DTU Byg

Combined radiant and convective indoor climate systems

Lauris Rezgals s111717
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Table of Contents

Introduction

1.1 Indoor environment quality

Nowadays people are spending more and more time in buildings and indoor spaces. The fast development of the high technologies has led office buildings to become a very common workplace for people. It is the environment where people spend most of the time during the day with the necessity of being productive. Therefore the indoor environment quality (IEQ) is very important. IEQ affects occupants' health, comfort and performance and depends on the thermal environment, indoor air quality (IAQ), lighting and acoustics.

Thermal environment is described by the air temperature, mean radiant temperature, air velocity and air humidity in the room. The perception of the thermal comfort also depends on the personal factors, such as activity level and clothing.

Unpleasant thermal environment can cause local thermal discomfort. The reasons can be draught (too high air velocity), temperature gradient (large vertical/horizontal temperature difference) and radiant asymmetry (cold/warm surrounding surfaces).

The indoor air quality is achieved by providing fresh air into the space and removing the pollution generated by occupants, devices, surface materials etc. Perceived air quality also depends on the temperature and humidity of the air – cool and dry air is felt fresher [1].

1.2 Present ventilation and air conditioning systems

Mixing ventilation

Mixing ventilation is a total volume ventilation principle that is very commonly used to ventilate spaces. Typically the supply air is provided in the room above the occupied zone from the ceiling or wall-mounted diffusers with a lower temperature than the room temperature. The air is provided from the diffusers with a high jet momentum and the supplied air with the lower temperature mixes with the warmer room air creating a uniform environment and cooling the space (Figure 1.1). However, the supplied air dilutes the pollutants in the room to acceptable level but does not remove them. Because of the high jet momentum there is a draught risk, especially at the upper body parts.

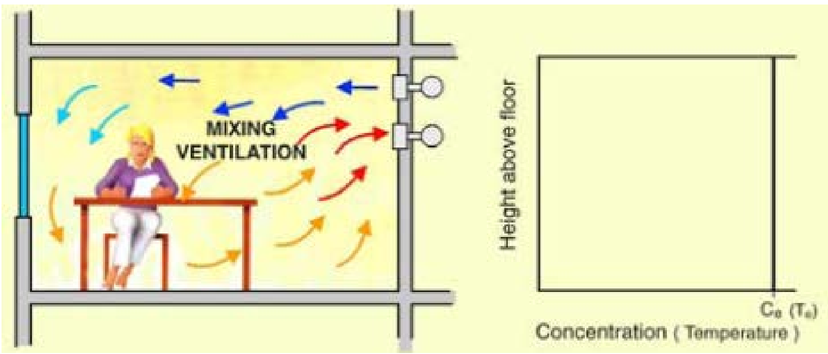


Figure 1.1: Mixing ventilation principle.

Displacement ventilation

Displacement ventilation is the second widely used ventilation principle. A cool fresh air is supplied in the room at low velocities close to the floor from air terminal devices (ATD), Figure 1.2. The supplied air reaches the heat sources, e.g. occupants, and is entrained up by the free convection providing cooling and removing the pollution in vertical direction. Therefore the temperature and the concentration of contaminants increase with height. The overhead convective heat gain is isolated and removed whereas with mixing ventilation it is mixed with the room air. Local discomfort risks associated with displacement ventilation are draught at the feet and the vertical temperature difference.

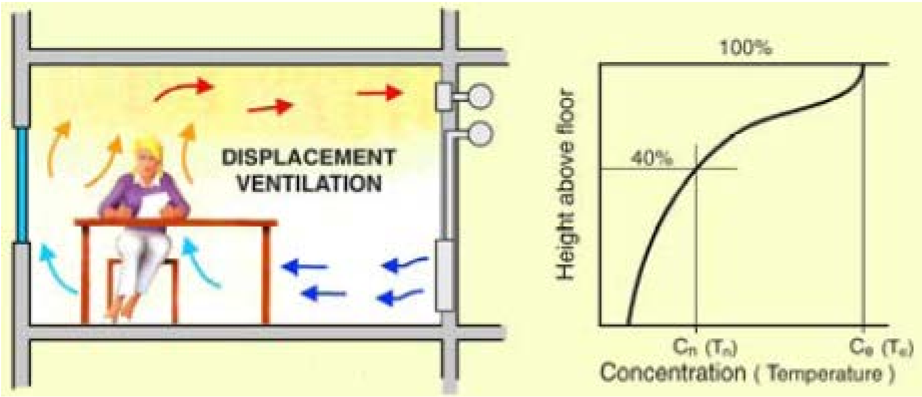


Figure 1.2: Displacement ventilation principle.

Chilled beams

Chilled beams are relatively new and advanced systems for ventilating and conditioning the air in the spaces. These systems are promising with regard to the energy saving which is achieved due to the high temperature cooling that they can provide. The main components of a chilled beam are a heat exchanger, a plenum box, air supply duct, water pipes and the frame with the slots. The working media is water.

There are different types of chilled beams. All chilled beams are divided into two main groups – active chilled beams and passive chilled beams.

Passive chilled beams are cooling coils placed close to the ceiling level. The working media is water. The warm room air flows through the heat exchanger by the natural convection. As a result, the air gets cooled and “drops down” back in the room, Figure 1.3. Passive chilled beams cool the air but do not provide fresh air to the room.

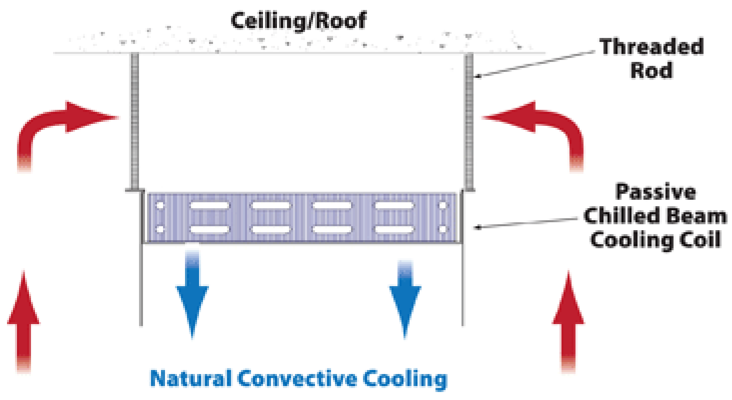


Figure 1.3: The working principle of a passive chilled beam.

The main difference between passive and active chilled beams is that active chilled beams are connected to the HVAC system where the air is processed and supplied to the chilled beam. Active chilled beams can be open or closed or with other configurations. The fresh air, which is called primary air, is supplied in a plenum box, where the pressure is increased because of the limited space. Along the plenum box small nozzles are placed. The air goes through these nozzles and forms small jets with high momentum. These jets induce or "take along" the warm room air, which first goes through the heat exchanger and gets cooled by forced convection. Then the primary air mixes with the cooled room air and then is supplied to the room via the slots along the chilled beam, Figure1.4.

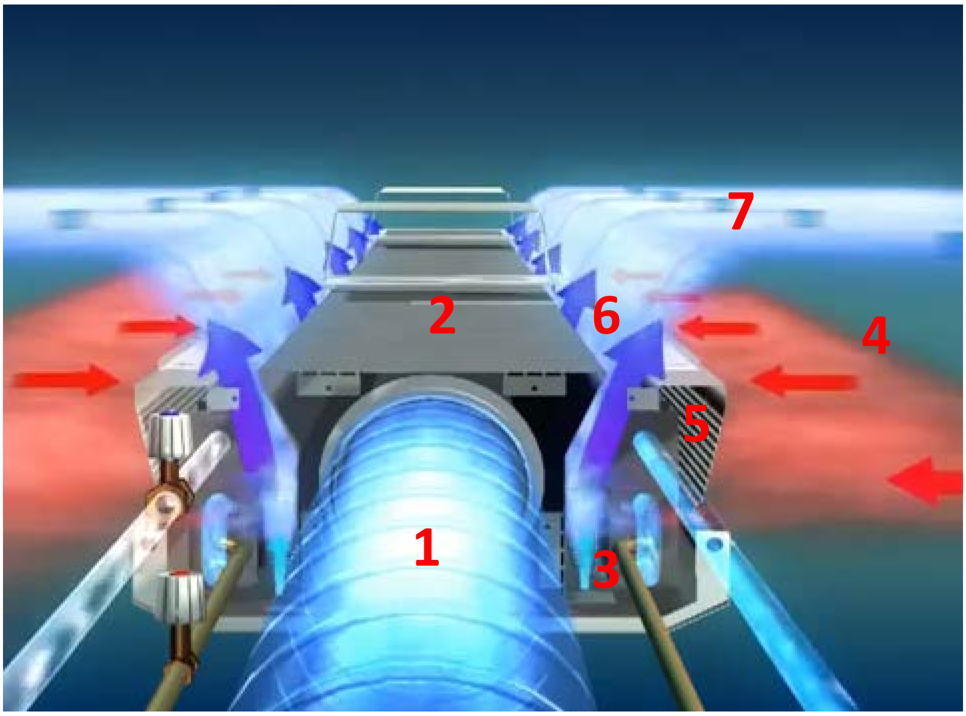


Figure 1.4: Working principle of active chilled beam (1 – primary air duct, 2 – plenum box, 3 – nozzle, 4 – warm room air, 5 – heat exchanger, 6 – slots, 7 – supplied air to the room).

Another variation of the chilled beams is chilled beam with radiant panels (CBR). In this case, radiant panels are connected to the chilled beam and the water goes through the panels and then to the cooling coil. This combination provides additional radiant cooling to the area below the chilled beam.

An important risk to be considered when chilled beams are used is the condensation. Chilled beams are not equipped with condensate draining systems. Therefore, the supplied air to the chilled beam must be dry enough and the temperature of the water must be controlled carefully not to exceed the dew point temperature of surrounding room air.

Chilled ceiling

The principle of chilled ceiling is to cool the space with radiant cooling. Chilled ceiling consists of panels that are made from heat conductive materials and pipes for the water, Figure 1.5. The water is circulating through the panels continuously and the panels hold a constant temperature. The absorbed heat is removed by the water.

Chilled ceiling can be combined with mixing or displacement ventilation to provide fresh air into the space and fulfill the requirements for acceptable indoor air quality, Figure 1.6.

Figure1.5: Chilled ceiling panel.

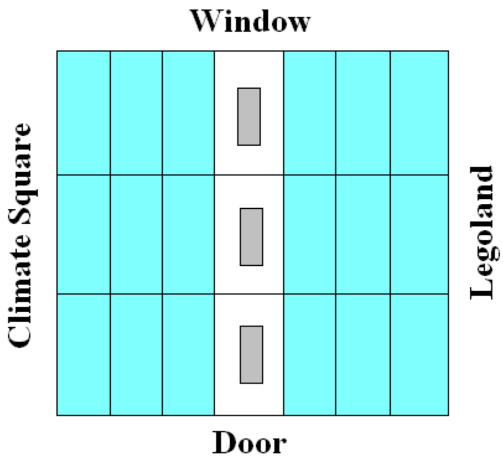


Figure 1.6: Chilled ceiling (18 panels in blue) combined with mixing ventilation (3 linear slot diffusers in grey).

When using chilled ceiling condensation risks have to be considered. A local thermal discomfort is possible because of a high radiant asymmetry. In addition, the air supplied tangentially from the diffuser (mixing ventilation) is in contact with the cold ceiling and therefore, when it reaches the walls, it drops down with high momentum. This can cause draught if occupants are located near the walls.

1.3 Benefits of combining convective and radiant cooling

Combining convective cooling with radiant cooling can lead to significant energy savings. If the radiant cooling effect is provided to the occupied zone, the background (air) temperature can be kept higher because the operative temperature due to the radiant cooling will be lower. Therefore, occupants will feel the air cooler.

The radiant cooling is provided by using water as the working media. Since density of air is 1.12 and of water 1000 kg/m^3 and specific heat capacity of water is 4.18 and of air 1.012 kJ/kg·K, water is 4000 times better heat conductor than air for the same volume of work fluid. Thus the energy necessary for the cooling is reduced. As a result, the necessary supply air is reduced because mainly it is needed for fresh air supply to fulfill the air quality requirements only. Therefore the duct dimensions, fans and HVAC requirements are reduced which also leads to energy savings (less air is moved and conditioned).

1.4 The importance of the room lay-out and heat load

The room lay-out is definitely an important factor that has to be considered when applying a ventilation system for particular space. The purpose of ventilation and air conditioning is to provide thermal comfort and acceptable air quality for the occupants in the occupied zone. However, the factors that determine the occupied zone can vary significantly between different lay-outs in the room. For example, sitting close to the window or away from the window during summer time makes a big difference and requires different parameters of the air supplied from the systems in order to ensure a comfortable environment.

The location of the cooling system indoors relative to the occupants is also important. For instance, occupants that are directly exposed to the chilled beam with radiant panels (sitting below the CBR) will feel different than those who are not, because of the additional more direct cooling effect from the radiant panels.

By testing different room lay-outs the system can be judged by its flexibility – the ability to provide thermal comfort regardless of the room lay-out.

The applied system should not only be flexible but also able to “adjust to” different heat loads in the room. There might be cases when the heat load is not constant and varies a lot, e.g. between summer and winter conditions if the building is not well isolated, different use of the room (office room with occasional meetings) etc.

If the system is able to operate effectively between different levels of heat load, it can be called “reliable”.

1.5 Justification

Nowadays there are many different kinds of ventilation systems that can be used to reach the desired indoor climate. The performance of both conventional ventilation systems (mixing ventilation) and modern ventilation systems (chilled ceiling with mixing ventilation, chilled beam, and chilled beam with radiant panels) under different levels of heat loads have been studied separately but not compared with each other in details. The performance of these systems under different heat source locations (room lay-outs) has not been studied. The comparison of these particular systems performance with respect to indoor thermal environment will be the purpose of this report.

2. Objectives

The objectives of this study are:

- To collect data of physical measurements comprising indoor environment and operation conditions of chilled beam, chilled beam with radiant panels, chilled ceiling with mixing ventilation and mixing ventilation systems.
- To study the effect of the room lay-out and the level of the heat load on the systems' performance on the generated thermal environment.
- To compare the systems with regard to the provided indoor thermal environment under different heat source locations (room lay-outs) and different levels of heat load.

3. Methods of measurements

3.1 Experimental facilities

Climatic chamber

The experiments were performed at the International Center for Indoor Environment and Energy (ICIEE), Department of Civil Engineering at the Technical University of Denmark. The chamber used was built in 2001 for multipurpose experiments and is well suited for long-term exposures. The temperature in the chamber can be controlled within the range 10 – 40°C and the relative humidity within the range 20-90 %. The HVAC system for this chamber can supply 43 – 612 m^3/h plus additional supply of 0 – 612 m^3/h. The chamber appears like a normal office room with the dimensions of 6.0 m x 4.7 m x 2.87 m (length x width x height). For this project the length was shorten by 1.1 m with an artificial well insulated wall which was used to simulate the window in the chamber. The free space was used as a control room behind the window.

The window area is 6.3 m^2 and consists of five Uponor panels with a wooden frame and a high conductive material and PE serpentine pipe inside. Water is circulating through the pipes and by controlling the temperature of the water, the necessary surface temperature of the "window" was achieved. Temperature of the water is controlled by a Julabo FC1200T thermo pump. The simulation window is covered with a metal plate painted in grey for maximum emissivity (Figure 3.1).

Figure 3.1: The simulated window with the sensors attached for surface temperature measurements.

The direct solar heat gain in the chamber was simulated with five electrical foils placed in the floor next to the window (Figure 3.2). The surface temperature of the foils was controlled by a voltage transformer with a potentiometer [1].

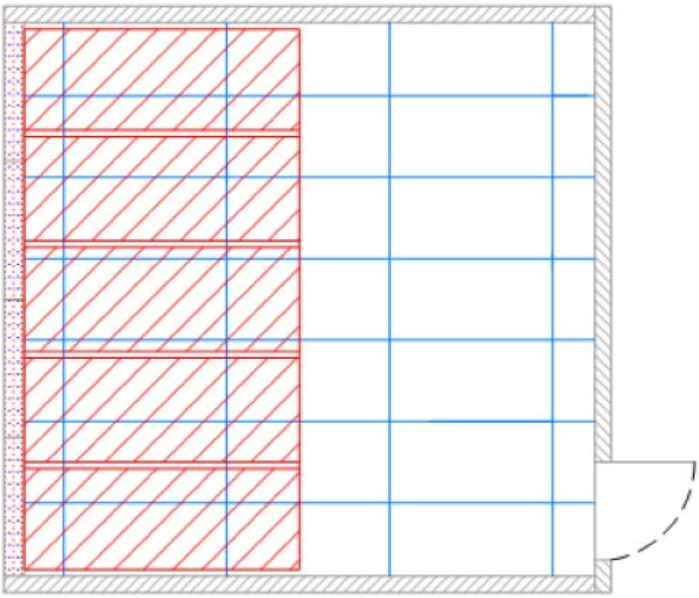


Figure 3.2: Electrical foils for direct solar heat gain simulation.

Systems used for the experiments

In this project four different systems were studied – chilled beam, chilled beam with radiant panels, chilled ceiling with mixing ventilation and mixing ventilation. All systems except mixing ventilation require both air and water.

The air supply system used for the experiments is shown in Figure 3.3 (front view) and Figure 3.4 (top view).

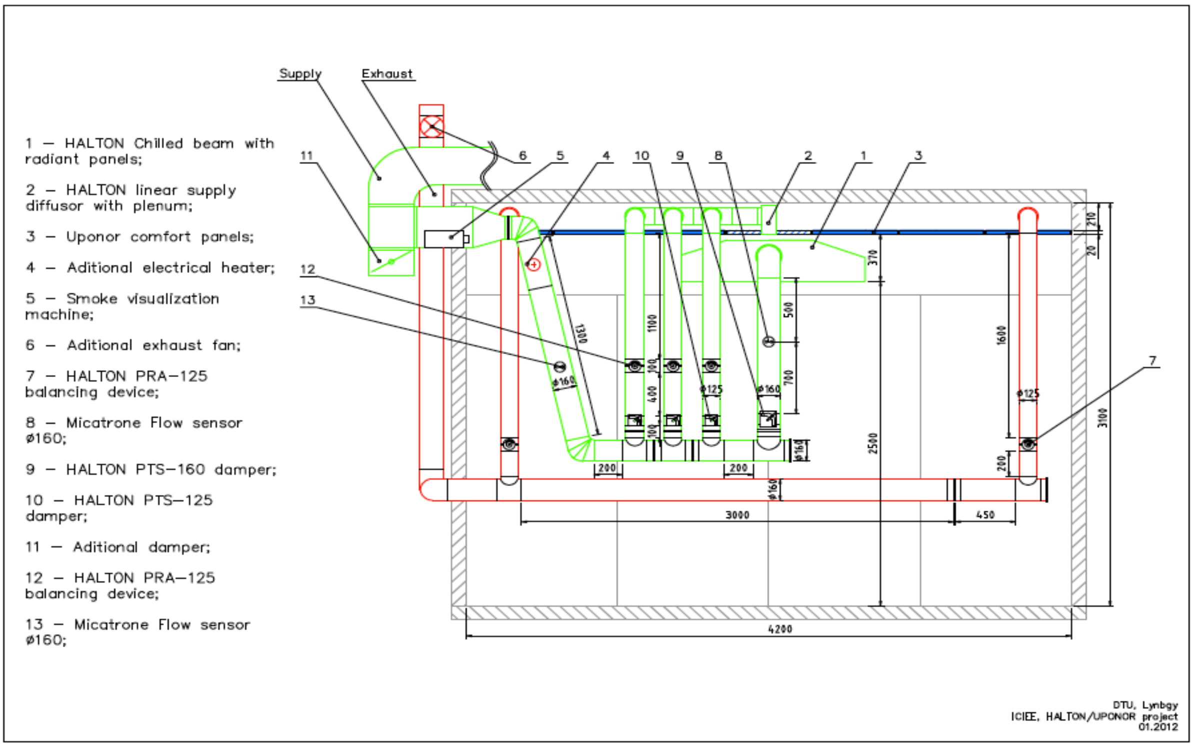


Figure 3.3: Air system used for the experiments in the chamber (front view).

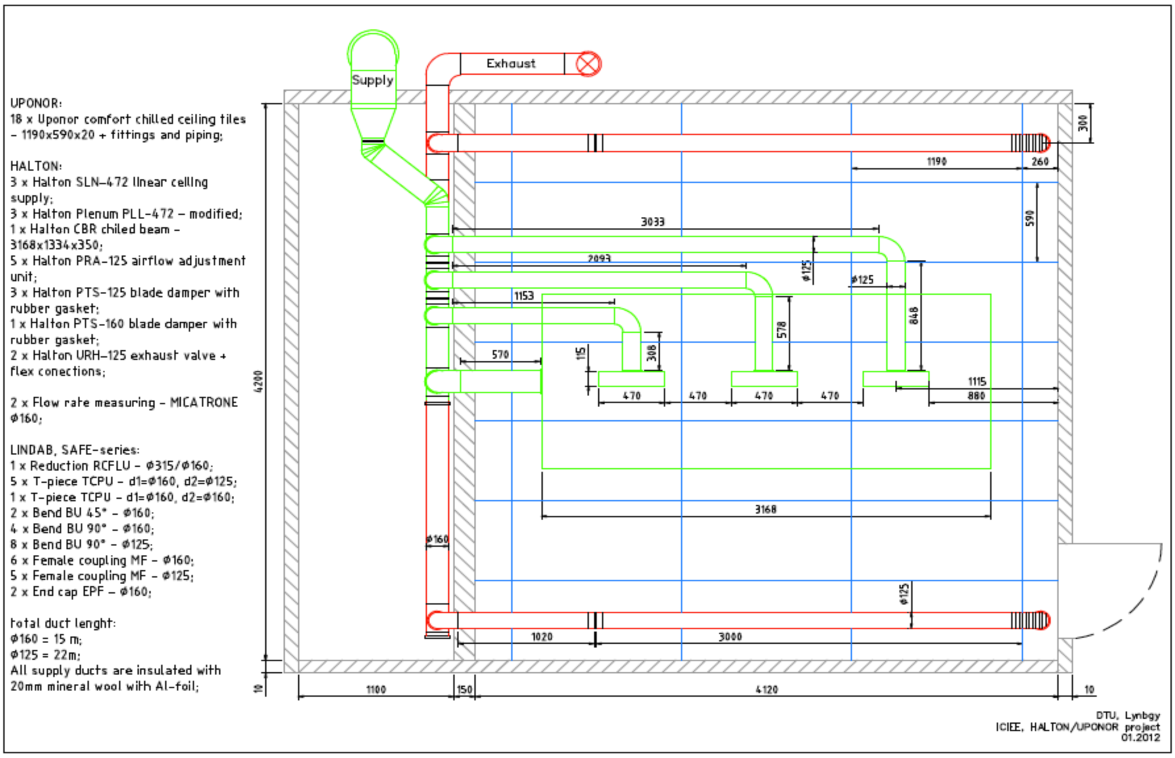


Figure 3.4: Air system used for the experiments in the chamber (top view).

The supply ducts are shown in green and the exhaust ducts are shown in red. The air handling unit (AHU) is located in the basement. The minimum flow rate that this system can provide was too high for the experiments. Because of the low air flow rate needed and the long distance from the AHU the supplied air temperature was not high enough. To ensure the necessary flow rate and temperature of the supplied air, an additional damper (11 in Figure 3.3) and additional electrical heater (4 in Figure 3.3) were used.

The air supply consists of four branches where the first three from the left are for the three linear slot diffusers (Figure 3.5) used for the mixing ventilation – one branch for each. The fourth branch is to supply the air to the chilled beam. Halton PRA balancing devices are used to distribute the air equally in each branch.

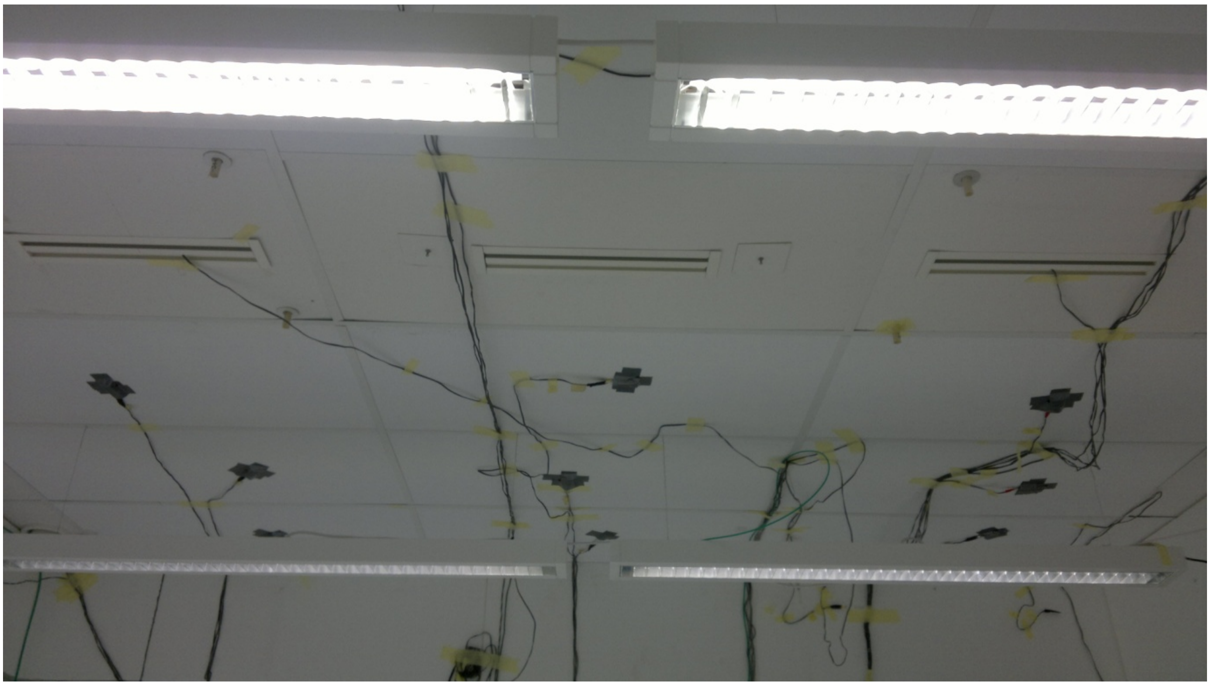


Figure 3.5: Three linear slot diffusers used for mixing ventilation.

The air is exhausted from the right and left corner of the room, Figure 3.6. The exhaust diffuser is shown in Figure 3.7. The exhausted air from each branch is sucked in the collector duct and then exhausted in the ambient space by a separate exhaust fan (6 in Figure 3.3). The fan is controlled with a voltage transformer potentiometer that allows balancing the system, i.e. supply air is equal to the exhaust air. Similar to the supply, the air in both branches is balanced by using PRA balancing devices from Halton.

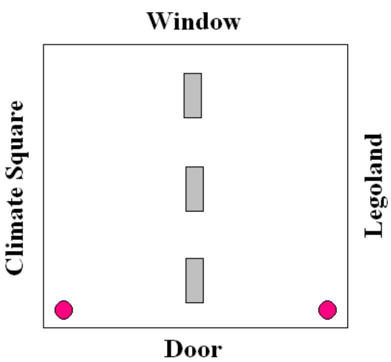


Figure 3.6: Linear slot diffusers in grey and exhaust diffusers in red – top view.

Figure 3.7: Exhaust diffuser used for the experiments.

Cold water used in the chilled beam, chilled beam with radiant panels and chilled ceiling (combined with mixing ventilation) is provided by a main chiller at the research facilities. The chiller is located in the basement and works in mode 7°C/12°C. This temperature is too low for the experiments and therefore a separate system is used, which is integrated between the main chiller and the ventilation systems used in the chamber, Figure 3.8.

The blue line is the supply water from the chiller and the purple dashed line is the return water from the systems installed in the chamber.

The flow rate to the water cooling systems in the chamber is controlled by a circulation pump included in the additional water control system. The temperature control of the water is based on mixing between supply and return water. For this purpose a mixing valve is mounted on the return pipe. This mixing valve is controlled by a main temperature controller (Yokogawa UT150). This is a closed loop control system where the input signal for the controller is given by a temperature sensor Pt100. If the water temperature is too low, the mixing valve closes and water circulates in a closed loop through the water cooling systems. If the temperature cannot be increased enough by mixing with the return water, an electrical heater is turned on to heat the supply water to the set temperature. If the water temperature is too high, the mixing valve opens and more chilled water from the main chiller is supplied in the system decreasing further the water temperature [1].

