

Experimental study of energy performance in low-temperature hydronic heating systems

Arefeh Hesaraki^{*1}, Eleftherios Bourdakis², Adnan Ploskić¹, Sture Holmberg¹

¹*Division of Fluid and Climate Technology, School of Architecture and the Built Environment, KTH Royal Institute of Technology, Stockholm, Sweden*

²*International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, DTU, Copenhagen, Denmark*

Abstract

Energy consumption, thermal environment and environmental impacts were analytically and experimentally studied for different types of heat emitters. The heat emitters studied were conventional radiator, ventilation radiator, and floor heating with medium-, low-, and very-low-temperature supply, respectively. The ventilation system in the lab room was a mechanical exhaust ventilation system that provided one air change per hour of fresh air through the opening in the external wall with a constant temperature of 5 °C, which is the mean winter temperature in Copenhagen. The parameters studied in the climate chamber were supply and return water temperature to the heat emitters, indoor temperature, and heat emitter surface temperature. Experiments showed that the mean supply water temperature for floor heating was the lowest, i.e. 30 °C, but it was close to the ventilation radiator, i.e. 33 °C. The supply water temperature in all measurements for conventional radiator was significantly higher than ventilation radiator and floor heating; namely, 45 °C. Experimental results indicated that the mean indoor temperature was close to the acceptable level of 22 °C in all cases. For energy calculations, it was assumed that all heat emitters were connected to a ground-source heat pump. Analytical calculations showed that using ventilation radiator and floor heating instead of conventional radiator resulted in a saving of 17% and 22% in heat pump's electricity consumption, respectively. This would reduce the CO₂ emission from the building's heating system by 21 % for the floor heating and by 18 % for the ventilation radiator compared to the conventional radiator.

Keywords: Low-temperature hydronic heating systems, Energy performance, Thermal environment, Experimental study, Ventilation radiator, Floor heating

* Corresponding author: Tel.: +46 8 790 48 84; fax: +46 8 790 48 00.
E-mail address: arefeh.hesaraki@byv.kth.se
Address: Brinellvägen 23, 100 44 Stockholm, Sweden

Nomenclature

COP	coefficient of performance
D_h	degree hours per year, $^{\circ}\text{C}\cdot\text{h}\cdot\text{year}^{-1}$
E	energy demand by building, $\text{kWh}\cdot\text{year}^{-1}$
E_{el}	electrical energy consumption by heat pump, $\text{kWh}\cdot\text{year}^{-1}$
g_{el}	CO_2 emission factor for electricity, $\text{kg}_{\text{CO}_2}\cdot\text{kWh}^{-1}$
GSHP	ground-source heat pump
$h_{gl,surf}$	total heat transfer coefficient at inner glazing surface of windows, $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
n	number of measurements
$P_{passive, indirect}$	passive (indirect) internal and external heat gains, W
Q_{tot}	specific heat loss, $\text{W}\cdot\text{K}^{-1}$
SD	standard deviation
U_{wall}	heat transfer coefficient of wall, $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
U_{window}	heat transfer coefficient of windows, $\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
x_i	measured parameter
x_m	mean value of measured data
θ_{base}	base temperature at which the heat loss from building is equal to heat generated by active heating system, $^{\circ}\text{C}$
$\theta_{gl, surf}$	window' surface temperature, $^{\circ}\text{C}$
θ_i	indoor temperature, $^{\circ}\text{C}$
θ_o	outdoor temperature, $^{\circ}\text{C}$

1. Introduction

The level of temperature supplied to the heat emitter in buildings plays a major role in primary energy consumption and environmental impacts. In addition to increasing the earth's temperature every year, as more and more buildings are becoming energy-efficient due to better thermal insulation, less infiltration and more efficient heating and ventilation systems, heat losses from buildings are decreasing. All these changes could be the reasons to reduce the need to supply the heating system with water at temperature as high as previously. As the temperature to the heat emitter decreases, heat losses from the heat production unit and from distribution pipes decrease, and consequently, more renewable and low-quality energy sources can be used [1]. Boerstra et al. [2] defined different supply temperature levels; namely: 55°C for medium-, 45°C for low-, and 35°C for very-low-temperature heat emitters. The main principle of low-temperature heating system is to provide the same thermal comfort as medium-temperature heating system, while using a lower supply temperature [3]. Supporting a low-temperature heat emitter with a heat pump is thermally efficient. Generally, the thermal efficiency of a heat pump improves by one to two percent for every degree by which the supply water temperature is reduced [4]. Heat pumps have been recognized for many years as an energy-efficient and sustainable heat source that, by utilizing renewable energy, uses three to four times less electrical energy to deliver the same amount of heat as a direct electrical heater. By

