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Field measurements and numerical analysis on operating modes of a radiant floor heating aided by a warm air system in a large single-zone church

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ABSTRACT

Space heating can constitute 60–80% of the total energy use of buildings in cold climates. Efficient heating techniques in buildings still rely on operating strategies. In this paper, a church with radiant floor heating in a cold climate is taken as a case of a large single-zone building to analyze the energy use for heating. Field measurements and numerical analysis are both used in the study. Different operating modes of heating, including intermittent heating and constant set-point heating, are compared for energy saving, reliability on indoor climate, and thermal comfort. The intermittent heating by an all-air system with supplied air temperature control results in the highest energy use. The constant set-point air temperature radiant floor heating aided by a warm air system (return air temperature control) is least affected by outdoor temperature with the best reliability and met the thermal comfort requirements throughout the heating season. The main novelty is that an operating mode of cyclic set-point air temperature is proposed. It is found that the small thermal inertia of heating systems should be preferred when the operating mode of cyclic set-point temperature is used to reduce the warm-up period. The results suggest how to operate and reduce the energy use of radiant floor heating systems in a large single-zone building.

1. Introduction

Buildings worldwide account for approximately 30% and 40% of the total final energy and primary energy use, respectively [1]. In cold climates, space heating constitutes the majority of energy use in residential and non-residential buildings. Buildings for religious activities such as churches are one category of non-residential buildings, which consume a large amount of energy. Thermal comfort and energy-saving have always been a concern throughout the history of church heating technology development. Initially, iron ovens were used for space heating and to raise the thermal comfort for sermons [2]. With the emergence of low-pressure steam technology and the implementation of fire protection regulations, iron stoves were gradually replaced. In the 1920s or 1930s, when the medieval or early modern churches were restored in Sweden, central heating was installed for the first time in the country [3]. The churches were heated intermittently for energy saving, often unheated for five or six days a week. However, it was found later that the intense heat from the large radiators mounted on walls causes dust particles to stick to the wall surfaces and damage the mural paintings [2]. Electrical heating was considered a technically and aesthetically feasible way of heating a church at that time, as compared to a stove or a central heating system using firewood, even if it was not ideal from the point of view of operational costs [4].

Without appropriate building designs and efficient heating, ventilation, and air-conditioning systems, large space buildings exhibit poor thermal comfort and high energy use [5]. Different heating systems and devices such as convector heating units, warm-air heating, pew heating, and radiant floor heating are used for space heating in church buildings. Convector heaters can deliver enough heat, but the evolution in the relative humidity may increase the risk of damaging the historical works of art [6]. Warm-air intermittent heating systems were often installed when renovating churches until December 2012 (European standard UNE-EN 15759-1) [7,8]. During field measurements, it was found that hot-air systems induce wide fluctuations in indoor thermal–hygrometric (T/RH) conditions, in turn translating into a stratification of temperature and relative humidity. As a result, the air remains cold at pew height while heat accumulates in the upper

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surface of the church space [8,9]. Pew heating provides heating of the local area of the pews by convector or pipe heating [10]. This kind of heating system provides heat directly, without causing large periodic air temperature and relative humidity changes. Thus, it can reduce the risk of mechanical stress in wooden artworks and panel or canvas paintings, fresco soiling, and cyclic dissolution-recrystallization of soluble salts in the masonry, and it is energy-efficient due to its low thermal inertia and relatively fast warming up period [11,12]. To evaluate the local pew heating, Camuffo et al. [11] defined a skin temperature (T) considering the desired level of thermal comfort. A mean skin T, of 30–33 °C corresponds slightly cool to neutral thermal sensations [11]. However, for churches in a severe cold climate, local area heating of pews may cause cold walls and ceilings. Thermal asymmetry also needs to be evaluated for pew heating in cold climates.

Radiant floor heating (RFH) is considered to have better thermal comfort [13,14], and when used in large spaces, the vertical temperature distribution has good uniformity [15]. This is because, in RHF, the floor which is the source of heat radiation has a better angle factor than other surfaces (walls, ceiling, windows, etc.) [16]. Hence all inner surfaces of the room can be heated relatively evenly using RHF. In a Danish church, RHF is installed to maintain thermal comfort at a reduced temperature and the large windows are double glazing installed to prevent convective air movements [17]. These changes have improved the indoor thermal climate in that church, so the annual average variation in RH is limited to 30%–60% with little diurnal fluctuation [17]. A low-temperature electric RHF is also used in Salepçioğlu mosque [18]. The simulation results indicate that the use of RHF has substantially increased the thermal comfort of the main worship space in the mosque [18]. In practice, the RFH has evolved from heavy radiant floor heating to light-weight radiant floor heating (LRFH) [19–21]. Compared with the traditional heavy RFH, the LRFH does not have a concrete layer. (The detailed constructions can be found in reference [21,22]). Therefore, LRFH has the advantages of convenient construction, lightweight, faster room heating, and compactness. Further, as compared to heavy radiant floor heating, LRFH has a lower thermal resistance [22].

Though the heating devices have become more efficient, church heating systems still have the problem of high energy use [23]. Another issue faced by churches is the drastic variation of relative humidity, which caused desiccation, warping, and paint flaking on wooden surfaces arts, pipe organs, and cultural relics collection [23,24]. Permanent heating church buildings can prevent the problem of relative humidity, but continuous heating will result in unnecessary use of energy, as it is sparsely occupied for the majority of the time, and infrequently used to its maximum occupancy [25].

Due to the large differences in the construction, geometry, glazing, usage, location, and monumental value of church buildings, there is no ‘one size fits all’ solution and each building needs to be considered independently for devising optimum solutions for creating a suitable indoor climate and saving energy of HVAC systems [26,27]. Different churches have different emphases on the indoor climate. In churches with historic wooden organs, frescos, or relics such items are often protected at the expense of visitors’ thermal comfort [28]. Condensation and deformation of pipe organs are more vital problems for such churches [29]. However, in church buildings which does not have components/equipment of historical significance, the HVAC system should be energy efficient while providing thermal comfort to its occupants [30]. The challenge could be enhanced in old and listed buildings, which are part of the cultural heritage. When attempting to make such buildings more energy-efficient, a non-invasive approach is often needed to ensure these heritage values are sustained for future generations [31]. Intermittent heating [32] dehumidification, and adaptive ventilation strategies [23] have been studied in specific churches.

In this work, the energy use and the thermal comfort of a protected church building for different operating modes of RFH aided by a warm air heating system are compared and analyzed. A cyclic set-point temperature operating mode is proposed and additionally evaluated for energy-saving possibilities.

2. Method

2.1. Description of the church and local climate

Teg’s church (Tegskyrkan), a church in northern Sweden, is used as a case study to investigate the energy use and the occupants’ thermal comfort. Construction of the church started in 1964, and the building was inaugurated in 1969. Largely unaltered, the church due to its unique architecture, has been designated as a cultural heritage and is a protected building. It is a modernist church made of concrete. The church is almost square-shaped and has a choir oriented towards the courtyard to the south. The building space is a combination of multiple interconnected cubes, with a total floor of 27.0 m $\times$ 24.0 m. The highest ceiling in the church is above the altar with 20 m, while the height of the main hall is 16 m. The total window area is 133.5 m^2, and the window to wall ratio of the building is 8.4%. The geometry and dimensions are illustrated in Fig. 1(a) – (b). The church can accommodate up to 500 visitors. The church is divided into two occupied zones. The first is the main occupied zone, which is in the main hall. The second is about 40 m^2 and located on the second floor, where seats are provided. The church is open from 8:30 to 16:00, except on Thursday and Saturday. The church is open till 20:00 on Thursdays for evening service while it is closed on Saturdays. On a normal weekday, depending on climate, 1 – 20 people visit the church, while 30 – 40 people attend the Sunday service. The church occasionally hosts large events such as concerts when it could be occupied by 300 – 500 people.