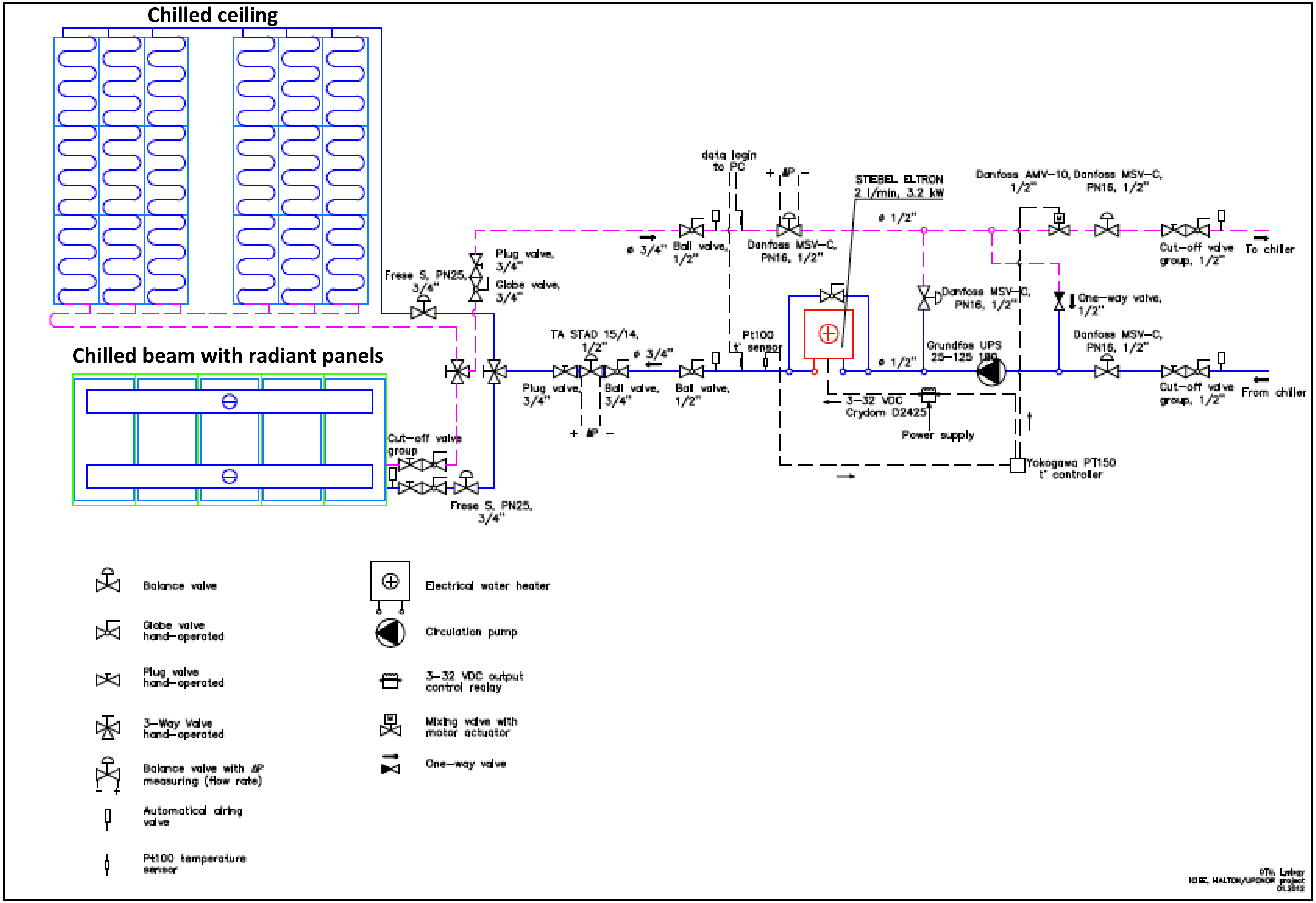


Figure 3.8: Cold water system.

The switching between chilled beam and chilled ceiling systems is achieved by 3-way valves on the supply and return water pipes, Figure 3.8. Before the chilled beam a cut-off valve is mounted for easy and fast disconnecting from the water line system.

The purpose of this project is to study both chilled beam and chilled beam with radiant panels as already mentioned. In order to switch from one mode to another, a 3-way manual valve was used (Figure 3.9).

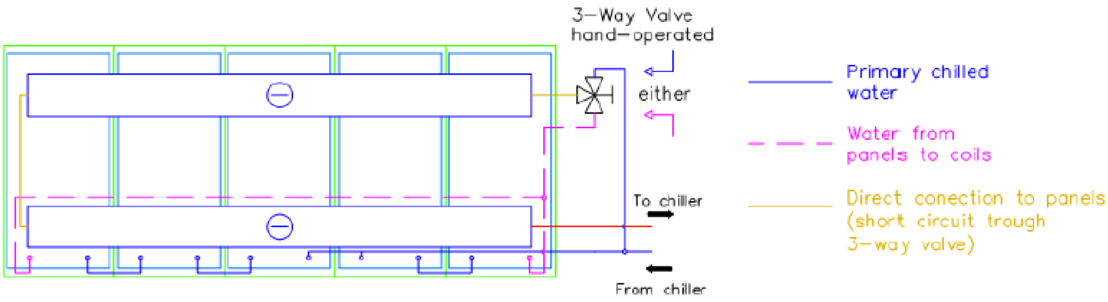


Figure 3.9: Switching between chilled beam and chilled beam with radiant panels.

When chilled beam with radian panels is used, the supply water (blue line) first goes through the radiant panels (5 panels in a row), then it (purple dashed line) goes into the cooling coils (yellow line) and then returns to the chiller (red line).

When only chilled beam is used, the 3-way valve is turned to the position so that the return (purple dashed) line is closed and the supply water goes directly to the cooling coils [1].

The chilled beam is made by Halton. The dimensions are 3.2 m x 1.6 m x 0.35 m (length x width x height) and weights 120 kg, Figure 3.10.

Figure 3.10: Chilled beam with radiant panels made by Halton.

This chilled beam has two heat exchangers – one on each side. The plenum box for the supplied fresh air is in the middle. The cool air is supplied to the room symmetrically from both sides.

Figure 3.8 also shows schematically the chilled ceiling system. The water goes through the pipes installed in the panels (blue line) and returns to the chiller (purple dashed line). The return line is made with a loop in order to balance the water flow rate through the panels.

The chilled ceiling consists of 18 radiant panels divided in 6 parallel groups with 3 panels in series in each group. There are no radiant panels in the middle row because in this row the linear slot diffusers are installed (Figure 3.11). The total area of the panels is 12.96 m^2 or 75% of the total floor area.

Figure 3.11: Chilled ceiling radiant panels with sensors attached and linear slot diffusers in the middle row.

3.2 Experimental conditions

Studied cases

In total 32 different cases were studied, Table 3.1. Four different systems were used – chilled beam (CB), chilled beam with radiant panels (CBR), chilled ceiling combined with mixing ventilation (CCMV) and mixing ventilation alone (MV). Each system was tested for 4 different room set-ups (lay-outs), Figure 3.12. The red wall is the simulation window, blue squares are the tables, blue circles are the occupants (thermal manikins) and with the blue line is the position of the chilled beam shown.

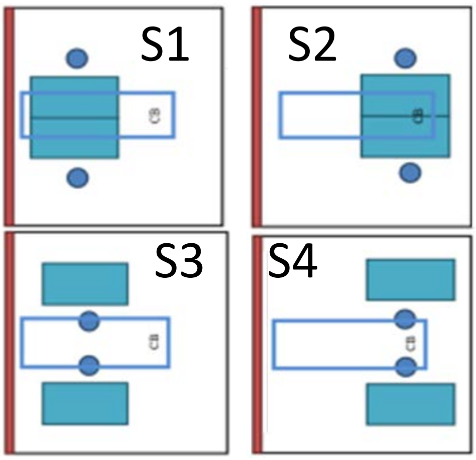


Figure 3.12: Room set-ups studied.

In set-up 1 (S1) and set-up 2 (S2) the occupants are facing each other, the tables are together with a partition in between (Figure 3.13). The occupants are not sitting exactly below the chilled beam and hence are not directly exposed to it. The difference between S1 and S2 is that in S1 occupants are sitting close to the window.

Figure 3.13: Room set-up 1.

In set-up 3 (S3) and set-up 4 (S4) the occupants are sitting turned with backs to each other (Figure 3.14). Also they are directly exposed to the chilled beam (sitting right below the chilled beam). In S3 occupants are sitting closer to the window compared to S4.

Figure 3.14: Room set-up 3.

All set-ups for all systems were studied under two different heat loads – design heat load or high heat load of 65 W/m^2 (H as “high”) and usual heat load or low heat load of 39 W/m^2 (L as “low”), Table 3.2.

Table 3.1: All cases studied in the project.

Case No.	System	Heat load	Room set-up
1	Chilled beam (CB)	Design (H)	S1
2	Chilled beam (CB)	Design (H)	S2
3	Chilled beam (CB)	Design (H)	S3
4	Chilled beam (CB)	Design (H)	S4
5	Chilled beam (CB)	Usual (L)	S1
6	Chilled beam (CB)	Usual (L)	S2
7	Chilled beam (CB)	Usual (L)	S3
8	Chilled beam (CB)	Usual (L)	S4
9	Chilled beam with radiant panels (CBR)	Design (H)	S1
10	Chilled beam with radiant panels (CBR)	Design (H)	S2
11	Chilled beam with radiant panels (CBR)	Design (H)	S3
12	Chilled beam with radiant panels (CBR)	Design (H)	S4
13	Chilled beam with radiant panels (CBR)	Usual (L)	S1
14	Chilled beam with radiant panels (CBR)	Usual (L)	S2
15	Chilled beam with radiant panels (CBR)	Usual (L)	S3
16	Chilled beam with radiant panels (CBR)	Usual (L)	S4
17	Chilled ceiling combined with mixing ventilation (CCMV)	Design (H)	S1
18	Chilled ceiling combined with mixing ventilation (CCMV)	Design (H)	S2
19	Chilled ceiling combined with mixing ventilation (CCMV)	Design (H)	S3
20	Chilled ceiling combined with mixing ventilation (CCMV)	Design (H)	S4
21	Chilled ceiling combined with mixing ventilation (CCMV)	Usual (L)	S1
22	Chilled ceiling combined with mixing ventilation (CCMV)	Usual (L)	S2
23	Chilled ceiling combined with mixing ventilation (CCMV)	Usual (L)	S3
24	Chilled ceiling combined with mixing ventilation (CCMV)	Usual (L)	S4
25	Mixing ventilation (MV)	Design (H)	S1
26	Mixing ventilation (MV)	Design (H)	S2
27	Mixing ventilation (MV)	Design (H)	S3
28	Mixing ventilation (MV)	Design (H)	S4
29	Mixing ventilation (MV)	Usual (L)	S1
30	Mixing ventilation (MV)	Usual (L)	S2
31	Mixing ventilation (MV)	Usual (L)	S3
32	Mixing ventilation (MV)	Usual (L)	S4

Set-up for the measurements

The physical parameters measured in the chamber were operative temperature, air temperature, air velocity, surface temperature and radiant asymmetry.

Operative temperature, air temperature and air velocity were measured at different points in the chamber. The measured grid can be seen in Figure 3.15.

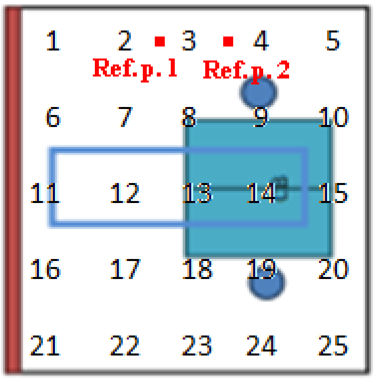


Figure 3.15: Measuring points in the chamber.

The measurements were not performed at points 1, 6, 11, 16 and 21. At these points according to the results from a previous study with three of the systems, i.e. CB, CBR and CCMV [1] air velocities were very low and air temperature was constant and close to 28°C. Thus it was decided to omit these points.

Because of inaccessibility the measurements were not performed at the following points:

- 7, 12 and 17 for set-up 1;
- 9, 14 and 19 for set-up 2;
- 7 and 17 for set-up 3;
- 9 and 19 for set-up 4.

The measurements of air temperature, operative temperature and air velocity were performed at 8 heights - 0.05, 0.10, 0.3, 0.6, 1.1, 1.7, 2.0 and 2.4 m. For this purpose a stand with the sensors was prepared. See Figure 3.16.

The reference air and operative temperature was measured at two points - between measuring points 2 and 3 and between points 3 and 4 respectively. At both locations the temperatures were measured at 1.1 m from the floor, Figure 3.15 and Figure 3.16.

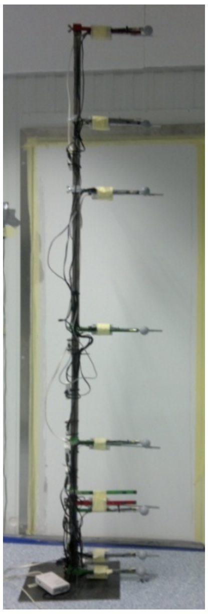
a)  b) 

Figure 3.16: a) Stand with the sensors at eight different heights (left) and b) stands with the sensors at the reference points (right).

In total 60 thermistor sensors were used to measure the surface temperature and air temperature at different locations. The locations are shown in Figure 3.17.

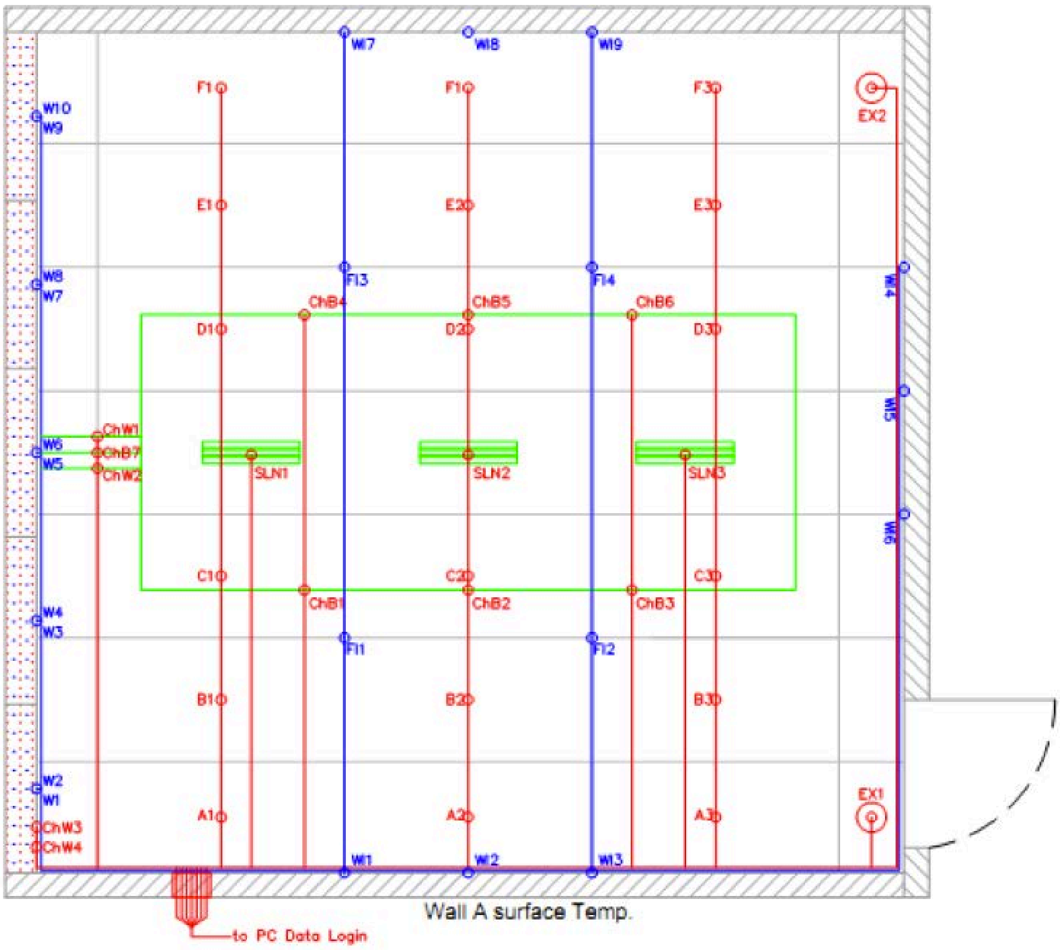


Figure 3.17: Different locations for the surface temperature measurements in the chamber.

The temperature was measured for surfaces such as walls, floor, ceiling, window, chilled beam radiant panels and water system pipes. Air temperature was measured in supply and exhaust diffusers, chilled beam air linear supply slots and exhausts.

Operating conditions

According to the ISO Standard 7730 (2005) [2], category B of thermal environment is what this study is aiming for. The target temperature in the room was 26°C at 1.1 m height from the floor. The supply air temperature was designed to be 16°C. Higher temperature could bring too much moist in the room which could increase the risk from condensation on the radiant panel surfaces.

The two heat loads used for the measurements were design heat load of 1100 W (65 W/m^2) and usual heat load of 650 W (39 W/m^2), Table 3.2. Heat sources were 2 occupants (thermal manikins), 2 lap-top computers, lighting, window surface and direct solar load on the floor simulated by the electrical foils. The window surface temperature at design heat load was 34°C and the direct solar load on the floor was 250 W but at usual heat load the window surface temperature was 30°C and no direct solar load was simulated on the floor.

The air supply flow rate is calculated according to the standard EN 15251 (2007) [3] for category B. There are 2 occupants in the office. According to the standard 7 L/s are required per person. That sums up to 14 L/s. Additional air is supplied for the building emissions. Assuming the office to be low polluting, the required flow rate is 0.7 L/s per m^2. In total the minimum required air flow rate is 26 L/s, which has been kept constant for all of the cases.

Table 3.2: Operating conditions for the systems studied.

Target room air temperature: 26°C		Design (high) heat load			Usual (low) heat load		
Room area: 17,1 m^2		Number	W per unit	W in total	Number	W per unit	W in total
Chilled beam (CB)	Occupants, [W]	2	87	174	2	87	174
	Computers, [W]	2	65	130	2	65	130
	Lighting, [W]	4	40	160	4	40	160
	Window surface load, [W]	-	403	403	-	201	201
	Direct solar load on the floor, [W]	-	250	250	-	-	-
	Total load, [W]	1117			665		
	Total load per room area, [W/m2]	65			39		
	Supply air temperature, [°C]	16			16		
	Supply air flow rate, [L/s]	26			26		
	Supply water temperature, [°C]	15,9			21,4		
	Supply water mass flow rate, [kg/s]	0,12			0,12		
Chilled beam with radiant panels (CBR)	Occupants, [W]	2	87	174	2	87	174
	Computers, [W]	2	65	130	2	65	130
	Lighting, [W]	4	40	160	4	40	160
	Window surface load, [W]	-	403	403	-	201	201
	Direct solar load on the floor, [W]	-	250	250	-	-	-
	Total load, [W]	1117			665		
	Total load per room area, [W/m2]	65			39		
	Supply air temperature, [°C]	16			16		
	Supply air flow rate, [L/s]	26			26		
	Supply water temperature, [°C]	17			21,7		
	Supply water mass flow rate, [kg/s]	0,067			0,067		
Chilled ceiling combined with mixing ventilation (CCMV)	Occupants, [W]	2	87	174	2	87	174
	Computers, [W]	2	65	130	2	65	130
	Lighting, [W]	4	40	160	4	40	160
	Window surface load, [W]	-	403	403	-	201	201
	Direct solar load on the floor, [W]	-	250	250	-	-	-
	Total load, [W]	1117			665		
	Total load per room area, [W/m2]	65			39		
	Supply air temperature, [°C]	16			16		
	Supply air flow rate, [L/s]	26			26		
	Supply water temperature, [°C]	15,5			20,7		
	Supply water mass flow rate, [kg/s]	0,12			0,07		
Mixing ventilation (MV)	Occupants, [W]	2	87	174	2	87	174
	Computers, [W]	2	65	130	2	65	130
	Lighting, [W]	4	40	160	4	40	160
	Window surface load, [W]	-	403	403	-	201	201
	Direct solar load on the floor, [W]	-	250	250	-	-	-
	Total load, [W]	1117			665		
	Total load per room area, [W/m2]	65			39		
	Supply air temperature, [°C]	16			16		
	Supply air flow rate, [L/s]	90			55		

3.3 Instrumentation

Air velocity

For velocity measurements inside the chamber multichannel low velocity thermal anemometer AirSpeedSys 5000 with eight omnidirectional velocity sensors was used. The velocity sensors (shown in Figure 3.18) are based on the hot wire technology and are omnidirectional. Thus they measure the speed of air. The range of the velocity measured by the instrument is from 0.05 m/s up to 5 m/s. The accuracy of the velocity (in fact, speed) measurement is 0.02 m/s ±1% of the reading in the range between 0.05 m/s and 1 m/s [1]. All probes were precisely calibrated for velocity measurements prior to the experiments.

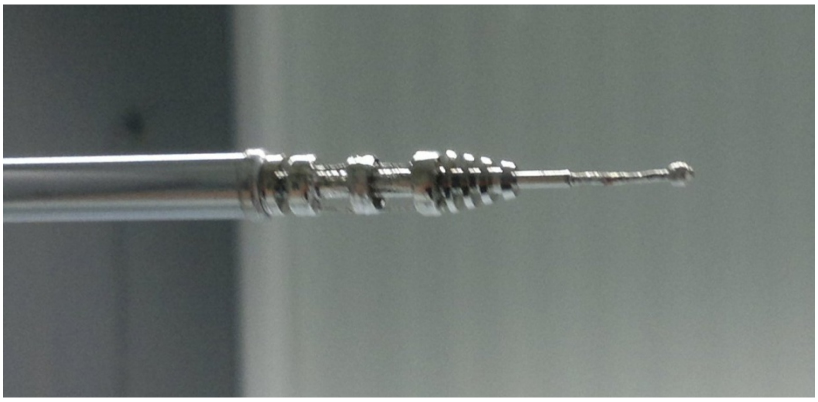


Figure 3.18: Air velocity sensor.

Operative and air temperature

The air and operative temperature measurements were performed by specially developed sensors (Figure 3.19) at the ICIEE at the Technical University of Denmark. Both sensors use thermistor to measure the voltage change as a function of the temperature change and record that voltage in real time by a portable data logger with two separate voltage channels (HOBO logger) [1]. Prior to the experiments both type of sensors were calibrated.

Figure 3.19: Sensors for the operative temperature (sphere) and air temperature measurements and HOBO data logger.

Thermistors

To measure supply, exhaust and surface temperatures Craftemp Astra thermistor sensors were used. The sensor measures the resistance in Ohms. Twenty of sixty sensors were connected to a data acquisition unit Agilent where the measured resistance was recorded (Figure 3.20). The measured resistance was transformed into temperature using the following equation:

$$t = \frac{3893.74}{\ln\left(\frac{R}{0.01294}\right)} - 273\ [°C]$$

Where “t” is the temperature [°C] and R is the measured resistance [Ohm].

The rest of the thermistors were connected to a computer and by using software the measured resistance was directly transformed into the corresponding temperature. This way it was possible to follow the temperatures simultaneously while measuring.

The accuracy of the sensors is ±0.2°C. All the sensors were calibrated before the experiments.

Figure 3.20: Craftemp Astra thermistor (left) and data acquisition unit Agilent (right).

Thermal manikins

The two thermal manikins used for the experiments and named “Kirsten” and “Pernille” are 1.7 m in height and are shaped to resemble accurately the body of an average Scandinavian woman. The manikins bodies are made of a 0.003 m fiberglass coated polystyrene shell. Kirsten consists of 23 body segments, while Pernille consists of 17 body segments. Each of these segments is equipped with heating and temperature measuring wiring controlled by a computer program so as to maintain a surface temperature equal to the skin temperature of a person in a state of thermal comfort at the actual activity level, and thus realistically to recreate the free convection flow surrounding the human body [1]. The control of the manikin is described by Tanabe et al. (1994) [4].

List of all manikins` all body segments for both Kirsten and Pernille are listed in Table 3.3, the manikins are shown in Figure 3.21.

Table 3.3: Body segments of thermal manikins Kirsten and Pernille.

Kirsten	Pernille
Crown	Top of head
L. Face	Head
R. Face	
Back of neck	
L. Hand	L. Hand
R.Hand	R. Hand
L.Forearm	L. Forearm
R.Forearm	R. Forearm
L. Upper arm	L. Upper arm
R.Upper arm	R. Upper arm
L.Chest	Chest
R. Chest	
Back	Back
Back side	
Pelvis	Pelvis
R. Back Thigh	R. Thigh
R. Front thigh	
L. Back thigh	L. Thigh
L. Front thigh	
R.Low.Leg	R. Low leg
L.Low.Leg	L. Low leg
R.Foot	R. Foot
L.Foot	L. Foot

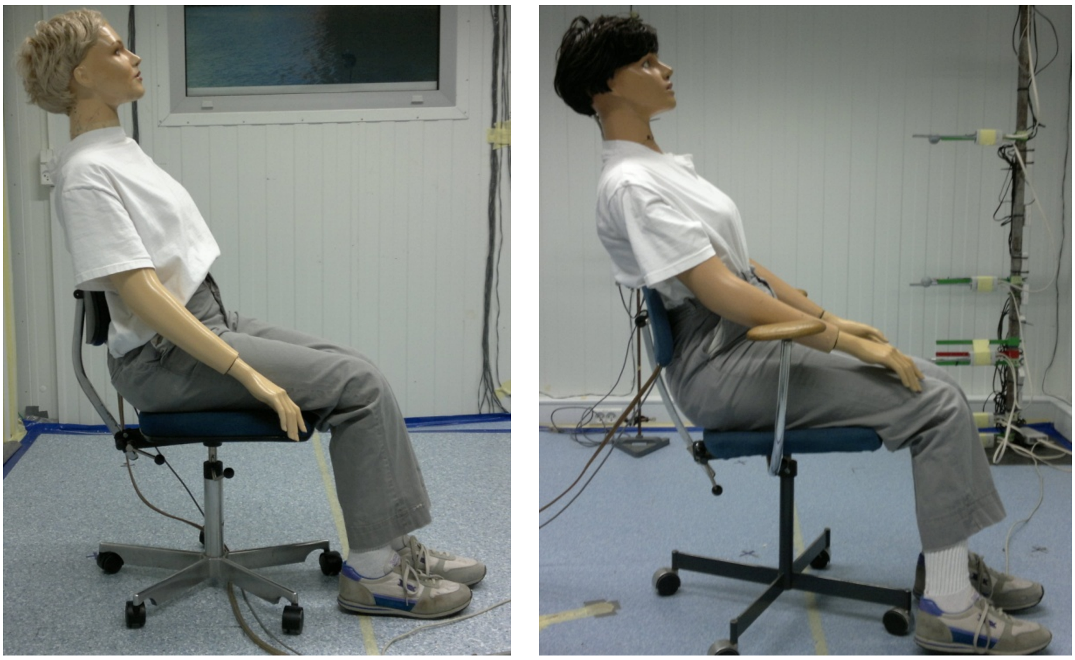


Figure 3.21: Thermal manikins Kirsten (left) and Pernille (right).

Radiant asymmetry

The radiant temperature asymmetry was measured by a sensor made by Brue and Kjaer. The sensor has two sides A and B. The radiant asymmetry is read as the difference between the radiant temperature of side A and side B (A minus B). The value was shown by a Brue and Kjaer indoor climate analyzer 1213 to which the sensor was connected. Both the sensor and the indoor climate analyzer can be seen in Figure 3.22.

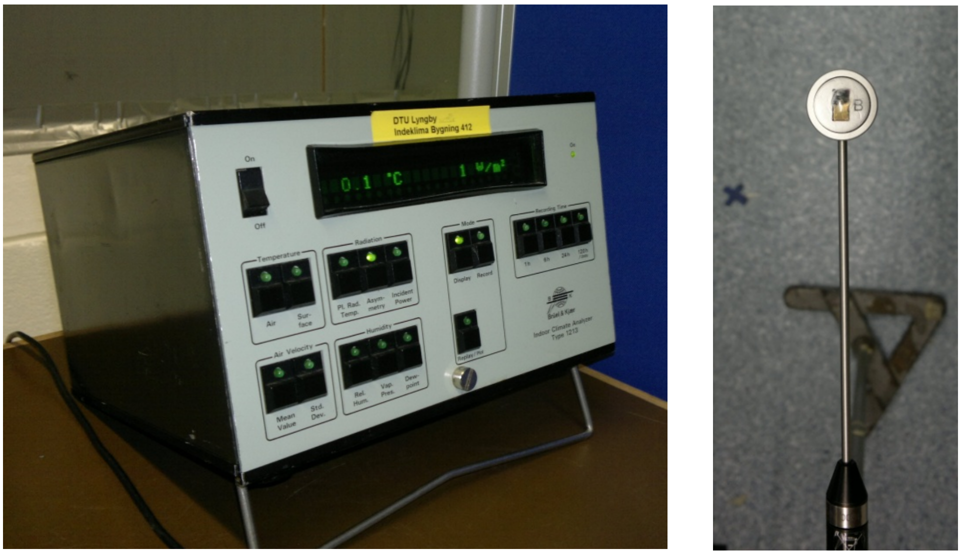


Figure 3.22: B&K indoor climate analyzer (left) and radiant asymmetry sensor (right).

Air flow

To control and observe the air flow rate in supply and exhaust, the pressure difference was measured. A Micromanometer FCO510 was connected to the Micatrone Flow sensors and Halton PRA balancing devices (7, 8, 12, and 13 in Figure 3.3) and the ΔP (Pa) was read (Figure 3.23).

Figure 3.23: Measuring of the pressure difference.

When using the Micatrone Flow sensor (for supply to the chilled beam), the air flow rate is calculated according to the formula given by the producer:

$$V = A \times k_m \times \sqrt{\Delta P}\,, [l/s]$$

Where:

V – volume flow rate (l/s);

A – cross section of the duct (m^2);

k_m – coefficient depending on the diameter of the duct (specified by the producer);

ΔP – pressure difference (Pa).

When using the Halton PRA balancing device, the air flow rate is calculated according to the formula:

$$V = k \times \sqrt{\Delta P}\,, [l/s]$$

Where:

k – coefficient depending on the turns of the balancing device`s knob, (specified by the producer).

Water flow

To control and be able to measure the flow rate of the water in the supply line a balance valve Tour & Anderson (TA STAD 15/14) was installed. The valve has differential pressure connections where a Computer Balancing Instrument TA II was connected and the water flow rate was calculated based on the pressure difference generated by the valve.

