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Highlights

- Two thermostat controls based on air/operative temperature were studied in offices.
- The two controls were compared in convective/radiant systems in three locations.
- Convective system with operative temperature control was better in thermal comfort.
- The controls had less impact on thermal comfort and energy use to radiant systems.
- Different thermostat control was suggested based on system type and orientation.

Using Thermostats for Indoor Climate Control in Offices: The Effect on Thermal Comfort and Heating/Cooling Energy Use

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Abstract

The most commonly used thermostat control variable in heating, ventilating and air-conditioning systems is air temperature. However, people's thermal comfort responds to operative temperature more directly than air temperature. Will the adoption of operative temperature based control lead to better thermal comfort and how will this affect the energy use? To get a better understanding about these questions, simulations have been performed based on three heating and cooling systems in three different geographical locations (Copenhagen, Paris and Rome). The three system are fan-coil system representing convective system and two radiant systems: floor heating/cooling system and radiant ceiling heating/cooling panel system.

The results showed that air temperature based thermostat control and operative temperature based thermostat control had different impacts on fan-coil system and radiant systems. For fan-coil system, the use of operative temperature based thermostat control had better thermal comfort conditions and higher energy use than that of air temperature based thermostat control. For the two radiant systems, the results were the opposite. The results were almost the same in different locations. Besides, the thermal comfort difference between the two controls of north office was smaller than that of south office. For fan-coil system, in south office, compared with air temperature based thermostat control, the hours of Cat. I ($-0.2 \leq PMV \leq 0.2$) increased 8.3% for building in Copenhagen, 8.8% for Paris and 14.2% for Rome and hours of Cat. IV ($PMV < -0.7$ or $PMV > 0.7$) decreased 2.5%, 3.9% and 7.1% respectively when operative temperature based thermostat control was used. Meanwhile, total energy supply increased 13.7% in Copenhagen, 14.3% in Paris and 12.7% in Rome. For radiant systems, the total energy use reduced 3.3% to 8.3% depending on location and type of system when operative temperature based control was used. With this reduction of energy use, thermal comfort in

south office was still within recommended criteria in international standards in most cases. Based on the results, it is suggested that air temperature based thermostat control be used in fan-coil system and operative temperature based thermostat control be used in radiant system in north office. For south office, operative temperature based thermostat control was considered better for fan-coil system and could be more energy efficient when utilized in radiant heating and cooling systems.

Key words Room temperature control, air temperature, operative temperature, energy use, thermal comfort, heating/cooling system, convective system, radiant system

1 Introduction

In HVAC (heating, ventilating and air-conditioning) systems the most commonly used parameter to control thermal environment in buildings is the air temperature (T_a). This is accomplished either by a thermostat installed in each zone (room) or by a 'central' thermostat controlling the conditions of several zones based on the average air temperature set point of those zones. Control of the air temperature alone is not sufficient in some situations. For example, during summer, people in perimeter zones of highly glazed buildings are generally uncomfortable because of radiation from sun or high temperature of the window surface. The effects of thermal radiation on thermal comfort have long been known, and were quantified by Hardy [1] and Gagge et al. [2]. The concept of "operative temperature" (T_{op}) was introduced earlier by Gagge [3] to account for the combined effect of mean radiant temperature (T_r), and air temperature. In all existing standards that deal with criteria for the indoor environment, requirements are expressed using Operative Temperature as room temperature [4-6].

The following equation is used to calculate T_{op} [4]:

$$T_{op} = \frac{h_c T_a + h_r T_r}{h_c + h_r} \quad (1)$$

Where h_c is convective heat transfer coefficient for the human body, $W/(m^2 \cdot K)$; h_r is linear radiative heat transfer coefficient for the human body, $W/(m^2 \cdot K)$.

In equation (1), h_c is influenced by air velocity. In most indoor office spaces with low air velocities (< 0.2 m/s), the equation can be simplified to:

$$T_{op} = 0.5 \times (T_a + T_r) \quad (2)$$

As T_{op} is more closely related to the influence of room temperature on thermal comfort compared to T_a , the use of T_{op} based thermostat control (TC- T_{op}) have been studied and compared with T_a based thermostat control (TC- T_a) in a few research papers.

Jain et al. [7] compared the effects of the two controls on energy use in perimeter zones of highly glazed buildings in New Delhi (India) through simulation. Their results had shown that energy use in highly glazed buildings can be under-estimated by the use of TC- T_a . In

simulation study, Julien et al. [8] found an increase of 18.7% on heating load when TC- T_{op} was compared with TC- T_a . Another simulation study recorded an increase of 7% in an office building in Hong Kong [9]. From these studies, the use of TC- T_a or TC- T_{op} would influence cooling /heating energy use.

