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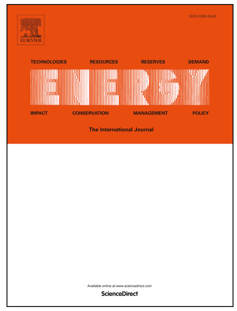

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A theoretical study of the effects of different heating loads on the exergy performance of water-based and air-based space heating systems in buildings

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Abstract

The present study investigated theoretically the exergy performance of floor heating, radiators, and air heating under three different space heating loads of 10, 30 and 50 W/m^2. The effects of different supply and return water temperatures were studied for the radiators, and the effects of different supply air temperatures were studied for the air heating system. All systems were assumed to be connected to a boiler. The floor heating system was further analyzed assuming an air-to-water heat pump, and a ground-source heat pump.

Floor heating was the optimal heating system due to its low exergy demand. The separation of thermal environmental conditioning and ventilation was an efficient solution. The results prove thermodynamically that renewables (i.e., ground source heat in the present study) and low temperature heating systems (i.e., floor heating in the present study) are a resource- and exergy-efficient combination proven by their exergy efficiencies up to 10.5%. The critical COP concept was validated (2.57 in this study).

The power use of auxiliary components might seem negligible in terms of energy; however, it is critical in terms of exergy as it affects the exergy performance drastically. The relative importance of auxiliary power becomes more critical at low space heating loads.

Keywords: exergy analysis; floor heating; radiators; air heating; auxiliary component; ground-source heat pump

1. Introduction

Heating and cooling systems in buildings consist of three main parts: heating and cooling plant (generation), distribution, and heat emission and removal. Heat emission and removal systems are also known as indoor terminal units. The energy performance of heating and cooling systems and the resulting indoor environments depend strongly on the choice of indoor terminal units [1]. Water-based and air-based indoor terminal units operate with different principles (i.e., they use different heat transfer media and rely on different heat transfer mechanisms, being mainly radiant, mainly convective or a combination of these) and operate at different temperatures. In addition, different forms of energy (i.e., heat and electricity) are used in building heating and cooling systems.

These characteristics require a holistic analysis, and an analysis method other than an energy-based method is needed for this purpose, such as exergy analysis. Exergy analysis enables the comparison of the effects of working temperatures and qualities of different energy sources and flows in systems [2], [3], [4]. Due to these characteristics, exergy analysis has been applied to a wide range of engineering systems, including heating, cooling, ventilation systems, and the built environment [5], [6], [7], [8], [9], [10].

More recently, Choi et al. [11], [12] developed methods to apply unsteady-state exergy analysis to conduction heat transfer and insulated building envelopes. Sayadi et al. [13] studied the exergy performance of heating and cooling systems in a building and pointed out the low exergy efficiency of the systems despite a high energy efficiency. The authors suggested that an exergy-based control strategy could be used to optimize exergy efficiency. A separate review article reached a similar conclusion and suggested that further research should be carried out on exergy-based control algorithms [14].

Several authors compared the exergy performance of different space heating and cooling systems, including various types of heating plants and indoor terminal units [15], [16], [17], [18]. Zmeureanu and Wu [19] compared energy and exergy performances of different residential heating systems and concluded that a radiant floor heating coupled to a ground-source heat pump had the highest exergy efficiency among the systems the authors studied. The authors also pointed out that a mechanical ventilation system should be used unless other natural means of ventilation are not possible. Kilkis [20] studied the possibilities of coupling radiant floors to air-source heat pumps, and showed that the supplementary boilers could be eliminated and this coupling can increase the Coefficient of Performance (COP) of heat pumps. Schmidt [21] compared different heat sources and emission systems (radiator and floor heating) and concluded that a floor heating system performed close to the ideal condition, i.e., the real exergetic demand of a zone. This was mainly due to the low temperature heating possibility of water-based radiant floor heating systems. Kazanci et al. [22], [23] compared the exergy performance of several heating and cooling systems, and concurred with Schmidt [21] that floor heating and cooling systems had lower exergy demands, and hence, lower exergy consumption than other systems such as radiators and air heating in the heating season, and air cooling in the cooling season. One limitation of these studies was that the systems were studied under only one space heating and cooling load, while it is crucial to study the performance of these systems under different space heating and cooling loads to properly characterize their performance, i.e., how differences in the rate at which heat must be provided to or removed from the indoor space affect the overall performance.

To address this gap, the present study compared theoretically, by means of calculations, the exergy performance of different water-based (floor heating, and radiators) and air-based heating systems under different space heating loads. The effects of different supply and return water temperatures were studied for the radiators, and the effects of different supply air temperatures were studied for the air heating system. All systems were assumed to be connected to a natural-gas fired boiler. The floor heating system was further analyzed assuming an air-to-water heat pump, and with a ground-source heat pump. The auxiliary exergy used for pumps and fans in the different systems was also compared. The calculation methodology that was developed could be applied to other building types, although in the present case a single-family house was used as a case study.

2. Details of the space heating systems analyzed

2.1. Case study building and the space heating loads

A detached, one-story, single-family house with a floor area of 66 m^2 and a conditioned volume of 213 m^3 was selected for the present analyses. This experimental house was located in Denmark. Kazanci and Olesen in [24] reported the results of extensive measurements of energy performance and the indoor environmental conditions in this house. In the analyses reported in the present study, measurements from the experimental house were used when possible (e.g., hydronic floor heating structure, pumps, fans, and so forth). The construction details, description and details of the heating, cooling and ventilation systems of the experimental house are given in [22], [25], [26].

It was assumed that the heating demand of the house was addressed with different water-based (floor heating, and radiators) and air-based (air heating via warm air supply to the indoor space) heating systems. The space heating load used in [22] was 31 and 33 W/m^2 for the air-based and water-based heating systems, respectively, which corresponded to the heating loads of the experimental house. In the present study, space heating loads of 10 and 50 W/m^2 for the air-based system, and 12 and 52 W/m^2 for the water-based systems were assumed in addition to the heating loads used in [22]. The heating load of 10 (12) W/m^2, which is 21 W/m^2 less than 31 (33) W/m^2, was low enough to make a heating system analysis reasonable and feasible. This value corresponds to the peak heating demand suggested by the Passive House Institute [27]. The heating load of 50 (52) W/m^2 was selected so that it had the addition of 19 W/m^2, from 31 (33) to 50 (52) W/m^2, and was high enough to represent the trend of the results. Although the reduction of 21 W/m^2 and the addition of 19 W/m^2 are not exactly the same, preliminary analyses showed that these three heating loads were representative to characterize how the heating systems performed under different conditions. These two additional heating loads correspond to two hypothetical houses that are otherwise identical to the experimental house, although their building envelopes perform differently due to less or more heat loss to the outdoor environment. The reason for the slightly higher space heating loads in the water-based heating cases is explained in the following.

2.2. Determination of the necessary values for calculations

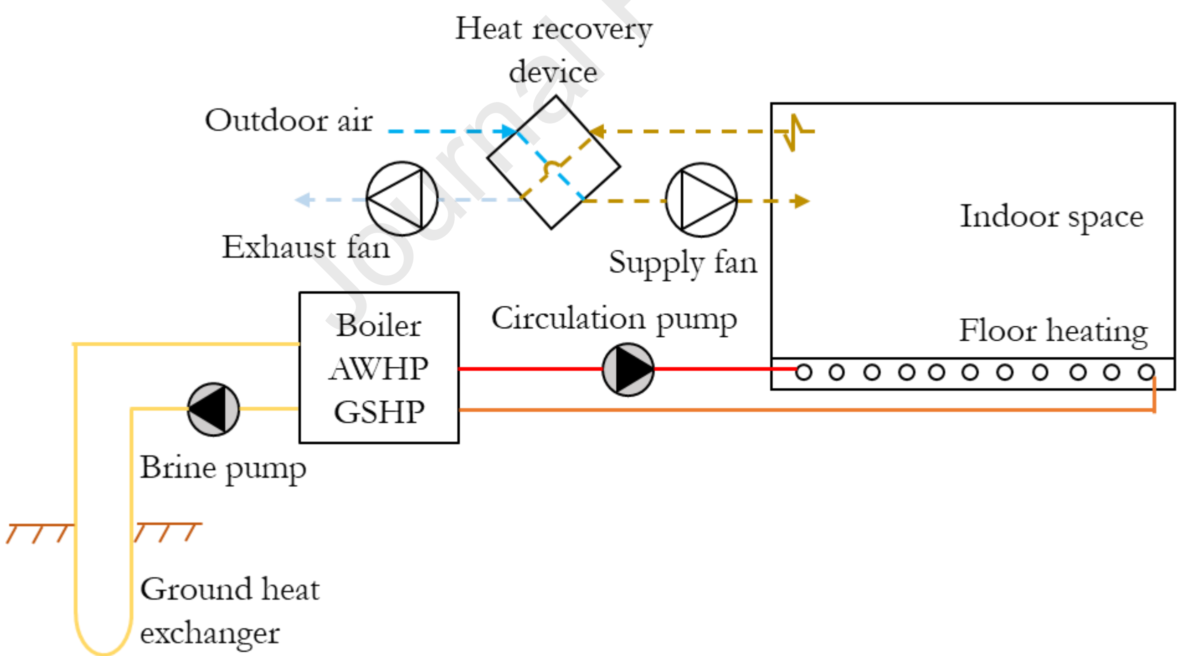
2.2.1. Common values for all systems

All calculations were carried out under a steady-state condition. The outdoor air temperature was taken to be -5°C, which was also the environmental (reference) temperature for the exergy calculations, and the room temperature was assumed to be 20°C (air temperature and mean radiant temperature were considered to be equal to each other and the indoor air was assumed to be fully mixed). An infiltration rate of 0.2 ach was assumed.

