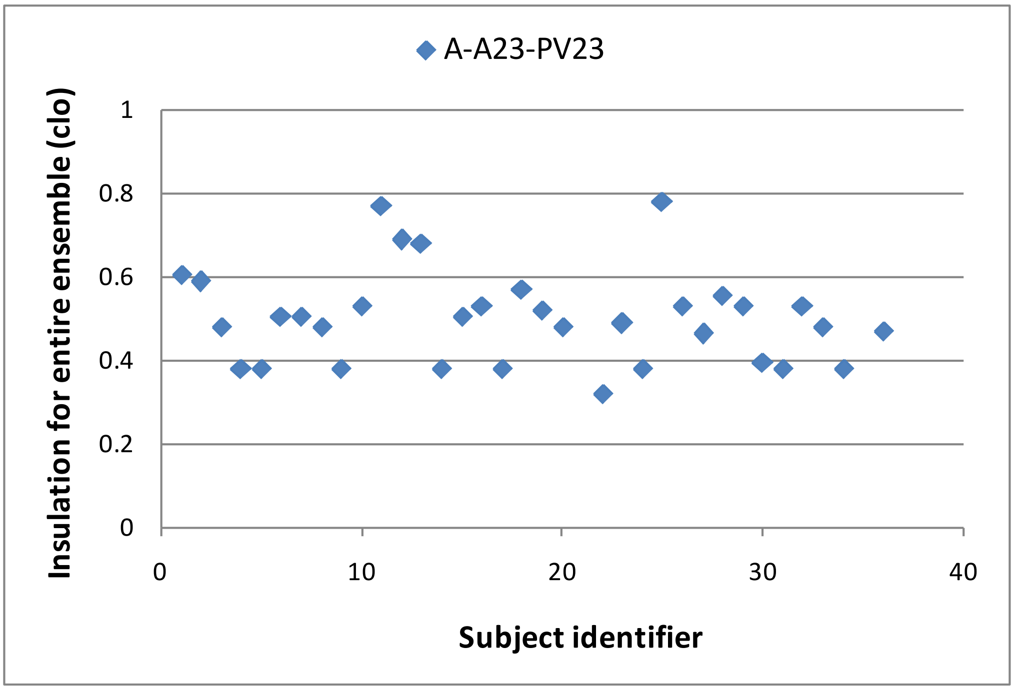


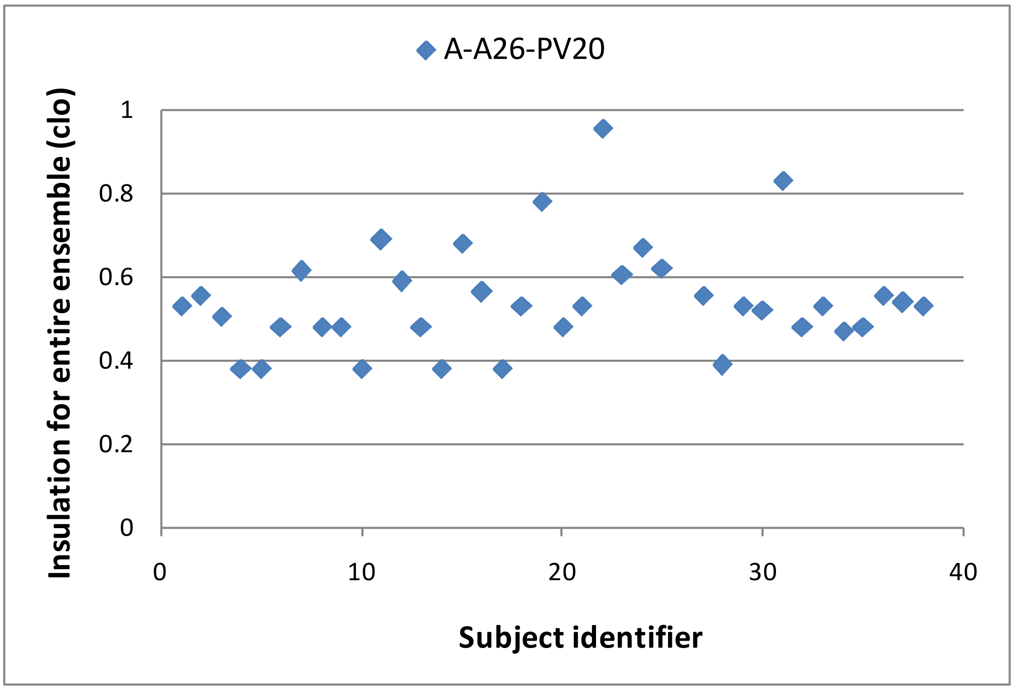

D-A26-PV20
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

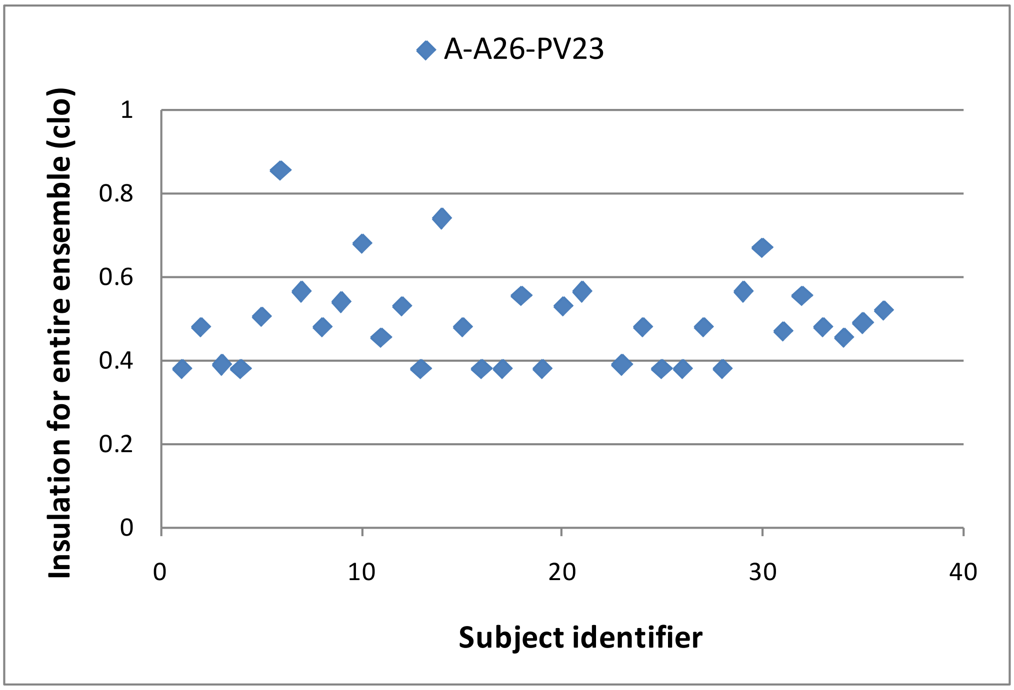

D-A26-PV23
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

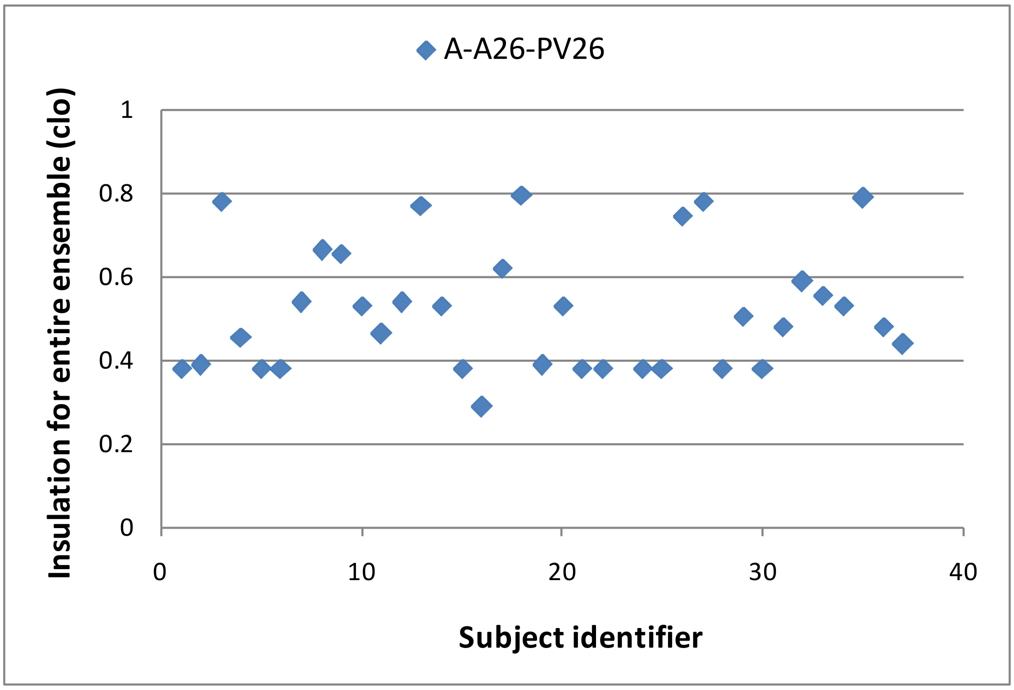