Kontes et al. [10] studied two different buildings to study the effect of TC- T_a and TC- T_{op} on thermal comfort. Based on their simulation results, they confirmed the inability of widely-used air temperature control methodologies to provide a structured and systematic way of representing and controlling thermal comfort in buildings. They also concluded that simply adoption of TC- T_{op} could not ensure thermal comfort.

Currently, most studies are based on simulations. Theoretically, the basic difference between two thermostat controls, in simulation, is that during TC- T_a , T_a is maintained at a fixed design value (set-point) within a dead band, T_{op} is allowed to fluctuate. And it would be vice versa if TC- T_{op} is used [9]. The impact of TC- T_a and TC- T_{op} on thermal comfort and energy use would be small if the difference between T_a and T_{op} was small. For convective heating/cooling system where radiant surfaces (cool/warm windows etc.) are involved in rooms, T_a may be different from T_{op} . When radiant heating/cooling system are used, in summer the radiant cooler surface may compensate for the warm window [11,12], while in winter a heated surface may compensate for a cool window. This difference between convective system and radiant system may lead to different effect of TC- T_a and TC- T_{op} , which should be studied intensively.

This paper summarizes the results of dynamic computer simulations of energy use and indoor thermal environment of offices conditioned by means of different HVAC systems: a commonly used fan-coil system (FCS), floor heating/cooling system (FHCS), and radiant ceiling heating/cooling panel system (CHCPS). The potential influence on thermal conditions and energy use resulting from using the selected systems was quantified.

2 Methods

The simulations described in this paper were performed using the IDA ICE (version: 4.8) building simulation tool [13]. It allows simultaneous performance assessments of all building characteristics such as facade and wall construction, window glazing, HVAC systems, controls, indoor air quality, human thermal comfort and energy use.

2.1. Building model

The building model used in all simulations included a central module of an office building with two offices separated by a corridor (Fig.1). Both the corridor and each of the offices were treated as separate zones. The selection of the building module was based on earlier work by Olesen and Dossi [14], and also used in other simulations [15,16]. The total floor area used in modeling was 48.4 m². The building was a light construction with steel frame and thermal insulation. The thermal characteristics of the building components

are given in Table 1. The orientation of offices were north and south representing extreme comparison of solar radiation conditions. All spaces adjacent to the central module were assumed to have the same internal temperatures as the test space, therefore only heat transmission through external walls was considered.

Fig.1 Section of the room module (reproduced from Olesen and Dossi [14]). Dimensions in m.

Table 1 Thermal characteristics of the building components [15]

Building construction	Building components	Thickness (mm)	Density (kg/m ³)	Thermal conductivity (W/mK)	Specific heat capacity (Wh/kgK)	Emissivity dim. less	Overall u value (W/m ² K)
Floor/ceiling	Screed	45	2000	1.4	0.28	0.94	1.238
	Acoustic insulation	20	50	0.04	0.42		
	Concrete	180	2400	2.1	0.28		
Outside wall	Aluminum	2	2600	200	0.28	0.30	0.375
	Insulation	100	30	0.04	0.28		
	Aluminum	2	2600	200	0.28		
Internal wall	Plasterboard	25	900	0.21	0.28	0.82	0.482
	Insulation	60	20	0.04	0.28		
	Plasterboard	25	900	0.21	0.28		
Windows	3 pane glazing	Solar heat gain coefficient, dim. less				0.45	2.09

2.2. Locations and meteorological data

Simulations were performed for three locations in Europe, each representing a different climate: Copenhagen (cold winter and cool summer), Paris (less-cold winter and warm summer) and Rome (warm winter and hot summer). The Typical meteorological year design data from ASHRAE Fundamentals 2013 [17] was used as climate data input. Average monthly outdoor temperatures of the three cities are shown in Fig.2.

In this study, the dynamic simulation of each case was carried out throughout a whole year (from 1st Jan. to 31st Dec. 2018, considering eleven public holidays besides normal

weekends) to compare the different effects of the two control modes on the cooling and heating conditions.

Fig.2 Average monthly outdoor temperature.

Total internal heat load in the office zones was considered approximately 630 W/zone (32 W/m²), which was set according to reference [15]. The load included heat production by two occupants, present in the building during week days from 8:00 to 12:00 and from 13:00 to 17:00. Occupants' metabolic rate was considered to be approximately 1.2 met; that corresponds to about 238 W [5]. Clothing insulation was set to 0.5 clo for simulations corresponding to the cooling period and 1.0 clo during the heating period. Office equipment (two computers and a printer) were estimated to produce 350 W (long wave radiation fraction 50%) and ceiling light 50 W (convective fraction 50%) of heat. Heat loads from equipment and light had the same time schedule as occupants as it could be assumed that people switched off the light and their computers during lunch break. Moisture production by occupants was taken into account in the zone model, calculation of the moisture load was done according to EN ISO 7730 [5]. In addition, there was 100 W ceiling light as the only heat source in the corridor zone.