2013, more than half of detached and semi-detached dwellings in Sweden have had heat pumps installed [5]. Therefore, as the number of heat pumps sold in Sweden and Europe increases, there is a growing need to adjust the temperature of heat emitters to this change in order to attain greater efficiency; that is, to use low-temperature heat emitters. In addition, some recent studies have focused on low-temperature district heating, known as the fourth generation of district heating networks [6]. This means that, in the future, the supply water temperature to buildings connected to district heating – that is, more than 90 % of Swedish apartment buildings – will also decrease. Therefore, there is also a need to renovate existing apartment buildings to be adapted for a lower supply water temperature. A reduced return temperature level also favors district heating networks in terms of higher efficiency of heat generation plants, heat pumps, and solar collectors. Furthermore, with a requirement for lower supply temperature level surplus and waste heat could be used as an efficient heat source in a district heating system.

Using a low-temperature heating system is also more sustainable due to a reduction in the generation of carbon dioxide. For every degree reduction of the supply temperature in a heating system, the carbon dioxide emission decreases by 1.6 % [4]. Ploskic [4] showed that by using a water supply temperature of 40°C instead of 55°C , heat pump efficiency would increase by 25 % and the carbon dioxide emissions would decrease by 24 %.

In some investigations [7-9], low-temperature panel heating had better indoor air quality than a high-temperature heating system. This was due to the correlation between the temperature of heating surface and particle deposition and also the mite population. In addition, thermal comfort increases by a greater share of radiant heat transfer and lower vertical temperature gradient in a room with low-temperature panel heating, which makes it possible to reduce the indoor temperature. This would also decrease ventilation heat losses. Experimental investigations by Zhao et al. [10] showed better thermal environment and energy savings with low-temperature heating and high-temperature cooling compared to the jet ventilation system in China's International Airport. Primary annual energy calculation based on Energy Performance of Buildings Directive (EPBD) by Olesen and Carli [11] showed that a low-temperature heating system connected to a ground-source heat pump has better overall energy performance than a conventional radiator connected to a boiler or an air to water heat pump. Hasan et al. [12] studied the performance of low-temperature heating systems in terms of energy consumption and thermal comfort. Their results showed that although the supply temperature decreased to 45 °C in a conventional radiator, the indoor temperature never dropped below 20 °C. This was due to an oversized conventional radiator for a well-insulated building.

Maivel and Kurnitski [13] investigated the distribution and emission losses for a low-temperature heating system compared to a high-temperature heating system installed in different building types located in a cold and Central European climate. Their results showed that, depending on the building type, climate condition and heating period, using high-temperature heating with 70 °C supply temperature has 4-40 % higher losses compared to low-temperature heating system with 40 °C supply temperature. Nagy et al. [14] developed a model to investigate the influence of retrofit measures to the supply water temperature. Their results showed that the supply water temperature to the existing building without any retrofit measures can be decreased by 10 °C; that is, from 55 to 45 °C, without sacrificing thermal comfort. In addition, they showed that improving the building's insulation would allow to decrease the supply temperature to 40 °C and saving energy by 60 % compared to the reference case.

There are different types of low-temperature room heaters in which the large surface area or improved forced convection makes it possible to reduce the supply water temperature without sacrificing the heat output. Examples include

panel heating such as floor, ceiling or wall heating, or forced-convection radiator such as ventilation radiators [15], or add-on fan radiators [16]. In a ventilation radiator, the ventilation supply is placed behind the radiator; see Fig. 1. This combination increases the forced convection heat transfer and makes it possible to pre-heat the supply air before it enters the room. An experimental investigation [15] showed that an efficient ventilation radiator produced twice as much heat output as a conventional radiator under the same conditions. This was due to the high convective heat transfer by combining it with incoming air, and also the large temperature difference between cold incoming air and the heat emitting surface.