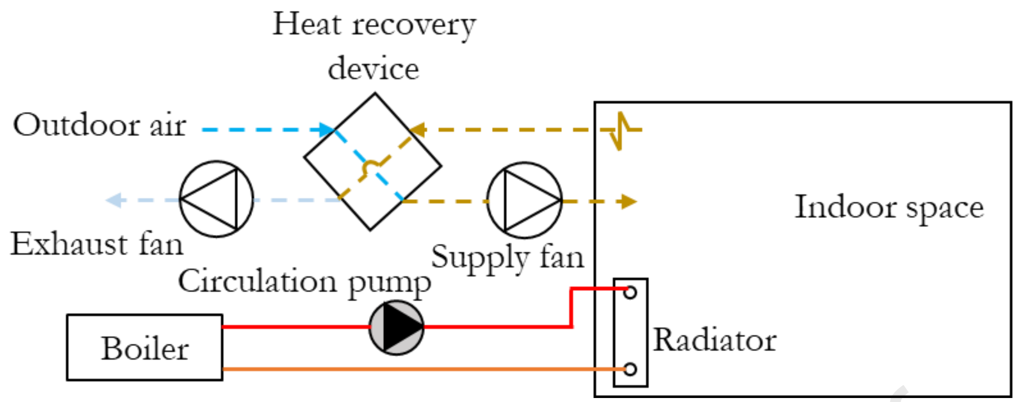
In the air heating cases, the supply air flow rate was determined based on the space heating load and the temperature difference between the supply air and room temperature. A ventilation rate of 0.5 ach was used in the water-based heating cases with a supply air temperature of 16.3°C [28].

The reason for the slightly higher space heating load in the water-based heating cases is that, in air-based heating cases, the warm air was supplied into the space at temperatures higher than the room temperature of 20°C (i.e., at a supply air temperature of 25, 30 or 35°C), therefore it did not contribute to the space heating load. However, in the water-based heating systems, there was not active heating of the supply air but only passive heating through the heat recovery unit. The supply air temperature in the water-based heating cases was 16.3°C, which was lower than the room temperature of 20°C, and therefore it resulted in a slight increase in the space heating load.

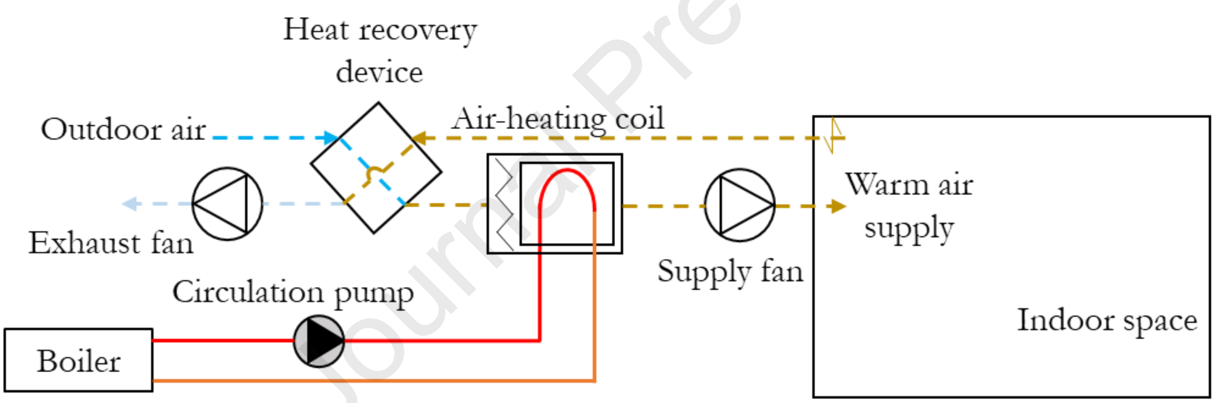
Figure 1 shows the schematic drawings of the systems analyzed.



a) Floor heating



b) Radiator heating



c) Air heating

Figure 1. Schematic drawings of the heating systems that were analyzed: (a) Floor heating, (b) Radiator heating, (c) Air heating (AWHP: Air-to-water heat pump, GSHP: Ground-source heat pump).

2.2.2. Floor heating

The active floor heating area was 45 m^2, which corresponded to 68% of the total floor area. The same floor heating structure as described in [22] was used with a floor covering thermal resistance of 0.05 m^2K/W

(corresponding to a wooden floor covering with a thermal conductivity of 0.13 W/mK and a thickness of 6.5 mm).

Water flow rate, supply and return water temperatures, and floor surface temperatures were calculated for each space heating load, according to [29], [30], [31]. The floor heating system was assumed to be connected to different heating plants, i.e., a boiler, an air-to-water heat pump (AWHP), and a ground-source heat pump (GSHP). Table 1 summarizes the floor heating systems.

Table 1. Summary of the floor heating cases.

Space heating load [W/m^2]	Case name	Specific heat output from floor heating [W/m^2]	Supply / return water temperature [°C]	Floor surface temperature [°C]	Water flow rate [kg/h]
12	FH_10W/m^2	17.7	26.3 / 22.3	21.9	171
33	FH_30W/m^2	48.4	33.0 / 29.0	24.7	469
52	FH_50W/m^2	76.5	39.3 /35.3	27.1	741

2.2.3. Radiator heating with different working temperatures

Three sets of working temperatures were studied under all space heating loads. These temperatures were 45/35, 55/45, and 70/55 (supply/return water temperature in °C) [32]. The radiator type was assumed to be a double panel steel radiator with extended surface area (fins) [33]. The required flow rates in the radiators were determined according to the space heating load and the difference between the supply and return water temperatures at each radiator. The radiators were assumed to be connected to a boiler.

The average surface temperatures of the radiators were assumed to be 0.3°C lower than the simple average value of supply and return water temperatures. This value was determined so that not only the energy balance but also the entropy and exergy balance equations were satisfied, as explained in [22]. Table 2 summarizes the radiator heating cases.

Table 2. Summary of the radiator heating cases.

Space heating load [W/m^2]	Case name	Supply / return water temperature [°C]	Surface temperature [°C]	Water flow rate [kg/h]
12	R45_10W/m^2	45 / 35	39.7	68
	R55_10W/m^2	55 / 45	49.7	68
	R70_10W/m^2	70 / 55	62.2	46
33	R45_30W/m^2	45 / 35	39.7	188
	R55_30W/m^2	55 / 45	49.7	188
	R70_30W/m^2	70 / 55	62.2	125
52	R45_50W/m^2	45 / 35	39.7	297
	R55_50W/m^2	55 / 45	49.7	297
	R70_50W/m^2	70 / 55	62.2	198

2.2.4. Air heating with different supply air temperatures

The air heating system was assumed to consist of a heat recovery device and an air-heating coil. The heat recovery device (a cross-flow heat exchanger) had a heat recovery efficiency of 85% (sensible heat). Outdoor air at -5°C entered the heat recovery device and left it at 16.3°C, which was also the temperature of the air entering the air-heating coil. Different supply air temperatures of 35 [22], 30 and 25°C were assumed for each space heating load.

The air-heating coil was connected to a boiler. The supply and return water temperatures to and from the air-heating coil were 50 and 39°C, respectively [33]. For each supply air temperature under different space heating loads, new supply air flow rates were calculated, and the heat output from the air-heating coil was adjusted accordingly by varying the water flow rate. Table 3 summarizes the air heating cases.

Table 3. Summary of the air heating cases.

Space heating load [W/m^2]	Case name	Supply air temperature [°C]	Air flow rate [m^3/h] (ach)	Water flow rate in air-heating coil [kg/h]
10	AH35_10W/m^2	35	132 (0.6)	65
	AH30_10W/m^2	30	199 (0.9)	71
	AH25_10W/m^2	25	397 (1.9)	91
30	AH35_30W/m^2	35	410 (1.9)	201
	AH30_30W/m^2	30	614 (2.9)	221
	AH25_30W/m^2	25	1229 (5.8)	281
50	AH35_50W/m^2	35	662 (3.1)	324
	AH30_50W/m^2	30	993 (4.7)	357
	AH25_50W/m^2	25	1986 (9.3)	454

2.2.5. Pump and fan powers

For the floor heating and radiator cases, it was assumed that there was a pump circulating the water between the boiler or heat pump and the floor loops or radiators. Additionally, two identical fans for air supply and exhaust were assumed. For the air heating cases, it was assumed that there was a pump circulating the water between the boiler and the air-heating coil, and two identical fans for air supply and exhaust.

The pump power for different cases was obtained from the pump specifications (performance curve of a real pump) as a function of the water flow rate and the required pressure increase [22].

Measurements from the experimental house were used to determine the fan power. The measurements from the house showed that the air-handling unit (AHU) was using 68 W with a ventilation rate of 0.5 ach (105 m^3/h) [25], which corresponds to a total Specific Fan Power (SFP) of 2331 W/(m^3/s) for two fans, and 1166 W/(m^3/s) for one fan. This SFP value is in the SFP 3 category according to EN 13779:2007 [34]. Assuming that the fans for the air heating cases are also in the SFP 3 category (1200 W/(m^3/s)), the fan power was calculated as a function of the air flow rates. Table 4 summarizes the pump and fan power for the different cases.

Table 4. Summary of the pump and fan powers for different heating cases (all cases with boiler).

Space heating load [W/m^2]	Heating system	E_{pump} [W]	E_{fans} [W]*	E_{total} [W]
12	Floor heating	24	67.9	91.9
33	Floor heating	27.5	67.9	95.4
52	Floor heating	34	67.9	101.9
12	R45 and R55	21.5	67.9	89.4
	R70	21	67.9	88.9
33	R45 and R55	24.5	67.9	92.4
	R70	23	67.9	90.9
52	R45 and R55	26	67.9	93.9
	R70	25	67.9	92.9
10	AH35	21.5	88.3	109.8
	AH30	21.5	132.4	153.9
	AH25	22	264.8	286.8
30	AH35	25	273	298
	AH30	25.5	409.5	435
	AH25	26	819.1	845.1
50	AH35	26.5	441.3	467.8
	AH30	26.5	662	688.5
	AH25	27.5	1324	1351.5

* Includes both supply and exhaust fans, power input to these two fans are identical.