Windows in the office zones were equipped with only internal solar shading (light woven drape). When the solar radiation incident on an inside glass pane exceeded 200 W/m², internal drape would be activated. When the internal shading was drawn, the total shading coefficient of the window was multiplied by a factor of 0.71 and the short-wave shading coefficient of the window by a factor of 0.67. The effect of the internal blinds on the U-value of the window was considered by a coefficient of 0.87. The solar shading devices had simple two-position (on/off) control.

2.3. Description of the simulated systems

For each type of simulated HVAC system, the indoor units cooperated with a ventilation system with heat recovery (effectiveness was 0.6). The ventilation system operated on weekdays from 8:00 to 12:00 and 13:00 to 17:00 with lunch break. The supply airflow was 2 L/(s.m²), which was about 2.6 air change per hour (ACH), (to keep acceptable indoor air quality [18]) with constant supply temperature of 19 °C. The supply/exhaust airflow ratio was 1. Besides this mechanical ventilation, natural infiltration was set as 0.5 ACH under

pressure difference of 50 pa, which was $0.39 \text{ L/(s.m}^2\text{)}$ under this pressure difference. In the simulation, this natural infiltration would change according to the variation of climate file and would much lower than 0.5 ACH.

The temperature of offices and corridor was set to $25 \pm 1 \text{ }^\circ\text{C}$ for cooling and $21 \pm 1 \text{ }^\circ\text{C}$ for heating. The cooling would be turned on when the temperature is higher than $26 \text{ }^\circ\text{C}$ and be turned off when the temperature is lower than $24 \text{ }^\circ\text{C}$. The heating would be turned on when the temperature is lower than $20 \text{ }^\circ\text{C}$ and be turned off when higher than $22 \text{ }^\circ\text{C}$. There was sensor of T_a or T_{op} in each zone to achieve individual zone control.

(a) Fan-Coil System (FCS)

The main aim of FCS simulation was to compare the effect of two thermostat controls on a conventional system, which does not involve any radiant heating or cooling surfaces. The indoor unit of FCS had constant air volume with air flow (circulating air) rate of $5 \text{ L/(s.m}^2\text{)}$ for each zone. The supply air temperature would be adjusted according to the difference between the set point and tested room temperature. The fan-coils operated during occupied hours and were totally turned down during lunch break. The supply water temperature was $15 \text{ }^\circ\text{C}$ for cooling and $40 \text{ }^\circ\text{C}$ for heating. The delta T of water circulation was $5 \text{ }^\circ\text{C}$ under design power for cooling and $10 \text{ }^\circ\text{C}$ for heating under design power.

(b) Floor Heating/Cooling System (FHCS)

This is one of the typical radiant HVAC systems in buildings with water pipes buried in the concrete floors. In the simulation, water in the floor pipes circulated from 7:00 to 17:00 during weekdays (an hour earlier before occupancy) considering the slower heat transfer process of this type of system. In the water circuit of each zone, a three-way valve was used to control the mass flow. The supply water temperature for cooling was $15 \text{ }^\circ\text{C}$ and $40 \text{ }^\circ\text{C}$ for heating. The delta T of water circulation was $5 \text{ }^\circ\text{C}$ under design power for both heating and cooling. In the simulation the lowest floor surface temperature was $21.8 \text{ }^\circ\text{C}$ and there was no condensation.

(c) Radiant Ceiling Heating/Cooling Panel system (CHCPS)

This is another radiant HVAC system, in which water pipes run through suspended panels below the ceiling. In this system, the panel is not embedded in the building construction. The operation time was same as the FHCS. The set point of supply water for cooling was $19 \text{ }^\circ\text{C}$ and $40 \text{ }^\circ\text{C}$ for heating. The delta T of water circulation was $3 \text{ }^\circ\text{C}$ for cooling and $10 \text{ }^\circ\text{C}$ for heating at design power. Considering the direct contact of cooling panel with air, to avoid condensation the supply water temperature of CHCPS was set higher than FCS and FHCS.

2.4 Analysis of the results

The amount of the energy used to condition the indoor space in the building module was determined and compared for all simulated HVAC systems. In the analysis, only two parts of energy (kWh) delivered to a particular HVAC system were considered: (1) zone

heating/cooling – thermal energy supplied to indoor units; (2) ventilation system heating/cooling – thermal energy supplied to heating/cooling coils in air handling units to condition the air supplied to the zones. Since the main concern was to compare the effect of two thermostat controls, other energy cost by fans, pumps etc. was not included. The annual heating/cooling energy use per square meter of the space (kWh/m².yr) was calculated.