D-A26-PV26
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier


A-A23-PV20
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

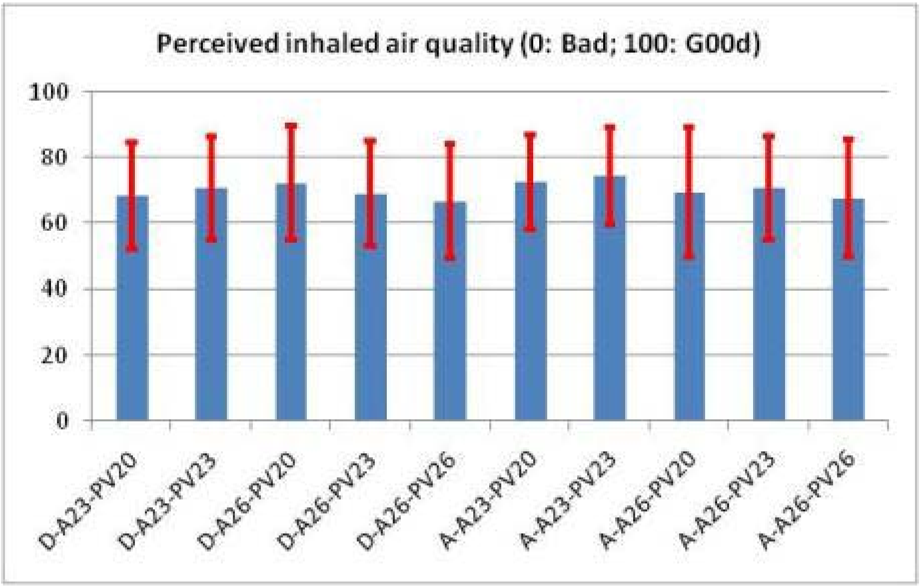

A-A23-PV23
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

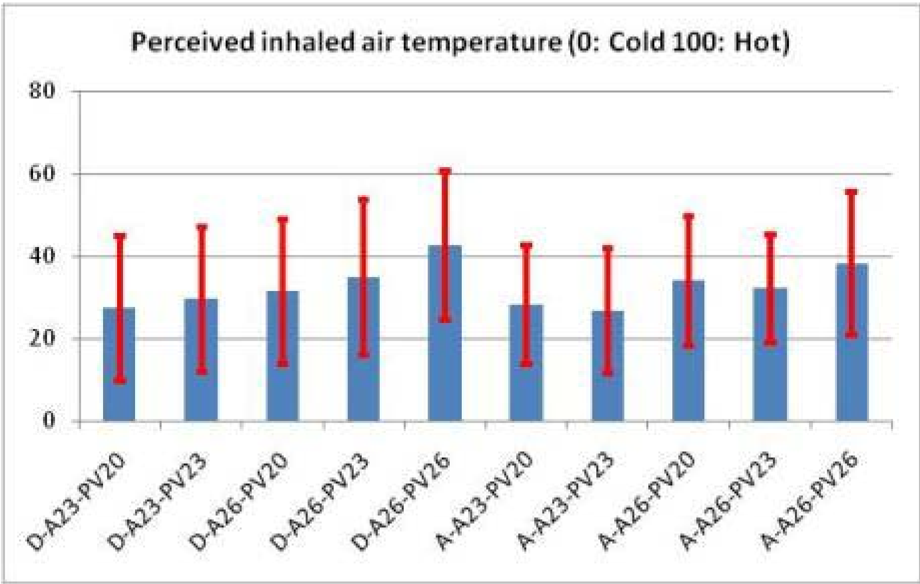

A-A26-PV20