2.2.6. Heat and power generation

In the first part of the analyses, it was assumed that the space heating systems were connected to a natural-gas fired condensing boiler with an efficiency of 90% [4], [35]. The ratio of chemical exergy to the higher heating value of natural gas was 0.93 [4]. It was assumed that the electricity provided to the heat pump, pumps, and fans was generated in a remote, natural-gas fired power plant. The conversion efficiency at the power plant, transmission and distribution efficiencies combined was assumed to be 0.35 [4].

In the second part of the analyses, it was assumed that the floor heating system was connected to an air-to-water heat pump, and to a ground-source heat pump. The COP of the air-to-water heat pump was obtained from the manufacturer's datasheets as a function of the outdoor air temperature and the temperature of the water leaving the condenser (assumed to be equal to the supply water temperature for the floor loops). The methodology described in [10] and [36] was used to estimate the COP of the ground-source heat pump. The brine pump was assumed to be identical to the circulation pump and, hence, the same pump specifications were used to calculate the power usage.

Table 5 summarizes the COP, electricity input to the heat pumps and the electricity input to brine pumps for the cases in which the floor heating was connected to an air-to-water heat pump (Floor heating + AWHP) or a ground-source heat pump (Floor heating + GSHP). For the floor heating cases where the floor heating was connected to an air-to-water heat pump or a ground-source heat pump, the pump and fan powers were the same as listed in Table 4.

Table 5. Summary of COPs, electricity inputs to the heat pumps and brine pumps for FH+AWHP and FH+GSHP cases.

Space heating load [W/m^2]	Heating system	COP [-]	E_{HP} [W]	$E_{brine\ pump}$ [W]
12	Floor heating + AWHP	2.78	285.8	N/A
33	Floor heating + AWHP	2.63	828.9	N/A
52	Floor heating + AWHP	2.33	1477.4	N/A
12	Floor heating + GSHP	4.33	183.5	24.6
33	Floor heating + GSHP	3.57	611.2	29.2
52	Floor heating + GSHP	3.08	1118.4	33.8

3. Basic definitions of exergy and calculation methodology

3.1. Basic definitions

For any system under consideration, it is possible to obtain the exergy balance equation from energy and entropy balance equations. In its general form, an exergy balance equation can be obtained as follows [4]. The energy and entropy balance equations for a system are

$$[Energy\ input] = [Energy\ stored] + [Energy\ output] \quad (1)$$

$$[Entropy\ input] + [Entropy\ generated] = [Entropy\ stored] + [Entropy\ output] \quad (2)$$

In its general form, exergy = energy – entropy · T_o. It is then possible to obtain the exergy balance equation as Eq. (1) – Eq. (2) · T_o.

$$[Exergy\ input] - [Exergy\ consumed] = [Exergy\ stored] + [Exergy\ output] \quad (3)$$

where [Exergy consumed] = [Entropy generated] · T_o, and T_o is the environmental (reference) temperature [K], in which the system and its components are situated. The storage terms in Eqs. (1), (2) and (3) disappear under steady-state conditions.

Eqs. (1), (2) and (3) indicate that every system consumes a part of the supplied exergy and that entropy is generated. This applies to both heating and cooling systems in buildings.

3.2. Heating exergy load

The heating exergy load is the demand that the heating system has to address [2]. It is defined as

$$X_{heating} = Q_{heating}\left(1 - \frac{T_o}{T_i}\right) \quad (4)$$

where $X_{heating}$ is the heating exergy load [W], $Q_{heating}$ is the space heating (thermal energy) load [W], T_o is outdoor (environmental) temperature [K] and T_i is the operative temperature [K].

3.3. Exergy supplied to the indoor space by heating systems

The exergy supplied to the indoor space from heated floor surfaces, radiators, and from warm air are given in Eqs. (5), (6) and (7), respectively:

$$X_{FH,out} = Q_{heating}\,(1 - \frac{T_o}{T_{S,FH}}) \quad (5)$$

$$X_{R,out} = Q_{heating}\,(1 - \frac{T_o}{T_{S,R}}) \quad (6)$$

$$\Delta X_{AH,out} = V_{sa} c_a \rho_a \left\{(T_{sa} - T_i) - T_o \ln \frac{T_{sa}}{T_i}\right\} \quad (7)$$

where $X_{FH,out}$ is the exergy supplied from the heated floor to the indoor space [W], $T_{S,FH}$ is the average temperature of the heated floor surface [K], $X_{R,out}$ is the exergy supplied from the radiator to the indoor space [W], $T_{S,R}$ is the average surface temperature of the radiator [K], $\Delta X_{AH,out}$ is the net exergy supplied by warm air to the indoor space (the difference in the exergy flows between the supply air and the indoor air) [W], V_{sa} is the volumetric flow rate of supply air [m^3/s], c_a is the specific heat capacity of air [J/kgK], ρ_a is the density of air [kg/m^3], and T_{sa} is the temperature of the supply air [K].

The exergy consumed in the indoor space is the difference between the exergy supplied to the indoor space and the heating exergy load.

3.4. Exergy consumption in the floor, radiator, air-heating coil, and heat recovery device

The exergy consumption in the floor structure and in the radiator can be obtained from the exergy balance for these terminal units as

$$\Delta X_W + E_{pump} - X_c = X_{out} \quad (8)$$

$$\Delta X_w = X_{w,supply} - X_{w,return} \quad (9)$$

where ΔX_w is the difference between the rate of exergy of the supply and return water (net exergy input) [W], E_{pump} is the pump power as indicated in Table 4 for floor heating and radiator cases [W], $X_{w,supply}$ is the exergy of the supply water flow into the floor or radiator [W], $X_{w,return}$ is the exergy of the return water flow from the floor or radiator [W], X_c is the exergy consumption rate within the terminal unit [W], and X_{out} is the exergy supplied to the indoor space from the terminal unit [W], given by Eq. (5) for floor heating and by Eq. (6) for radiator heating, respectively. Eqs. (8) and (9) apply to both floor heating and radiator heating.

The exergy of the supply and return water flows is calculated as

$$X_w = V_w c_w \rho_w \left\{ (T_w - T_o) - T_o \ln \frac{T_w}{T_o} \right\} \tag{10}$$

where X_w is the rate of exergy delivered by the water flow [W], V_w is the volumetric flow rate of water [m^3/s], c_w is the specific heat capacity of water [J/kgK], ρ_w is the density of water [kg/m^3], and T_w is the temperature of the water [K].

The exergy consumption in the air-heating coil in the AHU can be obtained as

$$\Delta X_w + E_{pump} + E_{supply,fan} - X_c = \Delta X_a \tag{11}$$

$$\Delta X_w = X_{w,supply} - X_{w,return} \tag{12}$$

$$\Delta X_a = X_{a,out} - X_{a,in} \tag{13}$$

where E_{pump} is the pump power [W] and $E_{supply,fan}$ is the supply fan power [W] as indicated in Table 4 for the air heating cases, $X_{w,supply}$ is the rate of exergy supplied by the water entering the air-heating coil from the boiler [W], $X_{w,return}$ is the rate of exergy contained by the water leaving the air-heating coil to the boiler [W], $X_{a,out}$ is the rate of exergy contained by the air leaving the air-heating coil [W], and $X_{a,in}$ is the rate of exergy contained by the air entering the air-heating coil [W]. $X_{w,supply}$ and $X_{w,return}$ are calculated from Eq. (10) using the respective temperatures. $X_{a,in}$ and $X_{a,out}$ are calculated using the following Eq. (14).

$$X_a = V_a c_a \rho_a \left\{ (T_a - T_o) - T_o \ln \frac{T_a}{T_o} \right\} \tag{14}$$

where X_a is the rate of exergy delivered by the air flow [W], V_a is the volumetric flow rate of air [m^3/s], and T_a is the temperature of the air flow [K].

The exergy consumption in the heat recovery device can be obtained through the exergy balance equation for the heat recovery device as

$$X_{outdoor\ air} + X_{exhaust\ air} + E_{exhaust,fan} - X_c = X_{inlet\ air} + X_{discharge\ air} \tag{15}$$

where $X_{outdoor\ air}$ is the rate of exergy flowing in by the intake air from outdoors (=0) [W], $X_{exhaust\ air}$ is the rate of exergy brought in by the exhaust air from the indoor space [W], $E_{exhaust,fan}$ is the exhaust fan power [W] as indicated in Table 4 for air heating cases, $X_{inlet\ air}$ is the rate of exergy contained by the inlet air (supply air for the floor heating and radiator heating cases, and the air entering the air-heating coil in the air heating cases) [W], and $X_{discharge\ air}$ is the rate of exergy contained by the discharge air, which is discarded to the environment after the heat recovery [W]. Eq. (14) is used for calculating $X_{outdoor\ air}$, $X_{exhaust\ air}$, $X_{inlet\ air}$, $X_{discharge\ air}$.

3.5. Exergy input to the boiler, power plant, and the ground-source heat pump

The exergy input to the boiler can be calculated using Eq. (16).

$$X_{in,boiler} = \frac{Q_{boiler}}{\eta_{boiler}} r \tag{16}$$

where $X_{in,boiler}$ is the exergy input to the boiler through natural gas [W], Q_{boiler} is the rate of thermal energy to be provided by the boiler [W], η_{boiler} is the boiler efficiency, and r is the ratio of chemical exergy to higher heating value of natural gas (0.93).

For the floor heating cases when a heat pump was assumed to be the heat source, the exergy input required at the power plant for generating the electricity input to the heat pump was calculated as

$$E_{HP} = \frac{Q_{heating}}{COP} \tag{17}$$

$$X_{in,power\ plant} = \frac{E_{HP}}{\eta_{TOT}} r \tag{18}$$

where E_{HP} is power (electricity) input to the heat pump [W], COP is the coefficient of performance as given in Table 5, $X_{in,\ power\ plant}$ is the rate of exergy input to the power plant from the natural gas [W], and η_{TOT} is the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (0.35). Eqs. (17) and (18) apply both for the air-to-water heat pump and the ground-source heat pump.

The exergy input required at the power plant for the pump and fans was also calculated using Eq. (18) by replacing the E_{HP} with respective pump power (E_{pump} and $E_{brine\ pump}$) and fan power (E_{fans}).