For one specific system in different locations, percent change of total cooling and heating energy use, ΔE , was calculated according to Eq.(3).

$$\Delta E = 100 \times \frac{E_{Ta} - E_{Top}}{E_{Ta}} \quad (3)$$

Where E_{Ta} is annual heating or cooling energy use (kWh/m².yr) with TC- T_a and E_{Top} is annual heating or cooling energy use (kWh/m².yr) with TC- T_{op} for a specific HVAC system. The PMV index was calculated by the simulation software based on air temperature, mean radiant temperature, air humidity and air velocity in the office zones, clothing insulation and metabolic heat production [5]. In the calculation of PMV, the velocity in zones was assumed to be 0.1 m/s. Statistics of the thermal comfort conditions was conducted to determine the ability of the simulated systems to keep the comfort limits specified by current standards (Table 2). The thermal comfort conditions were evaluated for both of the zones (north and south office).

Table 2 Recommended thermal comfort levels according to standards [5,6,19]

Category	Thermal state of body as a whole			
	PMV	PPD (%)	Operative temperature (°C)	
			During cooling	During heating
I	-0.2 to 0.2	<6	23.5-25.5	21.0-23.0
II	-0.5 to 0.5	<10	23.0-26.0	20.0-24.0
III	-0.7 to 0.7	<15	22.0-27.0	19.0-25.0
IV	<-0.7 or >0.7	>15	<22.0 or >27.0	<19.0 or >25.0

3 Results and discussions

3.1 Cooling and heating energy use

The annual heating and cooling energy of the simulated HVAC systems for three locations is summarized in Table 3.

Table 3 Annual heating and cooling energy use

Location	Type of system	Type of control	Cooling (kWh/m ² .yr)			Heating (kWh/m ² .yr)			Total cooling and heating (kWh/m ² .yr)
			Ventilation	Indoor units	Total	Ventilation	Indoor units	Total	
Copenhagen	FCS	TC-T _a	1.44	13.03	14.47	9.5	1.77	11.27	25.74
		TC-T _{op}	1.39	15.68	17.07	9.23	2.96	12.19	29.26
		Δ E	3.5%	20.3% ^c	18.0%	2.8%	67.2%	8.2%	-13.7%
	FHCS	TC-T _a	1.46	15.76	17.22	8.71	4.71	13.42	30.64
		TC-T _{op}	1.46	14.25	15.7	8.61	4.72	13.33	29.03
		Δ E	0.0%	9.6%	8.8%	1.1%	-0.2%	0.7%	5.3%
	CHCPS	TC-T _a	1.42	16.48	17.9	8.33	6.45	14.78	32.68
		TC-T _{op}	1.43	14.86	16.29	8.52	5.17	13.69	29.98
		Δ E	-0.7%	9.8%	9.0%	-2.3%	19.8%	7.4%	8.3%
Paris	FCS	TC-T _a	4.47	17.92	22.39	5.25	0.39	5.64	28.03
		TC-T _{op}	4.33	21.5	25.83	5.33	0.88	6.21	32.04
		Δ E	3.1%	-20.0%	-15.4%	-1.5%	-125.6%	-10.1%	-14.3%
	FHCS	TC-T _a	4.52	21.04	25.56	5.05	1.75	6.81	32.37
		TC-T _{op}	4.54	19.28	23.82	4.96	1.62	6.58	30.4
		Δ E	-0.4%	8.4%	6.8%	1.8%	7.4%	3.4%	6.1%
	CHCPS	TC-T _a	4.35	21.96	26.31	4.9	2.63	7.53	33.84
		TC-T _{op}	4.37	20.07	24.44	4.93	1.84	6.77	31.21
		Δ E	-0.5%	8.6%	7.1%	-0.6%	30.0%	10.1%	7.8%
Rome	FCS	TC-T _a	19.95	32.45	52.4	0.95	0.02	0.97	53.37
		TC-T _{op}	19.16	39.67	58.83	1.2	0.13	1.33	60.16
		Δ E	4.0%	-22.2%	-12.3%	-26.3%	-550.0%	-37.1%	-12.7%
	FHCS	TC-T _a	20.26	35.11	55.37	1.05	0.19	1.24	56.61
		TC-T _{op}	20.36	33.27	53.63	0.99	0.14	1.13	54.76
		Δ E	-0.5%	5.2%	3.1%	5.7%	26.3%	8.9%	3.3%
	CHCPS	TC-T _a	19.71	36.96	56.67	1.06	0.27	1.33	58
		TC-T _{op}	19.94	34.88	54.82	1.01	0.16	1.17	55.99
		Δ E	-1.2%	5.6%	3.3%	4.7%	40.7%	12.0%	3.5%

(^c Percent change of ΔE beyond the range of ±5% are printed in bold.)