Although the church location is subarctic, the outdoor climate is maritime, due to the proximity to the sea. It makes winters and summers relatively mild. The annual average temperature is 3.84 °C, and the annual average relative humidity is 77.5%. The heating period is often 8 months stretching from September to May. During the heating season, the average outdoor air temperature is 0.30 °C. The monthly average temperature varies between −10 °C and 0 °C during the winter months and +12 °C and +20 °C during the summer months [33]. Field measurements and numerical simulations are carried out to study the energy use and indoor climate of the church (see sections 2.3 and 2.4).

2.2. Building envelops and HVAC facilities

The envelope of the church consists of concrete with insulation materials. The windows are double glass, and the window frame is aluminum. The airtightness data for the church is not available. The thermography images showed some small leakages around the main door (Fig. 2a). Swedish National Board of Housing, Building and Planning’s (Boverket) regulations from 2002 stipulates the airtightness of residential buildings and other buildings as 0.8 L/(s m^2) and 1.6 L/(s m^2), respectively [34]. The airtightness of historic churches in Sweden that were built in 1792–1907 is reported to vary from 0.59 L/(s·m^2) to 4.24 L/(s·m^2) at 50 Pa pressure difference [35,36]. Taking into account the air leakage shown in thermography images, the large volume of the building and the airtightness of historic churches mentioned earlier, the airtightness of Teg’s church is assumed to be not lower than 2.4 L/(s m^2) (equivalent to 0.88 ACH). Electrical RFH is used to heat the building. Electric

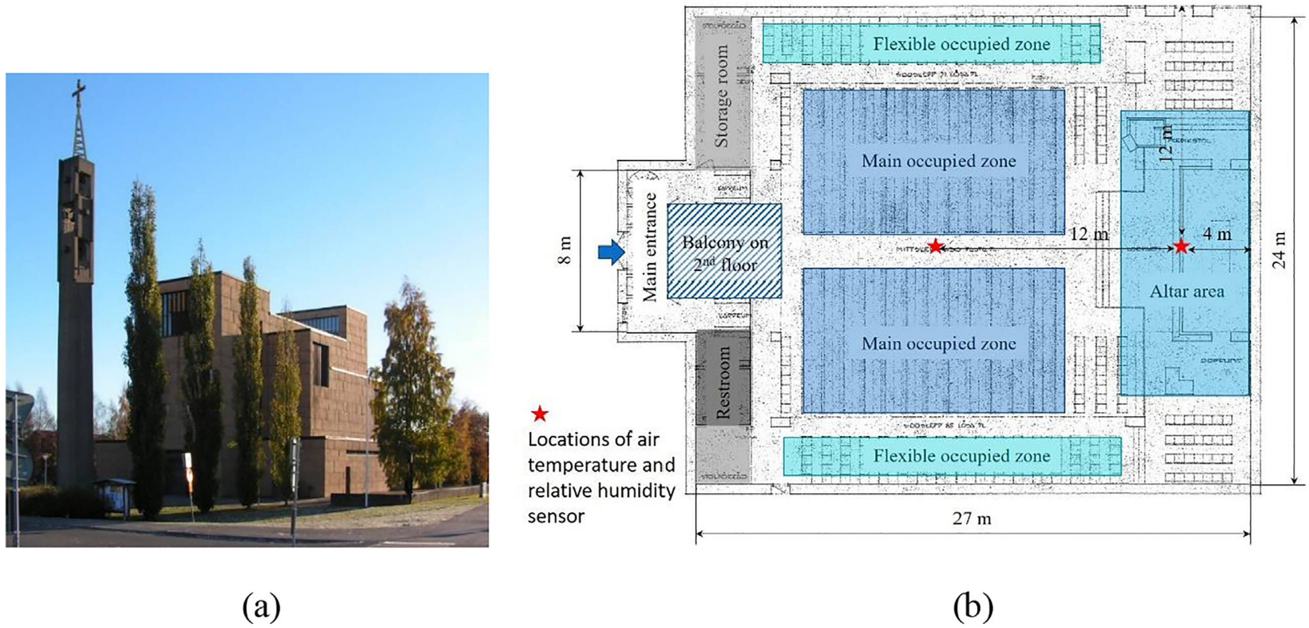

Fig. 1. Pictures (a) and plan view (b) with dimensions of the church (unit: m).

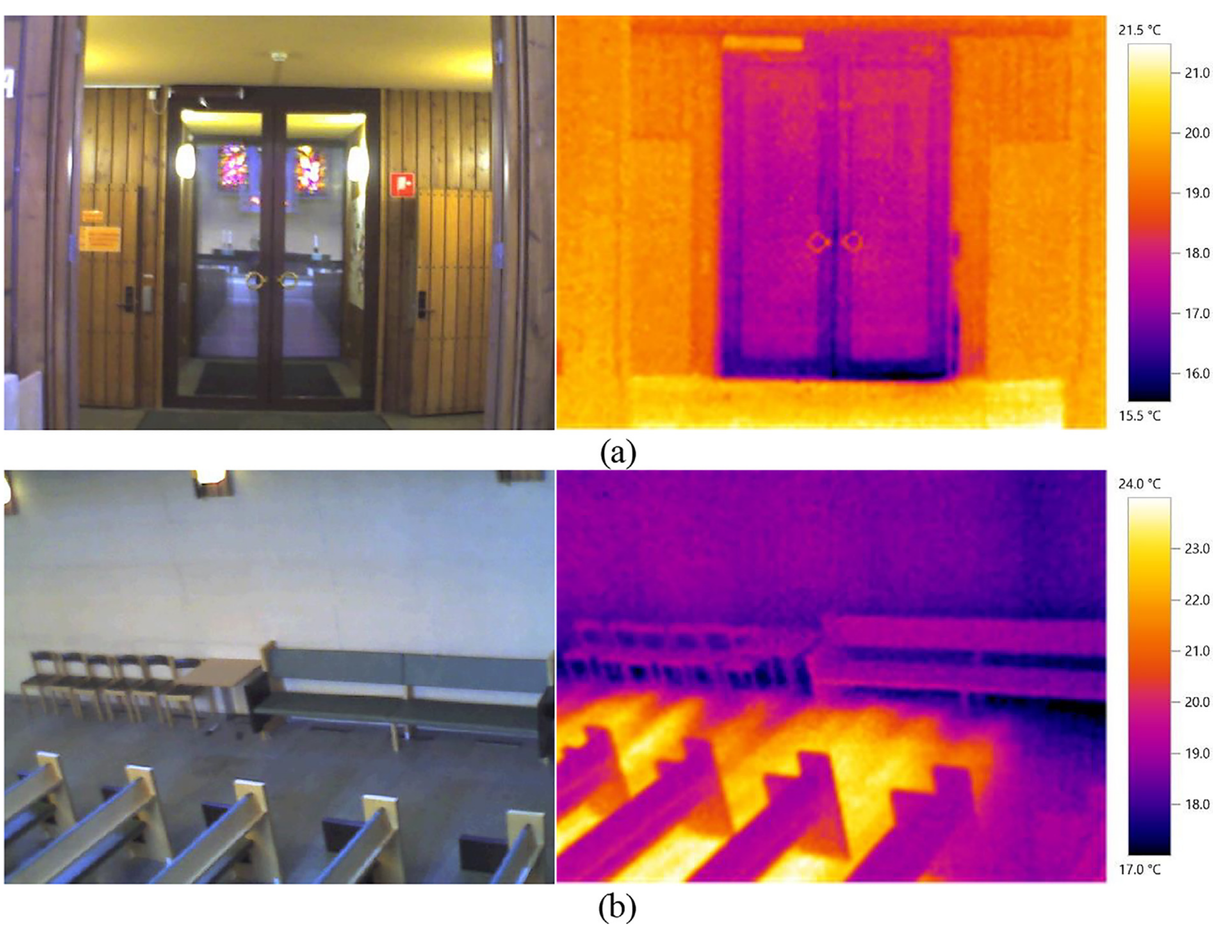

Fig. 2. Main door and floor heating units and thermography images.

cables for heating are placed under the floor, the edges of which can be seen through the thermography image, shown in Fig. 2(b). To meet the fresh-air requirements, the church also has a warm air system with an air supply at floor level and return on opposite sides on the floor. The RFH with the warm air system is similar to the radiant floor heating system with displacement ventilation

Table 1
Measuring instruments and its technical data.