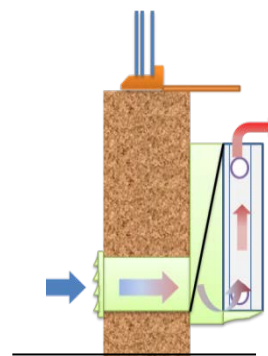


Figure 1 Schematic of ventilation radiator; that is, the combination of supply ventilation with radiator to preheat the cold supply air and increase the efficiency of the radiator

In the add-on fan radiator, fans are placed below the radiator panels to boost the convection heat transfer. A previous study [16] showed that having five fans below the radiator increases the heat output to almost twice that of a conventional radiator. This was due to increasing the convection heat transfer along the radiator surfaces. Added fans below the radiator consumed very small amounts of electricity, that is, the ratio of electricity consumption by the fan to increasing the heat output of the radiator was between one and two percent.

For floor heating, the temperature difference between supply and return is decreased due to the preferred homogenous temperature along the floor. This would require a higher flow rate, followed by increasing auxiliary hydraulic pressure loss and work for the circulation pump [11] compared to a hydronic radiator system. In addition to higher circulation pump work, higher primary energy consumption in floor heating could be due to high heat loss to the ground because the heating system is not completely interior and is embedded in the envelope [17]. However, this problem could be solved by adding more insulation under the floors. Sarbu and Sebarchievici [18] investigated

thermal comfort, energy savings, environmental impacts, and economic performance of low-temperature heating systems, including floor heating, wall heating, ceiling heating, and a combination of floor and ceiling heating. Their study showed that floor-ceiling heating provided the best performance in terms of energy consumption, environmental impacts, and operating cost.

Statistical data showed that an average Swedish detached and semi-detached house built between 2001 and 2012 had annual energy consumption of 84 kWh·m⁻² for heating and domestic hot water [19]. Energy measurements by Hesaraki and Holmberg [20] for five semi-detached houses built in 2011 equipped with low-, and very-low-temperature heat emitters, including floor heating on the first floor and ventilation radiator on the second and third floors, showed an average annual heating requirement of 48 kWh·m⁻² for heating and hot water. This consumption was 43 % less than that in an average detached and semi-detached house built between 2001 and 2012.

Previous studies have focused on the energy consumption of low-temperature heat emitters, leaving the question of how much the actual savings are compared to conventional radiator under the same conditions still to be addressed. However, the laboratory measurements, simulations and field tests have indicated that low- and very-low-temperature heat emitters could be an energy-saving alternative compared to conventional radiators [15, 20]. To evaluate and compare the performance of low- and very-low-temperature heating systems, including ventilation radiators and floor heating, with conventional radiators, laboratory measurements were conducted. Afterwards, a detailed comparison of energy performance, thermal environment and environmental impacts in conventional radiator, ventilation radiator, and floor heating was made.

2. Methodology

The methodology consists of evaluating a medium-, low- and very-low-temperature heat emitters in a climate chamber test facility at Technical University of Denmark (DTU).

2.1. Climate chamber description

The geometrical dimensions of climate chamber were 4 x 4.2 x 2.7 (m); see Fig. 2. The simulated room there represented a living room where two persons are at home during the evening. For this reason, no solar external heat gains were simulated. The room was simulated with one

external wall, including a cold window, and three internal adiabatic walls with an adiabatic floor and ceiling. The U value of windows and wall were considered to be 2.10 and 0.25 (W·m⁻²·K⁻¹), respectively, corresponding to an average multi-family building built between 1976 and 2005 [21].

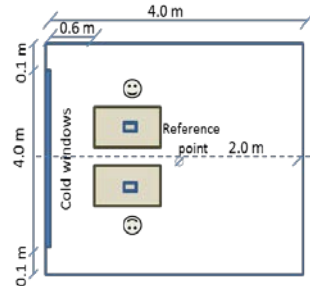


Figure 2 Climate chamber study at DTU University

Two seated and relaxed persons, two laptops and two lamps were used as internal heat source generators, creating a total of 204 W of heat. The ventilation rate was set to 1 air change per hour, corresponding to 12.6 l·s⁻¹ with an inlet constant temperature of approximately 5 °C, which is the average winter temperature in Copenhagen. In addition, since the climate chamber was not very tight, 10 % leakage was assumed when calculating total heat loss. The outdoor temperature of 5 °C was used to calculate temperature of the cold windows; see Eq. (1). In Eq. (1), for the total heat transfer coefficient at the inner glazing surface, the mean value of 7.6 W·m⁻²·K⁻¹ was used [22].

$$\theta_{gl,surf} = \theta_i - \frac{U_{windows}}{h_{gl,surf}} (\theta_i - \theta_o) \quad (1)$$

where θ_i and θ_o are the indoor and outdoor temperatures (°C), respectively, U_{window} and $h_{gl,surf}$ are the heat transfer coefficient of windows and the inner glazing surface of windows (W·m⁻²·K⁻¹), respectively.