To make the comparison more clearly, these data are also shown in Fig.3.

As expected that the lowest cooling energy use was observed in Copenhagen and the highest in Rome, while the highest heating energy use was in Copenhagen and the lowest in Rome. With a moderate climate, both cooling and heating energy in Paris were in the middle of the other two cities.

In general, for FCS less heating and cooling energy were supplied with TC- T_a for the three locations, while for radiant systems more energy was used with TC- T_a . For the three systems, the cooling energy difference between the two controls was higher than 5% when the building was located in Copenhagen and Paris. For Rome, the difference was less than 5% for all systems except FCS.

The difference of simulated heating energy use between the two controls was higher than 5% for FCS and CHCPS in Copenhagen and Paris, while for FHCS, the difference was less than 5%. In Rome, the heating energy use between TC- T_a and TC- T_{op} was larger than 5% for all systems. However, the total heating energy used in Rome was very small, less than 10% of that of Copenhagen, therefore this difference would not have significant meaning.

(a) Cooling energy use in simulated year

(b) Heating energy use in simulated year

Fig.3 Annual heating and cooling energy use

The energy use on typical days were further compared. Mondays and Wednesdays were chosen in winter and summer respectively, because Monday is the first day after weekend with heavy loads, while Wednesday is a mid-weekday with relatively moderate loads. The

hourly energy supply on 15th Jan. (Monday), 17th (Wednesday), 16th Jul. (Monday) and 18th (Wednesday) are shown in Fig.4. As ventilation system was turned off during lunch break (12:00-13:00), there were sudden drop of heating/cooling energy during this period. Besides, FCS was also turned off during this time and the fluctuation of FCS was more significant than radiant systems.

The change of heating or cooling energy between the two controls showed the same tendency with that of the whole year.

On 15th Jan., for FCS the total heating energy with TC-T_a was smaller than that of TC-T_{op}, and for FHCS and CHCPS the heating energy with TC-T_a became larger than that of TC-T_{op} in both Copenhagen and Paris. It could be noticed that the heating energy became smaller in the afternoon. In fact, in the afternoon heating energy was supplied only for ventilation (to heat the outdoor air to 19 °C). In Rome, heating energy was only supplied in the morning and in the afternoon cooling was needed. Because of the internal heat gains in the offices, the demand for heating would decrease when the outdoor temperature rises and the solar radiation increases in the afternoon. For a city like Rome, which has warm climate in winter cooling was, therefore, needed in the simulation.

On 16th Jul., for FCS the total cooling energy with TC-T_a was smaller than that with TC-T_{op} for the three locations. For FHCS and CHCPS the cooling energy with TC-T_a became higher than that of TC-T_{op} when the building was located in Copenhagen and Paris. When located in Rome, the cooling energy difference between the two controls could be neglected for radiant systems.

Compared to Mondays, on Wednesdays heating/cooling energy became smaller in the three locations.

Total heating on 15th Jan. (Monday)

Total heating on 17th Jan. (Wednesday)

Total cooling on 16th Jul. (Monday)

Total cooling on 18th Jul. (Wednesday)

(a) Copenhagen

(b) Paris

(c) Rome

Fig.4 Change of total heating/ cooling energy on typical days

3.2 Thermal conditions of offices

3.2.1 Statistics of thermal comfort

The statistics of thermal comfort conditions in simulated cases are shown in Fig.5. There are 2000 occupancy hours in the simulated year and the percentage of comfort levels or category (Cat.) according to EN 15251 [6] among occupied hours are shown on the left. From this figure, thermal comfort of north office was better than south office in all cases. North office had more hours of Cat. I and less hours of Cat. IV than south office in corresponding cases. For north office, the maximum hours of Cat. IV was 44 h in all cases, which was less than 3% of the occupied hours; for south office, the number was 214 h, almost 11% of the occupancy hours. The Cat. IV condition happened mostly because of overheating, e.g. the operative temperature was higher than 25 °C in winter and 27 °C in summer. In the building model, the north or south office had only one external wall and window and there were internal heat load during occupied hours. The solar radiation in the south office increases the risk of overheating. The south office would be more sensitive in response to the two controls. Based on this, south office would be discussed with more details.

(a) North office

(b) South office

Fig.5 Statistics of thermal comfort levels

From Fig.5 (b), with TC- T_{op} , thermal comfort of FCS became better than that of TC- T_a for all cities. Compared with TC- T_a , the hours of Cat. I increased 8.3% for building in Copenhagen, 8.8% for Paris and 14.2% for Rome. Meanwhile, the hours of Cat. IV decreased 2.5% for Copenhagen, 3.9% for Paris and 7.1% for Rome. For radiant systems, the results were just the opposite that thermal comfort was better with TC- T_a . There were more hours of Cat. I and fewer hours of Cat. IV with TC- T_a for both FHCS and CHCPS. The comparisons in the three cities showed the same results.