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

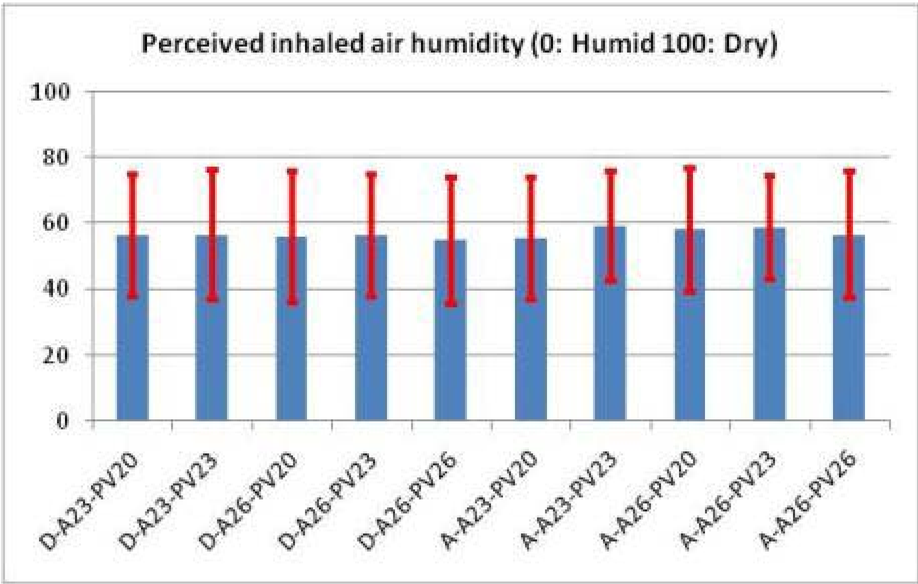

A-A26-PV23
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

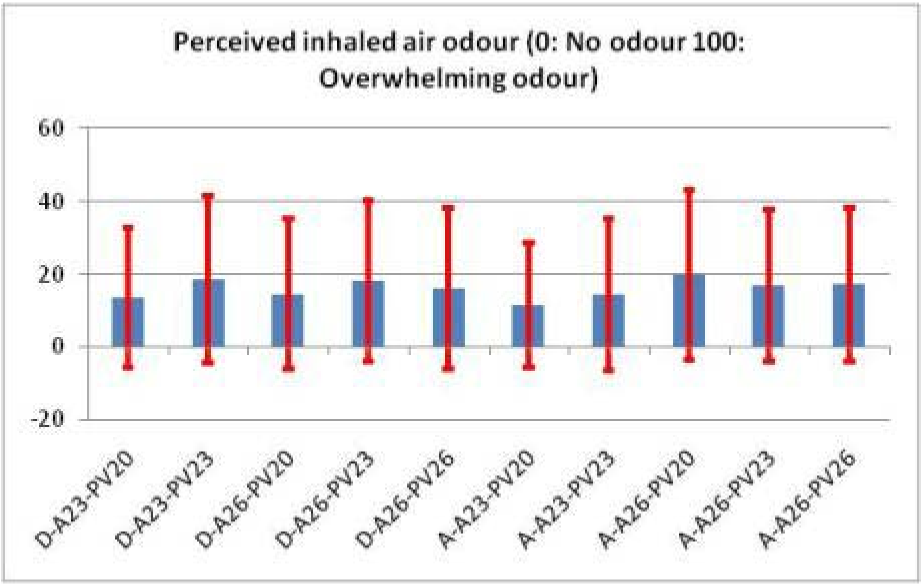

A-A26-PV26
Insulation for entire ensemble (clo)
1
0.8
0.6
0.4
0.2
0
0
10
20
30
40
Subject identifier