The exergy balance equation for the ground-source heat pump is as follows [7], [37].

$$\Delta X_{brine} + E_{brine\ pump} + E_{HP} - X_{C,HP} = \Delta X_w \tag{20}$$

where ΔX_{brine} is the net rate of exergy input to the heat pump by the brine (which is also the net exergy input to the brine from the ground) [W], $E_{brine\ pump}$ is the brine pump power as given in Table 5 for GSHP cases [W], $X_{C,HP}$ is the exergy consumption rate in the ground-source heat pump, and ΔX_w is the net rate of exergy input to the water circulating through the condenser of the heat pump [W] calculated by Eq. (9). ΔX_{brine} is obtained by Eq. (21) as

$$X_g - X_{c,g} = \Delta X_{brine} \tag{21}$$

$$X_g = Q_g \left(1 - \frac{T_o}{T_g}\right) \tag{22}$$

$$\Delta X_{brine} = V_{brine} c_{brine} \rho_{brine} \left\{ \left(T_{brine,out,ground} - T_{brine,in,ground} \right) - T_o \ln \frac{T_{brine,out,ground}}{T_{brine,in,ground}} \right\} \tag{23}$$

where X_g is the rate of warm exergy flow from the ground to brine [W], $X_{c,g}$ is the exergy consumption rate in the ground [W], Q_g is the rate of thermal energy extracted from the ground [W], T_g is the undisturbed ground temperature [K] (281.5 K [26]), V_{brine} is the volumetric flow rate of brine in the ground heat exchanger [m^3/s], c_{brine} is the specific heat capacity of brine [J/kgK] (3898 J/kgK), ρ_{brine} is the density of brine [kg/m^3] (1027 kg/m^3), $T_{brine,out,ground}$ is the temperature of brine leaving the ground [K] (278.2 K), and $T_{brine,in,ground}$ is the temperature of brine entering the ground [K] (275.2 K).

3.6. Exergy efficiency definition

Exergy efficiency provides an alternative way to evaluate and compare the performance of different space heating systems. Exergy efficiency can be described as follows [23].

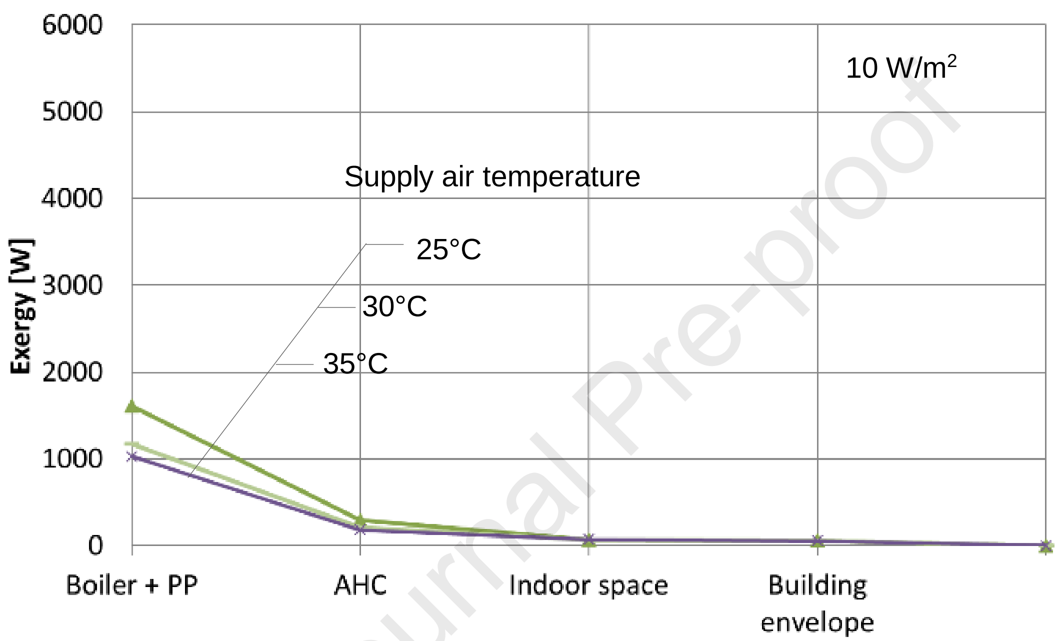
$$\eta_x = \frac{X_{heating}}{X_{in,total}} \tag{24}$$

where η_x is the conventional exergy efficiency description, which is the ratio of space heating exergy load to the total exergy input to the system.

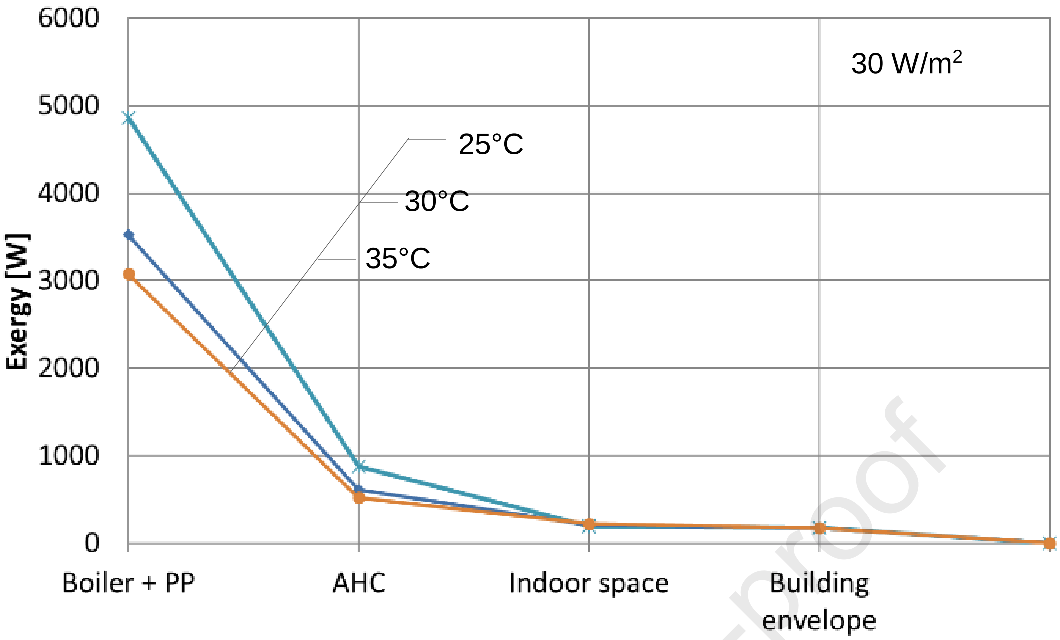
4. Results and discussion

4.1. Exergy performance of different space heating systems

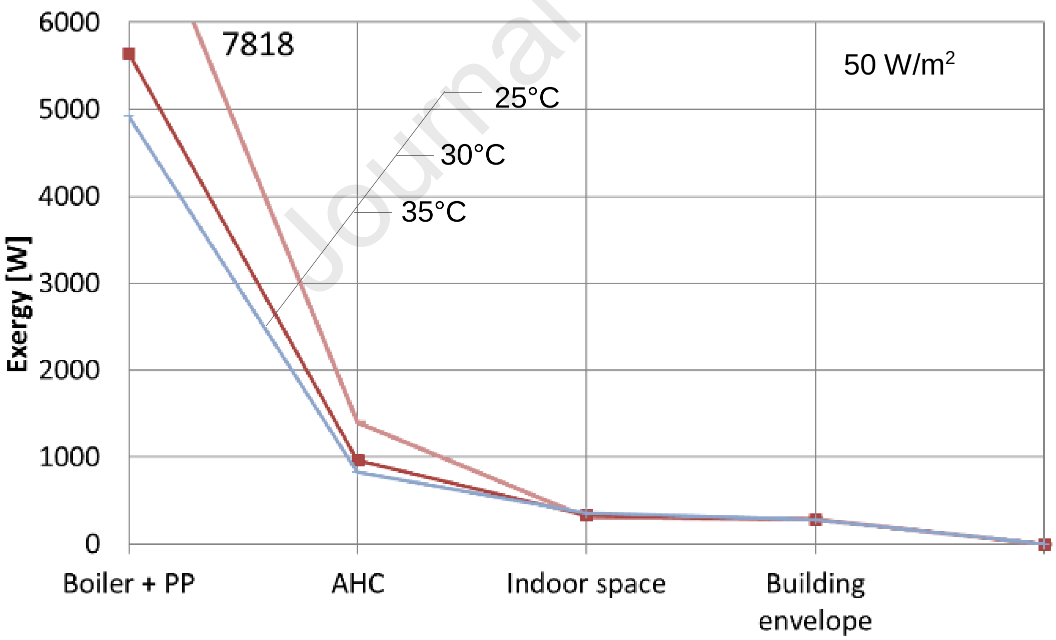
Figure 2 shows the exergy flow patterns from supply, via consumption, to demand for air heating under different space heating loads.



a)



b)



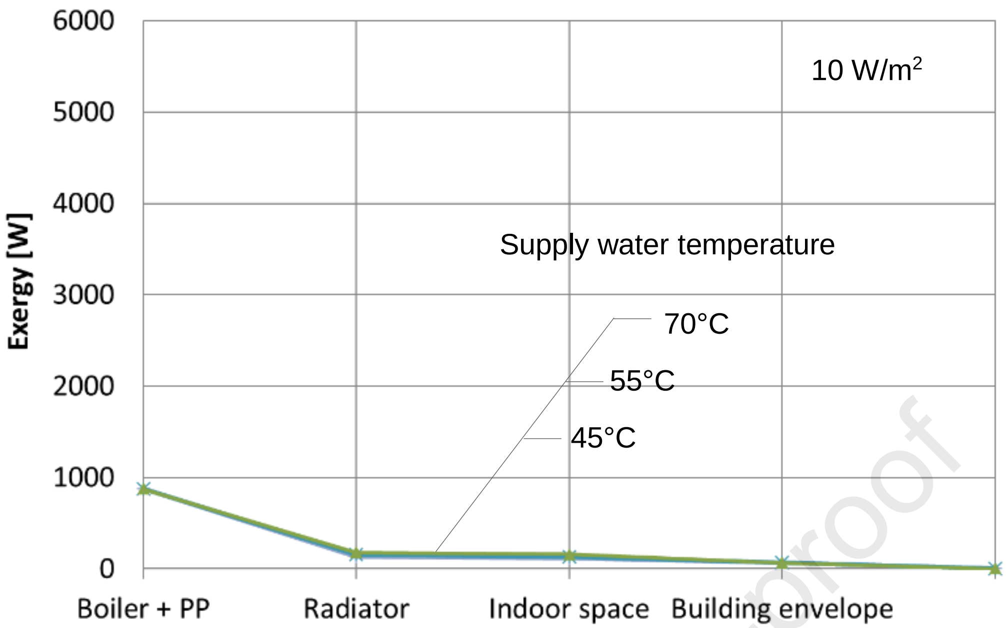
c)

Figure 2. Exergy flows for air heating under different space heating loads: (a) 10 W/m^2, (b) 30 W/m^2, (c) 50 W/m^2, (PP: Power plant, AHC: Air-heating coil).