3.4 Procedure

Monitoring of the operating conditions

Before any measurements were started it was ensured that all the operating conditions (described in Chapter 3.2, Table 3.2) of the systems were right for the particular case. This includes the supply and exhaust air flow and supply air temperature, water supply mass flow and water supply temperature, artificial window surface temperature and operative and air temperature at the reference points in the room.

Supply and exhaust air flow and supply water mass flow was measured as described in Chapter 3.3.

Air, water and artificial window surface temperature was measured by the thermistor sensors (described in Chapter 3.3) located at the corresponding places.

To be sure that the target temperature of 26°C in the chamber was achieved the reference operative temperature was observed at Reference point 1 (between measuring points 2 and 3) for room set-up 2 and set-up 4, and at reference point 2 (between measuring points 3 and 4) for room set-up 1 and set-up 3. The purpose was to follow the reference air temperature, which was less affected from the manikin. However, for results and analysis only reference temperature measured at Reference point 2 was used since the temperatures did not differ significantly (0.2°C or 0.8%).

It was also checked whether the thermal manikins are working properly and are in steady state.

It was checked whether the direct solar load on the floor was simulated or not (on or off) depending on the case studied. Lap-tops in the chamber were checked to be sure that they were working properly and the total heat load in the room was correct.

If all the conditions were fulfilled the physical measurements were started.

Measurements of air velocity, air temperature and operative temperature

Measurements of operative temperature, air temperature and air velocity at 8 different heights were performed at one measuring point at a time. Logging time for the velocity and air temperature measurements (measurements with anemometer) was 5 minutes and averaging time was also 5 minutes. It means that every 5 minutes the mean velocity and mean air temperature were calculated and recorded. As a result, one velocity value and one air temperature value were acquired at each height, which was the mean velocity and mean air temperature for the 5-minute logging. Then the stand was moved to the next measuring point. After waiting for 5 min the next measurement was initiated. This was needed in order to make sure that the airflow pattern in the chamber was not affected by the movement of the stand to the next location.

Operative temperature was measured during all points without interruption. When analyzing the operative temperature, only values from the logging time of the anemometer were used. The sampling rate was 30 seconds. Measuring process can be seen in Figure 3.24.

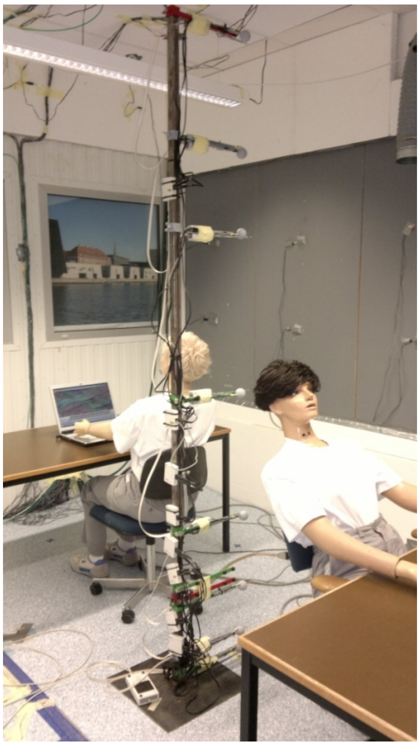


Figure 3.24: Changing the measuring point after logging of 5 minutes.

Measurements of surface temperature

All the thermistors for surface and air temperature measurements were also measuring all the time and then the data was processed as already explained above. Sampling rate of the thermistors was 10 seconds. In order to achieve more precise measurements of the surface temperatures, a thermal paste and insulation strip was used to improve the conductivity and eliminate the air temperature impact, Figure 3.25.

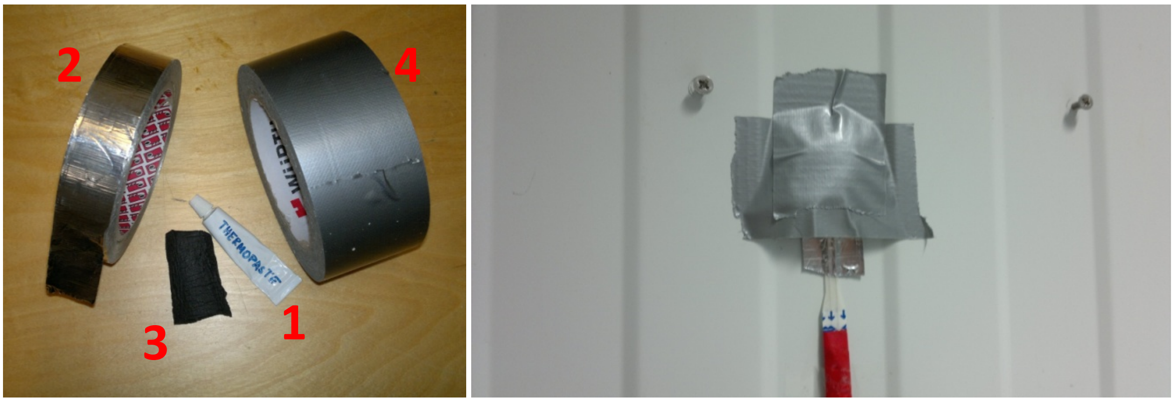


Figure 3.25: Components used to improve the conductivity and accuracy of the surface measurements (1 – thermal paste, 2 - silver tape, 3 – insulation strip, 4 – duct tape).

Measurements with thermal manikins

The purpose of the thermal manikins was to simulate occupants in the office room. Both of the manikins were dressed with typical summer clothes – light shoes, cotton socks, panties, light cotton trousers and a T-shirt. The overall insulation of clothes was 0.5 Clo for each manikin. Both of the manikins had short-hair wigs.

The parameter obtained from the manikins was the manikin based equivalent temperature. The manikin based equivalent temperature is the temperature of a uniform enclosure (without radiation heat exchange and under low velocity of air, v_a<0.05 m/s) in which a thermal manikin with realistic skin surface temperature would lose heat at the same rate as it would in the actual environment - Tanabe et al. (1994) [4].

For this purpose both of the manikins were calibrated in a special climate chamber at the research facility prior to all the measurements. The uniform environment in the chamber was achieved by the piston flow supply from the whole floor area. To avoid vertical velocities, a plywood plates were placed on the floor just below the manikins to avoid upward air movement. The chamber walls consisted of vinyl fabric and air space in front of the real walls. Air low was supplied between the vinyl fabric and the wall to eliminate the effect of radiation due to temperature difference between the walls and the air. A partition was placed between the manikins to avoid the radiation interaction between them.

The manikins were calibrated at five different temperatures – 18, 21, 24, 27 and 30°C. The temperature in the chamber was measured using a mercury thermometer with an accuracy of ±0.1°C.

During the calibration both manikins were dressed identically as during the measurements.

Both of the manikins were logged during all the measuring points. The sampling rate was 10 seconds.

The equivalent temperature was calculated according to the equation:

$$Q = h \times (t_s - t_{eq}) \quad \rightarrow \quad t_{eq} = t_s - \frac{Q}{h}, [°C]$$

Where:

Q – measured mean heat flux (W/m^2);

h – heat transfer coefficient found from the calibration (W/(m^2·K);

t_s – measured mean surface temperature (°C);

t_{eq} – equivalent temperature (°C).

Measurements of radiant asymmetry

After all the points in the chamber were measured, logging of the manikins was disabled and the radiant asymmetry measured. The radiant asymmetry was measured at the locations where the manikins were placed during that case. Measurements were performed at heights 0.6 m and 1.1 m and in 3 directions, Table 3.4 and Figure 3.26.

Table 3.4: Side order of the radiant asymmetry measurements for all studied cases.

Window A	Door B
Climate Square A	Legoland B
Floor A	Ceiling B

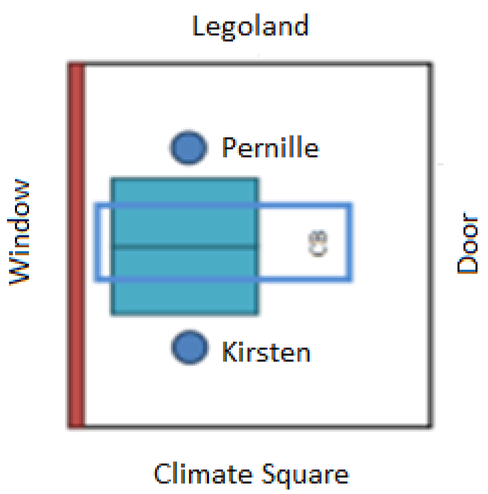


Figure 3.26: Legend of the sides for radiant asymmetry measurements.

First, one of the manikins was moved away, but remained in the room, and the stand with the radiant asymmetry sensor fixed at the necessary height was placed, Figure 3.27.

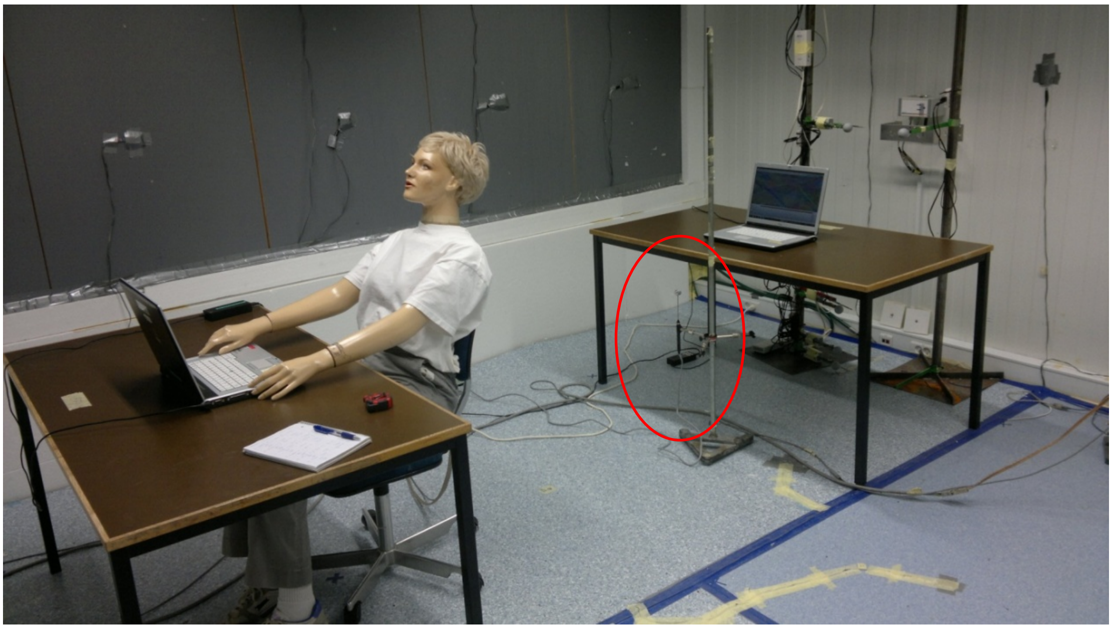


Figure 3.27: Measurements of the radiant asymmetry.

The radiant asymmetry was measured for 2-3 minutes. Then the direction was changed. After waiting for 2-3 minutes the new measurement was made. This was done at both manikins' positions, for the 2 heights and all 3 directions on each height, giving in total 12 measurements per case.

Transition to the next case

One or two cases were measured per day. If two cases were measured then both of the cases were of the same heat load and with the same system. The only change was the room set-up. Before measuring the second case it was left for at least 1 hour in order to achieve steady state. The criterion for steady state was stable air temperature at both reference points.

If the heat load or ventilation system had to be changed then the new conditions were left for at least one night before the next measurements.

Before initiating any measurements all the operating conditions were observed as described in Chapter 3.4.

3.5 Data Analysis

Normalized temperature

Because of the differences in reference point temperatures between the cases, a normalized temperature was used in order to be able to compare the results. The normalized temperature is a dimensionless parameter, which is calculated as the average temperature from all the measured points at a particular height divided by the mean reference point temperature for that particular case. It is a ratio which shows how uniform environment the system is able to provide in the room – the closer the ratio to value of 1, the more stable is the system.

Normalized air temperature (or air temperature factor TF_a), normalized operative temperature (TF_o) and air velocity in height will be shown for each room set-up comparing between all four systems tested and for each system comparing among all four set-ups tested in Chapter 4.

4. Results

The results are shown with the following abbreviations:

Systems:

CB – chilled beam

CBR – chilled beam with radiant panels

CCMV – chilled ceiling combined with mixing ventilation

MTVV (MV) – total volume mixing ventilation

Room set-ups:

S1 – set-up 1

S2 – set-up 2

S3 – set-up 3

S4 – set-up 4

Heat loads:

H – high (design) heat load

L – low (usual) heat load

4.1. Thermal environment

Reference temperatures

The target temperature in the room was 26°C. This was controlled by measuring the air temperature in the reference point. The mean reference point air temperatures for all 32 cases measured are listed in Table 4.1.

Table 4.1: Mean reference air temperature at 1.1 m between measuring points 3 and 4.

System	Heat load	Set-up	Mean reference air temperature (°C)	STD	Min	Max
MV	Design (H)	1	25,8	0,06	25,7	25,9
		2	25,9	0,07	25,8	26
		3	25,9	0,06	25,8	26
		4	26,0	0,04	25,8	26
	Usual (L)	1	25,8	0,06	25,7	25,9
		2	26,0	0,06	25,8	26
		3	25,9	0,02	25,9	26
		4	25,8	0,04	25,7	25,8
CCMV	Design (H)	1	26,0	0,05	25,9	26,1
		2	26,2	0,04	26,1	26,3
		3	25,9	0,03	25,9	26
		4	25,8	0,04	25,7	25,8
	Usual (L)	1	26,0	0,03	25,9	26
		2	26,3	0,02	26,3	26,3
		3	26,2	0,03	26,2	26,3
		4	26,1	0,06	26	26,2
CBR	Design (H)	1	26,3	0,03	26,3	26,4
		2	26,6 (between 2 and 3)	0,16	26,4	26,8
		3	26,3	0,08	26,1	26,4
		4	26,2 (between 2 and 3)	0,08	26,4	26,7
	Usual (L)	1	26,0	0,07	25,9	26,1
		2	26,2	0,07	26,1	26,3
		3	25,9	0,03	25,8	26
		4	26,0	0,04	25,9	26,1
CB	Design (H)	1	25,7	0,07	25,6	25,9
		2	26,4	0,03	26,4	26,4
		3	25,6	0,09	25,4	25,7
		4	26,3	0,27	25,8	26,5
	Usual (L)	1	26,0	0,06	25,9	26,1
		2	25,8	0,02	25,8	25,9
		3	26,0	0,07	25,9	26,1
		4	26,2	0,03	26,2	26,3

As mentioned before, only reference temperature between the measuring points 3 and 4 is used for the analysis. The mean reference air temperature is obtained as the mean temperature from all the logging time intervals (5 min logging for each measuring point).

It can be seen that reference point temperatures are close to 26°C. Although there is some variation, the reference point temperature during each case was kept stable as the standard deviations show, Table 4.1.

Air temperature

The Figure 4.1 and Figure 4.2 show that only between heights 1.7 m and 2.4 m the air temperature is higher than the reference point temperature. The reason for that could be the lighting which makes the average air temperature at that height higher.

A stable system seems to be chilled ceiling combined with mixing ventilation because in most of the set-ups at both design and usual heat load the normalized temperatures are relatively close to 1. This can be explained with the fact that the chilled ceiling covered 75% of the whole ceiling so it provided more homogeneous cooling of the room.

Chilled beam appears to be not very stable - at design heat load for set-up 2 and set-up 4 particularly (Figure 4.3). The air temperature in the room is lower than in the reference point (TF_a less than 1). This can be because the air supplied by the chilled beam drops faster compared to the CBR, CCMV and MTVV, i.e. the air "drops" down before reaching the wall where the reference point was set (see Appendix 2).

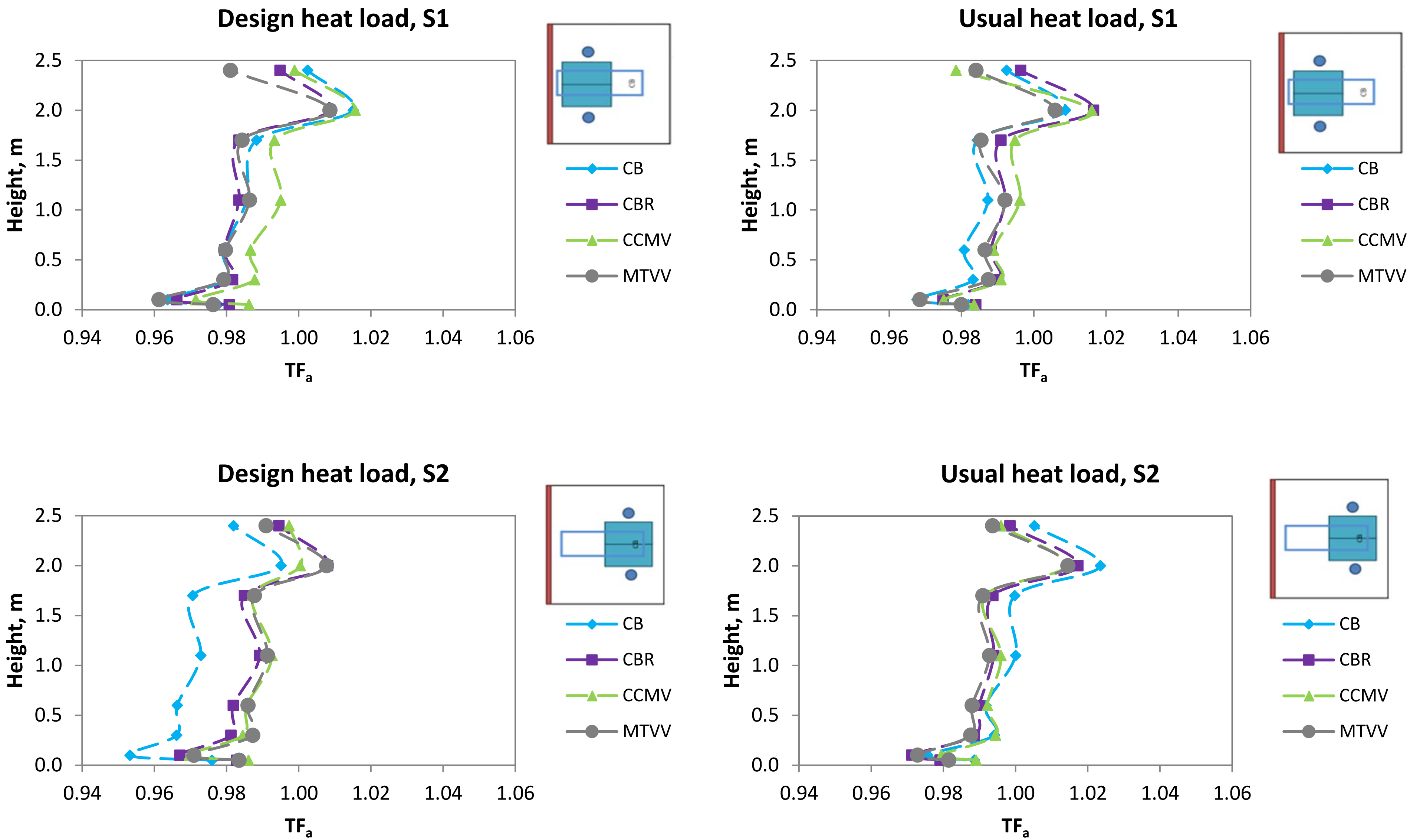


Figure 4.1: Air temperature factor (normalized temperature) in height of set-up 1 and 2 at design and usual heat load.

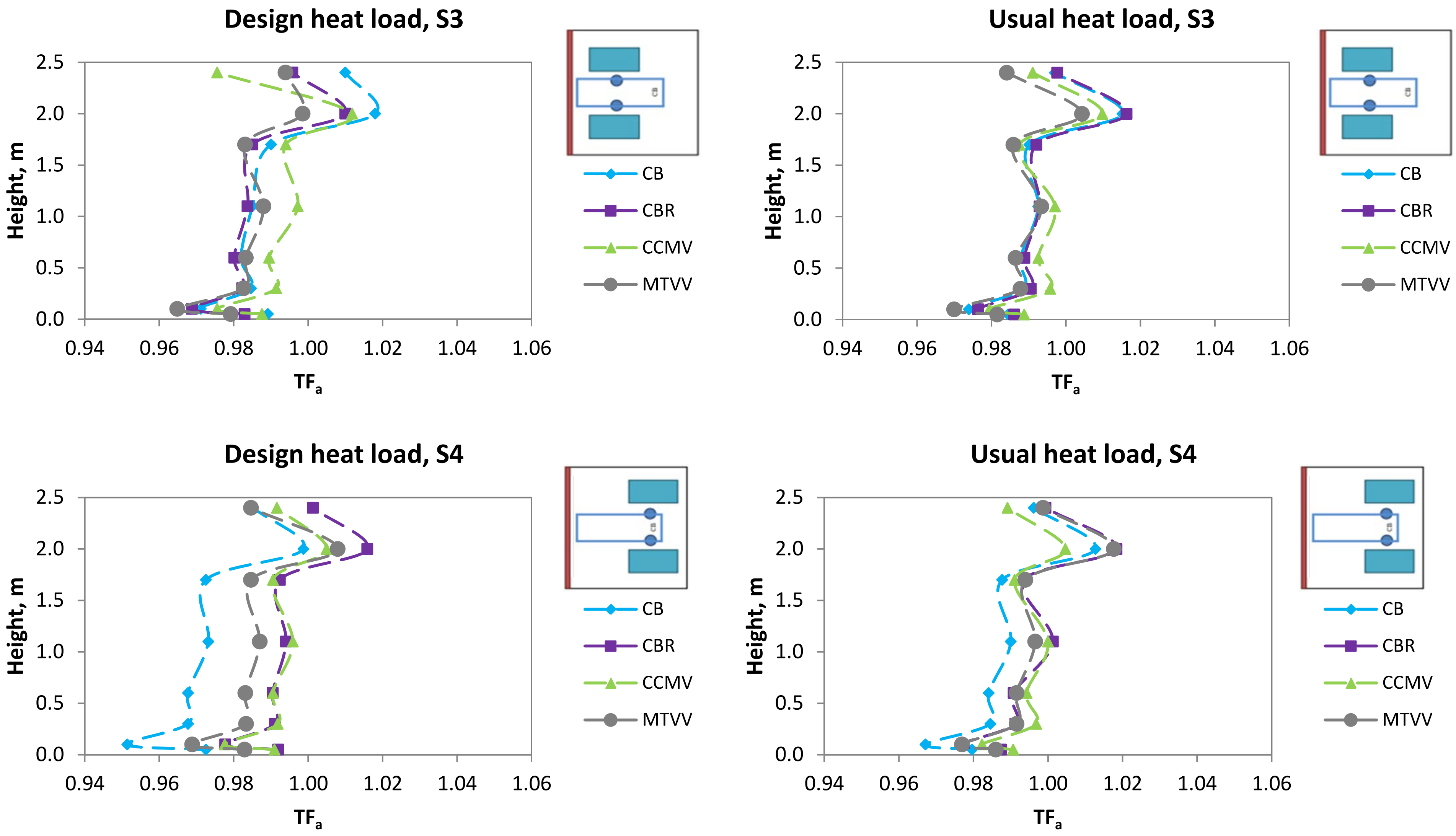


Figure 4.2: Air temperature factor (normalized temperature) in height of set-up 3 and 4 at design and usual heat load.

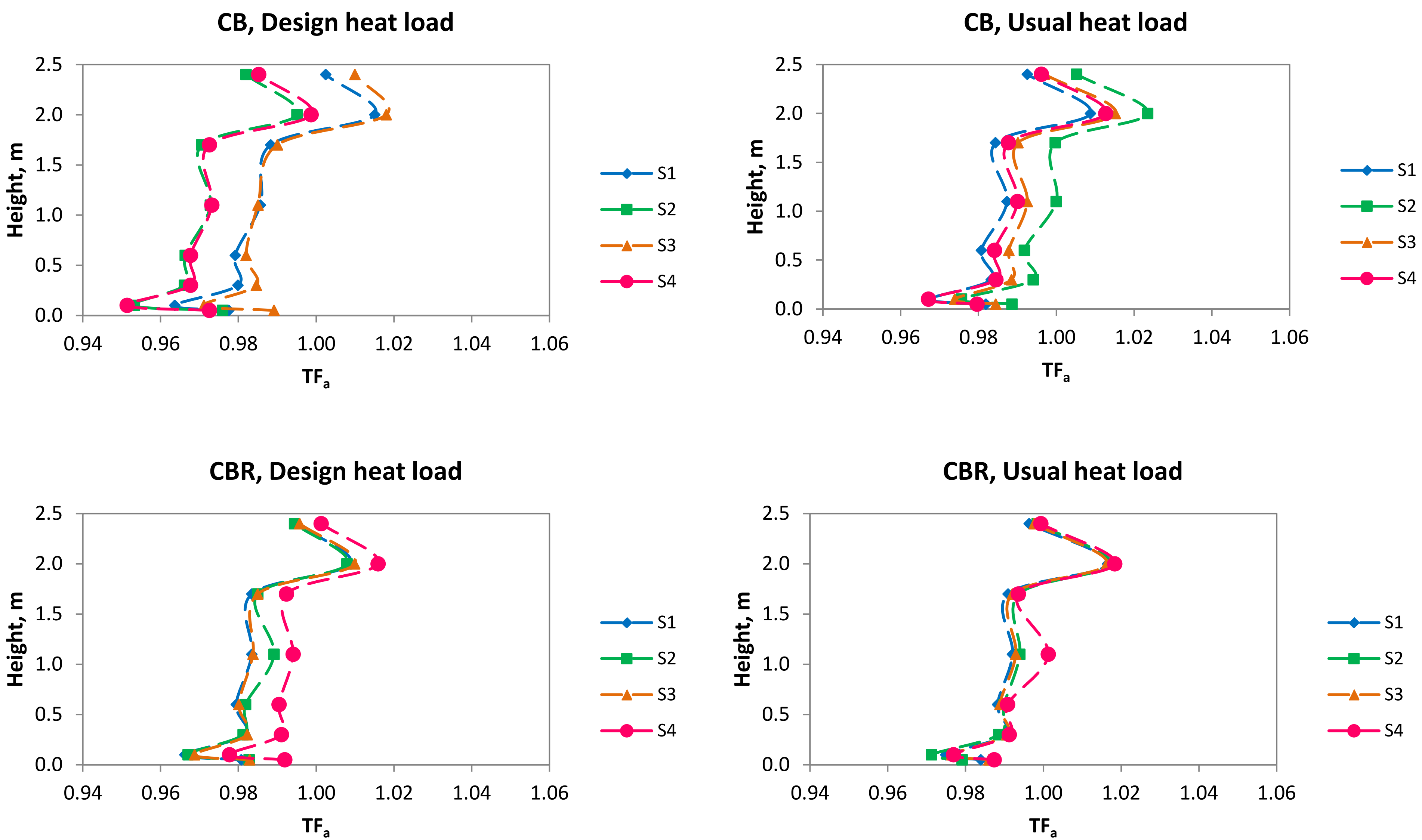


Figure 4.3: Air temperature factor (normalized temperature) in height of CB and CBR at design and usual heat load.

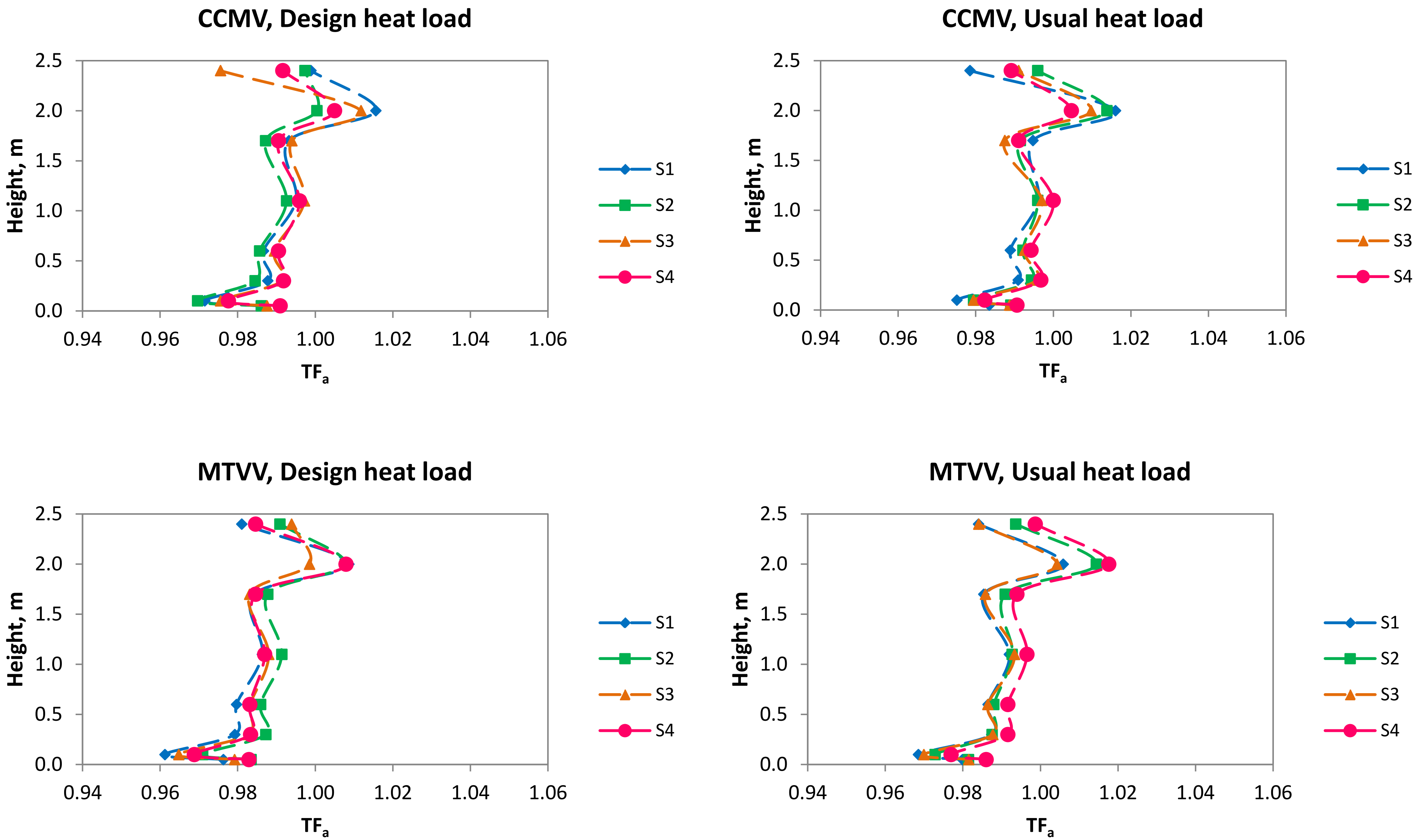


Figure 4.4: Air temperature factor (normalized temperature) in height of CCMV and MTVV (mixing total volume ventilation) at design and usual heat load.