Function	Type	Accuracy	Range
Temperature	RTR-52A	±0.3 °C	−20 – +80 °C
Thermography	Testo 875	±2% of mV	−30 – +100 °C
Mean radiant temperature	Testo Globe thermometer TC type K	Class 1 of standard EN 60584-2	0 – +120 °C
Relative humidity	ERS	±10%	0–100%
Air velocity	Testo Thermal velocity probe	± (0.03 m/s +4 % of mV)	0–20 m/s

Table 2
Envelope materials.

Material and thickness (mm)	Average U-value [W/(m^2·K)]	Total area (m^2)
Wall (concrete 100 mm + mineral wool 150 mm + concrete 250 mm)	0.22	1673
Floor (floor coating 5 mm + concrete 50 mm + light insulation 100 mm + concrete 150 mm)	0.29	688
Roof (light insulation 150 mm + concrete 150 mm)	0.23	688
Window (double-glazed)	2.9	141

(RFH/DV). The capacity of the warm air system is 1800 m^3/h. The supply air temperature is kept approximately at 17 °C to avoid vertical air temperature stratification. There are three exhaust fans with a total air volume of 290 m^3/h for the visitors on the second floor.

In recent years, the church is only heated by the RFH with the three exhaust fans in operation. The warm air system for fresh air has not been used since 2010. The RFH is controlled manually by the operational staff and is based on the data from the air temperature sensor in the church. The building manager reported high HVAC energy use and dissatisfaction with the thermal comfort. Heating by the RFH with exhaust fans on is referred to as the current daily operating mode, which is used as the reference case in this study.

2.3. Field measurements

Various indoor climate parameters of the church such as air temperature, relative humidity, and air velocity were measured from 1st February to 31st August. The outdoor dry-bulb temperature and relative humidity were collected from measurements of the previous ten years. The air temperature and relative humidity were measured mainly for validating the simulation. The measurements were carried out at two locations in the church, (i) at the center of the main occupied zone and (ii) at the center of the altar, as shown in Fig. 1(b). At the center of the main occupied zone, measurements were conducted using a globe thermometer, anemometer, air temperature, and relative humidity sensors. These measuring instruments were situated 22 m from the front wall. Air temperature and relative humidity sensors were at the centreline of the longitude direction (12 m from both sidewalls). The anemometer and the globe thermometer were on both sides of the air temperature and relative humidity sensors, respectively 0.5 m from the air temperature and relative humidity sensors. At the center of the altar, only air temperature and relative humidity sensors were situated. This measurement position was 4 m from the front wall and 12 m from the sidewalls. Parameters were measured at a height of 1.5 m. Table 1 summarized the measuring instruments and their type, accuracy, and measuring range.

2.4. Numerical simulations

The energy use of the church was simulated by IDA-ICE software. IDA-ICE takes outdoor meteorological parameters, geometric dimensions and physical properties of building envelope, occupants, equipment and other heat sources, and forms of an HVAC system such as hot water heating systems and related parameters as input variables. The envelope materials and schedule of occupants are provided in Table 2 and Table 3, respectively. When the measurements were conducted, the current daily operation mode of HVAC facilities was chosen, wherein only the floor heating and the exhaust fans worked simultaneously. In this way, the measuring results can be matched with the actual energy use records, which facilitates the parallel comparison and verification of energy use and indoor climate results during numerical model verification. IDA-ICE calculates energy use based on the assumption that the room temperature is uniformly distributed. The vertical temperature stratification in large spaces may cause errors in the simulation results. However, Teg's church uses the RFH and according to studies, a relatively uniform vertical temperature distribution in the room is achieved when RFH is used for heating [15,37]. Therefore, even though the church has large vertical space, IDA-ICE can still be used.

The meteorological data (ASHRAE IWEC2) for simulation are imported into the model. Meteorological data in IDA-ICE is from ASHRAE's International Weather for Energy Calculations 2.0 (IWEC2). ASHRAE IWEC2 provides 'typical year' weather files of 3012 international locations specifically for building energy calculations. There are observations for each IWEC2 location, at least four times per day of wind speed and direction, sky cover, and dry-bulb and dew-point temperature for 12–25 years. Meteorological data is one of the important input variables of energy simulation. To verify the meteorological data used in the numerical simulation, the recorded data for the past ten years measured by the weather station at Umeå University was also used. The coldest month is February, while the warmest month is July. The differences between IDA and ICE and climate station records for the monthly average outdoor dry-bulb temperature are 0.02–1.96 °C

Table 3
Occupants' schedule.

Working hours	Number of people visiting the church
Monday–Friday: 8:30–16:00	1–5 in winter, 10–20 in summer
Thursday: 19:00–20:00	10–15
Sunday: 11:00–13:00	30–40
Large event (occasionally in a year)	100–500

*The church is closed on Saturdays. The lights are switched on from 08:30–16:00 on all weekdays, while on Thursdays it is switched on till 20:00 and on Sundays, the lights are on from 09:00–15:00.

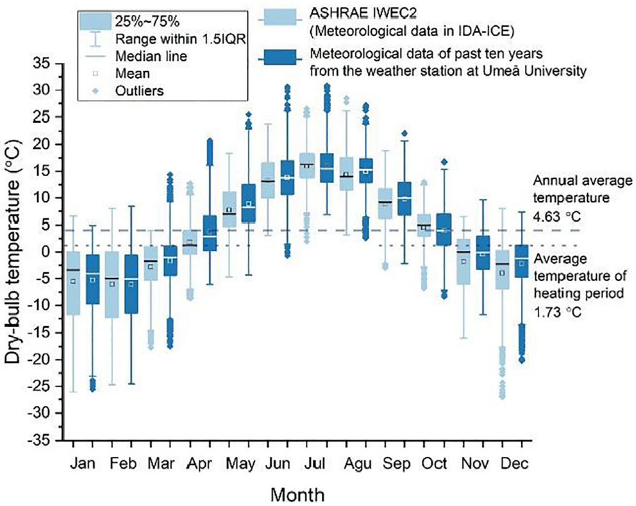

Fig. 3. Outdoor dry-bulb air temperature.

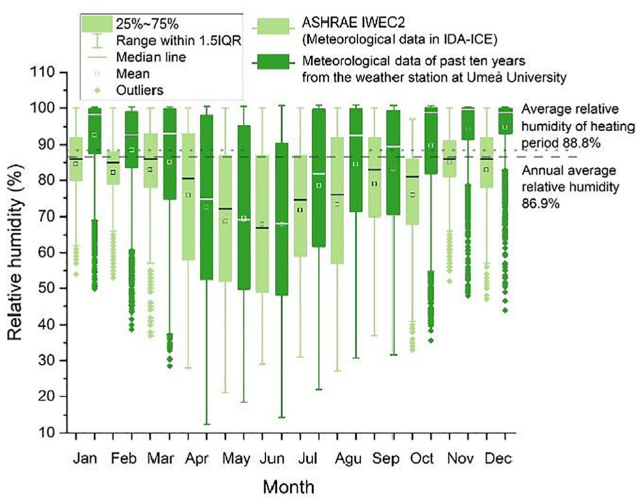

Fig. 4. Outdoor air relative humidity.