The European standard EN 15251 [6] suggests a methodology for long-term evaluation of the indoor environment. The standard accepts that indoor conditions can be outside the comfort range during 3–5% of the working hours per year. Taking into account upper limit later range – 5% of the working hours per year outside the Cat. III (normal level of expectation, PPD < 15%, corresponds to the Cat. C in EN ISO 7730 [5]), the north office met this requirement in all conditions. For south office, with FCS, when located in Paris and Rome, the proportion of Cat. IV hours was around 10% (8% in Paris and 11% in Rome). For all the other conditions, this proportion was less than or a little bit higher than 5% (with maximum of 5.7%, which happened to FHCS with TC- T_{op} in Rome).

This analysis indicated that for the FCS, the use of TC- T_a had poor thermal comfort condition in south office of Paris and Rome, while with TC- T_{op} thermal comfort can be improved effectively. For radiant systems, the thermal comfort condition with TC- T_a was better than that with TC- T_{op} .

3.2.2 T_a and T_{op} in south office

The temperatures in the south office were compared further to explain the effect of two controls on FCS and two radiant systems. Fig.6 shows the change of T_a and T_{op} in south

office on typical days. The sudden change from 12:00 to 13:00 was caused by the lunch break.

During cooling, on 16th Jul., for FCS, T_{op} was higher than T_a in three locations. In summer, surface temperature of external wall and window in south office was higher than T_a , which led to higher mean radiant temperature and thus caused T_{op} higher than T_a . With TC- T_{op} , T_a became lower than that of TC- T_a . And more cooling energy should be supplied and thermal comfort would be better with TC- T_{op} . For the two radiant systems, T_a was higher than T_{op} . The radiant system using cooling floor or ceiling, which has lower surface temperature than T_a , brought about lower mean radiant temperature. Thus with TC- T_{op} , T_a was higher than that of TC- T_a for radiant cooling systems. And less cooling energy could be supplied with TC- T_{op} for these systems.

During heating, on 15th Jan., in Rome, the change of T_a and T_{op} for the two controls in three systems were similar to that in summer, especially in afternoon when cooling was supplied. The T_a / T_{op} in south office were close to 25 °C, which meant that this office did not have a need for heating.

When the building was located in Copenhagen and Paris, T_a was higher than T_{op} using FCS, which made T_a of TC- T_{op} higher than that of TC- T_a . Therefore, more heating energy would be supplied using TC- T_{op} . In these two locations, for FHCS, T_a was slightly higher than T_{op} and there was almost no difference between two controls. As floor was heated, the higher floor surface temperature could compensate the low temperature of other surfaces and increase mean radiant temperature. Therefore, the difference between T_a and T_{op} was small and the influence of two controls became similar too. This tendency was also true for CHCPS in Paris. For CHCPS in Copenhagen, results were slightly different on Monday. As Copenhagen was cold in winter, T_a and T_{op} were about 18°C before occupancy, the ceiling panels would be heated at startup time. Because the surface temperature of ceiling panel was high (around 32°C), the increase of T_{op} was faster than T_a . Thus, with TC- T_{op} the heating time was shorter than that of TC- T_a (see Fig.4 (a)). And less heating energy was supplied by TC- T_{op} . With TC- T_a , T_a and T_{op} became higher than that of TC- T_{op} in the afternoon. This was why more heating energy was supplied to CHCPS with TC- T_a during heating.

Compared with Monday, T_a and T_{op} on Wednesday showed the same trend (though with less variation) in both heating and cooling season.

On 15th (Monday) Jan.

On 17th (Wednesday) Jan.

On 16th (Monday) Jul.

On 18th (Wednesday) Jul.

—□— Ta, TC-Ta —○— Top, TC-Ta —△— Ta, TC-Top —◇— Top, TC-Top

(a) Copenhagen

On 15th Jan.

On 17th (Wednesday) Jan.

On 16th (Monday) Jul.

On 18th (Wednesday) Jul.

—□— Ta, TC-Ta —○— Top, TC-Ta —△— Ta, TC-Top —◇— Top, TC-Top

(b) Paris

On 15th Jan.

On 17th (Wednesday) Jan.

Fig.6 Change of T_a and T_{op} in south office during occupied hours on typical days

Fig. 7 shows the mean value of T_a , T_{op} of every month (during occupied hours) and the temperature difference between T_a and T_{op} in south office, which can reflect the impact of different control methods on temperatures with the change of seasons.