Operative temperature

In Figure 4.5 and Figure 4.6 it can be seen that chilled ceiling combined with mixing ventilation (CCMV) at design heat load for set-up 2, 3 and 4 (also set-up 1 but less noticeable) provides lower operative temperature throughout the room than in the reference point. In fact, the operative temperature decreases in height. The reason is the radiant cooling from the chilled ceiling panels – the higher from the floor (closer to the panels) the more significant is the cooling effect. In these results the average temperature from all measuring points is taken into account and since chilled ceiling is across the whole ceiling, the cooling effect can be clearly seen.

However, Figure 4.8 shows that this tendency is present with the high (design) heat load only. With the low or usual heat load the operative temperature is kept very stable throughout the room by all systems. It could mean that with a high heat load to be removed, the cooling effect becomes more significant with the CCMV than when usual heat load has to be removed.

A decrease of the operative temperature with the height is not observed with CBR at any case. The reason for that could be that as mentioned before, average temperature from all measuring points is taken into account. The radiant cooling from CBR has an effect for several measuring points only and not all of them (see Appendix 3). The impact could not be great enough to affect the average value.

Mostly all systems keep the operative temperature stable ad close to the reference value of 26°C, especially at the low (usual) heat load.

Figure 4.7 and 4.8 show that room set-up has an effect on the systems' performance at some cases. Chilled ceiling combined with mixing ventilation is able to decrease the operative temperature better at set-up 3 and set-up 4 when the two manikins are sitting close together with backs to each other. This way a strong heat flow is generated which is successfully removed by the CCMV. Again CCMV performs efficiently when greater amount of het has to be removed.

According to the ISO Standard 7730 (2005) [2] vertical air (in fact operative) temperature difference between head and ankles (1.1 m and 0.1 m) for category B is recommended to be lower than 3°C in order to avoid local thermal discomfort. Under all 32 cases this requirement is fulfilled. The values can be seen in Appendix 5.

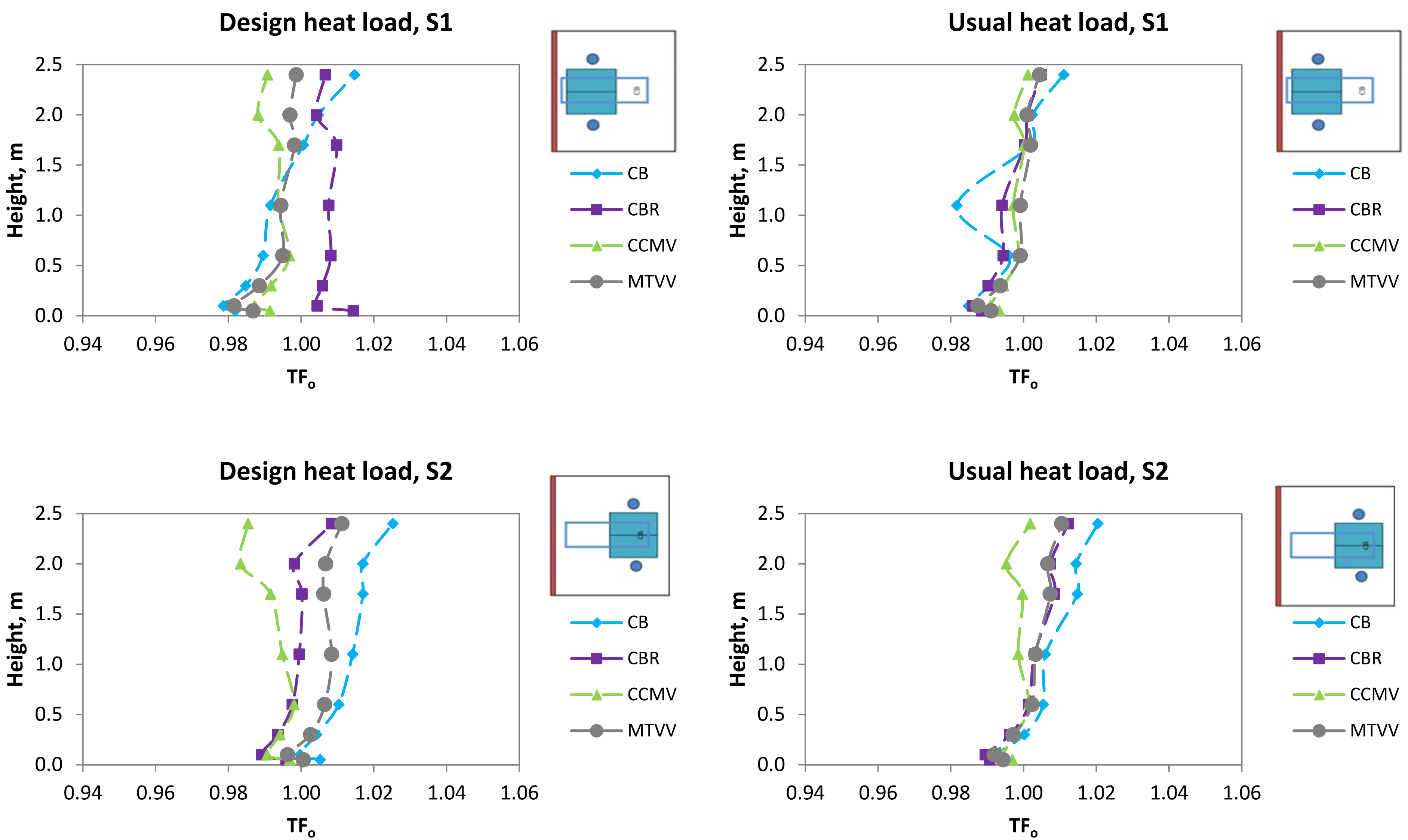


Figure 4.5: Operative temperature factor (normalized temperature) in height of set-up 1 and 2 at design and usual heat load.

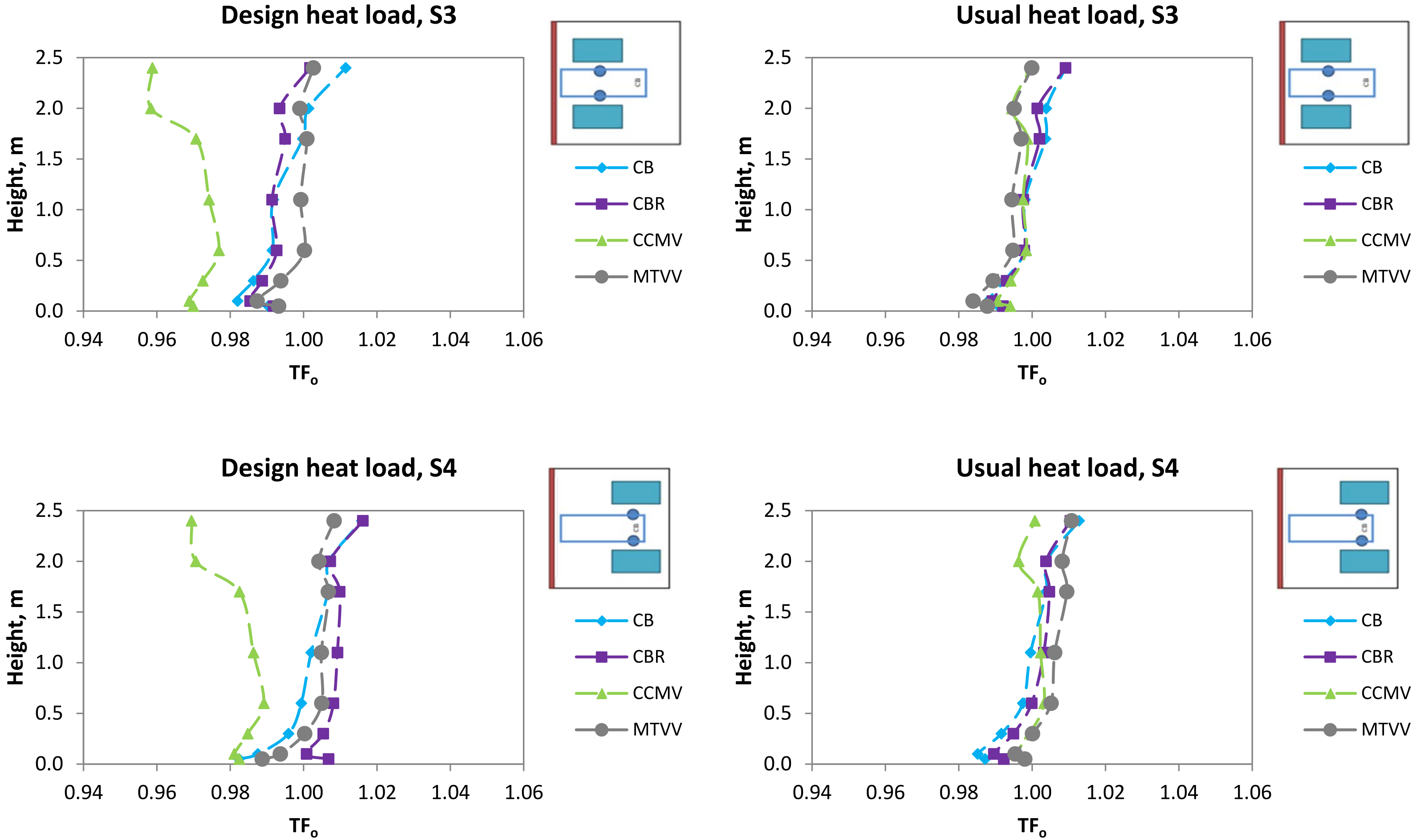


Figure 4.6: Operative temperature factor (normalized temperature) in height of set-up 3 and 4 at design and usual heat load.

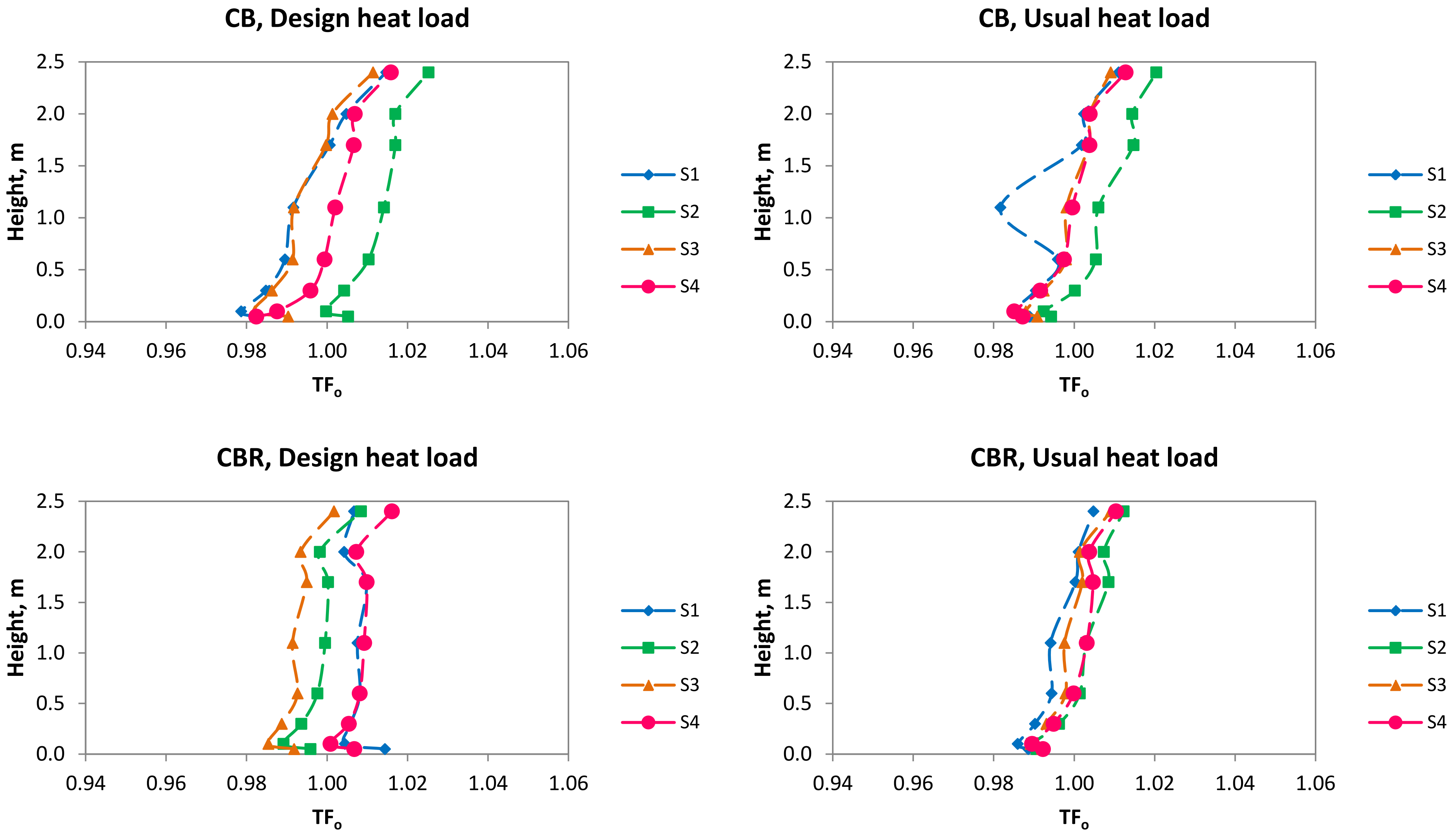


Figure 4.7: Operative temperature factor (normalized temperature) in height of CB and CBR at design and usual heat load.

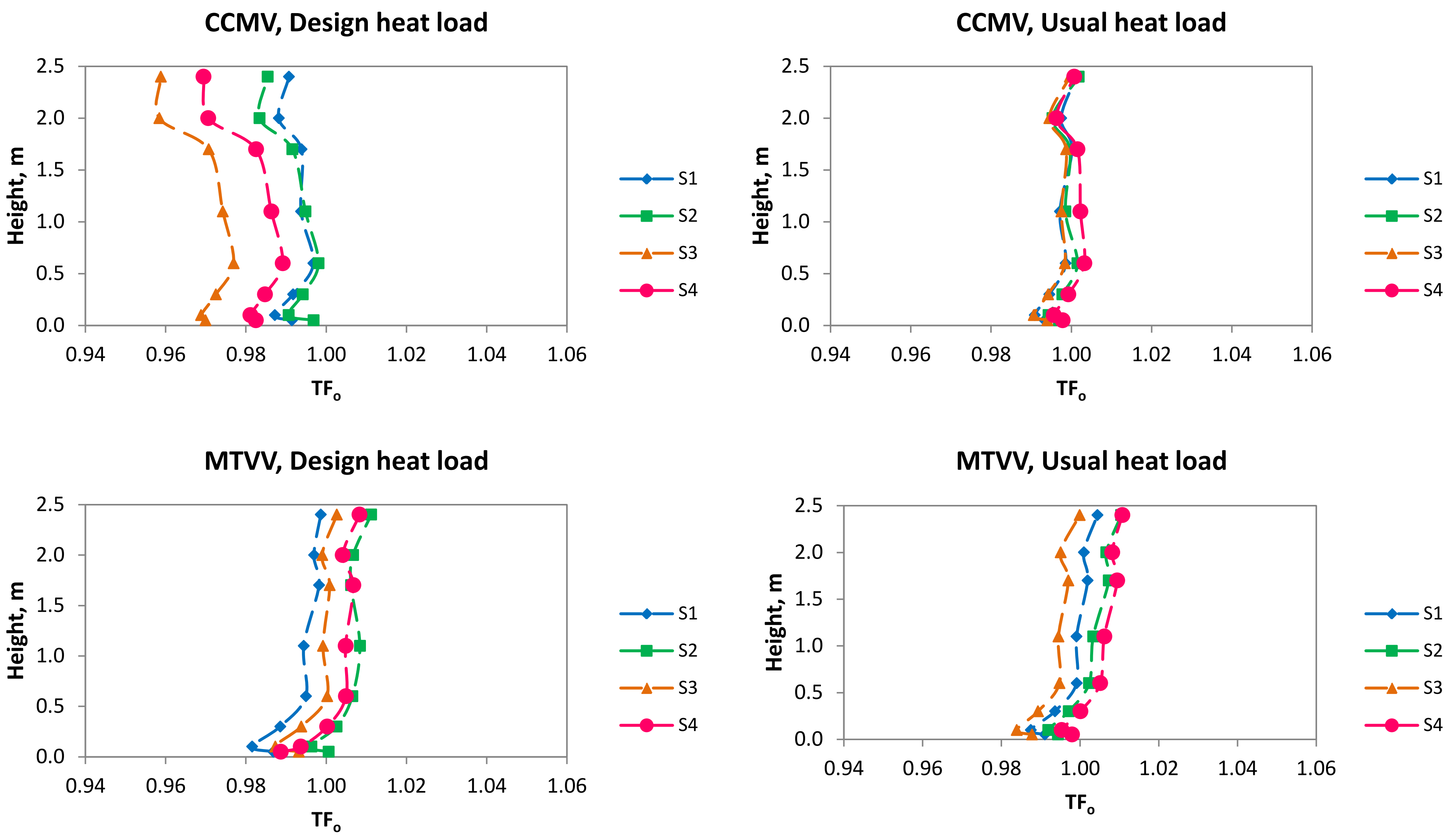


Figure 4.8: Operative temperature factor (normalized temperature) in height of CCMV and MTVV at design and usual heat load.

Air velocity and draught

Figure 4.9 and Figure 4.10 show that the highest air velocities are measured close to the floor at heights 0.05 m and 0.1 m but are all bellow the recommended limit of 0.2 m/s (ISO 7730, 2005). The air jet with a high momentum reaches the walls, falls down and glides over the floor with high velocities. Velocities are also slightly higher at 2 m. This is probably because of the supply air jets spread from the air terminal devices. Air velocities in the occupied zone are lower than 0.2 m/s, ISO 7730 (2005).

Slightly lower velocities are achieved with chilled ceiling combined with mixing ventilation (CCMV). Although the air supply flow rate is the same as for chilled beam and chilled beam with radiant panels (26 L/s, Table 3.2) air velocities are lower with CCMV at most of the cases. The reason could be that the air movement pattern in the room is better with CCMV. Only two linear slot diffusers were used with CCMV (the middle one was not used) which increases the momentum of the jets but also leaves some space for the supply air jets to spread and decrease in velocity, and not interact with other jets. Also because the jets are in contact with the chilled ceiling it makes the air to “drop” also before it has reached the walls to some extent. The radiant panels cover 75% of the ceiling area so this process occurs evenly and not drastically throughout the room. This is not the case with chilled beam – the air supply jets most probably reach the walls with high velocities, drops down and glide over the floor with relatively higher velocities.

The highest velocities are achieved with mixing ventilation (MTVV), which was expected because mixing ventilation performs cooling by providing cooler air in the room only. In order to remove the heat load, relatively high supply air flow rate is needed (55 L/s under low heat load and 90 L/s under high heat load while 26 L/s were needed for both heat loads for all the other systems, Table 3.2). The cross section area of the linear slot diffusers is small enough thus high volume air is supplied in the room with high velocity due to the small openings of the diffusers.

Room set-up has a little impact on the air velocity for all four systems, Figure 4.11 and Figure 4.12.

According to the ISO Standard 7730 [2] draught rate for category B, which this project is aiming for is recommended to be lower than 20%. Number of locations where draught rate is higher than 20% for every case is listed in Table 4.2. Draught rate only from 0.05 m to 1.7 m above the floor is taken into account since this is the occupied region where most people would be able to feel the draught.

Table 4.2: Number of points and percentage of all points where draught rate is higher than recommended (20%) from 0.05 m to 1.7 m above the floor.

System	Set- up 1				Set- up 2				Set- up 3				Set- up 4			
	H	%	L	%	H	%	L	%	H	%	L	%	H	%	L	%
CB	4	24	3	18	4	24	4	24	3	18	0	0	8	47	3	18
CBR	2	12	0	0	2	12	2	12	2	12	1	6	2	12	2	12
CCMV	1	6	0	0	0	0	0	0	1	6	0	0	0	0	0	0
MTVV	13	76	1	6	7	41	0	0	10	59	0	0	7	41	0	0

Where:

CB – chilled beam

CBR – chilled beam with radiant panels

CCMV – chilled ceiling combined with mixing ventilation

MTVV – mixing total volume ventilation

H – high (design) heat load

L – low (usual) heat load

Clearly draught rate is higher with the high heat load. The highest draught rate occurs with mixing ventilation alone and second highest with chilled beam. Explanation for the high draught rate for the mixing ventilation alone could be the same as for the high velocities, i.e. the high supply air flow rate. The supply air flow rate for the other systems were the same under high heat load as under low heat load. In order to remove the heat load, supply water temperature for these systems was changed – it was decreased under high heat load conditions. Therefore the air temperature supplied from the chilled beam (and CBR) to the room was lower under high heat load conditions than under low heat load conditions. Because the air was cooler, it could "drop" faster and create higher draught rate (increased local mean air velocity and decreased local air temperature).

With low (usual) heat load draught rates are significantly lower. However, chilled beam and chilled beam combined with radiant panels both have more points where draught rate is above the recommended 20% than mixing ventilation alone. Almost all of these points are by the walls but about half of them are right behind the manikins. It means that occupants might feel the draught.

Chilled ceiling combined with mixing ventilation does not cause any problems with draught rate as expected. The explanation could be the same as for the velocities mentioned above.

The maximum draught rate from 0.05 m to 1.7 m above the floor for each measured point for all cases can be seen in Appendix 6.

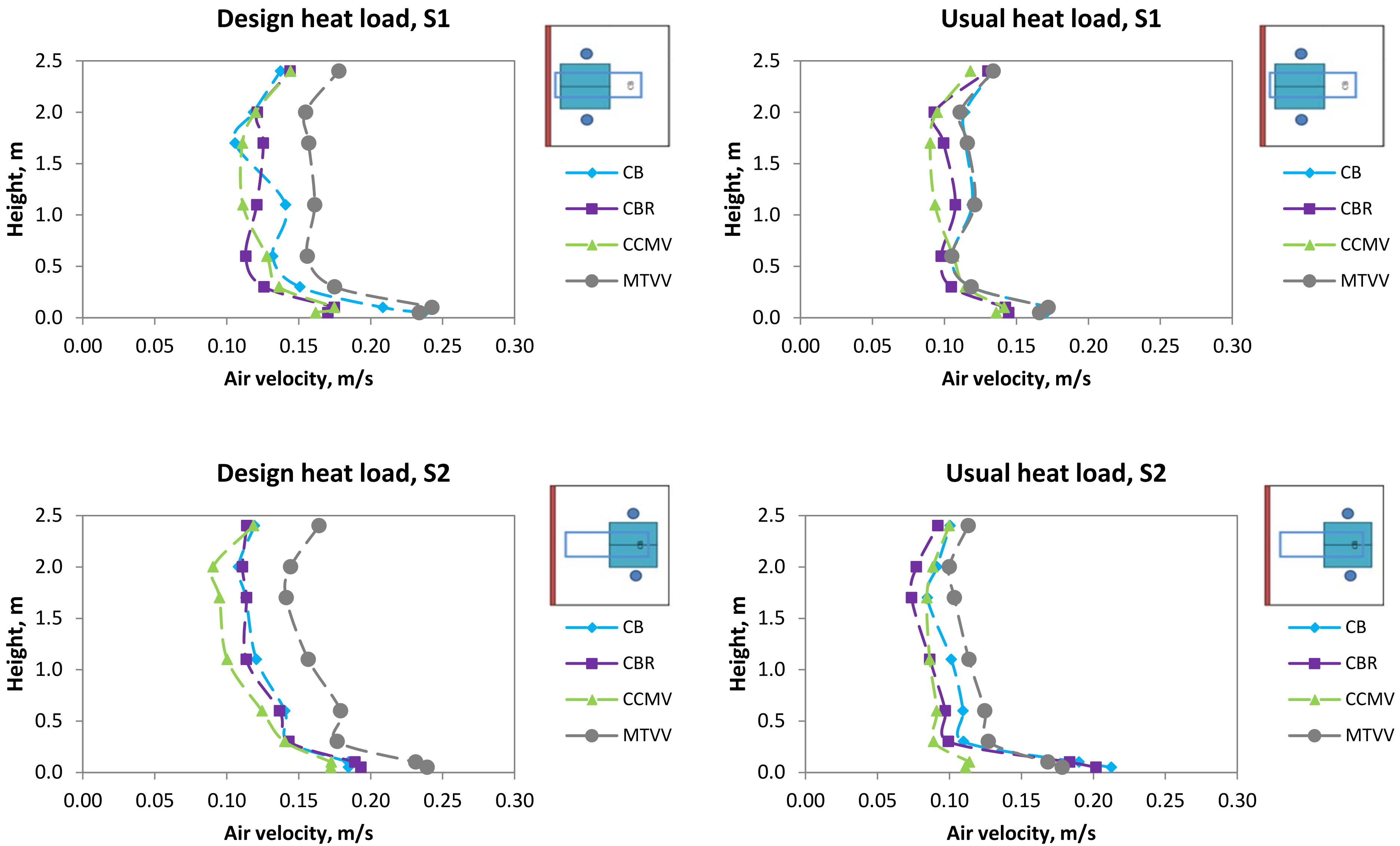


Figure 4.9: Air velocity in height of set-up 1 and 2 at design and usual heat load.

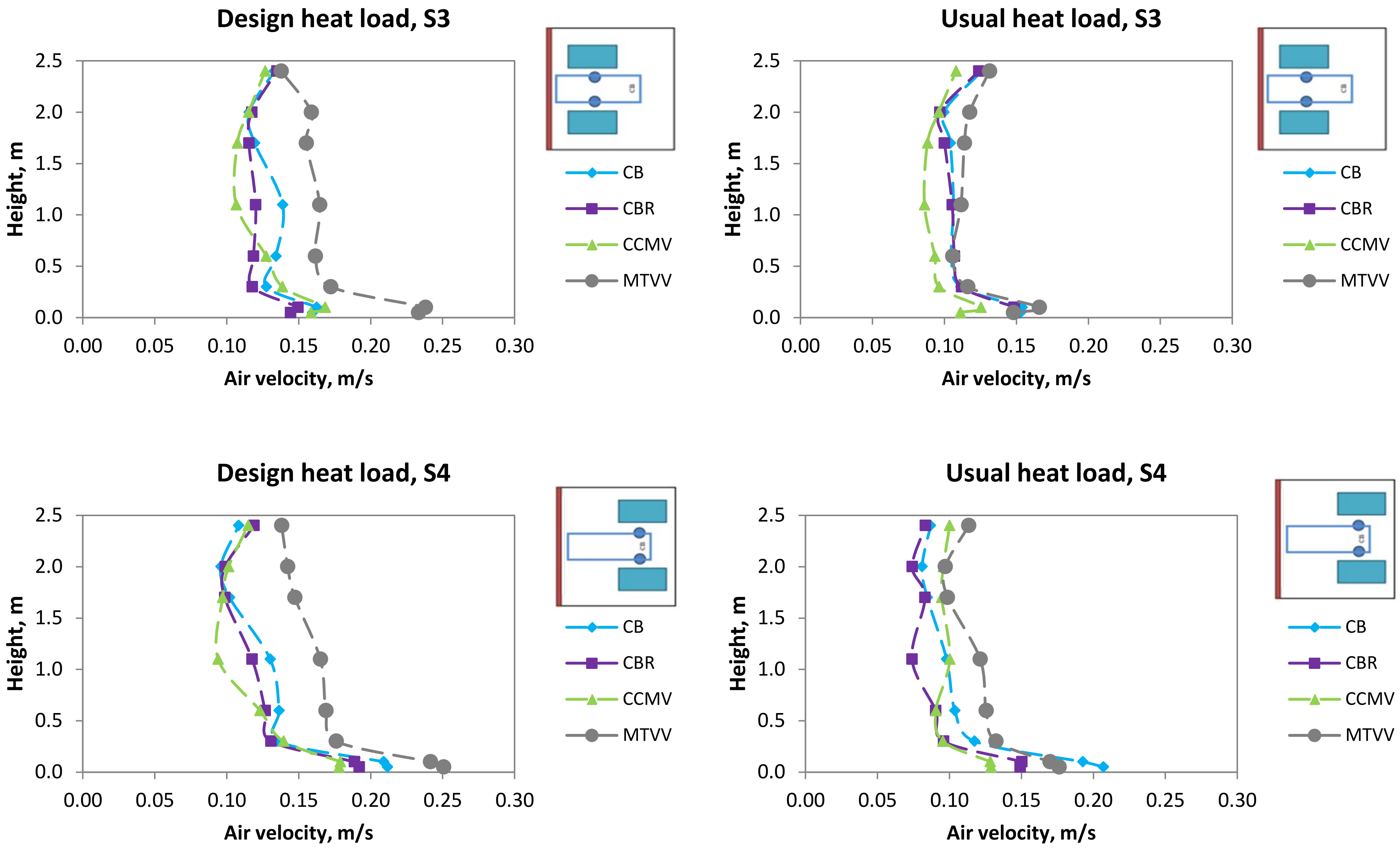


Figure 4.10: Air velocity in height of set-up 3 and 4 at design and usual heat load.