(Fig. 3). For the monthly average outdoor relative humidity (Fig. 4), the differences between IDA and ICE and the climate station records are 0.1% – 13.8%. The average humidity is highest in November, while June is the driest month. Based on the above data, the Root Mean Square Error (RMSE) between the meteorological data in IDA-ICE and the weather station data from Umeå was found to be within 2 °C for the outdoor temperature and 15% for relative humidity. Hence ASHRAE IWEC2 data is used in the simulation.

2.5. Source of energy use and heat balance in the building

The actual energy use (measured electricity use) of the church for the previous ten years is taken from the energy utility company's records. In practice, the outside climate doesn't always follow the 'typical year'. The RFH is controlled according to the variation of the outdoor weather to keep a constant indoor air temperature. Thus, the electricity use for the RFH varies from year to year. To minimize the impact of manual interferences on indoor temperature settings and the uncertainty of weather changes, average values of energy use for each month (average of ten years) are used. Besides, the set-point air temperature of 19, 20, 21, 22 °C is respectively simulated based on the current operating mode. The

Table 4
Conditions of numerical simulations.

Operating mode	Heating system	Average U-value of windows [W/(m^2·K)]	Floor heating Set-point temperature (°C)	Warm air Supplied air temperature (°C)	Return air temperature (°C)	Fresh air rate [L/(s·m^2)]
Case 1*	RFH with exhaust fans	2.9	Air temperature at 20 (Duration: 24 × 7)	–	–	0.12
Case 2	RFH with exhaust fans	1.9	Air temperature at 20 (Duration: 24 × 7)	–	–	0.12
Case 3		1.4				
Case 4		0.6				
Case 5	All-air system	2.9	–	18 – 55	20 (06:00 – 16:00)	3.49
Case 6			–	55 (06:00 – 16:00)	–	3.49
Case 7	RFH aided by warm air system	2.9	Air temperature at 20 (Duration: 24 × 7)	18– 55	20 (08:30 – 16:00)	0.3
Case 8			Floor surface temperature at 23 (Duration: 24 × 7)	18 – 55	20 (08:30 – 16:00)	
Case 9			Air temperature at 20 (Duration: 24 × 7)	18 (08:30 – 16:00)	–	
Case 10			Floor surface temperature at 23 (Duration: 24 × 7)	18 (08:30 – 16:00)	–	
Case 11	RFH aided by warm air system	2.9	Air temperature at 20 °C (06:00–16:00) and lowered to 15 °C during 16:00–06:00	18 – 55	20 (08:30 – 16:00)	0.3
Case 12	Lightweight floor radiant heating aided by warm air system					
Case 13	Convector heating aided by warm air system					

*Case 1 is the current daily operating mode (reference case).

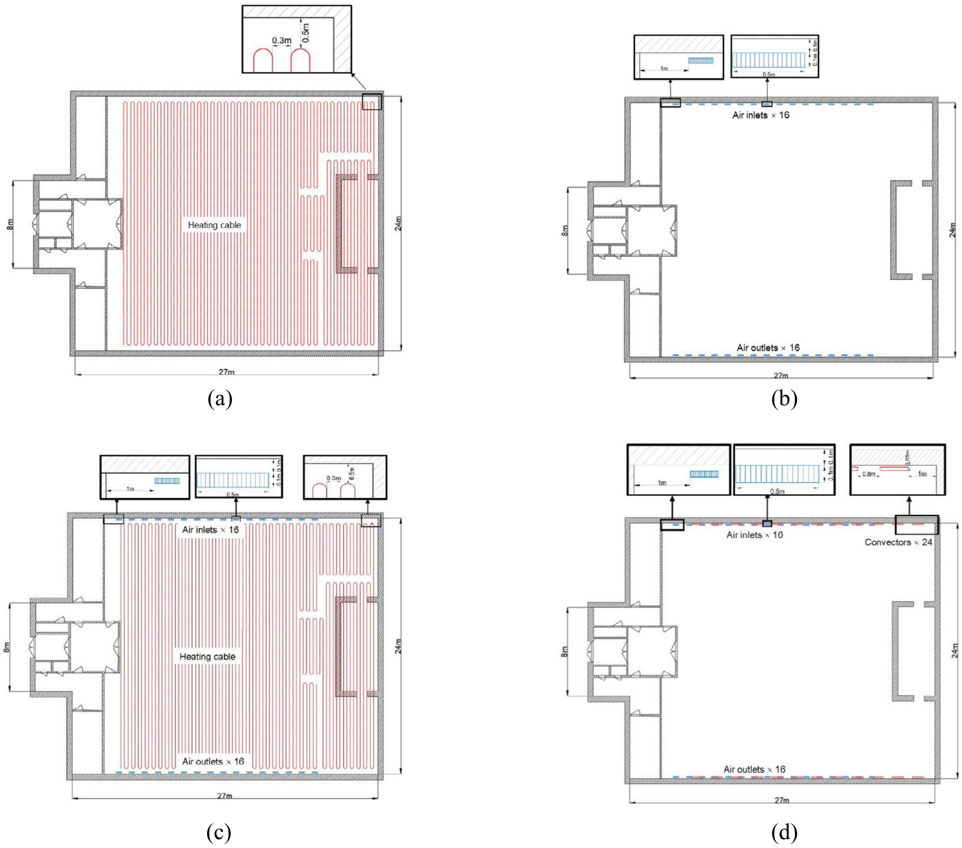

Fig. 5. Layouts of different heating systems: (a) the electric radiant floor heating; (b) the warm air heating; (c) the electric floor radiant heating with warm air system; (d) the electric convector heating with warm air system.

average value of the simulated results is calculated to compare with the average energy use records.

The heat balance in the indoor space can be expressed as Eq. (1).

$$Q_{RFH} + Q_{In} + Q_S = Q_{env} + Q_{infil} \tag{1}$$

where, Q_{RFH} is the heat supplied by the radiant floor heating system; Q_S is the solar heat gain; Q_{In} is the total internal heat gains generated by indoor lights, equipment, and occupants; Q_{env} is the heat losses from envelopes; Q_{infil} is the heat losses via air infiltration. All the units are in kWh. For a non-residential building with continuous space heating, the solar heat gain is not included to calculate the space heating load for the worst-case scenario [15]. In practice, the solar heat gain could partly compensate for the building heat loss. Therefore, the solar heat gain can't be ignored in the energy simulation, especially from March to May (after the vernal equinox) in the northern hemisphere.

2.6. Operating modes for heating

As mentioned previously, the management is not satisfied with the current daily operating mode of HVAC. Accordingly, this study analyzed other operating modes, such as constant set-point temperature heating, intermittent heating, and cyclic set-point temperature heating. In the constant set-point temperature mode, the church is heated at a constant temperature by RFH for 24 × 7, and the heating will be supported by a warm air system only during the opening hours of the church (8:30 – 16:00). In the intermittent heating, the church is heated by an all-air system from 06:00 and the all-air system is completely shut off when the church is closed at 16:00. The operating mode with cyclic set-point temperature is also realized by the combination of the RFH and the warm air system. In this mode, the RFH keeps the same set-point temperature as that of the constant set-point temperature heating during the daytime (06:00 – 16:00), while it works at a lower set-point temperature instead of completely shutting down at night (from 16:00 to the next morning 6:00). The daytime includes the opening hours and the recovering time. The 'recovering time' is defined as the time needed to raise the operative temperature from the lower set-point temperature at night to at least 19 °C[1].