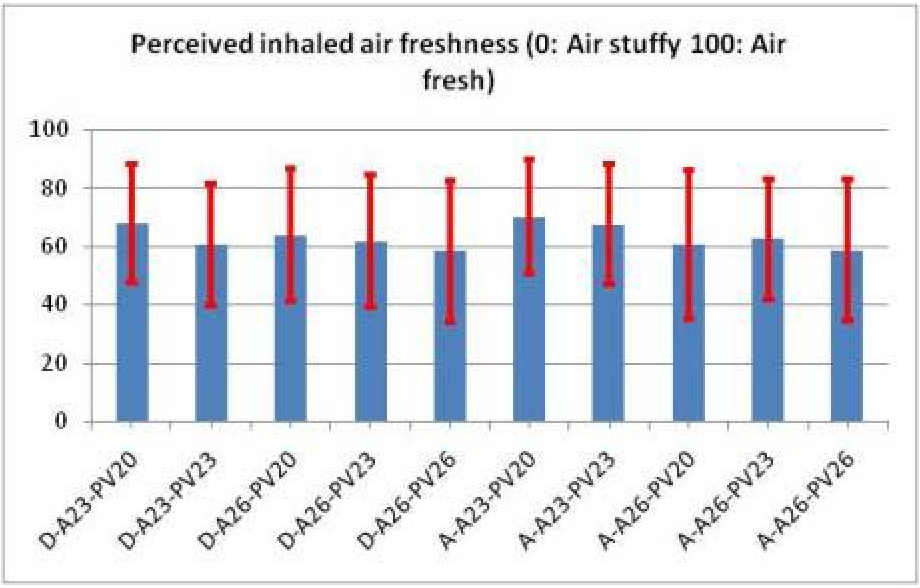
The results of indoor air quality



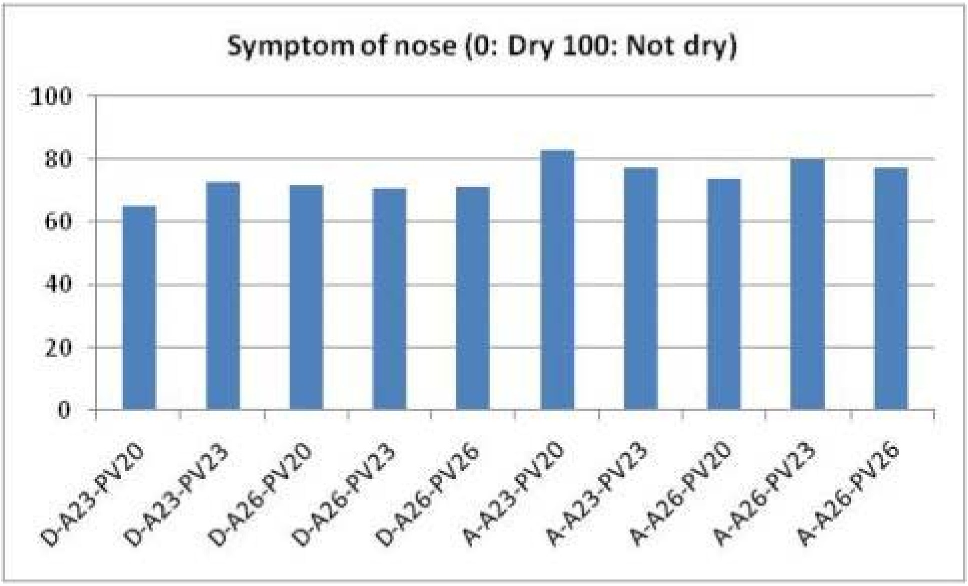


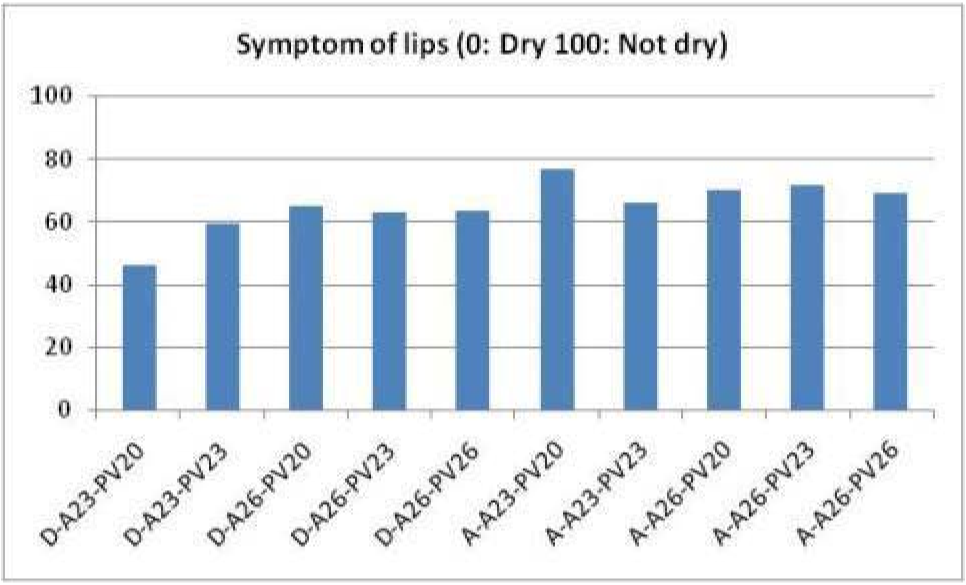

Perceived inhaled air humidity (0: Humid 100: Dry)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26

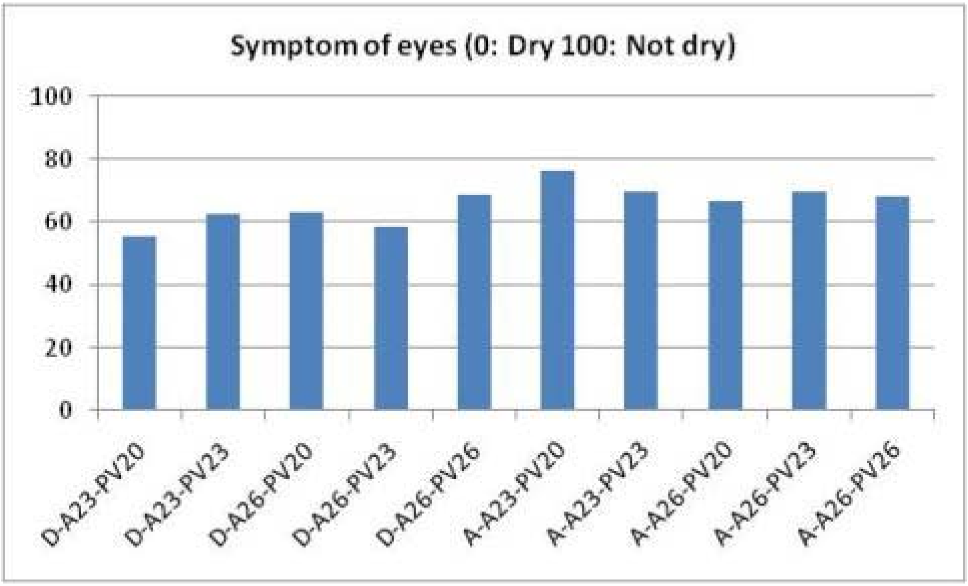

Perceived inhaled air odour (0: No odour 100: Overwhelming odour)
60
40
20
0
-20
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26