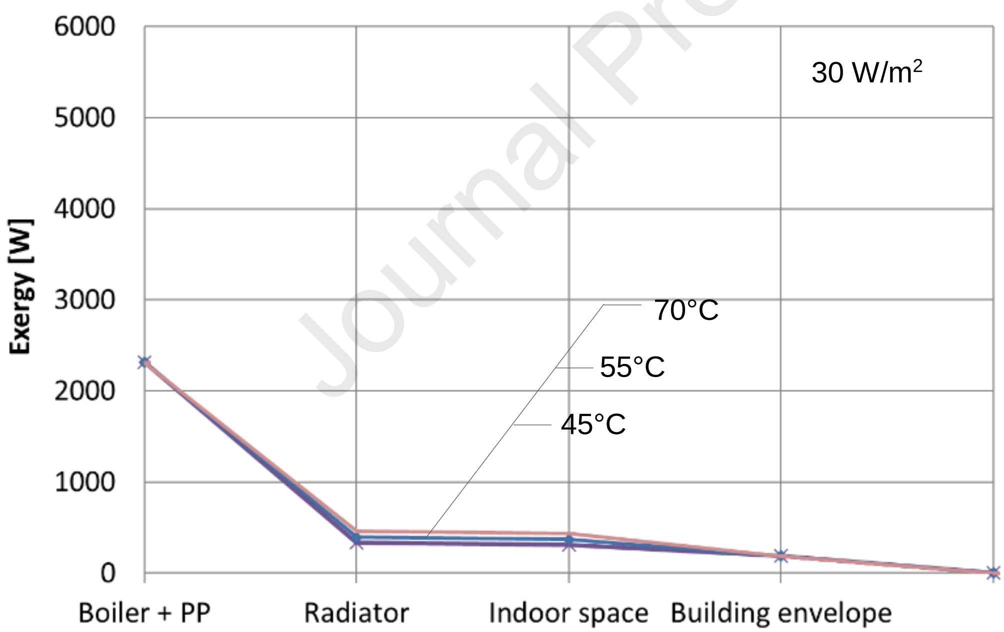
For the air heating systems under the same space heating load, the exergy load of the space was the same, i.e., the points on the vertical lines represented by “Building envelope” are the same in the three curves. However, the systems behave differently depending on the supply air temperature. Under the same space heating load ($Q_{heating}$), exergy consumption in the space, which is the difference in exergy value between “Indoor space” and “Building envelope”, increases, though marginally, with the increasing supply air temperature due to the increasing difference between the supply air temperature and room temperature. On the other hand, exergy consumption in the air-heating coil, which is the difference in exergy value between “AHC” and “Indoor space”, decreases with the increasing supply air temperature (a lower supply air temperature requires a larger amount of air to be supplied to the indoors and results in a larger temperature difference between the supply air and the circulating water in the air-heating coil). A higher exergy consumption rate in the air-heating coil to be able to meet the same space heating load implies that a higher exergy input to the air-heating coil, and hence to the entire system, is necessary. This can also be observed in the required total exergy inputs i.e., the sum of exergy inputs from natural gas to the boiler and to the power plant.

A higher total exergy input was required to meet the same space heating load with decreasing supply air temperature. Under 10, 30 and 50 W/m^2 heating loads, the air heating system with a supply air temperature of 25°C required 56%, 58% and 58% higher exergy input than the air heating system with a supply air temperature of 35°C, respectively. The results in Figure 2-b) and Figure 2-c) show that air heating system with a supply air temperature of 25°C under a 30 W/m^2 space heating load required a total exergy input that was very close to that of the air heating system with a supply air temperature of 35°C under 50 W/m^2 space heating load. This shows that the supply air temperature in air heating systems could be optimized to obtain the lowest possible total exergy input to the system under a given space heating load and that the fan power is critical for the overall system performance. This is discussed in further details in 4.4. Auxiliary energy input to the heating systems.

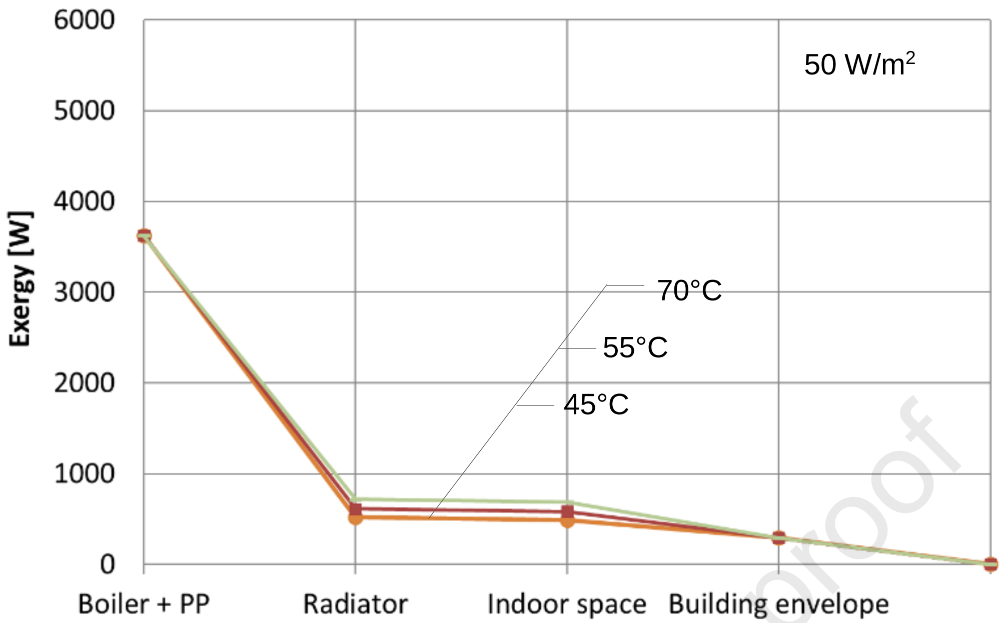
Figure 3 shows the exergy patterns from supply, via consumption, to demand for radiator heating under different space heating loads.


6000
5000
4000
3000
2000
1000
0
Exergy [W]
10 W/m2
Supply water temperature
70°C
55°C
45°C
Boiler + PP
Radiator
Indoor space
Building envelope

a)


6000
5000
4000
3000
2000
1000
0
Exergy [W]
30 W/m2
70°C
55°C
45°C
Boiler + PP
Radiator
Indoor space
Building envelope

b)



c)

Figure 3. Exergy flows for radiator heating under different space heating loads: (a) 10 W/m², (b) 30 W/m², (c) 50 W/m² (PP: Power plant).

The radiator heating systems with different working temperatures have to address the same heating exergy loads under the same space heating loads. The exergy consumption rate in the indoor space increases slightly as the working temperatures increase and with increasing space heating loads. The exergy consumption rate in the radiators is almost the same under the same space heating loads (due to the assumption of the average surface temperature being 0.3°C lower than the average water temperature [22]) but increases as the space heating load increases. The relatively low exergy consumption rate in the radiators compared to the other components in the system can be observed in Figure 3 as this part of the curve (the difference between "Radiator" and "Indoor space") was almost flat. Under the same space heating load, the total exergy input was almost the same regardless of the working temperatures of the radiators, mainly because a boiler is not a flexible heat source that can adjust its inputs and outputs, due to the combustion process involved.

Figure 4 shows the exergy flows for floor heating connected to a boiler, under different space heating loads.

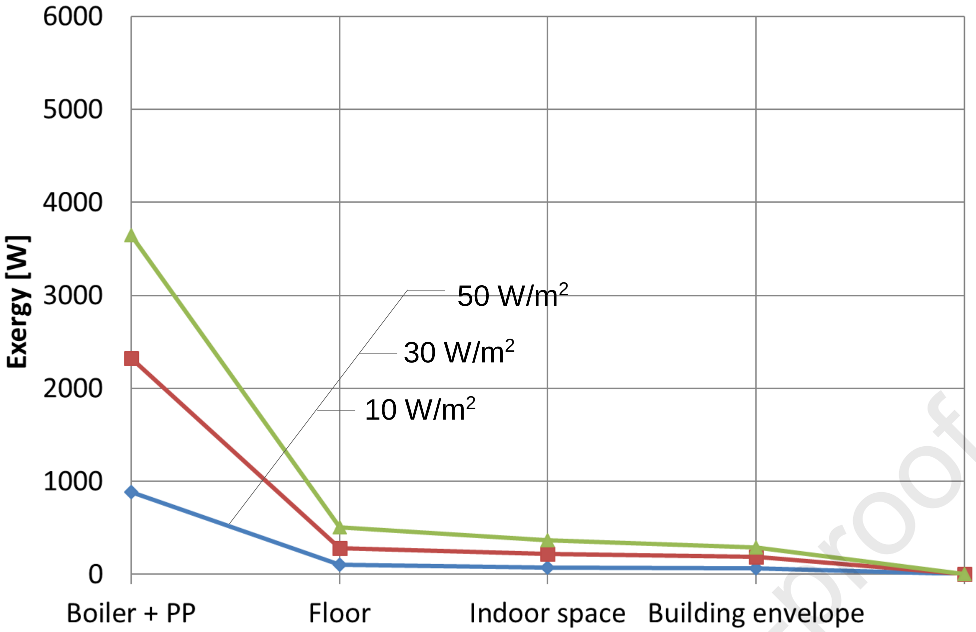


Figure 4. Exergy flows for floor heating under different space heating loads (PP: Power plant).