Heat losses and passive heat gains in the room caused a heat demand of 20 W·m⁻² to be covered by an active heating source. The boundary conditions for measurements are shown in Table 1.

Three types of heat emitters were used: conventional radiator, ventilation radiator, and floor heating; see Fig. 3. For floor heating and conventional radiator, a fresh air supply diffuser was placed above the window. In case of ventilation radiator, cold ventilation air was preheated by passing through the radiator's panels before entering the room. In order to simulate a living room, the exhaust air duct was placed at low level in the climate chamber. This was similar to the case in which the air was exhausted from below the door in the living room; as there is no exhaust device in this room.

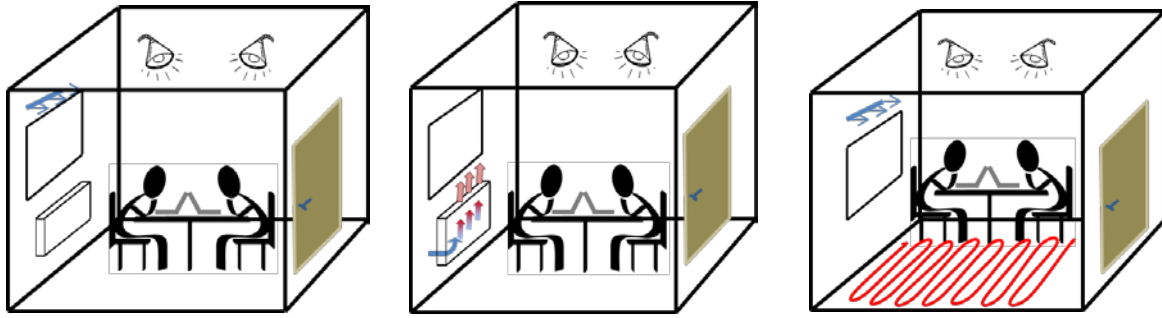


Figure 3 Conventional radiator (left), ventilation radiator (middle) and floor heating (right) used for investigation

Table 1 Boundary conditions for experiments in the DTU climate chamber

Transmission loss	Ventilation loss		Heat gains		
$U_{\text{external wall}}, \text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	0.25	Ventilation type	Mechanical exhaust	Passive heat sources	2 persons, 2 lamps, 2 laptops
External wall area, m^2	3.26	Ventilation rate, $\text{l}\cdot\text{s}^{-1}$	12.6	Active heat sources	Floor heating, Ventilation radiator, Conventional radiator
$U_{\text{window}}, \text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$	2.10	Leakage rate, $\text{l}\cdot\text{s}^{-1}$	1.26		
Window area, m^2	6.40	Ventilation temperature, $^{\circ}\text{C}$	5.0		
Window temperature, $^{\circ}\text{C}$	16.6				
Outdoor temperature, $^{\circ}\text{C}$	5.0				

All measurements were made in a steady state condition; that is, constant ventilation temperature, constant ventilation flow rate, and constant internal heat gains. Steady state condition was reached after three hours for each measurement. Measurements were conducted for approximately 45 hours for each type of heat emitter.

2.2. Thermal environment and heat output of heat emitters

To measure the water, air, and surface temperatures, plastic-coated paper probes with a sensor made of a film thermistor on an aluminium oxide substrate with accuracy of $\pm 0.3^{\circ}\text{C}$ were used. To avoid radiant heat exchange, a small piece of thermal insulation together with aluminium foil to shield the sensors was used. The mean indoor temperature was measured in the center of the room at a height of 1.1 m, representing the breathing zone of a seated person. In addition, the surface temperatures of windows and heat emitter were measured at different points. To measure the temperature of ventilation supply the sensor was placed on the supply air diffuser. Furthermore, the temperatures of inlet and outlet water of the heat emitters were monitored inside the pipes right before and after passing through the heat emitter, respectively.