The heating schemes analyzed are categorized into four groups and presented in Table 4. Case 1 is the current daily operating setting with existing windows, which is used in the numerical model verification. The set-point air temperature (20 °C) used in all simulations is obtained according to the current daily operation based on the operational staff's long-term practice. An analysis is carried out for four groups of different heating modes. The first group (Case 2 – 4) is to study the impact of energy-efficient windows based on the current daily operating mode. Case 1 – 4 only uses the electric radiant floor heating, shown in Fig. 5(a). The second group (Case 5 – 6) is the intermittent heating mode only using a warm air system, shown in Fig. 5(b). The third group (Case 7 – 10) is for the constant set-point temperature heating with the warm air system. In the second and the third group, the control strategy of the all-air system and the floor heating are compared. The last group (Case 11 – 13) is to study the thermal inertia of space heating systems in cyclic set-point temperature heating modes. Case 7 – 11 all use the electric floor radiant heating with a warm air system, shown in Fig. 5(c). Case 12 uses electric floor radiant heating with a warm air system, but lightweight floor radi-

[1] Assuming air velocity as 0.1 m/s, the minimum operative temperature for the acceptable indoor climate is 19.5 °C.

ant heating is used. The layout of Case 12 is the same as Fig. 5(c). Case 13 uses convector heating with a warm air system. Convector heating is used when the existing heavy radiant floor heating cannot be modified into a lightweight floor radiant heating system and convectors are evenly arranged along the wall, shown in Fig. 5(d).

The heat transfer coefficient of windows can be as low as 0.7–1.0 W/(m^2·K) or even lower [38,39]. In this study, different U-values from the IDA-ICE database are chosen. Case 5 and 6 are intermittent heating by all-air systems (scenario when the entire indoor heat load is provided by hot air supply without floor heating) with return or supplied air temperature control. The return air temperature control aims to achieve a constant ambient temperature within the room by varying the temperature of the air being supplied by the warm air heating units. The upper limit of the supplied air temperature (55 °C) was decided as per [40]. In the supplied air temperature control system the warm air heating units maintain the supplied air at a constant temperature of 55 °C. Case 7 – 10 are RFH aided by the warm air system with constant set-point temperature. The warm air system is switched on during 06:00–16:00. Two types of constant set-point temperature are analyzed: (i) constant set-point air temperature (Case 7 and 9), and (ii) constant set-point floor surface temperature (Case 8 and 10). The average floor surface temperature during the entire heating period was measured to be 23.4 °C and accordingly the setpoint floor temperature for the analysis was chosen as 23 °C. The control methods of the warm air heating system in Case 7, 8 and that of Case 9 and 10 are different. The return air temperature control is used in Case 7 and 8, while the supplied air temperature control is used in Case 9 and 10. Case 11 – 13 are RFH, lightweight floor radiant heating, and convector heating aided by the warm air system with cyclic set-point temperature, respectively. Three fresh air flow rates are used in different scenarios: 0.12 L/(s·m^2) is used in Case 1–4, according to the exhaust fan's air rate, 3.49 L/(s·m^2) is the capacity of the existing air handler of the all-air system in the church. 0.3 L/(s·m^2) is used for Case 7–13, in which the warm air system supports RFH by providing fresh air. The requirement of fresh air in the places of religious worship (ANSI/ASHRAE Standard 62.1–2019) [41].

2.7. Reliability and thermal comfort index

Both the air and average radiant temperature influence occupants' thermal comfort. Accordingly, the average of the air temperature and the mean radiant temperature is used to evaluate the indoor climate [16,42]. In practice, the building insulation and space heating systems' ability to maintain an acceptable indoor climate is limited. The outdoor weather varies. The indoor climate is not always kept as it is designed. The indoor air temperature will be affected by the outdoor temperature. For a given condition of the building insulation and heating capacity, the reliability of the heating system will be better if the indoor climate is least affected by the outdoor climate. To evaluate the reliability of indoor climate for ensuring thermal comfort under different operating modes, the ratio of the increment of the indoor operative temperature (ΔT_{Op}) to the increment of outdoor temperature (ΔT_{Out}) is proposed to explain the influence of the outdoor temperature on the indoor temperature.

PMV/PPD (Predicted Mean Vote / Predicted Percentage of Dissatisfied) is also used in this study to evaluate the indoor climate created by different operating modes. The PMV/PPD model is originally developed by Fanger using heat-balance equations and empirical studies about skin temperature to define comfort [43]. The model is one of the most recognized thermal comfort models and is adopted by the ANSI/ASHRAE Standard 55 [44] the ISO 7730 Standard [45] and the EN 16798–1 Standard [46]. In the calculation of PMV-PPD, the value of air temperature, the relative humidity, and the mean radiant temperature were taken from the IDA-ICE simulation results. The airspeed, 0.1 m/s, the mean value of the measuring results was used. The metabolic rate was 1.0 met (occupant seated and quiet). The clothing level was 1.30 col (trousers, long-sleeve shirt, long-sleeve sweater, t-shirt, suit jacket, and long underwear bottoms).

Though field measurement and numerical validation have been conducted in Teg's church, air infiltration, the vertical temperature difference, thermal bridge, and so on were not directly measured.

3. Results and discussion

3.1. Measurement results and numerical model validation

The energy simulation model is established based on the input parameters (meteorological data, Table 2 – 4) and the building parameters. The electricity use and the indoor air parameters (air temperature and relative humidity) were validated. The simulated result of average annual electricity use is 1.238×10^5 kWh (set-point air temperature as 20 °C), while the average measured result is 1.240×10^5 kWh. The calibration of the simulated energy use mentioned in section 2.5, resulted in a total deviation of 0.19% between the energy use records and the simulation results (Fig. 6), which reduces the performance gap [47]. The main reason for the minor deviation between simulation and measurement results may be due to the control method of RFH. In the simulation, the proportional-integral (PI) controller is used to realize operating mode control by the set-point air temperature. However, the staff manually adjusts the air temperature of the church based on the inputs from air temperature sensors. The manual adjustment method is not as accurate as the PI controller to keep the indoor air temperature at 20 °C.

The average measured and the simulated results of indoor air temperature and relative humidity from February 5 to May 15, 2020, are presented in Fig. 7. The average measured air temperature during the study period is 20.3 °C, while the simulated result from IDA to ICE is 20.1 °C. The difference between the simulated and the measured temperature is less than 1.0 °C for approximately 95% of the time. Although fluctuation of air humidity of the indoor air and outdoor air is different, the average value of simulated and the measured air relative humidity are 24.4% and 20.3%, respectively. The fluctuation of outdoor relative humidity has a relatively greater impact on indoor relative humidity. In practice, the hygroscopicity of the envelope material and indoor facilities has a delay and attenuation effect on the relative humidity change. When air is heated, the partial pressure of water vapor in the air does not change and hence the specific humidity does not change. The relative humidity does not have any effect on energy use. Besides, the envelope of the church is mainly made of concrete. The moisture absorption effect of concrete material is not significant in this study [48]. Therefore, although there is a difference between simulated and measured relative humidity, the one-dimensional heat, moisture, and air simultaneous model (HAM-Wall) [49,50] is not adopted. As the difference between simulated and measured values for energy use, air temperature, and relative humidity are insignificant, the numerical model of energy simulation is validated.

3.2. Heat balance and energy-saving potential

As per the heat balance of the building, the percentage of solar heat gain in the total heat gain increases from 9.0% to 69.5% (from February to May, shown in Fig. 8). The RFH is the main source of building heat gain in January (94.6%), February (88.9%), October (75.7%), November (92.3%), and December (95.9%). The internal

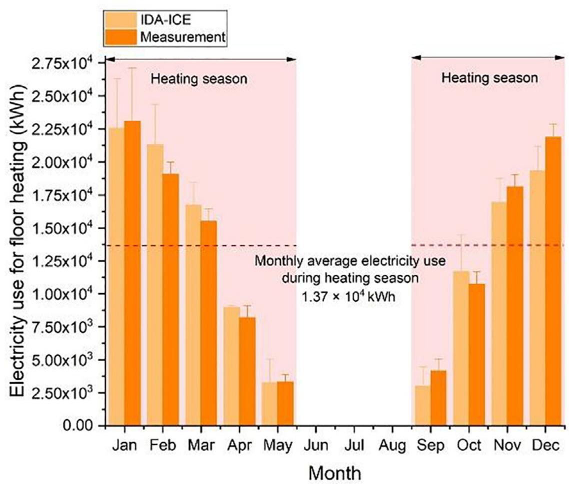

Fig. 6. Electricity use of floor heating by simulation and measurement.