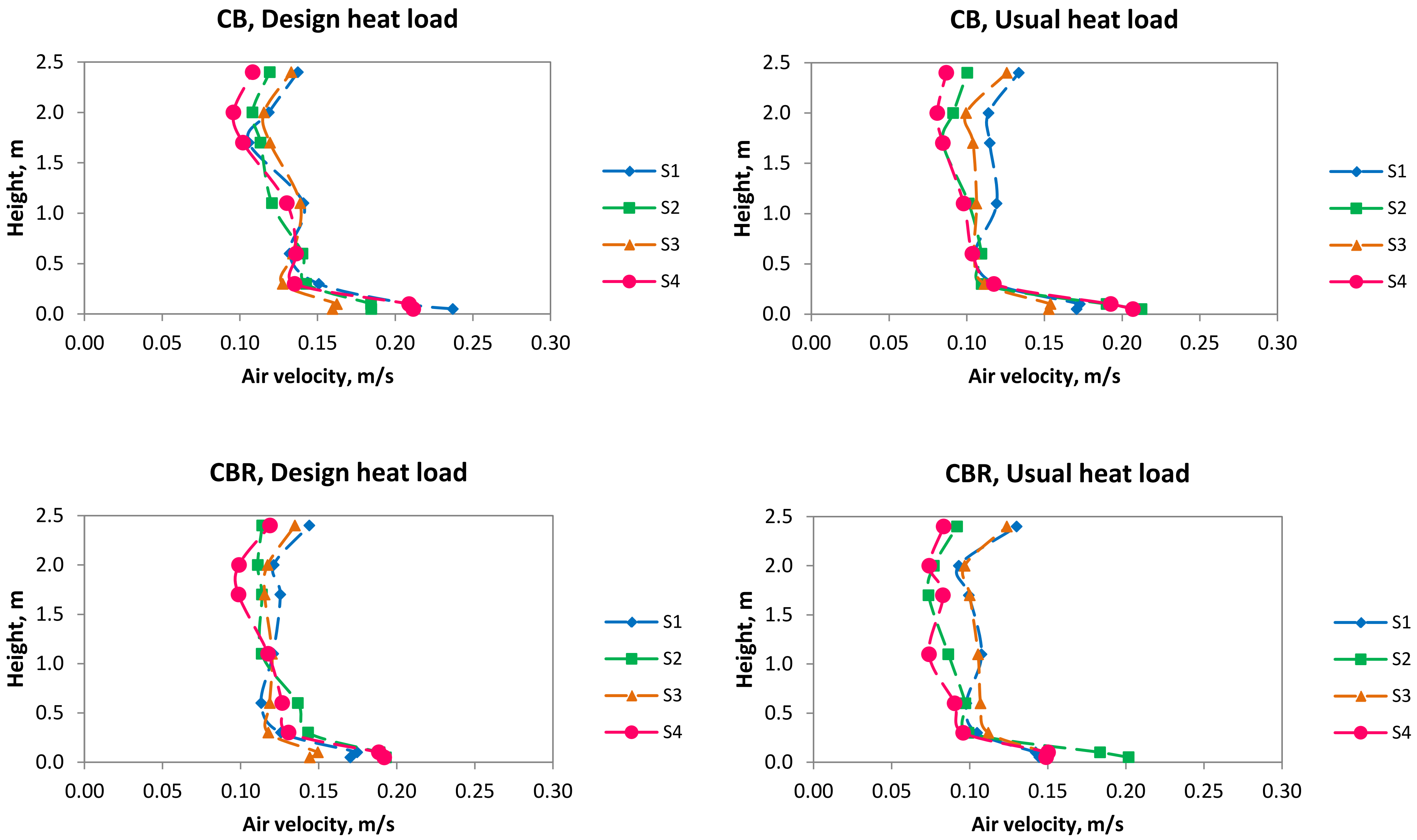


Figure 4.11: Air velocity in height of CB and CBR at design and usual heat load.

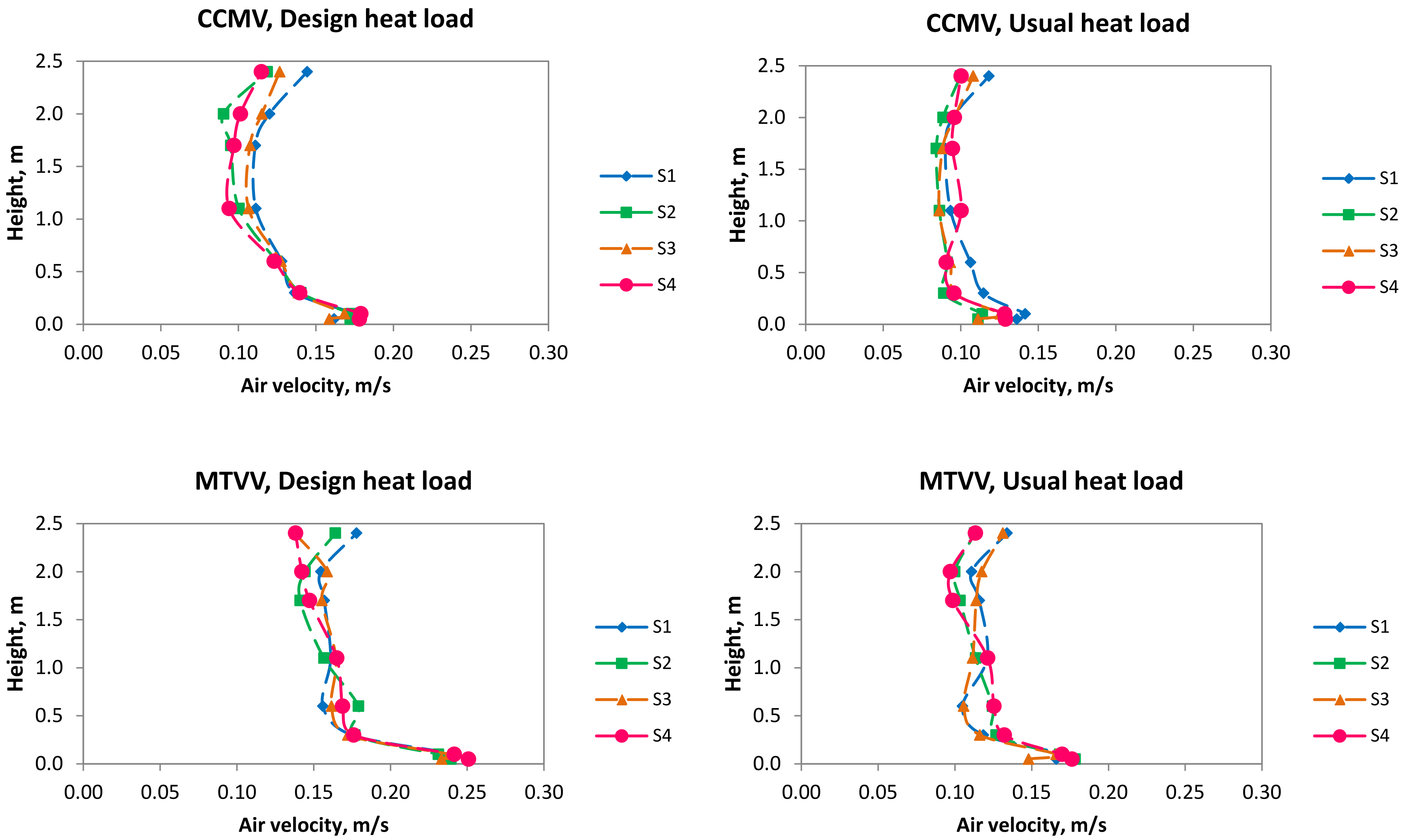


Figure 4.12: Air velocity in height of CCMV and MTVV (mixing total volume ventilation) at design and usual heat load.

The average values of air temperature, operative temperature and air velocity of all the measured points at every height can be seen in Appendix 1.

All measured values of air temperature, operative temperature and air velocity at every point and at all 8 heights for all 32 cases can be seen in Appendix 2, Appendix 3 and Appendix 4 respectively.

Maximum values of air temperature, operative temperature, air velocity and draught with information at which point and which height can be seen in Appendix 8.

4.2 Manikins

Cooling and warming effect

In order to see what is the temperature effect (cooling or warming) for each body part, the reference point temperature (the temperature measured between measuring points 3 and 4 for every case) was subtracted from the calculated equivalent temperature of each body part. If the difference is positive, it means that a person would feel warming at that particular body part. If the difference is negative, a cooling effect would be felt. The greater the difference, the greater is the effect.

The calculated manikin based equivalent temperature of all the body parts of both manikins at all cases can be seen in Appendix 7.

Hereafter figures of the temperature differences for all body parts for both manikins at all cases will be shown.

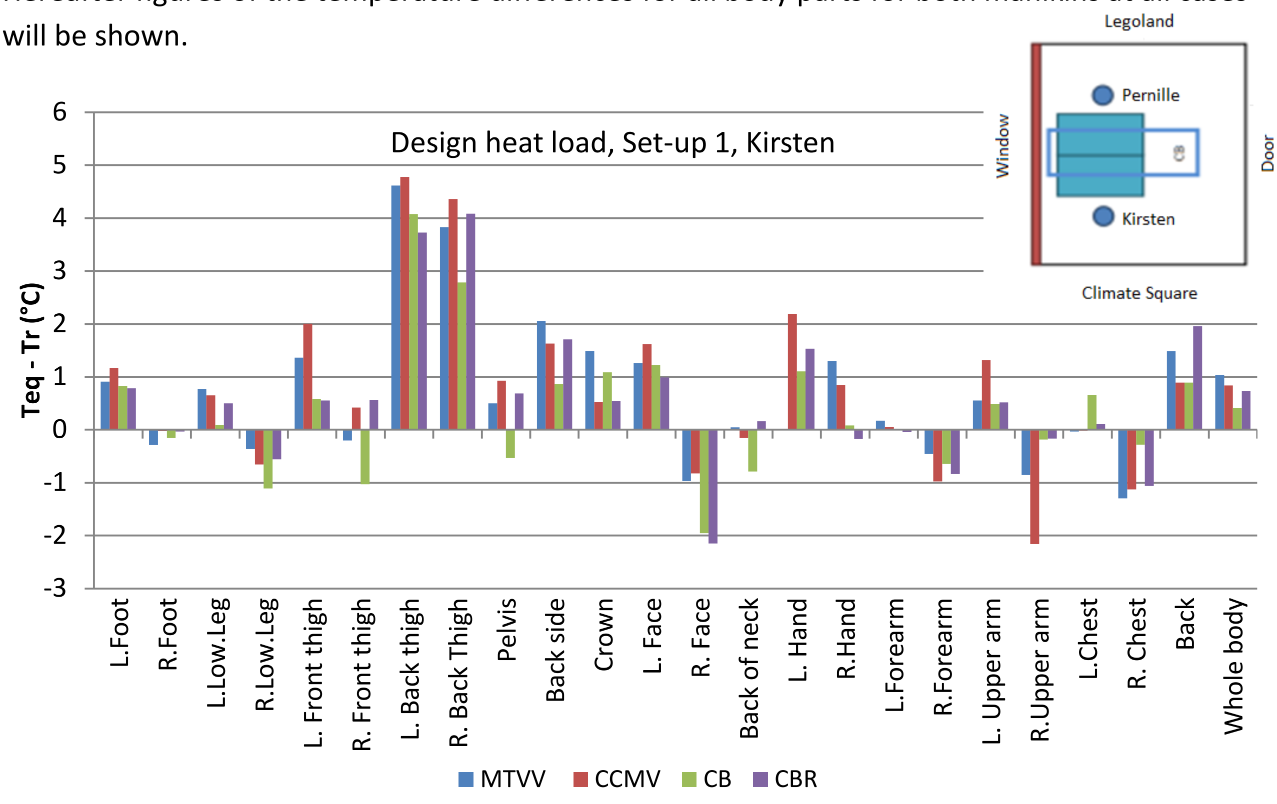


Figure 4.13: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 1 with design heat load.

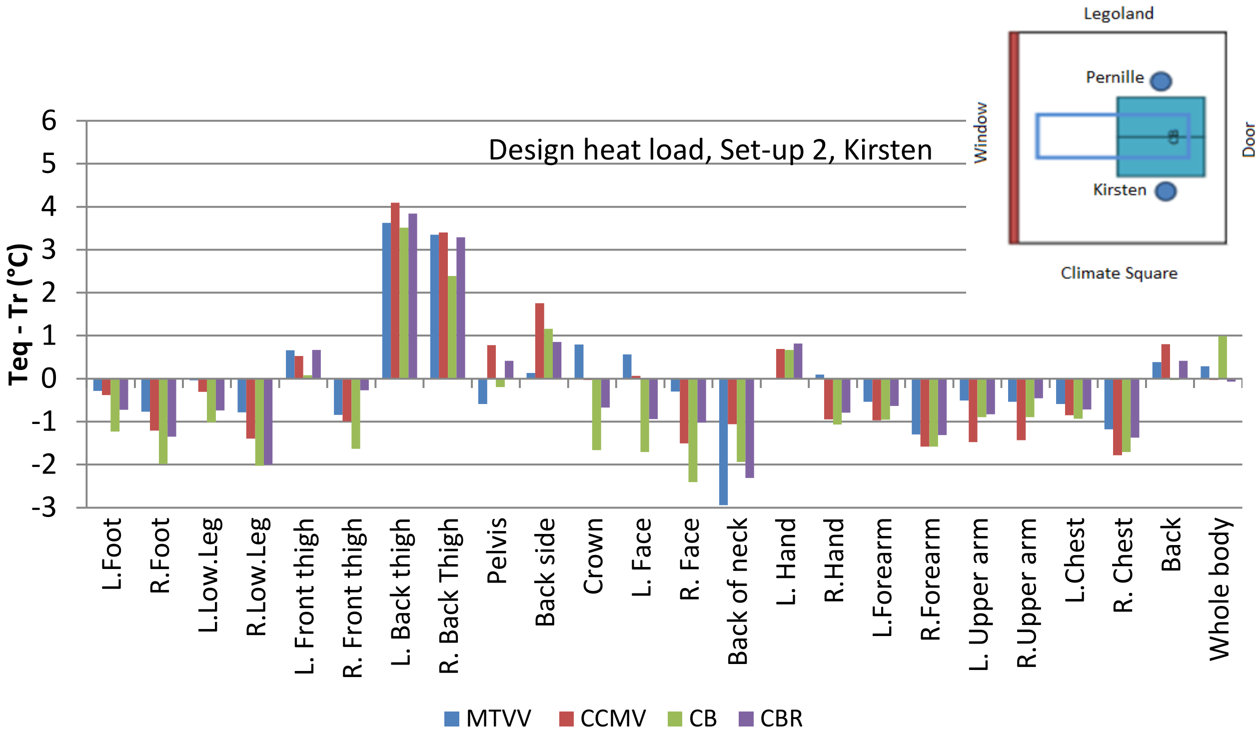


Figure 4.14: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 2 with design heat load.

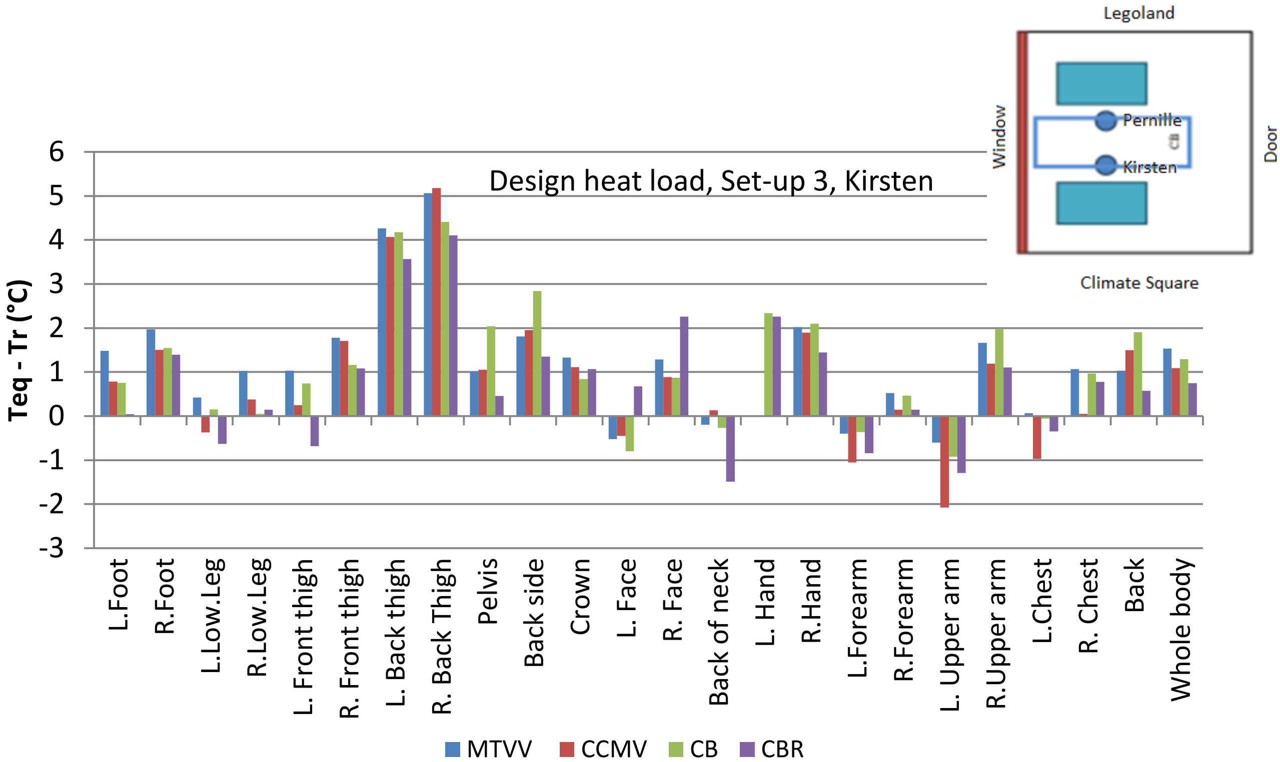


Figure 4.15: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 3 with design heat load.

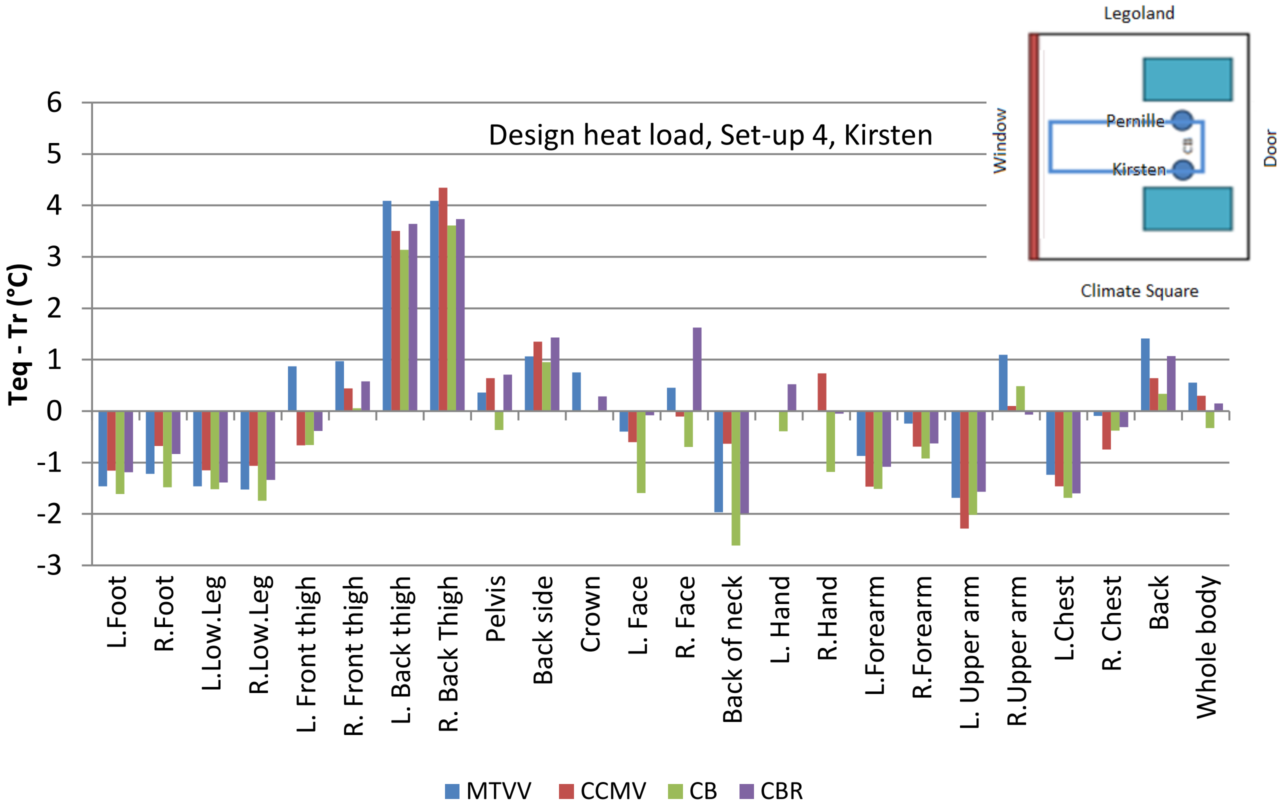


Figure 4.16: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 4 with design heat load.

It can be seen that a stronger cooling effect is felt under set-ups 2 and 4 compared to set-up 1 and 3. These are the set-ups where manikins are sitting away from the window. The heating effect of the window is clearly notable. At set-up 1 and 3 most of the body parts are warmed because of the window. However, the parts that are cooled are from the right side of the body at set-up 1 and left side of the body at set-up 3 – the parts that are facing the opposite wall from the window.

Back thighs of the manikin are warmed in all cases and that is because these body parts are in contact with the chair.

Comparing the MBET for the head region (crown, left face, right face and back of the neck) between set-ups 2 and 4, it is seen that greater cooling effect is achieved under set-up 2 where the manikin is not located exactly below the chilled beam. However, that does not mean that there is no additional cooling from the radiant panels of the chilled beam. The reason might be the close presence of the other manikin at set-up 4.

It can be seen that the chilled beam provides most of the cooling. The cooling effect is also noticeable under the thermal conditions generated by the chilled ceiling combined with mixing ventilation.

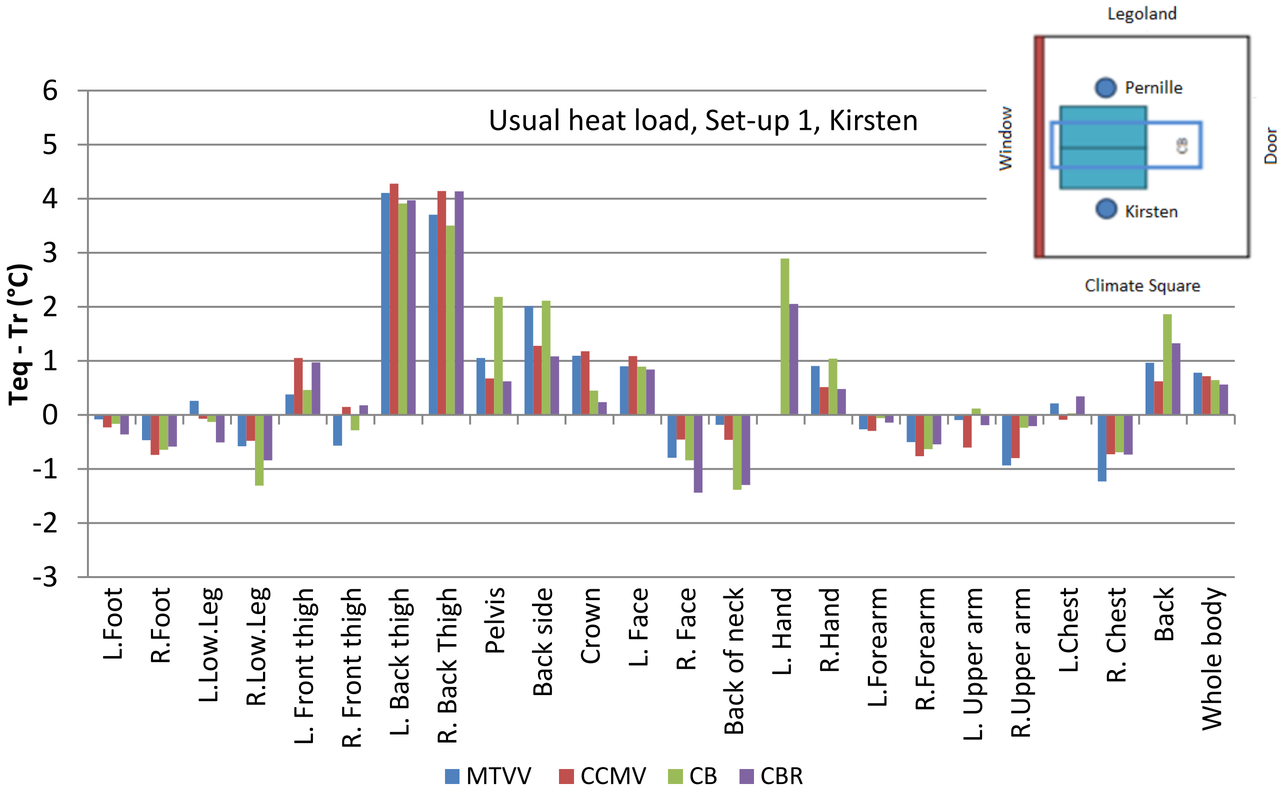


Figure 4.17: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 1 with usual heat load.

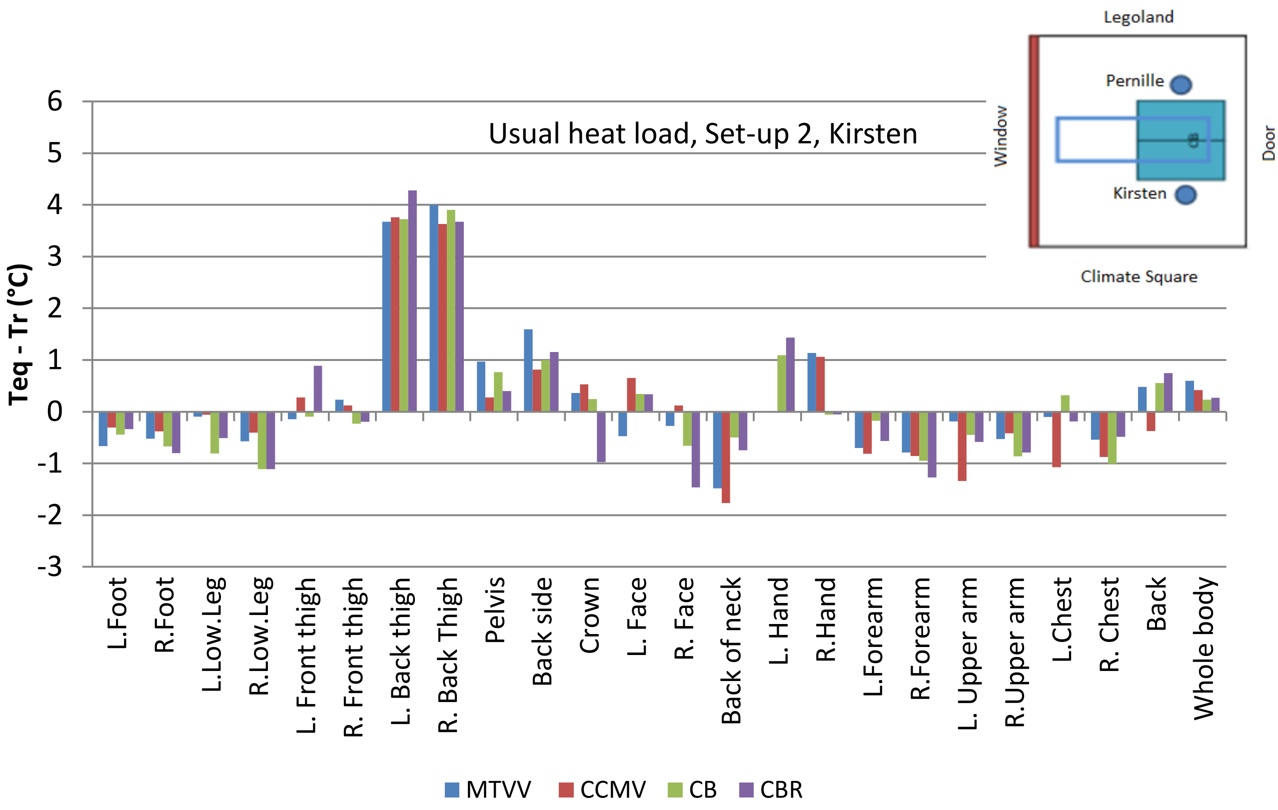


Figure 4.18: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 2 with usual heat load.