The heat output of each heat emitter was calculated based on the mean room temperature, the ventilation supply temperature, the water

supply, and the return temperature using a calculation code given by the manufactures. Heat

output of radiators and floor heating given in manufacturer's heat emission data were in accordance with European Norms of EN 442 and EN 1246, respectively [23, 24]. The maximum deviation of $\pm 2\%$ in heat output compared to other certificated laboratories [25] should be kept for holding the license for testing, according to the EN. Therefore, the thermal power given by the manufacturer's data is reliable with high precision.

2.3. Uncertainty analysis

To analyse the uncertainty integrated in the measurements, the standard deviation (SD) of variables representing statistical error was considered. Since the entire population were taken, population standard deviation was used; see Eq. (2).

$$SD = \sqrt{\frac{1}{n} \sum_{i=1}^n (x_i - x_m)^2} \quad (2)$$

where n is the number of measurements, x_m is mean value of measured data, and x_i is measured data.

2.4. Energy consumption and environmental impacts of heat emitters

The degree-hours (D_h) method was used to calculate energy demand; see Eq. (3). This is a simplified method for calculating the building energy demand for active heating. The degree hours mainly depend on the building location and the base temperature.

$$E = Q_{tot} \cdot D_h \quad (3)$$

where Q_{tot} is the specific heat loss ($W \cdot K^{-1}$) and D_h is degree hours per year ($^{\circ}C \cdot h \cdot year^{-1}$).

The base temperature is defined as the outside temperature above which the building needs no active heating. In this temperature, the heat loss from the building is equal to the heat generated by the active heating system. The difference between the base temperature and the desired indoor temperature is covered by internal and external heat gains from sources such as occupants, equipment, lighting, and solar energy; this is called passive or indirect heat ($P_{passive, indirect}$). For the climate chamber under study, the passive or indirect heat was calculated to be 204 W based on two sitting people, each generating 80 W, and two lamps and two laptops, generating 44 W. Using Eq. (4), the base temperature for the studied room was calculated as 15 $^{\circ}C$; that is, specific heat loss of 31.4 $W \cdot K^{-1}$ and an indoor temperature of 22 $^{\circ}C$. The average annual degree hours during 2010-2014 for Copenhagen, with annual mean temperature of 8.6 $^{\circ}C$ and base temperature of 15 $^{\circ}C$, was 63,912 $^{\circ}C \cdot h \cdot year^{-1}$ [26].

$$\theta_{base} = \theta_i - \frac{P_{Passive, indirect}}{Q_{tot}} \quad (4)$$

where θ_{base} and θ_i are indoor and base temperatures ($^{\circ}C$), respectively, $P_{passive, indirect}$ is heat contribution from internal and external heat gains (W).

In order to analyze the data in terms of energy consumption, it was assumed that the heat emitters were connected to a ground-source heat pump (GSHP). Therefore, to calculate energy consumption, the total energy demand was divided by the coefficient of performance (COP) of heat pump, see Eq. (5). To find the mean annual COP of the heat pump, a commercial program called Vitocalc 2010, developed by Viessmann, was used [27].

$$E_{el} = \frac{E}{COP} \quad (5)$$

With respect to environmental impacts, the CO_2 emissions from heating systems were calculated based on heat pump's electrical energy consumption (E_{el}) and specific CO_2 emission factor for electricity (g_{el}), which is 0.041 $kg_{CO_2} \cdot kWh^{-1}$ in Sweden [28]; see Eq. (6).

$$CO_2 = g_{el} E_{el} \quad (6)$$

3. Results and Discussion

Figs. 4, 5, and 6 show the supply and return temperature for the heat emitters, the supply temperature for ventilation, and the mean room temperature at the reference point for the conventional radiator, ventilation radiator, and floor heating, respectively. Due to the high mass flow rate to all types of heat emitter, the mean temperature difference between return and supply was between 4 and 5 ($^{\circ}C$). The mean supply water temperature for floor heating was the lowest, i.e. 30 $^{\circ}C$, but it was close to the ventilation radiator, i.e. 33 $^{\circ}C$. The supply water temperature in all measurements for conventional radiator was significantly higher than ventilation radiator and floor heating; namely, 45 $^{\circ}C$.