The exergy consumption rate in the indoor space and floor increased as the space heating load increased. Even though the exergy consumption increased with the increasing space heating load, these curves are much flatter until the boiler and the power plant, compared to the other two heating systems. This is mainly because of the low exergy demand of the floor heating system as identified previously by Schmidt [21] and Kazanci et al. [22]. The high exergy consumption in the boiler shows that a boiler is not a good match to such a low exergy demand due to the inherent limitation of the combustion process; a flame temperature higher than 1000°C in the boiler necessarily results in the largest portion of the whole exergy consumption.

For the 10 W/m^2 heating load, the exergy required at the floor was 25%, 35% and 43% lower than required by the radiators of the R45, R55 and R70 cases, respectively. For the 30 W/m^2 heating load, the exergy required at the floor was 15%, 28% and 38% lower than required by the radiators in the R45, R55 and R70 cases, respectively, and for the 50 W/m^2 heating load, the exergy required at the floor was 3%, 18% and 30% lower than required by the radiators in the R45, R55 and R70 cases, respectively. This was despite the fact that floor heating systems required slightly higher water flow rates than in the radiator heating cases due to the lower temperature difference between supply and return water flows requiring a larger flow rate, and hence, slightly higher power inputs to the circulation pumps. These results show that with an increasing space heating load, higher working temperatures are needed in the floor heating system and its benefits compared to R45 decrease as the working temperatures become closer; however, considerably less exergy was required compared to the R55 and R70 cases. This emphasizes the importance of reducing the space conditioning loads in the design phase.

For the 10 W/m^2 heating load, the exergy required at the floor was 65%, 51% and 44% lower than required by the air-heating coils of the AH25, AH30 and AH35 cases, respectively. For the 30 W/m^2 heating load, the exergy required at the floor was 67%, 53% and 45% lower than was required by the air-heating coils of the AH25, AH30 and AH35 cases, respectively, and for the 50 W/m^2 heating load, the exergy required by the floor was 64%, 48% and 39% lower than was required by the air-heating coils of the AH25, AH30 and AH35 cases, respectively. This was despite the fact that the floor heating had a slightly higher space heating exergy load than the air heating cases.

Despite the differences in the exergy required at the floor and radiator, the total exergy input for the floor heating and radiator heating systems was almost the same for the same space heating loads i.e., 1% higher for 10 W/m^2 heating load, and even less than 1% for 30 and 50 W/m^2 heating loads. This was because both systems had to provide the same amount of heat, i.e., they had the same space heating load and they were connected to a boiler.

For the 10 W/m^2 heating load, the total exergy input (sum of the exergy inputs to the boiler and to the power plant) for the water-based heating systems was 45%, 25% and 14% lower than that required by the air heating systems with supply air temperatures of 25, 30 and 35°C, respectively (Figure 2-a) and the blue line in Figure 4). For the 30 W/m^2 heating load, the total exergy input for the water-based heating systems was 52%, 34% and 24% lower than that required by the air heating systems with supply air temperatures of 25, 30 and 35°C, respectively (Figure 2-b) and the red line in Figure 4), and for the 50 W/m^2 heating load, the total exergy input for the water-based heating systems was 53%, 35% and 26% lower than that required by the air heating systems with supply air temperatures of 25, 30 and 35°C, respectively (Figure 2-c) and the green line in Figure 4).

These results show that the floor heating is the most efficient heating system among the heating systems compared due to its low exergy demand for achieving the same space heating effect, mainly because it is water-based and operates with low temperature heating principle. The results also show that to benefit from this low exergy demand, an appropriate heat source that can this demand with a low exergy supply is required otherwise, it is not possible to benefit fully from the low exergy demand of a heating system.

4.2.Floor heating connected to different heat sources

Figure 5 and Figure 6 show the exergy flow patterns from supply, via consumption, to demand for floor heating connected to an air-to-water heat pump (AWHP), and to a ground-source heat pump (GSHP) under different space heating loads, respectively.

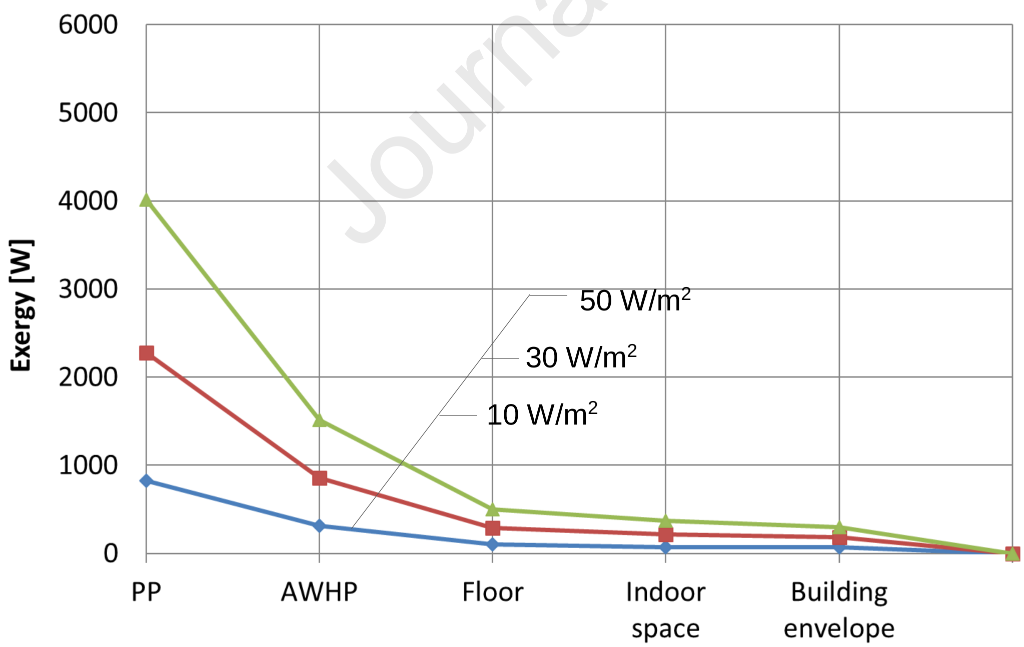


Figure 5. Exergy flows for floor heating connected to an air-to-water heat pump under different space heating loads (PP: Power plant, AWHP: Air-to-water heat pump).

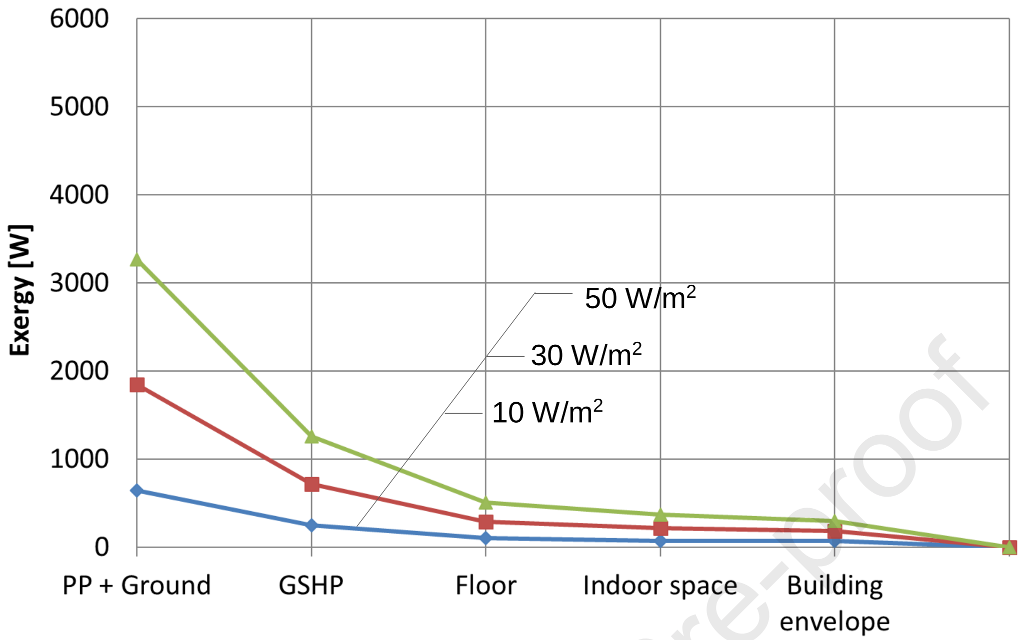


Figure 6. Exergy flows for floor heating connected to a ground-source heat pump under different space heating loads (PP: Power plant, GSHP: Ground-source heat pump).

In Figure 4 - Figure 6, the floor, indoor space and building components are the same; however, the heat source is different. In Figure 4, the exergy input consisted of the inputs to the boiler and to the power plant, in Figure 5 the exergy input was the input to the power plant from natural gas, and in Figure 6, the exergy input consisted of the input to the power plant from natural gas and the warm exergy input from the ground to the brine circulating in the ground heat exchanger.