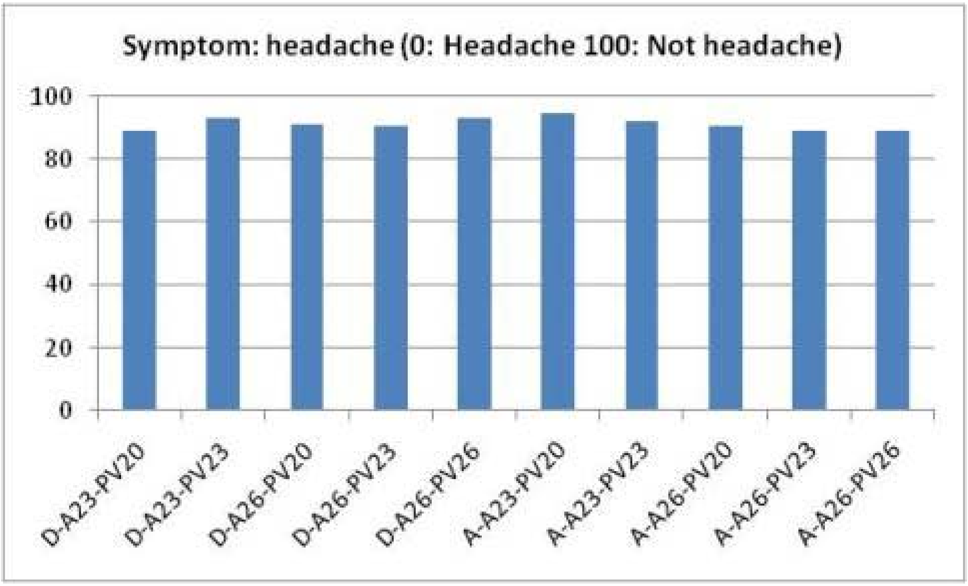

Perceived inhaled air freshness (0: Air stuffy 100: Air fresh)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26

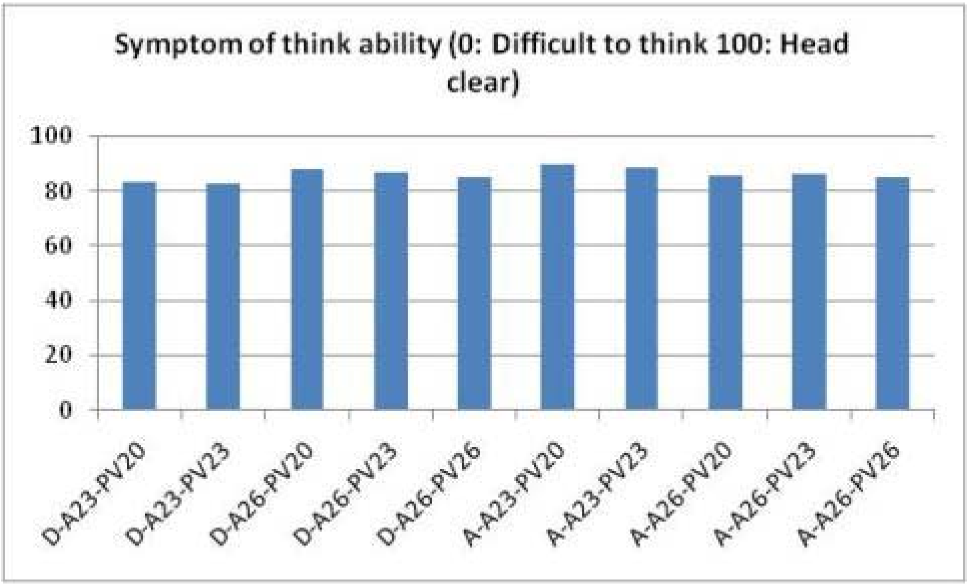
The results of indoor of sick building syndromes






Symptom of eyes (0: Dry 100: Not dry)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26


Symptom: headache (0: Headache 100: Not headache)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26