From Fig.7 (a) and (b), the variation of T_a and T_{op} throughout the year were the most significant in Copenhagen, the changes became smaller in Paris, while Rome had the smallest changes. This was due to the climate difference in three locations.

For FCS, with TC- T_a the average T_{op} and T_a became lower than that of TC- T_{op} in heating (in Copenhagen and Paris) and changed to the opposite during cooling in all locations.

For FHCS, during heating average T_a were basically the same under the two controls, and the difference between T_{op} was also negligible (in Copenhagen and Paris); during cooling with TC- T_{op} both T_a and T_{op} tended to be a little higher than that of TC- T_a in the three locations.

For CHCPS, during heating (in Copenhagen and Paris) with TC- T_a the average T_{op} and T_a were higher than their counterpart of TC- T_{op} and turned to be opposite in cooling period in all cities.

It can be noticed that in Copenhagen the difference of average T_{op}/T_a among the six cases in April and October became smaller than that of other months. This was because these two months were mild compared to others and the building had little demand for cooling and heating. This situation can also be found when located in Paris, in which the two mild months were February and November.

It can also be noticed that in all locations T_a of FCS with TC- T_{op} was the lowest, while T_{op} of CHCPS with TC- T_a was the lowest during cooling.

Fig.7 Monthly change of T_a , T_{op} in south office

According to Fig.7 (c), the difference between T_a and T_{op} differed between FCS and radiant systems. For FCS, T_a was higher than T_{op} during heating (in Copenhagen and Paris), while T_{op} was higher than T_a during cooling for both controls in all locations. For radiant systems, T_a was higher than T_{op} for the whole year. This was one of the difference between convective system and radiant system. As it had been stated before, when convective system was used, cool window would lead to lower T_{op} in winter, while warm window would result in higher T_{op} in summer. As to radiant system, it had heating surface to compensate cool window in winter and cool surface to compensate warm window in summer. Coexistence of cool and warm surfaces in radiant systems caused this difference. Besides, the difference between T_a and T_{op} was around 0.3 K for the two radiant systems, and the change with months and locations were not significant.

3.2.3 PMV index

The change of PMV in the south office on typical days are shown in Fig.8.

As it had been illustrated in Fig.6 and 7 that T_a and T_{op} in Rome were the highest among three cities during heating, the PMV in Rome was also the highest on winter days and was above zero for most of time. The PMV difference among six cases was small in Rome.

For Paris, on typical heating days PMV became smaller and was around zero for most cases. When FCS was used, with TC- T_a PMV was the lowest among the six simulations. For radiant systems, the PMV difference between the two controls was almost invisible. When located in Copenhagen, there was a relatively larger difference of PMV among the six simulations in January. CHCPS with TC- T_a had the highest PMV among all cases, while FCS with TC- T_a had the lowest. The PMV difference between the two cases was about 0.4. For CHCPS, PMV was closer to zero with TC- T_a than with TC- T_{op} . For FHCS, the PMV difference between two controls was negligible. FCS would have better thermal comfort index when TC- T_{op} was used.

For cooling time, on typical days of Jul., radiant systems had lower PMV with TC- T_a and FCS with this control had the opposite result. For same location, the PMV difference among the six cases was small except in Rome. CHCPS with TC- T_a achieved lowest PMV for most conditions.

The change of PMV on Wednesdays were similar to that on Mondays in the three cities. The analysis of PMV showed the similar results with the analysis of T_a/T_{op} based on typical days.

(a) 15th Jan. (Monday)

(b) 17th Jan. (Wednesday)

(c) 16th Jul. (Monday) (d) 18th Jul. (Wednesday)

Fig.8 Change of PMV index on typical days in south office

4 General discussion

As expected, the energy demand of each simulated system differed markedly among the tested locations. For same location, the energy use of three different systems shows some difference. For the same system under different control modes, energy use was not identical. In the same location, if combining the energy use with thermal comfort, it could be found that energy use can be connected with thermal comfort conditions. The corresponding cooling/heating energy supply will increase in the system or control mode that achieves better thermal comfort conditions. In the three systems, the control mode leading to better thermal comfort consumed more cooling and heating energy than the other one. For example, thermal comfort of FCS was highly improved using TC- T_{op} compared with TC- T_a . Meanwhile, with TC- T_{op} both heating and cooling energy were increased in all locations for FCS system. The effect of TC- T_a and TC- T_{op} on thermal comfort and energy use in the three climate conditions was almost the same.

In the analysis, much of the time that south office was out of comfort range was due to overheating. In the simulation, windows were kept closed all the time. In practice, night ventilation in hot season or natural ventilation during mild season can be applied to improve comfort and reduce energy use [20]. Especially for Rome, which had needs of cooling in winter, natural ventilation could completely meet the needs. When combined with natural ventilation, the comparison of the two control modes should be studied further.