Table 2 shows the results of the thermal environment for each case. As can be seen, the mean room temperature at the reference point was 22.1 $^{\circ}C$ for all cases when the supply ventilation temperature and the cold window's temperature were approximately 5.0 and 16.6 ($^{\circ}C$), respectively. The mean surface temperature of the ventilation radiator was 11.2 $^{\circ}C$ less than the conventional radiator. This was due to the lower temperature in supply water, and also cold ventilation air blowing between radiator's panels. In the case of floor heating, the mean temperature of the floor surface was 23.2 $^{\circ}C$, which according to the ASHRAE 55-2010 standard is within the acceptable range. Using degree-hours method, the annual energy demand was calculated as 119 $kWh \cdot m^{-2}$; that is, multiplying the specific heat loss of 31.4 $W \cdot K^{-1}$ by the degree hours of 63,912 $^{\circ}C \cdot h \cdot year^{-1}$. In addition, energy consumption and CO_2 emissions were calculated as electrical energy consumed by a ground-source heat pump; see Table 3. As can be seen in Table 3, energy consumption and CO_2 emissions from floor heating were lower compared to the other systems. Energy was reduced by 17 and 22 (%), when using ventilation radiator and floor heating, respectively, compared to conventional radiator. In addition, CO_2 emission savings were 18 and 21 (%) for ventilation radiator and floor heating, respectively, compared to conventional radiator.

3.1. Uncertainty analysis of laboratory measurements

Calculations of uncertainty analysis showed that the largest standard deviation was for the heat output of the floor heating; namely 6 %, and the smallest deviation of 1 % was for the surface, air and water temperature; see Table 4.

Table 2 Thermal environment measurements for each type of heat emitters (all units are in °C)

Type of heat emitter	Mean ventilation temperature	Mean room temperature	Mean water temperature (supply/ return)	Mean window surface temperature	Mean surface temperature of heat emitter
Floor heating	5.0	22.1	30.0/25.0	16.5	23.2
Ventilation radiator	5.0	22.2	33.0/29.0	16.7	26.8
Conventional radiator	5.1	21.9	45.0/40.0	16.5	38.0

Table 3 Energy performance and CO₂ emissions from each type of heat emitters

Type of heat emitters	COP of GSHP	Mean heat output	Annual energy consumption (saving)	Annual CO ₂ emissions (saving)
Floor heating	4.5	336 W	27 kWh·m ⁻² (22 %)	18.6 kg (21 %)
Ventilation radiator	4.2	350 W	28 kWh·m ⁻² (17 %)	19.3 kg (18 %)
Conventional radiator	3.5	329 W	34 kWh·m ⁻² (-)	23.4 kg (-)