Symptom of think ability (0: Difficult to think 100: Head clear)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
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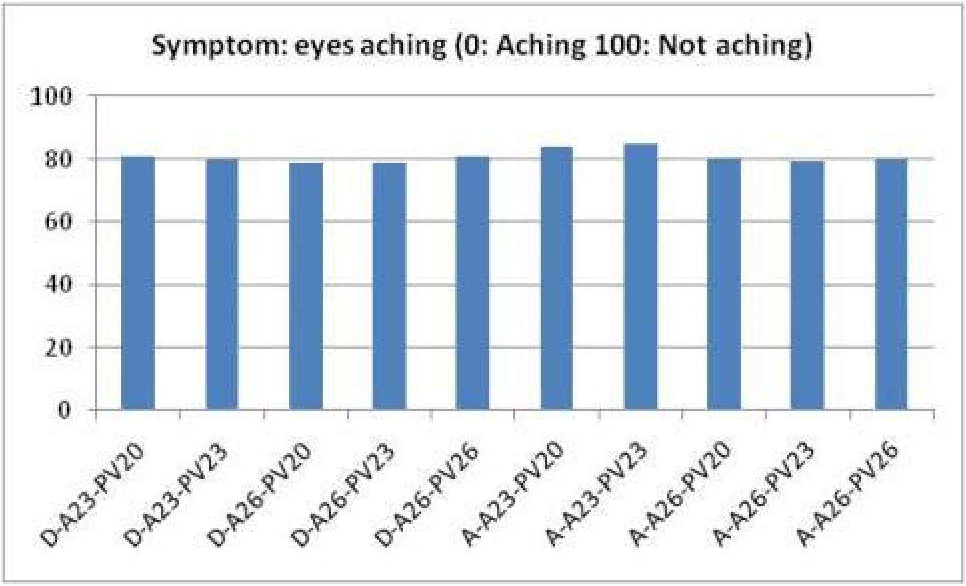
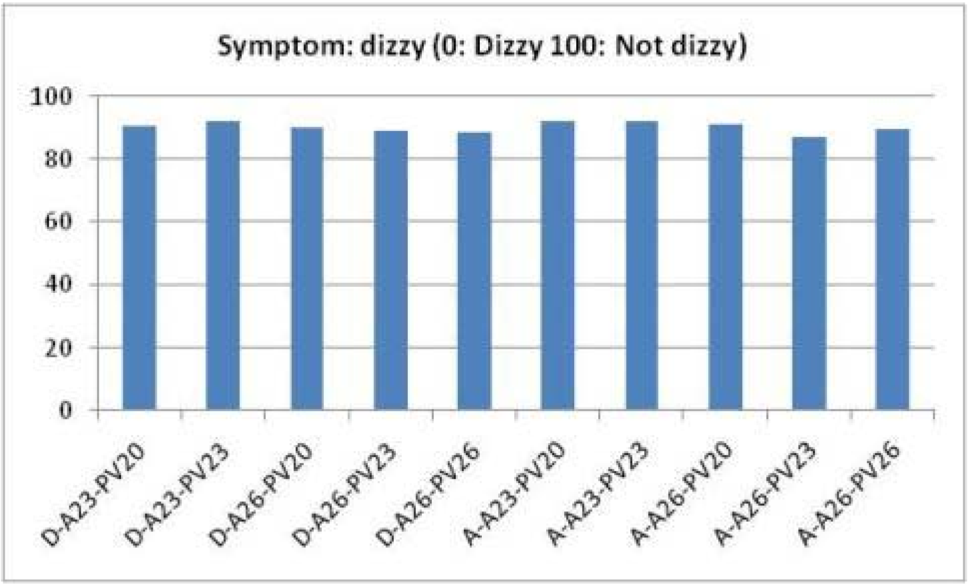
A-A26-PV23
A-A26-PV26


Symptom: eyes aching (0: Aching 100: Not aching)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26


Symptom: dizzy (0: Dizzy 100: Not dizzy)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
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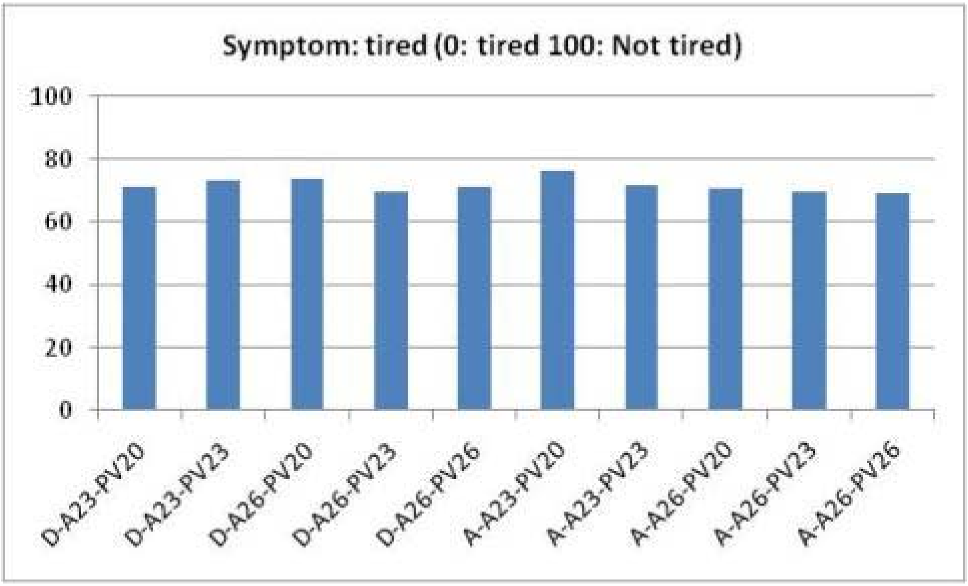
A-A26-PV20
A-A26-PV23
A-A26-PV26