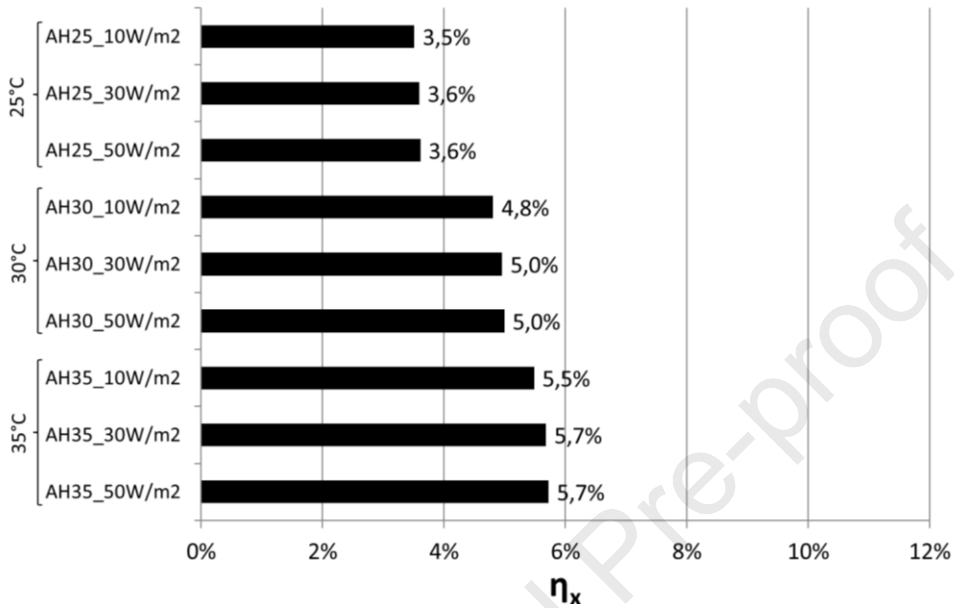
For the 10 W/m^2 heating load, the total exergy input to the floor heating connected to AWHP and to the floor heating connected to GSHP was 7% and 27% lower, respectively, than that of the floor heating connected to a boiler. For the 30 W/m^2 heating load, the total exergy input to the floor heating connected to AWHP and to the floor heating connected to GSHP was 2% and 21% lower, respectively, than that of the floor heating connected to a boiler. For the 50 W/m^2 heating load, the total exergy input to the floor heating connected to AWHP was 10% higher than that of the floor heating connected to a boiler, while the total exergy input to the floor heating connected to GSHP was 11% lower than that of the floor heating connected to a boiler. These results and the flatter shape of the curves of floor heating connected to GSHP indicate that the ground-source heat pump was a better match to the low exergy demand of the floor heating system. This performance was partly due to the higher COP value of the GSHP compared to the AWHP, but mainly due to the warm exergy input from a nearby renewable energy source, i.e., the ground. The higher COP values of the GSHP than those of the AWHP were also due to the use of the surrounding ground as the heat source, instead of the surrounding air. In an earlier study, Shukuya [4] showed that since the surrounding air does not contain any exergy, an air-source heat pump is essentially a machine to divide the exergy (incoming electricity) into warm and cool exergy and make use of these exergies according to the space conditioning requirement. A detailed explanation of the warm and cool exergy concepts can be found in [4] and [7].

The warm exergy flow from the ground to the brine was 29, 74 and 110 W (0.64, 1.64 and 2.44 W/m^2–active floor area) under the 10, 30 and 50 W/m^2 space heating loads, respectively. These relatively small warm exergy inputs from the ground resulted in considerable reductions in the exergy input from fossil fuels; a reduction of 178, 427 and 754 W of exergy input from natural gas can be achieved compared to the AWHP case. A similar analysis was also carried out by Kazanci et al. [23] for a ground heat exchanger and a crawl-space under cooling conditions.

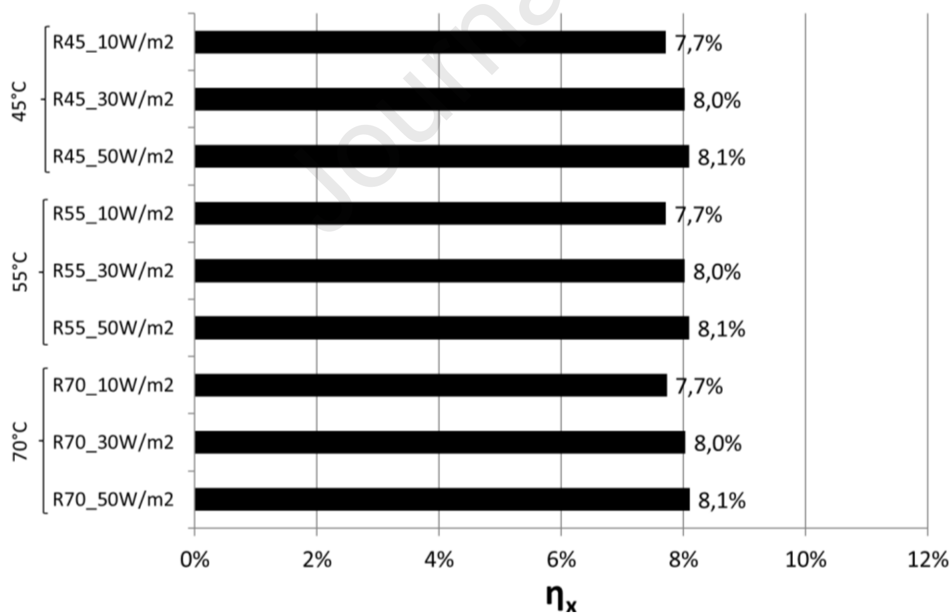
Under the 50 W/m^2 space heating load, floor heating connected to AWHP required a higher exergy input than floor heating connected to boiler. This was mainly due to the low COP of the AWHP. The possibility of such a system behavior was identified by Kazanci et al. [22], who pointed out that there was a critical COP value that a heat pump needed to exceed to be more beneficial than a boiler. In that study, a COP value of 2.57 was identified as the critical COP for the same floor heating and AWHP system under consideration. In the present study, the COP of the AWHP under 50 W/m^2 space heating load was 2.33, confirming the critical COP discussion of [22].

4.3.Exergy efficiency of the systems studied

Figure 7 shows the calculated exergy efficiency values for the different heating systems.



a)



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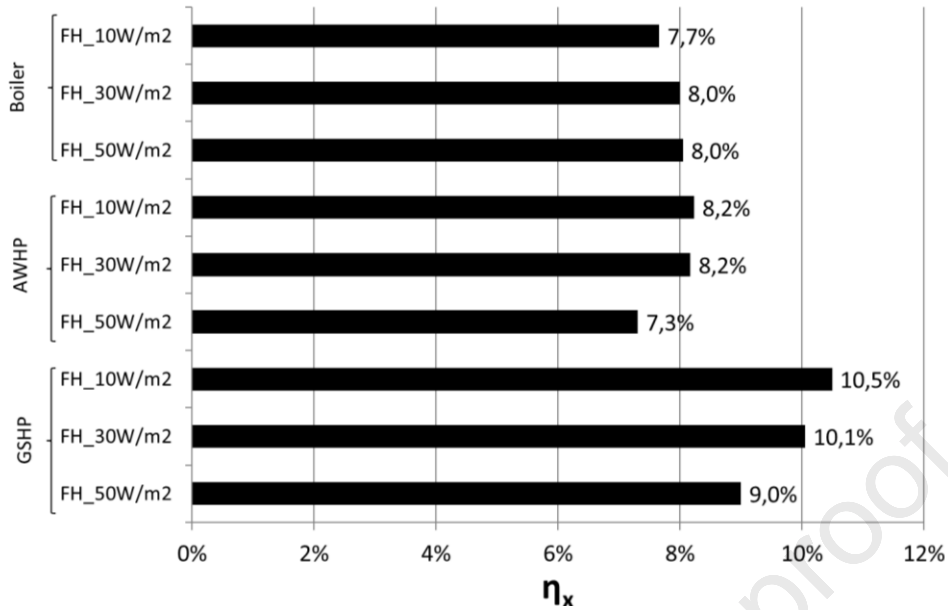

c)

Figure 7. Exergy efficiencies of the different heating systems: (a) air heating, (b) radiators, (c) floor heating.

The exergy efficiency values of the air heating cases show that under the same space heating load, a higher supply air temperature resulted in a higher overall exergy efficiency. This is mainly due to the lower exergy input to the fans at higher supply air temperatures, as opposed to air heating cases with lower supply air temperatures, which require larger supply air flow rates for providing the same space heating effect compared to the systems with higher supply air temperatures.

Water-based heating systems had higher exergy efficiencies than that of the air heating system. Radiator heating and floor heating connected to a boiler had closely similar exergy efficiency values. Floor heating connected to AWHP had slightly higher exergy efficiency values under 10 and 30 W/m^2 space heating loads than radiator heating and floor heating connected to a boiler; however, it had a lower exergy efficiency under the 50 W/m^2 space heating load. This finding supports the previously discussed critical COP concept.

Floor heating connected to GSHP had the highest exergy efficiency among the studied systems with values of 10.5% to 9%. This confirms that low-temperature radiant floor heating and the ground (i.e., warm exergy extracted from the ground) is a good match because of their low exergy demand and low exergy supply, respectively.

For the floor heating systems connected to GSHP, the ratio of the warm exergy input from the ground to the total exergy input to the system was 4.5%, 4.0% and 3.4% under 10, 30 and 50 W/m^2 space heating loads, respectively. This tendency among the values of the ratio and space heating loads explains the decreasing exergy efficiency, as a higher exergy input via electricity is needed with the increasing space heating load, emphasizing the importance of keeping space heating loads low.

4.4. Auxiliary energy input to the heating systems

The required pump powers, fan powers and their totals are summarized in Table 4. Based on those values, Figure 8 shows the required exergy input to the power plant for providing power to the auxiliary components (calculated using Eq. (18)).