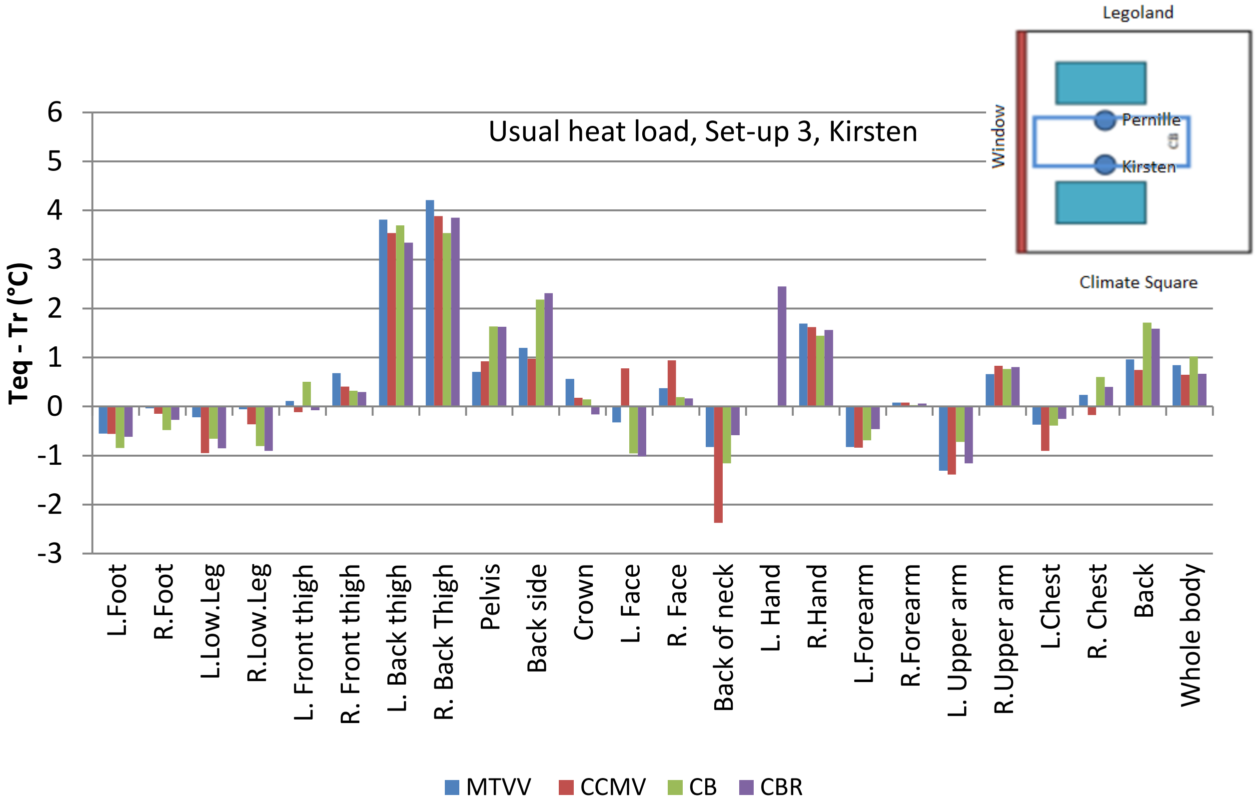


Figure 4.19: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 3 with usual heat load.

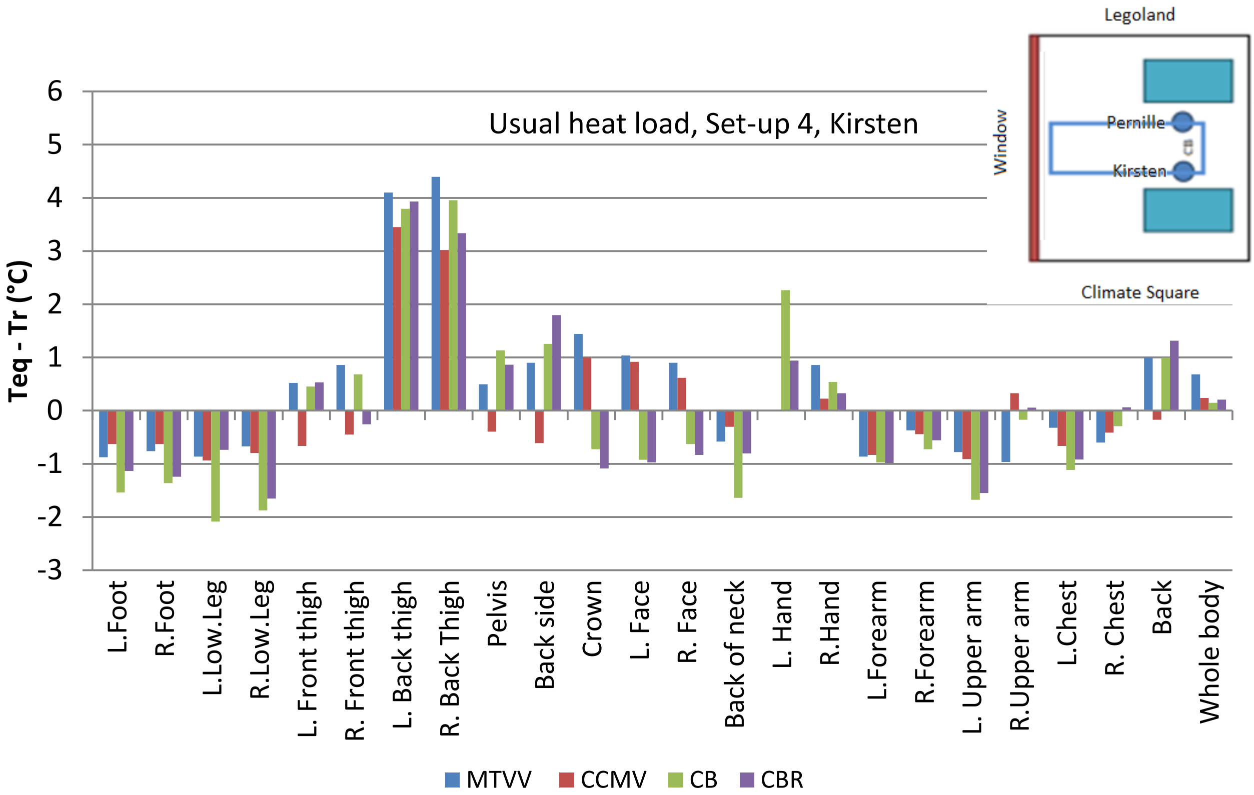


Figure 4.20: Equivalent and room temperature differences of Kirsten (23-body segment manikin) at set-up 4 with usual heat load.

With usual heat load in the room the similar tendencies are observed as with the design heat load cases. However, the MBET differences are not that high, which means that overall the warming and cooling effect, will be felt less drastically.

In Figure 4.20 the effect of the direct exposure to the chilled beam can be seen at head level. Chilled beam and chilled beam with radiant panels provide cooling while chilled ceiling combined with mixing ventilation and mixing ventilation alone have a warming effect. The reason for this could also be that lamps are aligned with chilled beam and chilled beam with radiant panels in the room. Therefore there is more air movement around the lamps and the radiation from the lamps is reduced.

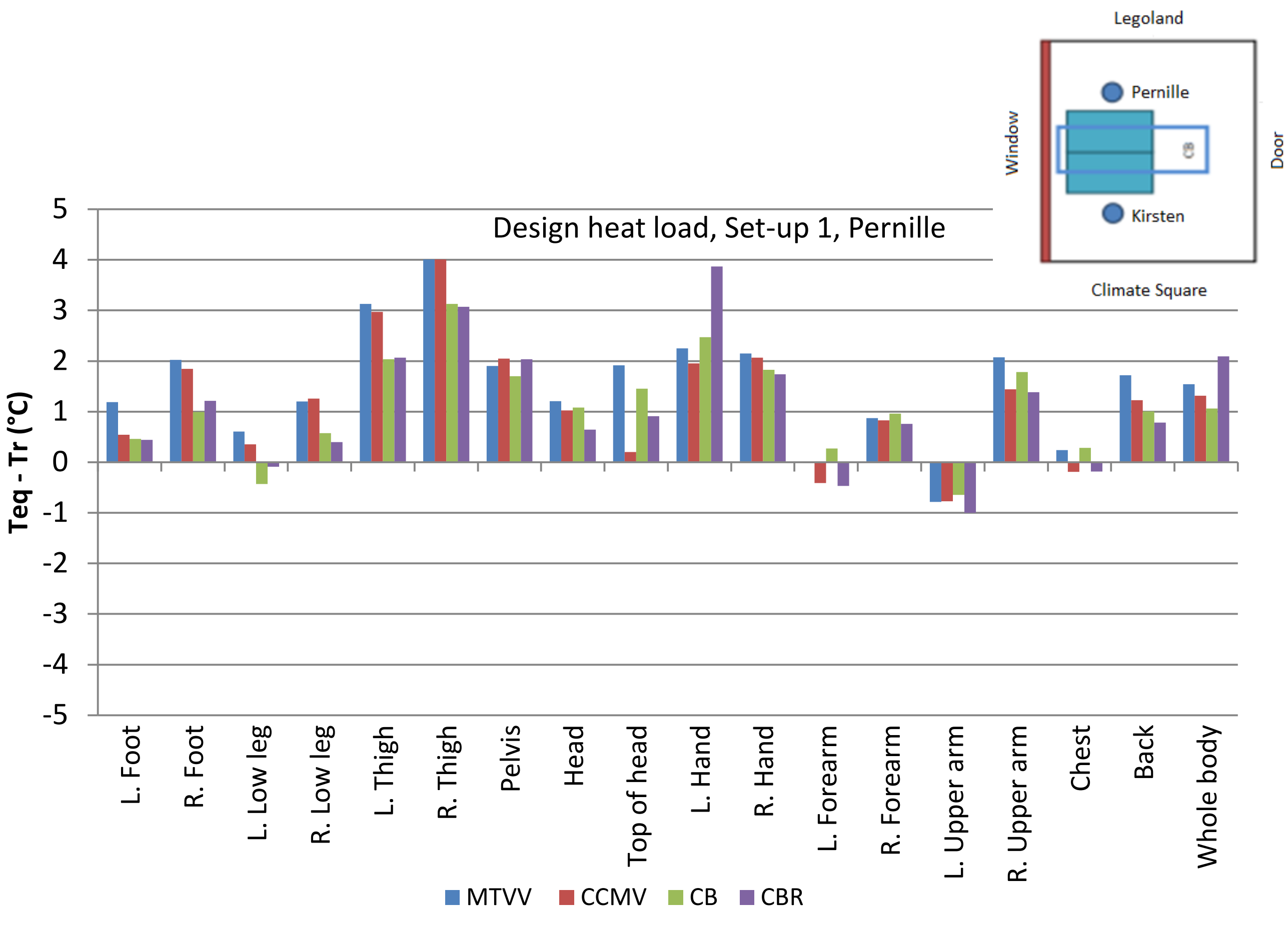


Figure 4.21: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 1 with design heat load.

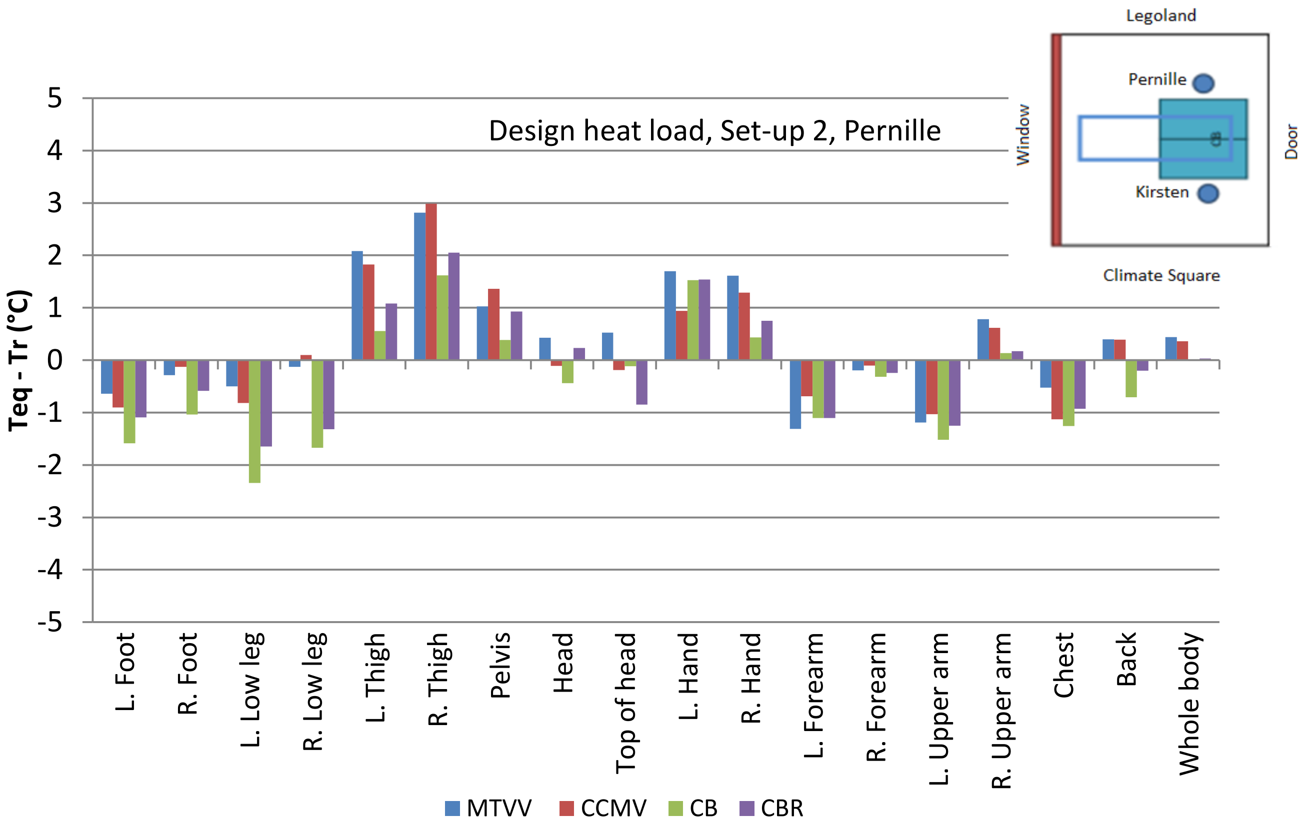


Figure 4.22: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 2 with design heat load.

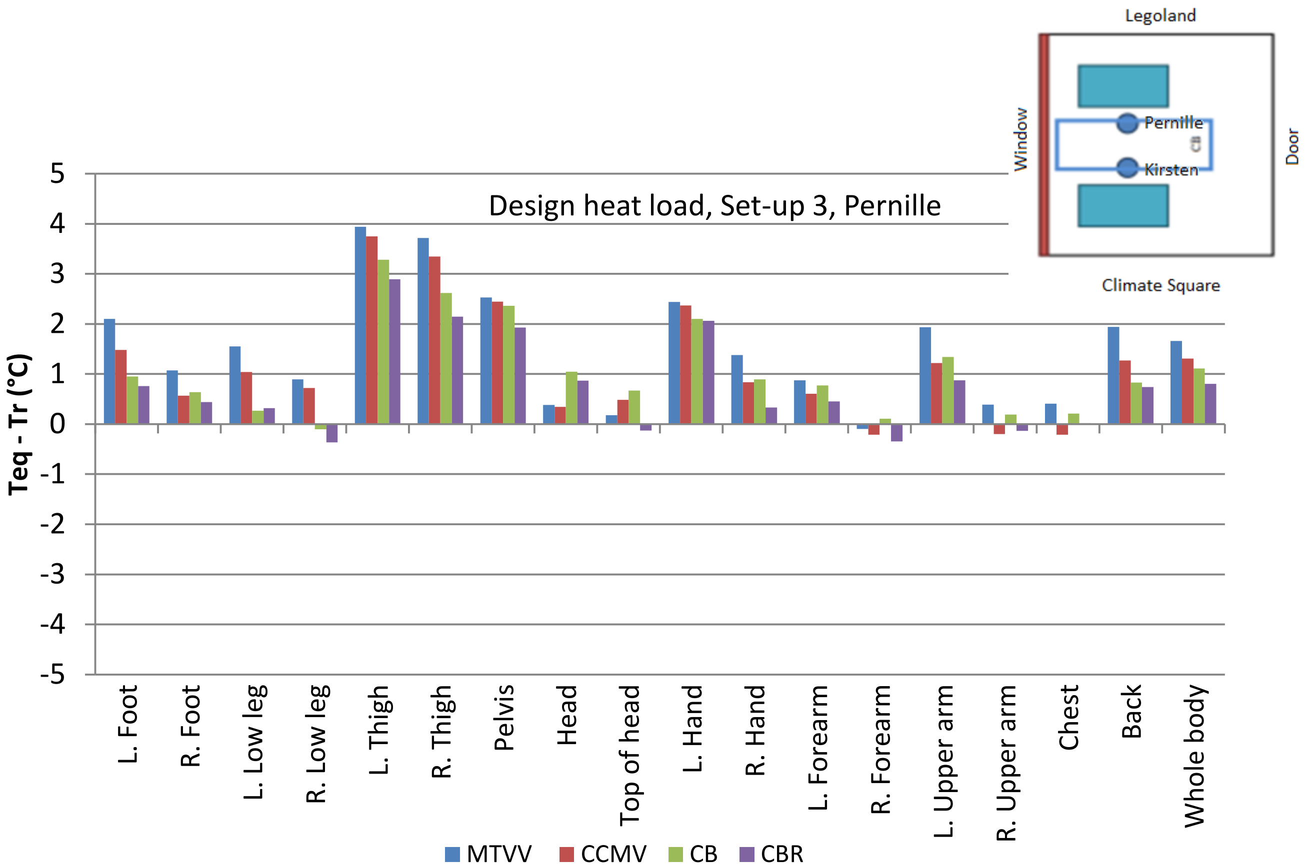


Figure 4.23: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 3 with design heat load.

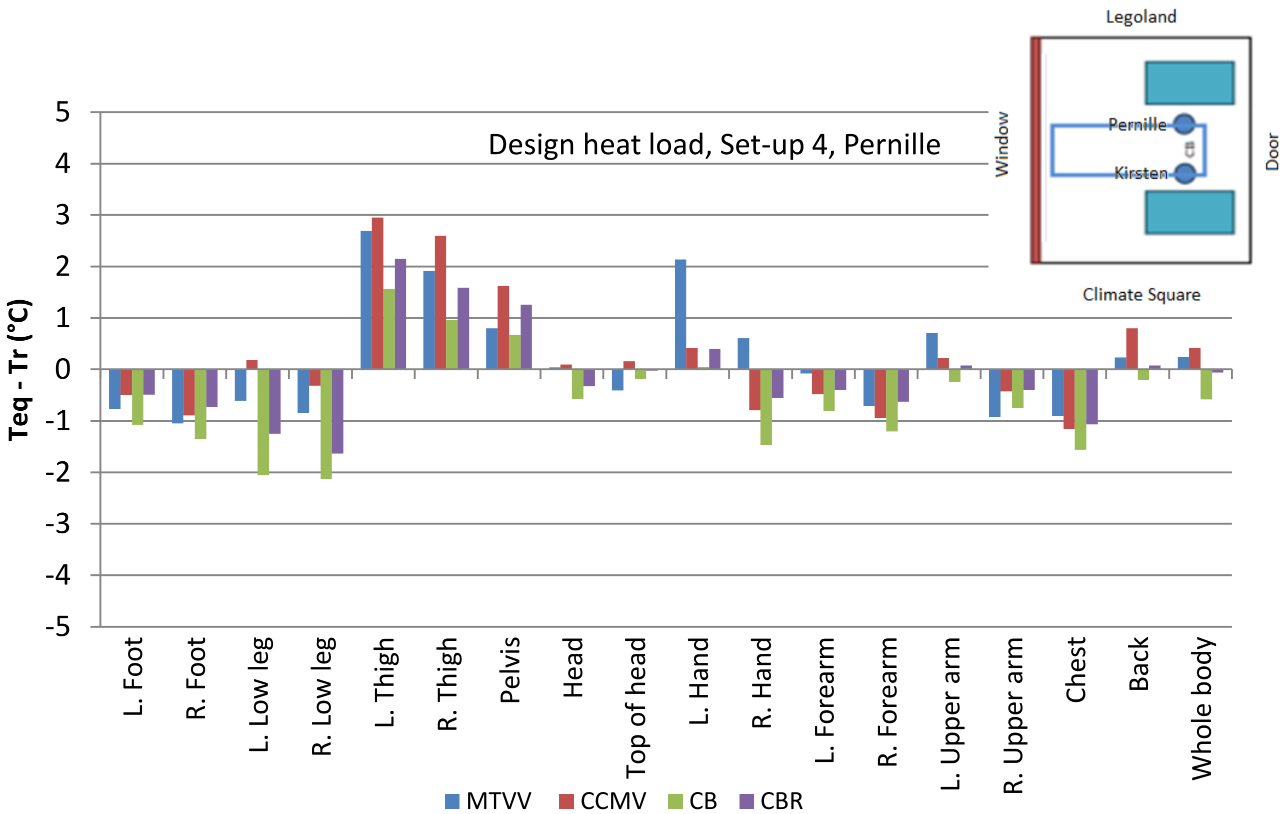


Figure 4.24: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 4 with design heat load.

Figures 4.21 to 4.24 show similar results for Pernille as for Kirsten. The room set-up has clearly an effect on the body cooling/warming. When the manikin is sitting closer to the window, most of the body segments will be warmed. When the manikin is sitting away from the window, higher cooling effect is achieved.

Comparing the top of the head at set-up 1 and set-up 3, it can be seen that when the manikin is exposed directly to the chilled beam (set-up 3), the warming effect is smaller than when the manikin is not sitting directly under the chilled beam (set-up 1) especially for the CBR case. In this case the panels of the CBR are also closer to the head of the manikin compared to the chilled ceiling case. The beam was lifted 2.5 m from the floor.

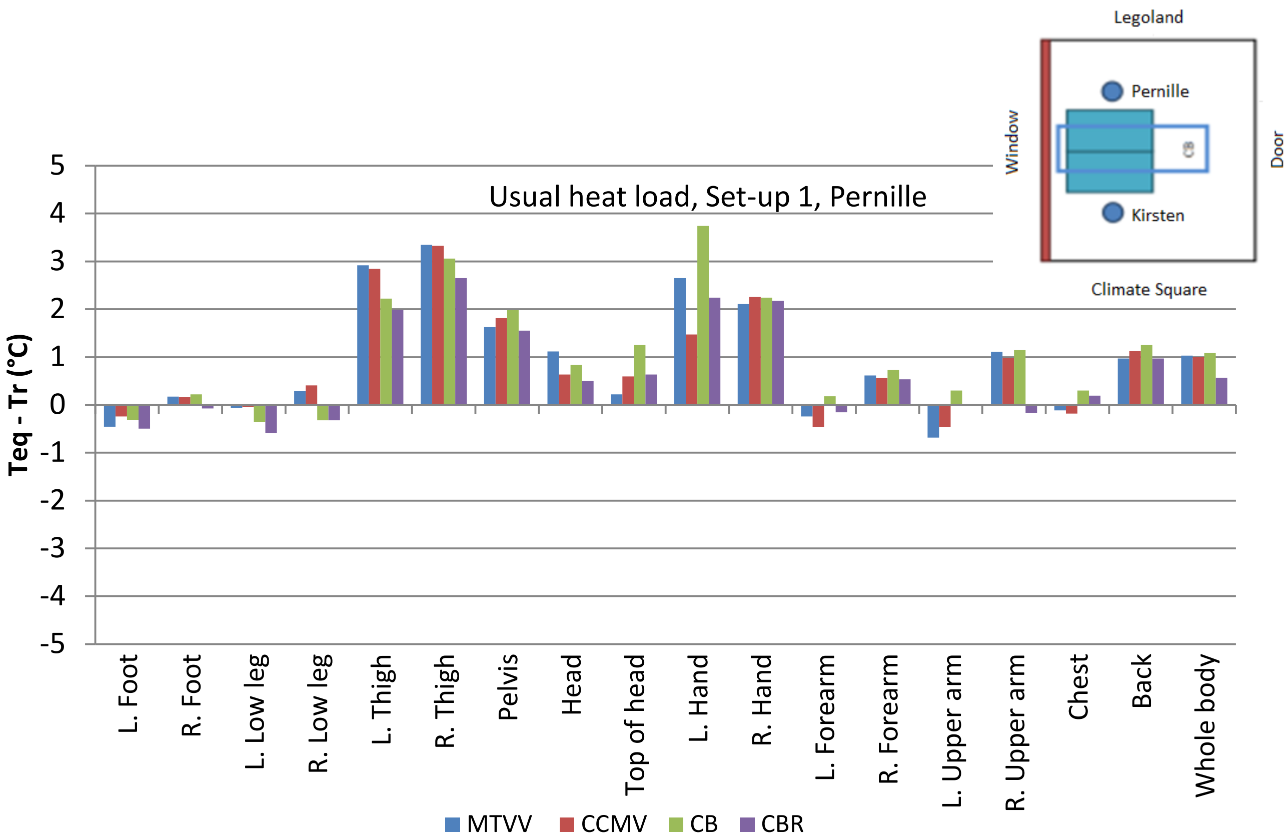


Figure 4.25: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 1 with usual heat load.

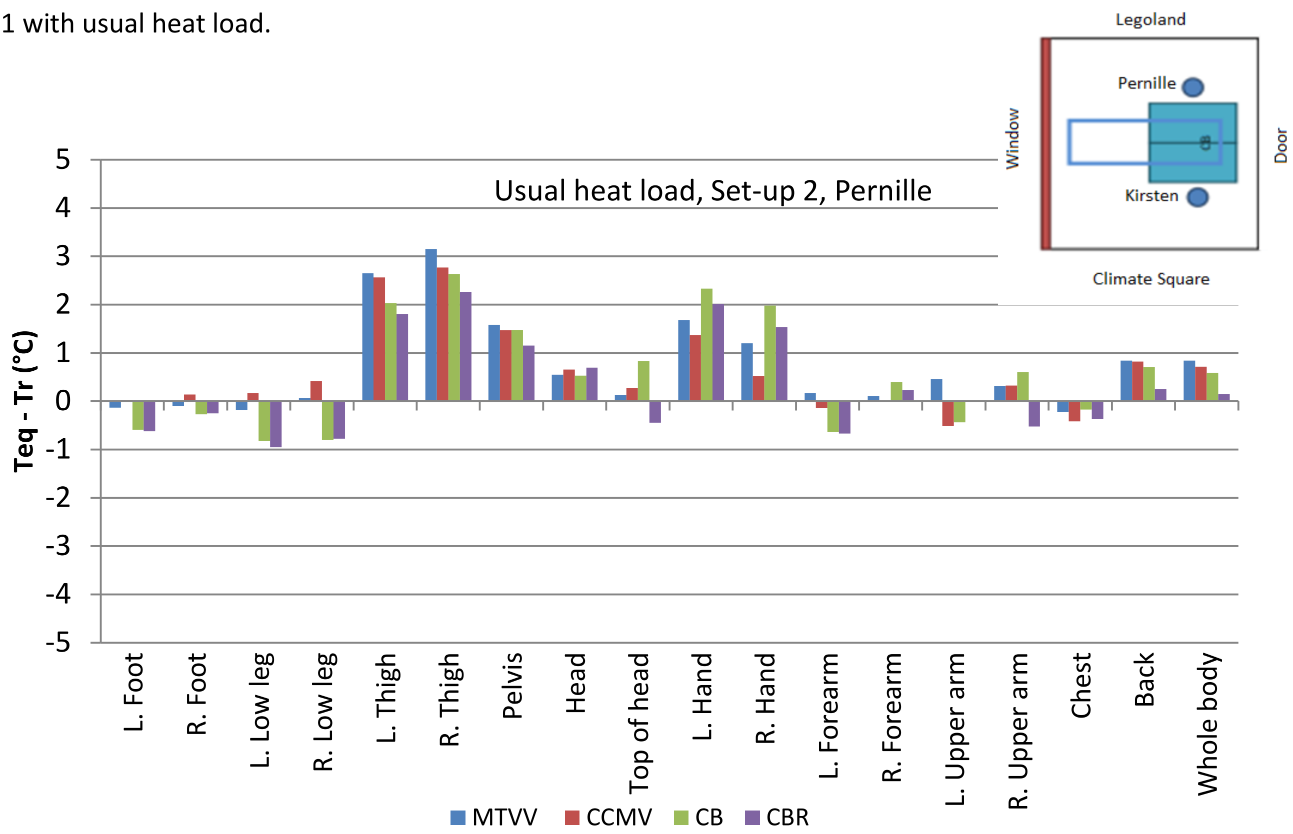


Figure 4.26: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 2 with usual heat load.