Symptom: tired (0: tired 100: Not tired)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
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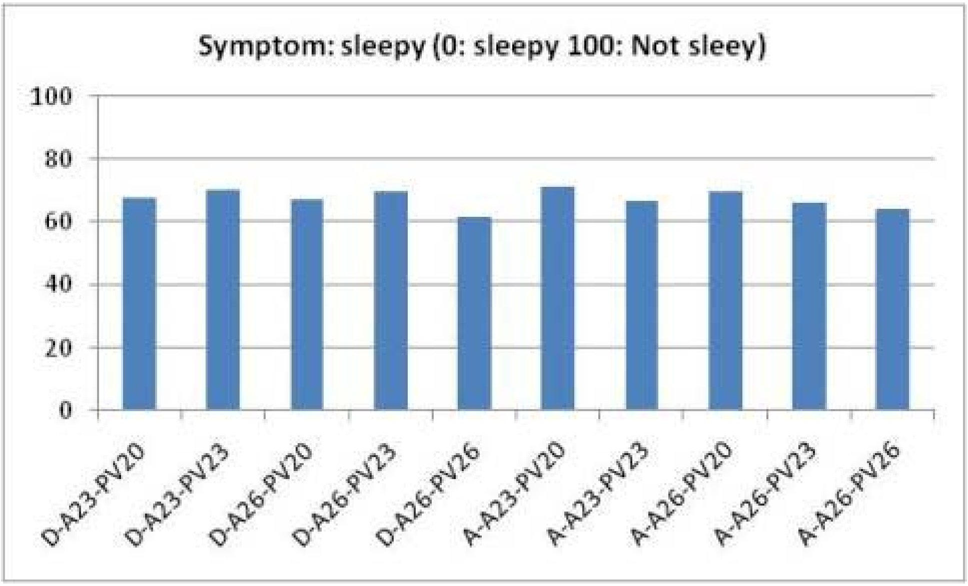
A-A26-PV26



Symptom: sleepy (0: sleepy 100: Not sleey)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
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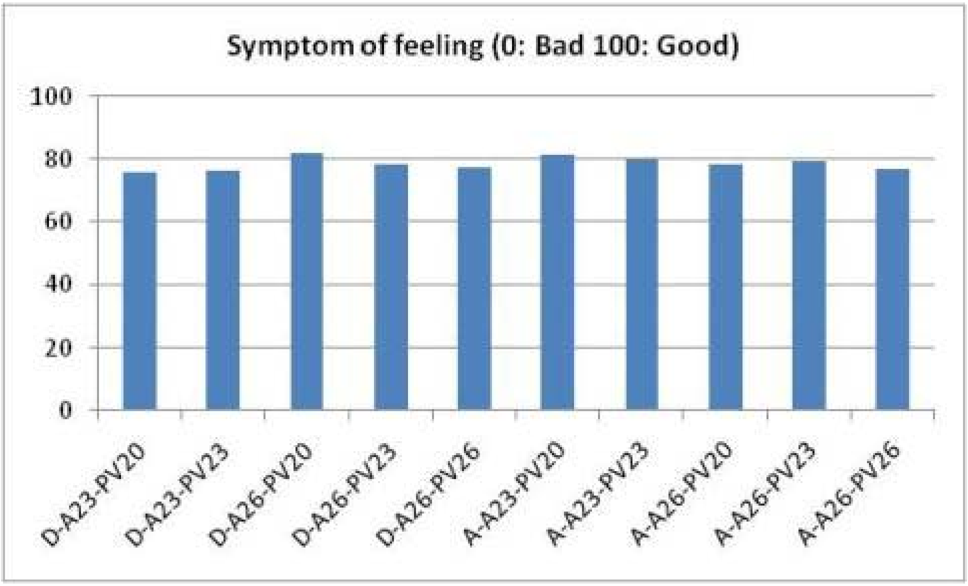
A-A26-PV23
A-A26-PV26



Symptom of feeling (0: Bad 100: Good)
100
80
60
40
20
0
D-A23-PV20
D-A23-PV23
D-A26-PV20
D-A26-PV23
D-A26-PV26
A-A23-PV20
A-A23-PV23
A-A26-PV20
A-A26-PV23
A-A26-PV26

Appendix C: List of publication

1. Raphael, B., Y CHEN, S C Sekhar and K W Tham, "Towards Intelligent Building Systems: Evaluating User Acceptance of Automatic Control". *Computing in Civil and Building engineering*, ed. Walid Tizani (2010). Nottingham: The University of Nottingham. (International conference on computing in civil and building engineering, 30 Jun - 2 Jul 2010, Nottingham, United Kingdom)

2. CHEN, Y, S C Sekhar, K W Tham and B. Raphael*, "Personalized Ventilation Control: Perception of Indoor Air Quality". *Clima 2010, 10th REHVA World Congress, "Sustainable Energy Use in Buildings"* (2010). Antalya: REHVA. (Clima 2010, 9 - 12 May 2010, Antalya, Turkey)

3. CHEN, Y, B. Raphael, S C Sekhar and K W Tham, "Energy Performance of Personalized Ventilation". *Clima 2010, 10th REHVA World Congress, "Sustainable Energy Use in Buildings"*, (2010). Antalya: REHVA. (Clima 2010, 9 - 12 May 2010, Antalya, Turkey)