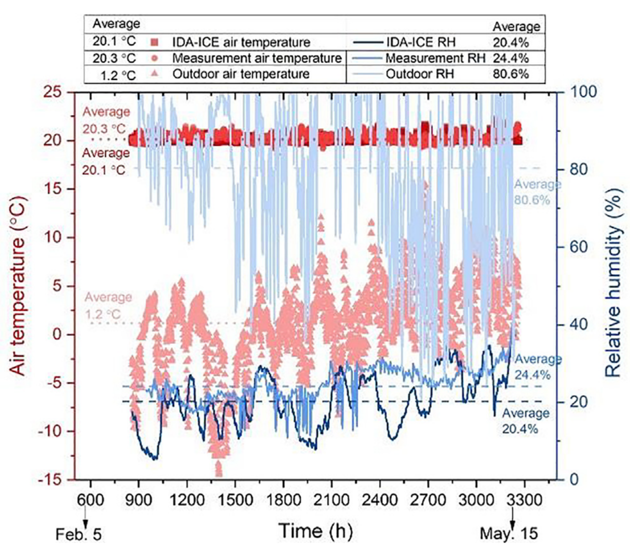

Fig. 7. Air temperature and relative humidity by simulation and measurement.

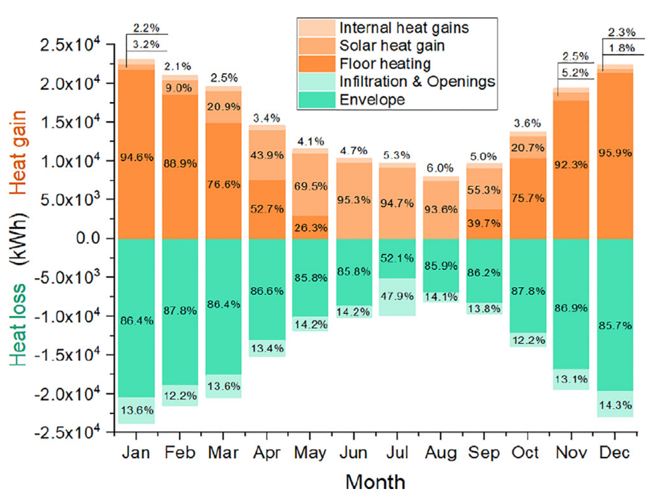

Fig. 8. Heat gains and losses from the case study church.

heat gains, accounting for 2.2% – 4.1%, are relatively stable during the heating season.

The church's heat loss mainly includes the heat loss by the envelope and the heat loss via infiltration and openings. The building heat loss is dominated by the envelope (85.9% – 87.8%) except during the non-heating season. In July, the heat loss from infiltration and openings causes a little more than 50% of the total heat loss from the building. This is because, in July, the main entrance of the church was opened during opening hours for airing to remove the indoor heat. The results show that the main energy-saving potential could be achieved by improving the building envelope performance. Some building codes of EU countries set requirements on specific heat transmission loss concerning the building envelope (U-values) [51]. Teg's church, being a protected building, needs to preserve its external and interior walls. Hence there are limitations to altering/modifying the building's envelope. However, the window glazing doesn't have historical significance. Accordingly, two options to reduce the energy use of the church building are studied: replacement of windows to energy-efficient windows and optimizing the operating modes of RFH according to the church's opening hours.

3.3. Thermal performance of windows

Large windows are common in public buildings for aesthetic purposes and maximum utilization of natural light [52,53], however, it could result in increased energy use. An energy-saving of approximately 27% can be achieved if the existing church windows are replaced with energy-efficient windows with the U-value of 0.6 W/(m^2·K). The reduction in energy use can be approximated as proportional to the reduction of the U-value of windows within 0.6 W/(m^2·K) $\leq$ U $\leq$ 2.9 W/(m^2·K) (R^2 = 1.00). With the same U-value of windows, when the set-point air temperature increases, energy use also increases, as shown in Fig. 9. To reflect the degree of the influence of U-value reduction on heating energy use reduction under different set-point temperatures, the equivalent temperature index ΔT_{u} is defined.

$$\Delta T_{\mathrm{u}} = \frac{\Delta Q}{\Delta U \cdot \tau \cdot A_{\mathrm{win}}} \tag{2}$$

where, ΔQ is the diminution in heating energy use, kWh; ΔU is the reduction of U-value, W/(m^2 K); τ is the total hours of the heating season, h; A_{win} is the total window area, m^2.

In this study, when the set-point air temperature is varied from 19 to 22 °C, ΔT_{u} varies from 6.8 to 9.0 °C. The higher the indoor set-point temperature, the greater the equivalent temperature. It means that the energy-efficient windows have a higher potential for energy-saving. Hence the significance of using energy-efficient windows increases with increasing the set-point temperature. Besides, the reduction of U-value has significantly reduced the ratio of the heat loss of the windows to the total building envelope heat loss, from 42% (U = 2.9 W/(m^2·K)) to 14% (U = 0.6 W/(m^2·K)).

3.4. Operating modes of heating

3.4.1. Intermittent and constant set-point temperature heating

The energy use of the seven different operating modes (Case 1, Case 5 – 10) is relatively close to each other, ranging from 1.152×10^5 – 1.336×10^5 kWh. For the RFH aided by the warm air system (Case 7 – 10), the energy use is mainly due to the RFH system, accounting for 90% – 96% of the total energy use. Compared with the operating mode of RFH with exhaust fans, the energy use of the all-air system intermittent heating with supplied air temperature control increases by 11.7%, while the energy use of other operating modes (Case 5, 7 – 10) is reduced by 2.1% to 3.7% (Fig. 10). Due to

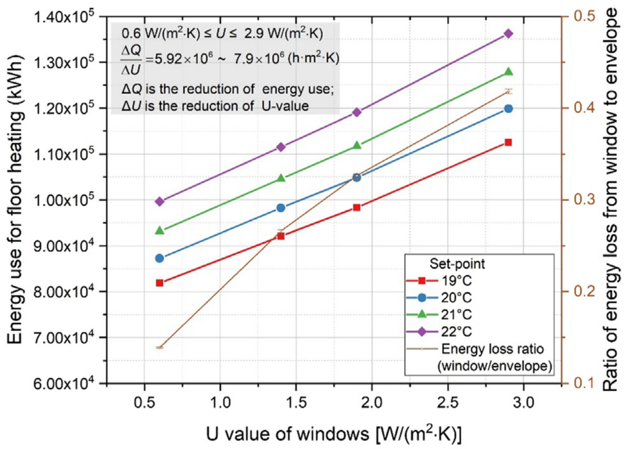

Fig. 9. Electricity use for floor heating for different U-value windows.