The results showed that non-radiant system of FCS had better thermal comfort conditions when TC- T_{op} was used in all locations, while for radiant systems, thermal comfort will be improved using TC- T_a . During heating, FCS heated the air directly and T_r was lower than T_a , while for cooling T_r was higher than T_a . Under the same set point, T_a would be higher in heating and lower in cooling with TC- T_{op} than that with TC- T_a , thus the overall comfort was improved and more energy should be supplied. Radiant systems had a high temperature surface in heating and a low temperature surface in cooling, which caused difference compared to FCS. The simulation software calculated one T_r and T_{op} in a space. In practice, when the sensor is located in different positions (such as close to the window, radiant surface or be under solar radiation directly), the measured T_r and T_{op} would be different. This may bring influence on the control of system and should be studied in the future.

The influence of control methods was more salient to south offices. South office is exposed to solar radiation that affects T_r , which will further influence T_{op} . Even if the south and north office has same T_a , T_{op} would be different. That is why the use of T_a and T_{op} sensors in control will have more significant impact on south offices. As to north office, the difference between the two controls were quite small. Based on this, different thermostat sensors can be considered for the control of the south and north spaces.

In the building model, the window occupied 30% of the total external wall in each office. When this ratio is changed (for example, external wall is constructed by glass curtain wall with ratio of 100%) it would affect T_r and T_{op} in related spaces. The effect of two controls on energy and thermal comfort may be different [7]. Therefore, this factor should be taken into account in future studies.

In the simulation the radiant systems were turned on an hour earlier than FCS on weekdays considering that radiant systems might heat or cool the space slower than FCS. However, the energy use of FCS and radiant systems on typical days (Fig. 4) showed that heating/cooling energy was supplied at similar increasing trend at the beginning of working hours, which implied that it may be unnecessary to turn on the radiant system earlier.

Moreover, heating supply on 15th Jan. in Copenhagen showed some difference between the two radiant systems. T_{op} of CHCPS grew faster than T_a with both controls, while T_{op} and T_a increased similarly for FHCS. This was caused by the heat transfer difference of these two systems. FHCS releases more convective heat as heat goes upward naturally during heating and both T_a and T_{op} increase at the same time, while CHCPS has less convective heat with downward warm surface and T_a cannot be heated as quickly as T_{op} at the beginning of the system. Therefore, FHCS usually works better than CHCPS for heating. During cooling, the supply water temperature of FHCS was set as 15 °C and 19 °C for CHCPS. In the simulation, the lowest temperature of cooling surface was 21.8 °C for FHCS and 21.3 °C for CHCPS, which meant that ceiling panels can provide

better cooling than floor surface. This explained the result that CHCPS had better thermal comfort conditions than FHCS under the same control conditions during cooling, especially when located in Rome.

5 Conclusions

The results of simulation showed that thermostat control based on air temperature and thermostat control based on operative temperature had different impact on fan-coil system and radiant systems.

- For fan-coil system, the use of thermostat control based on operative temperature was better for thermal comfort; but with higher energy use than that of thermostat control based on air temperature. When located in Paris and Rome, the application of thermostat control based on operative temperature can reduce hours of Cat. IV to less than 5% in south office. Meanwhile, total heating and cooling energy increased 14.3% in Paris and 12.7% in Rome.
- For radiant systems, the use of thermostat control based on operative temperature will save heating and cooling energy (3.3% to 8.3% depends on location and type of system) compared to the use of thermostat control based on air temperature. In south office, for floor heating/cooling system and radiant ceiling heating/cooling panel system, the thermal comfort condition with thermostat control based on operative temperature was within recommended criteria in international standards in most cases.
- The analysis indicated that the influence of two controls on thermal comfort were similar to north and south offices. However, the two controls had more influence on thermal comfort to south office than to north office.
- For all simulation cases, the thermal comfort of north office satisfied the demand of Cat. III and Cat. II in international standards.
- From the point of view of energy saving, thermostat control based on air temperature can be used in fan-coil system and thermostat control based on operative temperature be used in radiant system in north office.
- For south office, the selection of thermostats should satisfy the requirement of thermal comfort and then energy saving can be considered.
- Consequently, for south office thermostat control based on operative temperature is considered better for fan-coil system and can be more energy efficient when utilized in radiant systems.

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Conflict of Interest Statement

We declare that we have no financial and personal relationships with other people or organizations that can inappropriately influence our work, there is no professional or other personal interest of any nature or kind in any product, service and/or company that could be construed as influencing the position presented in, or the review of, the manuscript entitled.

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