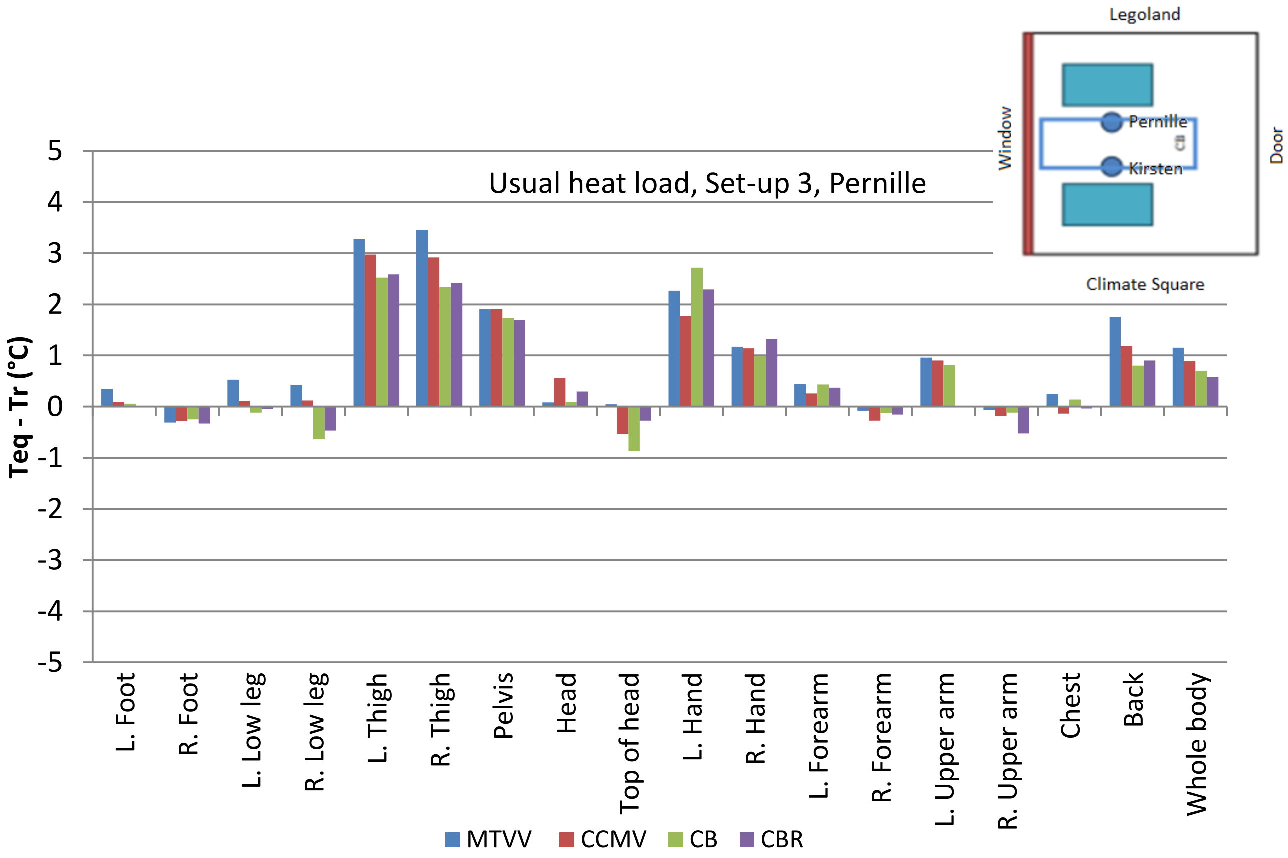


Figure 4.27: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 3 with usual heat load.

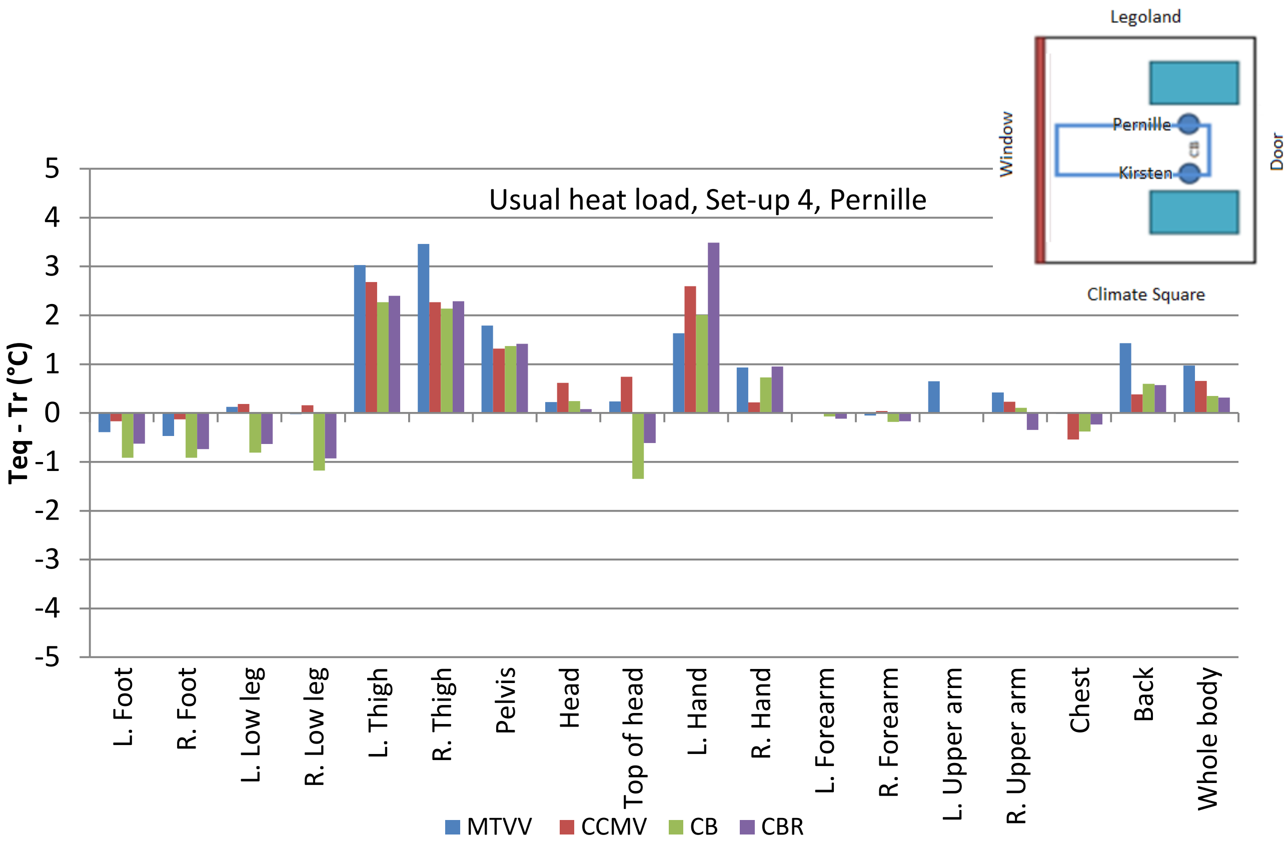


Figure 4.28: Equivalent and room temperature differences of Pernille (17-body segment manikin) at set-up 4 with usual heat load.

With usual heat load the warming and cooling effect is felt at a lower degree for Pernille similarly as for Kirsten. In the Figures 4.25 to 4.28 with low heat load can be well seen which body segments are felt warmer than the reference temperature at all room set-ups – thighs, pelvis, hands and back. That is because all these body parts are in contact with furniture or the lap-top. Thighs, pelvis and part of back are in contact with the chair. Hands were places on a warm lap-top computer.

Equivalent temperature difference from side to side and from head to feet

The equivalent temperature difference between the left side and the right side of the body, and between top of the head (crown) and feet of the manikins is shown in the following Figures 4.30 to 4.37. The difference between the sides is analyzed for the forearms of the manikins.

The results are shown for all the cases and for the 2 manikins. Figure 4.29 shows the positions of the manikins at each room set-up.

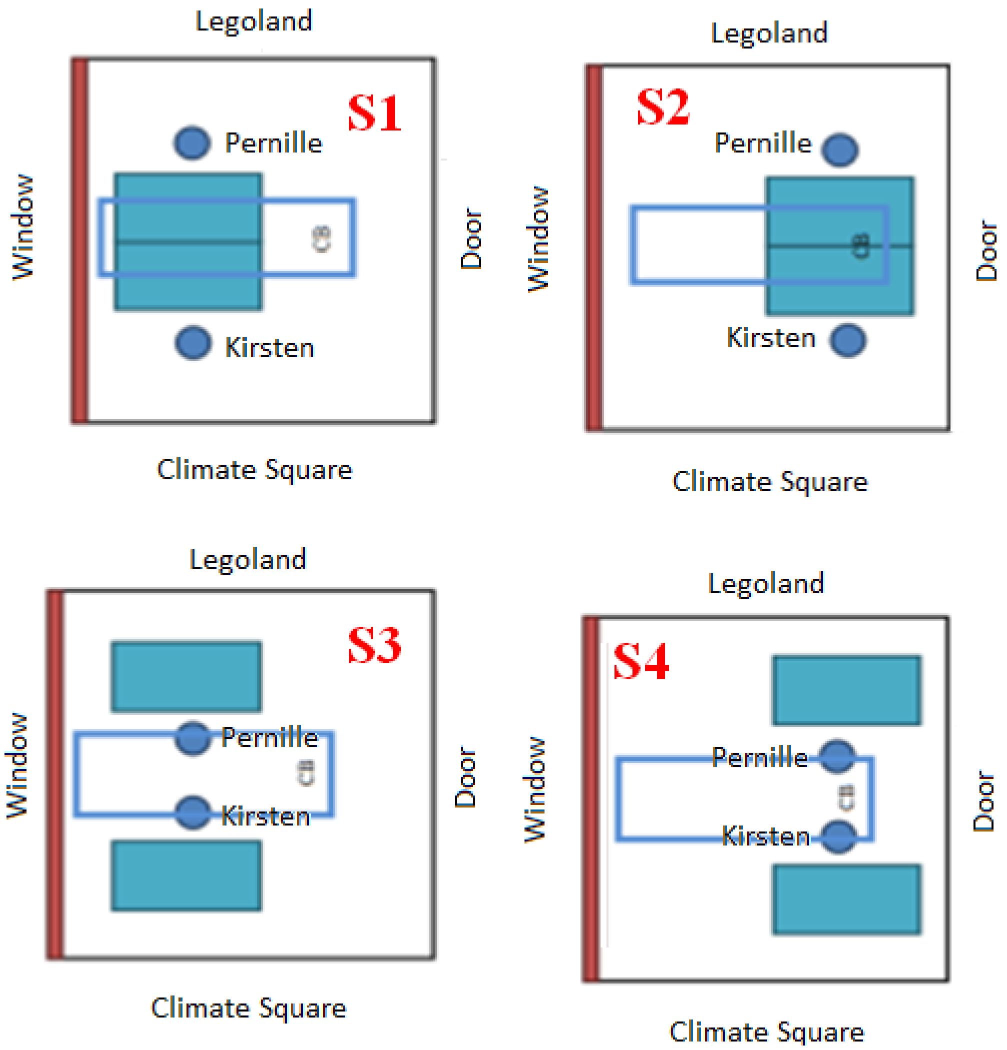


Figure 4.29: All four room set-ups and positions of the manikins.

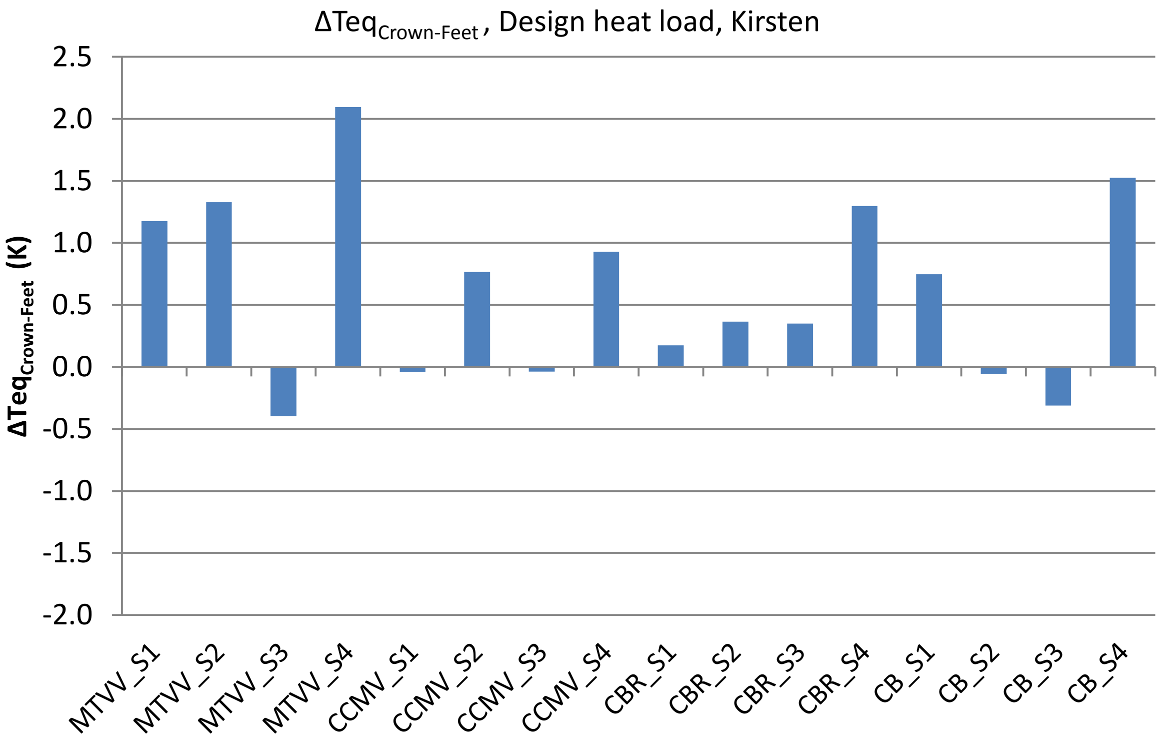


Figure 4.30: Equivalent temperature difference between crown and feet of Kirsten–design heat load.

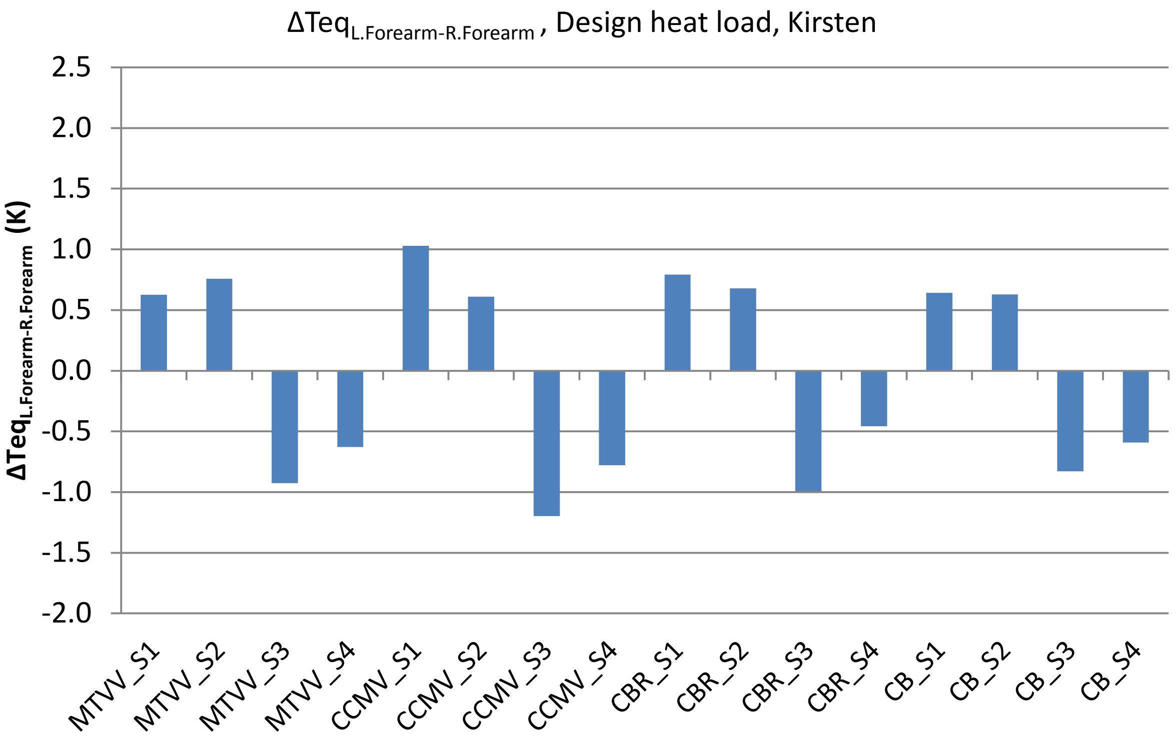


Figure 4.31: Equivalent temperature difference between left and right forearm of Kirsten–design heat load.

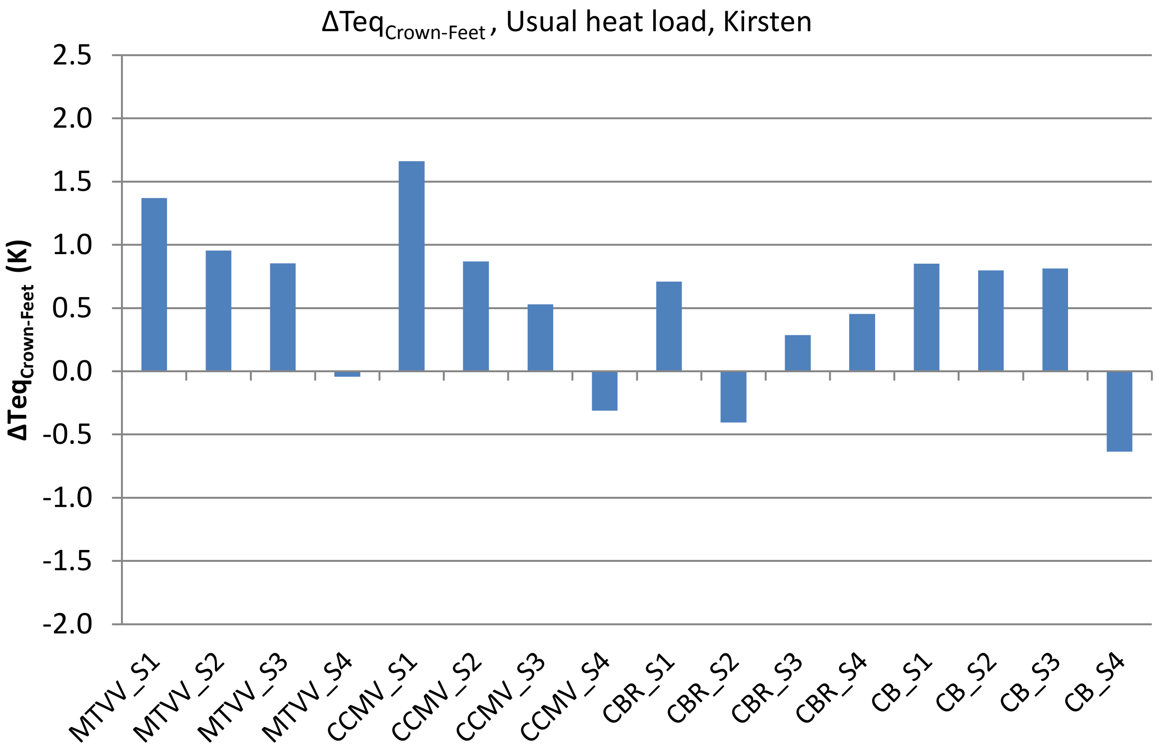


Figure 4.32: Equivalent temperature difference between crown and feet of Kirsten–usual heat load.

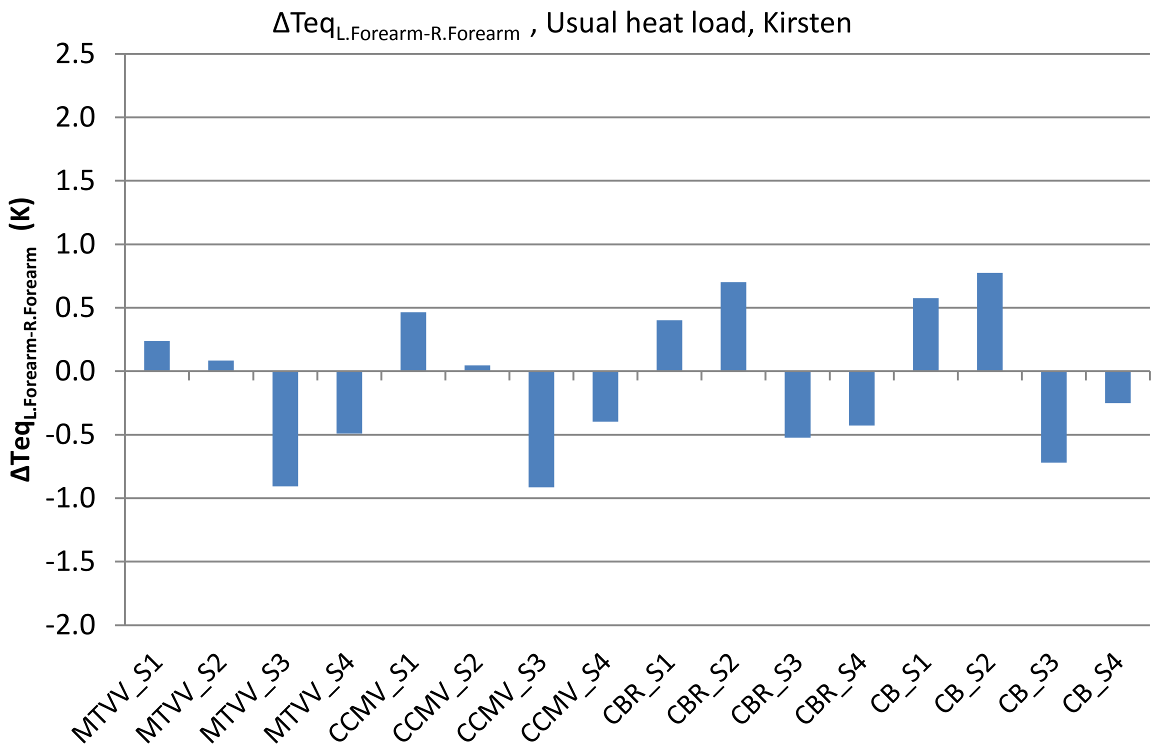


Figure 4.33: Equivalent temperature difference between left and right forearm of Kirsten–usual heat load.

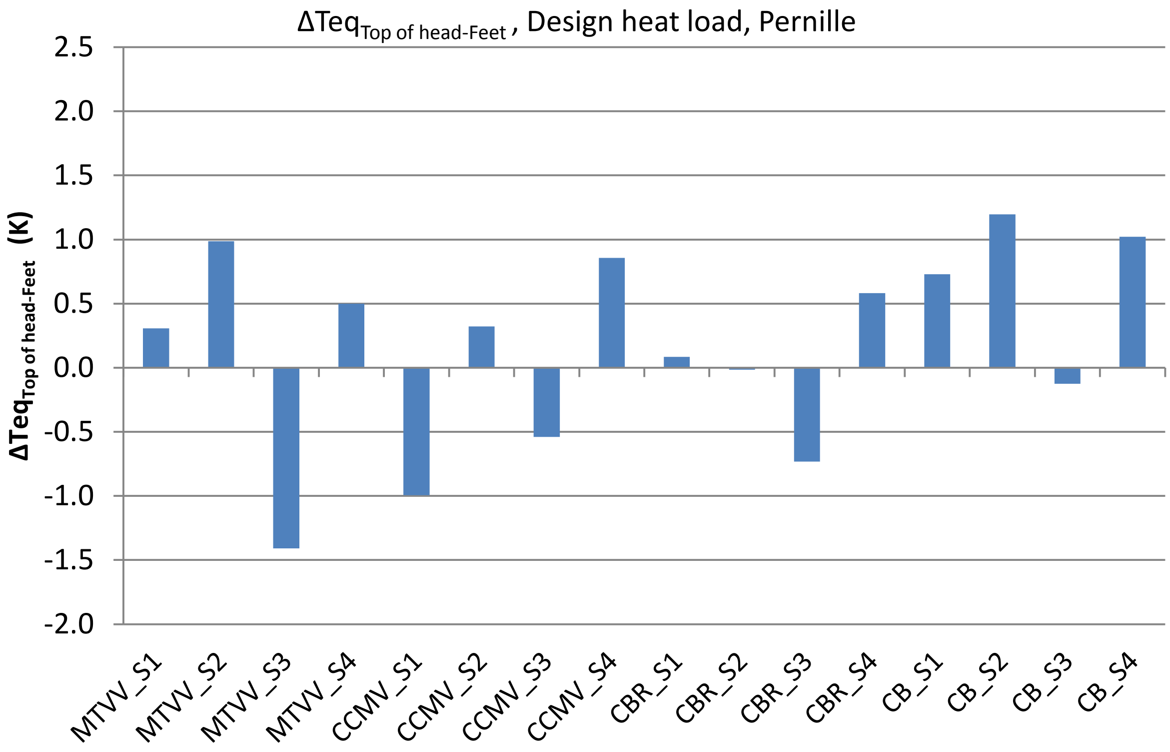


Figure 4.34: Equivalent temperature difference between top of head and feet of Pernille–design heat load.

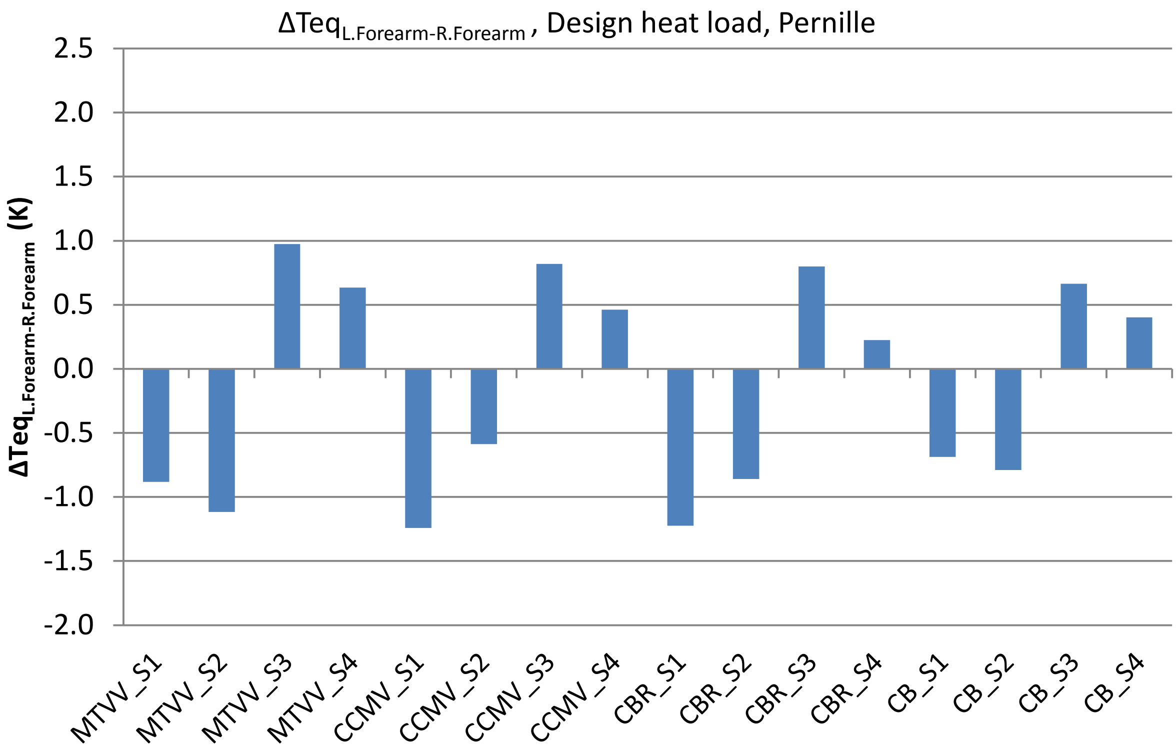


Figure 4.35: Equivalent temperature difference between left and right forearm of Pernille–design heat load.

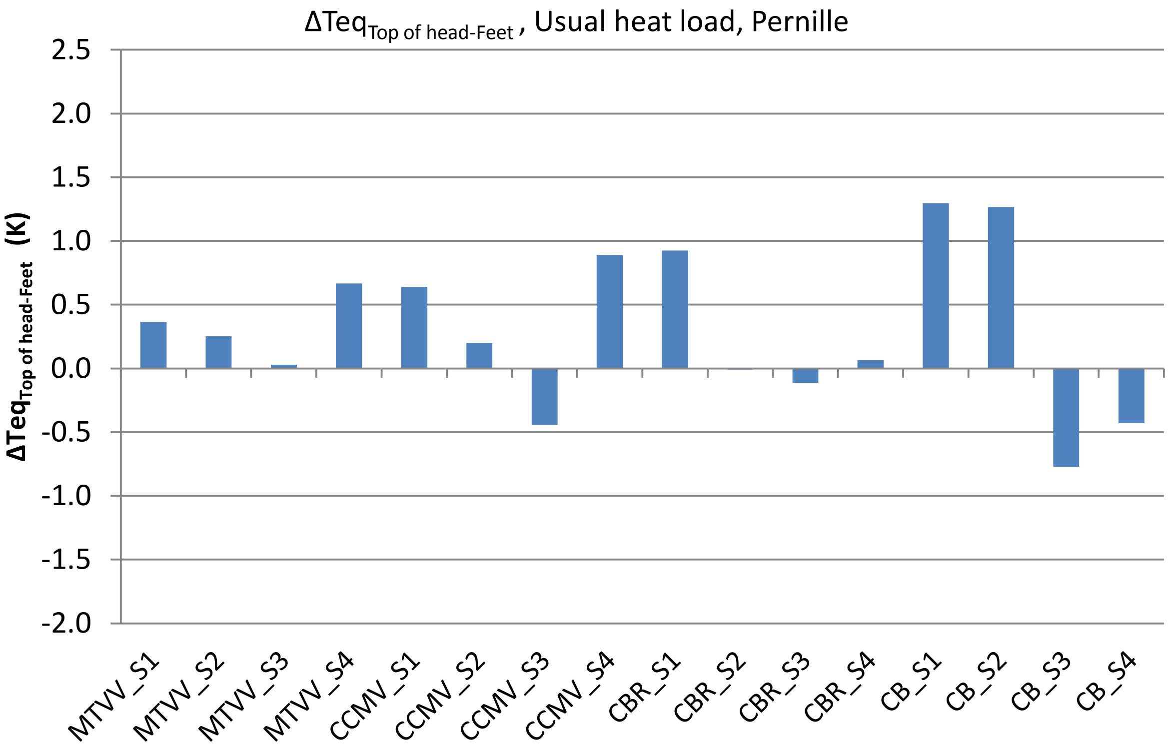


Figure 4.36: Equivalent temperature difference between top of head and feet of Pernille–usual heat load.