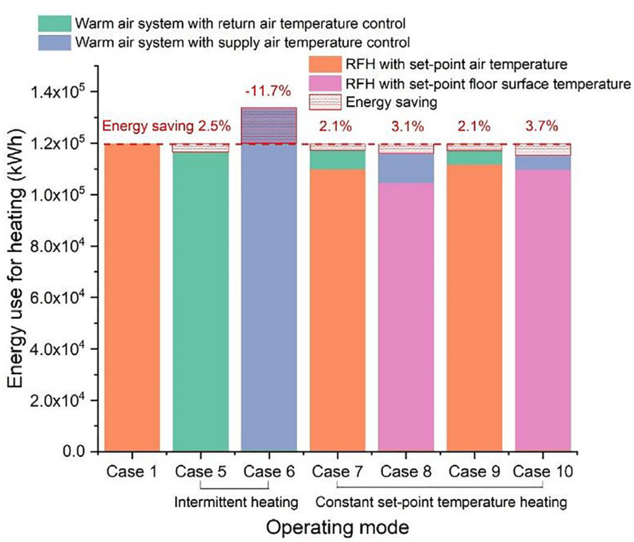

Fig. 10. Energy use for heating of strategies with constant set-point temperature.

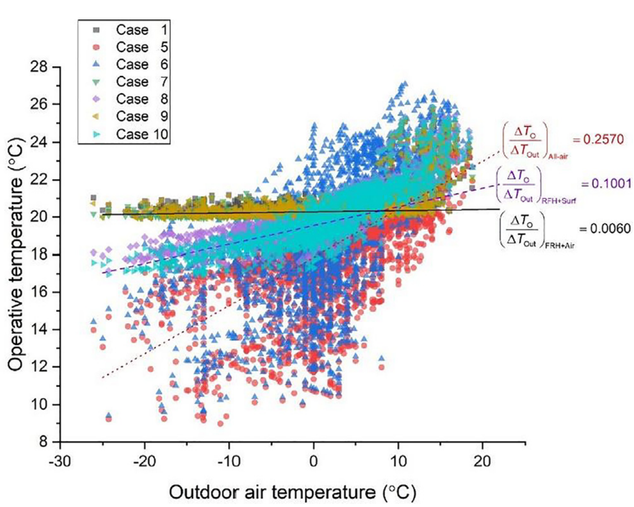

Fig. 11. Operative temperature of operating modes with constant set-point temperature.

the low outdoor temperature in winter (an average of 1.7 °C) and the large church hall volume, energy use can't be reduced in the all-air system intermittent heating with a constant supplied air temperature (Case 6). When the outdoor temperature is high, the temperature of the supplied air keeps the set value of 55 °C. Thus, the indoor air is overheated. Wang et al. conducted on-site measurement and showed that the energy-saving rate of the intermittent heating system is only about 5% compared with the continuous heating system [54]. It can be seen that these operating modes with intermittent heating and constant set-point temperature heating have limited energy-saving potential. The reliability and the thermal comfort of these operating modes are studied to provide a reference for further optimization of operating strategies.

It is found that the indoor temperature is affected by the outdoor temperature at different levels under different operating modes (Fig. 11). When operating modes of the RFH with set-point air temperature (Case 1, 7, and 9) are chosen, the ratio of the increment of the operative temperature to the increment of outdoor temperature $\left(\frac{\Delta T_{Op}}{\Delta T_{Out}}\right)_{RFH+Air} = 0.0060$, followed by operating modes of the RFH with floor surface temperature (Case 8 and 10) $\left(\frac{\Delta T_{Op}}{\Delta T_{Out}}\right)_{RFH+Surf} = 0.1001$ and $\left(\frac{\Delta T_{Op}}{\Delta T_{Out}}\right)$ for all-air system intermittent heating (Case 5 and 6) is 0.2570. Accordingly, the operating modes of RFH with set-point air temperature are relatively more reliable, while the all-air system intermittent heating is found to be the least reliable.

To compare the thermal comfort of intermittent heating (Case 5 and 6) and constant set-point air temperature heating (Case 7 – 10), the PMV/PPD results of different operating modes are shown in Fig. 12(a) – (f). Since the PMV/PPD analysis of Case 2 – 4, 11 – 13 are similar to that of Case 7 during the opening hours, those detailed results aren't shown here. The operating modes of RFH with set-point air temperature (Case 7 and 9) can meet the thermal comfort requirements (EN 16798-1 [46]) throughout the heating season. The intermittent heating (Case 5 and 6) has situations with an unsatisfied cold feeling. For operating modes of the RFH with set-point floor surface temperature (Case 8 and 10), the hours below the III categories (last level of thermal comfort according to EN 16798 [46]) of PMV/PPD account for 15.8% and 26.4%, respectively, of the total opening hours (2192 h) during the heating season. For Cases 5–6, 8, and 10, the results of PMV/PPD are worse than that of III categories, which meant that most people will feel uncomfortable. The unsatisfied hours' proportions when using intermittent heating (Case 5 and 6) are 58.1% and 37.4%, respectively, which are much larger than operating modes of the RFH with set-point floor surface temperature. It is found that the intermittent heating with return air temperature control has an unsatisfied cold and an unsatisfied warm feeling, as it has a constant air supply temperature, shown in Fig. 12(a). Overheating (PMV > +0.7) results when the outdoor air temperature increases and the overheated hour's proportion is 1.4%. The cyclic set-point air temperature RFH aided by the warm air system (return air temperature control) is investigated to study the possibilities of additional energy-saving strategies.

3.4.2. Cyclic set-point temperature heating

The analysis shows that intermittent heating (Case 5 and 6) has the problem of an unsatisfied cold feeling during nearly half of the heating season, and also results in higher energy use (Fig. 10, 12(a) and (b)). Contrarily, a constant set-point temperature for space heating (Case 7 and 9) ensured the required thermal comfort, however, it also resulted in high energy use. The operating mode of cyclic set-point temperature works at a lower set-point temperature during the evening/night. The potential energy savings of cyclic set-point temperature operating modes lies in the relatively lower

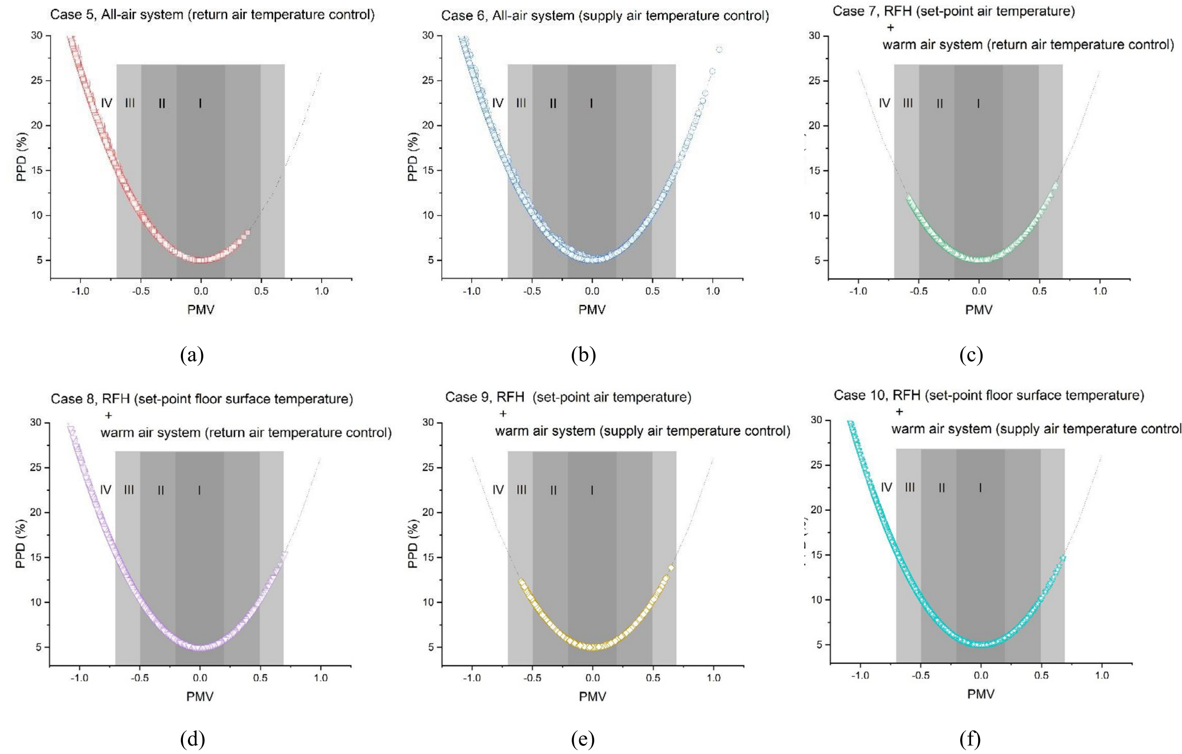

Fig. 12. PMV-PPD of operating modes with the constant set-point air temperature of floor heating (Airspeed: 0.1 m/s; Metabolic rate: 1.0 met; Clothing level: 1.30 col.)