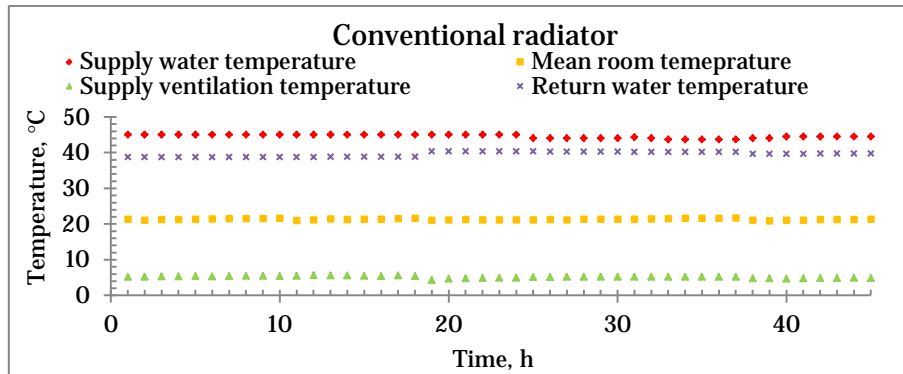


Figure 4 Temperature measurements for conventional radiator

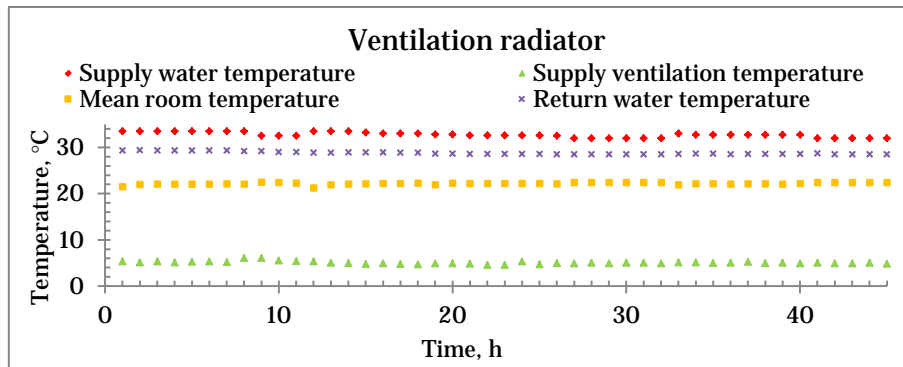


Figure 5 Temperature measurements for ventilation radiator

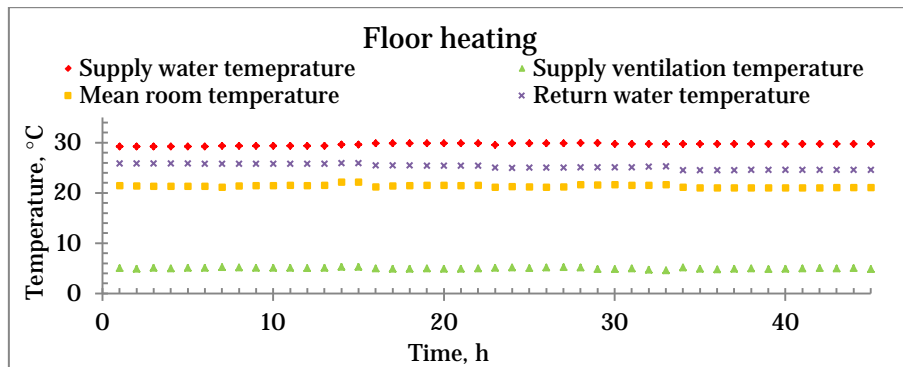


Figure 6 Temperature measurements for floor heating

Table 4 Standard deviation of measured parameters

Standard deviation	Water temperature	Air temperature	Surface temperature	Heat output
Floor heating	1 % (0.2 °C)	1 % (0.2 °C)	1 % (0.1 °C)	6 % (21 W)
Ventilation radiator	1 % (0.3 °C)	1 % (0.2 °C)	1 % (0.2 °C)	4 % (13 W)
Conventional radiator	1 % (0.3 °C)	1 % (0.2 °C)	2 % (0.3 °C)	2 % (8 W)

4. Conclusion

In this study, we experimentally and analytically investigated the performance of different room heaters with different supply temperatures in terms of energy consumption, environmental impacts, and thermal environment. The three heat emitters studied were floor heating, ventilation radiator, and conventional radiator as very-low-, low-, and medium-temperature heat emitters, respectively. The experimental investigation was conducted in a climate chamber at Technical University of Denmark (DTU). In the investigated room, due to negative pressure, $12.6 \text{ l}\cdot\text{s}^{-1}$ fresh air was introduced with a constant temperature of $5 \text{ }^\circ\text{C}$ from above the windows in the case of the conventional radiator and floor heating, and from behind and then through the radiator in the case of the ventilation radiator. Considering an evening situation with no external solar heat gains, internal heat gains came from two seated persons, two laptops and two lamps, generating in total 204 W . The considered U value for external wall ($0.25 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$) and window ($2.1 \text{ W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$) in the simulated room represented a room in an average multi-family building built between 1976 and 2005. Results showed that all heat emitters provided an acceptable level of thermal comfort in the room with $22 \text{ }^\circ\text{C}$ at the reference point. Measurements showed that the supply water temperature to the ventilation radiator and floor heating was around $33 \text{ }^\circ\text{C}$ and $30 \text{ }^\circ\text{C}$, respectively. These temperatures were much (between $12\text{--}15 \text{ }^\circ\text{C}$) lower than conventional radiator for covering the same heat demand of $20 \text{ W}\cdot\text{m}^{-2}$ under the same conditions; the supply water temperature for the conventional radiator was approximately $45 \text{ }^\circ\text{C}$. In case of connecting to a ground-source heat pump, the energy consumed for ventilation radiator and floor heating was 17 and 22 (%) lower than with a conventional radiator, respectively. This would cause 21 and 18 (%) lower CO_2 emissions for floor heating and ventilation radiator, respectively, compared to a conventional radiator. As shown by this study, in order to achieve greater efficiency and savings in energy and environmental impacts in the heating system of buildings there is a need to decrease supply temperature to the heating system. In low-temperature heating systems losses from production units, distribution pipes and emitters are reduced. In addition, higher efficiency is attained from heat pumps and fourth generation of district heating networks.

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