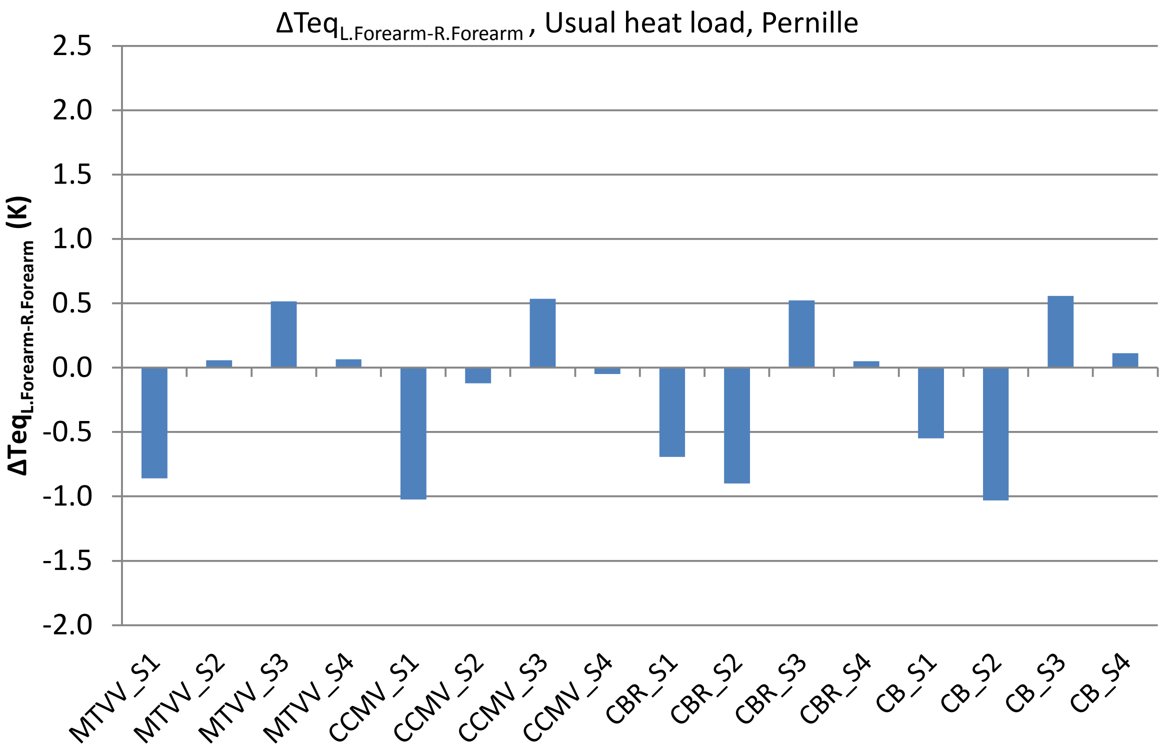


Figure 4.37: Equivalent temperature difference between left and right forearm of Pernille–usual heat load.

For both Kirsten and Pernille the equivalent temperature difference between left and right forearm under all cases show the impact of the window and the direct solar load. The equivalent temperature of the side, which is closer to the window, is always higher than the side, which is away from the window.

The difference is approximately 1 K with design (high heat load) and 0.5 K with usual (low) heat load.

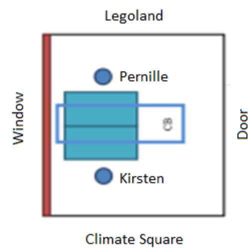
The equivalent temperature difference between top of head (crown) and feet when compared between set-up 1 and set-up 3, and set-up 2 and set-up 4 of CBR, is inconsistent. At some cases set-ups 2 and 4 provide lower equivalent temperature at head region than it is at feet level as would be expected since at these set-ups manikins are sitting directly below the radiant panels of the chilled beam. However, there are cases with opposite results. The reason could be that the cooling effect from the panels is still present but not strong enough to be seen due to the high floor temperatures, the presence of the lamps or impact of the manikins sitting close to each other, for example.

Radiant asymmetry

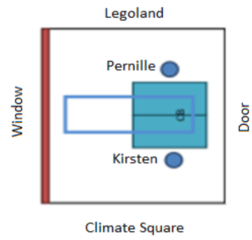
Hereafter the measured radiant asymmetry values are listed in tables (Table 4.11 to 4.14) for every system used.

Table 4.11: Radiant asymmetry – chilled beam.

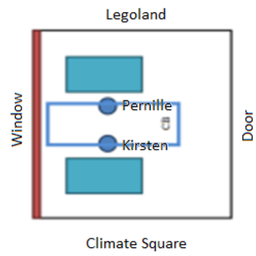
Radiant asymmetry		CB_H_S1				CB_L_S1			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m²	K	W/m²	K	W/m²	K	W/m²
window A / door B	1,1 m	3,5	22	3,3	21	2	13	1,6	10
	0,6 m	2,4	15	2,6	16	1,4	8	1,2	7
floor A / ceiling B	1,1 m	0,6	4	0,7	4	-0,5	-3	-0,4	-2
	0,6 m	0,9	6	1,1	7	-0,9	-5	-0,8	-5
Climate Square A / Legoland B	1,1 m	-0,4	-2	0,6	4	-0,6	-4	1,1	7
	0,6 m	0	0	0	0	-0,1	-1	0,4	2



Radiant asymmetry		CB_H_S2				CB_L_S2			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m²	K	W/m²	K	W/m²	K	W/m²
window A / door B	1,1 m	2,1	13	2,1	13	1,1	7	1	6
	0,6 m	2	12	2,1	13	1,1	7	1	6
floor A / ceiling B	1,1 m	-0,2	-1	0	0	-0,3	-2	-0,2	-1
	0,6 m	-0,7	-4	-0,6	-4	-0,6	-4	-0,7	-4
Climate Square A / Legoland B	1,1 m	0,9	6	1,3	8	-1,1	-7	1,3	8
	0,6 m	-0,2	-1	0,2	1	-0,2	-1	0,5	3



Radiant asymmetry		CB_H_S3				CB_L_S3			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m²	K	W/m²	K	W/m²	K	W/m²
window A / door B	1,1 m	3,6	22	4	25	1,8	11	1,8	11
	0,6 m	2,8	17	3	19	1,1	7	1,2	7
floor A / ceiling B	1,1 m	0,1	1	0,5	3	-0,1	0	0	0
	0,6 m	0,1	1	0,9	6	-0,9	-6	-0,9	-5
Climate Square A / Legoland B	1,1 m	-0,1	-1	-0,4	-2	0,7	4	-0,5	-3
	0,6 m	-0,6	-4	0,1	1	-0,2	-1	0,3	2



Radiant asymmetry		CB_H_S4				CB_L_S4			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m²	K	W/m²	K	W/m²	K	W/m²
window A / door B	1,1 m	2,2	14	2,2	13	1,2	7	1,3	8
	0,6 m	2,1	13	2	13	1,3	8	1,2	7
floor A / ceiling B	1,1 m	-0,1	0	0,3	2	-0,2	-1	0,1	1
	0,6 m	-0,7	-5	-0,4	-3	-0,8	-5	-0,6	-4
Climate Square A / Legoland B	1,1 m	-0,1	-1	-0,2	-1	0,4	2	-0,5	-3
	0,6 m	-0,6	-4	0,6	4	-0,2	-1	0,3	2

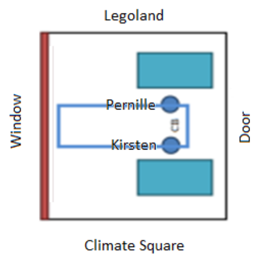
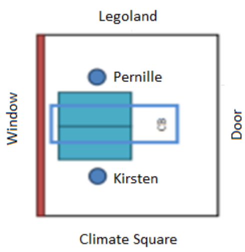
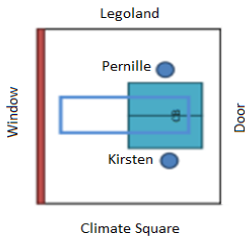


Table 4.12: Radiant asymmetry – chilled beam with radiant panels.

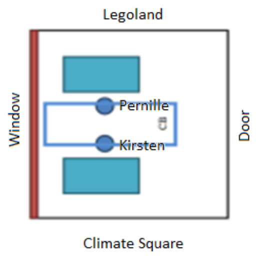
Radiant asymmetry		CBR_H_S1				CBR_L_S1			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	3,7	2,3	3,5	22	1,9	12	1,8	11
	0,6 m	2,6	16	2,7	17	1,2	7	1,1	7
floor A / ceiling B	1,1 m	1,3	8	1,4	9	-0,1	-1	0,1	1
	0,6 m	1,8	11	2,1	13	-0,5	-3	-0,8	-5
Climate Square A / Legoland B	1,1 m	-0,8	-5	0,3	2	-0,6	-4	1	6
	0,6 m	0,1	1	0,2	1	-0,2	-1	0,3	2



Radiant asymmetry		CBR_H_S2				CBR_L_S2			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	2,2	13	1,7	10	1,3	8	1	6
	0,6 m	2,4	15	1,9	12	1,1	7	0,9	5
floor A / ceiling B	1,1 m	0,6	4	0,9	6	0,1	1	-0,5	-3
	0,6 m	0,1	1	-0,2	-1	-0,3	-2	-0,6	-3
Climate Square A / Legoland B	1,1 m	-0,5	-3	0,5	3	-1,3	8	1,3	8
	0,6 m	-0,2	-1	0	0	-0,5	-3	0,3	2



Radiant asymmetry		CBR_H_S3				CBR_L_S3			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	3,7	23	4	25	2	12	1,9	12
	0,6 m	2,7	17	3	18	1,3	8	1,3	8
floor A / ceiling B	1,1 m	1,1	7	1,8	11	0,4	2	0,2	1
	0,6 m	0,9	6	1,8	11	-0,6	-4	-0,6	-4
Climate Square A / Legoland B	1,1 m	0,5	3	-0,4	-2	0,6	3	-0,5	-3
	0,6 m	-0,2	-1	-0,1	-1	-0,3	-2	0,3	2



Radiant asymmetry		CBR_H_S4				CBR_L_S4			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	1,8	11	2	12	1,2	7	1,1	7
	0,6 m	1,8	11	1,9	11	1,1	7	1,1	7
floor A / ceiling B	1,1 m	1	6	1,3	8	0,3	1	0,5	3
	0,6 m	0	0	0	0	-0,4	-4	-0,4	4
Climate Square A / Legoland B	1,1 m	0,4	2	-0,8	-4	0,4	2	-0,6	-4
	0,6 m	-0,2	-1	0,3	2	-0,2	-2	0,2	1

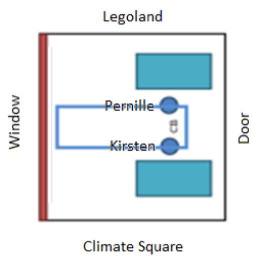
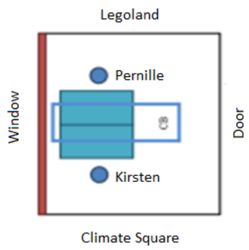
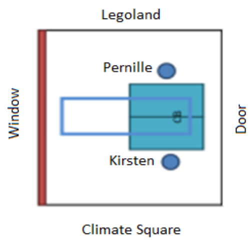


Table 4.13: Radiant asymmetry – chilled ceiling combined with mixing ventilation.

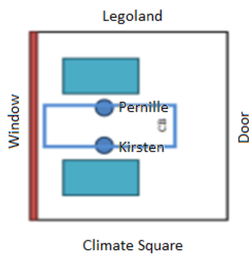
Radiant asymmetry		CCMV_H_S1				CCMV_L_S1			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	4	25	Error	Error	1,8	11	1,7	10
	0,6 m	3,2	19	2,9	17	1,2	7	1,3	8
floor A / ceiling B	1,1 m	2,7	16	2,3	14	0,5	3	0,8	5
	0,6 m	2,8	17	3,2	20	0	0	0,1	0
Climate Square A / Legoland B	1,1 m	-0,4	-3	Error	Error	-0,5	-3	1,2	7
	0,6 m	0	0	0,9	6	-0,2	-1	0,4	3



Radiant asymmetry		CCMV_H_S2				CCMV_L_S2			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	1,8	11	1,8	11	0,8	5	0,7	4
	0,6 m	1,7	10	2	12	0,6	4	0,5	3
floor A / ceiling B	1,1 m	2,1	13	2,5	15	0,8	5	0,8	5
	0,6 m	1,3	8	1,1	7	0,3	2	0,3	2
Climate Square A / Legoland B	1,1 m	-1	-6	1	7	-0,6	-4	1	6
	0,6 m	-0,6	-4	0,4	3	-0,3	-2	0,5	3



Radiant asymmetry		CCMV_H_S3				CCMV_L_S3			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	4	25	4,1	25	1,8	11	2	12
	0,6 m	3,2	20	3,1	19	-1,4	-9	1,5	9
floor A / ceiling B	1,1 m	2,8	17	3	19	0,7	4	0,7	4
	0,6 m	2,4	15	2,8	17	0	0	0,2	1
Climate Square A / Legoland B	1,1 m	-0,2	-1	-0,3	-2	0,2	1	-0,3	-2
	0,6 m	0,2	1	0,2	1	-0,3	-2	0,5	3



Radiant asymmetry		CCMV_H_S4				CCMV_L_S4			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	1,9	12	1,8	11	0,7	4	0,9	5
	0,6 m	2	12	1,9	11	0,8	5	0,8	5
floor A / ceiling B	1,1 m	2,4	14	2,3	14	0,8	5	0,9	5
	0,6 m	1,2	7	1	5	0,3	2	0,3	2
Climate Square A / Legoland B	1,1 m	0,5	3	-0,2	-1	0,3	2	-0,4	-3
	0,6 m	-0,1	-1	0,3	2	-0,2	-1	0,5	3

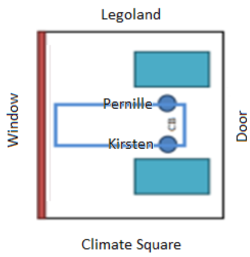
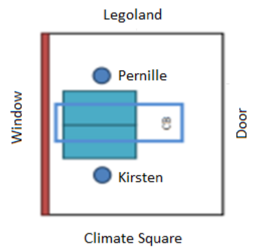
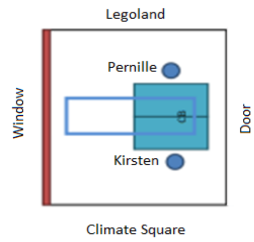


Table 4.14: Radiant asymmetry – mixing ventilation.

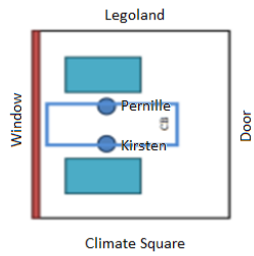
Radiant asymmetry		MTVV_H_S1				MTVV_L_S1			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	3,6	22	3,5	21	1,6	10	1,6	10
	0,6 m	2,8	18	2,8	18	1,2	7	1,3	8
floor A / ceiling B	1,1 m	1,2	7	1,2	8	-0,2	-1	0	0
	0,6 m	1,4	9	1,5	9	-0,7	-4	-0,7	-4
Climate Square A / Legoland B	1,1 m	-0,3	-2	0,5	3	-0,7	-5	0,9	5
	0,6 m	0,4	2	0,1	1	-0,1	-1	0,1	1



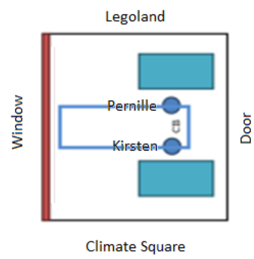
Radiant asymmetry		MTVV_H_S2				MTVV_L_S2			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	2	13	2,1	13	0,9	5	0,8	5
	0,6 m	1,9	12	2,1	13	0,8	5	0,7	5
floor A / ceiling B	1,1 m	0,4	2	0,5	3	-0,1	0	0,1	1
	0,6 m	-0,1	-1	-0,2	-1	-0,4	-2	-0,5	-3
Climate Square A / Legoland B	1,1 m	-0,7	-4	1,1	7	-0,8	-5	1	6
	0,6 m	-0,2	-1	0,5	3	-0,3	-2	0,4	2



Radiant asymmetry		MTVV_H_S3				MTVV_L_S3			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	3,7	23	3,9	24	1,7	10	1,9	12
	0,6 m	3	19	2,8	18	-1,4	-8	1,3	8
floor A / ceiling B	1,1 m	1,1	7	1,5	9	-0,3	-2	0,2	1
	0,6 m	1,1	7	1,7	10	-0,7	-4	-0,6	-4
Climate Square A / Legoland B	1,1 m	0	0	-0,3	-2	0,4	3	-0,8	-5
	0,6 m	-0,2	-1	-0,1	-1	-0,3	-2	0,3	2



Radiant asymmetry		MTVV_H_S4				MTVV_L_S4			
		Kirsten		Pernille		Kirsten		Pernille	
		K	W/m^2	K	W/m^2	K	W/m^2	K	W/m^2
window A / door B	1,1 m	2,3	14	2,3	14	0,8	5	1	6
	0,6 m	2,3	14	2,3	14	0,9	5	1	6
floor A / ceiling B	1,1 m	0,6	4	0,6	4	0,2	1	0,5	3
	0,6 m	-0,2	-1	-0,1	-1	-0,4	-3	-0,4	-3
Climate Square A / Legoland B	1,1 m	0	0	-0,2	-1	0,4	2	-0,5	-3
	0,6 m	-0,3	-2	0,5	3	-0,4	-3	0,2	2



The results show that at all cases radiant asymmetry is relatively high between window and door when compared with other directions, especially under the high heat (design) load. The warm window is an important factor for the radiant asymmetry. However, according to the ISO Standard 7730 [2] requirement for category B of radiant asymmetry with warm wall (window) is below 23°C, which is fulfilled. Occupants are not likely to feel local discomfort due to warm wall or window.

When comparing radiant asymmetry between floor and ceiling at 1.1 m between set-ups 1 and 3 (not direct exposure to the CB) and set-ups 2 and 4 (direct exposure to the CB) for cases with CBR, it can be seen that for the set-ups 2 and 4 the radiant asymmetry is higher. Since floor was measured with side "A" of the sensor and ceiling with side "B" of the sensor, and the radiant asymmetry is calculated as A minus B, it means that the cooling effect of the radiant panels is present. However, it could also be that the temperature of the floor is making the difference. The radiant asymmetry difference between set-ups 1 and 3 and set-ups 2 and 4 is roughly 0.4 K or 40%.

5. Discussion

The purpose of the study was to compare chilled beam (CB), chilled beam with radiant panels (CBR), chilled ceiling combined with mixing ventilation (CCMV) and total volume mixing ventilation (MTVV) with regard to provided thermal indoor environment under different levels of heat load and heat distribution in the room. The achieved results were compared based on the requirements of ISO Standard 7730 [2] for thermal environment category B.

It was expected that chilled beam with radiant panels will be able to provide lower operative temperature in the occupied zone due to the radiant cooling from the panels. As a result, background temperature (air temperature) could be kept higher, which would lead to energy savings. Similar effect was expected from chilled ceiling combined with mixing ventilation. In fact, CCMV was expected to provide this effect independent from the room set-up. The study proved that CCMV was able to provide lower operative temperature compared to the other systems. However, it was not the same at all room set-ups and under both heat loads. This is explained below.

First, the stability of the systems was analyzed. It was judged how homogenous the thermal environment generated by the systems was with regard to air and operative temperature. For this purpose the temperature normalized index was introduced. The closer the temperature throughout the room to the reference point temperature, the more stable the system is, i.e. it managed to provide more homogenous thermal environment inside the conditioned and ventilated space.

The most homogenous environment from air temperature perspective was provided by CCMV. This could be explained with the fact that radiant cooling panels covered 75% of the ceiling area. It means that the cooling effect from the system is evenly distributed throughout the room volume. The least homogenous environment from air temperature perspective was provided by the chilled beam. One of the reasons could be the air movement pattern in the room. Visualization needs to be performed in order to validate this. No impact of room set-up and heat load level was observed for all systems.

With regard to operative temperature, CCMV was able to provide lower operative temperature than the other three tested systems for the high (design) heat load conditions. The operative temperature decreased with height showing the cooling effect from the radiant panels. However, this was not observed during low (usual) heat load conditions. With usual heat load in the room all systems kept almost the same operative temperature in the room as it was at the reference point. This leads to the conclusion that the cooling effect from CCMV is more effective when high heat loads have to be removed from the room. Although the same effect was expected from CBR, it was not observed. The reason could be that in this analysis all measured points are taken into account but CBR provides the cooling effect only at few points (points below the chilled beam) and this cooling effect may not be strong enough to affect the average value of all points. This result is confirmed by the plane measurements shown in Appendix 3. Room set –ups did not have consistent effect on the systems performance. However, CCMV was able to lower the operative temperature more at set-up 3 and set-up 4 when the two manikins were sitting close together with backs to each other. This would make strong heat flow above the two manikins, which was removed by CCMV very successfully.

Requirement from [2] for operative temperature difference between head (1.1 m) and ankles (0.1 m) for category B is to be less than 3°C. This requirement was fulfilled with all studied systems and for all studied cases.

For all systems under all studied cases the highest air velocities occurred close to the floor level, which then decreased with increasing the height. The reason most probably is the air movement pattern in the room. The air movement was observed and corrected by [1] using a smoke visualization. It was necessary to prevent the air to fall too early therefore only two linear diffusers were used with CCMV in order to increase the momentum of the jets. Because of these changes and the fact that the room is relatively small with high heat load, the air supply jet from the diffusers and chilled beam, spreads along the ceiling, glides down the walls and reaches the floor with high velocity. The air velocities in the room are higher for the high (design) heat load conditions compared to the low (usual) heat load conditions for all systems. Room set-up did not have significant impact on air velocities in the room. The lowest air velocities and least draught rate are provided by CCMV. As mentioned before, the reason probably is that only two linear slot diffusers are used. Although this increases the momentum of the jets, the air has more space to spread and not interact with other jets. At the same time the air is in contact with the chilled ceiling (75% of the area) which cools the air and it has the tendency to fall also before reaching the walls. The highest air velocities are provided by mixing ventilation, which was also expected as the heat from the room is removed by air only (higher supply air flow rate compared to the other systems is needed).

The results of draught rate analysis in the room are much determined by the measured air velocities. According to [2] draught rate requirement for category B is <20%. This requirement is not completely fulfilled for the high heat load conditions. At 76% of all points measured (all room set-ups into account), the draught rate exceeds the recommended maximum level with mixing ventilation, second highest (47%) is with chilled beam. However, draught risk decreases considerably for the usual heat load conditions. (6% MTVV and 24% CB). CCMV exceeds the recommended value only at 2 measured points with the high heat load in the room and completely fulfils the requirement with the low (usual) heat load in the room.

The difference between the manikin based equivalent temperature and the reference point temperature (cooling/warming effect on the manikins' body parts), the difference between MBET for top of head and MBET for feet and the horizontal difference between the MBET for the left forearm and the MBET for the right forearm, showed clearly the impact of the room set-up. The simulated window with the direct solar load on the floor caused warming effect to the body parts that were closer, while those body parts that were further away from the window were felt as cooler. The heat load in the room determines how strongly the warming or cooling effect was felt – it was felt to a greater degree during high heat load conditions than during low heat load conditions. With low (usual) heat load in the room the occupants would feel slightly cooler.

The cooling effect for the top of head (crown), when the manikins were directly exposed to the CBR (sitting directly below the radiant panels) was inconsistent. At some cases it was observed while at others not at all. This might lead to a conclusion that the cooling effect is not strong enough to be noticeable.

The highest radiant asymmetry is documented to be between the window and door due to the warm window surface. The radiant asymmetry was measured only at the locations were the manikins were sitting. According to [2] requirement of radiant asymmetry for warm wall (window) for category B is <23 K. This requirement was fulfilled: the highest radiant asymmetry measured was only 4 K. Occupants are not likely to feel local thermal discomfort due to presence of warm wall/window.

According to the results and analysis the chilled ceiling combined with mixing ventilation (CCMV) performed slightly better compared to the other three systems.

6. Conclusions

All systems performed equally well and managed to keep the required thermal environment for category B.

The differences in performance of the systems are rather small and it is hard to recognize which system is the best with regard to the thermal indoor environment.

High (design) heat load reveals the differences in performance of the systems better - with low (usual) heat load all the systems perform very similar.

Room set-up (heat distribution) effect on the systems' performance is inconsistent; the set-up has an impact on the warming/cooling effect of the thermal manikins' (occupants') body parts.

Air temperature

Air temperature is higher between heights 1.7 m and 2.4 m than the air temperature at the reference point for all cases.

The most homogenous environment with regard to air temperature was provided by the chilled ceiling combined with mixing ventilation (CCMV). The least stable system was the chilled beam (CB).

Heat load and room set-up has insignificant effect on the systems` performance with regard to provide homogenous air temperature field.

Operative temperature

During high (design) heat load conditions chilled ceiling combined with mixing ventilation was able to provide lower operative temperature throughout the room than the other three systems.

During low (usual) heat load conditions all systems provided equally homogenous environment with regard to operative temperature.

From all the systems chilled beam (CB) has the most noticeable tendency to provide lower operative temperature at floor level, which then increases with height.

Room set-ups do not have consistent effect on the systems` performance with regard to provide homogenous operative temperature field.

Chilled ceiling combined with mixing ventilation (CCMV) decreases the average operative temperature noticeably (by up to 1°C at some heights) for the high heat load conditions.

Requirements for operative temperature difference between head (1.1 m) and ankles (0.1 m) based on ISO Standard 7730 [2] for category B are fulfilled with all systems and for all studied cases.

Air velocity and draught

The highest air velocities occur at floor level and then decrease with height, and increases again between 2 m and 2.4 m for all systems and for all studied cases.

Lowest air velocities are provided by chilled ceiling combined with mixing ventilation (CCMV).

Highest air velocities are provided by mixing ventilation (MTVV).

Air velocities are higher under the high (design) heat load conditions than under the low (usual) heat load conditions.

Room set-up does not have significant impact on air velocities in the room.

The requirement for recommended draught rate for category B according to ISO Standard 7730 [2] is not fulfilled for more points under the high (design) heat load than under the low (usual) heat load.

The highest draught rates occur with mixing ventilation (MTVV) (draught rate is higher than recommended at up to 76% of all points measured), second highest occur for chilled beam (CB) (up to 47%) under the high (design) heat load conditions.

For the low (usual) heat load conditions points at which the draught rates exceed the recommended level are significantly less.

Chilled ceiling combined with mixing ventilation (CCMV) does not fulfill the requirements only at two points under the high (design) heat load conditions. For the low (usual) heat load conditions draught rate requirements are fulfilled at all measured points.

Cooling and warming effect for the body parts of the manikins

Chilled beam (CB) and chilled beam with radiant panels (CBR) were able to provide the strongest cooling effect (up to 2.5 K) and for the most body parts compared to the other systems.

The warming and cooling effect is felt better under the high (design) heat load conditions compared to that under the low (usual) heat load conditions.

More body parts are cooled at set-up 2 and set-up 4 where the manikins are sitting away from the window.

Body parts directly exposed to the window are felt as warmed while the body parts on the other side are felt cooler.

Vertical and horizontal equivalent temperature differences of the body

The equivalent temperature for the body parts that are closer to the window is always higher than for the body parts that are away from the window. During high (design) heat load conditions this difference exceeds 1 K, while for the low (usual) heat load conditions it is more than 0.5 K.

Equivalent temperature difference between top of head (crown) and feet for the chilled beam with radiant panels (CBR) is inconsistent – at some cases when the manikins are directly exposed to the CBR the cooling effect for the top of the head (crown) is observed, while for others it is not observed.

Radiant asymmetry

The highest radiant asymmetry is measured between window and door.

Requirements for the radiant asymmetry due to warm wall (window) for category B according to ISO Standard 7730 [2] are fulfilled.

7. References

1. Kalin V. Kostov, 2012, Design, balancing and test of active chilled beam with and without radiant panels, and chilled ceiling combined with mixing ventilation.
2. ISO 7730 (2005) Ergonomics of the Thermal Environment – Analytical Determination and Interpretation of Thermal Comfort Using Calculation of PMV and PPD Indices and Local Thermal Comfort Criteria, 3rd ed., International Standards Organization, Geneva.

3. EN 15251 (2007) Indoor environmental input parameters for design and assessment of energy performance of buildings addressing indoor air quality, thermal environment, lighting and acoustics, European Committee for Standardization, Brussels.
4. S. Tanabe et. al., E. A. Arens, F. S. Bauman, H. Zhang, T. L. Madsen, 1994, Evaluating thermal environments by using a thermal manikin with controlled skin surface temperature, ASHRAE Transactions, Vol. 100, Part 1.