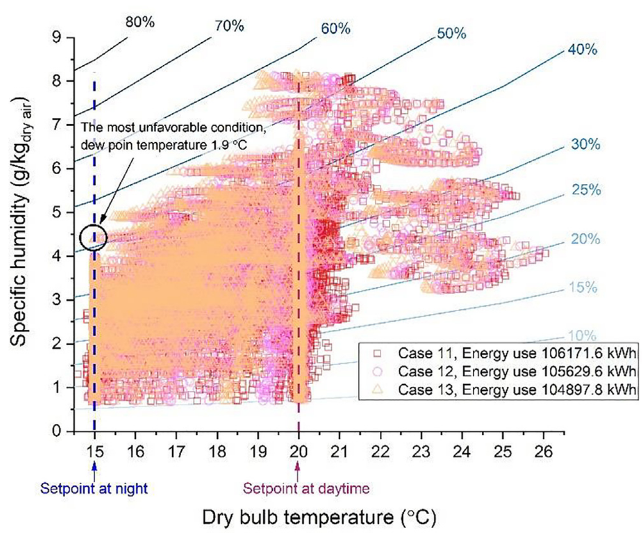

Fig. 13. Indoor climate of different operating modes with cyclic set-point temperature in a psychrometric chart.

temperature at night. To ensure thermal comfort, the indoor air temperature should be raised to the normal level before the church opens. The thermal inertia of the space heating system determines the 'recovering time'.

For Case 11 – 13, the air temperature of RFH is set at 20 °C and 15 °C during 06:00 – 16:00 and from 16:00 to the next morning 06:00, respectively. The total heat energy use of Case 11 – 13 is simulated to 1.062×10^5 kWh, 1.056×10^5 kWh, and 1.049×10^5 kWh, respectively. The average indoor operative temperature during opening hours, achieved by Case 11 – 13 is 20.15 °C, 20.00 °C, and 19.88 °C, respectively (Table 5). It can be found that the indoor climate created by Case 11 – 13 is also similar for the whole year (Fig. 13). The data scatter formed two bold lines at the two set-points, 15 °C at night and 20 °C during the daytime (Fig. 13). Most of the psychrometric parameters lie between the two set-points. The greater the thermal inertia of the heating system, the more hours are needed for increasing air temperature above 20 °C. Besides, the lowest temperature of the inner surface of the window is 6.8 °C at night. The most unfavorable condition of air is 15.0 °C with a relative humidity of 41%, whose corresponding dew point temperature is 1.9 °C. The lowest temperature of the inner surface of the window is about 5.0 °C greater than the dew point temperature. It can be concluded that in all the cyclic set-point temperature operating modes, the air relative humidity is free from condensation problems even with outdoor air relative humidity of 100%.

Table 5
Statistic data of indoor climate during daytime created by Case 11–13.

		Case 11	Case 12	Case 13	Max difference rate
Operative temperature (°C)	Average	20.15	20.00	19.88	1.34%
	Minimum	18.18	18.20	18.27	0.50%
Air temperature (°C)	Average	20.23	20.09	20.07	0.79%
	Minimum	18.05	18.01	18.03	0.11%
Relative humidity (%)	Average	23.0	23.2	23.2	0.87%
	Minimum	51.7	53.8	55.5	7.35%

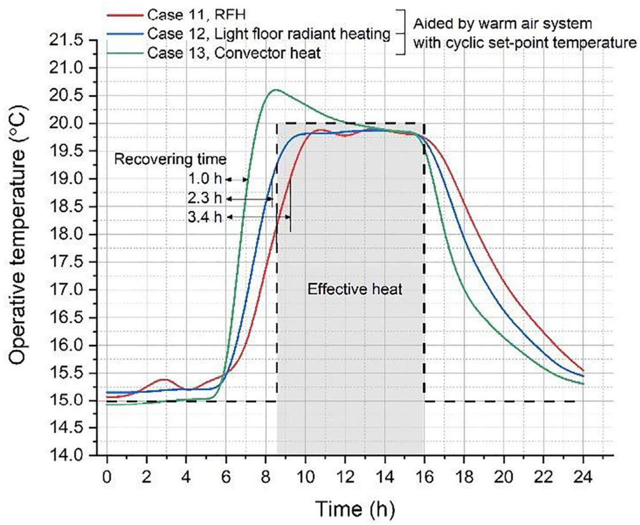

Fig. 14. Recovering time and operative temperature of the coldest day.

To reflect the 'recovering time' of different operating modes with cyclic set-point temperature, December 22 (the day with the lowest dry-bulb temperature of the outdoor environment) is selected to analyze the most unfavorable condition. The operating modes with cyclic set-point temperature can save at least 9.33% heating energy as compared to the operating modes with constant set-point temperature heating. This is because the church only needs to be heated during the day's opening hours, shown as the gray area in Fig. 14. It is called 'effective heat' by Wang et al. [42]. The operating modes with cyclic set-point temperature can keep the indoor air temperature profile as close as possible to the effective heat (the black dashed line in Fig. 14). The cyclic set-point temperature heating turns down the set-point temperature during the closing hours of church. Accordingly, the operating modes with cyclic set-point temperature minimize the ineffective heat by reducing the heat loss in periods when heating is not required. The convector heating aided by the warm air system (Case 12) results in the shortest recovering time of one hour, followed by the lightweight floor radiant heating aided by the warm air system (Case 13) with recovering time of 2.3 h. The RFH aided by the warm air system (Case 11) has the longest recovering time of 3.4 h. Therefore, when operating modes with cyclic set-point temperature are used, the small thermal inertia of the air heating system should be preferred [55]. In this way, the recovering time can be reduced and thereby can save more energy.

4. Conclusions

In this paper, field measurement and numerical analysis were carried out in a large single-zone building to study the energy use and the thermal comfort indices of radiant heating aided by the warm air heating system. The numerical simulation was verified by the measuring results of the electricity use, indoor air temperature, and relative humidity. It can be concluded that:

(1) The energy saving potential mainly lies in the improvement of the thermal performance of the building envelope. Energy savings of up to 27% can be achieved if the existing windows are replaced with energy-efficient windows. For historical buildings, within the scope allowed by the protection requirement, installation of energy-efficient saving windows or providing additional inside glazing could reduce energy use.

(2) The energy use of constant set-point temperature strategies can be reduced by 2.1% to 3.7% by varying the operating modes. The relaibility evaluation of the different operating modes of heating suggest that the indoor climate with intermittent heating created by the all-air system is most susceptible to changes in outdoor temperature. For buildings with tall indoor space, especially in a cold climate, intermittent heating by the all-air system may not be appropriate.

(3) The main difference between the three studied operating modes of RFH with cyclic set-point temperature is the recovering time for reaching the set-point temperature. Heating systems with lower thermal inertia should be preferred to shorten the recovering time. For the RFH with cyclic set-point temperature, when determining the lower set-point air temperature, the risk of condensation on windows should be avoided. The operating modes proposed in this work can serve as a reference to operating a heating system in a large single-zone church building in a cold climate.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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