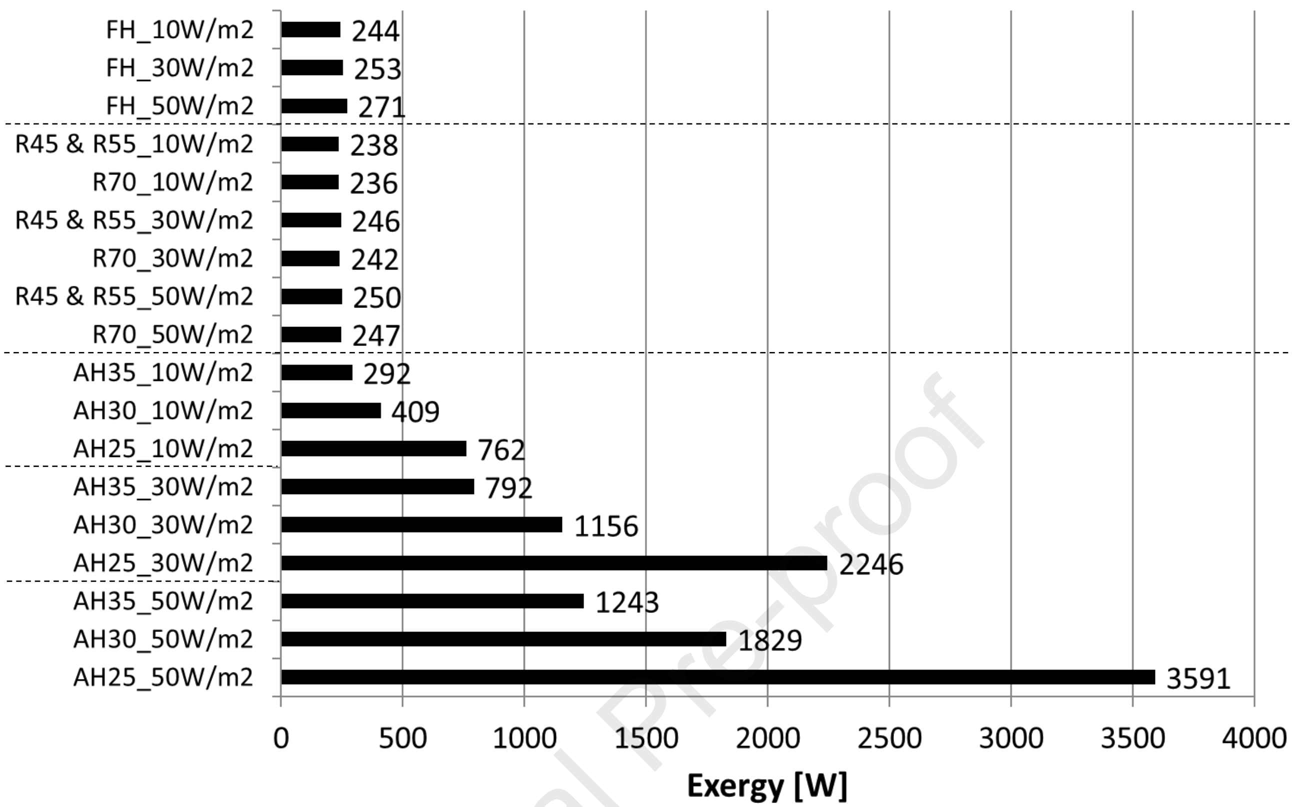

Figure 8. Required exergy input to the power plant for providing electricity to the auxiliary components.

All water-based heating cases (floor heating and radiators) had a ventilation rate of 0.5 ach; the differences between the water-based heating cases were therefore due to the differences in pump power requirements. The floor heating cases had a slightly higher pump power than that of the radiator heating cases due to the smaller temperature drop between the supply and return water flows, leading to a larger water flow rate.

In air-based heating cases, the differences between the cases were due to the different pump power required for each water flow rate in the air-heating coil, but mainly due to the differences in fan power. The air heating systems required higher auxiliary power and correspondingly, a higher exergy input to the power plant for providing electricity to the auxiliary components. For a given space heating load, the auxiliary power required for the air heating systems increased with decreasing supply air temperature. The difference in the required auxiliary power between the water-based heating systems and the air heating system became larger at higher space heating loads. These results indicate a clear benefit for the water-based heating systems regarding the required power for auxiliary components such as pumps and fans.

Table 6 compares the energy and exergy use for auxiliary components with space heating energy and exergy loads.

Table 6. Comparison of the energy and exergy use for auxiliary components and space heating (all cases with boiler)*

Case	FH_10W/m^2	FH_50W/m^2	AH35_10W/m^2	AH25_10W/m^2	AH35_50W/m^2	AH25_50W/m^2
$Q_{heating}$ [W]	794	3442	662	662	3310	3310
$X_{heating}$ [W]	68	294	56	56	282	282
$E_{pump}/Q_{heating}$	3%	1%	3%	3%	1%	1%
$E_{pump}/X_{heating}$	35%	12%	38%	39%	9%	10%
$E_{fans}/Q_{heating}$	9%	2%	13%	40%	13%	40%
$E_{fans}/X_{heating}$	100%	23%	156%	469%	156%	469%
$E_{total}/Q_{heating}$	12%	3%	17%	43%	14%	41%
$E_{total}/X_{heating}$	136%	35%	194%	508%	166%	479%
$X_{in,pp,aux}/X_{in,total}$	23%	7%	25%	39%	23%	37%

* E_{pump}, E_{fans} and E_{total} are given in Table 4. $X_{in,pp,aux}$ for different cases is given in Figure 8. $X_{in,total}$ is the sum of $X_{in,pp,aux}$ and $X_{in,\ boiler}$ (calculated by Eq. (16)).

The ratio of pump power to space heating load (the third line in Table 6) shows that in terms of energy, the pump power can be negligible as it is only 1 to 3% of the space heating load; however, this is not the case when comparing pump power to the space heating exergy load (the fourth line in Table 6). In the case of exergy, the pump power is no longer negligible and can be up to 35% of the space heating exergy load for floor heating and up to 39% of the space heating exergy load for air heating. A similar trend is observed for fans, although the fan power can be up to 40% of the space heating load of air heating systems with 25°C supply air temperature.

According to the exergetic comparison, fan power is much larger compared to space heating exergy load; up to 469% of the space heating exergy load in air heating systems with 25°C supply air temperature.

These trends are reflected in the ratio of the sum of the auxiliary powers for pumps and fans to the space heating energy and exergy loads. This ratio was lower for the floor heating systems compared to the air heating systems. This was an expected result, as the floor heating system is a water-based heating solution and relies mainly on the floor structure for heat emission to the indoor space while the ventilation system is used only for ensuring an acceptable air quality. In the air heating systems, space conditioning relies on the warm air supply to the indoor space. The exergy input required at the power plant for providing the necessary power to the auxiliary components can be up to 39% of the total exergy input to the system in air heating cases and up to 23% in the floor heating cases with low space heating loads (the bottom line in Table 6).

These results show that even though the power use of auxiliary components might seem negligible in terms of energy, it is critical in terms of exergy. The results also show that the relative importance of the power use of the auxiliary components increases and becomes even more critical with decreasing space heating loads. This behavior becomes particularly important when considering global trends and building regulations, which try to minimize the heating and cooling demands of buildings.

4.5. Overall discussion

In the present study, the selected space heating systems were compared only based on their exergy performance. The exergy performance of the systems is crucially important and should be considered during system selection; however, in practice the selection of a heating system could depend on more practical criteria such as costs, control requirements (e.g., zone vs. room control), local building traditions and the location (such as connection to a district heating network, regulations regarding the use of ground and so forth). Thermo- and exergoeconomic [38] analyses can be included in future studies to link the exergy performance of the studied systems and economic consequences.

During the current COVID-19 pandemic and the need to reduce the spread of the disease, several guidelines [39], [40] suggest eliminating air recirculation in indoor spaces and suggest the use of 100% fresh air from outdoors. This requirement limits the use of conventional air-conditioning systems, which usually rely on a mix of recirculation air and outdoor air. A feasible alternative could be the separation of heat emission to and removal from indoor spaces and ventilation, such as a combination of radiant heating and cooling systems with dedicated outdoor air systems (DOAS), which can minimize the risk of spreading airborne pathogens. The water-based heating systems analyzed in the present study used this principle and their benefits in terms of exergy performance were proven.

The present study used exergy analysis to identify the importance of auxiliary components for the overall performance of a heating system. In the present analyses, the auxiliary components consisted of pumps and fans that move the heat transfer medium. In buildings, there are several other auxiliary components such as valves, dampers, actuators and sensors that use electricity. As for pump and fan powers, their power use is also critical from the exergy point of view and their selection should be made by carefully considering their implications for the overall performance of the heating system.

It is common to mention the need for reducing space heating and cooling loads by making improvements such as better building envelope design, while the power use of auxiliary components is commonly overlooked. The present study shows thermodynamically that the power use of auxiliary components is critical for overall system performance, as auxiliary components use electricity as input, which is 100% exergy.

5. Conclusion

The conclusions from the present study are as follows.

1) Floor heating was the optimal heating system under different space heating loads due to its low exergy demand for achieving the same space heating effect, mainly due to being water-based and due to its low

temperature heating principle. Exergy analyses showed that the separation of space conditioning and ventilation was an efficient solution.

2) Floor heating connected to ground-source heat pump had the highest exergy efficiency among the studied systems with values of 10.5% to 9%, depending on the space heating load. A floor heating system should be connected to an appropriate heat source that can match the low exergy demand of floor heating systems with a low exergy supply. The present study proved thermodynamically that renewables (i.e., ground in the present study) and low temperature heating systems (i.e., floor heating in the present study) are a resource- and exergy-efficient match.
3) The critical COP concept [22], which identified a minimum COP value that a heat pump has to exceed to be more beneficial than a boiler, was confirmed and was 2.57 in this study.
4) A relatively small warm exergy input from the ground (29, 74 and 110 W under the 10, 30 and 50 W/m^2 space heating loads, respectively) resulted in large reductions in the exergy input from fossil fuels; 178, 427 and 754 W of exergy input from the natural gas could be saved compared to the air-to-water heat pump under the 10, 30 and 50 W/m^2 space heating loads, respectively.
5) The water-based heating systems required considerably lower auxiliary power than the air heating system. The power use of the auxiliary components might seem negligible in terms of energy, but it is crucially important in terms of exergy, as they use electricity, which is 100% exergy. The relative importance of the auxiliary power use becomes more critical with decreasing space heating loads.

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Highlights

- Radiant floor heating was the optimal heating system
- Separation of space thermal conditioning and ventilation was an exergy-efficient solution
- Proved thermodynamically that renewables and radiant systems is an exergy-efficient match with exergy efficiency up to 10.5%
- Developed and validated the critical COP concept, which was 2.57 in this study
- Power use of auxiliary components is critical for the system exergy performance

Declaration of interests

☒ 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.

☐The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: