Low Temperature Heating and High Temperature Cooling in Buildings

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Cover photo: Courtesy of Solar Decathlon Europe

This PhD thesis is based on the following publications. Parts of these publications have been used directly or indirectly in the thesis.

Paper 1: Kazanci, O. B., & Olesen, B. W. (2015). IEA EBC Annex 59 - Possibilities, limitations and capacities of indoor terminal units. Energy Procedia - 6th International Building Physics Conference, IBPC 2015, 78, 2427–2432.

Paper 2: Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Exergy performance of different space heating systems: A theoretical study. Building and Environment, 99, 119-129.

Paper 3: Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy. Building and Environment, 100, 102-113.

Paper 4: Kazanci, O. B., Liu, X., Jiang, Y., & Olesen, B. W. (2016). Comparison of air-based and water-based space heating and cooling systems using entransy. **Submitted to Building and Environment on 2nd of April 2016.**

Paper 5: Schøtt, J., Andersen, M. E., Kazanci, O. B., & Olesen, B. W. (2016). Simulation study of the energy performance of different space heating methods in plus-energy housing. In Proceedings of the 12th REHVA World Congress, CLIMA 2016. Aalborg. **Accepted.**

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Paper 8: Andersen, M. E., Schøtt, J., Kazanci, O. B., & Olesen, B. W. (2014). Analysis of a plus-energy house for improved building and HVAC system design. In Proceedings of ROOMVENT 2014, 13th SCANVAC International Conference on Air Distribution in Rooms (377-384). Sao Paulo: Institute for Technology Research - IPT and Jovem Benedicto de Moraes.

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Publications that are part of the PhD study but not included in the thesis are the following.

Kazanci, O. B., Skrupskelis, M., Olesen, B. W., & Pavlov, G. K. (2013). Solar sustainable heating, cooling and ventilation of a net zero energy house. In Proceedings of CLIMA 2013 – 11th REHVA World Congress and the 8th International Conference on Indoor Air Quality,

Ventilation and Energy Conservation in Buildings (5513-5522). Prague: Society of Environmental Engineering (STP).

Kazanci, O. B., & Olesen, B. W. (2013). The effects of set-points and dead-bands of the HVAC system on the energy consumption and occupant thermal comfort. In Climamed'13 - 7. Mediterranean Congress of Climatization Proceedings Book (224-232). Istanbul: Turkish Society of HVAC & Sanitary Engineers.

Kazanci, O. B., & Olesen, B. W. (2014). Air and operative temperature measurements in a plus-energy house under different heating strategies. In Proceedings of ROOMVENT 2014, 13th SCANVAC International Conference on Air Distribution in Rooms (312-319). Sao Paulo: Institute for Technology Research - IPT and Jovem Benedicto de Moraes.

Kazanci, O. B., & Olesen, B. W. (2014). Sustainable Plus-energy Houses: Final Report. Kgs. Lyngby: Technical University of Denmark.

Kazanci, O. B., & Olesen, B. W. (2015). Horizontal temperature distribution in a plus-energy house: Cooling season measurements. In Proceedings of the 2015 ASHRAE Annual Conference. American Society of Heating, Refrigerating and Air-Conditioning Engineers.

Kazanci, O. B., & Olesen, B. W. (2015). Thermal indoor environment and energy consumption in a plus-energy house: Cooling season measurements. Energy Procedia - 6th International Building Physics Conference, IBPC 2015, 78, 2965–2970.

Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Energy and exergy performances of air-based vs. water-based heating and cooling systems: A case study of a single-family house. In Proceedings of the 2016 ASHRAE Annual Conference. St. Louis, MO: American Society of Heating, Refrigerating and Air-Conditioning Engineers. **Accepted.**

Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Effects of floor covering resistance of a radiant floor on system energy and exergy performances. In Proceedings of the 12th REHVA World Congress, CLIMA 2016. Aalborg. **Accepted.**

Kazanci, O. B., Shukuya, M., & Olesen, B. W. (2016). Energy and exergy analyses of the benefits and limitations of air-to-air heat recovery. In Proceedings of the 9th International Conference on Indoor Air Quality Ventilation & Energy Conservation in Buildings, IAQVEC 2016. Seoul. **Submitted.**

Full list of publications, which includes the co-authored publications, can be found in DTU's webpage.

Preface

The work presented in this thesis has been carried out at the International Centre for Indoor Environment and Energy (ICIEE), Department of Civil Engineering (BYG), Technical University of Denmark (DTU) within the period 15th of January 2013 – 15th of April 2016. Some parts of this thesis have been carried out in parallel to an international research project from International Energy Agency (IEA), Energy in Buildings and Communities (EBC) Programme, Annex 59 – High Temperature Cooling and Low Temperature Heating in Buildings.

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I saved the best for last, I am forever grateful to my parents and to Linette, for all their support, understanding, stimulation and giving me the space to pursue what I wanted. I have learned over the years that the distance does not matter when the bond is so strong.

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Ongun Berk Kazanci

Abstract

A heating and cooling system could be divided into three parts: terminal units (emission system), distribution system, and heating and cooling plant (generation system). The choice of terminal unit directly affects the energy performance, and the indoor environment in that space. Therefore, a holistic system evaluation is necessary to ensure an optimal indoor environment for the occupants and to achieve energy efficiency simultaneously. Low temperature heating and high temperature cooling systems are one of the possible approaches to heat or cool indoor spaces in buildings.

In this thesis, a single-family house designed for plus-energy targets and equipped with a radiant water-based floor heating and cooling system was studied by means of full-scale measurements, dynamic building simulations and thermodynamic evaluation tools. Thermal indoor environment and energy performance of the house were monitored for one year while different control strategies were tested. Theoretical analyses consisted of comparing the performance of different heating and cooling systems using energy, exergy, and entransy methods under steady-state conditions. Dynamic simulations were used to study the energy performance of heating and cooling systems for achieving the same thermal indoor environment.

The results show that it is crucial to minimize the heating and cooling demands in the design phase since these demands determine the terminal units and heat sources and sinks that could be used. Low temperature heating and high temperature cooling systems (a radiant water-based floor heating and cooling system in this study) proved to be superior to compared systems, evaluated with different system analysis tools; energy, exergy, and entransy.

Radiant systems should be coupled to appropriate heating and cooling sources, and energy requirements of auxiliary components (pumps, fans, etc.) should be minimized. Radiant systems could be coupled to renewable heat sources and sinks (e.g. ground), which would result in considerable energy savings. Water-based heating and cooling systems require considerably less auxiliary energy compared to air-based systems. Exergy analysis can be used to optimize a system holistically where different quality energy forms, such as electricity and heat, are used.

Control of the radiant system and its interaction with the ventilation system are critical for an optimized operation. Measurements, simulations, and calculations proved that a system in which the radiant system heats or cools the space and the ventilation system only provides the required amount of fresh air for indoor air quality concerns is the optimal solution. Application of radiant floor heating is particularly beneficial in high-ceilinged spaces, as it can provide a uniform temperature distribution and decrease heat losses due to thermal stratification.

To obtain the most rational use of available resources, energy analysis alone is not sufficient. It is not enough to consider only the quantity of energy; the temperatures and temperature differences within a system should also be considered.

Although a single-family house was used for evaluations in this thesis, the results and developed calculation methodologies can be applied to a wider range of buildings using similar heating and cooling systems.

Resumé

Et opvarmnings- og kølesystem kan inddeles i tre kategorier: varme- og køleterminaler, fordelingssystem samt varme- og køleanlæg. Valget af varme- og køleterminaler vil påvirke den energimæssige ydeevne samt indeklimaet i den pågældende zone. På baggrund af dette er det nødvendigt med en holistisk system evaluering, for at sikre et optimalt indeklima for brugerne samtidig med at det er energieffektivt. Lav temperatur opvarmning og høj temperatur kølesystemer er en mulig tilgang hvorved bygninger kan opvarmes eller køles.

I nærværende afhandling blev et enfamiliehus, designet som et plus energi hus, udstyret med vandbaseret strålings-system via gulvvarme og køling, undersøgt ved hjælp af fuldskala målinger, dynamisk bygningssimulering samt termodynamiske evalueringsmetoder. Husets termiske indeklima og energimæssige ydeevne blev målt over et år mens forskellige kontrolstrategier blev testet. Teoretiske analyser blev udført ved at sammenligne de forskellige opvarmnings- og kølingssystemers ydeevne, ved at bruge energi, exergy og entransy metoder. Dynamiske simuleringer blev brugt til at studere den energimæssige ydeevne af opvarmnings- og kølesystemer for at opnå det samme termiske indeklima.

Det er vigtigt at minimere opvarmnings- og kølebehovet i designfasen idet disse behov stiller krav til hvilke varme- og køleterminaler og varme- og køleanlæg der kan bruges. Nærværende undersøgelser viser, at lav temperatur opvarmning og høj temperatur kølesystemer (i dette tilfælde vandbaseret gulvarme og køling strålingssystem) er overlegene i forhold til de sammenlignet systemer undersøgt med forskellige analyseværktøjer: energi, exergy og entransy.

Strålingssystemer bør bruges med passende opvarmnings- og kølekilder og energikravet til auxiliary komponenter (pumper, ventilatorer osv.) bør begrænses. Strålingssystemer kan bruges sammen med vedvarende energikilder som eksempelvis jorden, hvilket vil give betydelige energibesparelser. Vandbaseret opvarmnings- og kølesystemer kræver betydelig mindre energi til auxiliary komponenter sammenlignet med luftbaseret systemer. Exergy analyser kan bruges til at udføre en holistisk optimering af systemer hvor der anvendes forskellige kvalitets energiformer såsom elektricitet og varme.

Styring af strålingssystemer og dets interaktion med ventilationssystemer er kritiske for en optimeret drift. Målinger, simuleringer og beregninger udført har vist, at den mest optimale løsning er et system, hvor strålingssystemet opvarmer eller køler rummet uden affugtning samtidig med ventilationssystemet kun tilfører den nødvendige mængde friskluft der skal til at opretholde et tilladeligt indeklima. Anvendelse af gulvvarme som strålesystem er særlig fordelagtigt i højloftede rum, da systemet kan give en ensartet temperatur fordeling imens varmetabet mindskes grundet den termiske lagdeling.

Energi analyser alene er ikke tilstrækkelige til, at opnå den mest rationelle udnyttelse af tilgængelige ressourcer. Det er ikke nok blot at overveje kvantiteten af energi, men temperaturer og temperaturforskelle i et system bør også overvejes.

Til trods for, at et enfamiliehus blev anvendt i nærværende afhandling, kan resultaterne og de udviklet beregningsmetoder anvendes på en bredere vifte af bygninger.

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List of abbreviations

ACH	Air change per hour
AHU	Air-handling unit
CAV	Constant air volume
CFD	Computational Fluid Dynamics
COP	Coefficient of performance
DOAS	Dedicated outdoor air system
EBC	Energy in Buildings and Communities Programme
EPBD	Energy Performance of Buildings Directive
HTC	High temperature cooling
HVAC	Heating, ventilation and air-conditioning
IAQ	Indoor air quality
IEA	International Energy Agency
LTH	Low temperature heating
nZEB	Nearly zero-energy building
PAQ	Perceived air quality
PCM	Phase change material
PV	Photovoltaic
PV/T	Photovoltaic/thermal
SBS	Sick building syndrome
TABS	Thermally active building system
THIC	Temperature and humidity independent control
VAV	Variable air volume

1. Introduction

When striving for energy efficiency in buildings, it should not be overlooked that people spend most of their time indoors [1]. The main task of buildings is to provide a comfortable and safe environment for occupants, which stimulates a healthy and productive living. Therefore, buildings are built for people, the occupants, and not to save energy. Thus, energy savings and energy efficiency in buildings should not be achieved at the cost of occupants' indoor environmental discomfort. Instead, these goals should be achieved simultaneously.

This creates a challenge, and it means that buildings should provide the desired indoor environments with the minimum use of energy, and not just by using any energy resource, but by using energy resources with low grade (low exergy). The quantity and the quality of the energy used by buildings have remarkable effects on the global energy infrastructure and on the global level of greenhouse gas emissions.

Heating, cooling and ventilation systems are crucial components in buildings, because they allow controlling the thermal indoor conditions and air quality to meet occupant needs. However, these systems are also responsible for a large part of the energy used in buildings, therefore improvements in heating, cooling and ventilation systems in buildings can have significant effects on individual buildings, but also on the global scale.

Several approaches exist to heat or cool buildings. Low temperature heating (LTH) and high temperature cooling (HTC) systems are one of these approaches. This thesis focuses on one of the examples of LTH and HTC systems: radiant water-based heating and cooling systems.

1.1. Hypothesis and objectives

The main hypothesis of this thesis is that low temperature heating and high temperature cooling systems (radiant systems in this thesis) are the optimal choice for heating and cooling indoor spaces. This hypothesis was proven using measurements, dynamic simulations, and thermodynamic evaluation tools.

The first objective of this thesis was to evaluate the thermal indoor environment and energy performance of a plus-energy house equipped with a radiant floor heating and cooling system by means of full-scale measurements, and to provide improvement suggestions based on the measurement results and based on parametric analysis using a dynamic simulation model.

The second objective was to use this house as a case study for the theoretical evaluations (using energy, exergy, and entransy) and for the dynamic simulations to compare the performances of different space heating and cooling systems for achieving the same thermal indoor environment.

1.2. Outline of the thesis

This thesis consists of theoretical, numerical and experimental parts. A plus-energy house designed and constructed in 2012 for an internal student competition, Solar Decathlon Europe, was used as a case study for the evaluations.

A certain part of the work in this thesis was carried out in parallel to an international research project from International Energy Agency (IEA), Energy in Buildings and Communities Programme (EBC), Annex 59 – High Temperature Cooling and Low Temperature Heating in Buildings. The theoretical and experimental studies in this thesis provided inputs to Annex 59.

The work in this thesis can be divided as follows:

- An overview of terminal units was prepared (**Paper 1**);
- The house was used as a case study for the theoretical and numerical analyses. Theoretical analyses consisted of comparing the performance of different heating and cooling systems using energy, exergy, and entransy methods under steady-state conditions. Dynamic simulations were used to study the energy performance of the heating and cooling systems (**Paper 2 to Paper 6**);
- Thermal indoor environment and energy performance (energy production versus use) of the house were monitored for one year while different control strategies were tested. Measurements and dynamic simulations were used to provide improvement suggestions regarding the thermal indoor environment and energy performance of the house (**Paper 7 to Paper 9**).

This thesis is structured as follows. After this introductory chapter, Chapter 2 summarizes the operational characteristics of the chosen terminal units and provides a literature review of radiant heating and cooling systems. Chapter 3 describes the single-family house that was used for the analyses. Chapter 4 presents the results of the theoretical and numerical analyses of different space heating and cooling systems. Chapter 5 presents the results of the measurements and parametric analysis. Chapter 6 gives the overall conclusions of the thesis. Chapter 7 identifies topics to be investigated further. Appendix A provides a table, which summarizes the capabilities, limitations, and operational characteristics of the chosen terminal units. Appendix B consists of the published, accepted, and submitted papers.

2. Background

In this chapter, the most commonly used and available terminal units are identified. Their operational characteristics are summarized, together with certain aspects that are related to all terminal units. A table that summarizes the capabilities, limitations and operational characteristics of the chosen terminal units can be found in Appendix A – Terminal unit table. The first part of this chapter is based on Paper 1.

The second part of this chapter consists of literature review on radiant surface heating and cooling systems.

2.1. Indoor terminal units and their characteristics

Several approaches can be used to heat or cool indoor spaces in buildings. Heating and cooling systems in buildings can be considered to consist of three main parts: heating and cooling plant, the system carrying the heat transfer medium-distribution system, and heat (and/or moisture) emission and removal system-terminal unit.

Indoor terminal units are active building elements that use different heat transfer mechanisms and media to emit or remove heat (and/or moisture) to or from indoor spaces (e.g. hydronic radiant heating and cooling systems, fan-coil units, active beams, etc.).

Terminal units mainly rely on convection (natural or forced), radiation or a combination of both. In HVAC systems used throughout the world (e.g. Europe, North America, Asia, etc.), energy sources and energy generators are similar and the main global differences between HVAC systems are often the indoor terminal units [2].

Terminal units are the closest components of heating and cooling systems to the indoors and to the occupants. Indoor temperature and humidity fields depend on the chosen terminal units, and therefore the choice of terminal units has a direct impact on overall and local thermal comfort. This choice also affects the whole heating and cooling system because it determines the temperature levels of the heating-cooling medium to be used, dimensioning of the auxiliary components (pumps, fans, etc.), heating and cooling plants, and even the heat sources and sinks that could be used, which in turn determines the energy performance of the whole system.

Terminal units differ from each other according to certain criteria:

- Possibilities (heating, cooling, ventilation–fresh air, humidification, and dehumidification);
- Methods of heat emission or removal (convection, radiation, or a combination of both);
- Maximum heating and cooling capacities;
- Medium of energy distribution (air, water, or electricity);
- Thermal mass and storage capacity;
- Total or local volume conditioning.

These criteria are discussed further for the chosen terminal units, which are:

- Hydronic radiant heating and cooling systems;
- All-air systems (mixing, displacement, and personalized ventilation);
- Beams (passive and active).

These terminal units were chosen based on the low temperature heating and high temperature cooling possibility, and on the medium of energy distribution.

2.1.1. Hydronic radiant heating and cooling systems

A hydronic (water-based) radiant heating and cooling system refers to a system where water is the heat carrier (medium of energy distribution) and more than half of the heat exchange with the conditioned space is by radiation [3].

Radiant heating and cooling systems can be divided into three [3]:

- Radiant heating and cooling panels;
- Pipes isolated from the main building structure (radiant surface systems);
- Pipes embedded in the main building structure (Thermally Active Building Systems, TABS).

They are low temperature heating and high temperature cooling systems. Therefore, the heat carrier (water) circulating in the pipes has low temperatures in heating and high temperatures in cooling operation. In some TABS constructions (hollow core concrete decks), also air has been used as a heat carrier, and electricity can also be used in some heating applications.

Floor, wall and ceilings can be used as surfaces that provide heating or cooling to the space. Hydronic radiant surface systems are capable of addressing only sensible heating and cooling loads. Therefore, they require a ventilation system to address the latent loads (to regulate humidity) and to provide the ventilation rates required for indoor air quality concerns [3]. Radiant heating and cooling systems enable lower airflow rates than all-air systems, in which the entire heating and cooling loads are addressed by the ventilation system [4].

Heat emission to or removal from the space is by a combination of radiation and convection. Total heat exchange coefficients (combined convection and radiation) for floor heating, wall heating and ceiling heating are 11, 8, 6 W/m^2K, and for floor cooling, wall cooling and ceiling cooling are 7, 8, and 11 W/m^2K, respectively [3]. The radiant heat transfer coefficient can be used as a constant value of 5.5 W/m^2K, with an error of less than 4% [5]. The difference in total heat transfer coefficients is due to the natural convection. An overview of natural convection coefficients is given in [6].

Based on the acceptable surface temperatures (comfort and dew point concerns [3]), and assuming a room operative temperature of 20°C for heating and 26°C for cooling cases, the maximum heating and cooling capacities can be estimated. The maximum floor (occupied zone) heating and cooling capacities are 99 W/m^2 and 42 W/m^2, wall heating and cooling capacities are 160 W/m^2 and 72 W/m^2, and ceiling heating and cooling capacities are 42 W/m^2 and 99 W/m^2, respectively. In the perimeter zones of the floor, it is possible to obtain a maximum heating capacity of 165 W/m^2 [3].

Different construction types of radiant systems can be found in [3]. The design, test methods, control and operation principles of radiant panels are given in ISO 18566:2013 [7], while the design, dimensioning, installation and control principles of embedded radiant systems are given in ISO 11855:2012 [8].

Fig. 1 and Fig. 2 show examples of different radiant system applications.

Fig. 1. Examples of a cooling panel [3].

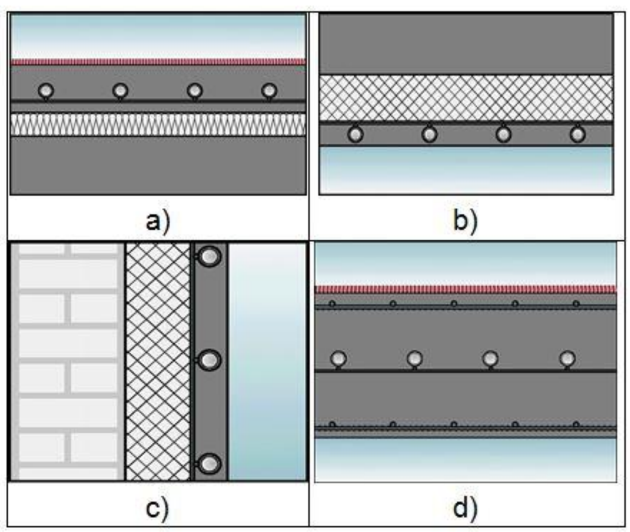


Fig. 2. Examples of embedded radiant systems, a) Floor, b) Ceiling, c) Wall, d) TABS [9].

2.1.2. All-air systems

There are eight commonly applied ventilation (air distribution) strategies in indoor spaces. These strategies are mixing ventilation, displacement ventilation, personalized ventilation, hybrid air distribution, stratum ventilation, protected occupied zone ventilation, local exhaust ventilation, and piston ventilation [10]. For ventilation (air-conditioning) systems, the main method of heat emission and removal is convection and the medium of energy distribution is air. Mixing, displacement, and personalized ventilation systems are further described as follows.

2.1.2.1. Mixing ventilation

Mixing ventilation (mixing room air distribution) intends to dilute the polluted and warm (or cool) room air with clean, cooler (or warmer) supply air. The aim is to achieve a uniform temperature and contaminant distribution in the occupied zone [11]. It is possible to heat or cool a space by mixing ventilation. It is also possible to provide dehumidified and conditioned outdoor air (fresh air). Typical supply air temperature ranges between 34°C and 14°C [10]. The obtained heating and cooling effect will depend on the ventilation rate. In some countries, there are regulations concerning the highest permissible supply air temperature, e.g. in Denmark, the highest permissible supply air temperature is limited to 35°C [12]. It is not recommended to have a higher temperature difference than 10°C between the supply and room air to achieve proper mixing [11]. According to [13], a specific cooling load of 90 W/m^2 can be handled with mixing ventilation systems.

2.1.2.2. Displacement ventilation

Displacement ventilation (displacement room air distribution) is based on displacing the polluted room air with fresh air (conditioned outdoor air) [10]. The cool fresh air is supplied with low velocity (0.25-0.35 m/s [14]) at or near the floor, and the supplied air rises by the effects of momentum and buoyancy forces [10,14]. It is possible to provide cold, dehumidified and conditioned outdoor air with displacement ventilation. Although it could be possible to provide warmer air than the room air with displacement ventilation (e.g. to heat an unoccupied room before the occupancy [15]), it is not common and it is not recommended due to possible short-circuiting of the supply air. Typically, the supply air temperature can be as low as 18°C [10]. The cooling load that a floor current displacement system can handle is 30-35 W/m^2 according to [14] and 50 W/m^2 according to [13].

2.1.2.3. Personalized ventilation

Apart from the two mainly applied total volume air distribution principles (mixing and displacement air distribution), another air distribution strategy is personalized ventilation, and it aims at supplying the clean and cool air close to an occupant before it is mixed with the room air [11,16]. The most important advantage of personalized ventilation compared to the total volume conditioning systems is its potential to provide clean, cool and dry air at inhalation [16,17]. According to [10], the supply air temperature can be as low as 20°C in cooling and as high as 28°C in heating mode. However, it should be noted that perceived air quality (PAQ) might be a problem with the increased supply air temperature [18,19] and ventilation effectiveness may decrease depending on the chosen air supply location and terminal.

The required ventilation rates can be calculated based on EN 15251:2007 [20], (this standard is currently under revision [21]), CR 1752:1998 [22], and ASHRAE 62.1-2013 [23].

Fig. 3 shows examples and principles of different ventilation approaches.

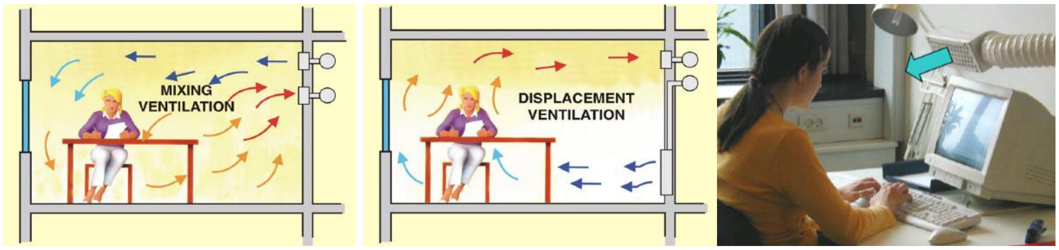


Fig. 3. Examples and principles of different ventilation strategies: mixing ventilation (left), displacement ventilation (middle), personalized ventilation (right) [24].

2.1.3. Beams

Although these systems are known as chilled beams, a recent publication refers to them as beams, and therefore this terminology will be used in the following [25]. Beams (passive and active) are room air recirculation devices that can heat or cool (sensible) a space using water as the energy distribution medium. Beams also operate with low temperature heating and high temperature cooling principle.

The operation principles of beams are similar to fan-coil units. They are similar to a fan-coil unit installed in the ceiling, although beams benefit, and they are specially developed for, from the low temperature heating and high temperature cooling principle. In beam systems, high chilled water temperatures of 14-18°C (typically around 14.5°C), and low heating water temperatures of 32-45°C are typically used [25].

Active beams can provide conditioned primary air to a space since they are coupled to the main air-handling unit [25]. Fresh air is delivered to the space by a decoupled ventilation system in passive beam applications. Beams cannot humidify or dehumidify the room air directly because they operate in dry (non-condensing) conditions but it is possible to control the latent loads and to address the ventilation requirements with active beams [25]. Heat emission to and removal from the space mainly takes place by convection.

2.1.3.1. Passive beams

The performance of passive beams relies on natural convection [25]. In passive beams, the medium of energy distribution from the plant is water. It is possible to heat and cool a space with passive beams but it is not possible to provide fresh air to the space. Although heating is possible with passive beams, in most applications, passive beams are used for cooling only and therefore a separate heating system should be used [25]. In addition, ventilation needs should be addressed by a complementing system (e.g. by an air-handling unit) [25]. It is recommended to use passive beams when the total sensible cooling load is in the range of 40-80 W/m^2 [26].

2.1.3.2. Active beams

The performance of active beams relies on convection that is caused by induction [25]. It is possible to heat, cool and provide fresh air to a space by active beams. In active beams, the medium of energy distribution is both air (fresh air from the air-handling unit) and water from the heating or cooling plant. Active beams can typically be used when the total sensible cooling (air and water) load is less than 120 W/m^2 in comfort conditions [25,26]. The optimum operating range (for achieving thermal comfort in sedentary type occupancy) is 60-80 W/m^2 [26]. For the heating case, the optimum operating range is a heating load of 25-35 W/m^2 and a maximum heating load of 50 W/m^2 [26].

The testing and rating procedures of passive and active beams are provided in EN 14518:2005 [27] and in EN 15116:2008 [28], respectively.

Fig. 4 shows the airflow and operation schemes of passive and active beams.

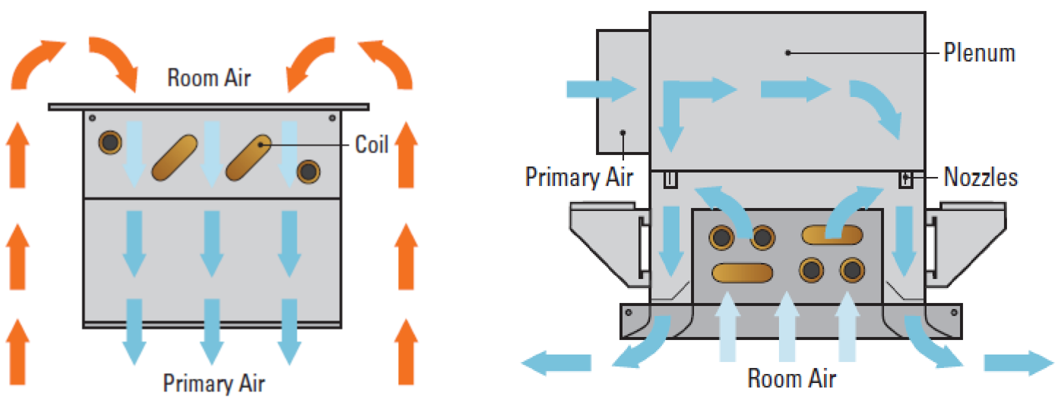


Fig. 4. Airflow schemes of a passive (left) and an active beam (right) [25].

2.1.4. Fan-coil units

Fan-coil units are another type of terminal units that are commonly used worldwide; however, they were not considered in details since they are not necessarily low temperature heating and high temperature cooling systems. They mostly recirculate the room air but in some cases, they can also take in outdoor air. Further information regarding fan-coil units can be found in [13,29].

2.1.5. Other systems

The descriptions, characteristics, operation principles and other information regarding other terminal units that were not considered in this thesis (radiators, radiant tubes, convectors, etc.) can be found in [13,29].

Le Dréau has also provided a classification scheme for air-based and radiant heating and cooling systems [30].

2.1.6. Limiting factors

When considering the operation of different heating and cooling systems and selection of terminal units, there are certain limiting factors depending on occupant thermal comfort (overall and local), dew point, and occupant safety and health.

The thermal comfort of occupants in a space will be directly influenced by the choice of terminal unit in that space. Chosen terminal unit will influence air speed, air temperature, mean radiant temperature, and for some of them also humidity.

Different thermal indoor environment categories are defined based on the operative temperature in international standards [20,31]. Although the overall thermal comfort may be satisfactory, local thermal discomfort can cause dissatisfaction with the thermal environment.

In addition to providing the required overall thermal comfort, the following phenomena should also be considered:

- Draught;
- Vertical air temperature difference;
- Warm or cool floors;
- Radiant asymmetry.

Temperature drifts in buildings should also be controlled and it should not exceed the limits given in respective standards [32]. The same applies for noise; noise from terminal units should be limited and the sound pressure levels should not be exceeded for the given type of space [20].

International standards (EN 15251:2007 [20], EN ISO 7730:2005 [31], ASHRAE 55-2010 [32]) should be followed to provide the optimal indoor environment for the occupants, considering indoor air quality, thermal indoor environment, acoustics and lighting.

When considering the surface temperatures for radiant surface heating and cooling systems, in addition to the surface temperature and radiant asymmetry limitations given in international standards, another limiting factor is the dew point. The formation of dew on cooled surfaces should be prevented, and it is recommended to maintain the supply temperature to the radiant loops above the dew point. It should be noted that terminal units could also be used to compensate for radiant asymmetry from cold surfaces (windows) and draught from cold surfaces.

Occupant safety and health also puts certain limitations on permissible surface temperatures. The wall surface temperatures with wall heating systems should be designed so that the risk of pain by touching with bare skin is minimized. This happens when the surface temperature exceeds approximately 40°C [3]. The perception of high surface temperature will depend on the thermal properties (thermal conductivity and specific heat capacity) of the surface material. Dew point and draught concerns should be considered when designing a wall cooling system, although recent studies show that draught is not a factor that limits the cooling capacity of a radiant wall [30].

The thermal mass of the terminal unit will have an influence on the energy use and on the dimensioning of the heating and cooling plants. In addition, it will also determine how quickly the conditions in a space can be changed. It is not enough to consider the thermal mass of the terminal unit alone and it should be considered together with the thermal mass of the building itself, as thermal mass of the building will affect the thermal indoor environment and will interact with the terminal unit.

On the supply-side, the choice of heat source and sink for the heating and cooling system will depend on the terminal unit, geographical conditions, and on the regulations (location-specific opportunities such as district heating and cooling, utilization of sea-water for cooling, etc.). A holistic approach that considers the supply, distribution and terminal units is required for a proper system and component selection.

Dynamic building simulation software, Computational Fluid Dynamics (CFD), and experimental methods could be used to evaluate the performance of different terminal units, in terms of energy performance, resulting temperature and humidity fields, and occupant thermal comfort.

2.1.7. Summary

In addition to the summarized characteristics, there are other factors to consider when selecting a heating and cooling system and its terminal unit:

- Once the heating and cooling demands of the building are determined, the most energy efficient and environmentally friendly heating and cooling system should be chosen to meet these demands;
- Related international standards regarding system design, operation, implementation and evaluation should be followed;
- Initial (capital) and operational costs should be considered;
- The chosen type of heating and cooling strategy (terminal unit) will have a direct effect on the occupant thermal comfort. Criteria such as noise, draft, vertical air temperature difference, etc. are limiting factors for the choice and application of different terminal units. International standards (EN 15251:2007 [20], EN ISO 7730:2005 [31], ASHRAE 55-2010 [32]) should be followed to provide the optimal thermal comfort for the occupants;
- It is crucial to consider the auxiliary energy use (pumps, fans, valves, dampers, sensors, etc.) associated with each terminal unit;
- Availability (depending on the location, natural resources, district heating and cooling network, etc.) of the energy sources and sinks, and the possibility of coupling with the terminal unit should be considered;
- Control possibilities, principles (e.g. individual room or zone control, control based on flow rate, supply temperature, average temperature, as a function of air temperature, operative temperature or outdoor temperature) and dynamic behaviors of the terminal units should be considered, e.g. ventilation systems try to keep a constant room temperature while TABS allows a certain temperature drift and keeps the operative temperature within the comfort range rather than a constant value [4]. Another example of the dynamic behavior is the difference between a TABS and a radiant panel, where the radiant panel will be able to affect the thermal conditions in

the room faster than a TABS construction, due to its significantly lower thermal mass.

When deciding on which terminal unit to use in a space, all of these factors and the provided possibilities, limitations, and capacities should be considered.

2.2. Radiant surface heating and cooling systems

In addition to the basic operation principles of radiant systems described in 2.1.1, a more detailed literature review is presented in the following.

2.2.1. Introduction

The principle of contemporary radiant heating and its examples have several thousand years of history, especially in Asia [33,34]. In South Korea, hot water floor heating system is the main heating method for residential buildings, and even in high-rise buildings. A recent study suggests that about 95% of all buildings in South Korea and about 85% of all buildings in Northern China use radiant floor heating [33,34].

There have been applications of radiant heating and cooling throughout the 20th century, and its application started to increase starting from early 1990s especially in central and northern Europe [35]. Regarding hydronic radiant cooling, although there have been some applications in late 1930s and late 1950s in Europe; afterwards the interest in them has faded until early 1990s. The interest in using hydronic radiant cooling systems started to grow due to several disadvantages associated with all-air (air-conditioning) systems such as high energy use, draught, noise, and sick building syndrome (SBS) [36,37].

Meierhans [35,38] reported a TABS installation in an office building in Switzerland, and identified several benefits of using of radiant systems. These were higher transport efficiency of water-based systems compared to air-based systems, phase shifting possibility between the peak load and its discharge to the environment, free cooling using cold night air-water heat exchanger (eliminating compressor cooling), and reducing the required airflow rates of supply air. The self-regulating effect of TABS was also identified. The author suggested that after the completion of an installation, a monitoring period of one to two years is necessary. This would enable to optimize and fine-tune the operation of the heating and cooling system of the building.

Simmonds [39] emphasized the effects of mean radiant temperature on occupant thermal comfort, and pointed out that it is inefficient to address radiant loads with convective methods of air-conditioning systems. Simmonds also showed that radiant system could be successfully used to condition spaces such as a multifunctional space, a retail store, and an office space. When compared to a variable air volume (VAV) system, the hydronic system uses less space, requires less ductwork and works with less noise.

Feustel and Stetiu [37] reviewed several characteristics and potentials of using hydronic radiant cooling systems. Their findings revealed several benefits for hydronic radiant cooling systems including improved comfort (reduced air movement, reduced convection), high temperature cooling, reduced peak power requirements, reduced transport energy, improved indoor air quality due to elimination of recirculation air, improved space use and reduced building costs due to reduced size of distribution system, and possibility of using alternative cooling sources. The authors also pointed out that an accurate sizing and an adequate humidity control are required.

2.2.2. Energy performance

Several authors compared performances of radiant systems to air-based systems. Olesen and Mattarolo [40] compared the energy and thermal comfort performance of TABS, radiant panel and radiant floor system to a conventional all-air system using EnergyPlus simulation software, under climate conditions of Copenhagen, Denmark. The authors showed that the radiant systems performed better in terms of providing a comfortable thermal indoor environment, with reduced energy use and reduced peak loads. The authors also showed that it is possible to reduce the peak power demand in cooling operation by 26% with TABS, 20% with radiant panel, and by 6% with the radiant floor system compared to the conventional all-air system.

Fabrizio et al. [41] compared the energy and thermal comfort performances of all-air, fan-coil, radiant floor, and radiant ceiling systems in several European cities using EnergyPlus. Their results show that for achieving similar thermal comfort conditions, a radiant system coupled with an appropriate heating source or sink will reduce energy use and CO_2 emissions compared to all-air systems. The authors also point out that the greatest reductions will be achieved in climates with a higher cooling demand than heating demand.

After a simulation study using IDA ICE, Kolarik et al. [42] showed that compared to a VAV system, TABS was able to reduce the primary energy use by 16% in a moderate climate (San Francisco, CA), and by 50% in hot-humid (Miami, FL) and hot-dry (Phoenix, AZ) climates.

Jeong et al. [43] compared energy performances of a radiant panel system combined with a dedicated outdoor air system (DOAS) to a VAV system with air-side economizer control. They showed that radiant/DOAS system required 37% lower supply airflow rate compared to VAV system, resulting in 29% less energy use for fans. The chiller energy use of the radiant/DOAS system was 25% less than the VAV system, and the chiller size was reduced by 29%. Although the pumping power was higher in the radiant/DOAS system compared to the VAV system, this was compensated by the fan energy use reduction. The authors showed that, in total, the radiant/DOAS could reduce system energy use by 42% compared to the VAV system.

Imanari et al. [44] compared thermal comfort, energy and cost performances of a radiant ceiling panel system with a conventional air-conditioning system. Radiant system reduced the energy used for transporting air by 20%, which resulted in 10% less total energy use compared to the air-conditioning system. This also enabled energy cost reductions of 9 to 10%.

Niu et al. [45] compared the energy performance of a radiant ceiling cooling system coupled to a desiccant cooling system. They showed that it is possible to apply radiant cooled ceiling and save up to 44% of the total energy use compared to a conventional constant air volume (CAV) system, in a hot-humid climate (Hong Kong). The authors also showed that the combined system required only 20% of the airflow rate required for the conventional system (i.e. 78% fan energy saving).

Stetiu [46] studied the potential of energy and peak power savings by using radiant cooling systems compared to all-air systems in commercial buildings in several US climates. The author showed that a properly designed and controlled radiant cooling system can be used at any location in US, and the average energy and peak power savings potentials were 30% and 27%, respectively. The potential of energy savings varied between 17 to 42%, and the potential of peak power savings varied between 22 to 37%, as a function of the climate.

2.2.3. Indoor environmental quality

Field studies, laboratory studies, and studies using CFD have been carried out to study the indoor environments created by radiant systems, and to compare the indoor environments created by all-air systems to those created by radiant systems. Some of the indoor environmental problems associated with all-air systems (air-conditioning) are draught, too high vertical air temperature differences, noise, and SBS symptoms [36,37,44,47]. The use of radiant systems can help reduce, or even eliminate, these complaints.

Corgnati et al. [48] showed that in an office room, draught risk caused by the direct drop of the cold air jet into the occupied zone could be reduced by using radiant ceiling cooling panels. Another study by Niu and Kooi [49] showed that the vertical air temperature gradients could be reduced by using cooled ceiling systems while still maintaining a satisfactory ventilation effectiveness.

In a field study in an office in Tokyo, Japan, Imanari et al. [44] showed that radiant ceiling cooling system improved the overall thermal comfort of the occupants, and also reduced draught and vertical air temperature differences compared to a conventional air-conditioning system. Similar results were obtained from a study in Canada [47], where a field study of occupant thermal comfort in a building equipped with radiant slab cooling showed a uniform thermal indoor environment with very low radiant temperature asymmetry, reduced draught rate and reduced vertical air temperature difference.

De Carli and Olesen [50] reported the results of field measurements of operative temperatures in four buildings, which used radiant heating and cooling systems, and showed that radiant systems are capable of providing acceptable thermal indoor environments even when the outdoor temperatures exceed 30°C. Martinez et al. [51] reported field measurements from a low-energy office building in Madrid, Spain, and showed that a properly designed building equipped with TABS can significantly lower the energy use and at the same time achieve high indoor environmental quality.

Boerstra et al. [52] compared low temperature heating systems to conventional systems in terms of thermal comfort, indoor air quality (IAQ) and safety. The authors reported that with low temperature heating systems, thermal comfort is improved through reduced vertical air temperature differences, reduced radiant temperature asymmetry, reduced temperature fluctuations, reduced draught risk, and more comfortable floor temperatures. Improvements in IAQ are possible due to reduced dust mites, lower air temperatures (improved PAQ), and less SBS symptoms. Safety benefits include lowered risk for hand burning and physical injuries. The authors point out some disadvantages with low temperature heating systems such as warming up time and radiant asymmetry near windows; however, those disadvantages were mostly encountered in poorly designed cases.

A recent study in Hyderabad, India compared the energy, thermal satisfaction and cost performance of an all-air system (an optimized VAV system) with a radiant system (radiant cooling with DOAS) [53]. The two systems were installed in identical buildings (floor area of 11613 m^2 each) and this enabled to test these cooling systems side-by-side. The results showed that after two years of operation, the radiant system used 34% less energy compared to the VAV system. A survey carried out among the employees showed that the radiant system provided a more satisfactory thermal indoor environment for the occupants; 45% and 63% of “satisfied or very satisfied” with VAV system compared to the radiant system, respectively.

2.2.4. Costs and space requirements

Initial and operational costs of heating and cooling systems are crucial to consider although it is not straight-forward to make an objective comparison of these costs globally, e.g. labor costs, and equipment costs for heating and cooling plants and terminal units could vary significantly in different countries.

Imanari et al. [44] pointed out that when radiant ceiling cooling system are used, smaller air-handling units can be used, which leads to reduced costs for AHUs, ducting and piping. They estimated that depending on the price of the radiant panel used, the payback time was between 1 to 17 years (based on standard prices of January 1994).

For a typical European office building, Kalz and Pfafferott [54] provided investment cost estimates for passive cooling (20 €/m^2), mechanical night-ventilation (32 €/m^2), fan-coil (85 €/m^2), radiant cooling ceiling panel (138 €/m^2), and for TABS (117 €/m^2).

Simmonds [55] showed that there is little or no difference between the installation costs of a VAV system and a radiant ceiling system for an office in US. The maintenance cost was about 7% lower for the radiant system. There are also utility (energy) cost savings associated with radiant systems.

Hoenmann and Nussle [56] compared costs and space requirements of a radiant cooling system (aluminum panel) and of a VAV system. For initial costs, they showed a break-even point for the radiant system at a cooling load of 50 W/m^2 and at a ventilation rate of 3 ACH. They also showed that the space requirements for shafts and equipment rooms were 36% and 28% less for the radiant system compared to the VAV system, respectively. The authors also pointed out that further space savings are possible by reducing the plenum height from 0.4 m for the VAV system to 0.2 m for the radiant system. Further savings can be achieved for radiant systems integrated to the ceiling [37].

Sodec [57] compared costs associated with a radiant ceiling cooling system and a VAV system. The results showed that above a cooling load of 45 to 55 W/m^2, the initial costs of a radiant ceiling cooling system could be below the VAV system, up to 20%. In terms of initial costs and energy costs, the radiant ceiling cooling system was more economical when the cooling load was higher. The ceiling cooling system required 40 to 55% less space compared to the VAV system. The author showed that with a proper layout and operation, the radiant ceiling cooling system is more economical than the VAV system.

The results of the side-by-side comparison study in Hyderabad, India [53] showed that the radiant system needed one air-handling unit compared to the six required by the VAV system, resulting in space and construction cost savings for the radiant system. Table 1 shows a detailed breakdown of the costs for the VAV and the radiant system. The cost of the radiant system was slightly lower than the VAV system.

Table 1. Cost comparison of the VAV and the radiant cooling system [a] [53].

	VAV	***Radiant***
Chiller	3,145,200	3,145,200
Cooling tower	1,306,400	1,306,400
HVAC low side works	22,839,000	15,310,000
AHUs, DOAS, HRW	5,118,200	2,878,900
Radiant piping, accessories, installation, etc.	0	9,075,800
Building automation system	6,184,000	6,584,000
Total cost (in Rupees and in Rupees/m^2)	38,592,800 / 3,327	38,300,300 / 3,302

[a] Values in Rupees.

2.2.5. Applications of radiant heating and cooling systems, and recent developments

Radiant heating and cooling systems have a wide range of applications such as in offices, residential buildings, workshops, laboratories, food storage cellars, meeting rooms [58], schools [59], museums [60], airports [61,62], sports halls, hangars and others [3]. Karmann et al. [63] provided an online database of buildings equipped with radiant systems.

Avoiding condensation is crucial for radiant systems and therefore its applications in humid climate zones require careful considerations. Different studies have shown that when properly designed, controlled and coupled with an appropriate ventilation system, radiant cooling systems can also be applied in hot-humid climate zones [45,46,53,61,64,65].

One special case and growing interest in the application of radiant floor cooling systems is in buildings with large glazed façades, such as airports, railway stations and so forth. The main reason behind this application is that radiant floor cooling systems remove the heat gains through direct solar radiation immediately (before it can heat up the space) and the cooling capacity of the floor cooling system increases considerably [66]. Different studies have shown that the cooling capacity of a floor cooling system exceeds the given maximum capacity of 42 W/m^2 and may even exceed 100 W/m^2, when there is direct solar radiation on the floor surface [3,67,68]. Another reason is that when floor cooling system is coupled with an appropriate ventilation strategy (e.g. displacement ventilation), this system combination allows conditioning only the occupied zone of the indoor space, in contrast to more traditional systems using air-conditioning.

In an application of radiant floor cooling in the New Bangkok International Airport, the energy demand was reduced by 30% compared to the reference case (mixed-air only cooling concept) [61]. Zhang et al. [62] reported another application from Xi'an Xianyang International Airport, where a combination of radiant floor and displacement ventilation system reduced the HVAC system energy use in the cooling period by 34% compared to a conventional jet ventilation system.

Radiant systems can be used in environments where vibrations, noise, and dust should be avoided, and steady temperature and humidity levels are required [58,60]. In an art museum in Austria, use of radiant systems coupled with a ground heat exchanger for space heating and cooling resulted in lowered airflow rates and space requirements. Energy and operation costs were less than 50% compared to other fully air-conditioned art museums [60].

Recent novel system applications of radiant systems include TABS (concrete deck) with integrated phase change materials (PCM) [69], TABS combined with diffuse ceiling ventilation [70,71], and temperature and humidity independent control (THIC) systems [72]. Other examples of recent developments in novel terminal units for low temperature heating and high temperature cooling systems can be found in [73].

2.2.6. Summary

Based on the reviewed literature, the characteristics of radiant heating and cooling systems can be summarized as follows:

- Low temperature heating and high temperature cooling systems;
- Possibility of coupling with natural heat sources and sinks (ground, lake- or sea-water, etc.) [37,66,74];
- Favorable operating conditions for heating and cooling plants (heat pumps, chillers, boilers, etc.) [37,66];
- Possibility of transferring peak heating and cooling loads to off-peak hours, and peak load reductions [74];
- Smaller-capacity heating and cooling plants, and downsized ventilation systems (air-handling unit capacity, ducts, etc.) [4,75];
- Reduced total energy use, including the energy use of auxiliary components such as pumps and fans;
- Lower heat losses [75,76];
- Free use of space, no obstacles, no cleaning requirements, quiet operation [66];
- Improved indoor air quality, uniform temperature distribution in indoor spaces, reduced risk of draught, and reduced vertical air temperature differences;
- Less space requirements (shafts, equipment rooms), lowered construction heights for each floor and saved building materials (reduced costs for the construction) [4];
- Possibility of initial, operational, and energy cost savings.

In order to have a completely satisfactory indoor environment and an optimally functioning system, following issues should be considered carefully:

- Fine-tuning of the system through monitoring the building in the first one to two years of operation [38], including control system;
- Proper load calculation and system dimensioning [37,75];
- Humidity control (latent loads) [37,66];
- Acoustic environment [4];
- Ventilation - radiant system interaction (avoiding simultaneous heating and cooling, avoiding condensation, providing the necessary amount of fresh air, etc.).

Radiant surface heating and cooling systems have a broad range of applications in different climates and in different building types. An effective and properly functioning application requires appropriate design and operation.

It is crucial to reduce the heating and cooling loads during the design phase for an effective use of radiant heating and cooling systems in buildings [37,77]. After this initial step, where the heating and cooling loads have been minimized, radiant systems can be used to heat or cool indoor spaces in an energy efficient and cost-effective way while providing a comfortable and healthy indoor environment for building occupants.

3. Details of the experimental house

This chapter describes the main envelope characteristics and the mechanical systems of the case study house that was used for the analyses. Only the most important characteristics are given in the following, further details can be found in Paper 7 and Paper 9.

In the experimental evaluation of the house, the following settings were used, while for the theoretical analysis, certain assumptions were made. These assumptions are given in Chapter 4.

3.1. Construction details

The house was a detached, one-story, single-family house with a floor area of 66.2 m^2 and a conditioned volume of 213 m^3. The house was constructed from pre-fabricated wooden elements that were made from layers of laminated veneer lumber boards, which in combination with I-beams in between formed the structural elements. The house was insulated with a combination of 200 mm mineral wool (between the boards of the structural elements) and 80 mm compressed stone wool fibers (40 mm on each side, outside the boards of the structural elements). The walls, roof and floor structures were formed by installing prefabricated elements in a sequential order, and the joints were sealed. The North and South glazing façades were inserted later and the joints between the glazing frame and the house structure were sealed. The house was supported on 200-300 mm concrete blocks and the space between the ground and the house's floor structure was covered, which created a crawl-space below the house.

Inside the house, there was a single space with a high and inclined ceiling, which combined kitchen, living room and bedroom. The technical room was completely insulated from the main indoor space, and had a separate entrance. The wall between the technical room and the indoor space was insulated with the same level of insulation as the envelope. The glazing façades were partly shaded by the roof overhangs. No solar shading was installed in the house except for the skylight window. All windows had a solar transmission of 0.3. The largest glazing façade was oriented to the North with a 19° turn towards the West. Fig. 5 shows the exterior views of the house.

Fig. 5. Exterior views of the house, seen from North-West (left) and South-West (right).

Table 2 shows the surface areas and thermal properties of the envelope.

Table 2. Thermal properties of the envelope.

	North	*South*	*East*	*West*	*Floor*	*Ceiling*
Walls, Area, [m^2]	-	-	37.2	19.3	66.2	53
Walls, U-value, [W/m^{2}K]	-	-	0.09	0.09	0.09	0.09
Windows, Area, [m^2]	36.7	21.8	-	-	-	0.74
Windows, U-value, [W/m^{2}K]	1.04	1.04	-	-	-	1.04

3.2. Details of the heating, cooling, and ventilation system

The sensible heating and cooling of the house relied on low temperature heating and high temperature cooling principle using the hydronic radiant system in the floor. The floor structure was a dry radiant system, consisting of a piping grid installed in the wooden layer. The details of the floor system were chipboard elements with aluminum heat distribution plates (thickness 0.3 mm and width 0.17 m), PEX pipe, 17x2.0 mm. Pipe spacing was 0.2 m. The available floor area for the embedded pipe system installation was 45 m^2, which is 68% of the total floor area. Fig. 6 shows the details of the floor structure used in the house.

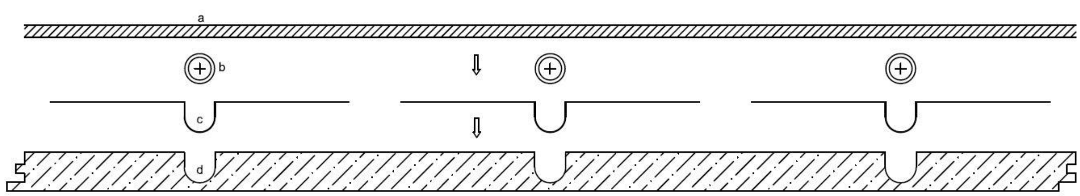


Fig. 6. a) Floor covering, b) Pipe, c) Heat distribution plate, d) Under-floor plate, 22 mm.

The heat source and sink of the house for space heating and cooling was outdoor air, using a reversible air-to-brine heat pump. The minimum, maximum cooling capacities and the nominal power input in the cooling mode were 4.01, 7.1, and 2.95 kW, respectively. The minimum, maximum heating capacities and the nominal power input in the heating mode were 4.09, 7.75, and 2.83 kW, respectively.

There was a flat-plate heat exchanger between the hydronic radiant system of the house and the air-to-brine heat pump. The pipes between the heat exchanger and the heat pump were filled with an anti-freeze mixture (40% ethylene glycol) to avoid frost damage during winter.

A mixing station, which linked the radiant floor heating and cooling system with the heat source and sink, and a controller adjusted the flow and supply temperature to the floor loops. The radiant system was controlled based on the operative temperature set-point that was inserted on a room thermostat (a matt gray half-sphere) in 0.5 K intervals and on the relative humidity inside the house to avoid condensation during summer.

The house was ventilated mechanically by an air-handling unit (AHU). The mechanical ventilation was only used to provide fresh air into the house since the main sensible heating and cooling terminal of the house was the radiant system. This also made it possible to have lower airflow rates compared to a system where space heating and cooling is mainly obtained by an air system [4]. The design ventilation rate was 0.5 ACH [78]. The intake air was taken from the crawl-space.

Passive and active heat recovery options were available in the AHU. The passive heat recovery was obtained by means of a cross-flow heat exchanger and this passive heat recovery system had an efficiency of 85% (sensible heat). By-pass was possible depending on the intake air temperature. The active heat recovery was achieved by means of a reversible air-to-water heat pump that was coupled to the domestic hot water tank. The AHU could supply fresh air at a flow rate of up to 320 m^3/h at 100 Pa. Humidification of the supply air was not possible due to the limitations of the AHU. The two air supply diffusers can be seen on the technical room wall in Fig. 7.

Fig. 7. Panoramic view of the interior (left), the measurement location (middle) and the globe and air temperature sensors (right).

The house was designed as a plus-energy house and the roof of the house was covered with photovoltaic/thermal (PV/T) panels. PV/T panels convert the incoming solar radiation to electricity (PV) and thermal energy (T). The electricity was generated by mono-crystalline cells and the total cell area was 50.8 m^2. The house was connected to the grid and there was no storage of electrical energy. The installed nominal power was cut down to 9.2 kWp by two inverters. The thermal part of the PV/T panels was not operational until the summer of 2014 and only limited data are available from this period; therefore, the thermal part is not described further in this study. Detailed information regarding the performance of the thermal part and its effects on the electricity production of the PV cells were reported in Paper 7.

4. Theoretical and numerical evaluation of low temperature heating and high temperature cooling systems' performance in a plus-energy house

This chapter presents the theoretical and numerical analyses that were carried out assuming that the house was conditioned using different space heating and cooling systems. The following part is based on Paper 2 to Paper 6. Detailed information can be found in those publications.

4.1. Introduction

The goal of providing comfortable and healthy indoor environments with minimal energy and resource use requires proper analysis tools. Therefore, there is a need to have accurate analysis methods during the design, simulation, testing and operation phases of heating, cooling, and ventilation systems to evaluate performance, find ways to improve energy efficiency and to quantify the energy and emission (CO_2 and other greenhouse gases) saving potentials. Different evaluation tools provide different insights into the system under consideration.

The European Energy Performance of Buildings Directive (EPBD) uses primary energy or CO_2 emission to evaluate energy efficiency. International Energy Agency's (IEA) Energy in Buildings and Communities Programme (EBC) Annex 37 (Low Exergy Systems for Heating and Cooling in Buildings) [79] and Annex 49 (Low Exergy Systems for High-Performance Buildings and Communities) [80] both used exergy as a measure of energy efficiency.

When designing heating and cooling systems, although heating and cooling load calculations and energy-based analysis are necessary for system dimensioning, they are not sufficient because energy analysis cannot quantitatively clarify the effects of working temperatures. Several studies have documented that energy analysis alone is not sufficient to understand completely energy use [81-83]. Exergy can be used for this purpose and exergy analysis articulates more precisely and accurately the different quality of energy sources and flows.

Exergy has a wide application range in engineering systems, including heating, cooling, ventilation systems and built environment [84-88]. Gonçalves et al. [89] compared energy and exergy performances of eight space heating alternatives under different climatic conditions. Zhou and Gong [90] studied the whole chain of exergy flows for a building heating and cooling system by hourly varying the reference state. Balta et al. [91], Lohani [92], and Lohani and Schmidt [93] studied different heat sources for building heating applications. Zmeureanu and Wu [94] studied the energy and exergy performance of residential heating systems. Schmidt [95] compared different heat sources and heat emission systems (radiator and floor heating), and concluded that the floor heating system performs close to ideal conditions (real exergetic demand of the zone). This is mainly due to the low temperature heating possibility of water-based radiant floor heating systems. Kilkis [96] studied the possibilities of coupling radiant floors to air-source heat pumps, and showed that it is possible to eliminate the supplementary boilers, and this coupling can increase the coefficient of performance (COP) of heat pumps.

In addition to heating needs, due to several cases of overheating in residential buildings [97,98], and due to increased requirements for comfort [36], cooling in residential buildings is becoming more important and almost a necessity.

Air-based or water-based systems can be used to heat or cool buildings. Although different studies have evaluated the performance of air-based and water-based heating and cooling systems in office buildings [41,44,53], so far there has only been little focus on residential buildings and dwellings regarding cooling systems and their exergy performance.

A recent annex from IEA EBC, Annex 59 (High Temperature Cooling and Low Temperature Heating in Buildings) [99,100], has used entransy as another tool to study heating and cooling systems in buildings. The annex aimed at finding ways of minimizing temperature differences in HVAC systems for high energy efficiency in buildings using entransy analysis. Entransy is defined as an object's heat transfer ability stemming from the analogy between heat conduction and electrical conduction [101,102].

So far, the application of entransy in analyzing buildings and building components is limited and only a few studies have been carried out. Zhang et al. [62,103] developed the entransy analysis method for different HVAC system components and for heat transfer mechanisms in indoor spaces. Zhang et al. used entransy analysis to compare the performance of an air-based cooling system to a radiant floor cooling system in an airport. The results of Zhang et al. show that entransy analysis can also be applied to study heating and cooling systems in buildings.

Energy, exergy, and entransy analyses were the tools used for theoretical evaluation of the performance of the studied heating and cooling systems.

4.2. Summary of the studied systems

For the theoretical and numerical analyses, it was assumed that the house was conditioned by different heating and cooling systems. The studied heating systems were warm-air heating with and without heat recovery, floor heating with different floor covering resistances, and radiator heating with different working temperatures. The heat source was a natural gas fired condensing boiler or an air-to-water heat pump, depending on the studied case.

The studied cooling systems were air cooling with different supply air temperatures to indoors and with different air intakes (from outdoors or from the crawl-space), and floor cooling with different space cooling loads and with different heat sinks. For air cooling cases, the effects of internal solar shading versus external solar shading on system performance were also investigated.

For all cases, in addition to the thermal flows, energy and exergy inputs to pumps and fans were also compared.

4.2.1. Space heating and cooling load calculations

Space heating and cooling load calculations were carried out under steady-state conditions and the location was assumed to be Copenhagen, Denmark. No solar heat gains were considered in heating season. Air temperature was assumed to be equal to mean radiant temperature for all cases.

A heat recovery unit on the exhaust air was used in warm-air heating with heat recovery, radiator heating and in floor heating cases. This heat recovery unit (a cross-flow heat exchanger) had a heat recovery efficiency of 0.85 (sensible heat) and, hence, the supply air temperature to the indoor space after the heat recovery was 16.3°C (temperature of the air entering the air-heating coil in warm-air heating with heat recovery). The design ventilation rate was 0.5 ACH in all radiator and floor heating cases, in order to provide the necessary amount of fresh air to the indoor space.

Table 3 summarizes the boundary conditions and the resulting space heating loads.

Table 3. Load calculation parameters and resulting space heating loads.

Indoor temperature [°C]	20
Outdoor temperature [°C]	-5
Internal heat gain [W/m^2]	4.5
Ventilation rate [ACH]	0.5
Infiltration rate [ACH]	0.2
Heating load – warm-air heating, total [W] and specific [W/m^2]	2048 / 31
Heating load – radiator heating and floor heating, total [W] and specific [W/m^2] [a]	2180 / 33

[a] The difference between the two space heating loads is due to the fresh air being supplied at 16.3°C with a ventilation rate of 0.5 ACH. This contributes to the space heating load, hence the higher load in radiator heating and floor heating.

For all cooling cases, an external shading was used, the intake air was from outdoors and the cooling plant was an air-to-water heat pump, unless otherwise stated. The design ventilation rate was 0.5 ACH for floor cooling cases and the intake air was not cooled.

In some of the cooling cases, the intake air was from the crawl-space instead of the outdoor air. In those cases, the fresh air temperature coming into the AHU or to the indoor space was 21.3°C. Details of the solar heat gain calculations are given in Paper 3.

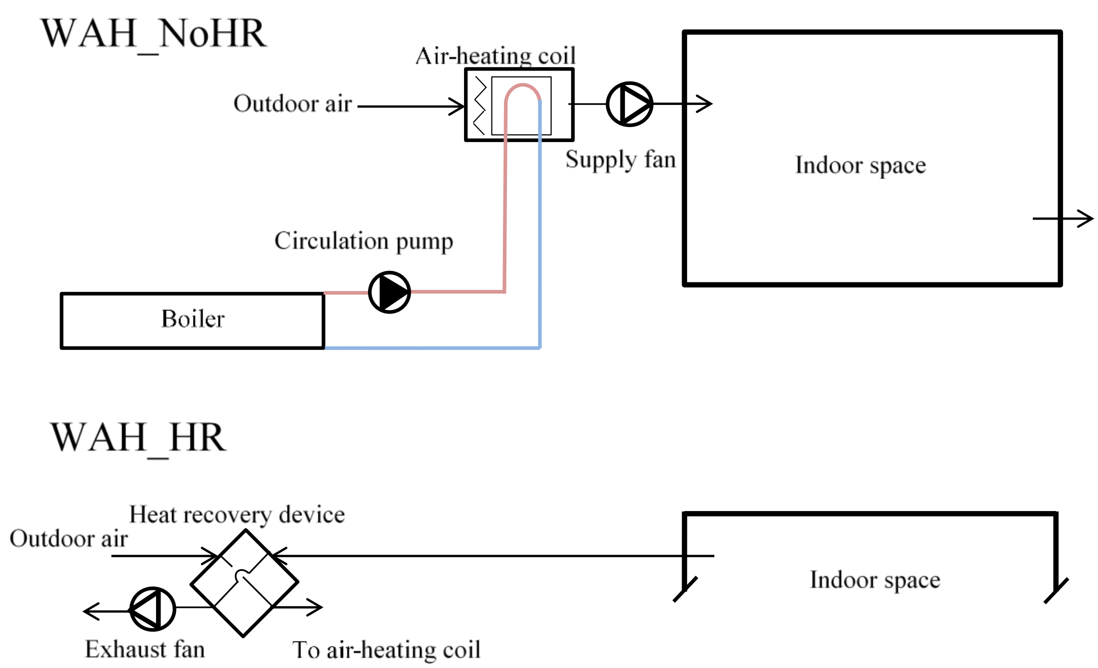
Table 4 summarizes the boundary conditions and resulting space cooling loads.

Table 4. Load calculation parameters and resulting space cooling loads.

Indoor temperature [°C]	26
Outdoor temperature [°C]	30
Air temperature in the crawl-space [°C]	21.3
Internal heat gain [W/m^2]	4.5
Ventilation rate [ACH]	0.5
Infiltration rate [ACH]	0.2
Cooling load – air cooling, total [W] and specific [W/m^2] [a]	3170 / 48
Cooling load – air cooling, total [W] and specific [W/m^2]	1042 / 16
Cooling load – floor cooling, total [W] and specific [W/m^2] [b]	1183 / 18
Cooling load – floor cooling, total [W] and specific [W/m^2] [c]	876 / 13

[a] Internal shading. [b] Fresh air intake from outdoors. [c] Fresh air intake from crawl-space.

Fig. 8 and Fig. 9 show the schematic drawings of the studied heating and cooling systems, respectively.



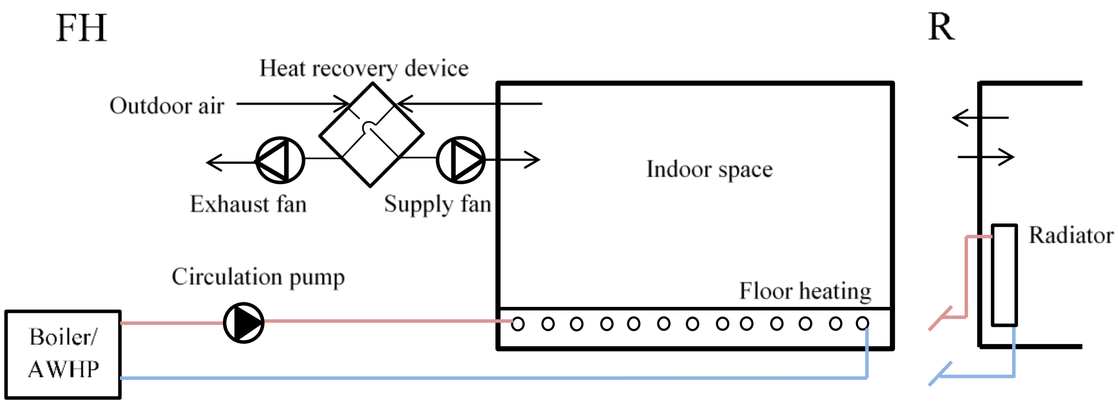
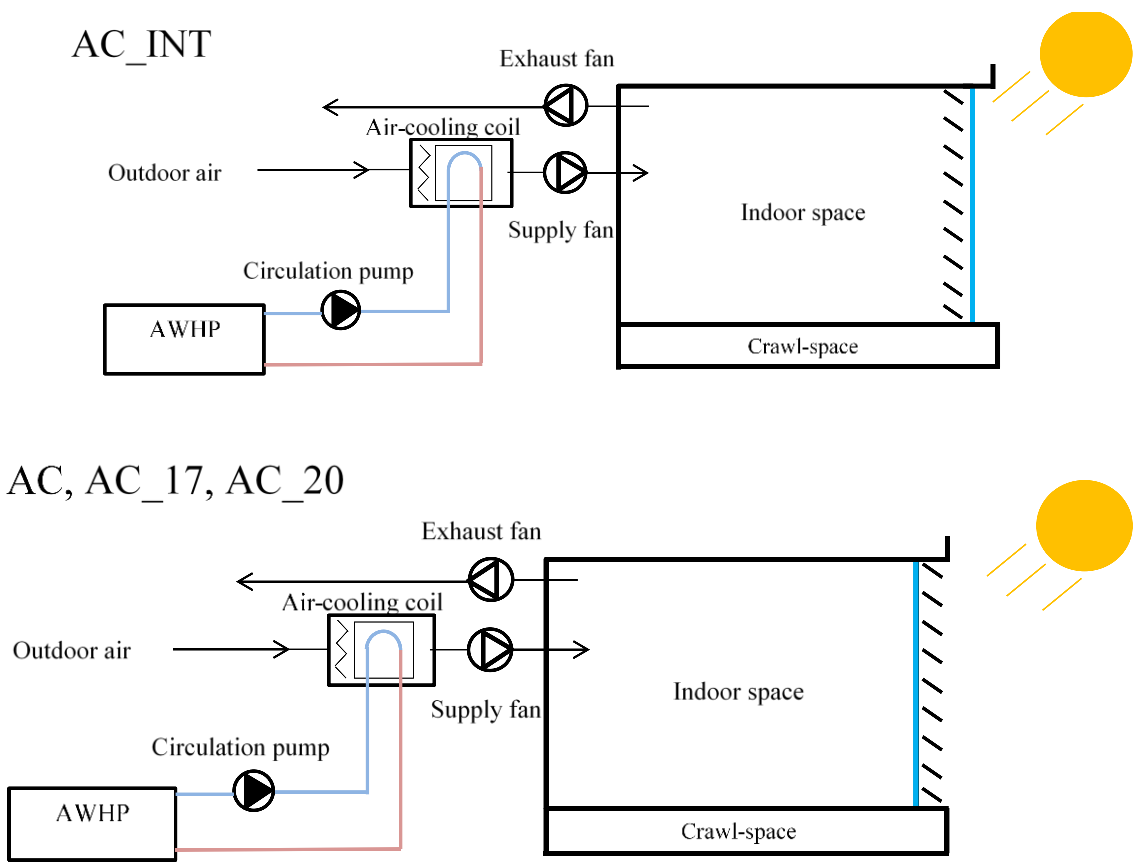
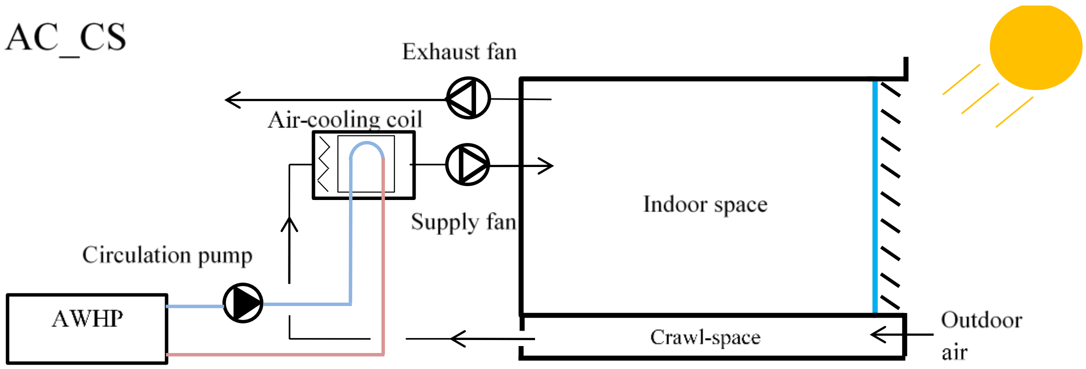


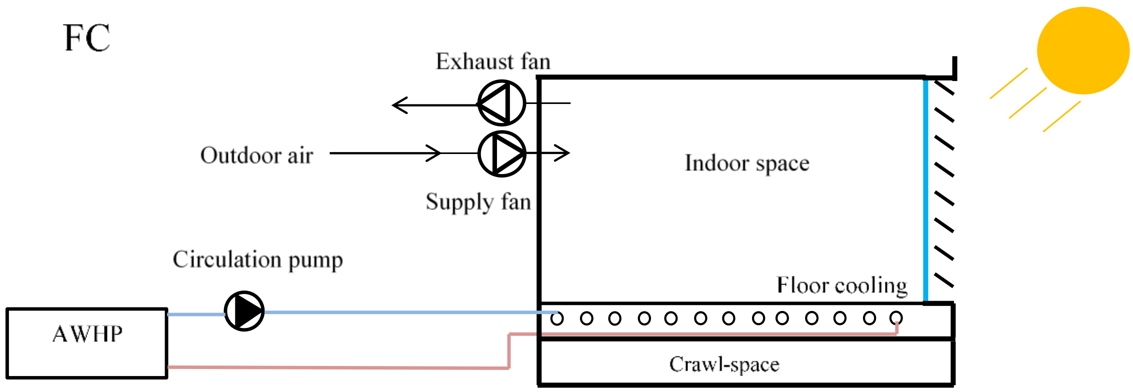
Fig. 8. Schematic drawings of the analyzed heating systems (WAH_NoHR: Warm-air heating without heat recovery, WAH_HR: Warm-air heating with heat recovery, FH: Floor heating, R: Radiator heating, AWHP: Air-to-water heat pump).



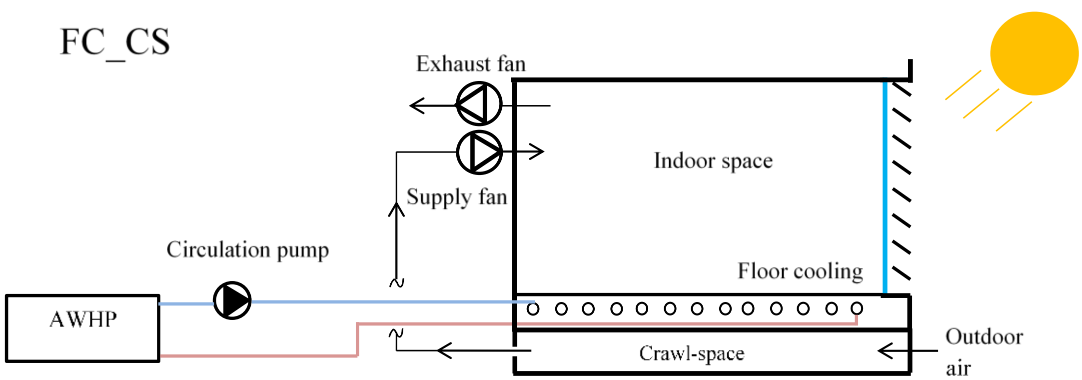
AC_CS


Exhaust fan
Air-cooling coil
Supply fan
Indoor space
Circulation pump
AWHP
Outdoor
air
Crawl-space

FC


Exhaust fan
Outdoor air
Supply fan
Indoor space
Circulation pump
Floor cooling
AWHP
Crawl-space

FC_CS


Exhaust fan
Indoor space
Supply fan
Circulation pump
Floor cooling
AWHP
Outdoor
air
Crawl-space

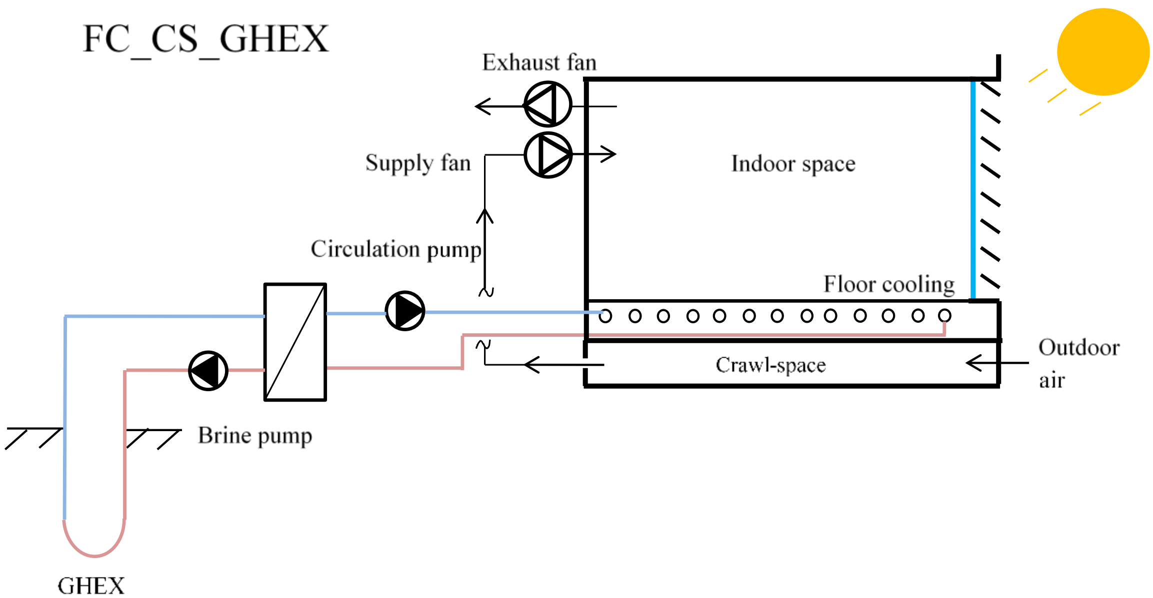


Fig. 9. Schematic drawings of the analyzed cooling systems (AC: Air cooling, INT: Internal shading, AWHP: Air-to-water heat pump, FC: Floor cooling, GHEX: Ground heat exchanger, CS: Crawl-space).

4.2.2. Warm-air heating and air cooling

For warm-air heating, supply air temperature of 35°C was chosen, which is limited by the national regulations in Denmark [12]. When there was no heat recovery (WAH_NoHR), outdoor air at -5°C was heated to 35°C. When there was heat recovery on the exhaust air (WAH_HR), it was possible to bring the outdoor air from -5°C to 16.3°C before it entered the air-heating coil.

The necessary heating rate for bringing the outdoor air at -5°C to the supply air temperature of 35°C was 5460 W, and it decreased to 2559 W with heat recovery. The supply and return water temperatures to and from the air-heating coil were 50°C and 39°C, respectively [104].

In air cooling cases, the intake air was passing through an air-cooling coil where it was cooled to the supply temperature. The water entered the air-cooling coil at 7°C and returned at 12°C. The heat to be removed from the intake air corresponds to the required amount of heat to lower the temperature of the intake air to the required supply temperature, which was 14°C, 17°C or 20°C for respective cases.

Table 5 summarizes the warm-air heating and air cooling cases.

Table 5. Summary of the warm-air heating and air cooling cases (IA: Intake air).

Case	*Supply air temperature [°C]*	*Ventilation rate [ACH]*	*Rate of heating or cooling to IA [W]*	*Mass flow rate in heating or cooling coil [kg/h]*
WAH_NoHR	35	1.9	5460	428
WAH_HR	35	1.9	2559	201
AC_INT [a]	14	3.7	4226	725
AC	14	1.2	1389	238
AC_17	17	1.6	1505	258
AC_20	20	2.5	1736	298
AC_CS [b]	14	1.2	634	109

[a] Internal shading. [b] Air intake from crawl-space.

4.2.3. Floor heating and cooling

Different floor covering resistances were assumed to investigate its effects on system performance: 0.05 m^2K/W (similar to a marble floor and mud-set [104]), 0.09 m^2K/W (actual value, similar to a light carpet [104]), and 0.15 m^2K/W (similar to a heavy carpet [104]), following the most common values given in standards [105]. The same floor covering material was used, wooden floor covering with a thermal conductivity of 0.13 W/mK, with 0.0065 m, 0.012 m and 0.0195 m thickness, respectively. For each resistance value, new water supply and return temperatures were calculated.

In order to provide the necessary heating to the indoor space, a specific heat output of 48.4 W/m2-floor heating area with an average floor surface temperature of 24.7°C was required. The floor surface temperature was the same for all floor heating cases. The temperature drop between the supply and return water in the radiant system was 4°C. The heat output, floor surface temperature, and the mass flow rate were calculated according to [3,105,106]. The mass flow rate was 469 kg/h.

In cooling cases, a floor covering resistance of 0.05 m^2K/W was used to minimize the effects of floor covering resistance on system performance. For FC_CS and FC_CS_GHEX, the heat to be removed by the floor was 876 W, and for FC it was 1183 W. These values correspond to cooling loads of 19.5 and 26.3 W/m^2-cooled floor area, respectively, and corresponding average floor surface temperatures of 23.2 and 22.2°C. The assumed temperature difference between supply and return water flows was 3°C for all cases. For FC_CS and FC_CS_GHEX, this resulted in a mass flow rate of 250 kg/h, and for FC it was 338 kg/h. The cooling output, floor surface temperatures and the mass flow rates were calculated according to [3,105-107].

Table 6 summarizes the floor heating and cooling cases.

Table 6. Summary of the floor heating and cooling cases.

Case	*Floor covering resistance [m^2K/W]*	*Supply water temperature [°C]*	*Return water temperature [°C]*
FH_LoRes	0.05	33	29
FH_MRes	0.09	35.8	31.8
FH_HiRes	0.15	39.8	35.8
FC	0.05	16.5	19.5
FC_CS [a] & FC_CS_GHEX [b]	0.05	18.6	21.6

[a] Air intake from crawl-space. [b] Air intake from crawl-space and floor cooling coupled to ground heat exchanger.

4.2.4. Radiator heating

The assumed radiator type was a double panel steel radiator with extended surface (fins) [104]. Four sets of working temperatures were assumed according to prEN 15316-2:2014 [108]: 45/35, 55/45, 70/55, and 90/70 (supply/return water temperature in °C). The required flow rates in the radiators were determined according to the space heating load and the temperature difference between the supply and return water flows to and from the radiator.

Table 7 summarizes the radiator heating cases.

Table 7. Summary of the radiator heating cases.

Case	*Supply water temperature [°C]*	*Return water temperature [°C]*	*Mass flow rate [kg/h]*
R_45	45	35	188
R_55	55	45	188
R_70	70	55	125
R_90	90	70	94

4.3. Results of exergy analysis

Fig. 10 shows the results of exergy analysis of floor heating systems with different floor covering resistances as an example. Detailed results of heating season exergy analysis are given in Paper 2 and some of the results that were used for exergy consumption and entransy dissipation comparison are given in Chapter 4.6.

Fig. 10. Exergy flows in floor heating systems with different floor covering resistances.

The main conclusions from the exergy analysis in heating season are as follows:

- Among the investigated cases, the floor heating system had the lowest exergy consumption and it performed better than other space heating systems in terms of required exergy input, and exergy consumption;
- Coupling a natural gas fired condensing boiler with the floor heating system is not an effective strategy due to the mismatch of the exergy supply and demand. Therefore, it does not allow taking advantage of the low exergy demand of the radiant floor heating system;
- Using a heat pump instead of a boiler as the heat source could provide a better match for the low exergy demand of the floor heating system, but there is a critical COP value (2.57 in this study) and only above this COP value, it is beneficial to use a heat pump instead of a boiler. It is crucial to account for the source of electricity supplied to the heat pump (e.g. generated at a remote, fossil fuel power plant or at a nearby renewable energy plant);
- Choice of floor covering has significant effects on the performance of the floor heating system and on the whole system. Higher floor covering resistance hinders the performance of the whole system. When comparing the cases with the lowest and highest floor covering resistances, the exergy consumption in the floor structure doubles and the required exergy input to the floor structure increases by 16% with a higher floor covering resistance. This means that to obtain the same space heating effect, higher electricity input is required to the heat pump, and 14% higher exergy input is necessary to the power plant where the electricity is generated. In order not to hinder the performance of a radiant system (floor heating, floor cooling, ceiling cooling and so forth), the covering resistance should be kept to a minimum, providing that structural and maintenance (e.g. wear and tear on floor coverings) concerns can be met and that the occupants are aesthetically satisfied;

- The water-based systems use lower auxiliary energy than the air-based heating system: about 42% and 68% lower compared to warm-air heating without and with heat recovery, respectively. This is a clear benefit of water-based systems over air-based systems. To benefit fully from the low-exergy heating potential of the low temperature heating systems, the dimensioning and choice of auxiliary components should be made carefully and this is clearly explained with the exergy analyses;
- In this study, the application of heat recovery was beneficial: the extra exergy input required to the power plant for the additional fan in the case of warm-air heating with heat recovery was 357 W, which is significantly smaller than the reduced exergy input to the boiler due to the application of heat recovery (2998 W). These values should be compared before applying heat recovery to ensure that it is beneficial.

Fig. 11 shows the results of exergy analysis of air cooling and floor cooling systems when the intake air was from outdoors, as an example. Detailed results of cooling season exergy analysis are given in Paper 3 and some of the results that were used for exergy consumption and entransy dissipation comparison are given in Chapter 4.6.

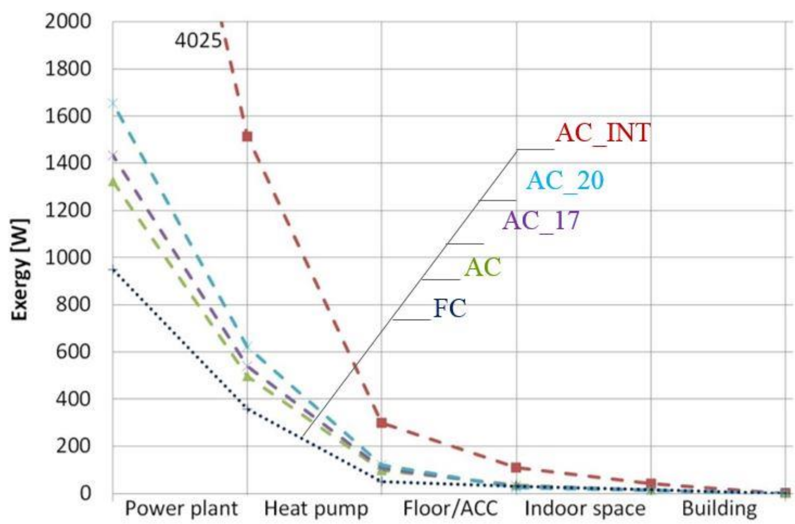


Fig. 11. Exergy flows in chosen air cooling and floor cooling systems (intake air is from outdoors, ACC: air-cooling coil).

The main conclusions from the exergy analysis in cooling season are as follows:

- Cooling (and heating) exergy demand of the building should be minimized from the beginning to achieve reasonable exergy efficiency and to allow the use of naturally available exergy sources and sinks. The cooling demand also influences the choice of indoor terminal units, e.g. with a high cooling load, it might not be possible to use radiant cooling systems due to dew point concerns. The choice of terminal unit is crucial since only with certain terminal units it is possible to use renewable energy resources, and since this choice directly affects occupant thermal comfort;

- The water-based radiant floor cooling performed better than the air-based cooling in terms of energy, exergy demand and consumption. When an air-to-water heat pump was used as the cooling source and the intake air was outdoor air, the exergy input required to the power plant was 28% smaller for the floor cooling system compared to the air cooling system. The water-based systems required 53% to 75% smaller auxiliary exergy input compared to air-based systems, due to the use of water as the main heat carrier medium;
- Cool exergy concept was used to quantify the available exergy in the crawl-space and in the ground. Integration of these natural exergy resources to the cooling system resulted in significant improvements in system performance. The use of cool exergy available in the crawl-space resulted in 54% and 29% smaller exergy input to the power plant for the air-based and water-based cooling systems, respectively. In these cases, only 4.5 W and 10.7 W of cool exergy provided from the crawl-space decreased the exergy input by natural gas to the power plant by 270 and 719 W, respectively;
- The coupling of ground with the radiant floor cooling system is effective since the exergy supply from the ground matches the low exergy demand of the floor cooling system. For floor cooling cases, it is possible to reduce the exergy input to the power plant by 90% and 93%, with the use of ground, and use of ground and crawl-space, respectively. Brine pump power should be kept to a minimum to truly benefit from the "free" cooling through the cool exergy stored in the ground;
- The benefit of coupling the ground with the floor cooling system was shown with the exergy efficiency values; 18% and 8.4% for the conventional and the new definition of the exergy efficiency, respectively, and 53.2% of the total exergy input to the system was from the ground. The decrease in efficiency indicates that the natural resources within our immediate surroundings should be used efficiently, and not exploited in ineffective ways through poor utilization.

4.4. Results of entransy analysis

Fig. 12 shows the results of entransy analysis of the floor heating system with a low floor covering resistance on a temperature-transferred heat (T-Q) diagram as an example. Detailed results of entransy analysis of different systems are given in Paper 4 and some of the results that were used for exergy consumption and entransy dissipation comparison are given in Chapter 4.6.

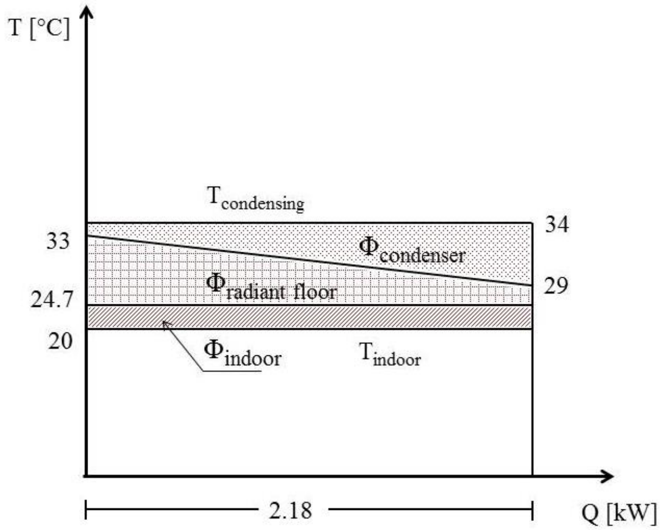


Fig. 12. T-Q diagram of the floor heating system with a low floor covering resistance (Φ: Entransy dissipation [kWK]).

The main conclusions from the entransy analysis are as follows:

- Among the investigated space heating systems, floor heating had the lowest total entransy dissipation: floor heating system with low floor covering resistance had 49% lower entransy dissipation compared to the radiator heating with lowest supply and return temperatures, and 59% lower compared to warm-air heating with heat recovery;
- In cooling operation, when the intake air was from outdoors, floor cooling system had 48% lower total entransy dissipation compared to air cooling. When the intake air was from the crawl-space, floor cooling had 36% lower total entransy dissipation compared to air cooling;
- Compared to the floor heating with high floor covering resistance, entransy dissipation in the floor structure was reduced by 31% in floor heating with medium floor covering resistance and by 52% in floor heating with low floor covering resistance. Total entransy dissipation was reduced by 19% in floor heating with medium floor covering resistance and by 33% in floor heating with low floor covering resistance compared to floor heating with high floor covering resistance. These system behaviors indicate that heat transfer potential was being wasted in the floor structure, and therefore the resistance between the heat transfer medium and the surface should be minimized to fully benefit from the low temperature heating and high temperature cooling potential of the floor heating and cooling system;
- Low temperature heating and high temperature cooling systems in buildings achieve higher performance in comparison to other space heating and cooling systems. This was shown in terms of entransy dissipation, using an example of these systems (radiant floor heating and cooling) and comparing its performance to other space heating and cooling systems;
- Decreased temperature differences within the whole system can be achieved by using higher cooling sink temperatures and lower heating source temperatures, and low

temperature heating and high temperature cooling systems enable this. Expressed in terms of entransy; lower entransy dissipation in a system shows a lowered temperature for heating and higher temperature for cooling, which would enable energy savings, improved resource and energy efficiency by allowing favorable operating conditions for heating and cooling plants (e.g. heat pumps, chillers, etc.), and by allowing use of natural heat sources and sinks.

4.5. Discussion

The detailed discussion of the results and other aspects are given in Paper 2 to Paper 6. The following part explains additional discussions.

4.5.1. Discussion about assumptions

In the air-based heating and cooling cases, the supply air was outdoor air and no recirculation was used. Use of recirculation air could have slightly improved the performance of air-based systems, however this does not mean that with recirculation air, air-based systems would have performed better than water-based systems. In addition, other studies comparing the performance of air-based versus water-based heating and cooling systems also used full outdoor air for reference [40], and the use of recirculation air in buildings is gradually being reduced due to distribution of pollutants within the building [37,43], and instead, use of dedicated outdoor air systems is increasing [43].

While the water-based systems had a ventilation rate of 0.5 ACH only for IAQ concerns, the air-based systems used a higher ventilation rate than the one required for IAQ concerns (Table 5); however, this might not necessarily mean that a higher satisfaction would be obtained, as PAQ might be a problem with the increased supply air temperature [18,19]. The fully mixed assumption might not hold in a real application and the ventilation effectiveness might decrease [22], especially due to thermal stratification and mixing effects, considering the shape of the indoor space of this house. A CFD model could have been used to determine the exact thermal indoor environment and air distribution patterns.

Certain issues also should be considered in relation to having the intake air from the crawl-space. Radon should be considered for different locations, i.e. a radon barrier might be needed in locations where radon is a concern. In addition to radon, also the quality of air coming from the crawl-space can cause problems with IAQ and it is crucial to consider this aspect in order not to cause any dissatisfaction to the occupants with the indoor environment. The moisture of the air coming in from the crawl-space can also be a problem, and the consideration of dehumidification needs may result in different energy inputs and system performance. The moisture of the air from the crawl-space could cause problems regarding condensation on the floor cooling unless properly dehumidified. The condensation can be avoided through a dew point control on the water supply temperature to the floor cooling loops.

In the present analyses, the heat transfer from the floor system toward the indoor space was evaluated as combined radiation and convection. Further studies could be carried out to separate the heat transfer mechanism into radiation and convection, and investigate its effects on system performance and on thermal indoor environment.

4.5.2. Discussion about entransy

There has recently been discussions in the scientific community concerning the applicability of entransy analysis to different systems. In this study, entransy was used as a tool to analyze heating and cooling systems in buildings and to find ways to reduce temperature differences in system components, as in IEA EBC Annex 59 [99]. Discussions and rebuttals on entransy can be found in [109-124].

4.6. Comparison of exergy consumption and entransy dissipation in system components

The following part reports an initial comparison between the results obtained from exergy and entransy analyses on the terminal units and indoor spaces. Other system components were not used for comparisons.

Both methods used a steady-state approach. The results obtained from exergy and entransy analyses were compared in terms of exergy consumption and entransy dissipation. The results were compared in relative terms; analyzing the changes in terminal units and in indoor space, and not on a whole system basis since the boundaries used for the analyses were different. In addition, two methods work with different units; exergy consumption has the unit W and entransy dissipation has the unit WK.

For heating and cooling seasons, the comparisons were made for terminal units (floor structure in floor heating and cooling, and air-heating and air-cooling coils in warm-air heating and air cooling cases, respectively) and for indoor spaces. When all of the heating or cooling systems were compared to each other, highest value of exergy consumption or entransy dissipation was chosen as the base case. The same applied when the systems were compared within the same heating or cooling method.

Table 8 and Table 9 show the exergy consumption and entransy dissipation comparison in heating and cooling operation, respectively. In Table 8 and Table 9, X_C is exergy consumption [W] and Φ is entransy dissipation [kWK]. Reductions in exergy consumption and entransy dissipation are given relative to the highest values, either among the whole heating or cooling methods or among the same heating or cooling method (floor heating with different floor covering resistances, etc.).

Table 8. Exergy consumption and entransy dissipation comparison in heating.

	Terminal unit				*Indoor space*			
	X_C [W]	Φ [kWK]	Reduction compared to base, X_C / Φ	Reduction compared within heating method, X_C / Φ	X_C [W]	Φ [kWK]	Reduction compared to base, X_C / Φ	Reduction compared within heating method, X_C / Φ
WAH_NoHR	479.6	161.1	-	-	46.3	15.4	81.5% / 83.4%	-
WAH_HR	137.0	48.3	71.4% / 70%	71.4% / 70%	46.3	15.4	81.5% / 83.4%	-
R_70	-	-	-	-	250.8	92.7	-	-
R_55	-	-	-	-	183.3	65.4	26.9% / 29.5%	26.9% / 29.5%
R_45	-	-	-	-	125.5	43.6	50.0% / 53.0%	50.0% / 53.0%
FH_HiRes	82.8	28.6	82.7% / 82.3%	-	31.2	10.3	87.6% / 88.9%	-
FH_MRes	58.4	19.8	87.8% / 87.7%	29.5% / 30.8%	31.2	10.3	87.6% / 88.9%	-
FH_LoRes	41.0	13.7	91.5% / 91.5%	50.5% / 52.1%	31.2	10.3	87.6% / 88.9%	-

Table 9. Exergy consumption and entransy dissipation comparison in cooling.

	Terminal unit				*Indoor space*			
	X_C [W]	Φ [kWK]	Reduction compared to base, X_C / Φ	Reduction compared within cooling method, X_C / Φ	X_C [W]	Φ [kWK]	Reduction compared to base, X_C / Φ	Reduction compared within cooling method, X_C / Φ
AC_INT	191.0	52.9	-	-	66.2	19	-	-
AC	62.8	17.4	67.1% / 67.1%	67.1% / 67.1%	21.8	6.2	67.1% / 67.4%	67.1% / 67.4%
AC_17	76.0	21.1	60.2% / 60.1%	60.2% / 60.1%	16.2	4.7	75.5% / 75.3%	75.5% / 75.3%
AC_20	96.7	27.0	49.4% / 49.0%	49.4% / 49.0%	10.7	3.1	83.8% / 83.7%	83.8% / 83.7%
AC_CS	19.0	5.1	90.1% / 90.4%	90.1% / 90.4%	21.8	6.2	67.1% / 67.4%	67.1% / 67.4%
FC	17.7	5.0	90.7% / 90.6%	-	15.2	4.5	77.0% / 76.3%	-
FC_CS	9.5	2.7	95.0% / 94.9%	46.3% / 46%	8.3	2.5	87.5% / 86.8%	45.4% / 44.4%
FC_CS_GHEX	9.5	2.7	95.0% / 94.9%	46.3% / 46%	8.3	2.5	87.5% / 86.8%	45.4% / 44.4%

For the compared cases, exergy analysis and entransy analysis show similar results, e.g. highest exergy consumption and entransy dissipation in terminal units occur in the air-heating coil and air-cooling coil, highest exergy consumption and entransy dissipation in the indoor space occur for R_70 in heating operation and in AC_INT in cooling operation, higher floor covering resistance increases exergy consumption and entransy dissipation, etc.

Relative changes indicated by exergy and entransy analyses also show very close values both when compared to the base case and when compared within the specific heating or cooling method.

Further analyses are needed to reach definitive conclusions, e.g. analyses on other heating and cooling systems, under different operation conditions and so forth. It should also be noted that even though a component-based analysis is important, a system-based analysis is required for global improvement and optimization. The exergy consumption and entransy dissipation values given in Table 8 and Table 9 are only for the ones chosen for comparison (terminal unit and indoor space). In exergy and entransy analyses, the processes were evaluated on a more holistic, system-basis (e.g. from the source until the last component in the process).

4.7. Dynamic simulations of the energy performance of different heating and cooling systems

Energy performance of the studied heating and cooling systems for achieving the same thermal indoor environment (operative temperature) was compared by means of dynamic building simulations using IDA ICE. The main results from the simulations are as follows, and details of the simulations can be found in Paper 5 and Paper 6.

4.7.1. Heating season

The main results from the heating season (1st of October to 30th of April) simulations show that:

- Implementing heat recovery in the air-handling unit can reduce the required energy input to the boiler by 47%;
- When the heat source was a boiler, radiators and floor heating used 9% less energy compared to warm-air heating with heat recovery, in total. The main difference was due to the auxiliary energy use, which was 38% less for water-based systems;
- When the heat source was a boiler, the effect of replacing the radiators with a radiant floor heating system on the total primary energy use was negligible. When the radiant floor system was coupled to a heat pump instead of the generic boiler, floor heating systems used 13% less energy, in total;
- Radiant floor heating system with low floor covering resistance coupled to an air-to-water heat pump used 22% less primary energy than warm-air heating with heat recovery, in total.

4.7.2. Cooling season

The main results from the cooling season (1st of May to 30th of September) simulations show that:

- When the fresh air intake was from outdoors, floor cooling system required 29% less primary energy in total, and up to 46% less energy input to the cooling plant compared to air cooling systems;
- When the fresh air intake was from the crawl-space, floor cooling system required 25% less primary energy in total, and 42% less energy input to the cooling plant compared to air cooling systems;
- Using naturally available heat sinks can result in considerable energy savings. In floor cooling cases, when the intake air is from the crawl-space, using a ground heat exchanger instead of a heat pump can result in 33% energy savings in total, and 82% less energy input to the cooling plant, which is now the ground, therefore the only energy use is by the brine pump in the ground loop;
- When the floor cooling system benefited both from the colder intake air from the crawl-space and from the ground heat exchanger, the overall energy use was 37% less compared to the floor cooling case when the intake air was the outdoor air and a heat pump was used as the cooling source. It was also possible to obtain the same space cooling effect with 84% less electricity input to the cooling plant (brine pump).

The results of the dynamic simulations show that water-based low temperature heating and high temperature cooling systems can be used to achieve considerable energy savings under similar thermal indoor environment conditions. LTH and HTC systems also enable coupling of natural energy resources to the heating and cooling systems due to their operating temperatures. It should be noted that the simulations only focused on the thermal indoor environment evaluated in terms of operative temperature and energy performance of different systems. The choice and their installation would depend on a combination of the factors summarized in 2.1.7.

4.8. Summary

Performance of different heating and cooling systems in the case study house was compared theoretically using energy, exergy, and entransy methods. The results were obtained by steady-state theoretical calculations, and by numerical methods using a dynamic building simulation software for energy analysis. The systems were compared under the same thermal indoor environment conditions (operative temperature).

All evaluation methods showed that the radiant floor heating and cooling system had the highest performance among the compared systems, not only in terms of energy but also in terms of exergy and entransy. This was mainly because of the low temperature heating and high temperature cooling possibility of the radiant system. The water-based systems use considerably less auxiliary energy compared to air-based systems.

In order to obtain the most rational use of resources, energy analysis alone is not sufficient. Exergy and entransy analyses showed that it is not only enough to consider the quantity of energy, and the temperatures and temperature differences in the system should also be considered. This finding agrees with that of Zhang et al. [125]. The two analysis methods used different boundaries and different units; therefore, no direct comparison was made between total exergy consumptions or total entransy dissipations. The results of relative improvements in terminal units (e.g. reduced floor covering resistance, etc.) and in indoor space obtained by exergy consumption and entransy dissipation calculations were very close.

Exergy analysis could be used to evaluate heat transfer processes and to study heating and cooling systems, and not only heat-to-work conversion processes. It provides a holistic approach to the system under consideration; from the heat source and sink until the reference environment. It enables comparison of different quality of heat sources and sinks, and evaluation of whole system performance. In addition, system components such as pumps and fans can be included in the exergy analysis and evaluation of the system performance, which will allow a more holistic and global improvement and optimization of the studied systems.

Entransy analysis could also be used to evaluate heat transfer processes and to study heating and cooling systems (e.g. heat transfer in indoor spaces between several heat sources and sinks). Temperature-transferred heat (T-Q) diagrams provide a straightforward way to illustrate these processes.

Both methods could be used to study a system but this would depend on the chosen system boundary, and a complete system approach that considers also the source of the energy is necessary. For example, in entransy analysis, it might not be possible to distinguish between a condenser and an electric heating element; however, this will have considerable effects on energy and exergy performances of the whole system. A strong link to the actual energy and resource use of the whole system is necessary. Further studies are required to determine the exact benefits and limitations of the entransy method.

The ultimate goal in heating and cooling of buildings is to decrease the temperature difference between different components from indoor terminal units to the heating and cooling plants (i.e. to decrease the overall temperature difference in the system, enabling the utilization of non-fossil resources such as ground or solar, and resulting in improved performance of heat pumps and chillers). A certain method or any combination of methods that serves this purpose will be useful for improving and optimizing building heating and cooling systems during the design and operation phases.

Dynamic calculations are widely available for energy analysis, and it is necessary to develop dynamic calculation methods for exergy and entransy analyses to obtain a complete view of system operation during longer periods.

5. Experimental and numerical evaluation of low temperature heating and high temperature cooling systems' performance in a plus-energy house

This chapter presents the measured thermal indoor environment and energy performance of the house evaluated over a one-year period, and the proposed improvement suggestions based on the measurements and dynamic simulations. The following part is based on Paper 7 to Paper 9. Further information can be found in those publications.

5.1. Introduction

Building energy codes are becoming tighter and nearly zero-energy building (nZEB) levels are dictated for new buildings by 2020 in the European Union [126]. A further goal is to design plus-energy houses, i.e. houses that produce more energy from renewable energy resources than they import from external resources in a given year, according to the definition given by the European Commission [127]. These trends are reflected in different initiatives such as the Passive House movement [128,129] and recently in the Active House Alliance [130]. The idea of plus-energy houses is also promoted with competitions such as Solar Decathlon [131], where multidisciplinary teams from universities compete to design, build and operate plus-energy houses. Plus-energy houses could have a significant role in the energy system in a number of ways: they can compensate for the old buildings that are too expensive to upgrade to nZEB levels, and they can act as small-scale power plants in the energy system.

The international focus on the residential sector is increasing [132] and there have been a number of surveys that monitored energy performance and indoor environment in passive houses [133-135]. One commonly encountered problem in low-energy and passive houses is overheating. Overheating has been reported from Denmark by Larsen and Jensen [136], from Sweden by Janson [137] and Rohdin et al. [138], and from Finland by Holopainen et al. [139]. Maivel et al. [98] also reported overheating in new apartment buildings in Estonia. Some of the main reasons of overheating are large glazing areas, poor or lack of solar shading, lack of ventilation [97], lack of thermal mass, and lack of adequate modeling tools in the design phase [140].

In addition to overheating, varying room temperatures [138,139], too low air temperatures in winter, stuffiness and poor air quality, and too low floor surface temperatures in winter [138] are some of the other problems that were reported from low-energy and passive houses.

There are still unsolved issues and problems to be addressed regarding low-energy, passive, and plus-energy houses. Therefore, designers, and engineers can benefit from research on the design, construction and operation of such buildings. In this context, a detached, one-story, single-family house, which was initially designed to be a plus-energy house, was operated from 26th of September 2013 to 1st of October 2014 to compare a number of different heating, cooling, and control strategies [141,142]. The thermal indoor environment and energy performance of the house were monitored during this period.

The performance of different strategies was evaluated in terms of the resulting thermal indoor environment, local thermal discomfort (vertical air temperature difference between head and ankles) and overheating, according to EN 15251:2007 [20], EN ISO 7730:2005 [31] and DS 469:2013 [12], respectively. The energy production and the energy use of the heating, cooling and ventilation systems of the house were used to evaluate its energy performance. Improvement suggestions regarding the design and operation of the building and regarding its heating and cooling systems are provided.

5.2. Materials and methods

5.2.1. Experimental set-up

The house was located in Bjerringbro, Denmark and it was unoccupied during the measurement period. The occupancy and equipment schedules (internal heat gains) were simulated by means of heated dummies.

The occupancy and equipment schedules were adjusted with timers. Two dummies were used to simulate occupants (the dummies had the same surface temperatures as a person would have) at 1.2 met (ON from 17:00 to 08:00 on weekdays and from 17:00 to 12:00 on weekends). One dummy (equipment #1, 120 W, 1.8 W/m^2) was always ON to simulate the house appliances that are always in operation. The fourth dummy (equipment #2, 180 W, 2.7 W/m^2) was used to simulate the house appliances that are in use only when the occupants are present. The fifth dummy was used to represent additional lights (180 W, 2.7 W/m^2, ON from 06:00 to 08:00 and from 17:00 to 23:00 until 27^{th} of May 2014, and after this date, ON from 20:00 to 23:00 every day). The house had ceiling mounted lights ON from 21:00 to 23:00 every day (140 W, 2.1 W/m^2). In addition, there was a data logger and a computer (80 W, 1.2 W/m^2), and a fridge (30 W, 0.4 W/m^2) which were always ON.

5.2.2. Measurements and measuring equipment

A number of physical parameters, energy use and production were measured and recorded. The temperatures (air and globe) were measured at heights of 0.1 m, 0.6 m, 1.1 m, 1.7 m, 2.2 m, 2.7 m, 3.2 m and 3.7 m at a central location in the occupied zone following EN 13779:2007 [143]. The measurements above the occupied zone were taken to evaluate the effects of thermal stratification based on different heating and cooling strategies. The stratification is particularly important for this house because of its high and inclined ceiling. The thermal stratification from the floor to the ceiling was used as an indicator of the performance of the heating strategy regarding heat loss from the conditioned space to the outdoors.

Globe temperatures were measured with a gray globe sensor, 40 mm in diameter. This sensor has the same relative influence of air- and mean radiant temperature as on a person [144] and, thus, at 0.6 m and 1.1 m heights will represent the operative temperature of a sedentary or a standing person, respectively. The air temperature sensor was shielded by a metal cylinder to avoid heat exchange by radiation. Both the globe and air temperature sensors have ±0.3°C accuracy in the measurement range of 10-40°C [145]. A portable data logger logged the output from the sensors. Fig. 7 shows a panoramic view of the interior of the house, the measurement location and the sensors used for air and globe temperature measurements.

The energy use of the air-to-brine heat pump, mixing station, and the controller of the radiant system were measured with wattmeters. The energy use of the AHU and energy production of the house (from the PV/T panels) were measured through a branch circuit power meter (BCPM). The wattmeters that were used to measure the energy use of the mixing station and the controller of the radiant system had an accuracy of ±2% ±2 W. The wattmeter that was used to measure the energy use of the air-to-brine heat pump had an accuracy of 3%. The BCPM's accuracy was 3% of the reading. A full specification of the parameters measured and the measuring equipment can be found in [146].

5.3. Experimental settings

5.3.1. Heating season

Different heating strategies were compared during the heating season. In the beginning of the heating season, floor heating was controlled according to different operative temperature set-points (without any ventilation). Afterwards, floor heating was supplemented by warm-air heating from the ventilation system, and during the last part of the heating season, the ventilation system was only used to provide fresh air (with passive heat recovery from the exhaust air) while floor heating was providing the required space heating. The design ventilation rate was 0.5 ACH.

Table 10 summarizes the most important boundary conditions for these strategies in the heating season (FH: Floor heating, HR: Heat recovery, HRPH: Heat recovery and pre-heating, corresponding to warm-air heating). The numbers in the abbreviations are related to the indoor temperature set-points.

Table 10. Periods and experimental settings of the different cases, heating season.

Period	*Average outdoor air temperature [°C]*	*Floor heating set-point [°C]*	*Ventilation*	*Case abbreviation*
26th of Sep to 21st of Nov	8.2	22	Off	FH22
21st of Nov to 18th of Dec	4.0	20	Off	FH20
18th of Dec to 16th of Jan	4.6	21	Off	FH21
16th of Jan to 10th of Feb	0.0	21	On, heat recovery and pre-heating [b]	FH21-HRPH
10th of Feb to 10th of Mar	5.0	20	On, heat recovery and pre-heating [b]	FH20-HRPH
10th of Mar to 3rd of Apr	5.5	21	On, heat recovery	FH21-HR
3rd of Apr to 1st of May [a]	9.0	20	On, heat recovery	FH20-HR

[a] The dummies simulating the occupants and a dummy (equipment #2) were OFF during this experimental period.

[b] Heat recovery refers to the passive heat recovery and pre-heating refers to the active heat recovery (warm-air heating) in AHU. The supply air temperature was between 30 to 34°C, except for the periods with low outdoor air temperatures when it dropped to 27°C.

5.3.2. Cooling season

The HVAC system was operated similarly during the cooling season. The house was cooled by floor cooling and was ventilated with the mechanical ventilation system (only passive heat recovery). Different operative temperature set-points and different ventilation rates were tested. Internal solar shading covering 20 m^2 (manually operated) was installed on the North façade on 30th of July 2014 and it was used in the fully down position until the end of the experiments.

Table 11 summarizes the most important boundary conditions for the strategies used in the cooling season (FH: Floor heating, CS: Cooling season, FC: Floor cooling, HV: Higher ventilation rate, S: Solar shading).

Table 11. Periods and experimental settings of the different cases, cooling season.

Period	*Average outdoor air temperature [°C]*	*Floor cooling set-point [°C]*	*Ventilation rate [ACH]*	*Solar shading*	*Case abbreviation*
1st of May to 27th of May [a]	14.7	20 [b]	0.5	No	FH20-CS
27th of May to 19th of June	18.7	25	0.5	No	FC25
19th of June to 13th of July	18.7	25	0.8	No	FC25-HV
13th of July to 30th of July	22.7	24	0.8	No	FC24-HV
30th of July to 21st of Aug	18.1	24	0.8	Yes	FC24-HV-S
21st of Aug to 1st of Oct	16.0	24	0.5	Yes	FC24-S

[a] The dummies simulating the occupants and a dummy (equipment #2) were OFF during this experimental period.
[b] Floor system was in heating mode, transition period.

5.4. Results and discussion

5.4.1. Heating season

The performance of different heating strategies was evaluated based on the indoor environment category achieved according to EN 15251:2007 [20], and the measured temperature stratification. The categories are given according to EN 15251:2007 [20] for sedentary activity (1.2 met) and clothing of 1.0 clo. Table 12 shows the indoor environment categories achieved with different heating strategies.

Table 12. The category of indoor environment based on operative temperature at 0.6 m height, heating season.

Indoor environment category/case	*FH22*	*FH20*	*FH21*	*FH21-HRPH*	*FH20-HRPH*	*FH21-HR*	*FH20-HR*	*Total, average*
Category 1 (21.0-25.0°C)	92%	2%	37%	22%	11%	67%	35%	45%
Category 2 (20.0-25.0°C)	97%	44%	92%	72%	61%	98%	77%	80%
Category 3 (18.0-25.0°C)	100%	95%	100%	93%	99%	100%	100%	98%
Category 4 [a]	0%	5%	0%	7%	1%	0%	0%	2%

[a] Category 4 represents the values outside Categories 1, 2, and 3.

Fig. 13 shows the operative temperature at 0.6 m height and the outdoor air temperature during the heating season.

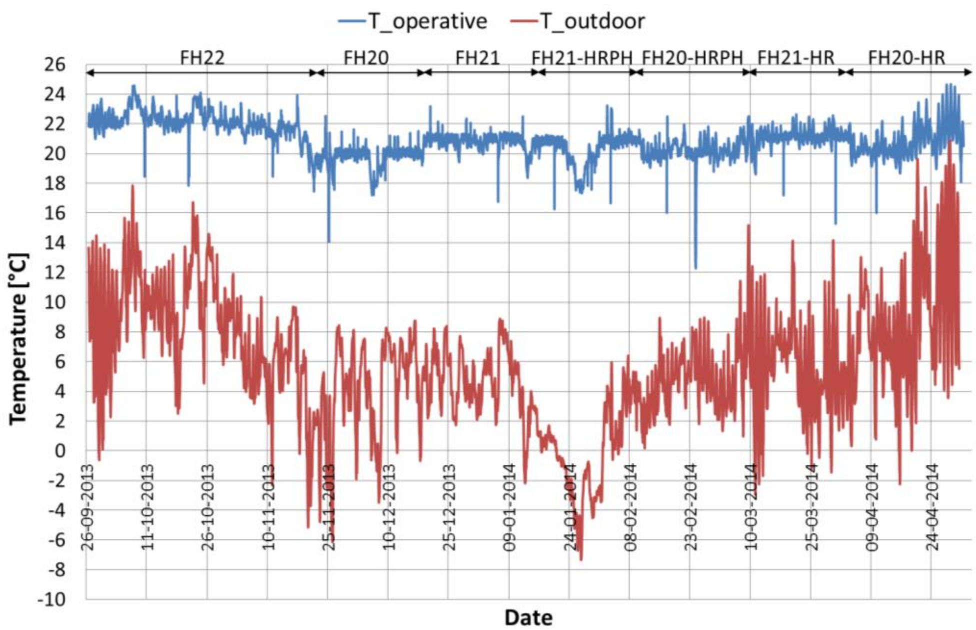


Fig. 13. Operative temperature and outdoor air temperature during the heating season.

The results presented in Table 12 and Fig. 13 show that even though different heating strategies were used, the overall performance regarding the indoor environment was satisfactory, i.e. 80% of the time in Category 2 according to EN 15251:2007 [20]. It may also be seen that there were periods when the indoor environment was outside Category 3: for 2% of the time it was in Category 4.

It was possible to keep the indoor operative temperature close to the set-point, although the systems struggled to achieve this when the outdoor temperatures were below -5°C (for 2% of the time only Category 4 was achieved). In addition to the increased heating demand, one possible explanation for this is that the lower outdoor air temperatures affected both the air-to-brine heat pump and the AHU.

The operative temperature set-point of 20°C was too low. This is because even though the ventilation system would be heating the indoor space, the floor heating system did not start the water circulation in the loops until the operative temperature had dropped below 20°C. This resulted in several periods with room temperatures below 20°C.

Thermal stratification is an inevitable physical phenomenon, particularly in a high-ceilinged space, and it can be used to analyze the indoor environment created by different heating strategies. Thermal stratification is important for occupant thermal comfort (due to local thermal discomfort) and for heat loss from the building. A high temperature gradient will increase the energy use and local thermal discomfort [11]. In Table 13, average air temperatures at selected heights are given based on the heating strategy.

Table 13. Time-averaged air temperature at the selected heights and the difference between highest and lowest measurement points, heating season.

Height/case	*FH22*	*FH20*	*FH21*	*FH21-HRPH*	*FH20-HRPH*	*FH21-HR*	*FH20-HR*	*Total, average*
0.1 m [°C]	21.7	19.2	20.3	19.5	19.5	20.8	20.4	20.4
1.7 m [°C]	22.3	19.7	20.7	20.7	20.7	21.2	20.9	21.1
2.2 m [°C]	22.3	19.6	20.6	20.9	21.0	21.1	20.8	21.1
3.7 m [°C]	22.6	20.0	21.0	22.3	22.3	21.5	21.2	21.7
Temperature difference between 3.7 m and 0.1 m [°C]	0.9	0.8	0.8	2.8	2.8	0.7	0.8	1.3

Fig. 14 shows the air temperature differences between the selected heights for the heating season.

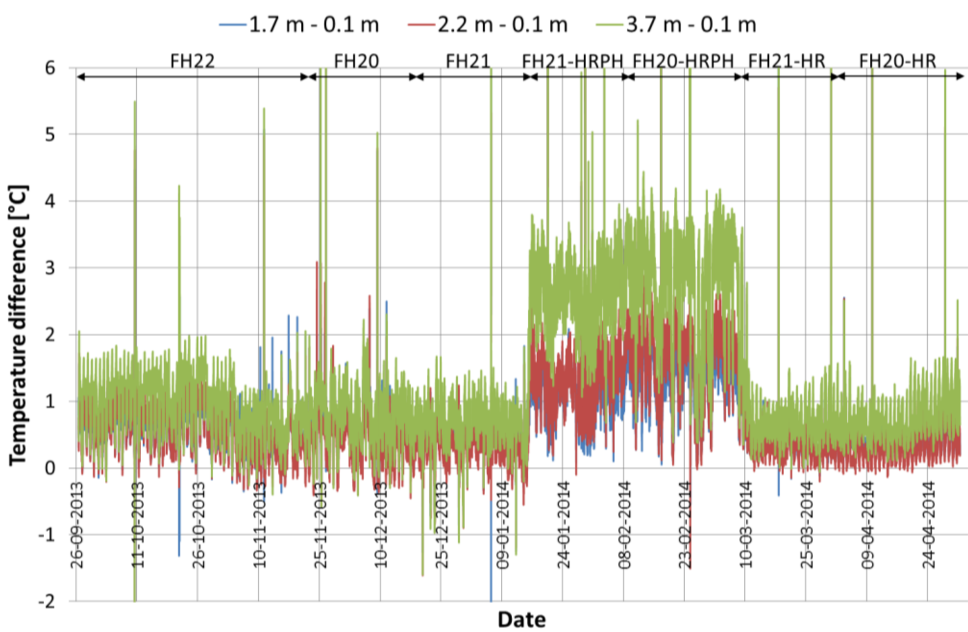


Fig. 14. Air temperature difference between the selected heights, heating season.

The results shown in Table 13 and Fig. 14 indicate that the thermal stratification inside the house was greatest when the floor heating was supplemented by warm-air heating from the ventilation system. On average, the temperature difference between the highest (3.7 m) and lowest (0.1 m) measurement points was 2.8°C when the floor heating was supplemented by warm-air heating while in other cases it was between 0.7°C and 0.9°C. Fig. 14 shows a clear increase in the temperature difference (thermal stratification) between the highest and lowest points when the floor heating was supplemented by warm-air heating, and the temperature difference reached almost 4°C in some periods.

Because of the lower density of the warm supply air compared to the room air, the supply air tends to flow along the ceiling and not to mix well with the room air. Due to this phenomenon and the thermal stratification, in the cases where the floor heating was supplemented by warm-air heating, the space above the occupied zone was being heated. This increases the heat loss from the indoor space, in particular, through glass façades with higher U-values compared to the external walls. Increased thermal stratification is a phenomenon to avoid (unless an underfloor air distribution or a displacement ventilation system is used); especially in a house with a high and tilted ceiling, as in this house, so it is recommended to use a radiant floor heating system in spaces with high ceilings.

5.4.2. Cooling season

The performance of different cooling strategies was evaluated based on the indoor environment categories given in EN 15251:2007 [20] for sedentary activity (1.2 met) and clothing of 0.5 clo. In addition, the hours above 26°C and 27°C were calculated following DS 469:2013 [12]. According to DS 469:2013 [12], 26°C should not be exceeded for longer than 100 hours during the occupied period and 27°C should not be exceeded for longer than 25 hours. Even though these specifications are given for offices, meeting rooms, and shops, it is considered applicable also for residential buildings, and according to DS 469:2013 [12], mechanical cooling would normally not be installed in residential buildings in Denmark.

Table 14 shows the indoor environment categories achieved, and the hours above 26°C and 27°C as a function of the cooling strategy, and Fig. 15 shows the operative temperature and outdoor air temperature during the cooling season.

Table 14. The category of indoor environment based on operative temperature at 0.6 m height, cooling season.

Indoor environment category/case	*FH20-CS*	*FC25*	*FC25-HV*	*FC24-HV*	*FC24-HV-S*	*FC24-S*	*Total, average*
Category 1 (23.5-25.5°C)	52%	56%	36%	54%	39%	22%	41%
Category 2 (23.0-26.0°C)	73%	72%	49%	72%	58%	36%	57%
Category 3 (22.0-27.0°C)	87%	87%	75%	91%	84%	72%	81%
Category 4	13%	13%	25%	9%	16%	28%	19%
Hours above 26°C	48	129	79	87	7	0	350 [a]
Hours above 27°C	19	71	38	34	0	0	162 [a]

[a] Although the overheating hours cannot be added directly for the different cooling strategies, their total is given to indicate the duration of overheating during the cooling season.

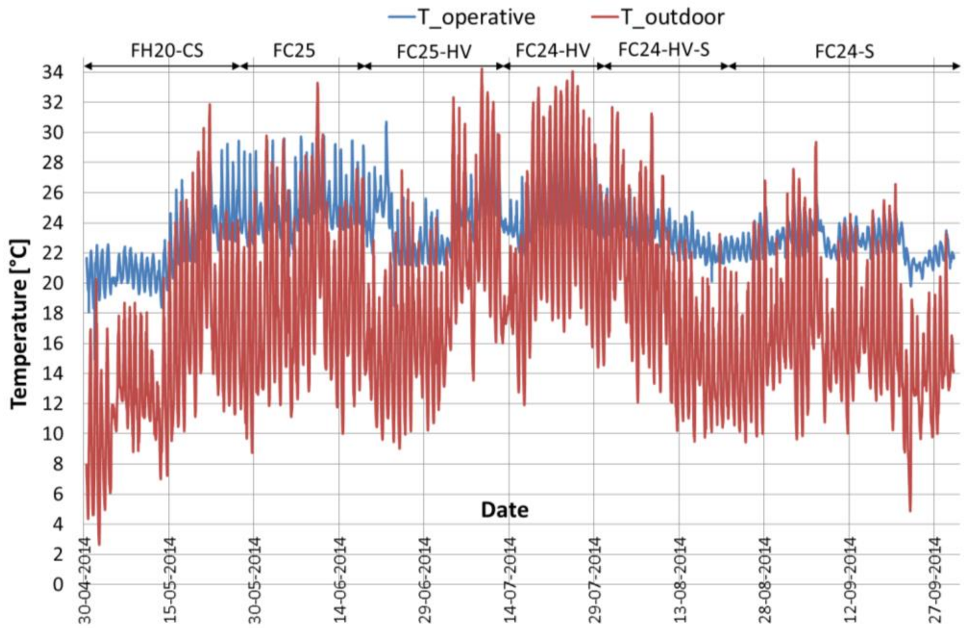


Fig. 15. Operative temperature and outdoor air temperature during the cooling season.

The house performed worse in the cooling season than in the heating season in terms of providing a satisfactory thermal indoor environment; for 57% of the time the operative temperature was in Category 2 and for 19% of the time it was outside the recommended categories in EN 15251:2007 [20]. This occurred mainly in the transition periods (i.e. May and September) and due to overheating, which was a problem during the cooling season, except in August and September. The hours above 26°C and 27°C exceeded the values recommended in DS 469:2013 [12]. Decreasing the operative temperature set-point and increasing the ventilation rate helped to address the increased cooling load, but with an increased energy use. This is mainly due to the longer operation of the floor cooling and to increased cooling effect of the supply air.

The results show that even though the floor system was in heating mode during most of May, which is a part of the transition period, floor cooling could have been activated in the second half of May, which would have reduced the hours above 26°C and 27°C, and improved the indoor environment.

The main reasons of overheating were the large glazing façades including the lack of solar shading, the orientation of the house and the lack of thermal mass to buffer sudden thermal loads. In the current location of the house, direct solar radiation from the South façade was not a problem, because of the orientation and longer overhang on the South façade. Most of the overheating hours were in the late afternoon (i.e. from 18:00 until sunset), when there was direct solar gain through the North façade.

In this study, operative temperature was used as an indicator of the thermal indoor environment, and vertical air temperature difference between head and ankles was used as an indicator for local thermal discomfort, but human thermal comfort is also affected by other factors such as floor surface temperature, radiant temperature asymmetry and draught [31]. All of these factors would have to be considered before any definitive conclusion on occupant thermal comfort can be drawn.

Vertical air temperature difference between head and ankle levels (for sedentary occupants, 1.1 m and 0.1 m above the floor, respectively) was also measured in heating and cooling seasons and the results are reported in Paper 9, together with the thermal stratification during the cooling season.

5.4.3. Energy performance

The energy performance of the house was evaluated by measuring the energy produced by the PV/T panels on the roof and the energy used by the HVAC system. Table 15 shows the monthly electricity production from the PV/T panels.

Table 15. Monthly electricity production from the PV/T panels (month/year) [a] [kWh].

10/13	*11/13*	*12/13*	*01/14*	*02/14*	*03/14*	*04/14*	*05/14*	*06/14*	*07/14*	*08/14*	*09/14*	*Sum*
214.6	143.0	61.9	59.7	168.8	310.2	507.3	621.1	665.1	519.7	405.0	308.1	4043.9

[a] Electricity production from the PV/T panels between 26th to 30th of September 2013 was 59.3 kWh.

The annual electricity production from the PV/T panels (4044 kWh) was lower than had been predicted by the simulations, 7434 kWh. This difference could have occurred for several reasons: it was observed during the competition period in Solar Decathlon Europe 2012 (17th to 28th of September 2012) that the output of the PV/T panels was lower than the expected values [146], the climate could have differed from the weather files used in simulations and the location was different. In addition to these factors, some of the PV/T panels were damaged during disassembly/assembly and transportation of the house, and finally, some trees around the house threw shadows on the PV/T panels.

The HVAC system's energy use consisted of the air-to-brine heat pump, mixing station, controller of the radiant system, and AHU. During the heating season, the set-point of the heat pump was 35°C until 21st of November 2013 and after this date, it was 40°C. The set-point of the heat pump was changed to 15°C on 27th of May 2014. Table 16 shows the energy use of the components for each heating and cooling strategy.

Heating degree days (HDD) and cooling degree days (CDD) were calculated for each case using a base temperature of 17°C and 23°C, respectively. Fig. 16 shows the total average energy use of the cases together with the calculated degree days following the methodology described by Quayle and Diaz [147].

Table 16. Energy use of the HVAC system components.

Case	*Heating degree days*	*Cooling degree days*	*Heat pump [kWh]*	*Mixing station [kWh]*	*Controller, radiant system [kWh]*	*AHU [kWh]*	*Total [kWh]*	*Total, average [kWh/day]*
FH22	496	-	518.9	15.5	5.3	0.0	539.7	9.5
FH20	350	-	438.3	11.3	3.8	0.0	453.4	16.8
FH21	361	-	460.6	15.0	5.0	0.0	480.6	16.6
FH21-HRPH	425	-	463.2	10.4	3.5	236.6	713.7	28.5
FH20-HRPH	337	-	307.2	2.2	0.8	221.2	531.4	18.9
FH21-HR	275	-	321.8	7.2	2.5	39.3	370.7	15.4
FH20-HR	220	-	304.6	4.3	1.5	47.5	358.0	12.8
FH20-CS	97	-	206.4	1.3	0.4	42.6	250.7	9.3
FC25	-	15	95.4	5.4	1.8	36.0	138.7	6.0
FC25-HV	-	20	110.1	3.2	1.4	75.2	189.8	9.0
FC24-HV	-	36	114.8	6.7	2.0	60.9	184.3	10.8
FC24-HV-S	-	12	105.6	3.8	1.2	78.8	189.3	8.6
FC24-S	-	6	145.9	0.6	0.2	65.7	212.4	5.3
Total	2561	89	3592.9	86.7	29.3	903.8	4612.7	-

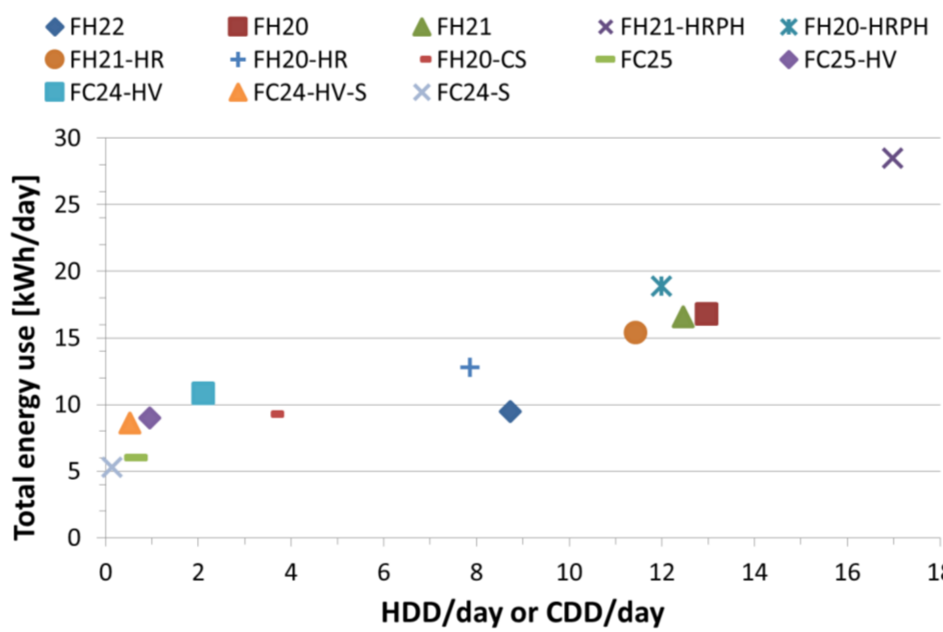


Fig. 16. Energy use per day versus heating or cooling degree days per day.

In Table 16, the heat pump energy use includes the heat pump cycle energy use, an integrated pump and the heat pump control system. The mixing station's energy use includes the circulation pump of the radiant floor heating system, a motorized mixing valve and the control unit. The energy use of the controller of the radiant system includes a control unit and an actuator at the manifold for each of the four loops. The AHU's energy use includes fans, control system, by-pass damper, internal heat pump cycle (active heat recovery), and its related equipment.

The results show that the energy use increased markedly when the warm-air heating (FH21-HRPH and FH20-HRPH) was in operation, and these strategies struggled to provide the intended thermal indoor environment despite the increased energy use. The energy use during the cases FH20 and FH21 were close to each other, but a better thermal indoor environment was achieved with FH21. The last two cases in the heating season, FH21-HR and FH20-HR, have lower energy use and achieved a better thermal indoor environment compared to the cases with the same set-points without ventilation (FH21 and FH20). The FH22 strategy had the lowest energy use (although it had the highest operative temperature set-point) and the most satisfactory thermal indoor environment, although this was partly due to the relatively high outdoor air temperatures during this period.

During the cooling season, the increased ventilation rate and lowered operative temperature set-point increased the energy use. This was expected, due to higher power input to the fans in the AHU and longer operation time of the pump in the floor cooling system. The increased energy use contributes to a better thermal indoor environment, but other strategies should be employed to reduce the cooling demand by means of energy efficient measures (e.g. lower ventilation rates when the house is unoccupied, natural ventilation when the outdoor conditions are suitable, decreased glazing area, solar shading, a better orientation of the house and so forth).

It would be possible to increase the energy performance of the house by means of simple modifications, e.g. 29% (1051 kWh) of the heat pump's energy use was due to the brine pump that is integrated into the heat pump and it was always ON due to an internal control algorithm. This energy use could have been reduced by synchronizing the circulation pump of the radiant system with the brine pump.

Table 17 summarizes the overall energy performance of the house during the measurement period. In the overall energy balance, positive values indicate energy surplus while negative values indicate energy deficit, i.e. the house used more energy than it produced on an annual basis.

Table 17. Overall energy balance of the house (yearly values).

Heat pump [kWh/m^2]	54.3
Mixing station [kWh/m^2]	1.3
Controller, radiant system [kWh/m^2]	0.4
AHU [kWh/m^2]	13.7
Energy use of the HVAC system, total [kWh] and per floor area [kWh/m^2]	4613 / 69.7
Electricity produced by the PV/T panels [kWh]	4044
Overall energy balance (production-use) [kWh]	-569

5.4.4. *Operation and performance*

The results from the measurement period show that the control of heating and cooling system through set-points and the interaction of the radiant floor heating and cooling system with the ventilation system have considerable effects on the thermal indoor environment and on the energy performance. The optimal system combination is when the floor heating and cooling system is emitting or removing the necessary heat and the ventilation system is used only to provide the necessary supply of fresh air without actively heating or cooling the intake air. This would simplify the system operation since only one system will heat or cool the indoor space, and lower airflow rates can be used since the only task of the ventilation system will be to provide fresh air.

The heating and cooling set-points should be carefully selected to avoid periods with too low or too high indoor temperatures. This requires choosing set-points so that the radiant floor heating and cooling system has enough time to provide the necessary indoor conditions. It should also be noted that a higher indoor temperature set-point in the heating season would result in higher energy use because of the longer operation time of the heating system, and because of the increased operating temperatures in the heating system which affect the heat pump performance. A similar effect will be observed in the cooling season with a lower indoor temperature set-point.

The fresh air intake for the AHU was from the crawl-space and this proved to be a beneficial approach: the air temperature in the crawl-space was higher than the outdoor air temperature during winter and lower than the outdoor air temperature during summer, so it buffered the variations in the outdoor air temperature to a certain extent.

Throughout the 12-month operation of the house, the heating and cooling systems were active with respective set-points also during the transition periods (i.e. May and September) but it is not practical to provide constant heating or cooling during the transition periods, therefore the heating and cooling system operation and the switchover between these modes require careful consideration. Operation of the systems needs to be improved to avoid unnecessary heating and cooling in the transition periods.

A recent study analyzed the horizontal temperature distribution in the present house and found that the radiant floor cooling created a uniform thermal indoor environment in the cooling season, despite the large glazing façades [148].

Based on the measurement results, a reversed orientation of the house, i.e. the façade with the longer overhangs towards the North, would have been more energy effective, as it would decrease the solar heat gain during the cooling season and increase the solar heat gain during the heating season. The results also show that the house would have benefited from a higher thermal mass to buffer the sudden thermal loads, especially during the periods in cooling season when there was direct solar gain and during the transition periods.

During the heating season, when the outdoor temperatures were below -5°C, the heating system of the house struggled to reach the operative temperature set-points. An evident reason for this was the increased heating demand. The air-to-brine heat pump and the AHU were also affected by the lower outdoor air temperatures. The initial design of the heating and cooling system incorporated a ground heat exchanger to obtain free cooling during summer and a coupled heat pump for the heating season, as the heat sink and source of the house [146]. This would have been beneficial, since the performance of a ground-coupled heat pump would not have been affected significantly by the varying outdoor air temperatures.

The house was initially designed as a plus-energy house; however, the measurement results show that it did not produce any energy surplus under the climate conditions in Denmark (Table 17). This is mainly due to the lower electrical output from the PV/T panels (4044 kWh) than estimated in the design conditions (7434 kWh), high heating and cooling energy use (4613 kWh), changes in the house's heating and cooling systems resulting in higher energy use than the design values, and differences in the simulations and in the actual components of the house.

The results of this study were not influenced by the users since there were no users and the heat gains were controlled. Therefore, the results are only related with the design and operation of the house and its systems.

During the experiments, it was not possible to make changes in the building envelope (except for the installation of the internal solar shading), so the modifications were on the heating, cooling and ventilation system operation. The design of the current building diverged from the ideal design of an energy efficient nZEB where the heating and cooling demands of the house would have been minimized in the design phase. The results of this study show that it is not enough and it is not energy efficient to address the heating and cooling loads through adjusting set-points and adjusting the flow rates of air and water.

Although the results reported in this study are for a particular house, the results regarding the systems and their operation have implications on a broader scale. In order to achieve high energy performance together with a comfortable thermal indoor environment, the initial step is to reduce the heating and cooling demands of the house as much as possible during the design, and this should be done in a holistic way considering orientation, shading, thermal mass, air-tightness, thermal bridges, etc. simultaneously. This is crucial since the demand determines the final energy use. After it is assured that the heating and cooling demands are minimized, the resulting demand should be addressed with the most energy efficient and environmentally friendly heating and cooling strategies.

The radiant low temperature heating and high temperature cooling systems enable the integration of natural heat sources and sinks (ground, night-time radiative cooling, solar, sea-water, etc.) into the heating and cooling systems and they also provide a draught-free and uniform thermal indoor environment, therefore they are a promising means of achieving high energy performance without sacrificing occupant thermal comfort.

Once it is ensured that the overall energy demand of the building (including plug loads) is minimized, the most energy efficient heating and cooling systems are chosen, and the control of these systems are optimized, then it would be possible to reach plus-energy targets even with a minimal contribution from the energy producing components in future buildings.

5.4.5. Performance improvement based on parametric analysis

The measurement results show that the house's performance could be improved in several ways. Therefore, the effects of different building and HVAC system parameters on the energy use and on the thermal indoor environment were parametrically studied by means of dynamic building simulations (IDA ICE) to propose improvement suggestions. The results are summarized in the following and reported in detail in Paper 8.

The analyzed parameters were window area, infiltration, thermal bridges, orientation of the house, exterior solar shading, natural ventilation, thermal mass, and increased radiant system area. The simulation model was validated with measurements from the house.

The improvements with the highest influence on both thermal indoor environment and on energy use were reducing the window area, reducing the infiltration and minimizing the thermal bridges. An optimized version of the house was proposed based on the simulations. The improved house model has 25% less window area, and an improved building envelope.

The improved house model performed considerably better than the reference case in terms of thermal indoor environment and energy performance. The duration in thermal indoor environment Category 1 of EN 15251:2007 [20] increased from 71% to 97% compared to the reference case, and operative temperatures were always within Category 2 of EN 15251:2007 [20]. Furthermore, the operative temperature never exceeded 26°C, indicating that there was no overheating. The annual energy use decreased by 68% compared to the reference case (both results are based on IDA ICE simulations).

These results show that it would have been possible to lower the energy use while improving the thermal indoor environment through careful analyses in the design phase.

5.5. Summary

A detached, one-story, single-family house designed for plus-energy performance was operated for one year. During this period, different heating and cooling strategies were compared and the energy performance of the house and its thermal indoor environment were monitored. The main conclusions are as follows:

- During the heating season, it was possible to provide the intended operative temperature in the occupied zone except for the periods when the outdoor air temperatures were below -5°C;
- Radiant floor heating combined with heat recovery from ventilation was the optimal heating strategy as it provided a uniform temperature distribution within the space and decreased the heat losses due to thermal stratification;
- The performance of the house in terms of maintaining a comfortable thermal indoor environment was worse in the cooling season than in the heating season. Overheating was a significant problem, and the main reasons for this were the large glazing façades, the orientation of the house, the lack of solar shading, and the lack of sufficient thermal mass to buffer the sudden thermal loads;

- The operation of the heating and cooling system during the transition periods was problematic and this affected the thermal indoor environment and energy performance negatively;
- The house had a high heating and cooling demand that could easily have been reduced at the design phase. Although, it might be possible to address the excessive heating and cooling loads by adjusting set-points, water and airflow rates, these would result in increased energy use, as in the present study. It is crucial to minimize the demand before attempting to satisfy it in the most energy efficient way;
- Although the house was designed as a plus-energy house, it did not perform as one under the climatic conditions of Denmark. This was mainly due to the electrical output from the PV/T panels being lower than had been assumed in the simulations, but also to the unnecessarily high energy use of the house and the fact that the heating and cooling system differed from that of the initial design. It would be possible to increase the energy performance of the house by making simple modifications to the operating strategies and to the architectural design of the house. It would then be possible to achieve plus-energy levels with the same active (energy producing) components.

6. Overall conclusion

The detailed conclusions from theoretical, numerical, and experimental analyses are given in respective chapters and in publications. The following is the overall conclusion of the thesis.

A single-family house was used as a case study for the evaluations in this thesis; nevertheless, the results and developed calculation methodologies can be applied also to other buildings.

Heating and cooling demands of a building should be minimized in the design phase since these demands determine the heating and cooling systems (terminal units, heat sources and sinks) that could be used to address these demands. Once the overall energy demand of the building is minimized, plus-energy targets can be reached with minimal contribution from energy producing components such as photovoltaic/thermal panels.

After the heating and cooling demand of the building is minimized, the most energy efficient and environmentally friendly heating and cooling systems should be chosen. For space heating and cooling purposes, low temperature heating and high temperature cooling systems proved to be superior to other systems, evaluated with different system analysis tools; energy, exergy, and entransy.

A radiant hydronic floor heating and cooling system was used in this study to compare the performance of a low temperature heating and high temperature cooling system to other systems. In other applications, depending on factors such as geographical location, building type, dominant loads (heating or cooling) and so forth, the chosen system might be different (e.g. TABS); however, low temperature heating and high temperature cooling approach should still be employed.

The ultimate goal in heating and cooling of buildings should be to decrease the temperature difference between different components from indoor terminal units to the heating and cooling plants (i.e. to decrease the overall temperature difference in the system, enabling the utilization of non-fossil resources such as ground or solar energy, and resulting in improved performance of heat pumps and chillers) and radiant surface heating and cooling systems allow precisely this. A certain analysis method or any combination of methods that helps achieving this goal will be useful for improving building heating and cooling systems during the design and operation phases.

Heating and cooling systems should be considered as a whole, focusing on the overall system performance. Radiant systems should be coupled to appropriate heating and cooling plants, and the energy inputs to auxiliary components (pumps, fans, etc.) should be minimized. Radiant systems could be coupled to naturally available renewable heat sources and sinks (e.g. ground), and this would result in considerable energy and resource conservation. Hydronic heating and cooling systems require considerably less auxiliary energy compared to air-based systems. Exergy analysis can be used to optimize a system holistically where different quality energy forms, such as electricity and heat, are used.

In application, control of the radiant system and its interaction with the ventilation system are important for an optimized operation. The operation of the heating and cooling system during the transition periods requires careful considerations. Measurements, simulations, and calculations proved that a system in which the radiant system heats or cools the space and the ventilation system only provides the required amount of fresh air for indoor air quality concerns is the optimal solution. Application of radiant floor heating is particularly beneficial in spaces with high ceilings, as it can provide a uniform temperature distribution within the space and decrease heat losses due to thermal stratification.

Different analysis tools showed that to obtain the most rational use of resources, energy analysis alone is not sufficient. It is not only enough to consider the quantity of energy; the temperatures and temperature differences within a system should also be considered.

Dynamic calculations are widely available for energy analysis, and it is necessary to develop dynamic calculation methods for exergy and entransy analyses to obtain a complete view of the system operation during longer periods and under different boundary conditions.

The tools (energy, exergy, and entransy) to achieve global energy and resource efficiency in buildings are available; however, there is a need to make these tools more applicable and somewhat more user-friendly so that their use can be widespread and a broader group of designers, researchers, and engineers could use and benefit from them. Some of the possible ways to reach a broader application are to include these methods in building codes and standards (e.g. exergy-based performance evaluation instead of primary energy or CO_2 emissions), develop universal and standardized calculation procedures, and to develop a simulation software, or add-ons to existing building simulation software, that can dynamically calculate system performance based on different tools.

7. Future investigations

The following issues are worthwhile to investigate in the future:

- Total or separate treatment (radiation and convection) of heat transfer to and from the radiant surfaces, and its effects on load calculations, dimensioning of systems, and on system performance;
- Effects of heat (convective and radiation fraction) and moisture sources on the performance of terminal units;
- Applicability of radiant heating and cooling systems in renovation of buildings;
- Optimization of the heating and cooling system operation during transition periods. Can enough thermal mass let the house run without active heating or cooling during transition periods and how to evaluate thermal comfort during these periods?
- Benefits of using a predictive control algorithm to control the heating and cooling systems of the house, especially during the transition periods;
- Further studies to determine the exact benefits and limitations of the entransy method;
- Dynamic exergy and entransy calculation methods;
- Development of universal and standardized calculation procedures for different performance evaluation methods for achieving a widespread use.

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Appendix A – Terminal unit table

Table A.1. Chosen terminal units and their corresponding possibilities, limitations and operational characteristics (Y: Yes, N: No, NC: Not Common).

		Possibilities				***Method of heat emission or removal***				***Capacity (W/m²)***		***Medium of energy distribution***		
Name of terminal	**Type**	Heating	Cooling	Ventilation (fresh air)	Humidification + Dehumidification	Convection	Mainly convection	Radiation	Mainly radiation	Heating	Cooling	Air	Water	Electricity
Radiant systems	Floor [a]	Y	Y	N	N	Y	N	Y	Y	99	42	Y	Y	Y
	Wall	Y	Y	N	N	Y	N	Y	Y	160	72	N	Y	Y
	Ceiling	Y	Y	N	N	Y	N	Y	Y	42	99	Y	Y	Y
Air systems [b]	Mixing ventilation	Y	Y	Y	Y	N	Y	N	N	34°C	14°C	Y	N	N
	Displacement ventilation	Y/N	Y	Y	Y	N	Y	N	N	NC	18°C	Y	N	N
	Personalized ventilation	Y/N	Y	Y	Y	N	Y	N	N	NC	20°C	Y	N	N
Beams	Passive	Y	Y	N	N	N	Y	N	N	NC	80	N	Y	N
	Active	Y	Y	Y	Y [c]	N	Y	N	N	50	120	Y	Y	N

[a] Floor in the occupied zone.

[b] For air systems, typical maximum and minimum supply air temperatures are provided [10]. The heating and cooling capacity will depend on the ventilation rate.

[c] Humidification and dehumidification is possible with the primary air and it should be done by the air-handling unit. Beams operate in dry, non-condensing conditions.

Note: The given values are only intended to provide guidance. The indicated capacities could vary depending on the application.

Appendix B – Published, accepted, and submitted papers

Paper 1: Kazanci, O. B., & Olesen, B. W. (2015). IEA EBC Annex 59 - Possibilities, limitations and capacities of indoor terminal units. Energy Procedia - 6th International Building Physics Conference, IBPC 2015, 78, 2427–2432.

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IEA EBC Annex 59 - Possibilities, limitations and capacities of indoor terminal units

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Abstract

Indoor terminal units can be defined as the building elements that use different heat transfer mechanisms and media to emit and remove heat or moisture from indoor spaces (e.g. hydronic radiant heating and cooling systems, fan-coil units, active beams). Indoor temperature and humidity fields depend on the chosen terminal units.

Terminal units differ in their capabilities of addressing sensible and latent loads, methods of heat emission or removal, maximum heating and cooling capacities, medium of energy distribution, and local or total volume conditioning.

In the present study, operation characteristics, possibilities and limitations of different terminal units were specified. Considered terminal units were radiant heating and cooling systems, all-air systems (mixing, displacement, and personalized ventilation), passive and active beams. The results were summarized in a table, which aims at providing a reference for terminal unit selection during the design phases of HVAC systems.

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Keywords: Indoor terminal unit; temperature and humidity field; radiant heating and cooling; ventilation; passive and active beams

1. Introduction

Indoor terminal units are active building components that emit or remove heat and moisture to indoor spaces. These indoor terminals mainly rely on convection (natural or forced), radiation or both. A recent research project from International Energy Agency (IEA), Energy in Buildings and Communities (EBC) Program [1], Annex 59 –

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High Temperature Cooling and Low Temperature Heating in Buildings [2] is studying the currently existing terminal units. A sub-group within the project, Subtask B – Indoor temperature/humidity field and terminal units, is aiming to summarize the indoor heat and moisture sources and the current methods to address them. A further goal of the project is to provide improvement suggestions to the currently existing terminal units and HVAC systems.

This paper summarizes the characteristics of the chosen terminal units (radiant systems, mixing, displacement, and personalized ventilation, passive and active beams) and it aims to function as a simple and reliable reference tool for these units. Possibilities (heating, cooling, ventilation, humidification and dehumidification of indoor air), method (heat transfer mechanism) of heat emission and removal from the indoor space, heating and cooling capacities, and the medium of energy distribution are specified for different terminal units. It is assumed that the reader is familiar with the described systems and concepts. The given values are only intended to provide guidance, and the indicated capacities could vary depending on the application.

2. Hydronic radiant heating and cooling systems

A hydronic (water-based) radiant heating and cooling system refers to a system where the water is used as the heat carrier (medium of energy distribution) and more than half of the heat exchange with the conditioned space is by radiation [3]. It is possible to divide the radiant heating and cooling systems into three: radiant heating and cooling panels, pipes isolated from the main building structure (radiant surface systems), and pipes embedded in the main building structure (Thermally Active Building Systems, TABS) [3]. The heat carrier (water) circulating in the pipes has low temperatures in the heating and high temperatures in the cooling operation. In some TABS constructions (hollow core concrete decks) also air has been used as a heat carrier, and also electricity can be used in heating applications.

Floor, wall and ceilings can be used as surfaces that provide heating or cooling to the space. Hydronic radiant surface systems require a ventilation system to address the latent loads and to provide the ventilation rates required for indoor air quality concerns [3]. Radiant heating and cooling systems enable lower air flow rates than all-air systems where the entire heating and cooling loads are addressed by the ventilation system [4].

The heat emission or removal from the space is achieved by a combination of radiation and convection. Total heat exchange coefficients (combined convection and radiation) for floor heating, wall heating and ceiling heating are 11, 8, 6 W/m^2K, and for floor cooling, wall cooling and ceiling cooling are 7, 8, and 11 W/m^2K, respectively [3]. The radiant heat transfer coefficient can be used as a constant value of 5.5 W/m^2K, with an error of less than 4% [5]. The difference in the total heat transfer coefficients stems from the natural convection. An overview of natural convection coefficients is given in [6].

Based on the acceptable surface temperatures (comfort and dew-point concerns [3]), and assuming an operative temperature of 20°C and 26°C in the room for heating and cooling cases, the maximum heating and cooling capacities can be obtained. The maximum floor (occupied zone) heating and cooling capacities are 99 W/m^2 and 42 W/m^2, wall heating and cooling capacities are 160 W/m^2 and 72 W/m^2, and ceiling heating and cooling capacities are 42 W/m^2 and 99 W/m^2, respectively. In the perimeter zones of the floor, it is possible to obtain a maximum heating capacity of 165 W/m^2 [3]. Different studies [3,7,8] have shown that the cooling capacity of a floor cooling system increases above the given maximum capacity of 42 W/m^2 and may even exceed 100 W/m^2, when there is direct solar radiation on the floor.

Different construction types of radiant systems can be found in [3]. The design, test methods, control and operation principles of radiant panels are given in ISO 18566:2013 [9], while the design, dimensioning, installation and control principles of embedded radiant systems are given in ISO 11855:2012 [10].

3. All-air systems

Currently, there are eight commonly applied ventilation strategies in buildings. These strategies are mixing ventilation, displacement ventilation, personalized ventilation, hybrid air distribution, stratum ventilation, protected occupied zone ventilation, local exhaust ventilation, and piston ventilation [11]. For the ventilation systems, the main method of heat emission and removal is convection and the medium of energy distribution is air. A review of different airflow distribution and ventilation systems in buildings can be found in [11]. Mixing, displacement, and personalized ventilation systems are further described in this paper.

3.1. Mixing ventilation

Mixing ventilation (mixing room air distribution) intends to dilute the polluted and warm (or cool) room air with clean, cooler (or warmer) supply air. The aim is to achieve a uniform temperature and contaminant distribution in the occupied zone [12]. It is possible to heat or cool a space by mixing ventilation. It is also possible to provide dehumidified and conditioned outdoor air (fresh air). Typical supply air temperature range for heating and cooling is up to 34°C and down to 14°C, respectively [11]. The obtained heating and cooling effect will depend on the ventilation rate. Also in some countries, such as Denmark, the highest permissible supply air temperature is limited to 35°C by regulations [13]. It is not recommended to have a higher temperature difference than 10°C between the supply and room air to achieve proper mixing [12]. According to [14] a specific cooling load of 90 W/m^2 can be handled with mixing ventilation systems.

3.2. Displacement ventilation

Displacement ventilation (displacement room air distribution) is based on displacing the polluted room air with fresh air (conditioned outdoor air) [11]. The cool fresh air is supplied with low velocity (0.25-0.35 m/s [15]) at or near the floor, and the supplied air rises by the effects of momentum and buoyancy forces [11,15]. It is possible to provide cold, dehumidified and conditioned outdoor air with displacement ventilation. Although, it could be possible to provide warmer air than the room air with displacement ventilation (e.g. to heat an unoccupied room before the occupancy [16]) it is not common and it is not recommended due to the short-circuiting of the supply air. Typically, the supply air temperature can be down to 18°C [11]. The cooling load that a floor current displacement system can handle is 30-35 W/m^2 according to [15] and 50 W/m^2 according to [14].

3.3. Personalized ventilation

Other than the two mainly applied total volume air distribution principles (mixing and displacement air distribution), another air distribution strategy is personalized ventilation, and it aims at supplying the clean and cool air close to an occupant before it is mixed with the room air [12,17]. The most important advantage of personalized ventilation compared to the total volume conditioning systems is its potential to provide clean, cool and dry air at inhalation [17,18]. According to [11], the supply air temperature can be down to 20°C in cooling and up to 28°C in heating mode, but it should be noted that perceived air quality (PAQ) might be a problem with the increased supply air temperature [19,20] and ventilation effectiveness may decrease depending on the chosen air supply location and terminal.

The required ventilation rates can be calculated based on EN 15251:2007 [21] (this standard is currently under revision [22]), CR 1752:1998 [23], and ASHRAE 62.1-2013 [24].

4. Beams

Although these systems are known as chilled beams, a recent guidebook [25] refers to them as beams, and this terminology will also be used in this paper. Beams (passive and active) are room air recirculation devices that can heat or cool (sensible) a space using water as the energy distribution medium. Active beams can also provide conditioned primary air to a space (they are coupled to the main air-handling unit) [25]. Fresh air is delivered to the space by a decoupled ventilation system in passive beam applications.

Beams cannot directly humidify or dehumidify the room air since they operate in dry (non-condensing) conditions but it is possible to control the latent loads and to address the ventilation requirements with active beams [25]. The method of heat emission and removal from the space takes place mainly by convection.

4.1. Passive beams

The performance of passive beams relies on natural convection [25]. In passive beams, the medium of energy distribution from the plant is water. It is possible to heat and a cool space with passive beams but it is not possible to provide fresh air to the space. Although heating is possible with passive beams, in most applications, passive beams are used for cooling only and therefore a separate heating system should be used [25]. Also the ventilation needs should be addressed by a complementing system (e.g. by an air-handling unit) [25]. It is recommended to use passive chilled beams when the total sensible cooling load is up to 40-80 W/m^2 [26].

4.2. Active beams

The performance of active beams relies on convection that is caused by induction [25]. It is possible to heat, cool and provide fresh air to a space by active beams. In active beams, the medium of energy distribution is both air (fresh air, from the air-handling unit) and water from the heating or cooling plant. Active beams can typically be used when the total sensible cooling (air and water) load is less than 120 W/m^2 in comfort conditions [25,26]. The optimum operating range (for a good thermal comfort in sedentary type occupancy) is 60-80 W/m^2 [26]. For the heating case, the optimum operating range is a heating load of 25-35 W/m^2 and a maximum heating load of 50 W/m^2 [26]. The specific heating and cooling capacities of beams can be found in [26] expressed in W/m.

The testing and rating procedures of passive and active chilled beams are given in EN 14518:2005 [27] and in EN 15116:2008 [28], respectively.

5. Other systems

Another type of terminal units that can be used for heating and cooling of buildings is a fan-coil unit. Information regarding fan-coil units can be found in [6,14,29].

The descriptions, characteristics, operation principles and other information regarding other terminal units that were not a part of this paper (radiators, radiant tubes, convectors, etc.) can be found in [14,29].

6. Discussion and conclusion

Possibilities, limitations and characteristics (heating and cooling capacities, governing heat transfer mechanisms and media of energy distribution) of the chosen terminal units are summarized in Table 1.

Table 1. The chosen terminal units and corresponding possibilities, limitations and characteristics (Y: Yes, N: No, NC: Not Common).

		Possibilities				Method of heat emission or removal				Capacity (W/m^2)		Medium of energy distribution		
Name of terminal	Type	Heating	Cooling	Ventilation (fresh air)	Humidification + Dehumidification	Convection	Mainly convection	Radiation	Mainly radiation	Heating	Cooling	Air	Water	Electricity
Radiant systems	Floor*	Y	Y	N	N	Y	N	Y	N	99	42	Y	Y	Y
	Wall	Y	Y	N	N	Y	N	Y	N	160	72	N	Y	Y
	Ceiling	Y	Y	N	N	Y	N	Y	N	42	99	Y	Y	Y
Air systems**	Mixing ventilation	Y	Y	Y	Y	N	Y	N	N	34°C	14°C	Y	N	N
	Displacement ventilation	Y/N	Y	Y	Y	N	Y	N	N	NC	18°C	Y	N	N
	Personalized ventilation	Y/N	Y	Y	Y	N	Y	N	N	NC	20°C	Y	N	N
Beams	Passive	Y	Y	N	N	N	Y	N	N	NC	80	N	Y	N
	Active	Y	Y	Y	Y***	N	Y	N	N	50	120	Y	Y	N

*: Floor in the occupied zone.
**: For air systems, typical maximum and minimum supply air temperatures are provided [11]. The heating and cooling capacity will depend on the ventilation rate.
***: Humidification and dehumidification is possible with the primary air and it should be done by the air-handling unit. Beams operate in dry, non-condensing conditions.

Dynamic building simulation softwares, Computational Fluid Dynamics (CFD), and experimental methods could be used to evaluate the performance of different terminal units, in terms of energy performance, resulting temperature and humidity fields, and occupant thermal comfort.

In addition to the characteristics that were addressed in this paper, there are several factors to consider when selecting a terminal unit (excluding the capital and operational expenditures):

- The chosen type of heating and cooling strategy (terminal unit) will have a remarkable effect on the occupant thermal comfort, and criteria such as noise, draft, vertical air temperature difference, etc. could be limiting factors for the choice and application of different terminal units. The international standards (EN 15251:2007 [21], EN ISO 7730:2005 [30], ASHRAE 55-2010 [31]) should be followed to provide the optimal thermal comfort for the occupants.
- It is crucial to consider the transportation and auxiliary energy consumption (pumps, fans, valves, dampers, sensors, etc.) associated with each terminal unit.
- Availability (depending on the location, natural resources, district heating or cooling network, etc.) of the energy sources and sinks, and the possibility of coupling with the terminal unit should be considered.
- Control possibilities and principles (e.g. individual room or zone control, control based on flow rate, supply temperature, average temperature, as a function of air temperature, operative temperature or outside temperature) and dynamic behaviors of the terminal units should be considered, e.g. ventilation systems try to keep a constant room temperature while TABS allows a certain temperature drift and keeps the operative temperature within the comfort range rather than a constant value [4]. Another example of the dynamic behavior is the difference between a TABS and a radiant panel, where the radiant panel will be able to affect the thermal conditions in the room faster than a TABS construction, due to its significantly lower thermal mass.

When deciding on which terminal unit to use in a space, all of these issues (occupant thermal comfort, transportation and auxiliary energy consumption, possible use of heat sources and sinks, control and dynamic behavior) and the possibilities, limitations, and capacities provided in this paper should be considered.

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Exergy performance of different space heating systems: A theoretical study

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ABSTRACT

Three space heating systems (floor heating with different floor covering resistances, radiator heating with different working temperatures, warm-air heating with and without heat recovery) were compared using a natural gas fired condensing boiler as the heat source. For the floor heating systems, the effects of floor covering resistance on the whole system performance were studied using two heat sources; a natural gas fired condensing boiler and an air-source heat pump. The heating systems were also compared in terms of auxiliary exergy use for pumps and fans.

The low temperature floor heating system performed better than other systems in terms of exergy demand. The use of boiler as a heat source for a low-exergy floor heating system creates a mismatch in the exergy supply and demand. Although an air-source heat pump could be a better heat source, this depends on the origin of the electricity supplied to the heat pump. The coefficient of performance (COP) of the heat pump has a critical value (2.57 in this study); it is beneficial to use a heat pump instead of a boiler only when the COP is above this critical value.

The floor covering resistance should be kept to a minimum, in order not to hinder the performance of the floor heating and the whole system. The exergy input to auxiliary components plays a significant role in the overall exergy performance of systems, and its effects become even more significant for low temperature heating systems.

1. Introduction

The choice of space heating system (terminal unit, heat source, etc.) in buildings has important effects on energy and exergy demands, occupant thermal comfort, and on global environment. Although heating load calculations and energy-wise analyses are necessary for system dimensioning, they are not sufficient since energy analyses cannot quantitatively clarify the effects of working temperature levels. Several studies have documented that energy analyses alone are not sufficient to completely understand energy use [1–3]. Exergy analysis can be used for this purpose. Exergy has a wide application range in engineering systems, including heating, cooling, ventilation systems and built environment [4–8]. Recently, exergy analysis has also been used to evaluate the operation and control of heating, ventilation and air conditioning (HVAC) systems in buildings [9–11].

Gonçalves et al. [12] compared energy and exergy performances of eight space heating alternatives under different climatic conditions. Zhou and Gong [13] studied the whole chain of exergy flows for a building heating and cooling system by hourly varying the reference state. Balta et al. [14], Lohani [15], and Lohani and Schmidt [16] studied different heat sources for building heating applications. Zmeureanu and Wu [17] studied the energy and exergy performance of residential heating systems. Schmidt [18] compared different heat sources and heat emission systems (radiator and floor heating), and concluded that the floor heating system performs close to ideal conditions (real exergetic demand of the zone). This is mainly due to the low temperature heating possibility of water-based radiant floor heating systems. Kilkis [19] studied the possibilities of coupling radiant floors to air-source heat pumps, and showed that it is possible to eliminate the supplementary boilers, and this coupling can increase the coefficient of performance (COP) of heat pumps.

The surface covering of a radiant floor heating system could be a

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subjective choice due to interior design of spaces, and this could have important effects on the performance of the floor heating system, and on the whole space heating system. Simmonds et al. [20], and Sattari and Farhanieh [21] studied the effects of floor covering on radiant floors. After a parametric study, Sattari and Farhanieh [21] concluded that the type and thickness of the floor covering are the most important parameters in the design of radiant heating systems.

This study compares the exergy performance of different space heating systems, using a single-family house as a case study. The house was considered to be conditioned with different space heating systems and the exergy performances of these systems were compared. The systems were floor heating with different floor covering resistances, radiator heating with different working temperatures, warm-air heating with and without heat recovery. These systems were also compared in terms of auxiliary exergy used for pumps and fans. The relative benefits of applying heat recovery in the ventilation system were also studied.

For the floor heating systems, the effects of floor covering resistance on the whole system performance was studied, and two heat sources were compared; a natural gas fired condensing boiler and an air-source heat pump.

2. Description of the case studies and determination of the key parameters

2.1. Construction details and description of the house

The house considered in this study was a detached, one-story, single-family house with a floor area of 66.2 m^2 and a conditioned volume of 213 m^3. The house was located in Denmark. It was constructed from wooden elements. The house was insulated with a combination of 200 mm mineral wool and 80 mm compressed stone wool fibers.

Inside the house, there was a single space combining different functions. The glazing façades were partly shaded by the roof overhangs. The largest glazing façade was oriented to the North with a 19° turn towards the West. Fig. 1 shows the exterior and interior views of the house. The surface areas and thermal properties of the envelope are given in Table 1.

2.2. Details of the heating, cooling, and ventilation systems in the house

The sensible heating and cooling of the house relied on the low temperature heating and high temperature cooling principle by the hydronic radiant floor. The floor heating system was a dry radiant system, consisting of a piping grid installed in the wooden layer. The details of the floor system were: chipboard elements, with aluminum heat conducting profiles (thickness 0.3 mm and length 0.17 m), PE-X pipe, 17 × 2.0 mm. Pipe spacing was 0.2 m. The available floor area for the embedded pipe system installation was 45 m^2, which is 68% of the total floor area. Fig. 2 shows the details of the floor structure.

Table 1
Thermal properties of the envelope.

	North	South	East	West	Floor	Ceiling
Walls, Area, [m^2]	–	–	37.2	19.3	66.2	53
Walls, U-value, [W/m^2K]	–	–	0.09	0.09	0.09	0.09
Windows, Area, [m^2]	36.7	21.8	–	–	–	0.74
Windows, U-value, [W/m^2K]	1.04	1.04	–	–	–	1.04

The heat source and sink of the house for space heating and cooling was outdoor air, using a reversible air-to-brine heat pump. The minimum and maximum cooling capacities and the nominal power input in the cooling mode were 4.01, 7.1, and 2.95 kW, respectively. The minimum and maximum heating capacities and the nominal power input in the heating mode were 4.09, 7.75, and 2.83 kW, respectively.

A flat-plate heat exchanger was installed between the hydronic radiant system of the house and the air-to-brine heat pump. The pipes between the heat exchanger and the heat pump were filled with an anti-freeze mixture (40% ethylene glycol) to avoid frost damage during winter.

A mixing station which linked the radiant system with the heat source and sink, and the controller of the radiant system controlled the flow and the supply temperature to the radiant system. The operation of the radiant system was based on the operative temperature set-point that was adjusted on a room thermostat and on the relative humidity inside the house to avoid condensation during summer.

The house was ventilated mechanically by an air handling unit (AHU), by which only the fresh air taken into the house was heated or cooled since the main sensible heating and cooling terminal was the radiant system. This also made it possible to have lower air flow rates compared to a system where space heating and cooling is mainly obtained by an air system [22]. The design ventilation rate was 0.5 ach [23].

There were two options for heat recovery in the AHU. One was obtained by means of a cross-flow heat exchanger with the heat recovery efficiency of 85% (sensible heat) and with a by-pass route. The other was achieved by means of a reversible air-to-water heat pump that was coupled to the domestic hot water tank. The former is the passive and the latter is the active heat recovery, respectively. The AHU could supply fresh air at a flow rate of up to 320 m^3/h (1.5 ach) at 100 Pa. Further details of the components and the system can be found in Refs. [24–26].

2.3. Determination of the design heating load

The dimensioning of the systems in the house was based on the

Fig. 1. Exterior (seen from South-West) and interior views of the house.

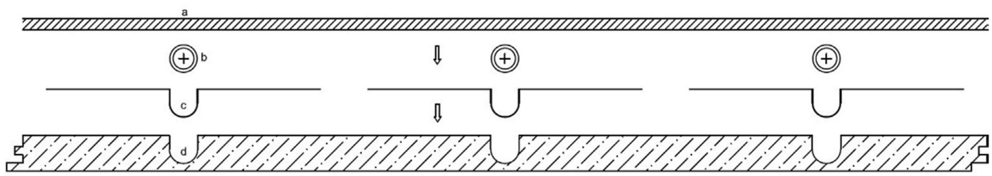

Fig. 2. a) Floor covering b) Pipe, 17 × 2.0 mm c) Heat distribution plate, 0.3 mm, d) Under-floor plate, 22 mm.

design outdoor condition, which in Denmark is an outdoor air temperature of −12 °C [27], though in the theoretical analyses in this paper the outdoor temperature was taken to be −5 °C, which is a more typical winter temperature. Previous experiments comparing the performance of different heating systems in a room had also assumed an outdoor temperature of −5 °C [28].

For all cases, the indoor temperature was considered to be 20 °C (air temperature and mean radiant temperature).

The calculations were carried out under steady-state conditions. No solar heat gains were considered. The internal heat gain was considered to be constant and 4.5 W/m^2, which represents two persons at 1.2 met and other household equipment. For the given heat recovery efficiency in the AHU, the supply air temperature after the heat recovery was 16.3 °C. The parameters used in the load calculations and the resulting space heating loads are given in Table 2.

2.4. Determination of the key parameters

Following the calculation of the space heating load, the parameters of the different space heating strategies necessary for the present exergy analyses were determined. It was assumed that the heating demand of the house was addressed with different heating systems; floor heating, radiator heating, and warm-air heating. The following assumptions were made during the calculations:

- In the actual house there was a heat exchanger between the radiant system and the heat pump, but for the calculations this heat exchanger was neglected and it was assumed that the water in the embedded pipes circulated directly through the boiler or the condenser of the heat pump. The same assumption applied to the radiator heating cases.
- The supply air was 100% outdoor air (no recirculation), and the indoor air was assumed to be fully mixed.
- The active heat recovery was not considered and the AHU was considered to be working in the passive heat recovery mode, as described in 2.2.
- It was assumed that there was no heat loss from the floor heating system, radiators, pipes and ducts to the outdoors.

The schematic drawings of the analyzed heating systems are given in Fig. 3.

2.4.1. Floor heating with different floor covering resistances

The effect of the floor covering resistance on the whole system performance of floor heating was studied through different floor covering resistances. The thermal resistances considered were 0.05 m^{2}K/W, 0.09 m^{2}K/W (actual value), and 0.15 m^{2}K/W, following the most common values given in respective standards [29]. The same floor covering material was used (thermal conductivity is 0.13 W/mK, wooden floor covering), but with 0.0065 m, 0.012 m and 0.0195 m thickness, respectively. For each resistance value, a new water supply temperature was calculated.

The load calculations showed that a specific heat output of 48.4 W/m^2-floor heating area with an average floor surface temperature of 24.7 °C was required to meet the necessary space heating demand. The heat output and surface temperatures were calculated according to [29] and [30]. The surface temperatures of the floor heating systems were the same for all cases, and this was achieved by adjusting the supply and return temperatures to the floor heating system. For all cases, the temperature drop between the supply and return water in the radiant system was 4 K. The mass flow rate was calculated based on EN 1264-3:2009 [31] and was found to be 469 kg/h. A summary of the floor heating cases is given in Table 3.

2.4.2. Radiator heating with different working temperatures

The assumed radiator type was a double panel steel radiator with extended surface (fins) [32]. Four sets of working temperatures were assumed according to prEN 15316-2:2014 [33]: 45/35, 55/45, 70/55, and 90/70 (supply/return water temperature in °C).

The required flow rates in the radiators were determined according to the space heating load and the temperature difference between the supply and return water temperatures to and from the radiator.

The average surface temperatures of the radiators were assumed to be 0.3 °C lower than the simple average value of inlet and outlet water temperatures. This value was determined so that not only the energy balance but also the entropy and exergy balance equations were satisfied. Table 4 summarizes the radiator heating cases.

2.4.3. Warm-air heating with and without heat recovery

In addition to the water-based heating systems, an air-based heating system was also analyzed and two cases, without and with a heat recovery device, as shown in Fig. 3 c) and d), were

Table 2
Parameters used in the load calculations and resulting space heating loads.

Indoor temperature [°C]	20
Outdoor temperature [°C]	−5
Internal heat gain [W/m^2]	4.5
Ventilation rate [ach]	0.5
Infiltration rate [ach]	0.2
Heating load – envelope (no ventilation), total [W] and per floor area [W/m^2]	2048/30.9
Heating load – envelope and heat recovery ventilation, total [W] and per floor area [W/m^2][a]	2180/32.9

[a] The difference between the two space heating loads is due to the fresh air being supplied at 16.3 °C with a ventilation rate of 0.5 ach. This contributes to the space heating load, hence the higher space heating load in the second case.

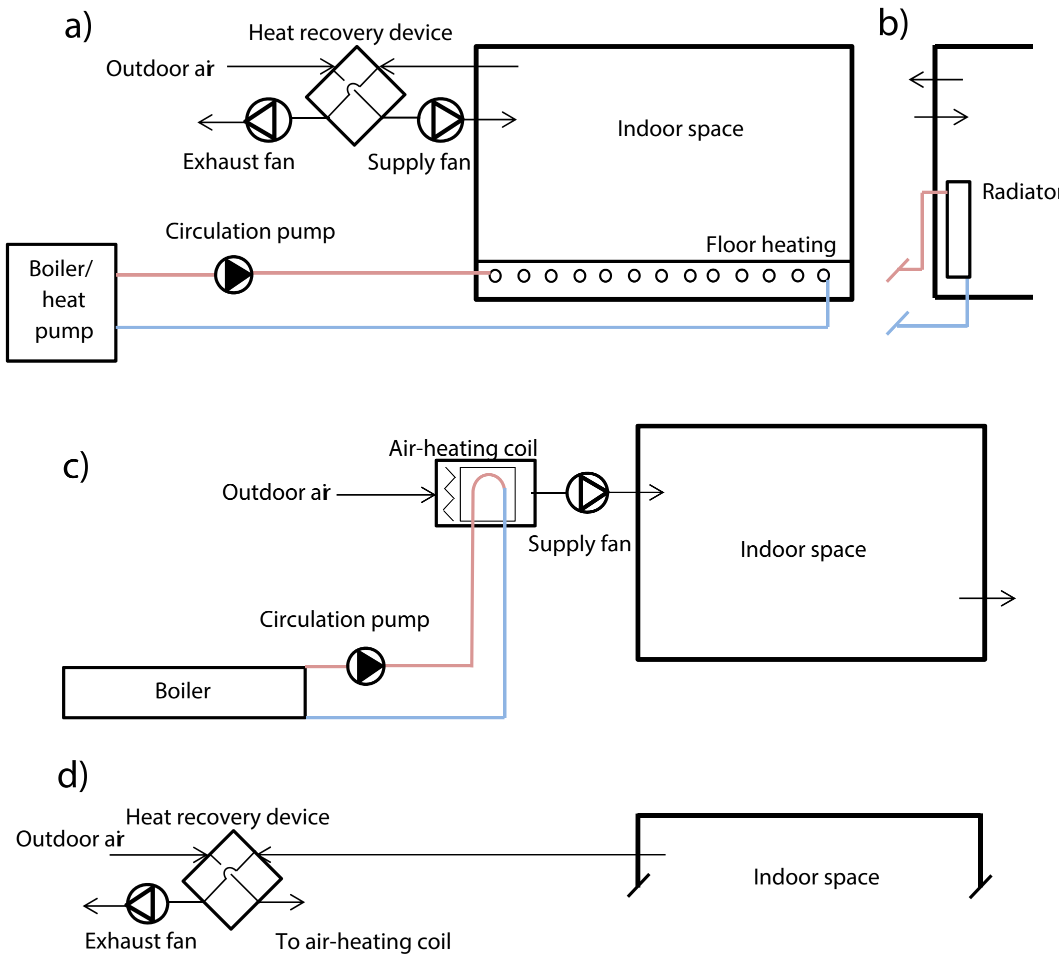

Fig. 3. Schematic drawings of the analyzed heating systems: a) Floor heating b) Radiator heating c) Warm-air heating without heat recovery d) Warm-air heating with heat recovery.

Table 3
Summary of the floor heating cases.

Case name	Floor covering resistance [m^2K/W]	Supply temperature [°C]	Return temperature [°C]	Heat pump COP [-]
FH_LoRes	0.05	33	29	2.63
FH_MRes	0.09	35.8	31.8	2.48
FH_HiRes	0.15	39.8	35.8	2.31

Table 4
Summary of the radiator heating cases.

Case name	Supply temperature [°C]	Return temperature [°C]	Surface temperature [°C]	Mass flow rate [kg/h]
R_45	45	35	39.7	188
R_55	55	45	49.7	188
R_70	70	55	62.2	125
R_90	90	70	79.7	94

studied.

The supply air temperature to the space was limited to 35 °C [27]. The necessary heating rate needed for bringing the outdoor air at −5 °C to the supply air temperature of 35 °C was 5460 W when there was no heat recovery, and it decreased to 2559 W with heat recovery. The necessary heat was supplied to the air by an air-heating coil which was connected to a boiler. The supply and return water temperatures to the air-heating coil were 50 °C and 39 °C, respectively [32].

The necessary air flow rate (1.9 ach) and the water flow rates in the air-heating coil are given in Table 5 together with a summary of the warm-air heating cases.

2.4.4. Fan and pump powers

The power to be supplied to fans and pumps that circulate the heat transfer medium in pipes or in ducts was determined as follows.

For the floor heating and radiator heating cases, it was assumed

Table 5
Summary of the warm-air heating cases.

Case name	Intake air temperature [°C]	Air temperature after HR [°C]	Supply air temperature [°C]	Air flow rate [m^3/h]	Mass flow rate in air-heating coil [kg/h]
WAH_NoHR	−5	–	35	410	428
WAH_HR	−5	16.3	35	410	201

that there is a pump which circulates the water between the boiler or heat pump and the floor loops and radiators. Additionally, two identical fans (supply and exhaust) were considered in the AHU. For the warm-air heating cases, a pump, which circulates the water between the boiler and the air-heating coil, was assumed. For the case without heat recovery, only a supply fan providing the necessary air flow to the indoors was assumed, while for the case with heat recovery, two identical fans were assumed.

The pump power for different cases was obtained from the pump specifications (performance curve of the installed pump) as a function of the water flow rate and the required pressure increase, assuming the pump actually installed in the house.

The measurements from the house were used to obtain the fan powers. The measurements from the house showed that the AHU was using 67.9 W with a ventilation rate of 0.5 ach (105 m^3/h) [26], which corresponds to a total specific fan power (SFP) of 2331 J/m^3 for two fans, and for one fan it corresponds to 1166 J/m^3. This SFP value is in SFP 3 category according to EN 13779:2007 [34].

Assuming that the fans for the warm-air heating cases are also in SFP 3 category (1200 J/m^3), the fan powers were calculated as a function of the air-flow rates.

Table 6 summarizes the pump and fan powers for different cases.

Table 6
Summary of the pump and fan powers for different heating cases.

	E_{pump} [W]	E_{fans} [W]	E_{total} [W]
FH_LoRes, FH_MRes, FH_HiRes	27.5	67.9	95.4
R_45, R_55	24.5	67.9	92.4
R_70	23	67.9	90.9
R_90	22	67.9	89.9
WAH_NoHR	27	136.5	163.5
WAH_HR	25	273	298

2.4.5. Heat and power generation

For the first part of the analyses, it was assumed that the heat generation for space heating for all cases was through a natural gas fired condensing boiler with an efficiency of 90% [5,35]. The ratio of chemical exergy to the higher heating value of natural gas was taken as 0.93 [5].

In the second part of the analyses, it was assumed that the floor heating was coupled to an air-to-water heat pump. The COP of the 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 equal to the supply water temperature to the floor loops).

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 [5].

3. Basic definitions of exergy and calculation methodology

3.1. Basic definitions

For any system in consideration, it is possible to obtain the exergy balance equation from energy and entropy balance equations. In general form, exergy balance equation is obtained as follows [5]:

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. Therefore it is possible to obtain the exergy balance equation as Eqs. (1)–(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], where the system and its components are situated in. The storage terms in Eqs. (1)–(3) disappear under steady-state conditions.

Eqs. (1)–(3) indicate that every system consumes a part of the supplied exergy and in the meanwhile entropy is generated. This applies for heating and cooling systems in buildings as well.

3.2. Heating exergy load

The heating exergy load [1] is the necessary load that the heating system has to address. 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 load [W], T_o is outdoor (environmental) temperature [K], and T_i is the operative temperature [K].

3.3. Exergy supplied to indoors

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

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

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

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

Where $X_{FH,out}$ is the exergy supplied from floor heating 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 radiator to the indoor space [W], $T_{S,R}$ is the average surface temperature of the radiator [K], $\Delta X_{WAH,out}$ is the net exergy supplied through 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].

Exergy consumed in the indoor space is the difference between the exergy supplied indoors and the heating exergy load.

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

The exergy consumptions in the floor structure and in the radiator are possible to obtain from the exergy balance for these terminal units as

$$\Delta X_W - X_c = X_{out} \tag{8}$$

$$\Delta X_w = X_{w,supply} - X_{w,return} \tag{9}$$

Where ΔX_w is the difference between the rate of exergy of the supply and return water (net exergy input) [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 indoors 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 both for floor heating and radiator heating.

The exergy of the supply and return water flows are 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 exergy of 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 water [K].

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

$$\Delta X_w - X_c = \Delta X_a \tag{11}$$

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

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

Where $X_{w,supply}$ is the exergy of the water entering the air-heating coil (from boiler) [W], $X_{w,return}$ is the exergy of the water leaving the air-heating coil (to boiler) [W], $X_{a,out}$ is the exergy of the air leaving the air-heating coil [W], and $X_{a,in}$ is the exergy of the air entering the air-heating coil [W]. The exergy of the water is calculated using Eq. (10). The exergy of the air is 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 exergy of 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 unit is obtained through the exergy balance equation set up for the heat recovery device:

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

Where $X_{outdoor\ air}$ is the exergy of the intake air from outdoors (=0) [W], $X_{exhaust\ air}$ is the exergy of the exhaust air (from the indoor space) [W], $X_{inlet\ air}$ is the exergy of the inlet air (supply air for the floor heating and radiator heating cases, and the air entering the air-heating coil in the warm-air heating case with heat recovery) [W], and $X_{discharge\ air}$ is the exergy of the discharge air (discarded to the environment after the heat recovery) [W]. Eq. (14) is used for calculating $X_{outdoor\ air}$, $X_{exhaust\ air}$, $X_{inlet\ air}$, $X_{discharge\ air}$.

In addition to the exergy consumption due to heat transfer in the heat recovery device, the discharged air also contains a certain amount of exergy. This exergy is totally consumed while the discharged air is completely discarded into the environment.

3.5. Exergy input to the boiler and the power plant

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

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

Where $X_{in,boiler}$ is the exergy input by natural gas to the boiler [W], Q_{boiler} is the rate of heat 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. Q_{boiler} is 2180 W for the floor heating and radiator heating cases as described in 2.3, while for warm-air heating cases without and with heat recovery, it is 5460 W and 2559 W, respectively, as described in 2.4.3.

For the floor heating cases when a heat pump was used as the heat source, the exergy input required at the power plant was calculated as follows:

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

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

Where E_{HP} is power (electricity) input to the heat pump [W], COP is the coefficient of performance, $X_{in,\ power\ plant}$ is the exergy input to the power plant through natural gas [W], η_{TOT} is the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid.

Exergy input required at the power plant for the pump and fans are calculated using Eq. (18) by replacing the E_{HP} with respective pump power (E_{pump}) and fan power (E_{fans}).

4. Results and discussion

4.1. Comparison of different space heating systems

The exergy inputs, outputs, and consumptions in different system components are shown in Fig. 4. The chains of exergy flows from the exergy input to the boiler to the environment are shown in Fig. 5. The exergy contained by natural gas is supplied to the boiler at a rate of 2253 W, and the exergy output from the boiler to the water circulating in the floor is 300 W (FH_HiRes). The difference between the input and output exergy is the exergy consumption in the boiler (due to combustion and exhausted gas through the chimney). The same relationship between input, output and consumption applies also to other components.

The heating exergy load [1] (input to the "Building" in Fig. 5) consists of the heat loss from the building envelope (transmission loss) and the heat required to bring the fresh air at 16.3 °C to the indoor temperature of 20 °C, therefore the heating exergy loads are the same for the floor heating and radiator heating cases (186 W). In the warm-air heating case, this load is different (174.6 W) since the

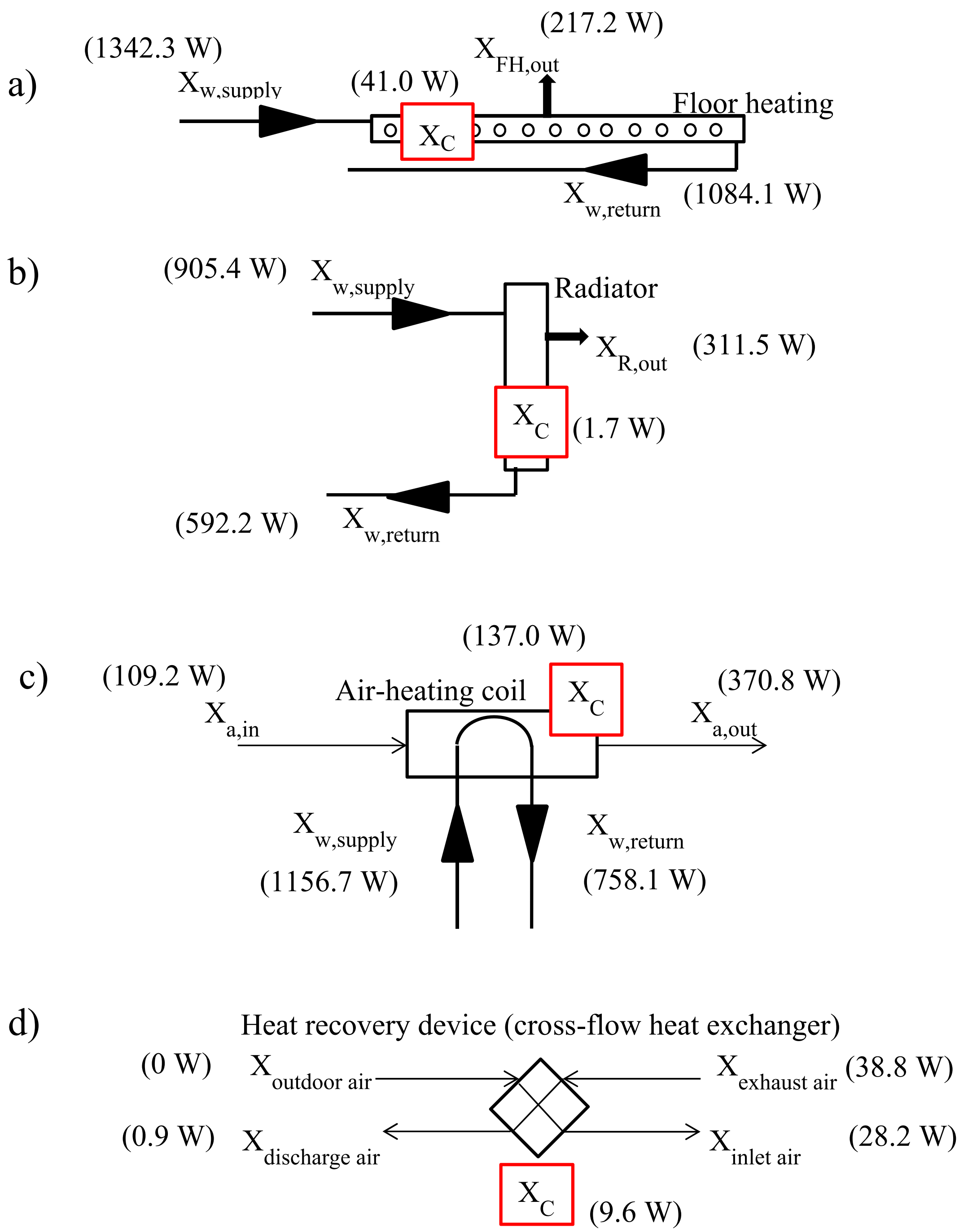

Fig. 4. Exergy input, output, and consumption in different system components: a) Floor (FH_LoRes) b) Radiator (R_45) c) Air-heating coil (WAH_HR) d) Heat recovery device (0.5 ach, floor heating and radiator cases). Values in the parentheses indicate the exergy values.

supply air has to be warmer than the indoor air.

Exergy consumption in the indoor space varies with different heating strategies. For the floor heating cases the exergy consumption in the space is 31.2 W, and this is the smallest among all cases. This is because of the low temperature heating possibility and the low surface temperature of the floor. For the radiator cases, the exergy consumption in the indoor space increases with the increasing average water temperature. Among the four radiator cases, the smallest exergy consumption in the indoor space is 125.5 W for R_45. Exergy consumption in the indoor space with the warm-air heating is 46.3 W, which is between the values of floor heating (31.2 W) and radiator cases (125.5 W). Nonetheless, it should be noted that this is only in the indoor space and does not consider the rest of the systems.

For different floor covering resistances, the energy coming into the floor structure (heat emitted to the indoor space) is the same but the exergy is different. The increased floor covering resistance results in higher average water temperature to obtain the same floor surface temperature to provide the necessary space heating. This inevitably results in increased exergy supply to and consumption within the floor structure. Compared to the low resistance case, in which 41 W is consumed, the exergy consumption

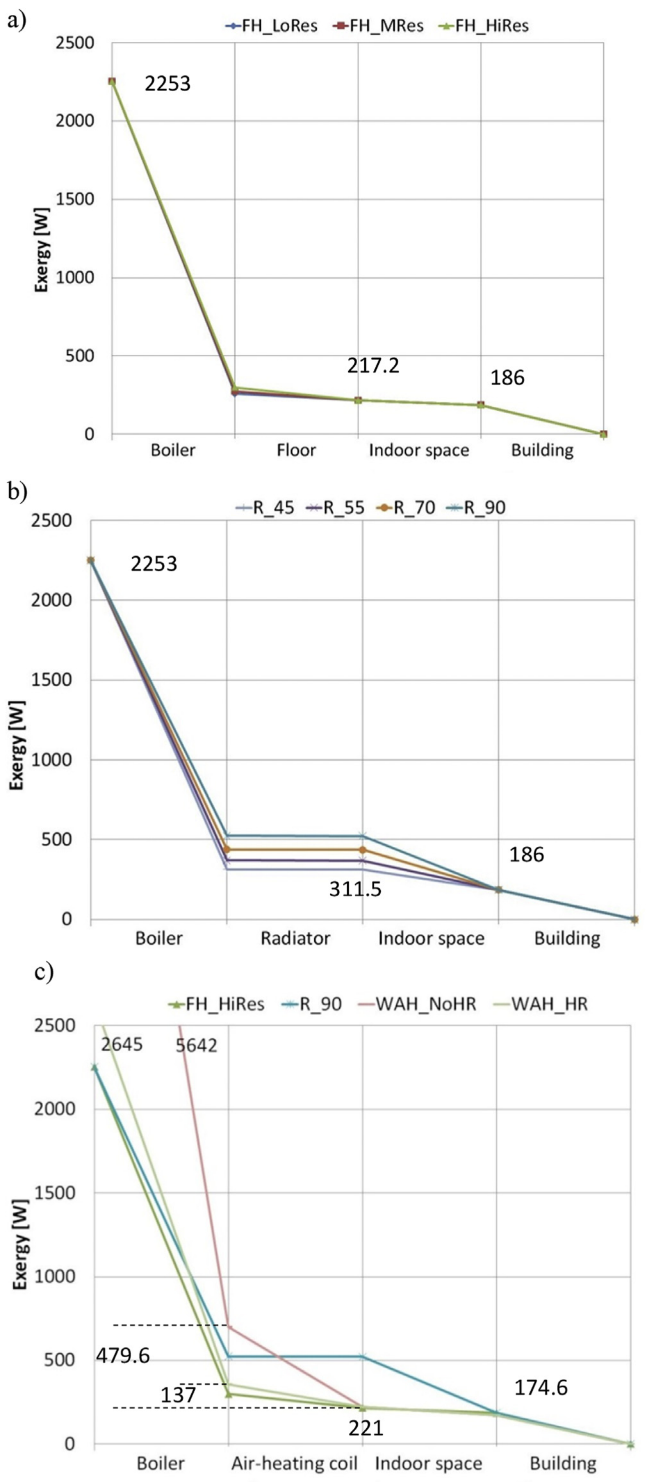

Fig. 5. Exergy flows for different heating strategies: a) Floor heating b) Radiator heating c) Warm-air heating.

increased by 42% and almost doubled for the cases with mid- and high floor covering resistances, respectively.

The exergy consumption within the radiators was very small, ranging from 1.2 W to 1.7 W, (0.2%–0.6% of net thermal exergy input to the radiators). This small exergy consumption was mainly due to the assumed surface temperature (due to the high conductivity of materials used for constructing radiators) of the radiators.

Fig. 5 shows that the largest exergy consumption in the terminal units (the air-heating coil for warm-air heating cases) occurred for the warm-air heating cases. The application of heat recovery on the exhaust air resulted in a significant decrease in the exergy consumption within the air-heating coil, from 479.6 W to 137 W, because with heat recovery the air temperature should be increased from 16.3 °C to 35 °C in the air-heating coil compared to no heat recovery where the air temperature should be increased from −5 °C to 35 °C. This is also reflected in the necessary energy and exergy inputs to the boiler.

The floor heating and radiator cases require the same fuel (exergy) input to the boiler (2253 W) while the required fuel input is the largest for the warm-air heating without heat recovery (5642 W), followed by the case with warm-air heating with heat recovery (2645 W).

The results show that the floor heating system requires the smallest exergy input to provide the same space heating. Higher floor covering resistance hinders the performance of the floor heating system. When comparing the cases with the lowest and highest floor covering resistances, the exergy consumption in the floor structure doubles and the required exergy input to the floor structure increases by 16% with a higher floor covering resistance. This underlines the importance of keeping the floor covering resistance to a minimum through selection of floor covering material and thickness to fully benefit from a low temperature heating system.

The radiator and warm-air heating cases have remarkably higher exergy demand compared to floor heating cases. The floor heating system with the highest floor covering resistance and the radiator with lowest average water temperature require relatively close exergy inputs (300 W and 313 W, respectively) although the floor heating system still requires lower exergy input. Among the investigated cases, the floor heating performed better than other space heating systems, in terms of required exergy input, and exergy consumption.

Regarding the exergy inputs to the heating plant, it can be observed that the use of a boiler does not allow taking advantage of the low exergy demand of the radiant floor heating system. The largest exergy consumption through the whole space heating process occurs in the boiler and this is mainly due to the combustion process inside the boiler.

Due to this mismatch of supply (high) and demand (low) exergy levels, an air-source heat pump was assumed to replace the condensing boiler to be the focus in the following analyses.

4.2. Comparison of the effects of different floor covering resistances on heat pump performance

The exergy flows for the whole process of space heating using a heat pump is shown in Fig. 6.

Fig. 6 shows that in order to obtain the same space heating effect, higher electricity input is required to the heat pump with increased floor covering resistance. This means that a higher exergy input is necessary to the power plant where the electricity is generated. This system behavior is because a higher floor covering resistance requires higher average water temperature for the floor heating system, which leads to a higher condensing temperature for the heat pump, decreasing the heat pump performance (lowering the COP), as given in Table 3.

When the overall exergy inputs to the boiler and to the power plant are compared, the results show that the use of a heat pump instead of a boiler is beneficial only when the floor covering resistance is kept to a minimum (FH_LoRes). For FH_MRes and FH_HiRes, the use of a boiler is a better choice, as far as the present cases are concerned. These results indicate that further

Fig. 6. Exergy flows for the floor structures with different floor covering resistances.

considerations are necessary regarding different heat sources.

The results suggest that the improved COP and lowered floor covering resistance has similar effects on the whole system performance, i.e. improvement with the same thickness (or material) is comparable to an improvement in the heat pump COP.

The results also show that the increased floor covering resistance has a similar effect on the heating plant (water-side of the radiant floor heating system) to an increase in heating load, although the space heating load itself is in fact constant; to obtain the same space heating effect, higher electricity input is required to the heat pump, and 14% higher exergy input is necessary to the power plant for FH_HiRes compared to FH_LoRes.

4.3. *The effects of the heat source*

The comparison of the boiler and the heat pump showed that there is a critical COP value (2.57 for the present calculations) for a heat pump, and the COP should be greater than this value for the heat pump to be beneficial over the boiler. This finding agrees with Kilkis [35] that it is crucial to consider and account for the source of the electricity used for the heat pump (generation plant, fuel input to the plant, distance to the utilization site, etc.).

In order to achieve a higher system performance, the improvement could be either on the power plant (higher conversion efficiency, on-site generated renewable electricity, etc.) or on the performance of the heat pump; increased COP through a better heat pump, by using a heat pump which is coupled to another heat source, such as ground, lake water, sea water, etc. Recent studies [5,36], showed that an air-source heat pump does not benefit from any heat source (ambient air does not contain any exergy as opposed to other heat sources and sinks which contain exergy), and it is basically a machine to separate the exergy (electricity) into cool and warm exergy.

In order to make a true use of the sustainable energy resources (ground, lake, sea, etc.), the first priority is to minimize the heating and cooling demands of the building. Such minimization would allow the use of a broad range of heat sources and sinks. With a high heating and cooling demand, the exergy demand necessarily becomes higher and it narrows the range of heat sources and sinks that can be used so the use of low-exergy ones becomes limited and the use of mid-to high-exergy content resources becomes inevitable.

Depending on the geographical availability, biomass or other sustainable fuels could be used in a boiler. To sustain such an operation, a prerequisite is to decrease the heating (or cooling) demand of the building as much as possible.

4.4. *Auxiliary energy input, and heat recovery in the AHU*

In addition to the thermal exergy analyses, the effects of electricity inputs to pumps and fans on the whole exergy consumption patterns were also considered. The results are presented in Fig. 7.

The floor heating cases require the largest pumping power due to the smallest temperature drop between the supply and return water flows among the investigated cases. When taking the pumping power into account, the overall performances of the floor heating case with the highest floor covering resistance, FH_HiRes, and the radiator with the lowest average water temperature, R_45, became close although the floor heating case still requires 5 W lower exergy input (300 and 313 W thermal exergy, and 253.5 and 245.5 W exergy for auxiliary components for FH_HiRes and R_45, respectively). A difference of 5 W might not seem significant from the energy viewpoint (thermal energy load is in the order of 10^2 to 10^3) but from the exergy viewpoint, it is significant. Considering radiative exergy emission and absorption, magnitudes of mW (in the order of 10^{-3}) can make a difference in the perception of the thermal environment [5], therefore a difference of 5 W exergy is not negligible and presents an advantage over the radiator solution.

The results also show that an air-based heating system requires large fan powers, resulting in a further decrease of the energy and exergy performance. This is mainly because larger flow rates and volumes are required to transport the same amount of heat with air compared to water-based systems and this emphasizes an advantage of water-based heating and cooling systems over air-based systems.

In the current study and in actual house design [24,26], the indoor space was mainly heated and cooled by the radiant floor heating and cooling system and the ventilation system was only used to provide the required amount of fresh air. When considering the pump and fan consumptions together with the thermal exergy values presented previously in 4.1, this approach proved to be an efficient approach.

The analyses showed that to fully benefit from the low-exergy potential of low temperature heating and high temperature cooling systems, the dimensioning of the auxiliary components (fans, pumps, valves, dampers, and so forth) should be carried out carefully. It is crucial to minimize the exergy demand of these components; especially for systems with a low temperature difference between the supply and return flows. This becomes clearer in terms

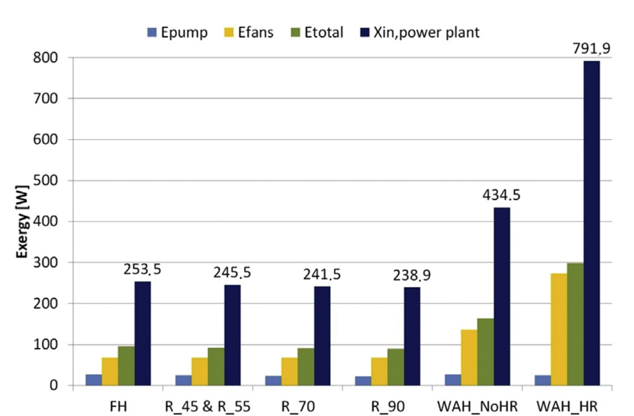

Fig. 7. Required exergy inputs to the pump, fans and to the power plant (four bars in each case indicate, from left to right, the exergy input to pump, to fans, their total, and the exergy input to the power plant).

of exergy: for the floor heating cases, the energy supplied to the pump is 1.3% of the heat provided from the floor to the indoor space while the exergy input to the pump is 12.7% of the exergy supplied from the floor to the indoor space (27.5 W pump power vs. 2180 W space heating load and 217.2 W exergy supplied from the floor to the indoor space, respectively).

There is a trade-off between the exergy gain with the heat recovery unit and the electricity necessarily supplied to the additional exhaust fan (also the extra fan power needed to cover the additional pressure drops). Depending on the local climate, ventilation rate and also on the efficiency of the heat recovery unit, it might not always be beneficial to have heat recovery. If the exergy input to the fan exceeds the exergy recovered from the exhaust air, this indicates that there is a threshold for which the heat recovery is beneficial. This issue should be carefully considered before applying heat recovery on ventilation systems.

When considering the warm-air heating cases without and with heat recovery, extra exergy input for the added exhaust fan is the crucial parameter to consider when evaluating the benefits of heat recovery. The extra exergy input required at the power plant for the additional fan in the case of warm-air heating with heat recovery is 357.4 W which is significantly less than the saved exergy input to the boiler due to the application of heat recovery (2998 W), therefore the application of heat recovery is justified for this case.

During the heat recovery from the exhaust air, the exergy consumption in the heat recovery unit is inevitable. Due to heat transfer between air streams, there is exergy consumption in the heat recovery unit and also due to the efficiency being less than 100% (exergy is necessarily consumed even if the 100% energy-wise efficiency is possible), it is not possible to fully recover the thermal exergy from the exhaust air. A certain amount of air with exergy is being discarded into the environment after the heat recovery, and hence it is lost. This amount of exergy would vary in different locations: in the present case of Denmark, due to low outdoor temperature, it has higher exergy compared to another location with a mild climate.

The exergy consumption in the heat recovery unit for the warm-air heating case is 37.2 W (40.6 W including the exergy of the discharge air), and for the floor heating and radiator cases is 9.6 W (10.5 W including the exergy of the discharge air).

4.5. Overall discussion

Although the heat losses from the distribution and emission systems were not considered in this study, heat losses should be minimized for high energy- and exergy-wise performance of heating systems.

The choice of the terminal unit has a direct and significant effect on the thermal comfort of occupants. The radiant low temperature heating and high temperature cooling systems (floor, ceiling, and wall heating or cooling) perform better than other room conditioning systems in terms of providing a thermally comfortable indoor environment due to their advantages of minimizing the risk of unpleasant air movement, and of creating a uniform thermal indoor environment [37,38].

Heat source and terminal unit selection would depend on the available products, costs, location of the building (connection to a district heating network, regulations regarding use of the ground, etc.), additionally, pressure drops would also change depending on the piping lay-out. Pressure drops will affect the necessary pumping powers. Given the overall possibilities and limitations of system selection, to achieve exergy-efficiency, the system and all its components should be considered as a whole from the beginning of the design process. It is crucial to minimize the demand of the building; this would enable to use a broad range of heat sources (ground, lake, etc. especially those to be found in our surroundings), among the available heat sources, the most environmentally friendly and sustainable heat source should be chosen to meet the exergy demand of the building.

5. Conclusion

Exergy performances of different space heating systems (floor heating, radiator heating, and warm-air heating) were compared in this study. In addition to the thermal exergy, these systems were also compared in terms of auxiliary exergy used for pumps and fans. The relative benefits of applying heat recovery on the ventilation system were also studied.

For the floor heating systems, the effects of floor covering resistance on the whole system performance was studied, and two heat sources were compared; a natural gas fired condensing boiler and an air-source heat pump.

The main conclusions from the analyses are as follows:

1. Among the investigated cases, the floor heating system had the lowest exergy consumption and it performed better than other space heating systems in terms of required exergy input, and exergy consumption.
2. Coupling of a natural gas fired condensing boiler with the floor heating system is not an efficient strategy due to the mismatch of the exergy supply and demand. Therefore it does not allow taking advantage of the low exergy demand of the radiant floor heating system.
3. Using a heat pump instead of a boiler as the heat source could provide a better match for the low exergy demand of the floor heating system, but there is a critical COP value and only above this COP value it is beneficial to use a heat pump instead of a boiler. It is also crucial to account for the source of electricity supplied to the heat pump (e.g. generated at a remote, fossil fuel power plant or at a nearby renewable energy plant).
4. Subjective choices on floor covering have significant effects on the performance of the floor heating system and on the whole system. Higher floor covering resistance hinders the performance of the whole system. When comparing the cases with the lowest and highest floor covering resistances, the exergy consumption in the floor structure doubles and the required exergy input to the floor structure increases by 16% with a higher floor covering resistance. This means that to obtain the same space heating effect, higher electricity input is required to the heat pump, and 14% higher exergy input is necessary to the power plant where the electricity is generated. In order not to hinder the performance of a radiant system (floor heating, floor cooling, ceiling cooling and so forth), the covering resistance should be kept to a minimum, providing that structural and maintenance (e.g. wear and tear on floor coverings) concerns can be met and that the occupants are aesthetically satisfied.
5. The present analyses showed that the water-based systems require lower auxiliary energy use than the air-based heating system. This is a clear benefit of water-based systems over air-based systems. To fully benefit from the low-exergy heating potential of the low temperature heating systems, the dimensioning and choice of the auxiliary components should be made carefully and this is clearly explained with the exergy analyses.
6. For a heat recovery unit to be beneficial, the thermal exergy gained from the heat recovery (from the exhaust air) must be greater than the exergy supplied to the exhaust fan, and this analysis should be carried out before deciding on the application of heat recovery. In the current analyses the application of heat recovery proved to be beneficial: the extra exergy input required at the power plant for the additional fan in the case of warm-air

heating with heat recovery was 357.4 W which is significantly smaller than the reduced exergy input to the boiler due to the application of heat recovery (2998 W).

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Note to the reader: The naming convention used in this article is different from the one used in the thesis. They represent the same systems, only with different names. The corresponding names are as follows: Case 1 is AC_INT, Case 2 is AC, Case 3 is AC_17, Case 4 is AC_20, Case 5 is AC_CS, Case 6 is FC, Case 7 is FC_CS, and Case 8 is FC_CS_GHEX.

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Theoretical analysis of the performance of different cooling strategies with the concept of cool exergy

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ABSTRACT

The whole chains of exergy flows for different cooling systems were compared. The effects of cooling demand (internal vs. external solar shading), space cooling method (floor cooling vs. air cooling with ventilation system), and the availability of a nearby natural heat sink (intake air for the ventilation system being outdoor air vs. air from the crawl-space, and air-to-water heat pump vs. ground heat exchanger as cooling source) on system exergy performance were investigated.

It is crucial to minimize the cooling demand because it is possible to use a wide range of heat sinks (ground, lake, sea-water, etc.) and indoor terminal units, only with a minimized demand. The water-based floor cooling system performed better than the air-based cooling system; when an air-to-water heat pump was used as the cooling source, the required exergy input was 28% smaller for the floor cooling system. The auxiliary exergy input of air-based systems was significantly larger than the water-based systems.

The use of available cool exergy in the crawl-space resulted in 54% and 29% smaller exergy input to the power plant for the air-based and water-based cooling systems, respectively. For floor cooling, the exergy input to the power plant can be reduced by 90% and 93%, with the use of ground, and use of the ground and the air in the crawl-space, respectively. A new approach to exergy efficiency was introduced and used to prove that the exergy supply from the ground matches well with the low exergy demand of the floor cooling system.

1. Introduction

Tightening targets for energy efficiency and energy use reduction in buildings have had significant effects both on residential and non-residential buildings in Europe [1]. The development of passive, low-energy, near zero-energy, and zero-energy buildings has been stimulated by these regulations and environmental concerns, and nearly zero-energy building (nZEB) levels are dictated for new buildings by 2020 in the European Union [1].

The international focus on the residential sector is increasing, and although the energy performance of buildings has increased, issues with the thermal indoor environment and air quality have been reported in low-energy and passive houses [2–4]. One prominent problem is overheating and it has been reported from Denmark [5], Sweden [3,6], Finland [4], and Estonia [7]. These findings indicate that cooling in residential buildings is becoming more important and almost a necessity.

Air-based or water-based systems can be used to heat or cool buildings. Although different studies have evaluated the performance of air-based and water-based heating and cooling systems for office buildings [8–10], and benefits of radiant panel heating and cooling in net zero-energy buildings [11], so far there has only been little focus on residential buildings and dwellings regarding cooling systems and their exergy performance.

In addition to the insights to different systems by energy analyses, exergy analyses articulate more precisely and accurately the different quality of energy sources and flows. “Cool” and “warm” exergy concepts enable us to quantify and to properly account for the “warmth” and “coolness” of a heat source or sink, and exergy flows from these sources and sinks [12–14].

In this study, the exergy performance of different space cooling systems was compared using a single-family house as a case study.

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The whole chain of exergy flows were considered from the source until the environment. The effects of cooling demand (studied by means of installing internal vs. external solar shading), space cooling method (floor cooling vs. air cooling with ventilation system) including auxiliary exergy use for pumps and fans, and the availability of a nearby cool exergy source (intake air for the ventilation system being outdoor air vs. air from the crawl-space, and air-to-water heat pump vs. ground heat exchanger as cooling source) on the system performance regarding energy, exergy demand and exergy consumption were studied. The cool exergy concept was used to analyze the crawl-space and the ground.

2. Analyzed space cooling systems

The eight different cooling systems that were studied in this paper are described here, before explaining the exergy calculation method that was used to perform the case studies.

2.1. Determination of the design cooling load

The studied house was assumed to be located in Copenhagen, Denmark. Construction details, description and details of the heating, cooling and ventilation systems of the actual house are given in Refs. [15] and [16].

The space cooling load was determined with the assumption of steady-state conditions. The outdoor air temperature was assumed to be 30 °C, which is also the environmental (reference) temperature for exergy calculations. For all cases, the indoor temperature was 26 °C (air temperature and mean radiant temperature). The relative humidity indoors was assumed to be 55%, resulting in a dew point temperature of 16.3 °C.

The house was supported on 30 cm high concrete blocks and this created a crawl-space between the ground and the house's floor structure. When the intake air was taken from the crawl-space, the fresh air temperature coming into the air handling unit (AHU) or to the indoor space was 21.3 °C, due to the pre-cooling of the outdoor air by the ground surface under the crawl-space.

The internal heat gain was assumed to be 4.5 W/m^2 which represents two persons at 1.2 met and other household equipment. For the floor cooling cases, a ventilation rate of 0.5 air change per hour (ach) was used to provide fresh air to the indoors [17]. For the air cooling cases, the supply air flow rate was calculated based on the cooling load. For all cases, an infiltration rate of 0.2 ach was assumed.

For Copenhagen, Denmark (56° Northern Latitude), in July at noon, assumed direct solar radiation on the South and West directions were 390 and 149 W/m^2, respectively, and the diffuse solar radiation was 32 W/m^2 [18]. The shading coefficients for internal and external solar shading were assumed to be 0.6 and 0.1, respectively (blinds, 45° inclination, light colored) [18]. The resulting space cooling loads for different cases are given in Table 2 and Table 3.

Table 1
Summary of the case studies.

Case	Shading	Cooling	Source	Intake air
1	Internal	AC	AWHP	OA
2[a]	External	AC	AWHP	OA
3[a]	External	AC	AWHP	OA
4[a]	External	AC	AWHP	OA
5	External	AC	AWHP	CS
6	External	FC	AWHP	OA
7	External	FC	AWHP	CS
8	External	FC	GHEX	CS

[a] Supply air temperatures and air flow rates are different for Cases 2–4. Further details are given in Table 3. AC: air cooling, FC: floor cooling, AWHP: air-to-water heat pump, GHEX: ground heat exchanger, OA: outdoor air, CS: crawl-space.

2.2. Details of eight cases studied

In order to compare the exergy performance of different cooling systems, the house was assumed to be cooled with a water-based radiant floor cooling system or an air cooling system with the supply of cold air from the air handling unit. The following assumptions were made during the calculation procedure:

- In the actual house, there was a heat exchanger between the radiant system and the heat pump, but for the calculations this heat exchanger was neglected and it was assumed that the water in the floor loops circulated directly through the evaporator of the heat pump. The same was assumed for the air-cooling coil in the AHU.
- The supply air was 100% outdoor air (no recirculation), and the indoor air was assumed to be fully mixed (mixing ventilation).
- It was assumed that there was no heat gain to the floor cooling system, pipes and ducts from the outdoors.

A summary of the investigated cases is given in Table 1, and schematic drawings of the eight cases are given in Fig. 1.

2.2.1. Floor cooling cases

For Case 7 and Case 8, the heat to be removed by the floor was 876 W, and for Case 6 it was 1183 W. This corresponds to a cooling load of 19.5 and 26.3 W/m^2-cooled floor area, respectively, and a corresponding average floor surface temperature of 23.2 and 22.2 °C. In order to achieve these surface temperatures, the required supply and return water temperatures were 18.6 and 21.6 °C for Case 7 and Case 8, and 16.5 and 19.5 °C for Case 6. For all cases, the temperature difference between supply and return water flows was assumed to be 3 °C. For Case 7 and Case 8, this resulted in a mass flow rate of 250 kg/h, and for Case 6 it was 338 kg/h. A floor covering resistance of 0.05 m^2 K/W was assumed for all cases to keep the effects of floor covering resistance on the system performance to a minimum [15].

The cooling output, floor surface temperatures and the mass flow rates were calculated according to [19–22]. The summary of floor cooling cases is given in Table 2.

Table 2
Summary of the floor cooling cases.

Case	Space cooling load [W]	Supply and return water temperature [°C]	Cooled floor surface temperature [°C]	Water flow rate [kg/h]
6	1183	16.5/19.5	22.2	338
7 & 8	876	18.6/21.6	23.2	250

2.2.2. Air cooling cases

The required ventilation rates were calculated based on the space cooling loads and the temperature difference between the supply air and room air temperatures. The water flow rate in the air-cooling coil was calculated based on the heat to be removed from the intake air and the temperature difference in the supply and return water flows to and from the air-cooling coil. The heat to be removed from the intake air corresponds to the required amount of heat to lower the temperature of the intake air to the required supply air temperature, which was 14 °C, 17 °C or 20 °C for respective three cases. The summary of the air cooling cases is given in Table 3.

coil.

The coefficient of performance (COP) of the heat pump was obtained from the manufacturer's datasheets as a function of out-

Table 3
Summary of the air cooling cases (IA: intake air).

Case	Space cooling load [W]	Supply air temperature [°C]	Ventilation rate [ach]	Rate of cooling to IA [W]	Water flow rate in the air-cooling coil [kg/h]
1	3170	14	3.7	4226	725
2	1042	14	1.2	1389	238
3	1042	17	1.6	1505	258
4	1042	20	2.5	1736	298
5	1042	14	1.2	634	109

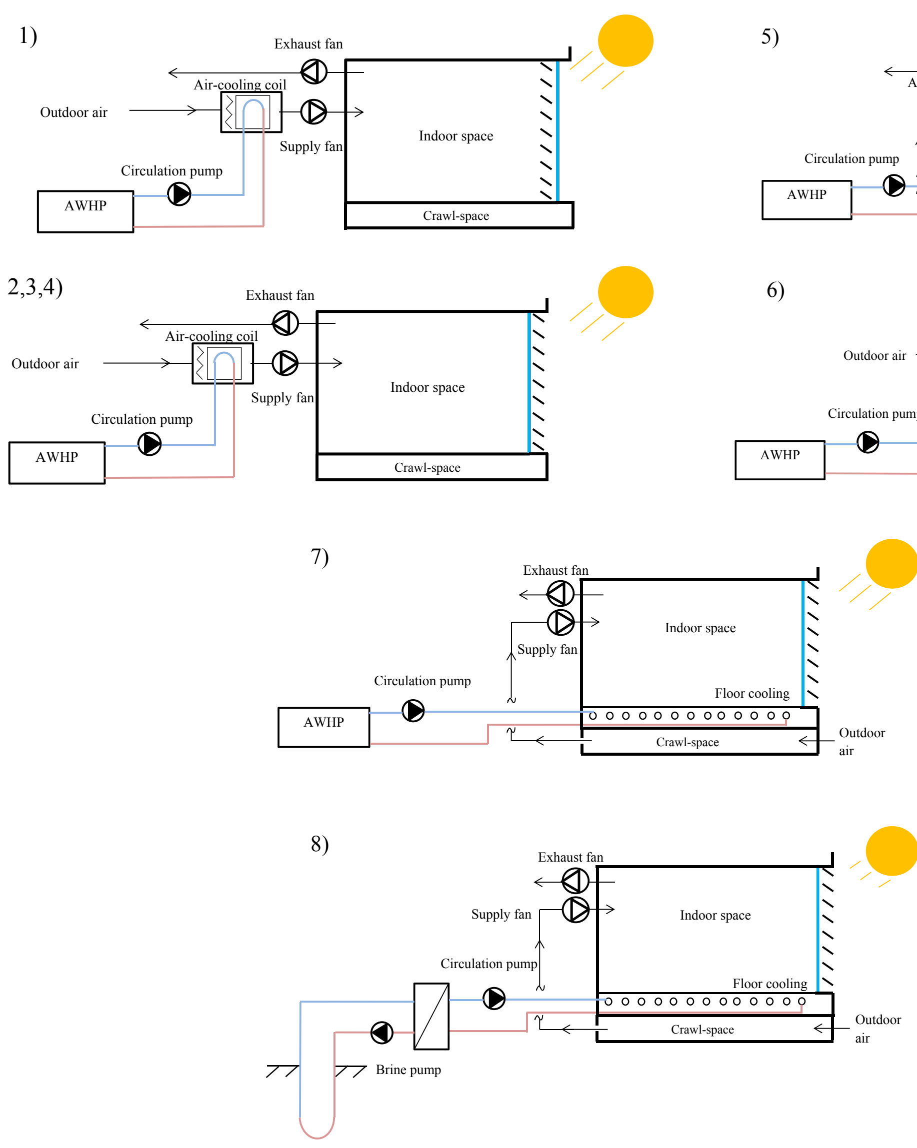

Fig. 1. Schematic drawings of the analyzed cooling systems.

2.2.3. Air-to-water heat pump, crawl-space, and ground heat exchanger

The temperature of the water leaving the evaporator of the air-to-water heat pump was assumed to be the same as the supply water temperature to the floor loops and to the air-cooling

door air temperature and the temperature of the water leaving the evaporator. For floor cooling cases, the required supply water temperature to the floor loops was used to obtain the COP, while for air cooling cases it was assumed that the supply and return water temperatures to and from the air-cooling coil were 7 and 12 °C, respectively. The resulting COP values were 3.42 for Case 7, 3.31 for Case 6, and 2.79 for air cooling cases.

In the actual house, the intake air was from the crawl-space and the measurements showed that the air in the crawl-space was warmer than the outdoor air in winter and colder than the outdoor air in summer [16].

Ground heat exchangers can be a good match to couple with high temperature cooling systems [19]. In this study, a single U-tube vertical heat exchanger was assumed to be coupled to the floor cooling system. It was assumed that there was a flat-plate heat exchanger between the floor system and the ground heat exchanger and a brine pump was circulating the anti-freeze mixture consisting of 30% propylene-glycol/water mixture.

The ground temperature of Copenhagen area was taken as 8.3 °C [23]. The incoming and outgoing liquid temperatures to and from the borehole were 17 and 13 °C, respectively. The corresponding mass flow rate in the borehole was 208 kg/h. It is possible to achieve the necessary cooling of the circulating liquid from 17 °C to 13 °C at the 40 m depth for the given borehole design. Further details of this ground heat exchanger are given in Refs. [23,24].

Since the temperatures of air in the crawl-space and of the deep ground are different from the outdoor (environmental) temperature, they contain a certain amount of cool (or warm) exergy and they act as immediate cool (or warm) exergy sources.

2.2.4. Fan and pump powers

The power to be supplied to fans and pumps that circulate the heat transfer medium in pipes or in ducts was determined as follows.

The pump power for different cases was obtained from the pump specifications as a function of the water flow rate and the required pressure increase, assuming the pump actually installed in the house.

The fan powers were determined from the measured data at the house. The measurements showed that the AHU was using 67.9 W with a ventilation rate of 0.5 ach (105 m^3/h), which corresponds to a specific fan power (SFP) of 1166 J/m^3 for one fan [16]. This SFP value is in SFP 3 category according to EN 13779:2007 [25]. The fan powers were calculated as a function of the air flow rates, assuming that the fans for the air cooling cases are also in SFP 3 category (1200 J/m^3).

Table 4 summarizes the pump, fan, and their total powers for eight cases studied.

3. Basic definitions of exergy and calculation methodology

3.1. Basic definitions

For any system, it is possible to obtain the exergy balance equation from energy and entropy balance equations together with the environmental temperature. In general form, exergy balance equation can be written as follows [13,26]:

Table 4
Summary of pump, fan, and their total powers for each case.

Case	E_{pump} [W]	E_{fans} [W]	E_{total} [W]
1	33.5	528.2	561.7
2	25.0	173.6	198.6
3	25.3	231.5	256.8
4	26.0	347.3	373.3
5	23.0	173.6	196.6
6	26.5	67.9	94.4
7 & 8[a]	25.2	67.9	93.1

[a] : The electricity input to the brine pump is not shown in this table, it is not considered as an auxiliary component but rather as a component similar to a heat pump, which is used to deliver the "coolness" from the ground to the floor loops.

$$[Exergy\ input] - [Exergy\ consumed] = [Exergy\ stored] + [Exergy\ output] \tag{1}$$

where [Exergy consumed] = [Entropy generated]·T_o, and T_o is the environmental (reference) temperature [K], where the system and its components are situated in. The storage term in Eq. (1) disappears for the analyses under steady-state conditions.

Eq. (1) indicates that every system consumes a part of the supplied exergy while at the same time the corresponding amount of entropy is generated.

A brief description of "cool" and "warm" exergy concepts are given in Appendix A. The following calculations were carried out manually and under steady-state conditions.

3.2. Cooling exergy load

The cooling exergy load is the required rate of exergy to be supplied to the indoor space to maintain the design indoor conditions, and it can be defined as

$$X_{cooling} = -Q_{cooling}\left(1 - \frac{T_o}{T_i}\right) \geq 0 \tag{2}$$

where $X_{cooling}$ is the cooling exergy load [W], $Q_{cooling}$ is the space cooling energy load [W], and T_i is indoor temperature (air and mean radiant temperatures) [K].

3.3. Exergy supplied to the indoor space

The exergy supplied to the indoor space through floor cooling and through the supply air can be calculated using Eqs. (3) and (4), respectively.

$$X_{FC,\ out} = -Q_{cooling}\left(1 - \frac{T_o}{T_{S,\ FC}}\right) \geq 0 \tag{3}$$

$$\Delta X_{AC,out} = V_{sa} c_a \rho_a \left\{ (T_{sa} - T_i) - T_o \ln\frac{T_{sa}}{T_i} \right\} \tag{4}$$

where $X_{FC,out}$ is the exergy supplied from floor cooling system to the indoor space [W], $T_{S,FC}$ is the average temperature of the cooled floor surface [K], $\Delta X_{AC,out}$ is the net exergy supplied by cold air to the indoor space (the difference in the amount of exergy carried 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 supply air temperature [K].

The exergy consumed within the indoor space can be obtained as the difference between the exergy supplied to the indoor space and the cooling exergy load.

3.4. Exergy input, output and consumption in the ground, flat-plate heat exchanger, floor, and air-cooling coil

In order to get a complete understanding of the exergy flows in the whole cooling system, it is necessary to start from the ground and to identify the exergy consumption processes. The net exergy input to the circulating anti-freeze mixture from the ground, ΔX_{ground} [W], is obtained from the following equation.

$$X_g - X_{c,\ ground} = \Delta X_{ground} \tag{5}$$

where

$$X_g = -Q_g\left(1 - \frac{T_o}{T_g}\right) \tag{6}$$

$$\Delta X_{ground} = V_g c_{pgw} \rho_{pgw} \left\{ (T_{g,out} - T_{g,in}) - T_o \ln \frac{T_{g,out}}{T_{g,in}} \right\} \tag{7}$$

X_g is the “cool” exergy flow rate from the ground to the anti-freeze mixture [W], $X_{c,ground}$ is the exergy consumption rate in the ground [W], Q_g is the rate of heat removed from the anti-freeze mixture to the ground [W], T_g is the ground temperature [K], V_g is the volumetric flow rate in the U-tube heat exchanger in the ground [m^3/s], c_{pgw} is the specific heat capacity of the anti-freeze mixture [J/kgK], ρ_{pgw} is the density of the anti-freeze mixture [kg/m^3], $T_{g,out}$ is the temperature of the anti-freeze mixture going out from the ground [K], and $T_{g,in}$ is the temperature of the anti-freeze mixture going into the ground [K].

The exergy consumption in the flat-plate heat exchanger, $X_{c,HEX}$ [W], is obtained from the exergy balance equation, Eq. (8).

$$\Delta X_{ground} - X_{c,\ HEX} = \Delta X_w \tag{8}$$

where

$$\Delta X_w = X_{w,supply} - X_{w,return} \tag{9}$$

ΔX_w is the net exergy input from the supply and return water [W], $X_{w,supply}$ is the rate of exergy carried by the supply water into the floor [W], $X_{w,return}$ is the rate of exergy carried by the return water from the floor [W].

The values of $X_{w,supply}$ and $X_{w,return}$ are calculated from the following equation.

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

where 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 supply or return water temperature [K].

The exergy consumption rate in the floor structure, $X_{c,floor}$ [W], is calculated from the following exergy balance equation.

$$\Delta X_W - X_{c,\ floor} = X_{FC,\ out} \tag{11}$$

where ΔX_w and $X_{FC,out}$ are given by Eqs. (9) and (3), respectively.

The exergy consumption in the air-cooling coil in the AHU, $X_{c,coil}$ [W], is obtained from

$$\Delta X_{w,coil} - X_{c,\ coil} = \Delta X_a \tag{12}$$

$$\Delta X_{w,coil} = X_{w,supply,coil} - X_{w,return,coil} \tag{13}$$

$$\Delta X_a = X_{a,out} - X_{a,in} \tag{14}$$

where $\Delta X_{w,coil}$ is the net exergy input by the water to the air-cooling coil [W], $X_{w,supply,coil}$ is the rate of exergy carried by the water entering the air-cooling coil (from the heat pump) [W], $X_{w,return,coil}$ is the rate of exergy carried by the water leaving the air-cooling coil (to the heat pump) [W], $X_{a,out}$ is the rate of exergy carried by the air leaving the air-cooling coil [W], and $X_{a,in}$ is the rate of exergy carried by the air entering the air-cooling coil [W].

Eq. (10) can be applied to the calculation of $X_{w,supply,coil}$ and $X_{w,return,coil}$. In the case of air instead of water, Eq. (10) is also applied with the replacement of the values of volumetric flow rate, specific heat capacity, density and respective temperatures from water to air. Eq. (10) is also used to calculate the rate of cool or warm exergy carried by the air flowing in from the crawl-space.

3.5. Exergy input to the power plant

It was assumed that the electric power supplied to the heat pump, pumps, and fans was generated in a remote, natural gas fired power plant. The exergy input required to the power plant can be determined from

$$E_{HP} = \frac{Q_{HP,\ cooling}}{COP} \tag{15}$$

$$X_{in,\ power\ plant} = \frac{E_{HP}}{\eta_{TOT}} r \tag{16}$$

where E_{HP} is power (electricity) input to the heat pump [W], $Q_{HP,cooling}$ is the rate of heat to be removed by the heat pump to the water circulating through the evaporator [W], COP is the coefficient of performance, $X_{in,\ power\ plant}$ is the exergy input to the power plant through natural gas [W], η_{TOT} is the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (assumed to be 0.35 [13]), and r is the ratio of chemical exergy to higher heating value of natural gas (assumed to be 0.93 [13]).

For the value of $Q_{HP,cooling}$, the space cooling energy load is used in the floor cooling cases, and the rate of heat to be removed from the intake air is used in the air cooling cases.

Exergy input required at the power plant for the pump and fans are calculated using Eq. (16) by replacing the E_{HP} with respective pump power (E_{pump}) and fan power (E_{fans}).

3.6. Exergy efficiency

One way of evaluating the exergy performance of cooling systems is to use exergy efficiency. Conventional definition of the exergy efficiency can be used but it may fail to capture the effects of exergy supply to the cooling system from the immediate natural exergy sources, such as the ground and the crawl-space. Therefore, three kinds of exergy efficiency as defined in Eqs. (17–19) were used:

$$\eta_{x,\ conventional} = \frac{X_{cooling}}{X_{in,power\ plant}} \tag{17}$$

$$\eta_x = \frac{X_{cooling}}{X_{in,power\ plant} + X_g + X_{crawl-space}} \tag{18}$$

$$\eta_{x,\ natural} = \frac{X_g + X_{crawl-space}}{X_{in,power\ plant} + X_g + X_{crawl-space}} = 1 - \frac{\eta_x}{\eta_{x,conventional}} \tag{19}$$

where $\eta_{x,conventional}$ is the conventional exergy efficiency, η_x is the exergy efficiency which takes into account the exergy supplied from the immediate natural exergy sources (one from the ground, X_g, and the other from the crawl-space, $X_{crawl-space}$ [W]), and $\eta_{x,natural}$ is the ratio of the exergy input from the immediate natural exergy resources to the total exergy input to the system.

4. Results and discussion

The main results of the analyses are presented in this chapter. Sensitivity of the results to the total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (η_{TOT}), to the SFP of fans, and to the brine pump power can be found in Appendix B. Sensitivity of the system exergy performance to these parameters is presented in Appendix C.

4.1. Comparison of different space cooling systems without a cool exergy source

The chains of exergy flows from the initial natural gas input to the power plant to the environment are shown in Fig. 2 for Cases 1, 2, 3, 4, and 6. For the power plant, the exergy contained by natural gas is supplied to the power plant as fuel and the electricity produced is supplied to the heat pump. The difference between the exergy input from natural gas and the output electricity is the exergy consumption in the power plant. The same relationship between input, output and consumption applies also to the other components in the chain.

Exergy inputs, outputs, and consumptions in different system components are given in Fig. 3. $X_{g,in}$ and $X_{g,out}$ are the rates of exergy carried by the anti-freeze mixture flowing into and out of the ground heat exchanger, respectively [W].

The exergy to be supplied to the power plant from natural gas is 4025 W for Case 1 and it is the largest among the investigated cases. This implies that the use of an internal solar shading device is not effective in reducing the cooling demand of the house. The large exergy input requirement is due to the large cooling load and also due to the way of addressing this load; cooling with an air-based system. Compared to the rest of the cases, exergy consumption in the rest of the system components is also the largest for Case 1. Case 1 clearly shows that it is crucial to minimize the cooling demand of the house.

In the rest of the cases in Fig. 2, an external solar shading device was employed. The exergy input required at the power plant has decreased remarkably compared to Case 1, and the exergy consumption in different system components has also decreased.

In Cases 2, 3 and 4, air cooling was employed to address the cooling load. Although the cooling exergy load itself is the same for all of these cases (13.9 W), the exergy required at the power plant in order to power the heat pump is different, due to different supply air temperatures assumed for each case (Table 3). The exergy consumption rate in the indoor space is 21.8 W, 16.2 W, and 10.7 W for Cases 2, 3 and 4, respectively. This trend is reversed for exergy consumption in the cooling coil, where the exergy consumption rates are 62.8 W, 76.0 W, and 96.7 W for Cases 2, 3 and 4, respectively. This is also reflected in the rest of the systems towards the source, where the differences between these cases in the heat

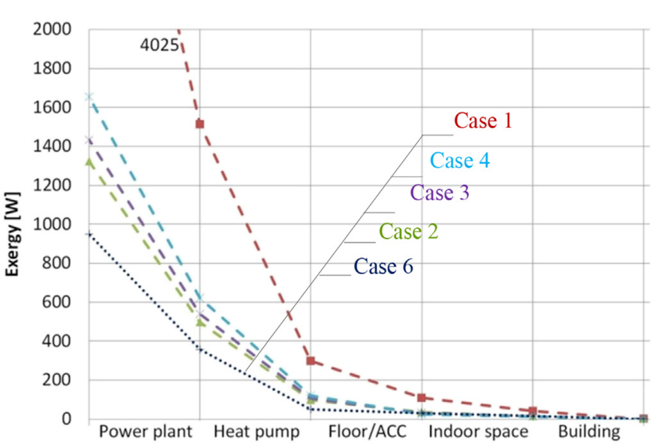

Fig. 2. Exergy flows for different cooling strategies without a cool exergy source (ACC: air-cooling coil).

pump and the power plant are clear in Fig. 2. Case 4 requires the highest exergy input among these three cases, followed by Case 3 and Case 2, therefore for the further analyses and comparisons, Case 2 will be used.

Among the cases presented in Fig. 2, Case 6 requires the lowest exergy input to the power plant, despite the cooling exergy load being slightly higher (15.8 W) than the air cooling cases. This is because of two reasons; the system being a water-based cooling system, and working at water temperatures close to room temperatures (high temperature cooling). The exergy input required at the power plant is 28% smaller for Case 6 (floor cooling) than Case 2 (air cooling).

4.2. The effects of immediate cool exergy sources on system performance

4.2.1. Cool exergy contained in the crawl-space

The crawl-space below the house acted as a buffer zone, where the air temperature was higher than the outdoor air temperature in the heating season and vice versa in the cooling season. This results in a warm or cool exergy storage effect in the crawl-space [12,13]. The outdoor air temperature, air temperature in the crawl-space, and the specific exergy contained by air in the crawl-space are shown in Fig. 4.

During the period from spring to autumn 2014, the maximum warm and cool exergy stored in the crawl-space were 131.1 J/m^3 and 287.9 J/m^3, respectively. Under the conditions considered in this study (air temperature in the crawl-space of 21.3 °C and an outdoor air temperature of 30 °C), the stored cool exergy density in the crawl-space corresponds to 152.7 J/m^3, which is lower than the maximum cool exergy density stored during this period. Fig. 4 shows that when the air temperature in the crawl-space is lower than the outdoor air temperature, there is cool exergy storage in the crawl-space.

Further cases were studied by modifying the boundary conditions of Case 6 and Case 2, in order to investigate the effects of this cool exergy storage on the whole system. That is, in Cases 7 and 5, it was assumed that the intake air was taken from the crawl-space, instead of the outdoor air. Fig. 5 shows the effects of the crawl-space (cool exergy storage) on the performance of different cooling systems.

In Fig. 5, the differences in the exergy input to the power plant between Case 2 and Case 5 (719 W, 54% reduction in exergy input to the power plant compared to Case 2), and between Case 6 and Case 7 (270 W, 29% reduction in exergy input to the power plant compared to Case 6) are due to the use of the cooler air at 21.3 °C in the crawl-space instead of the outdoor air at 30 °C as intake air. The rate of cool exergy provided from the crawl-space with the ventilation rates of 0.5 ach (Cases 6 and 7) and 1.2 ach (Cases 2 and 5) are 4.5 W and 10.7 W, respectively. The exergy consumption in the power plant, heat pump, floor, and in the indoor space is also decreased for Case 7 compared to Case 6. The exergy consumption in the power plant, heat pump, and cooling coil is decreased for Case 5 compared to Case 2.

It is worth noting that only 4.5 W and 10.7 W of cool exergy provided from the crawl-space can eliminate 270 and 719 W of exergy input by natural gas at the power plant, respectively. This is mainly because the actual cool exergy demand is very small so that making use of such small quantities of cool exergy results in a significant reduction on the supply side.

Fig. 5 shows that the air-based systems benefit more from the storage of cool exergy in the form of cooler air in a crawl-space compared to floor cooling. Although Case 5 requires less exergy input than Case 7, this does not mean that the air-based system performs better than the water-based system, due to the higher

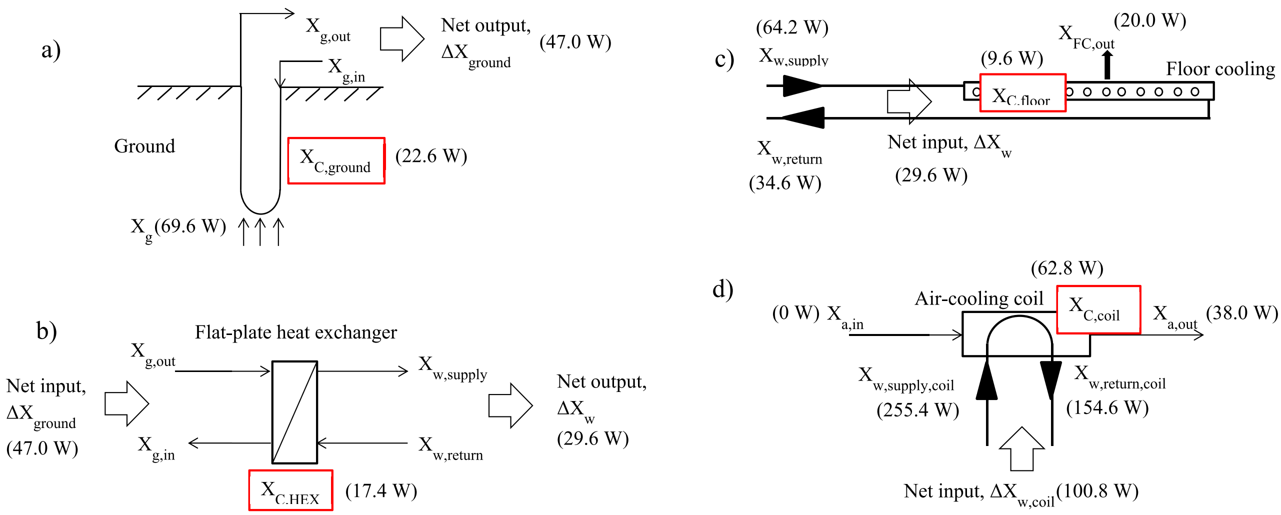

Fig. 3. Exergy input, output, and consumption in different system components: a) Ground b) Flat-plate heat exchanger c) Floor d) Air-cooling coil. a), b), and c) are given for Case 8. d) is given for Case 2. Values in the parentheses indicate the exergy values and the red boxes indicate exergy consumptions in the respective system components. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

auxiliary energy use, as will be presented in 4.3. It should also be noted that compared to Case 7, Case 5 has higher exergy consumption in the cooling coil (19.0 W) compared to the floor structure (9.6 W), higher exergy consumption in the indoor space (21.8 W vs. 8.3 W), and has a higher cooling exergy load (13.9 W vs. 11.7 W).

4.2.2. *Cool exergy contained in the ground*

Although the floor cooling performs better than the air cooling cases as presented in Fig. 2, a closer look at the exergy flow reveals that it is possible to increase the exergy performance of this system with a better match of the exergy demand and supply. This is achieved through the coupling of the floor cooling system with a ground heat exchanger. The results are presented in Fig. 6.

The brine pump in the ground loop was assumed to be identical in terms of performance to the circulation pumps. The brine pump has a similar function to a heat pump, which transports the cool exergy from its source (ground) to where it is needed. This is also shown in Fig. 6, where the orange line (Case 8, X_g) is the flow of cool exergy from the ground through the system components, house and to the environment. The green line (Case 8, $X_g + X_{NG}$) shows, in addition to the cool exergy flow from the ground, the exergy input at the power plant from natural gas to provide the brine pump with electricity.

When comparing Case 7 to Case 8, the exergy input at the power plant is decreased from 680 W (for heat pump) to 65 W (for brine pump) corresponding to a 90% reduction. This difference is due to the use of the cool exergy available in the ground. When considering also the cool exergy that was initially available in the ground (69.6 W), then the total exergy input is 134.6 W, as given in Fig. 6 (Case 8, $X_g + X_{NG}$). Compared to Case 7, the use of ground without any thermodynamic refrigeration cycle (the use of stored cool exergy available in the ground) is an effective way to match the low exergy demand of the floor cooling system.

Regarding the exergy flow of the AWHP, a possible improvement could be to power it with on-site generated electricity from a renewable energy resource instead of a power plant in a remote location, though this requires further considerations for a definitive conclusion.

The pump power is important for realizing the true benefits of the cool exergy stored in the ground, as it is an addition to the cool exergy from the ground (cool exergy of 69.6 W from the ground is comparable to the 65 W of exergy by natural gas at the power plant to provide the brine pump with 24.5 W of electricity). This is crucial for justifying the free cooling. Increasing pump power requirements (e.g. to use a deeper or additional ground heat exchangers than one borehole or a worse pump) will decrease the overall efficiency of the system, as shown in Appendix C.

The initial design of the heating and cooling system of the house relied on the ground as the heat source and sink, and a theoretical single U-tube ground heat exchanger was designed. The benefits of using the ground compared to an AWHP were justified in energy terms in previous studies [23,24] and the results obtained in this study justify this solution from the exergetic viewpoint.

Case 8 takes advantage of the cool exergy available both in the crawl-space (4.5 W) and in the ground (69.6 W). When comparing Case 6 to Case 8, the exergy input required at the power plant is decreased from 950 W (for heat pump) to 65 W (for brine pump) corresponding to a 93% decrease. This result emphasizes the benefits of using the naturally available heat sources and sinks in our immediate surroundings, although, a prerequisite of this is to limit the cooling (and heating) demand of the building as much as possible from the beginning of the design process. This implies that passive and active technologies should be well combined.

4.3. *Auxiliary exergy input*

In addition to the thermal exergy analyses, the effects of electricity inputs to pumps and fans on the whole exergy consumption patterns were also considered. The results calculated for all cases are presented in Fig. 7. In each case, there are four bars, which are, from left to right, the exergy input to pump, to fans, their total, and the exergy input to the power plant.

The results presented in Fig. 7 show that although the pump consumptions are within a close range, the fan consumptions vary greatly from floor cooling to air cooling cases. In the floor cooling cases, the ventilation system was only used to provide the necessary fresh air, while in air cooling cases, the ventilation system was used to remove all the necessary heat (cooling load), which resulted in larger ventilation rates. The large ventilation rates result in a relatively large auxiliary energy use compared to the floor cooling cases and this difference can be attributed to the difference between the air-based and water-based cooling approach. A high auxiliary energy use would decrease the energy and exergy-wise efficiency of the whole system (Appendix C).

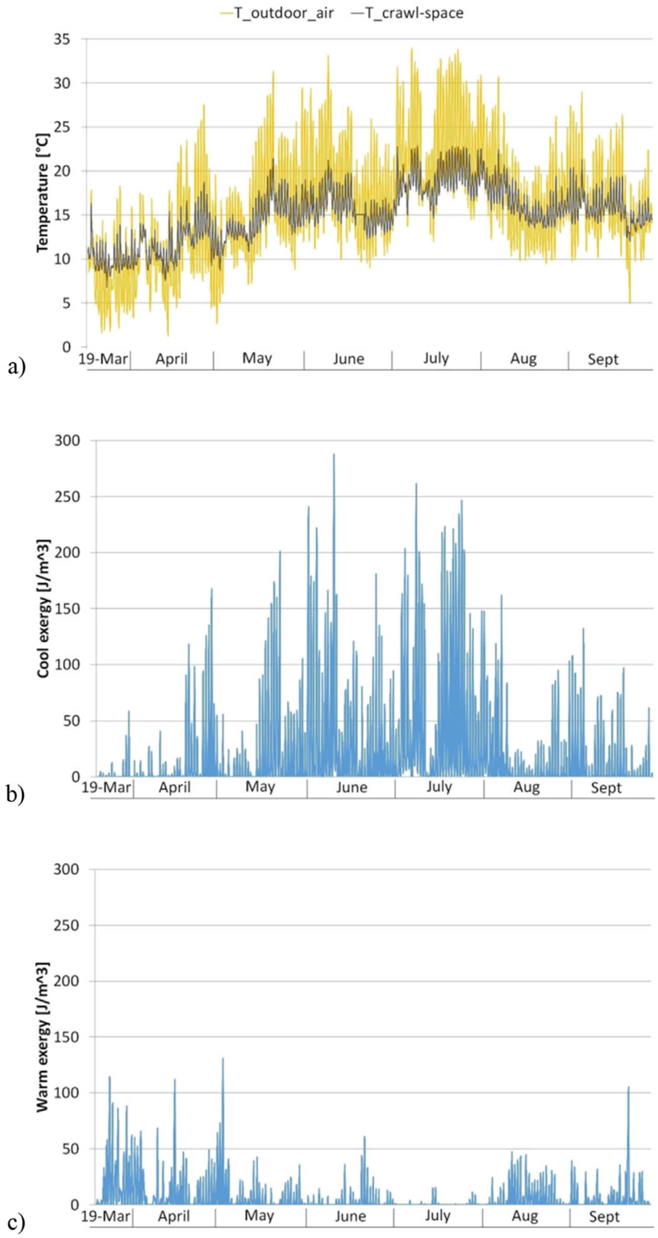

Fig. 4. a) Air temperatures outdoors and in the crawl-space b) Specific cool exergy contained by the air in the crawl-space c) Specific warm exergy contained by the air in the crawl-space.

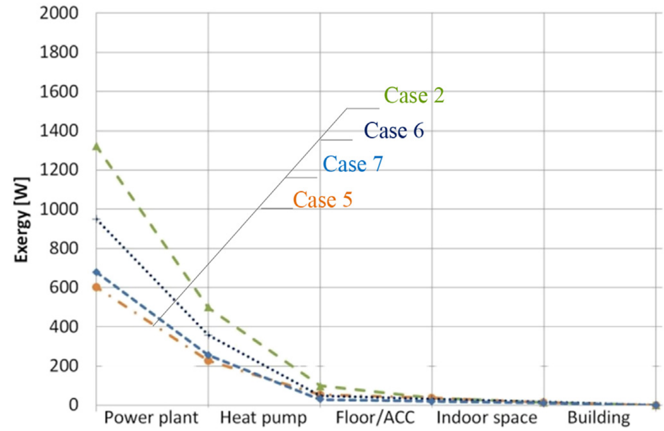

Fig. 5. Exergy flows for different cooling strategies with available cool exergy from the crawl-space (ACC: air-cooling coil).

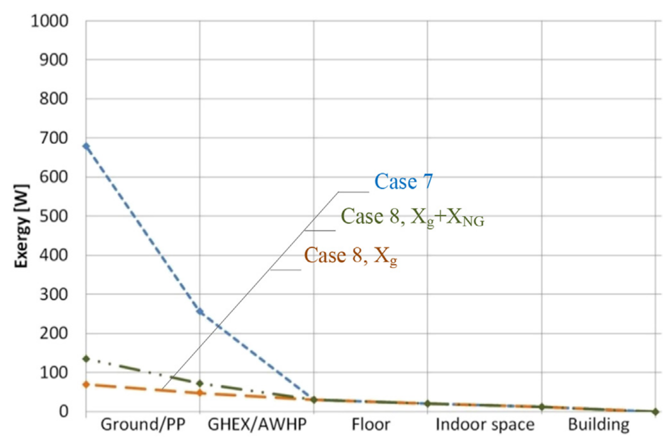

Fig. 6. Exergy flows for different cooling strategies with available cool exergy from the ground (PP: power plant).

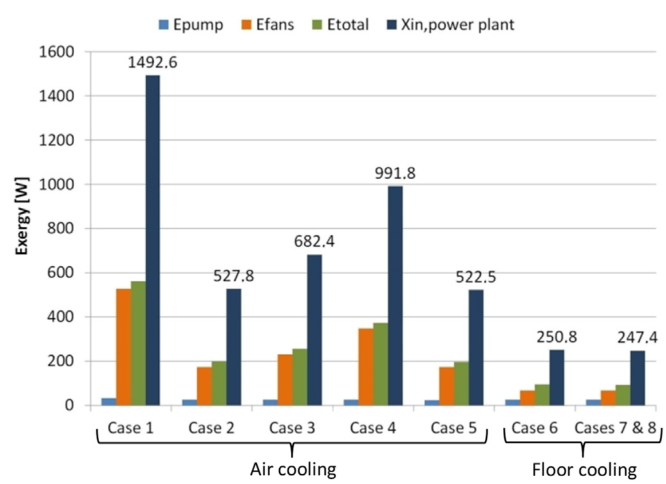

Fig. 7. Exergy inputs to the circulation pump, supply and exhaust fans and to the power plant.

Air-based systems require larger flow rates and volumes to transport the same amount of heat, cool or warm exergy compared to water-based systems because of the air's smaller specific heat capacity and density than water. This causes a larger power requirement for air to be transported and it emphasizes the advantage of water-based heating and cooling systems.

Fig. 7 shows that the auxiliary exergy input to the system can be substantial compared to the thermal exergy values, as presented in Figs. 2, 5 and 6. These results emphasize the importance of minimizing the auxiliary component exergy use in order to achieve a holistically high performing system.

4.4. Exergy efficiency

In addition to the total exergy input to the systems, exergy efficiency can also be used to evaluate the overall exergetic performance of heating and cooling systems. The resulting exergy efficiencies for the studied cases are given in Table 5.

The exergy efficiencies given in Table 5 do not include the exergy input to the auxiliary components; the inclusion of the auxiliary components in the exergy efficiency will decrease the exergy efficiencies. Appendix C presents the exergy efficiency values when the auxiliary components are also considered. The effects of specific fan power, and brine pump power on overall system performance can also be found in Appendix C.

Exergy efficiency values show that the floor cooling (Case 6) has slightly higher exergy efficiency than the air cooling (Cases 1 to 4) when the effects of immediate natural exergy sources are not considered. When the effects of naturally available resources are considered, the exergy efficiencies increase as in Case 5 compared to Case 2, and Case 8 compared to Case 6. This is mainly due to the decreased exergy input into the power plant to provide the required space cooling, and through a better match of the exergy demand and supply (Case 8).

The natural exergy efficiency values ($\eta_{x,natural}$) clearly show that only Cases 5, 7 and 8 benefit from the available natural exergy sources in the immediate surroundings. In Case 8, 53.2% of the total exergy input to the system was provided by the natural exergy sources, whereas this value was 1.8% for Case 5 and 0.7% for Case 7. According to this definition, in case of a building using conventional heating and cooling systems, which does not use any immediate sources of natural exergy (as in Cases 1 to 4, and 6), $\eta_{x,natural}$ becomes zero. If in a building (e.g. a zero energy building), all the exergy requirement is supplied with solar, ground, wind, etc., this means that the exergy supply to the building is 100% sustainable and $\eta_{x,natural}$ becomes unity.

The results show that the heat sinks within our immediate environment should be used wisely, since these are valuable natural resources and they should not be exhausted through poor utilization policies. This is partly reflected in the exergy efficiency of Case 8; when the cool exergy consumption from the ground is taken into account then the exergy efficiency decreases to 8.4% from 18.0%. This implies that the rational efficiency index is η_x rather than $\eta_{x,conventional}$.

4.5. Overall discussion

This study considered a cooling season operation but it is crucial to assure that the given systems can work effectively in the heating season in residential buildings. Exergy analyses and especially the warm exergy concept can be used to analyze the performance of heating systems in buildings. The exergy performance of different heating systems has been addressed in another publication [15].

Certain assumptions have been made during the calculation procedure. The steady-state assumption brings certain limitations especially regarding the consideration of thermal storage effects and transient behavior of buildings. There are also certain limitations regarding the occupant presence and comfort throughout the day, since different control strategies (e.g. setback control) can be used in a single day. The current calculations do not address these issues, dynamic exergy calculations or exergy calculations based on building performance simulations (indoor environment and energy) can be used to study these effects.

The obtained results are dependent on the values used in the calculations (e.g. power plant efficiency, pump and fan powers, etc.) therefore it is crucial to choose realistic values for each parameter. In this study, the necessary values were obtained from literature or from the actual house components, when applicable. The uncertainties associated with these assumptions were examined and confirmed by sensitivity analysis, and the results are described in Appendix B.

For the air cooling cases, it was chosen to keep the same supply and return water temperatures from the heat pump and to vary the water and air flow rates, corresponding to a change in pump and fan powers. Another approach could have been to vary the supply and return water temperatures from the heat pump, and hence to improve the COP. Latent loads were not considered and it was assumed that the cooling in the air-cooling coil was sensible.

The effect of cooling system on thermal indoor environment was not directly studied; rather a comparison of exergy performance of different cooling systems was made based on the same level of comfort. The method used to condition spaces (chosen terminal units, air-based or water-based system) has a direct influence on the thermal comfort of the occupants. Human thermal comfort should not be sacrificed for energy- and exergy-wise efficiency improvement alone and they should be achieved simultaneously. Water-based radiant systems perform better compared to other systems in terms of occupant thermal comfort; they minimize the risk of unpleasant air movement (draught), and they create a uniform temperature distribution in spaces [27,28].

The use of a crawl-space was applicable for this particular house but it might not always be applicable. The crawl-space has a similar function to an earth-tube dug into the ground, in fact an earth-tube also benefits from the cool exergy available in the ground. Radon should be considered for different locations, i.e. a radon barrier might be needed in locations where radon is a concern. In addition to radon, also the quality of air coming from the crawl-space can cause problems with the indoor air quality and it is crucial to consider this aspect in order not to cause any dissatisfaction to the occupants with the indoor environment. The moisture of the air coming in from the crawl-space can also be a problem, and the consideration of dehumidification needs may result in different exergy inputs and system performance. The moisture of the air from the crawl-space could cause problems regarding condensation on the floor cooling unless properly dehumidified. The condensation can be avoided through a dew point control on the water supply temperature to the floor cooling loops.

The usability of the heat sources and sinks that are found in our immediate environment (e.g. ground, lake, sea-water, etc.) will depend on the costs, regulations, and on geographical conditions. Nevertheless, the minimization of the cooling (and heating) demand is crucial since only with a reasonably low exergy demand, it would be possible to use the naturally available exergy sources and sinks in our surroundings.

A high cooling load would limit the use of radiant systems (floor, ceiling or wall cooling) due to dew point concerns. The required low water temperatures might also limit the use of natural heat sinks and hence the use of a refrigeration cycle might become necessary.

If radiant systems are to be used with a high cooling load, then the air must be dehumidified, although this is not the optimal

Table 5
Exergy efficiencies of different cases.

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
$\eta_{x,conventional}$ [%]	1.1	1.1	1.0	0.8	2.3	1.7	1.7	18.0
η_x [%]	1.1	1.1	1.0	0.8	2.3	1.7	1.7	8.4
$\eta_{x,natural}$ [%]	0	0	0	0	1.8	0	0.7	53.2

choice since it means that at another part of the system, water temperatures below the dew point are necessary to dehumidify the air, unless the air is dehumidified by other means (e.g. desiccant wheel). This operation strategy would partly off-set the benefits of high temperature cooling by the radiant floor cooling.

The optimal system design should be the one, in which the cooling demand is lowered as much as possible so that there is no need for dehumidification. Then high temperature cooling systems can be used which would enable the integration of heat sinks that can be found in our vicinity and this would increase the overall energy- and exergy-wise efficiencies.

5. Conclusion

The exergy performances of different space cooling systems were compared, using a single-family house as a case study. The main conclusions from the analyses are as follows.

1. Cooling exergy demand of the building should be minimized from the beginning to achieve reasonable exergy efficiency and also to allow the use of naturally available exergy sources and sinks. The cooling demand also influences the choice of indoor terminal units e.g. with a high cooling load, it might not be possible to use radiant cooling systems due to dew point concerns. The choice of the terminal unit is crucial since only with certain terminal units, it is possible to use natural resources, and since this choice directly affects the occupant thermal comfort.
2. The water-based radiant floor cooling system performed better than the air-based cooling with ventilation in terms of energy, exergy demand and consumption. When an air-to-water heat pump was used as the cooling source and the intake air was outside air, the exergy input required at the power plant was 28% smaller for the floor cooling system compared to the air cooling system. The water-based systems had remarkably smaller auxiliary exergy input compared to air-based systems, due to the use of water as the main heat carrier medium.
3. Cool exergy concept was used to quantify the available exergy in the crawl-space and in the ground. Integration of these natural exergy resources to the cooling system resulted in significant improvements in the system performance. The use of the cool exergy available in the crawl-space resulted in 54% and 29% smaller exergy input to the power plant for the air-based and water-based cooling systems, respectively. In these cases, only 4.5 W and 10.7 W of cool exergy provided from the crawl-space decreased the exergy input by natural gas to the power plant by 270 and 719 W, respectively.
4. The coupling of ground with the radiant floor cooling system is feasible since the exergy supply from the ground matches well with the low exergy demand of the floor cooling system. For floor cooling cases, it is possible to reduce the exergy input to the power plant by 90% and 93%, with the use of ground, and use of the ground and the crawl-space, respectively. Brine pump power should be kept to a minimum to truly benefit from the "free" cooling through the cool exergy stored in the ground.
5. The benefit of coupling the ground with the floor cooling system was shown with the exergy efficiency values; 18% and 8.4% for the conventional and the new definition of the exergy efficiency, respectively, and 53.2% of the total exergy input to the system was from the ground. The decrease in efficiency indicates that the natural resources within our immediate surroundings should be used efficiently, and not exploited in ineffective ways through poor utilization.

It should be noted that the obtained results are case-specific, i.e. based on the house design, location, and steady-state assumption, and, therefore the results can differ as a function of these factors.

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Appendix A. Definitions of cool and warm exergy

Thermal exergy can be categorized into two types as "cool" and "warm" exergies and this approach enables us to properly consider the "warmth" or the "coolness" of a heat source or sink [12,13]. An example is as follows: two tanks containing water are placed in an environment with an environmental temperature of T_o. One of the tanks is at a temperature of T_h and the other one is at a temperature of T_c, where $T_h > T_o$ and $T_c < T_o$. In the former case, the flow of energy and exergy are from the tank at T_h to the environment at T_o and this exergy corresponds to the flow of "warm" exergy, while in the other tank, the flow of energy is from the environment at T_o to the tank at T_c but the flow of exergy is from the tank to the environment and this flow of exergy is the "cool" exergy. It could be explained that, in cooling season, what is expected as a merit from this chilled water tank is the "cool" exergy and not the energy (the chilled water has a "lack" of energy). Further examples and more detailed descriptions of "cool" and "warm" exergies can be found in Refs. [13,14].

Appendix B. Sensitivity analysis

In order to quantify the effects of different assumptions on the results, sensitivity analyses were carried out on total efficiency including conversion efficiency of the power plant, distribution and transmission efficiencies of the grid (η_{TOT}), SFP of fans and on the brine pump power.

For the total efficiency (η_{TOT}), values of 0.3, 0.35 (used value), 0.4 and 0.45 were considered. Table B.1 shows the results. In Table B.1, the values indicate the exergy inputs to the respective system components for different cases.

Table B.1
Exergy input to system components and to the power plant as a function of total efficiency

		Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 8
									X_g	$X_g + X_{NG}$
Building [W]		42	14	14	14	14	16	12	12	12
Indoor space [W]		109	36	30	25	36	31	20	20	20
Floor/ACC [W]		300	98	106	121	55	49	30	30	30
Heat pump [W]		1515	498	539	622	227	358	256	47	72
Power plant [W]	η_{TOT}=0.35[a]	4025	1323	1433	1654	604	950	680	70	135
	0.3	4695	1543	1672	1929	704	1109	793	70	146
	0.4	3522	1158	1254	1447	528	831	595	70	127
	0.45	3130	1029	1115	1286	469	739	529	70	120

[a] Values presented for $\eta_{TOT} = 0.35$ correspond to the values presented in Figs. 2, 5 and 6.

The results of the sensitivity analysis for the total efficiency show that although the absolute values of exergy input to the power plant change, the relative effects and relative system performances are the same.

Although not shown in Table B.1, the variation of power plant efficiency will also affect the necessary exergy input to the power plant for auxiliary components: a lower power plant efficiency results in a larger exergy input being necessary to supply the same power for the pumps and fans.

For the SFP of the fans, values of 1200 J/m^3 (used value), 1000 J/m^3, 750 J/m^3 and 500 J/m^3 were considered. These values are in SFP 3, SFP 2 and SFP 1 categories according to EN 13779:2007, respectively. The SFP values in Cases 6, 7 and 8 were not changed, since they use the measured values from the house [16]. The SFP values were only varied for the air cooling cases and the results are given in Table B.2. Pump powers were not changed; therefore the differences are only due to the variation of fan SFP.

Table B.2
Exergy input to the power plant for auxiliary components as a function of fan SFP

	SFP	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7 & 8
$X_{in,\ power\ plant,\ aux}$ [W]	1200 J/m^3 [a]	1493	528	682	992	522	251	247
	1000 J/m^3	1259	451	580	838	446	251	247
	750 J/m^3	966	355	452	646	349	251	247
	500 J/m^3	674	259	324	454	253	251	247

[a] Values presented for SFP = 1200 J/m^3 correspond to the values presented in Fig. 7.

The results of the sensitivity analysis on fan performance show that even if the SFP were to be 500 J/m^3, the air-based systems cannot perform better than the water-based system in terms of exergy input required at the power plant for the auxiliary components. Even though the total exergy input required at the power plant becomes close for Cases 2 and 5 to water-based cooling systems' performance, it should be noted that water-based cooling cases used the actual measured values from the house in terms of fan energy use and therefore show the actual fan performance. Choosing better performing fans, as in this case with lower SFP values, will require other considerations, e.g. other products, costs, etc.

For the brine pump power, values of 10 W, 24.5 W (used value), 50 W and 75 W were considered. Table B.3 shows the effects of brine pump power on system performance.

Table B.3
Sensitivity of the results to the brine pump power in Case 8

	X_g [a]	$X_g + X_{NG}$ 24.5 W [a]	10 W	50 W	75 W
Building [W]	12	12	12	12	12
Indoor space [W]	20	20	20	20	20
Floor [W]	30	30	30	30	30
GHEX [W]	47	72	57	97	122
Ground/power plant [W]	70	135	96	202	269

[a] Values correspond to Fig. 6.

The results show that with an over-dimensioned pump, the whole system performance decreases; this indicates that in order to fully benefit from the available cool exergy in the ground, the dimensioning of the brine pump should be done carefully and the brine pump power should be minimized. An over-dimensioned pump will also mean reduced savings with the use of ground, compared to a system which uses an air-to-water heat pump, as in Fig. 6.

A sensitivity analysis on the power of the circulation pump was not carried out, since the characteristics of this pump were obtained from the actual component installed in the house. The pump has also a relatively small influence compared to the fan power.

The heat pump used in the calculations was not subject to a sensitivity analysis, because its performance was determined from tabulated data provided by the manufacturer as explained in 2.2.3. This allows the systems to be compared on a fair basis, using the same heat pump, and therefore reflecting the effects of actual operating conditions of air-based and water-based systems, e.g. the effects of water temperature leaving the evaporator of the heat pump.

It should be noted that heat pump plays a crucial role in the results and the effects of a given heat pump on the whole system performance should be considered for each case. The results presented in this study were obtained for one particular air-to-water heat pump.

The rest of the parameters used in the calculations either belonged to the actual design of the house or were actual components used in the house, therefore, further sensitivity analyses were not carried out.

Appendix C. Whole system exergy efficiency

Other than the total exergy input to the system (thermal exergy and for the auxiliary components), exergy efficiency can also be used to evaluate system performance. In addition to the efficiencies described based on thermal exergy values in 3.6, another exergy efficiency which takes the effects of auxiliary exergy use on the whole system performance into account was defined as follows.

$$\eta_{x,\ system} = \frac{X_{cooling}}{X_{in,power\ plant} + X_{in,\ power\ plant,\ aux}} \tag{C.1}$$

where $\eta_{x,system}$ is the whole system exergy efficiency, and $X_{in,\ power\ plant,\ aux}$ is the exergy input to the power plant through natural gas to provide the necessary electricity to the auxiliary components [W] (calculated as defined in 3.5 and the values are given in Fig. 7).

Table C.1 shows the whole system efficiencies for different cases.

Table C.1
Whole system exergy efficiencies for different cases

	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
$\eta_{x,system}$ [%]	0.77	0.75	0.66	0.53	1.24	1.32	1.26	3.75
$\eta_{x,system}'$ [%] [a]	0.9	0.88	0.79	0.66	1.63	1.44	1.42	5.57

[a] SFP = 500 J/m^3 for all systems, including the water-based cooling systems.

The results show that when the intake air is the outdoor air, water-based systems perform better than the air-based systems, i.e. they have a higher efficiency. The system performances are considerably affected by the availability of nearby natural heat sinks. When the cold air from the crawl-space is used as the intake air, the air-based system (Case 5) performs close to the water-based system coupled to a heat pump. When the water-based system is coupled to a ground heat exchanger (Case 8), it performs considerably better than all other system configurations.

If all systems have an SFP of 500 J/m^3, the overall performance of air-based systems increases compared to the previous cases. The water-based systems still perform better than the air-based systems when the intake air is the outside air.

When the intake air is from the crawl-space for the air-based system (Case 5), the air-based system performs better than the water-based system coupled to a heat pump; this is because the air-based system benefits from the available cool exergy in the crawl-space while the water-based systems either do not benefit from it (Case 6) or benefit in a limited amount compared to the air-based system (Case 7). These results also match with the results presented in 4.2.1. When the heat pump is replaced with a ground heat exchanger in water-based systems, water-based system performs considerably better than any other system configuration.

In order to examine the effects of brine pump power on the system exergy efficiency, two of the values given in Appendix B, 10 W and 75 W, were considered. With these values, the system exergy efficiency ($\eta_{x,system}$) turns out to be 4.28% and 2.62%, respectively. When an SFP of 500 J/m^3 is used ($\eta_{x,system}$'), the system exergy efficiency becomes 6.83% and 3.40%, respectively. Although these values are still considerably higher than the efficiencies of other systems, they also indicate that the brine pump power is a crucial parameter to be minimized, in order to properly and fully benefit from the available exergy in the ground.

The results show that the auxiliary exergy has a considerable effect on the overall system performance and therefore it should be considered carefully and minimized, in order to achieve an optimal system performance and to benefit from the naturally available heat sinks.

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Comparison of air-based and water-based space heating and cooling systems using entransy

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Abstract

Indoor spaces in buildings can be heated or cooled with different indoor terminal units. The choice of indoor terminal unit has significant effects on the overall energy performance of a heating and cooling system and on occupant thermal comfort.

There are different tools that can be used to evaluate the performance and to provide improvement suggestions to heating and cooling systems. This paper reports a theoretical investigation of the performance of different heating and cooling systems based on entransy analysis. The systems were compared in terms of entransy dissipation. The analyzed systems were assumed to be installed in a single-family house. These systems were warm-air heating with or without heat recovery on exhaust, radiator heating with different working temperatures, and floor heating with different floor covering resistances. In cooling case, the performance of air cooling was compared to floor cooling.

Floor heating system with low thermal resistance of the floor covering had the lowest entransy dissipation among the heating cases: 49% lower compared to the radiator heating with low supply and return water temperatures, and 59% lower compared to warm-air heating with heat recovery.

In cooling operation, when the air intake was from outdoors, floor cooling system had 48% lower total entransy dissipation compared to air cooling. When the intake air was from the crawl-space, floor cooling had 36% lower total entransy dissipation compared to air cooling.

The results show that low temperature heating and high temperature cooling systems perform superior compared to other space heating and cooling systems.

Keywords

Entransy dissipation; low temperature heating; high temperature cooling; warm-air heating; air cooling; radiator heating

Nomenclature

c_p	specific heat capacity [J/kgK]
En	entransy density [JK/m^3]
$\dot{En}_o$	rate of obtained entransy [WK]
$\dot{En}_s$	rate of supplied entransy [WK]
$\dot{En}_\phi$	rate of dissipated entransy [WK]
k	thermal conductivity [W/mK]
$\dot{m}$	mass flow rate [kg/s]

$\dot{q}$	heat flux [W/m]
Q	heat [W]
T	temperature [°C]
ε	effectiveness [-]
ρ	density [kg/m^3]

1. Introduction

Buildings are complex structures where several different systems interact with each other. Mechanical systems (heating, cooling and ventilation) are important components in buildings due to their direct effects on occupant thermal comfort, energy use and on environment on a global scale.

Indoor terminal units are a part of heating, cooling and ventilation systems in buildings and they can be defined as active building components emitting or removing heat (and moisture, depending on the system) to or from indoors. They are mainly air-based (relying on convection) or water-based (relying on a combination of radiation and convection) [1]. Terminal units are the closest components of the heating and cooling system to the occupants and therefore they determine, together with the building envelope, the indoor thermal conditions and occupant thermal comfort.

During the design, simulation, testing and operation phases of heating, cooling, and ventilation systems, there is a need to have accurate analysis methods, in order to evaluate energy performance, find ways to improve energy efficiency and to quantify the energy and emission (CO_2 and other greenhouse gases) saving potentials. Different evaluation methods provide different insights into the system under consideration.

Different methods have been used to study the performance of heating and cooling systems from different aspects. The European Energy Performance of Buildings Directive (EPBD) uses primary energy or CO_2 emission to evaluate energy efficiency. International Energy Agency's (IEA) Energy in Buildings and Communities Programme (EBC) Annex 37 (Low Exergy Systems for Heating and Cooling in Buildings) [2] and Annex 49 (Low Exergy Systems for High-Performance Buildings and Communities) [3] both used exergy as a measure of energy efficiency.

A recent annex from IEA EBC, Annex 59 (High Temperature Cooling and Low Temperature Heating in Buildings) [4], [5], has used entransy as another tool to study heating and cooling systems in buildings. The annex aimed at finding ways of minimizing temperature differences in HVAC systems for high energy efficiency in buildings using entransy analysis. Entransy is defined as an object's heat transfer ability stemming from the analogy between heat conduction and electrical conduction [6], [7].

Several studies compared the performances of water-based vs. air-based heating and cooling systems using energy, and exergy [8], [9], [10], [11], [12], [13]. So far, the application of entransy in analyzing buildings and building components is limited and only a few studies have been carried out. Zhang et al. [14], [15] developed the entransy analysis method for different HVAC system components and for heat transfer mechanisms in indoor spaces. Zhang et al. used entransy analysis to compare the performance of an air-based cooling system versus a radiant floor cooling system in an airport. The results of Zhang et al. show that entransy analysis can also be applied to study heating and cooling systems in buildings.

The present study reports the results of entransy analysis applied to different space heating and cooling systems, with a focus on indoor terminal units, using a single-family house as a case study. It was assumed that the house was heated or cooled with different systems. The studied systems were warm-air heating with or without heat recovery, radiator heating with different working temperatures, and floor heating with different floor covering resistances. In the cooling case, air cooling with different supply air temperatures to indoors with different air intakes (from outdoors or from the crawl-space beneath the house), and floor cooling (with different space cooling loads and with different heat sinks) were studied. For air cooling cases, the effects of internal solar shading vs. external solar shading on system performance were also investigated. The results are reported with the use of temperature-transferred heat (T-Q) diagrams and in terms of entransy dissipation.

2. Basic definition of entransy

Before different heating and cooling systems can be analyzed based on entransy, the entransy and entransy dissipation concepts should be clarified.

In recent years, entransy has been proposed as a new tool to evaluate heat transfer processes that do not involve heat-to-work conversion. Entransy is defined as an object's ability to transfer heat (also referred to as the potential energy of its heat) stemming from the analogy between heat conduction and electrical conduction. This analogy is explained in details in [6].

The entransy balance equation can be obtained from the energy conservation equation for heat conduction, without a heat source:

$$\rho c_P \frac{\partial T}{\partial t} = -\nabla \cdot \dot{\boldsymbol{q}} = \nabla \cdot (k \nabla T) \tag{1}$$

Multiplying both sides by T and writing the equation in terms of entransy results in entransy balance equation:

$$\rho c_P T \frac{\partial T}{\partial t} = \nabla \cdot (k \nabla T) T \tag{2}$$

$$\frac{\partial En}{\partial t} = \nabla \cdot (T k \nabla T) - E\dot{n}_{\phi} \tag{3}$$

Entransy density is then defined as

$$En = \frac{1}{2} \rho c_P T^2 \tag{4}$$

In Eq. (3), the term on the left-hand side is the variation of entransy density with time, the first term on the right-hand side is the entransy transfer ability related to heat transfer and the last term on the right-hand side is the entransy dissipation rate.

Entransy dissipation, loss of heat transfer ability, is a key concept in the entransy analysis method and it was used to compare different heating and cooling systems in this study. In an irreversible heat transfer process, heat and entransy are transferred, however entransy is dissipated (heat transfer ability is being lost). Hence, the entransy balance equation in an HVAC system can be written as follows.

$$\dot{En}_s - \dot{En}_\phi = \dot{En}_o \tag{5}$$

where the entransy dissipation rate is defined as:

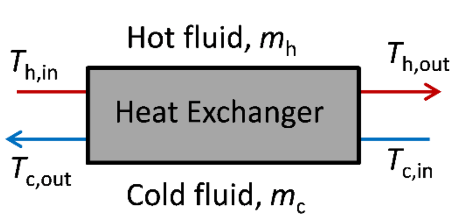
$$\dot{En}_\phi = k\,|\nabla T|^2 = -\dot{\boldsymbol{q}} \cdot \nabla T \tag{6}$$

Eq. (5) indicates that in an irreversible heat transfer process, entransy dissipation is inevitable. The first term on the left-hand side of Eq. (5) is the rate of supplied entransy, the second term on the left-hand side is entransy dissipation rate and the rate of obtained entransy is on the right-hand side of Eq. (5).

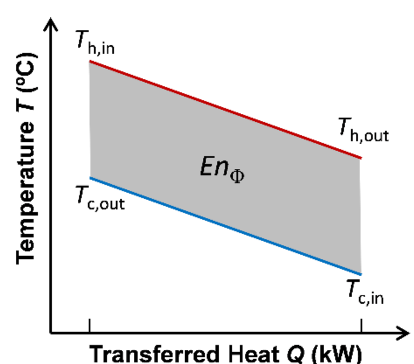
Eq. (6) shows that entransy dissipation is a function of the quantity of heat transferred and the corresponding temperature gradient, therefore a reduction of the temperature difference within a heating and cooling system will result in a reduction of the entransy dissipation.

This characteristic enables the utilization of temperature-heat (T-Q) diagrams to represent thermal (heat transfer) processes. T-Q diagrams are useful to show thermal processes and the consequent entransy dissipation since they take into account the transferred heat, respective temperatures and temperature differences. The areas between the lines in the T-Q diagrams correspond to entransy dissipation in respective system components.

Fig. 1 shows a counter-flow heat exchanger and its T-Q diagram.



a)



b)

Fig. 1. a) An example of a counter-flow heat exchanger, b) T-Q diagram of the heat exchange process [15], [16].

In Fig. 1 b), the shaded area in the T-Q diagram represents entransy dissipation during the heat transfer process in the heat exchanger. Assuming that the specific heat capacities of the fluids are constant, entransy dissipation in this process can be calculated as [15], [16]:

$$\dot{En}_{\phi} = \frac{1}{2}\left(T_{h,in} + T_{h,out} - T_{c,in} - T_{c,out}\right)Q \tag{7}$$

where $T_{h,in}$ and $T_{h,out}$ are inlet and outlet temperatures of the hot fluid [°C], and $T_{c,in}$ and $T_{c,out}$ are inlet and outlet temperatures of the cold fluid [°C], respectively.

Fig. 2 shows the T-Q diagrams of two mixing processes.

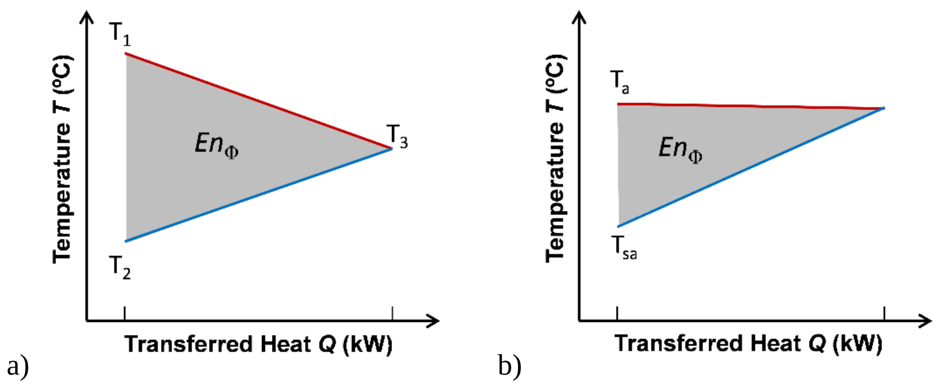


Fig. 2. T-Q diagrams of a) mixing of two fluids, b) mixing of supply air with room air [15], [16].

In Fig. 2 a), mixing of two fluids is shown, and in b), mixing of cold supply air with the warmer indoor air is shown. The shaded areas in both of the T-Q diagrams represent entransy dissipation. Entransy dissipation in a mixing process can be calculated as [15], [16]:

$$\dot{En}_{\phi} = \frac{1}{2}\, c_p\, \frac{\dot{m}_1 \dot{m}_2}{\dot{m}_1 + \dot{m}_2}\, (T_1 - T_2)^2 \tag{8}$$

If the specific heat capacities of the fluids are constant, Eq. (8) can be simplified to Eqs. (9) and (10), for Fig. 2 a) and b), respectively.

$$E\dot{n}_{\phi} = \frac{1}{2}\,(T_1 - T_2)Q \tag{9}$$

$$E\dot{n}_{\phi} = \frac{1}{2}\,(T_a - T_{sa})Q \tag{10}$$

where indices 1 and 2 denote different fluids, a denotes indoor air and sa denotes supply air.

The calculation procedure followed in this study is based on [14], [15]. Further details of the application of entransy analysis in buildings together with its applications to heat transfer mechanisms in indoor environments (conduction, convection and radiation) have been described thoroughly in IEA EBC Annex 59 documents [16], [17].

The concepts explained in this chapter were used in this study to analyze different heating and cooling strategies. As an example, the calculation of entransy dissipation in the floor heating systems was carried out in three steps; the first part was in the heat transfer in the condenser (from the refrigerant to the water), the second part was within the embedded pipe system (from the water in the pipes to the floor surface), and the third part was from the floor surface to the indoor space (to air by convection and to surfaces by radiation). These three values were summed to obtain the total entransy dissipation.

3. Description of the case studies and determination of the key parameters

The studied heating and cooling systems were assumed to be installed in a detached, one-story, single-family house, located in Copenhagen, Denmark. The house consisted of a single-space interior with a floor area of 66.2 m^2 and a conditioned volume of 213 m^3. Fig. 3 shows exterior and interior views of the house and Table 1 gives the thermal properties of the building envelope together with respective surface areas.

Fig. 3. Exterior (left) and interior (right) views of the studied house.

Table 1. Envelope thermal properties.

	North	South	East	West	Floor	Ceiling
Walls, Area, [m²]	-	-	37.2	19.3	66.2	53
Walls, U-value, [W/m²K]	-	-	0.09	0.09	0.09	0.09
Windows, Area, [m²]	36.7	21.8	-	-	-	0.74
Windows, U-value, [W/m²K]	1.04	1.04	-	-	-	1.04

The present study was of a real house but purely theoretical and therefore no actual measurements of heat flow in the house were made. The house was ventilated mechanically and the ventilation was only used to provide the necessary amount of fresh air for indoor air quality (IAQ) since the main heating and cooling terminal was the radiant floor. Further information (construction details, description and details of the heating, cooling and ventilation systems) regarding the house can be found in [18] and [19].

The following assumptions applied to all of the studied heating and cooling cases.

- The supply air was 100% outdoor air (no recirculation), and the indoor air was assumed to be fully mixed (mixing ventilation).

- It was assumed that there was no heat loss from the floor heating system, radiators, pipes and ducts to the outdoors in the heating season and there was no heat gain to the floor cooling system, pipes and ducts from the outdoors in the cooling season.
- In order to have a realistic comparison between the different systems, in all cases the required condensing and evaporation temperatures were obtained with the effectiveness-number of transfer units (ε-NTU) method with an assumed effectiveness value of 0.8 for water and 0.7 for air media, respectively.

The required parameters of the studied systems for entransy calculations were obtained as follows.

3.1 Heating season

3.1.1 Heating load

In order to determine the heating load, the methodology described in [12] was followed. The calculations were carried out under steady-state conditions. No solar heat gains were considered. The internal heat gain was constant and 4.5 W/m^2. For all cases, an outdoor temperature of -5°C [8] and an operative temperature of 20°C were assumed. When calculating ventilation losses, air temperature was assumed equal to operative temperature, for all cases.

A heat recovery unit on the exhaust air was used in warm-air heating with heat recovery case, radiator heating and in floor heating cases. This heat recovery unit (a cross-flow heat exchanger) in the air handling unit (AHU) had a heat recovery efficiency of 0.85, sensible heat [18], which implies that the supply air temperature to the indoor space after the heat recovery was 16.3°C (temperature of the air entering the air-heating coil in warm-air heating with heat recovery case). The design ventilation rate was 0.5 air change per hour (ach) in all radiator and floor heating cases, in order to provide the necessary amount of fresh air to the indoor space. An infiltration rate of 0.2 ach was assumed.

Table 2 summarizes the boundary conditions and the resulting space heating loads. Further details of the heating load calculations, required fan and pump powers, together with the schematic drawings of the analyzed heating systems are given in [12].

Table 2. Load calculation parameters and resulting space heating loads.

Indoor temperature [°C]	20
Outdoor temperature [°C]	-5
Internal heat gain [W/m^2]	4.5
Ventilation rate [ach]	0.5
Infiltration rate [ach]	0.2
Heating load – warm-air heating, total [W] and per floor area [W/m^2]	2048 / 31
Heating load – radiator heating and floor heating, total [W] and per floor area [W/m^2]*	2180 / 33

*: The difference between the two space heating loads is due to the fresh air being supplied at 16.3°C with a ventilation rate of 0.5 ach. This contributes to the space heating load, hence the higher space heating load in the second case.

3.1.2 Warm-air heating with and without heat recovery

In order to provide the necessary heating to the indoor space, a supply air temperature of 35°C was chosen, limited by the national regulations in Denmark [20]. When there was no heat recovery (WAH_NoHR), outdoor air at -5°C was heated to 35°C. When there was heat recovery on the exhaust air from the house (WAH_HR), it was possible to bring the outdoor air from -5°C to 16.3°C before it entered the air-heating coil.

The necessary heating rate needed for bringing the outdoor air at -5°C to the supply air temperature of 35°C was 5460 W, and it decreased to 2559 W with heat recovery. The necessary heat was supplied to the air by an air-heating coil that was connected to a boiler. The supply and return water temperatures to and from the air-heating coil were 50°C and 39°C, respectively [12], [21]. Table 3 summarizes the warm-air heating cases.

Table 3. Summary of the warm-air heating cases (HR: Heat recovery).

Case name	**Intake air temperature [°C]**	**Air temperature after HR [°C]**	**Supply air temperature [°C]**	**Air flow rate [m^3/h]**	**Mass flow rate in air-heating coil [kg/h]**
WAH_NoHR	-5	-	35	410	428
WAH_HR	-5	16.3	35	410	201

3.1.3 Radiator heating

Three sets of working temperatures were assumed for radiator heating cases: 70/55, 55/45 and 45/35 (supply/return water temperature in °C) [22]. The assumed radiator type was a double panel steel radiator with extended surface (fins) [21]. The required water flow rate in the radiators was determined according to the space heating load and the temperature difference between the supply and return water flows to and from the radiator.

The heat source for the radiators was assumed to be an air-to-water heat pump. The water in the radiators was assumed to be directly circulating through the heat pump. The required condensing temperatures were 57.5°C and 47.5°C for the 55/45 case and for the 45/35 case, respectively. This assumption did not directly apply to the 70/55 case, but the same temperature difference between the heating surface and the supply water temperature was kept, resulting in a temperature of 72.5°C. Table 4 summarizes the radiator heating cases.

Table 4. Summary of the radiator heating cases.

Case name	Supply temperature [°C]	Return temperature [°C]	Condensing temperature [°C]	Mass flow rate [kg/h]
R_45	45	35	47.5	188
R_55	55	45	57.5	188
R_70	70	55	72.5*	125

*: Heating surface temperature for R_70.

3.1.4 Floor heating

The floor heating system in the house covered an area of 45 m^2, which is 68% of the total floor area, and it consisted of chipboard elements, with aluminum heat conducting profiles (thickness 0.3 mm and length 0.17 m), PE-X pipe, 17x2.0 mm. Pipe spacing was 0.2 m. A detailed drawing of the floor structure is given in [12].

The thickness and material of the floor covering (hence the thermal conductive resistance) are important parameters affecting the thermal output from a floor heating system [23], [24], [25]. The choice of floor covering could be based on structural reasons or simply on personal preferences. Different floor covering resistances were studied to investigate its effects on system performance: 0.05 m^2K/W (similar to a marble floor and mud-set [21]), 0.09 m^2K/W (actual value, similar to a light carpet [21]), and 0.15 m^2K/W (similar to a heavy carpet [21]), following the most common values given in standards [26]. The same floor covering material was used, wooden floor covering with a thermal conductivity of 0.13 W/mK, with 0.0065 m, 0.012 m and 0.0195 m thickness, respectively. For each resistance value, new water supply and return temperatures were calculated. It was assumed that the floor heating system was coupled to an air-to-water heat pump and therefore a new condensing temperature was calculated for each resistance value. It was assumed that the water in the embedded pipes circulated directly through the condenser of the heat pump.

In order to provide the necessary heating to the indoor space, a specific heat output of 48.4 W/m^2-floor heating area with an average floor surface temperature of 24.7°C was required. Heat output and surface temperatures were calculated according to [26] and [27]. The floor surface temperature was the same for all cases. The temperature drop between the supply and return water in the radiant system was 4°C. The mass flow rate was calculated based on EN 1264-3:2009 [28] and was found to be 469 kg/h. Table 5 summarizes the floor heating cases.

Table 5. Summary of the floor heating cases.

Case name	**Floor covering resistance [m^2K/W]**	**Supply temperature [°C]**	**Return temperature [°C]**	**Condensing temperature [°C]**
FH_LoRes	0.05	33	29	34
FH_MRes	0.09	35.8	31.8	36.8
FH_HiRes	0.15	39.8	35.8	40.8

3.2 Cooling season

3.2.1 Cooling load

The space cooling load was calculated with the steady-state assumption. The outdoor air temperature was assumed to be 30°C and the operative temperature was 26°C. The internal heat gain was constant and 4.5 W/m^2. The design ventilation rate was 0.5 ach for floor cooling cases and the intake air was not cooled. An infiltration rate of 0.2 ach was assumed for all cases.

The house was supported on 30 cm high concrete blocks and this created a crawl-space between the ground and the house’s floor structure. In some of the cooling cases, the intake air was from the crawl-space instead of the outdoor air. In those cases, the fresh air temperature coming into the AHU or to the indoor space was 21.3°C [13], due to the pre-cooling of the outdoor air by the ground surface under the crawl-space.

Table 6 summarizes the boundary conditions in cooling season. Further details of the cooling load calculations (solar heat gains, etc.), required fan and pump powers, and the schematic drawings of the analyzed cooling systems are given in [13].

Table 6. Load calculation parameters in cooling season.

Indoor temperature [°C]	26
Outdoor temperature [°C]	30
Air temperature in the crawl-space [°C]	21.3
Internal heat gain [W/m^2]	4.5
Ventilation rate [ach]	0.5
Infiltration rate [ach]	0.2

In total five cases were considered for air cooling and three cases were considered for floor cooling. In the first air cooling case, an internal solar shading was used and this resulted in a space cooling load of 3170 W (48 W/m^2 floor area). For all other air cooling cases, an external solar shading was used and the resulting space cooling load was 1042 W (16 W/m^2 floor area).

For floor cooling cases, external shading was used. In the first floor cooling case, the fresh air was taken directly from outdoors and this resulted in a space cooling load of 1183 W (18 W/m^2 floor area). In other floor cooling cases the fresh air was taken from the crawl-space and the resulting space cooling load was 876 W (13 W/m^2 floor area). Table 7 summarizes the cooling cases.

Table 7. Summary of the cooling cases.

Case	Shading	Cooling	Source	Intake air	Space cooling load [W]
AC_INT	Internal	AC	AWHP	OA	3170
AC	External	AC	AWHP	OA	1042
AC_17	External	AC	AWHP	OA	1042
AC_20	External	AC	AWHP	OA	1042
AC_CS	External	AC	AWHP	CS	1042
FC	External	FC	AWHP	OA	1183
FC_CS	External	FC	AWHP	CS	876
FC_CS_GHEX	External	FC	GHEX	CS	876

*: Supply air temperatures and air flow rates are different for AC, AC_17 and AC_20. The abbreviations of these cases indicate the supply air temperatures to the indoor space (Table 8). AC: Air cooling, FC: Floor cooling, AWHP: Air-to-water heat pump, GHEX: Ground heat exchanger, OA: Outdoor air, CS: Crawl-space.

3.2.2 Air cooling

In air cooling cases, the intake air (either outdoor air or air from the crawl-space) was passing through an air-cooling coil where it was cooled down to the supply temperature. The water entered the air-cooling coil at 7°C and returned at 12°C. It was assumed that this air-cooling coil was coupled to an air-to-water heat pump [13].

The ventilation rates for air cooling cases were calculated based on the space cooling loads and the temperature difference between the supply and room air temperatures. The water flow rate in the air-cooling coil was calculated based on the heat to be removed from the intake air and the temperature difference between the supply and return water flows to and from the air-cooling coil. The heat to be removed from the intake air corresponds to the required amount of heat to lower the temperature of the intake air to the required supply air temperature, which was 14°C, 17°C or 20°C for respective cases. Table 8 summarizes the air cooling cases.

Table 8. Summary of the air cooling cases (IA: Intake air).

Case	Supply air temperature [°C]	Ventilation rate [ach]	Rate of cooling to IA [W]	Water flow rate in the air-cooling coil [kg/h]
AC_INT	14	3.7	4226	725
AC	14	1.2	1389	238
AC_17	17	1.6	1505	258
AC_20	20	2.5	1736	298
AC_CS	14	1.2	634	109

3.2.3 Floor cooling

The same floor structure as described in 3.1.4 was used for cooling. A floor covering resistance of 0.05 m^2K/W was used for floor cooling cases to minimize the effects of floor covering resistance on system performance.

For FC_CS and FC_CS_GHEX, the heat to be removed by the floor was 876 W, and for FC it was 1183 W. These values correspond to cooling loads of 19.5 and 26.3 W/m^2-cooled floor area, respectively, and corresponding average floor surface temperatures of 23.2 and 22.2°C. In order to obtain these surface temperatures, the required supply and return water temperatures were 18.6 and 21.6°C for FC_CS and FC_CS_GHEX, and 16.5 and 19.5°C for FC. The assumed temperature difference between supply and return water flows was 3°C for all cases. For FC_CS and FC_CS_GHEX, this resulted in a mass flow rate of 250 kg/h, and for FC it was 338 kg/h. The cooling output, floor surface temperatures and the mass flow rates were calculated according to [26], [27], [28], [29]. Table 9 summarizes the floor cooling cases.

Table 9. Summary of the floor cooling cases.

Case	Supply and return water temperature [°C]	Cooled floor surface temperature [°C]	Evaporation temperature [°C]	Water flow rate [kg/h]
FC	16.5 / 19.5	22.2	15.7	338
FC_CS & FC_CS_GHEX	18.6 / 21.6	23.2	17.8	250

In FC and FC_CS, it was assumed that the floor cooling system was coupled to an air-to-water heat pump and that the water in the floor loops circulated directly through the evaporator of the heat pump. The required evaporation temperatures for FC and for FC_CS are given in Table 9.

In FC_CS_GHEX, it was assumed that the floor cooling system was coupled to a single U-tube vertical heat exchanger in the ground. There was a flat-plate heat exchanger between the floor loops and the ground heat exchanger. A pump was circulating the brine consisting of 30% propylene-glycol/water mixture [13].

The incoming and outgoing brine temperatures to and from the borehole were 17 and 13°C, respectively, with a corresponding brine mass flow rate of 208 kg/h in the borehole. It is possible to achieve the necessary cooling of the brine from 17°C to 13°C at 40 m depth for this particular borehole, with the ground temperature of 8.3°C in Copenhagen, Denmark [19]. Further details of this ground heat exchanger are given in [19] and [30].

4. Results and discussion

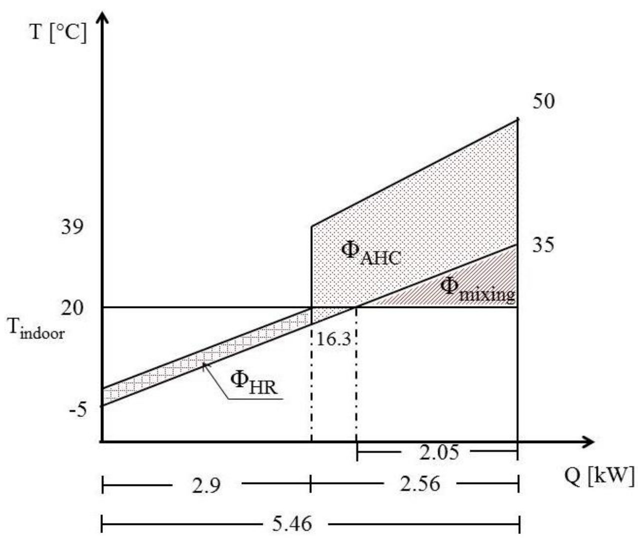
The results are presented using T-Q diagrams and entransy dissipation for the studied heating and cooling systems. The shaded areas between the lines in the following T-Q diagrams, indicated by Φ [kWK], represent entransy dissipation in respective system components.

In the following results, in warm-air heating and air cooling cases, entransy dissipation in the indoor space is referred to as mixing, while in radiator heating, floor heating and in floor cooling cases it is referred to as indoor. This is because the entransy dissipation in the indoor space is due to a combination of radiative and convective heat transfer in radiator heating, floor heating and floor cooling cases, while in warm-air heating and air cooling cases the heat transfer in the indoor space is convective.

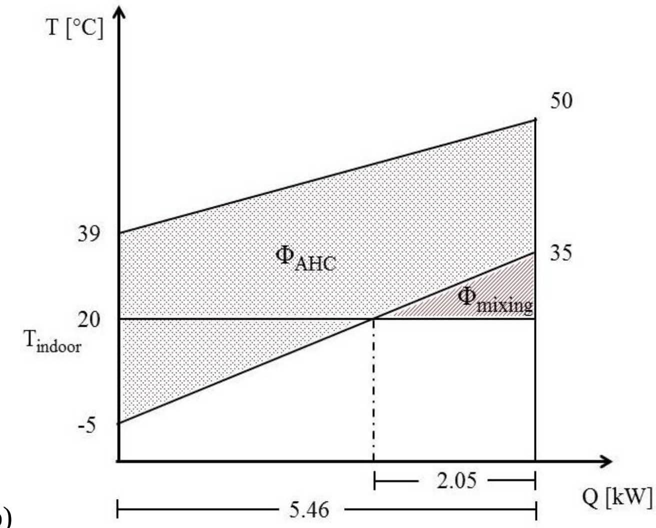
4.1. Heating systems

4.1.1 Warm-air heating

Fig. 4 shows the T-Q diagrams of warm-air heating with and without heat recovery on the exhaust. Table 10 shows the entransy dissipations in system components when warm-air heating was used to heat the indoor space.



a)



b)

Fig. 4. T-Q diagrams of a) warm-air heating with heat recovery on exhaust air (WAH_HR), b) warm-air heating without heat recovery on exhaust air (WAH_NoHR) (AHC: Air-heating coil, HR: Heat recovery).

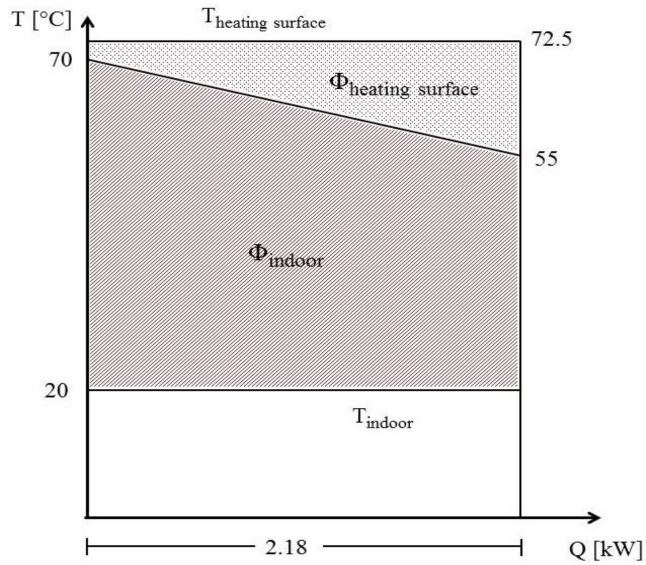
Table 10. Entransy dissipations in system components for warm-air heating cases.

Case	Φ_{AHC} [kWK]	Φ_{mixing} [kWK]	Φ_{HR} [kWK]	Φ_{total} [kWK]
WAH_HR	48.3	15.4	10.7	74.4
WAH_NoHR	161.1	15.4	-	176.5

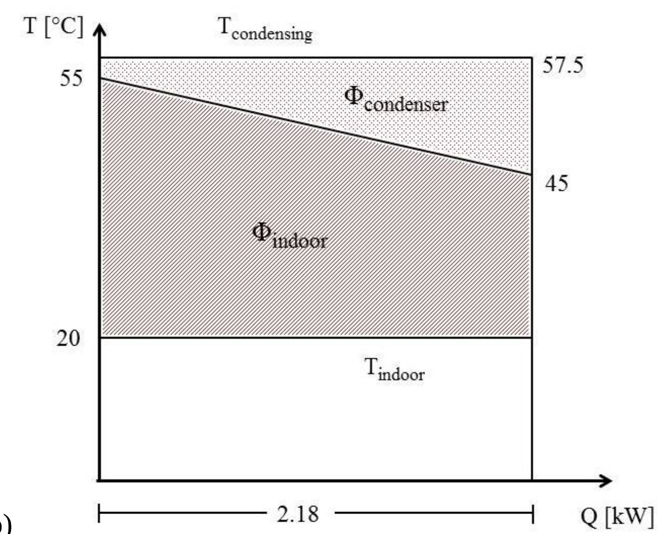
In warm-air heating cases, entransy dissipation due to mixing in the indoor air is the same for both cases due to the supply air temperature and the indoor air temperature being the same. The results show that heat recovery on the exhaust air is beneficial in terms of reducing entransy dissipation in the air-heating coil; a 70% reduction is obtained when heat recovery is applied compared to the case with no heat recovery. Although there is a certain amount of entransy dissipation in the heat recovery unit (10.7 kWK), total entransy dissipation is reduced by 58% with the application of heat recovery compared to the case without heat recovery.

4.1.2 Radiator heating

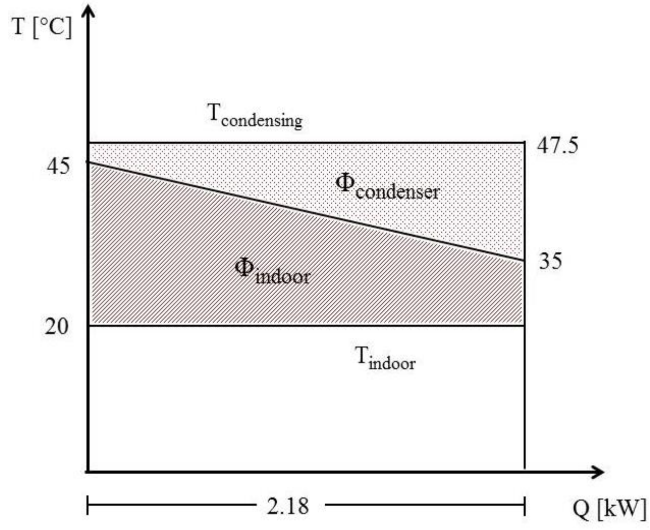
Fig. 5 shows the T-Q diagrams of radiator heating cases with different working temperatures and Table 11 shows the entransy dissipations in the system.



a)



b)



c)

Fig. 5. T-Q diagrams of radiator heating cases with working temperatures of a) 70/55 (R_70), b) 55/45 (R_55), c) 45/35 (R_45).

Table 11. Entransy dissipations in the system components for radiator heating cases.

Case	$\Phi_{condenser}$ [kWK]	Φ_{indoor} [kWK]	Φ_{total} [kWK]
R_70	21.8*	92.7	114.5
R_55	16.4	65.4	81.8
R_45	16.4	43.6	60.0

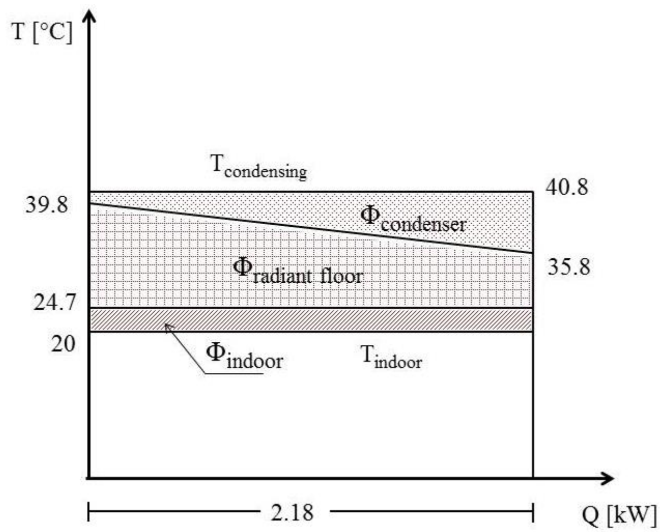
*: $\Phi_{heating\ surface}$ for R_70.

In the radiator systems analyzed, a large part of the total entransy dissipation is due to the entransy dissipation indoors; 81% for R_70, 80% for R_55 and 73% for R_45. This behavior is due to the radiator working temperatures being noticeably higher than the indoor temperature, as opposed to low temperature heating systems. The assumption of constant temperature on the heat source (condenser and heating surface) is also affecting these values.

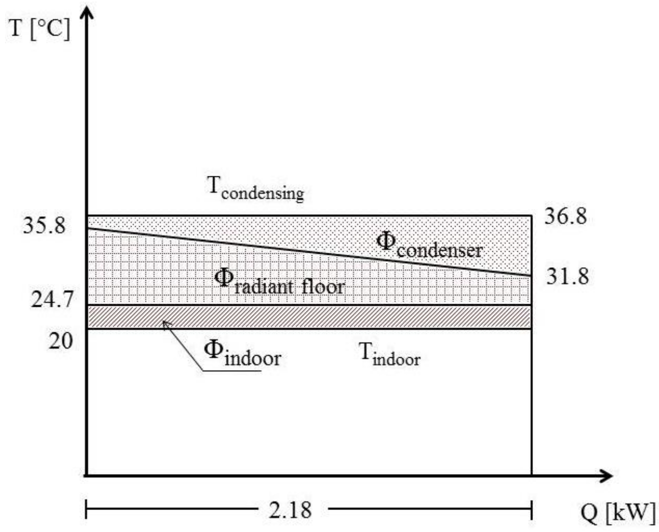
R_45 had 33% lower entransy dissipation in the indoor space compared to R_55, and 53% lower compared to R_70. The results also show that the total entransy dissipation in R_45 case was 27% lower compared to R_55 and 48% lower compared to R_70. These results show that reducing the working temperatures for heating is beneficial for reducing the entransy dissipation in the space heating process. This also allows reducing the total temperature difference in the system; from indoor temperature to the heat source temperature. This approach could particularly be useful in renovation of existing buildings.

4.1.3 Floor heating

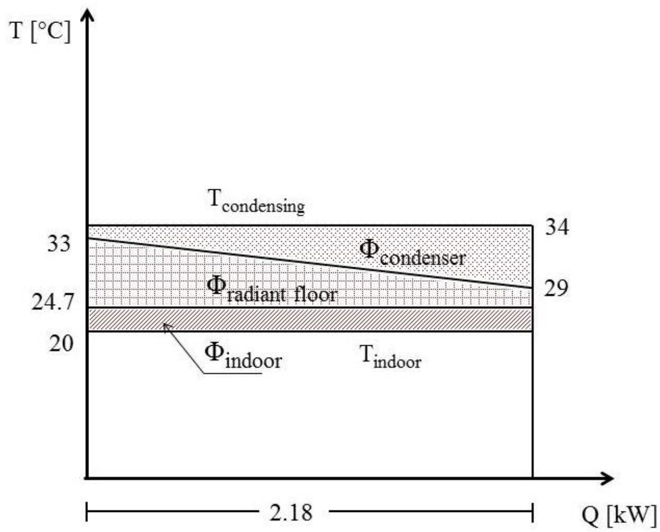
Fig. 6 shows the T-Q diagrams of floor heating cases with different floor covering resistances and Table 12 shows the entransy dissipations in system components.



a)



b)



c)

Fig. 6. T-Q diagrams of floor heating cases with different floor covering resistances: a) FH_HiRes, b) FH_MRes, c) FH_LoRes.

Table 12. Entransy dissipations in system components for floor heating cases.

Case	$\Phi_{condenser}$ [kWK]	$\Phi_{radiant\ floor}$ [kWK]	Φ_{indoor} [kWK]	Φ_{total} [kWK]
FH_HiRes	6.5	28.6	10.3	45.4
FH_MRes	6.5	19.8	10.3	36.6
FH_LoRes	6.5	13.7	10.3	30.5

The entransy dissipation in the indoor heat transfer process was the same for all floor heating cases. This is because the required floor surface temperature (24.7°C) and the indoor temperature (20°C) were the same for all cases. Furthermore, the entransy dissipation in the condenser was the same for all cases and this is because the assumed effectiveness value in the condenser was the same for all cases, which results in a 1 K higher condensing temperature than the water supply temperature to the floor heating system (energy transfer to the indoor space is the same for all floor coverings, therefore the condenser delivers the same amount of energy hence a constant temperature difference of 1 K). In a real case, there might be some additional loses in the distribution system with higher water temperatures, so it is probable that the temperature difference will not be a fixed value.

The differences in the entransy dissipation between floor heating cases are due to the heat transfer process from the water in the pipes to the floor surface. An increased floor covering resistance resulted in an increase of the required water supply temperature, in order to compensate for the increased conductive resistance of the floor structure and to achieve the same floor surface temperature.

Compared to the highest floor covering resistance case (FH_HiRes), entransy dissipation in the floor is reduced by 31% in FH_MRes and by 52% in FH_LoRes. In total, entransy dissipation is reduced by 19% in FH_MRes and by 33% in FH_LoRes compared to FH_HiRes. These system behaviors indicate that heat transfer potential was being wasted in the floor structure, and therefore, higher the floor covering resistance higher the entransy dissipation. The results show that it is important to minimize the unnecessary resistances, such as the floor covering resistance, in the heating system that could hinder the overall system performance.

4.1.4 Entransy dissipation based comparison of the heating systems

Among the analyzed space heating methods, floor heating cases had the lowest total entransy dissipation, indicating a clear benefit for radiant low temperature heating systems. The results also show that in order to fully benefit from the low temperature heating potential of the floor heating systems, the resistance between the heat transfer medium and the indoor space should be kept to a minimum. This was studied by means of floor covering resistance. Floor heating cases also had the lowest entransy dissipation in the indoor space among the investigated cases.

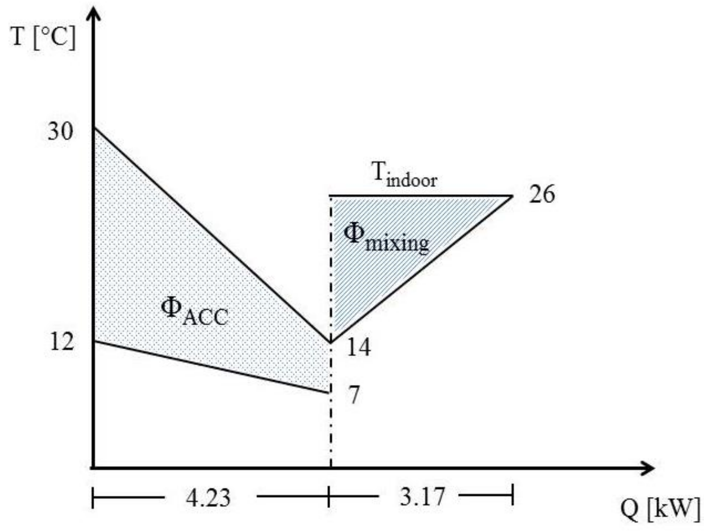
When comparing the total entransy dissipations of different heating systems, FH_LoRes had 49% lower entransy dissipation compared to R_45, and 59% lower compared to WAH_HR. Excluding the floor heating cases, R_45 had the lowest entransy dissipation followed by WAH_HR, R_55, R_70 and WAH_NoHR had the highest entransy dissipation.

The results show a clear benefit for the radiant floor heating system, which is an example of a low temperature heating system where the temperature of the heat transfer medium is close to the room temperature.

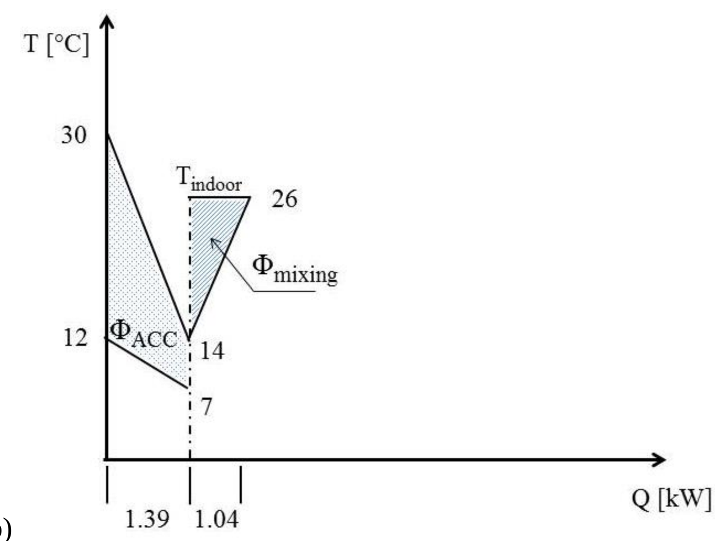
4.2. Cooling systems

4.2.1 Air cooling

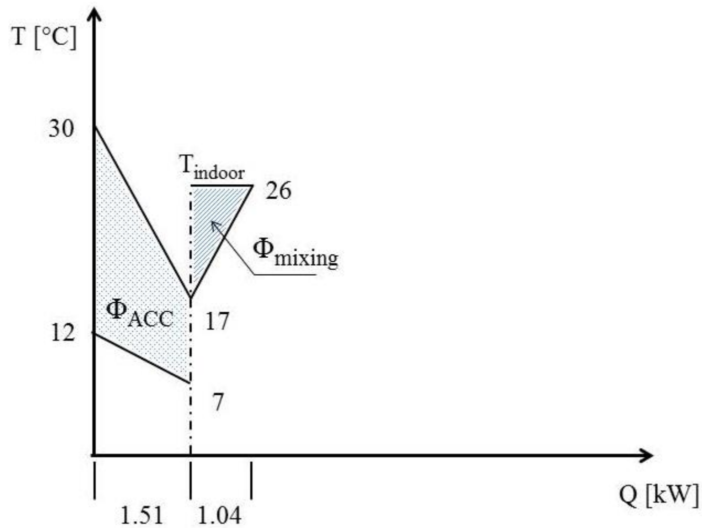
Fig. 7 shows the T-Q diagrams of air cooling systems. Table 13 shows the entransy dissipations in system components for air cooling cases.


T [°C]
30
12
T_{indoor}
26
Φ_{mixing}
Φ_{ACC}
14
7
4.23
3.17
Q [kW]

a)

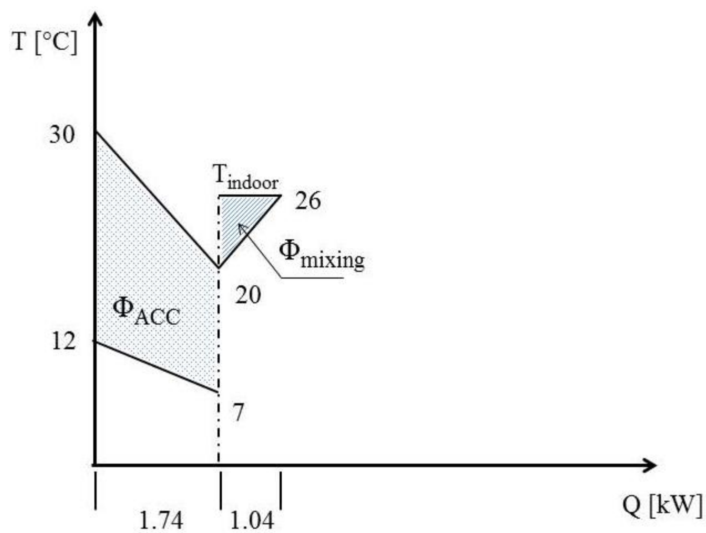

T [°C]
30
T_{indoor}
26
Φ_{mixing}
12
Φ_{ACC}
14
7
Q [kW]
1.39
1.04

b)



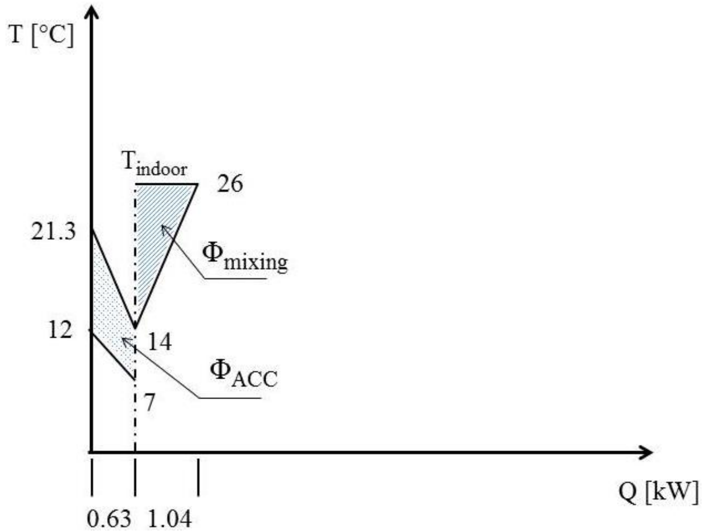
T [°C]
30
T_{indoor}
26
Φ_{mixing}
Φ_{ACC}
17
12
7
Q [kW]
1.51
1.04

c)



T [°C]
30
T_{indoor}
26
Φ_{mixing}
20
Φ_{ACC}
12
7
Q [kW]
1.74
1.04

d)



e)

Fig. 7. T-Q diagrams of air cooling cases: a) AC_INT, b) AC, c) AC_17, d) AC_20, e) AC_CS (ACC: Air-cooling coil).

Table 13. Entransy dissipations in the system components for air cooling cases.

Case	**Φ_{ACC} [kWK]**	**Φ_{mixing} [kWK]**	**Φ_{total} [kWK]**
AC_INT	52.9	19	71.9
AC	17.4	6.2	23.6
AC_17	21.1	4.7	25.8
AC_20	27.0	3.1	30.1
AC_CS	5.1	6.2	11.3

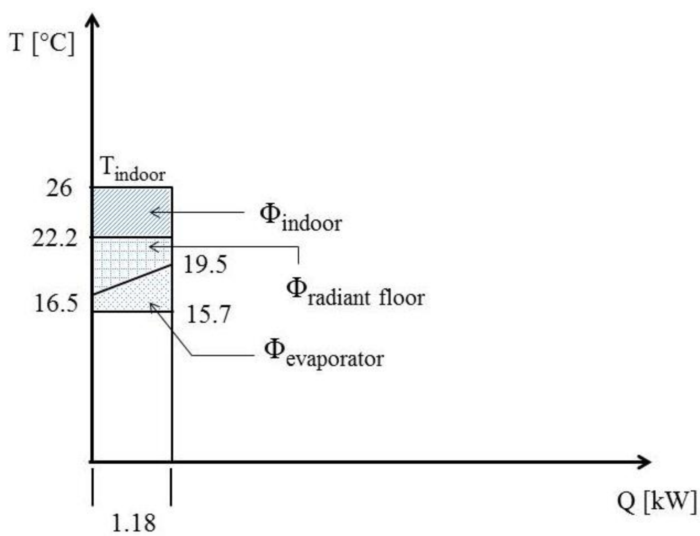
Among the air cooling cases, AC_INT had the highest total entransy dissipation. This was because of the high space cooling load compared to other cases due to the use of internal solar shading compared to external solar shading. The results show that high cooling load results in high entransy dissipation.

In the rest of the cooling cases, external solar shading was used. Among AC, AC_17 and AC_20, AC had the lowest total entransy dissipation followed by AC_17 and AC_20. Among these cases, AC had the lowest entransy dissipation in the air-cooling coil followed by AC_17 and AC_20, while this trend was reversed when considering the mixing entransy dissipation indoors. For further comparisons AC will be used.

Among the air cooling cases, AC_CS had the lowest entransy dissipation due to having the intake air from the crawl-space instead of the outdoor air. This results in a reduction in the entransy dissipation in the air-cooling coil; a 71% reduction compared to AC. This difference in the air-cooling coil resulted in an overall entransy dissipation reduction of 52% for AC_CS compared to AC.

4.2.2 Floor cooling

Fig. 8 shows the T-Q diagrams of floor cooling systems coupled to an air-to-water heat pump (FC and FC_CS) and to a ground heat exchanger (FC_CS_GHEX). Table 14 shows the entransy dissipations in system components.



a)

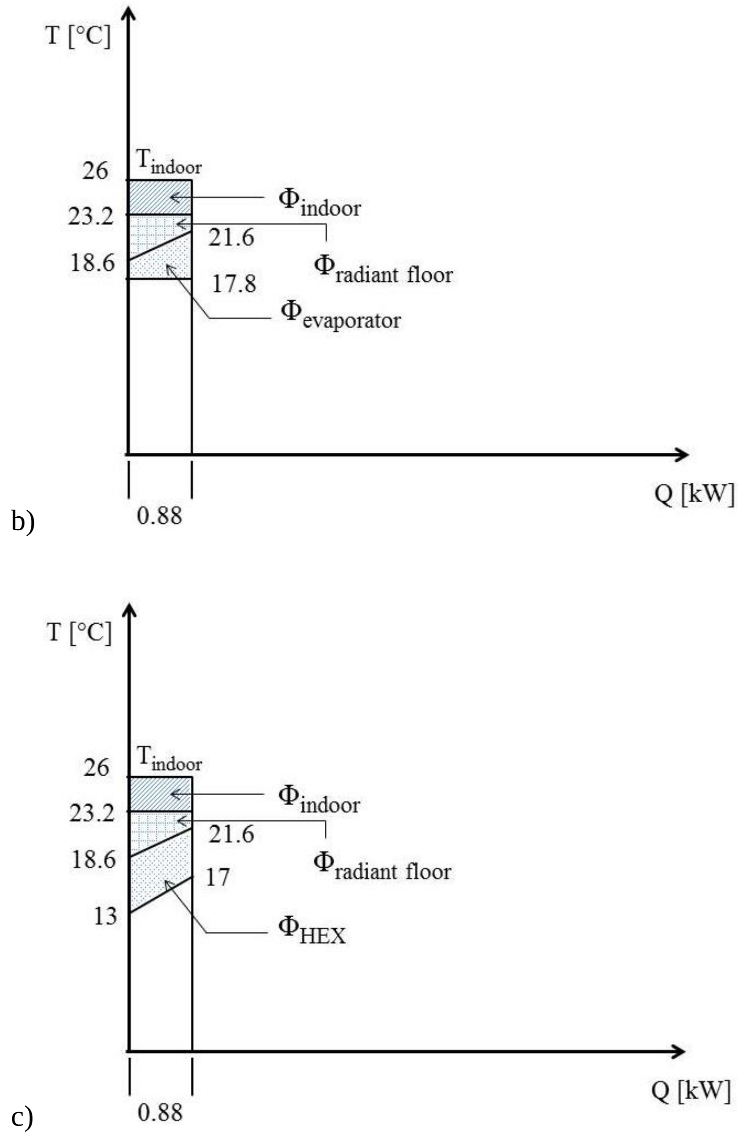


Fig. 8. T-Q diagrams of floor cooling cases: a) FC, b) FC_CS, c) FC_CS_GHEX (HEX: Heat exchanger).

Table 14. Entransy dissipations in the system components for floor cooling cases.

Case	$\Phi_{evaporator}$ [kWK]	$\Phi_{radiant\ floor}$ [kWK]	Φ_{indoor} [kWK]	Φ_{total} [kWK]
FC	2.7	5.0	4.5	12.2
FC_CS	2.0	2.7	2.5	7.2
FC_CS_GHEX	4.5*	2.7	2.5	9.7

*: Φ_{HEX} for FC_CS_GHEX, which is the flat-plate heat exchanger between the floor loops and the ground heat exchanger.

The results from FC and FC_CS show that reducing the space cooling demand through supply of fresh air from the crawl-space instead of the outdoor air has remarkable effects on the system entransy dissipation. FC_CS has 41% lower entransy dissipation compared to FC, due to the entransy dissipation reduction in system components (evaporator and radiant floor) and indoors. This is possible due to the decreased space cooling load enabling a higher cooled floor surface temperature and a higher evaporation temperature in the heat pump.

When the floor cooling was coupled to ground heat exchanger (FC_CS_GHEX), the entransy dissipations in the radiant floor and indoors were the same however the entransy dissipation in the heat exchanger between the ground loop and the floor cooling loops was higher than the entransy dissipation in the evaporator in FC_CS. This is mainly due to the temperature differences between the supply and return brine temperatures in the ground loop and the supply and return water temperatures in the floor loop. Although the overall entransy dissipation in FC_CS_GHEX is higher than FC_CS, it would still be a better option to couple the floor cooling to a ground heat exchanger compared to coupling it to an air-to-water heat pump [13] as long as the local conditions and regulations allow it.

4.2.3 Entransy dissipation based comparison of the cooling systems

The results of air cooling cases show the importance of reducing the cooling demand, as a priority. Different supply air temperatures result in different system behaviors with effects on the air-cooling coil and on the indoor conditions, and hence, on the total system performance. The air cooling case where the intake air was from the crawl-space, instead of the outdoor air, benefited considerably from the lower intake air temperature.

The comparison between AC and FC, and between AC_CS and FC_CS allow comparing the air-based cooling system performance with water-based cooling system performance in terms of entransy dissipation. The results show that total entransy dissipation in FC is 48% lower than AC. FC_CS has 36% lower total entransy dissipation than AC_CS. Entransy dissipations indoors and in the evaporator are also remarkably lower for floor cooling cases compared to air cooling cases (Table 14).

The results show the benefits of using a high temperature cooling system, floor cooling, in reducing the entransy dissipation during the space cooling process. This is possible due to working at temperatures close to room temperatures. The use of a high temperature cooling system also enables to integrate a renewable energy resource, ground, to the cooling system, which would considerably reduce the overall energy use by the system [13], [19].

4.3 Overall discussion

The results of both the heating and cooling cases indicate a clear benefit for the low temperature heating and high temperature cooling systems. Radiant floor heating and cooling systems are an example of low temperature heating and high temperature cooling systems. Floor heating systems performed better in terms of entransy dissipation than warm-air heating, and radiator heating. Floor cooling performed better in terms of entransy dissipation than air cooling.

The main advantage of water-based low temperature heating and high temperature cooling systems stems from the fact that the temperature of the heat transfer medium is close to the room temperatures, both in heating and cooling modes. This also enables decreasing the overall temperature difference in the system from the indoor temperature to the heat source or sink, indicating an improvement in the system performance [14].

In the investigated cases, a lower condensing temperature (e.g. lower floor covering resistance vs. higher floor covering resistance) and a higher evaporation temperature (e.g. intake air from crawl-space vs. outdoor air, in floor cooling cases) would indicate an improvement in the heat pump performance. The requirement of low water temperatures for heating and high water temperatures for cooling means that it would be possible to use naturally available heat sources and heat sinks instead of refrigeration cycles, which use a compressor (an air-to-water heat pump in this study).

The present study was of a real house but purely theoretical and therefore certain assumptions had to be made. The obtained results depend on the assumptions made and values might differ in a real application, however the general trends would be similar, due to the operation and heat transfer principles of the chosen air-based and water-based heating and cooling systems.

The present study used the entransy concept and entransy dissipation as the parameter to compare the chosen systems. In an actual installation, several other factors will influence the choice of heating and cooling system. These factors are costs (capital and operational), occupant thermal comfort (overall and local), auxiliary energy use (pumps, fans, control systems, etc.), availability of heat nearby natural heat sources and sinks and the possibility of coupling with the terminal units, control possibilities and dynamic behaviors of the terminal units [1]. Further discussion regarding the applicability of different heat sources and sinks, choice of terminal units, and regarding auxiliary components (pumps and fans) can be found in [12], [13].

In this study and in the actual design of the house [18], [30], the indoor space was heated and cooled by the radiant floor and the ventilation system was only used to provide the required amount of fresh air. The effectiveness of this approach was proven with previous studies in terms of exergy [12], [13], and the present study proves the effectiveness of this approach by the entransy method.

The ultimate goal in heating and cooling of buildings is to decrease the temperature difference between different components from indoor terminal units to the heating and cooling plants (i.e. to decrease the overall temperature difference in the system, enabling the utilization of non-fossil resources such as ground or solar, and resulting in improved performance of heat pumps and chillers). A certain method or any combination of methods that serves this purpose will be useful for improving and optimizing building heating and cooling systems, during the design and operation phases.

5. Conclusion

The present study used entransy analysis and compared different space heating and cooling systems' performance based on entransy dissipation. A single-family house was used as a case study. Warm-air heating, radiator heating and floor heating were compared in the heating season, while in the cooling season, air cooling and floor cooling were compared. Main conclusions from the analyses are the following.

- Among the investigated space heating systems, floor heating had the lowest total entransy dissipation: floor heating system with a low floor covering resistance (FH_LoRes) had 49% lower entransy dissipation compared to the radiator heating with low supply and return temperatures (R_45), and 59% lower compared to warm-air heating with heat recovery on the exhaust (WAH_HR).
- In cooling operation, when the intake air was from outdoors, floor cooling (FC) system had 48% lower total entransy dissipation compared to air cooling (AC). When the intake air was from the crawl-space, floor cooling (FC_CS) had 36% lower total entransy dissipation compared to air cooling (AC_CS).
- Floor covering resistance can be an important factor affecting the performance of radiant systems, including floor heating and cooling systems. Compared to the floor heating with high floor covering resistance (FH_HiRes), entransy dissipation in the floor structure was reduced by 31% in floor heating with medium floor covering resistance (FH_MRes) and by 52% in floor heating with low floor covering resistance (FH_LoRes). Total entransy dissipation was reduced by 19% in FH_MRes and by 33% in FH_LoRes compared to FH_HiRes. These system behaviors indicate that heat transfer potential was being wasted in the floor structure, and therefore the resistance between the heat transfer medium and

the surface should be minimized to fully benefit from the low temperature heating and high temperature cooling potential of the floor heating and cooling system.

- Low temperature heating and high temperature cooling systems in buildings achieve higher performance in comparison to other space heating and cooling systems. This was shown in terms of entransy dissipation in the present study, using an example of these systems (radiant floor heating and cooling) and comparing its performance to other space heating and cooling systems.
- Decreased temperature differences within the whole system can be achieved by using higher cooling sink temperatures and lower heating source temperatures, and low temperature heating and high temperature cooling systems enable this. Expressed in terms of entransy; lower entransy dissipation in a system shows a lowered temperature for heating and higher temperature for cooling, which would enable energy savings, improved resource and energy efficiency by allowing favorable operating conditions for heating and cooling plants (e.g. heat pumps, chillers, etc.), and by allowing use of natural heat sources and sinks.

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Simulation Study of the Energy Performance of Different Space Heating Methods in Plus-energy Housing

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Abstract

Due to a shortage of energy resources, the focus on indoor environment and energy use in buildings is increasing which sets higher standards for the performance of HVAC systems in buildings. The variety of available heating systems for both residential buildings and office buildings is therefore increasing together with the performance of the systems.

This paper reports the results of a simulation study carried out using the commercially available building simulation software IDA ICE. The considered house was designed as a plus-energy house and it was located in Denmark. The dynamic building simulation model has been validated and calibrated with measurement data from the house in a previous study. The studied systems were radiant floor heating, warm-air heating through ventilation system and radiator heating. The energy performance of systems for achieving the same thermal comfort was compared.

The effects of several parameters on system energy performance for each space heating solution were investigated; floor covering resistance of the floor heating system, having a heat recovery on the exhaust in the ventilation system, and different working temperature levels for the radiator heating. For all cases the heat source was a natural gas fired condensing boiler, and for the floor heating cases also an air-to-water heat pump was used to compare two heat sources. The systems were also compared in terms of auxiliary energy use for pumps and fans.

The results show that the investigated floor heating systems had the best performance in terms of energy with a total energy saving of 23% compared to warm-air heating with heat recovery. It can furthermore be coupled to other heat sources than a boiler. The floor covering resistance of the floor heating system should be kept to a minimum to fully benefit from the low temperature heating potential since an increased floor covering requires higher average water temperatures in the floor loops and decreases the COP of the heat pump. The water-based heating systems required significantly less auxiliary energy input compared to the air-based heating system.

Furthermore, the results show that low temperature heating systems, as seen in floor heating in this study, can contribute to achieving plus-energy targets by minimizing the energy use for space heating purposes while achieving necessary thermal comfort for the occupants.

Keywords - Floor heating, warm-air heating, radiator heating, plus-energy house

1. Introduction

The amount of energy used for space heating accounts for a large part of the total energy demand of a building placed in colder climates. As the focus on energy use in buildings is increasing, different types of heating systems are also increasing together with the performance of the systems.

This study focuses on determining the energy use for different heating systems under the same thermal comfort conditions. A detailed definition of the investigated heating systems can be found in [1] in which the systems are investigated with the use of the exergy concept.

The simulation model used for the investigation was based on the competition house 'Fold' designed and constructed by the Technical University of Denmark for the Solar Decathlon Europe 2012 competition. The house has a floor area of 66 m^2 and a conditioned volume of 213 m^3, and has two large glazing facades facing North (36.7 m^2) and South (21.8 m^2) respectively with a turn of 19° to the West.

Figure 1 shows the exterior views of the actual house.

Figure 1 – South façade (left) and North façade (right) of Fold

A detailed description of the house can be found in [2], [3].

2. Method

The investigation was carried out with the simulation software IDA ICE as the tool for assessing the thermal indoor environment and energy use. The initial IDA ICE model was constructed as a model with the properties as the actual house and has been validated with experimental measurements from the house [4]–[6].

A more detailed explanation of the general methods, internal gains, occupancy schedules etc. applied in this study can be found in [7].

For the investigations, the HVAC plant consists of a simple system mainly containing two 100 L water tanks for hot and cold water respectively. The cold water tank was only implemented due to limitations in the simulation software. The method of supplying the heat could quickly be altered without having to change any of the remaining properties of the HVAC system. A generic fuel heater was used in most of the investigated cases. The generic fuel heater heats the water in the hot water tank to the

desired temperature. For the floor heating cases, an air-to-water heat pump delivered the heating to the floor heating systems.
The investigation focused on three methods of supplying heat to the indoors: warm-air heating, heating with radiators and radiant floor heating. The main properties for the different cases are listed in Table 1, Table 2 and Table 3.

Table 1 – Case description for warm – air heating (WAH) [1]

Case	Heat recovery	Supply air temperature
WAH_NoHR	No	35°C
WAH_HR	Yes	35°C

Table 2 – Case description for radiator heating (R) [1]

Case	Supply temperature	Return temperature
R_45	45°C	35°C
R_55	55°C	45°C
R_70	70°C	55°C
R_90	90°C	70°C

Table 3 – Case description for radiant floor heating (FH) [1]

Case	Thermal resistance of floor covering	Supply temperature
FH_LoRes	0.05 m^2K/W	33°C
FH_MRes	0.09 m^2K/W	35.8°C
FH_HiRes	0.15 m^2K/W	39.8°C

For all cases the set-point for heating was set as 20°C and controlled by the operative temperature. The software normally controls the ventilation system by the air temperature. In order to get similar control systems, the ventilation system had to be controlled by the operative temperature. The solution was to implement a radiator with a negligible capacity of 1 W. The operative temperature reading from the radiator were connected to the ventilation system as a replacement for the air temperature input.

Radiators and radiant floor heating cases were set to be controlled by the operative temperature in the room with the use of a proportional controller.

The ventilation rate for the water-based systems was fixed at 0.5 h^{-1} with a supply temperature of 16.3°C with the use of heat recovery.

The properties of the radiators were calculated directly in the software from inputs of maximum power, supply temperature and return temperature. All radiators and floor heating systems were assumed to have a maximum capacity of 2500 W which was found as the required heating demand from the software. The radiant floor system was build up in the same manner as the one in the validated case with a temperature difference of 4°C between supply and return temperatures.

The radiant floor system was furthermore investigated in combination with an air-to-water heat pump instead of the generic fuel boiler. The properties of the heat pump were identical to the one used in the actual house with a COP of 3.47 and a total heating capacity of 8.73 kW [2], [4]. All simulations were performed with the weather file from the validated model, which contains weather data from the location of the house.

3. Results

The duration curves for the operative temperatures in the different cases are plotted in Figure 2. The temperatures are taken only for the investigated period from 1st of October to the 30th of April.

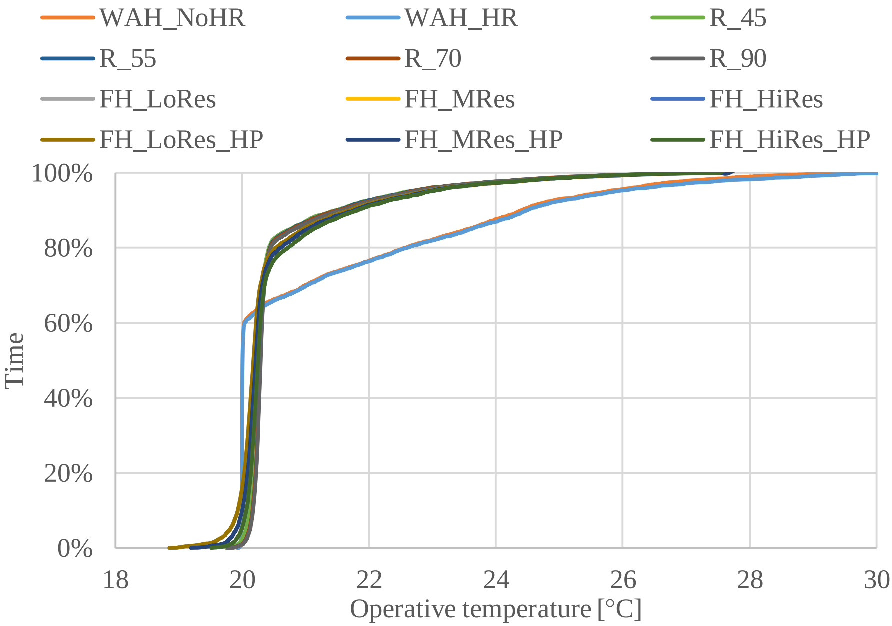

Figure 2 – Duration curves from simulations

The figure shows that the temperatures are clustered together for all water-based systems with a slight difference between water-based and air-based systems.

The plotted operative temperatures in Figure 2 show that the cases with warm-air heating had fewer hours below the set-point of 20°C compared to the water-based radiator and floor heating cases. The figure also indicates that the operative temperatures for the radiator and floor heating cases are more constant compared to the warm-air heating cases as the operative temperature is in the range of 20°C-21°C around 85% of the time. The time for the warm-air heating cases within these temperatures is around 70%.

In order to assess the thermal indoor environment, the temperature ranges stated in EN 15251 [8] was used. The ranges are given in Table 4 and the results are shown in Figure 3.

Table 4 - Temperature range for heating [8]

Category	Temperature range for heating (Clothing – 1.0 Clo)
I	21.0 – 25.0°C
II	20.0 – 25.0°C
III	18.0 – 25.0°C

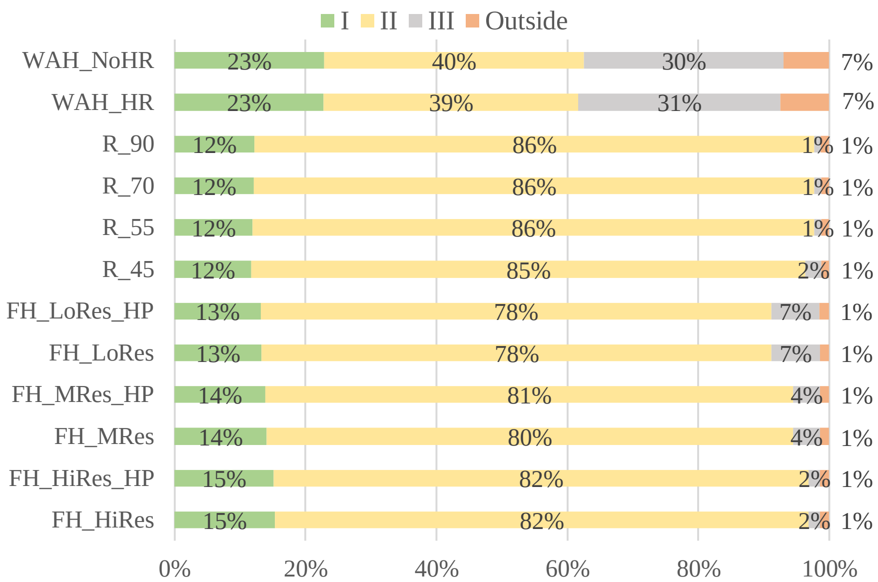

Figure 3 – Distribution of time in indoor environment categories

Figure 3 shows that the warm-air heating cases had the highest amount of time in indoor category I. The warm-air heating cases also had the largest fraction of time outside category I and II. The main reason for the increased time in category I is due to the ability of the warm-air heating to quickly adapt to the step changes whereas the water-based systems require longer time to change the thermal indoor environment, although the water-based system itself reacts immediately to the changes in the indoor environment. However, the increased time to affect the indoor environment for the water-based systems is also the reason for the small amount of time outside category I and II as the temperatures are kept more constant.

The primary energy use for heating and auxiliary energy for the different cases is shown in Table 5 together with the total primary energy use. A primary energy factor of 1 and 2.5 is used for Heating and HVAC aux respectively, except for the heat pump cases where an energy factor of 2.5 was used for heating as well [9].

Table 5 – Primary energy use (1st of October to the 30th of April)

	Heating [kWh]	HVAC aux [kWh]	Total [kWh]
WAH_NoHR	8693	944	9638
WAH_HR	4572	915	5487
FH_HiRes	4460	573	5033
R_90	4429	562	4991
R_55	4421	565	4985
R_70	4420	563	4983
FH_MRes	4397	577	4974
R_45	4392	566	4959
FH_LoRes	4348	581	4929
FH_HiRes_HP	3758	572	4330
FH_MRes_HP	3742	576	4318
FH_LoRes_HP	3697	580	4277

The table shows that in general the cases relying on warm-air heating used the most energy. They were followed by the radiator cases and finally the radiant floor heating cases. The floor heating system was found to be even more efficient when combined with a heat pump.

The differences between the radiator cases and the floor heating cases connected with a boiler are minimal. The point is underlined by the fact that the floor heating case with the highest thermal resistance performs worse in regards to the energy use compared to the radiator cases.

4. Discussion

The results with regard to the thermal indoor environment illustrated in Figure 2 and Figure 3 show that the temperatures inside the house were very close. The largest difference occurs between the air-based and radiant floor systems due to the difference in how quickly they can affect the thermal indoor environment.

It was a necessary condition that the thermal indoor environment was comparable in order to conduct a reasonable comparison of the energy use of the different heating systems.

The difference is clearly illustrated in Figure 2, where the amount of time below 20°C is higher for the water-based systems. The same figure also shows that the air-based systems have a larger amount of hours above 25°C resulting in approximately 7% outside any of the categories given in EN 15251 [8]. The performance of all systems may well be optimized by making individual control schemes.

The largest improvement was found between the two air cases with and without heat recovery. The improvement in energy use during the heating season was determined as 4150 kWh corresponding to an improvement of 43%. The main reason was found to be the amount of primary energy used for heating. Further analyses regarding the application of heat recovery can be found in [1].

The change of heating concept from warm-air heating to water-based radiators resulted in a reduction in the primary energy use. The main difference was found to be in the amount of primary auxiliary energy with a reduction of approximately 38% between warm-air heating with heat recovery and all water-based cases. The savings in energy use between warm-air heating and water-based radiator heating taken as an average of all four radiator cases was found to be 508 kWh corresponding to 9% less primary energy used.

The investigated radiator cases revealed that the working temperatures should be as low as possible in regards to the energy use. The difference was, however, found to relatively small (32 kWh between R_45 and R_90) corresponding to a difference of less than 1%.

Replacing the radiator systems with a radiant floor heating system was found to be negligible in regards to the total amount of primary energy used. The compared values were taken as average values for respective cases; R_45, R_55, R_70 and R_90 for the radiators and FH_LoRes, FH_MRes and FH_HiRes for the floor heating. If the radiant floor system was combined with a heat pump instead of the generic boiler 671 kWh was saved resulting in an improvement of 13%.

For the radiant floor heating cases in general it was found to be important to minimize the resistance caused by the floor covering as this would lead to an increased amount of energy used [10]. The differences between the high resistance cases and low resistances cases were found to have an average value of around 2%. The difference was expected to be higher, but it could be a result of favorable operating conditions for the heat pump.

The total difference between warm-air heating and a radiant floor heating system with low floor covering resistance and combined with an air-to-water heat pump

(FH_LoRes_HP) was 1210 kWh during the heating season which is an improvement of 22%.

All results are found with the use of a dynamic simulation tool which is highly dependent on the input given to the software. The initial simulation model was validated with actual measurements, but changes such as the two 100 L water tanks, etc. could have had an effect on the results.

The obtained results were in some cases very close to each other and in those cases summation errors could have an influence in determining the optimal solution. This is due to the output files given by the software which is given as hourly average values with only two decimals.

5. Conclusion

The energy performance for different space heating systems (warm-air heating, radiator heating, and radiant floor heating) was investigated under similar thermal indoor conditions. The investigations were carried out with a dynamic building simulation model.

The investigation found that implementing heat recovery in the air handling unit is necessary as it was simulated to save up to 43% of the total primary energy used for space heating.

The energy performance could be improved by means of using water-based systems instead of air-based systems. The energy savings were calculated to 9% for radiators and floor heating systems heated by a generic boiler. The main difference was found in the primary auxiliary energy use which was approximately 38% less for all the water-based systems.

The study found that the radiant floor systems had the best energy performance when coupled to an air-to-water heat pump. The total energy savings from warm-air heating with heat recovery to a radiant floor heating system coupled to an air-to-water heat pump were 22%, while achieving similar thermal indoor environmental conditions.

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Paper 6: Andersen, M. E., Schøtt, J., Kazanci, O. B., & Olesen, B. W. (2016). Energy performance of water-based and air-based cooling systems in plus-energy housing. In Proceedings of the 12th REHVA World Congress, CLIMA 2016. Aalborg. **Accepted.**

Energy Performance of Water-based and Air-based Cooling Systems in Plus-energy Housing

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Abstract

Energy use in buildings accounts for a large part of the energy use globally and as a result of this, international building energy performance directives are becoming stricter. This trend has led to the development of zero-energy and plus-energy buildings. Some of these developments have led to certain issues regarding thermal indoor environments, such as overheating.

Thermal comfort of occupants should not be sacrificed for energy efficiency but rather, these should be achieved simultaneously. Although the priority should be to minimize the cooling demand during the design, this is not always achieved and cooling might be needed even in residential buildings.

This paper focuses on the cooling operation of a detached, single-family house, which was designed as a plus-energy house in Denmark. The simulation model of the house was created in IDA ICE and it was validated with measurement data in a previous study. The effects of the cooling demand (internal vs. external solar shading), the space cooling method (floor cooling vs. air-cooling with ventilation system), and the availability of a nearby natural heat sink (intake air for the ventilation system being outdoor air vs. air from the crawl-space, and air-to-water heat pump vs. ground heat exchanger as cooling source) on the system energy performance were investigated while achieving the same thermal indoor conditions.

The results show that the water-based floor cooling system performed better than the air-based cooling system in terms of energy performance and also regarding the energy use of auxiliary components such as pumps and fans. The total reduction in primary energy used was 31% compared to the air-based systems with intake air from outdoors.

The integration of natural heat sinks into the cooling system of the house results in significant energy use reductions. The coupling of radiant floor with the ground enables to obtain "free" cooling, although the brine pump power should be kept to a minimum to fully take advantage of this solution. By implementing a ground heat exchanger instead of the heat pump and use the crawl-space air as intake air an improvement of 37% was achieved.

The cooling demand should be minimized in the design phase as a priority and then the resulting cooling load should be addressed with the most energy efficient cooling strategy. The floor cooling coupled with a ground heat exchanger was shown to be an effective means to minimize the energy use for cooling purposes, and this can contribute to achieving zero-energy or plus-energy targets in future buildings.

Keywords - Radiant floor cooling, air-cooling, ground heat exchanger, crawl-space

1. Introduction

As the requirements for energy use in buildings are tightening, the focus on development of low-energy buildings, zero-energy buildings etc. is increasing. The thermal properties of the building envelopes have improved so much that it leads to undesired overheating in many buildings if no precautions are taken.

A method to avoid overheating is by installing mechanical cooling systems in buildings. Several systems have been developed and different studies have investigated the thermal indoor environment and energy use of these systems [1]–[3].

This study focuses on the investigation of the thermal indoor environment and energy use of different cooling systems by using dynamic building simulation models. The analyzed cooling systems are defined in detail in [4].The investigated house 'Fold' was constructed for an international competition, Solar Decathlon Europe 2012, by the Technical University of Denmark. The house is a single-family, one storey house with a floor area of 66 m^2 and an internal volume of 213 m^3. The house has two large glazing facades facing North (36.7 m^2) and South (21.8 m^2) with a 19° turn to the West.

Further details of the studied house can be found in [5], [6].

Figure 1 shows the exterior views of the house.

Figure 1 – South façade (left) and North façade (right) of Fold

2. Method

A dynamic building simulation model was created in IDA ICE with the purpose of validating the results from the dynamic simulation model with actual measurements from the house. Temperature measurements and measurements of energy use for different components of the house were used in the validation of the model [7]–[9].

The model was constructed with internal gains for two occupants (1.2 met), lighting (180 W), and equipment corresponding to normal housing equipment (380 W). The occupants, lighting, and equipment were controlled after a predefined occupancy schedule given in Table 1.

Table 1 – Schedule for internal gains

	Power output	Schedule [On]
Equipment 1 + PC	120 W + 80 W	Always
Equipment 2	180 W	Weekdays: 17-08, Weekends: 17-12
2 occupants	1.2 Met per occ.	Weekdays: 17-08, Weekends: 17-12
Lighting	180 W	06-08 and 17-23

The design idea for 'Fold' was to control the ventilation system by the CO_2 concentration and have the radiant heating and cooling systems installed in the floor control the thermal conditions in the house.

The HVAC system used a reversible air-to-brine heat pump for the radiant systems as a heat source and sink, and an air handling unit with active and passive heat recovery provided the necessary fresh air. The cooling systems was given a set point of 26°C and was controlled according to the operative temperature.

In order to get a validated model, a weather file was constructed from a combination of measurements taken at the site and supplemented with measurements from a nearby weather station [10]. The temperatures and energy use for certain system parts were then compared to actual measurements from the building to validate and improve the precision of the simulation model.

In this study, alterations were made to the validated IDA ICE model for the current investigations. The HVAC plant was replaced with a system consisting of two 100 L water tanks, one for hot water and one for cold water. The hot water tank was not used in this investigation but had to be included for the simulation model to function. The cold water tank was cooled by a reversible air-to-water heat pump. The cold water tank supplied the required cooling for the cooling coil in the air handling unit or for the floor cooling system depending on the case.

The investigated parameters were: two different types of shading (internal vs. external shading), variation of supply temperatures for air-cooling, air-cooling vs. floor cooling and the effect of implementation of nearby heat sinks; intake air from outdoors vs. intake air from the crawl-space beneath the house, and having the floor cooling system coupled to a ground heat exchanger.

The effect of shading factors was investigated by making simulations with internal shading and external shading respectively. The shading factor was set to 0.6 for the internal shading and to 0.1 for the external shading. The shading was controlled by the solar gain through the windows and the shading was activated when the solar gain exceeded a value of 50 W/m^2 window area.

Three different cases of supply temperatures for air-cooling were investigated with external shading implemented. The investigated supply temperatures were 14°C, 17°C and 20°C. The case with a supply temperature of 14°C was also investigated with the intake air taken from the crawl-space instead of outdoor air. Figure 2 shows the outdoor air temperatures and the air temperatures in the crawl-space.

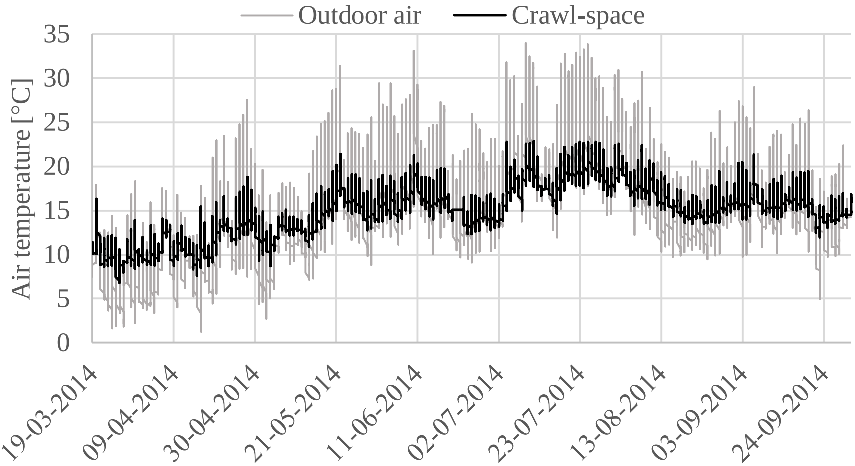

Figure 2 – Outdoor and crawl-space temperatures

The crawl-space acted as a buffer zone where the air temperature was lower compared to the outdoor air temperature in the cooling season and vice versa in the heating season.

Finally, three different cases of floor cooling were investigated. All floor systems had the same thermal properties as the one in the validated IDA ICE model. The first case used the outdoor air as intake air, the second case used the air from the crawl-space, and the third case also utilized the air from the crawl-space. Furthermore, in Case 8 the heat pump for the radiant system was replaced with a ground heat exchanger, thereby fully utilizing the available heat sinks.

Eight different cases were investigated in total. An overview of the differences is given in Table 2.

Table 2 – Case description [4]

Case	Shading	Cooling	Source	Intake air
1	Internal	AC	AWHP	OA
2	External	AC	AWHP	OA
3	External	AC	AWHP	OA
4	External	AC	AWHP	OA
5	External	AC	AWHP	CS
6	External	FC	AWHP	OA
7	External	FC	AWHP	CS
8	External	FC	GHEX	CS

AC – Air-cooling
FC – Floor cooling
AWHP – Air-to-water heat pump
GHEX – Ground heat exchanger
OA – Outdoor air
CS – Crawl-space

For Cases 2, 3 and 4 different supply air temperatures were used: 14°C, 17°C and 20°C, respectively. In Case 1 and 5 a supply temperature of 14°C was used. The simulation period was from 1st of May to 30th of September.

The control of the HVAC system was regulated by the operative temperature. The control by operative temperature was chosen in order to keep the same indoor conditions for all cases to evaluate the energy performance of the different systems.

The cooling cases are described in full detail in [4].

3. Results

The duration curve of the operative temperatures is given in Figure 3. The figure shows how the temperatures were distributed in the investigated period from 1st of May to 30th of September.

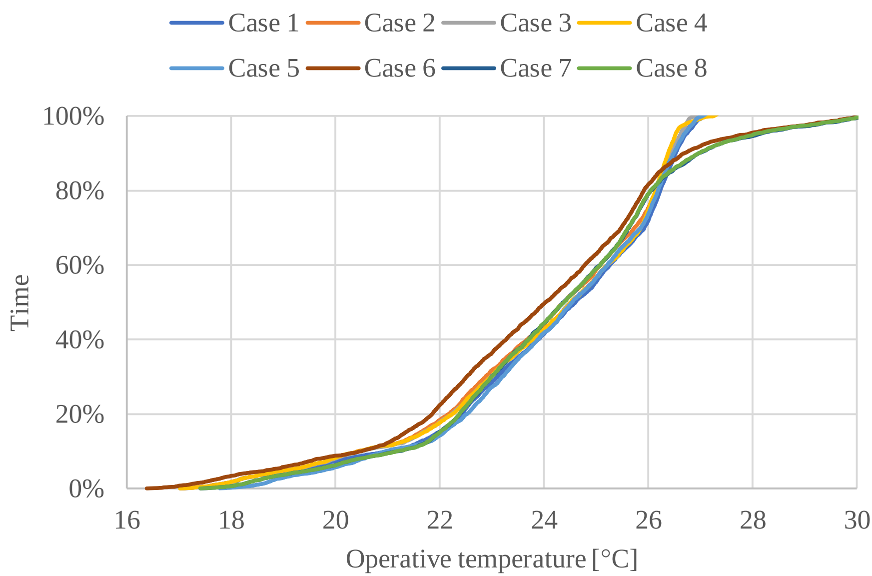

Figure 3 – Duration curves from simulations

Figure 3 shows that the temperatures for all simulations follow a similar pattern. However, there is a slight difference between the radiant and air-based cooling systems. The radiant cooling systems have warmer peak temperatures compared to the air-cooling cases. The duration above 26°C was in contrary found to be lower for the radiant systems compared to the air-cooling cases.

The thermal indoor environment was assessed with the use of standard values for temperature ranges given in EN 15251 [11], Table 3. The distribution can be seen in Figure 4.

Table 3 - Temperature range for cooling [11]

Category	Temperature range for cooling (Clothing – 0.5 Clo)
I	23.5 – 25.5°C
II	23.0 – 26.0°C
III	22.0 – 27.0°C

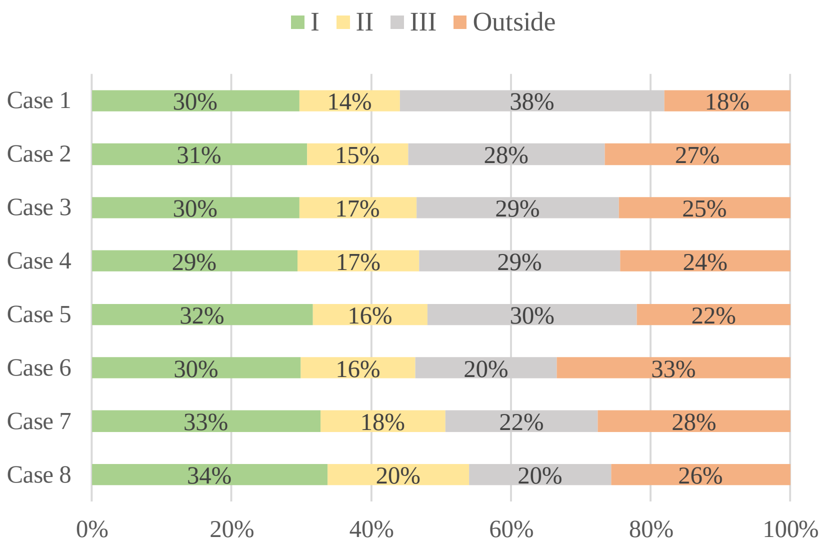

Figure 4 – Distribution of time in indoor environment categories

Figure 4 shows a similar trend to Figure 3. In general, the air systems are able to change the indoor conditions more rapidly due to the supply of cold air directly into the indoors, whereas the floor systems require longer time to change the thermal indoor environment, although the radiant system itself reacts immediately to the changes in the indoor environment.

The primary energy used for all simulated cases are shown in Table 4. The electric cooling corresponds to the primary energy use of the reversible air-to-water heat pump working in cooling mode and the brine pump in the ground loop for Case 8. The primary auxiliary energy use covers the air handling unit and pumps for the floor systems.

The primary energy factor was 2.5 according the Danish Building Code [12].

Table 4 – Primary energy use (1st of May to 30th of September)

	Electric cooling [kWh]	HVAC aux [kWh]	Total [kWh]
Case 1	765	524	1289
Case 4	472	671	1143
Case 2	591	442	1032
Case 3	515	516	1031
Case 5	475	445	920
Case 6	318	415	733
Case 7	277	416	693
Case 8	51	413	464

There is a clear distinction between the energy use of the radiant systems compared to air-based cooling systems with the solar gains. The radiant cooling system (Case 6) was found to have an average total energy use 31% lower than the air-based systems (Case 2-4) when outdoor air was used as the intake. With the crawl-space air as intake the radiant system (Case 7) had a reduced energy use of 25 % compared to the air-based system (Case 5).

4. Discussion

The results in Figure 3 and Figure 4, show that all systems performed comparably with regard to the thermal indoor environment. The radiant systems had higher peak temperatures as shown in Table 5.

Table 5 - Maximum temperatures for each case (1st of May to 30th of September)

Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8
27.2°C	27.1°C	27.0°C	27.3°C	27.0°C	30.4°C	30.5°C	30.4°C

The main reason for the difference was that the radiant systems cannot affect the thermal indoor environment as fast as the air-based systems. The tendency is shown in Figure 5, where the operative temperature for Case 3 (air-based system) and Case 7 (water-based radiant system) are plotted together with the solar gain and outdoor temperature for a hot summer day.

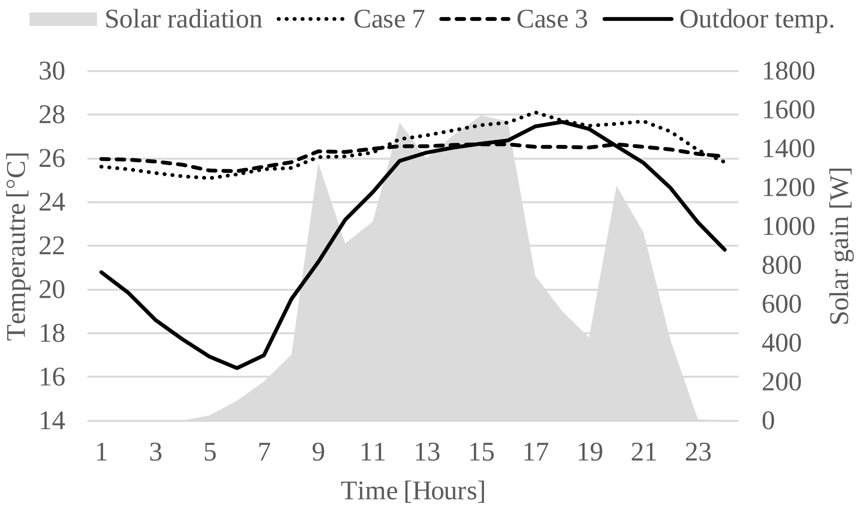

Figure 5 – Temperatures during a hot summer day

Figure 5 clearly illustrates that the air-based system is able to quickly adapt to the step changes by increasing the air change and keeping a constant operative temperature. The radiant system was slower to change the conditions resulting in higher peak temperatures. However, it should be noted that the radiant system reacts immediately to the changes in the indoor conditions due to its surface temperature. The performance of the radiant system in order to address this issue could be improved with a different control strategy.

It was a necessary condition that the thermal indoor environment was comparable in order to conduct a reasonable comparison of the energy use of the different setups.

The results show that there was a great difference in energy use for the radiant systems and air-based systems. The main difference occurred in the energy use for electric cooling. Furthermore, there was also associated a difference in auxiliary energy use. The reduction in total primary energy use from Case 2 to Case 6 was 299 kWh corresponding to 29%. The radiant systems were determined to be more energy efficient while still maintaining a similar indoor environment to the air-based systems.

The air-based systems had a large energy use for the fans in order to continuously sustain the set point especially at higher supply air temperatures. This point was underlined when comparing the two air-based cases, Case 2 and 4. Case 2 with the lower supply temperature had a total energy use that was 10% lower even though they had the same cooling demand. Case 2 used 119 kWh (20%) more primary energy for cooling of the air. However, this was offset with the energy use for auxiliary equipment which was 229 kWh (34%) less.

The results show that it was beneficial to use external solar shading compared to internal shading. By reducing the solar gains, the energy use was reduced by 20%.

The influence of natural heat sinks was also investigated. The investigation of the intake air for the AHU showed that there is a potential to lower the energy use for cooling. Case 2 and Case 5 have the same supply temperature but Case 5 utilizes the colder air temperatures in the crawl-space. Case 5 had an electric energy use of 116 kWh (20%) lower than Case 2. The main difference was the energy use for the cooling coil in the AHU as the auxiliary energy use was within 1% of each other.

The simulations showed that it was beneficial to have a floor cooling system coupled to a ground heat exchanger. Case 7 and Case 8 have the same supply water temperature but the only difference was that Case 8 utilizes a ground heat exchanger. Case 8 had 33% lower energy use compared to Case 7. When both of the investigated heat sinks were implemented (Case 8) the total energy use was lowered with 37% and the energy use for electric cooling with 84% compared to Case 6 with no heat sinks. There is, of course, associated with a large installation cost with the implementation of the borehole. However, it also provides a great possibility to lower the energy use in detached residential houses.

The simulation model has been validated with on-site measurements. However, the simulation models are not perfect and even though the original model was validated, assumptions were made in this study to simulate the different systems. For instance, two water tanks of 100 L each were implemented due to limitations in the software. The assumptions could have affected the outcome of the investigation.

5. Conclusion

The energy performances of different space cooling systems with similar thermal indoor environments were compared using dynamic building simulations, using a single-family house as a case study.

The investigation found that it was beneficial to lower the cooling demand by means of external solar shading compared to internal solar shading. It was concluded that the air-based systems have the ability to affect the conditions for the thermal indoor environment more rapidly compared to the radiant systems. In contrary, the water-based radiant systems were found to be the more energy efficient cooling systems as they have a lower energy use for electric cooling as well as auxiliary energy with a total energy saving of 31% with intake air from outdoors. Furthermore, the water-based systems achieved fewer hours of overheating compared to the air-based systems and provided more stable thermal conditions.

It was found that combining the radiant floor systems with a ground heat exchanger would be beneficial in terms of energy performance, especially if the energy used by the circulation pump was kept at a minimum. The simulation found that total energy use was reduced with 37% and the energy use for electric cooling with 84% when heat sinks was utilized.

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Sustainable heating, cooling and ventilation of a plus-energy house via photovoltaic/thermal panels

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ABSTRACT

Present work addresses the HVAC and energy concerns of the Technical University of Denmark's house, Fold, for the competition Solar Decathlon Europe 2012. Various innovative solutions are investigated; photovoltaic/thermal (PV/T) panels, utilization of ground as a heat source/sink and phase change materials (PCM).

The development of a building integrated photovoltaic/thermal (BIPV/T) system and its performance evaluation compared to a PV installation built of the same photovoltaic cells are also presented. Annual results show that having the combined PV/T system is more beneficial compared to having two separate systems.

PV/T panels enable the house to perform as a plus-energy house. PV/T panels also yield to a solar fraction of 63% and 31% for Madrid and Copenhagen, respectively.

The ground heat exchanger acts as the heat sink/source of the house. Free cooling enables the same cooling effect to be delivered with 8% of the energy consumption of a representative chiller.

The major part of sensible heating and cooling is done via embedded pipes in the floor and ceiling. Ventilation is used to control the humidity and to remove sensory and chemical pollution.

A combination of embedded pipes and PCM was simulated. Results show energy savings up to 30%, for cooling season in Madrid.

1. Introduction

Buildings play a key role within the 20-20-20 goals of the European Union due to the fact that they are responsible for 40% of the energy consumption within the member states [1]. Therefore an urgent and effective transition is necessary in order to reach the almost passive house levels dictated by various standards.

These goals are in parallel directions with the main goals of the competition, Solar Decathlon, where the main goal is to design, build and operate an energetically self-sufficient house that uses solar energy as the only energy source [2].

Technical University of Denmark, herein DTU, joined the competition, Solar Decathlon Europe 2012 with the house "Fold". During the course of this study, an entire HVAC system for a single family house has been designed, simulated and tested.

A house, other than just providing shelter, should also be able to provide necessary and optimal thermal comfort (including indoor air quality) for the occupants however this goal should be achieved with the lowest possible energy consumption. The design of the HVAC system intended to satisfy both of these needs. Innovation was a driving force and this was achieved via taking advantage of well-known and proven systems and integrating them into the HVAC system and coupling them with relatively less mature technologies.

The HVAC system of the house consisted of: ground heat exchanger (GHX), embedded pipes in the floor and in the ceiling, ventilation system (mechanical and natural), domestic hot water (DHW) tank and photovoltaic/thermal (PV/T) panels placed on the roof. The design methodology, further information about the components and main results are presented in the following sections.

2. Design of the house

The project being multi-disciplinary by its nature, some of the design values and parameters were fixed without the possibility

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of alteration. Also some of the design values were fixed due to the commercially available products and their capacities.

The house is a detached, one-storey, single family house with an indoor floor area of 66.2 m^2 and with a conditioned volume of 213 m^3. The design of the house intends to minimize heat gain to the house from the ambient. The house's largest glazing façade is oriented to the North side, with a 19° turn toward West.

The house is constructed from wooden elements. Walls, roof and floor structures are formed by placing prefabricated elements in a sequential order and sealing the joints. North and South glazed façades are inserted later and the joints between glazing frame and house structure are sealed.

Prefabricated house elements are made from layers of wooden boards, which in combination with I beams in between form structural part, and mineral wool insulation. The house is insulated with two types of insulation; 20 cm of conventional mineral wool and 8 cm of compressed mineral wool.

The glazing surfaces in North and South sides of the house are covered by the overhangs which eliminate direct solar radiation to the house during the summer season. For the winter season direct solar radiation enters the house, creating a favorable effect. No active shading systems were installed in the house except for the skylight window.

Inside the house, there is a single space combining kitchen, living room and bedroom areas. Shower and toilet areas are partly separated by partitions. Technical room is completely isolated from the main indoor space, having a separate entrance. Wall between technical room and indoor space is insulated with the same level of insulation as the outside walls. The house, structural element and respective areas can be seen in Fig. 1 and Table 1.

The house is fully functional therefore it is equipped with different appliances such as: PC, refrigerator/freezer, clothes washer, clothes dryer, dishwasher, oven, TV and DVD player. Electrical power of the installed equipment is 1.5 kW.

3. Design methodology of the HVAC system

With the given constraints on the system, an entire HVAC system for the house had to be designed following the ambitions given in Section 1.

To design the heating, cooling and ventilation system, load calculations were performed. Construction of the house was defined by the architectural design team. This design was taken as the basis for load calculations. Even though the idea behind the architectural design of the house was to adjust certain parameters including, orientation, tilt of the roof and walls, glazing areas, etc. this option was not realized in the simulations and in the calculations.

The initial design conditions required for the house to be fully functioning in two different climates: Denmark (Copenhagen) and Spain (Madrid). The resulting heating and cooling needs are as follows: maximum cooling load is 52.0 W/m^2, average cooling load is 35.2 W/m^2, maximum heating load is 45.6 W/m^2 and average heating load is 26.6 W/m^2, given the indoor floor area of 66.2 m^2.

Even though the design was mainly aimed at providing the comfort conditions during the competition period, it had to be assured that the house performs as intended all year round. This was implemented with different set-points in the simulations as explained in the respective section.

The only electrical energy source of the house is solar energy, utilized via photovoltaic panels placed on the entire roof area. The electrical system is designed to be grid-connected with no batteries. Coupled with the photovoltaic panels is the thermal system, which absorbs the heat produced by photovoltaic panels and utilizes it in the DHW tank, making combined photovoltaic/thermal system (PV/T).

Heating and cooling system of the house is water based, with low temperature heating and high temperature cooling principle. Heat source/sink is the ground, utilized via a borehole heat exchanger. Free cooling is obtained during the cooling season without any extra energy consumption other than the circulation pump and ground coupled heat pump is used to achieve the necessary supply temperature to the embedded pipes during the heating season.

As an addition to the space heating and cooling, ground heat exchanger could also be utilized for the PV/T cooling. Yet, initial evaluations showed that this concept was too expensive to be realized, since it requires extra capacity of the ground heat exchanger.

In order to regulate the air quality in the house, mechanical and natural ventilation systems are installed. The mechanical ventilation consists of two supply diffusers to the space and four exhausts (kitchen hood, bathroom, toilet and the clothes dryer).

To increase the building's thermal mass, an option of installing phase change material, herein PCM, into the structure of the building was considered. The model of active cooling using PCM was chosen. Pure PCM material is stored in a metal container. The container is equipped with a piping system, to discharge the heat stored in the material.

The house being high-tech, it stores great amount of machinery and electronic equipment which operate the house. All of these components release heat to the environment. As it is a need to limit heat production in the house, a solution is to isolate all equipment which is not used by the occupants on a daily basis. The equipment is placed in the technical room, which has no direct thermal connection to the inside area.

4. Design methodology of the PV/T system

In order to justify the advantages of combining electrical and thermal part in one element, various investigations were carried out. The main goal was to keep the cell temperature under control and keep the electrical efficiency close to the nominal value and also to utilize the heat that is gained from cooling the cells for the various heating needs of the house (domestic hot water, hot water consuming appliances, but not space heating). Special attention was given to the hydraulic division of PV/T panel, to the practical solution for dismountable joints between panels and to the common design of thermal and electrical parts.

4.1. Test of the thermal part

Parametric analyses were made in order to find out the panel's effectiveness in relation to different configurations of lateral pipes, 6 and 10 per meter (Fig. 2). It can be observed from Fig. 2 that spacing of 100 mm can utilize more solar energy than spacing of 166 mm. The most significant difference appears when the temperature difference between surrounding and PV/T surface is negligible.

Temperature fluctuation across the absorber plate for two different spacing of lateral piping in PV/T panel can also be seen in Fig. 2. The calculation was carried out for solar irradiation of 1000 W/m^2, 25 °C and no wind. The peaks indicate intermediate space between two pipes where the temperature raises the most. It was desired to have as even temperature over the absorber as possible, thus spacing of 100 mm was chosen.

The PV/T panel was tested at an outside testing facility, with a tilt of 67.5° from the horizontal and oriented to the true South, the test setup can be seen in Fig. 3.

The expressions used to calculate the efficiencies are as following:

$$\eta_{\text{Thermal with active cells}} = 0.422 - 5.628 \cdot \frac{\Delta T}{G} \tag{1}$$

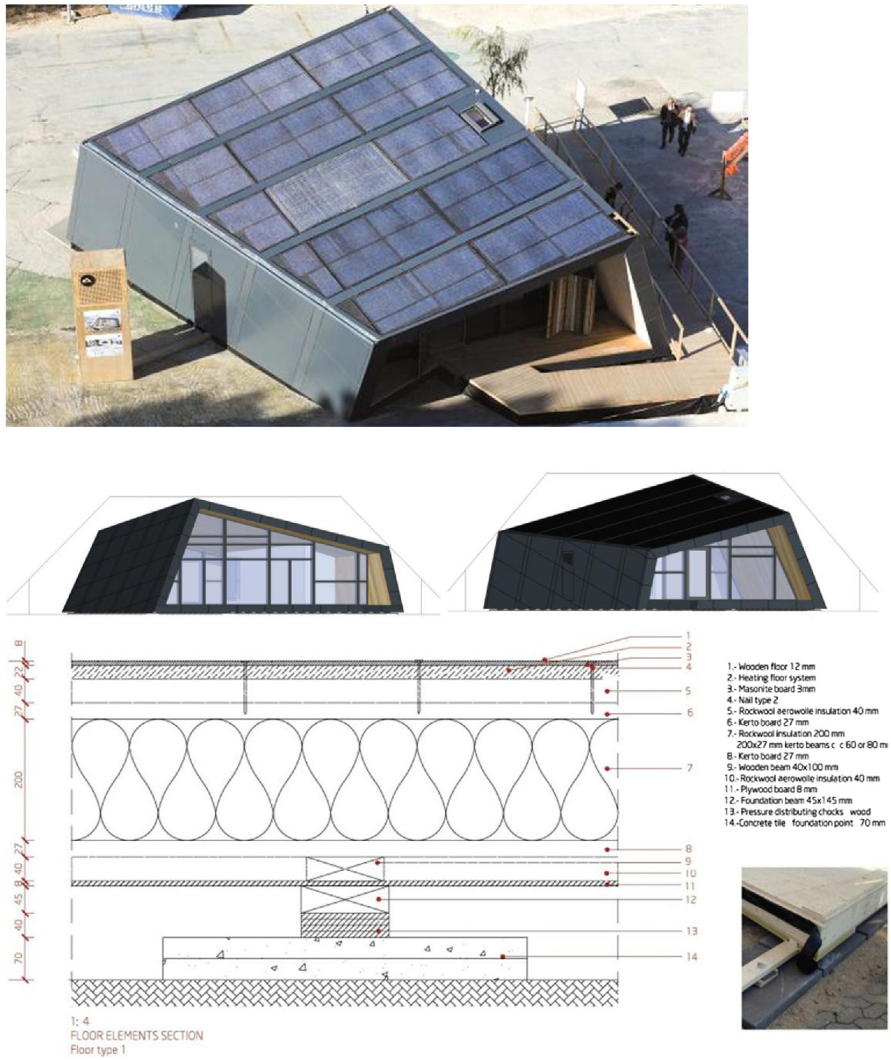

Fig. 1. Bird's-eye view of the PV/T system, North-East and South-West sides of the house and the structural element [3].

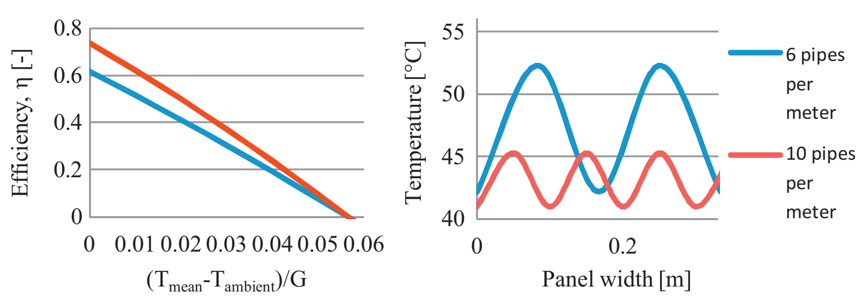

Fig. 2. Thermal efficiency and temperature fluctuation for two different spacing of lateral pipes [4].

Table 1
House construction details.

External walls	South	North	East	West	Floor	Ceiling
Area [m^2]	–	–	37.2	19.3	66.2	53
U-value [W/m^2-K]	–	–	0.09	0.09	0.09	0.09
Windows	South	North	East	West	Floor	Ceiling
Area [m^2]	21.8	36.7	–	–	–	0.74
U-value [W/m^2-K]	1.04	1.04	–	–	–	1.04
Solar transmission	0.3	0.3	–	–	–	0.3

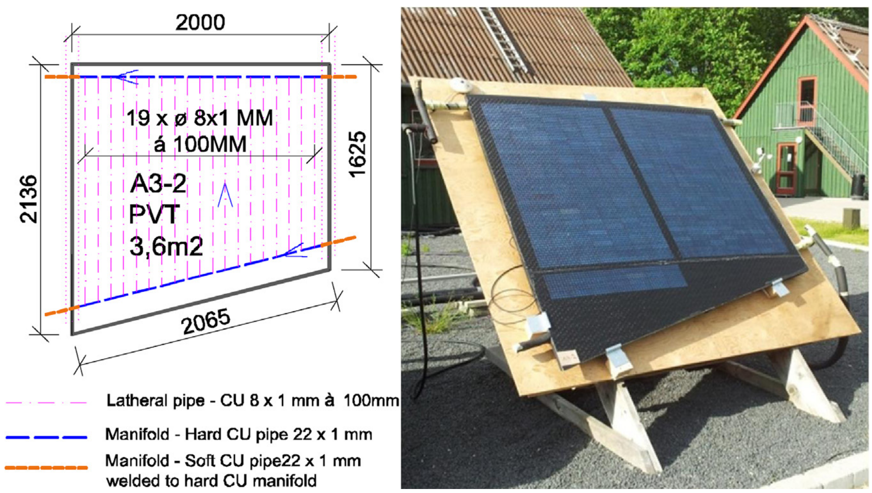

Fig. 3. The tested PV/T panel [5].

$$\eta_{\text{Thermal with passive cells}} = 0.483 - 5.485 \cdot \frac{\Delta T}{G} \quad (2)$$

Thermal efficiency was measured under two circumstances: with active and passive PV cells. In the case with active cells (1), 42.2% of solar irradiation was transformed to heat. During the measurement with passive cells (2), efficiency of 48.3% was reached. The difference is due to the conversion of irradiation into electricity (Fig. 4).

4.2. Test of the electrical part

Electricity was generated by mono-crystalline silicone cells. Squared cells were divided into three rectangular pieces with dimensions of 41 mm × 125 mm to decrease the risk of failure due to panel bending. It is also possible to cover a larger area with smaller cells. The by-pass diodes were integrated inside the lamination (8 and 14 cells per diode). Thus, in case of failure, only a certain number of cells are out of order and the panel still produces power. No junction boxes were used for the PV/T modules; only fixture for cable outlets.

The electrical testing was performed on the same test setup as the thermal test. Voltage and current were measured using the "Uganda" method, as seen in Fig. 5.

The expression used to calculate the efficiency is as following:

$$\eta_{\text{PV cells}} = 0.159 - 0.583 \cdot \frac{\Delta T}{G} \quad (3)$$

The results of the electrical tests (Fig. 6) showed that the efficiency curve depends on solar irradiation and temperature difference between the panel and the ambient. The electrical test of the panel was done under moderate Danish summer weather conditions, temperature and irradiation values can be seen in Table 2. The efficiency curve was idealized and divided into two zones representing electrical efficiency when the cells are actively cooled by fluid circulation and to when the panel is cooled only naturally. The three marked efficiency levels correspond to Standard Test Conditions but with varying panel temperature. The three different efficiency levels illustrate these scenarios; 32 °C for PV/T cooling via ground; 35 °C for PV/T charging the DHW tank and 66 °C for normal PV panel operation.

The electrical characteristics of the PV cells stayed unchanged regardless of the cooling mode, but the active cooling provided higher electrical efficiency, in comparison to PV with the same boundary condition.

A new type of efficiency, hybrid efficiency, was introduced. Up to 58% of the solar energy, that is incident on the surface of the PV/T panel, is utilized, as seen in (4). The hybrid efficiency (Fig. 7) represents the sum of the electrical efficiency and the thermal efficiency if both systems work simultaneously.

$$\eta_{\text{PVT hybrid}} = 0.583 - 6.281 \cdot \frac{\Delta T}{G} \quad (4)$$

5. HVAC system and control concept

The individual operation of the components of the HVAC system and operation of the system as a whole had to be controlled in order to assure optimal performance. This was mainly done on a seasonal basis (heating/cooling) and with more detailed conditions within each season.

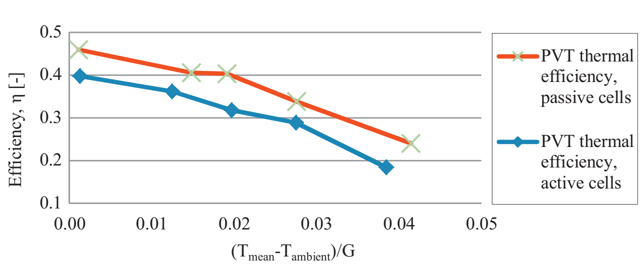

Fig. 4. Thermal efficiency of the PV/T panel [11].

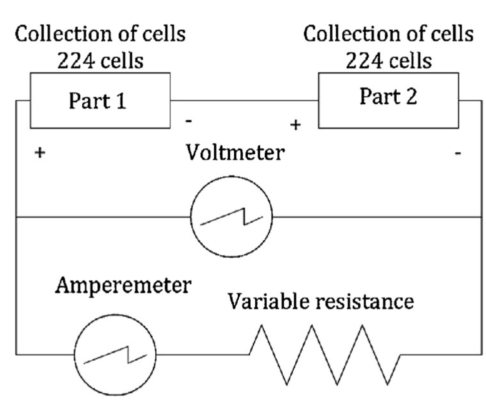

Fig. 5. Electrical scheme of the test setup [4].

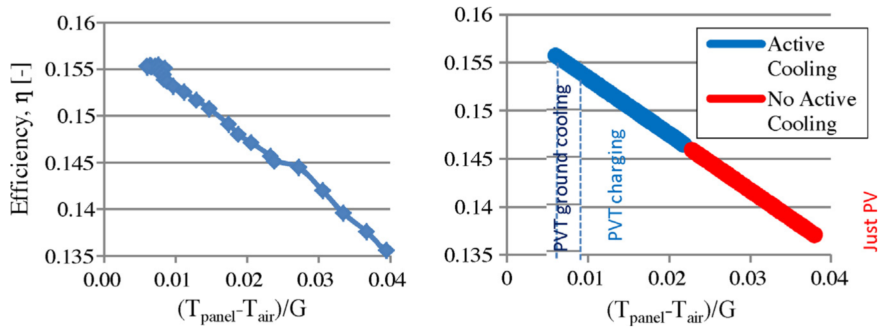

Fig. 6. Measured and idealized electrical efficiency of the PV/T panel [5].

Table 2
Active and no active cooling effect.

	Efficiency [%]	Panel temperature [°C]	Air temperature [°C]	Solar irradiation [W/m^2]
Active cooling	15.5	32 ± 0.5	22.5 ± 0.5	880–950
No active cooling	13.5	66 ± 0.5	22.5 ± 0.5	880–950

The most significant parameters of the HVAC system and how they interact with the rest of the components are presented in the following.

The ground heat exchanger was designed to be a borehole with a depth of 120 m, single U-tube configuration and with a diameter of 0.12 m. The inner and outer radii of the heat exchanger pipes were 0.013 m and 0.016 m, respectively. Obtained borehole resistance was 0.1 m-K/W and total resistance to the undisturbed ground (8.3 °C and 14.3 °C for Copenhagen and Madrid, respectively) was 0.37 m-K/W.

The space heating and space cooling in the house is provided by the pipes that are embedded in the floor and the ceiling. It is a dry radiant system, having piping grid installed under the wooden layer, with an aluminum layer for better thermal conductance. Space heating is only obtained by the embedded pipes in the floor and space cooling is obtained by embedded pipes in the ceiling and, if necessary, in the floor. The supply and return flows will be coming from/going into the installed ground heat exchanger. In order to control the water flow and the supply temperature, a mixing station is installed.

The details of the embedded pipe system are as following:

- Ceiling; foam board system, with aluminum heat conducting device, PEX pipe 12 mm × 1.7 mm. In total six circuits are designed for the ceiling system, with maximum flow rate in one circuit of 0.07 m^3/h.
- Floor; chipboard system, with aluminum heat conducting device, PEX pipe 17 mm × 2.0 mm. In total four circuits are designed for the floor system, with maximum flow rate in one circuit of 0.07 m^3/h for the cooling case, and 0.15 m^3/h for the heating case.

The installed air handling unit, herein AHU, can provide an air flow rate up to 320 m^3/h, which is 1.5 ach at 100 Pa. This flow rate fully covers the need for the design flow rate.

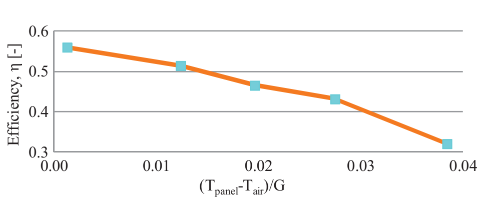

Fig. 7. Hybrid efficiency of PV/T panel [4].

AHU has two heat recovery systems: passive (cross flow heat exchanger) and active (reversible heat pump coupled with the DHW tank). Active heat recovery is obtained via a heat pump cycle that changes the evaporator/condenser in the supply air duct to the interior. This is achieved via a 4-way valve in the heat pump cycle. Passive heat recovery system has an efficiency of 88% (sensible heat). Thermal energy of the exhaust air is transported to the supply air. By pass mode is possible.

Mechanical ventilation gives more control over the parameters including temperature, relative humidity and CO_2 concentration however due to the use of fans it consumes a certain amount of energy (40 Wh/m^3). This amount of energy can be eliminated when the outside conditions are feasible for natural ventilation. Natural ventilation is possible via two windows in South and North façades and the operable skylight window.

PV/T part could also directly interact with the ground. PV/T part is intended to produce electricity and produce heat for domestic hot water and domestic appliances (dishwasher, clothes washer and clothes dryer).

PV/T area (67.8 m^2) is hydraulically divided into Part A (45.4 m^2, 3 × 3 m PV/T panels) and Part B (22.4 m^2, 2 × 2 m PV/T panels), for different control purposes and also for lowering the pressure drop on supply/return piping. Part A is solely intended to charge DHW tank. If there is any flow in Part A this is when there is a DHW need and the flow can only be directed to the DHW tank.

On the other hand, Part B serves for two purposes; charging the DHW tank and PV/T cooling. When there is a DHW need, Part B also contributes to the charging of the DHW tank. Initial simulations and calculations showed that the ground (one borehole) is not capable of providing necessary supply temperature to the embedded pipes when house cooling and PV/T cooling are active simultaneously. Therefore PV/T cooling option is only applicable when house does not need cooling.

The PV/T panels were interconnected in six separate electrical strings. The most of the strings were made of 448 full cells (three cut cells) with nominal voltage of 298 V (0.66 V per cell) and short circuit current of 8 A. Total installed nominal power was 10.8 kWp that was electronically cut down to 9.2 kWp by two inverters. In total, 9914 cells were used with a cell area of 50.8 m^2.

A drain-back tank was included in the thermal circuit, between the PV/T loops and the DHW tank. All piping in the level above the drain back tank was constructed with a minimal slope of 2% to the reservoir. In idle pump mode, the heat transfer medium is drained from the collectors into the 100 l reservoir

Table 3
Obtained results for both locations, annually (TRNSYS) [8].

	Copenhagen	Madrid
Heat pump, heat to load [kWh]	6932.3	4351.3
Energy balance, ground [kWh]	−3128.8	−548.6
Free cooling total [kWh]	1301.1	2042.6
Free cooling to the house [kWh]	1195.8	1661.0
Free cooling to the PV/T [kWh]	105.3	381.6

Table 4
Initial and average temperatures and heat balance of the ground after 10 years of operation (TRNSYS) [8].

	Copenhagen	Madrid
Initial ground temperature [°C]	8.3	14.3
Average ground temperature [°C]	7.8	14.2
Heat balance of the ground [MWh]	−28.7	−2.8

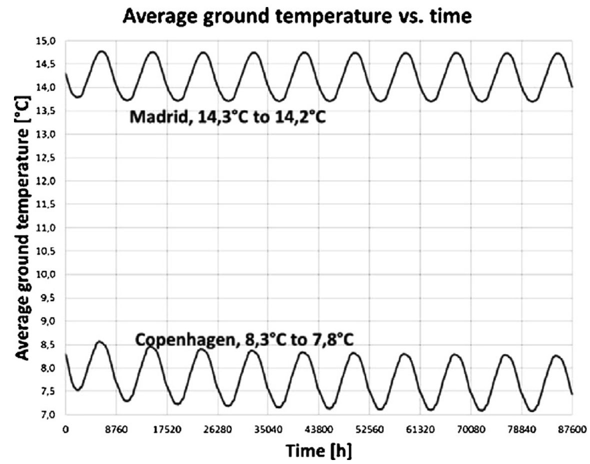

Fig. 8. 10-Year average ground temperatures for Copenhagen and Madrid (TRNSYS).

tank, from where the liquid fills the collectors when the pump starts.

Low-pressure drop of the solar thermal part with Tichelmann connection is using the drain-back tank system. This combination allows the operation of the system without any additional anti-freeze liquid, free of thermo-syphoning effect and without boiling or freezing risk in any climate around the world.

To keep as much equal flow rate per square meter across the array as possible, the flow was optimized by various diameters of piping and by balancing valves. The balancing was done in situ when the Fold was assembled in Madrid.

The DHW tank of 180 l is equipped with two spiral heat exchangers and an electric heater. One of the spiral heat exchangers is connected to the PV/T loops via the drain-back tank and the other one to the active heat recovery system of the ventilation. The top part of the tank (54 l) is heated by the electric heater (1.5 kW).

6. Dynamic simulations of the house

In order to evaluate all year round performance of the house, commercially available dynamic building simulation software, TRNSYS [6] was utilized.

Simulations were carried out for Copenhagen and Madrid. Weather files used were; International Weather for Energy Calculations (IWEC) and Spanish Weather for Energy Calculations (SWEC), respectively.

May to September months (both included) were considered as cooling season and rest of the months were considered as heating season. Relative simulation parameters were adjusted accordingly.

Same load profiles for occupants, lighting and equipment were implemented for Copenhagen and Madrid. There are two occupants in the house with 1.2 met. Occupants are assumed to be away from 8:00 to 16:00 during the weekdays and from 12:00 to 17:00 during the weekends.

The lighting load is 222 W (3.4 W/m^2). Lights are assumed to be ON from 05:00 to 08:00 and from 16:00 to 22:00 every day. Different equipment is ON and OFF during the day. Following values are expressed with respect to the maximum value; for the weekdays, load is 5% all the time except from 02:00 to 03:00 where load is 20% and except from 19:00 to 20:00 where load is 62%. For Saturday, from 7:00 to 8:00 the load is 15%, from 8:00 to 9:00 the load is 34% and for Sunday from 2:00 to 3:00 the load is 20%.

Set-points for the temperature have been defined as 21 °C ± 1 K for heating and 25 °C ± 1 K for cooling season, following [7]. These values refer to the Category 2 of comfort conditions for living spaces in residential buildings in the respective standard.

Also an investigation using dynamic building simulation software BSim was made to evaluate if PCM application in the designed house would bring the desired effect of decreasing energy consumption for heating and cooling. In total four cases were simulated, starting with the simplest conventional structure and at last having a structure which is fully packed with PCM:

- 50 mm PCM layer in direct contact with indoor space (green column).
- Embedded pipe system in wooden construction (blue column).
- 50 mm PCM layer covered with 10 mm plywood layer (orange column).
- 50 mm PCM layer on the ceiling, covered by 10 mm plywood layer and embedded pipe system in the wooden floor structure (violet column).

Only cooling season was investigated due to limitation of the software. Results of these simulations are presented in the following section.

7. Results

Presented results are mostly from simulations and the respective simulation software is indicated in the parentheses where applicable.

Simulation results for the designed ground heat exchanger are presented in (Table 3).

Also long-term behavior of the ground has been investigated, results are presented in Table 4 and Fig. 8.

Table 5
Obtained results for PV/T panels and DHW consumption, annual (calculations and TRNSYS) [5].

Unit	Variable	PV/T (heating mode)	PV	Solar thermal	PV/T – (PV + T)
%	Efficiency	15.34 + 36.6	13.59	42.8	
kWh/year	Net annual el. en. balance; Copenhagen[b]	7434 + 242[a]	7214	259	+203 (2.6%)[a]
kWh/year	Net annual el. en. balance; Madrid[b]	11,393 + 495[a]	10,970	530	+388 (3.3%)

[a] Heat transferred to electricity in a way, how much electricity would be used to charge the 180 l DHW to 60 °C by a heat pump (COP 3.28 for heating).
[b] Solar fraction was obtained to be 62.7% for Madrid and 30.5% for Copenhagen, annually [8].

Table 6
Energy consumption by building need (TRNSYS) [9].

	Copenhagen	Madrid
Heating [kWh/m^2]	31.6	20.7
Cooling [kWh/m^2]	0.5	1.0
Ventilation [kWh/m^2]	0.7	5.2
DHW [kWh/m^2]	7.3	3.8
Rest of the electricity [kWh/m^2]	5.6	4.4
Total electricity consumption [kWh/m^2]	45.6	35.1
Total primary energy consumption [kWh/m^2]	114.1	105.2
Total energy balance (electricity) [kWh/m^2]	66.7	137.0

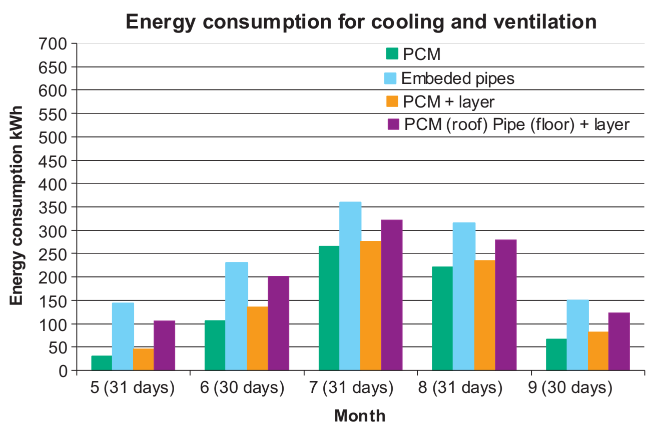

Fig. 9. Energy consumption during cooling season in Madrid for different constructions (BSim) [9].

Presented results for PV/T system are obtained from calculations and simulations (Table 5).

In order to evaluate the house on an all year round basis, also simulations have been carried out and the results of the simulations are presented in Table 6.

The results of the dynamic simulation for the PCM application in the house are presented in Fig. 9 (cooling season is May–September).

8. Discussion

The evaluations show that the house performs as a plus-energy house on an annual basis however it should be kept in mind that the results are aggregated values over a year and the time of energy production and consumption does not necessarily correspond to each other.

For both of the locations, the highest contribution to the energy consumption is from the heating demand. This is mainly due to the North and South glass façades. This effect somewhat offsets the positive effect of the low *U*-value of the walls.

Embedded pipes in the floor and ceiling are advantageous in achieving the goal of energy efficient heating and cooling, mainly due to the high temperature cooling and low temperature heating concept enabling the natural resources to be integrated into the HVAC system, in this case being ground heat exchanger.

Free cooling effect is possible to observe for both of the locations, taking Madrid as an example, the same amount of cooling would have been delivered with 848 kWh of electricity compared to 65 kWh of electricity, if it were to be done with a representative chiller. For the heating case, long-term effects should be considered and kept in mind while realizing this design.

PV/T panels enable the house to be self-sufficient and even produce more energy than it consumes on the electrical side and PV/T panels also contribute significantly to the heat demand for the domestic hot water consumption.

The maximum thermal efficiency of the PV/T panel, with passive solar cells was measured as 48%, when the PV/T panel was cooled by water at 20 °C. With active solar cells the maximum efficiency was decreased by 6–42%.

Even though the mechanical ventilation provides better control over the important comfort parameters, natural ventilation possibilities should be exploited until the limits in order to save energy.

The results from the BSim simulation proved that using thermal mass such as PCM decreases energy consumption for cooling. The highest energy savings using PCM appear in early and late cooling season months, up to 30%. At the peak month energy consumption using PCM is lower, yet only approximately by 20%, compared to conventional water based cooling system.

9. Conclusion

The main goal of this study was to design the heating, cooling and ventilation system of DTU's house for the competition, Solar Decathlon Europe 2012 and to power it with photovoltaic/thermal panels however it was not limited to this extent. Further evaluations were carried out regarding different energy saving and energy efficiency mechanisms.

The competition rules regarding temperature, relative humidity and indoor air quality were on the focal point of the design however an all year round approach was utilized in order to assure that the house and its systems can perform as close as possible to optimum. Keeping these constraints in mind, the components and the HVAC system were designed, simulated and fortunately most of these components were tested and evaluated in full-scale once the house was erected in Denmark and later during the competition in Madrid. During the competition period, it was observed that the designed system is capable of meeting the requirements regarding comfort conditions during most of the time [10].

One of the aims of the study was to develop a commercially available PV/T product because currently, the conditions of the PV/T market are very tentative. Many manufacturers promote a number of various combinations of different technologies, but they offer only the end-product which is not the key to sustain the proclaimed hybrid efficiency. A well designed combination of thermal and electrical part can ensure the proclaimed rise of effectiveness and let the PV/T technology to excel in the advantages compared to the conventional solutions.

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ANALYSIS OF A PLUS-ENERGY HOUSE FOR IMPROVED BUILDING AND HVAC SYSTEM DESIGN

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Abstract

The purpose of this study is to evaluate the energy consumption and indoor environment of a plus-energy house located in Denmark. The house is a detached, single-family house built for the Solar Decathlon Europe competition in 2012 by the Technical University of Denmark. The house is currently being used as a full-scale experimental facility. The measurement period considered in this study is from September 26, 2013 to April 3, 2014.

Measurements from the house are compared to the results obtained from a simulation model of the house in the commercially available building simulation software, IDA ICE. Different parameters are investigated in order to improve the energy performance and the indoor environment of the house. Some of the investigated parameters are the orientation of the house, positioning and areas of the windows, thermal bridges, and infiltration. Various improvement suggestions are presented based on the parametric studies and the comparison of the results.

It was found that there is an average standard error of 2.7 % when comparing the simulation results with the obtained experimental data regarding the operative temperature. The simulated energy consumption of the heat pump is 13.7 % lower than the measurements. The ventilation system is determined to have 23 % lower energy consumption in the simulated model when compared to the measurements.

The improved model has a 25 % reduced window area, and an improved building envelope. The study concludes that the improved model reduces the total energy consumption by 68 %. The time in thermal indoor environment category I, according to EN 15251, is improved from 71 % to 97 %.

Keywords: Energy consumption, indoor environment, IDA ICE, full-scale measurements, plus-energy house.

1 Introduction

Energy consumption in buildings will continue to be a vital part of achieving the goal of reducing the greenhouse gas emissions. As the buildings evolve to be more energy efficient, some undesired effects are also appearing. A result of this evolution is an increase of thermal discomfort for the occupants due to factors such as overheating and poor indoor air quality.

There has been a growing international focus on reducing the energy consumption to reduce the impacts of global warming. The energy used in buildings represents 41 % of the total energy demand in Europe, Olsen (2014), which encourages the development of zero energy houses and plus energy houses, Greenmatch (2014). The low energy houses have, as a result of the low infiltration, low U-values and often due to large glazing façades, a tendency to overheat resulting in discomfort for the occupants, Larsen (2011).

This study is focusing on a plus-energy house, Fold, which was constructed for the international student competition Solar Decathlon Europe 2012 by the Technical University of Denmark. Fold is a plus-energy house because it produces more energy than it consumes on a yearly basis, Kazanci et al. (2013), Kazanci et al. (2014). The house was located in Denmark and since September 2013, it has been used as a full-scale experimental facility where different heating, cooling and ventilation strategies had been tested. Various physical parameters were measured, as well as the energy production and consumption. Even though the house is classified as a plus-energy house due to the electricity production by the photovoltaic/thermal panels placed on the roof, there is potential for improvement regarding the energy consumption and indoor environment.

The main goal of this study is to provide improvement suggestions (i.e. lowered energy consumption and improved indoor environment) based on the parametric analyses, carried out with computer simulations which are validated with measurements. One of the main focuses has been on the window area and the respective heating demand, as indicated by previous studies on the house, Kazanci et al. (2012), Kazanci et al. (2014).

The effects of different parameters which affect the energy performance and indoor environment of the house were studied via the commercially available building simulation software IDA ICE, EQUA (2014). Operative temperatures obtained from the simulation model and from the house were compared in order to validate the simulation model. The energy consumption of different components of the HVAC system was also compared.

2 House details

The house considered in this study is a single family, detached, one-storey house with a floor area of 66 m^2 and a conditioned volume of 213 m^3. The house had two large glazing façades oriented to North and South. The largest glazing façade was oriented towards the North with a 19° turn towards the West. The house and its two glass façades may be seen in Figure 1:

Figure 1 *–The South façade (left) and the North façade (right)*

The U-values of the different construction parts are shown in Table 1:

Table 1 – *U-values for different types of constructions*

Glazing façades	1.04 W/m^2K
Roof	0.090 W/m^2K
Walls	0.089 W/m^2K
Floor	0.088 W/m^2K

The HVAC system of the house consisted of a reversible air-to-brine heat pump, water-based radiant heating and cooling system and an air handling unit that is coupled with the domestic hot water tank via a reversible air-to-water (or water-to-air, depending on the operation mode) heat pump.

There was a flat plate heat exchanger placed in between the hydronic loops of the house and the heat pump in order to avoid frost damage during the winter months.

The main sensible heating and cooling strategy of the house relied on the low temperature heating and high temperature cooling principle via the hydronic radiant system. There were pipes embedded in the floor and in the ceiling structure. The embedded pipes in the ceiling were designed to be used for cooling purposes only while the embedded pipes in the floor could be used for heating as well as cooling during the peak loads. Four loops were located in the floor and six loops in the ceiling. A mixing station was installed in the system, in order to control the flow to the individual loops, the flow rate, and the supply temperature to the embedded pipes.

The indoor air quality (ventilation) was regulated by an air handling unit. Passive and active heat recovery options were available. The passive heat recovery was obtained via a cross-flow heat exchanger while the active heat recovery was obtained via a reversible air-to-water heat pump cycle that was coupled to the domestic hot water tank. The air handling unit could supply fresh air at a flow rate of 320 m^3/h at 100 Pa. The design ventilation rate was determined to be 0.5 ach. Humidification of the supply air was not possible due to the limitations of the air handling unit.

The infiltration was determined to a value of 5.3 l/s pr. m^2 at an induced pressure of 50 Pa. The test was conducted according to the methods described in EN 13829 (2001).

3 Physical measurements and the simulation model of the house

During the measurement period, the air and globe temperatures were measured at different heights (from 0.1 m to 3.7 m). The measurements of the globe temperatures at a height of 1.1 m (operative temperature) were used for the comparison of the measurements with the simulation results. The measurement location and equipment may be seen in Figure 2:

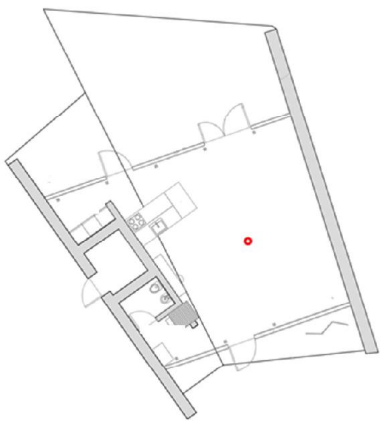


***Figure 2** – The location of the measurements(left) and the measurement equipment (right)*

The geometry of Fold was modeled in Google SketchUp to construct the advanced shapes of the house. The geometry was then imported into IDA ICE where windows, constructions and other parameters were specified. The Early Stage Building Optimization (ESBO) tool in IDA ICE was used as a base for the implementation of the HVAC system. The ESBO HVAC system was then modified in order to represent the real system. The HVAC system in the simulation model was constructed as close as possible to the real system, however an exact copy was not possible to obtain because some of the components from the real system were not available in IDA ICE.

A weather file was constructed with experimental weather data located 25 km away in a straight line from the house to eliminate as many sources of errors as possible. The weather file consisted of six parameters: dry-bulb temperature, relative humidity, wind direction, wind speed, direct normal radiation and diffuse radiation on a horizontal surface. The first four parameters were obtained directly from the datasheets of the weather station. The solar radiation parameters were calculated from the global solar radiation in the datasheet according to the method described in Kragh et al. (2002).

4 Results

The results from the measurements, simulations, comparisons and the results of the improvement suggestions are presented.

4.1 Temperature

Figure 3 shows the operative temperature from the measurements and from the simulation results.

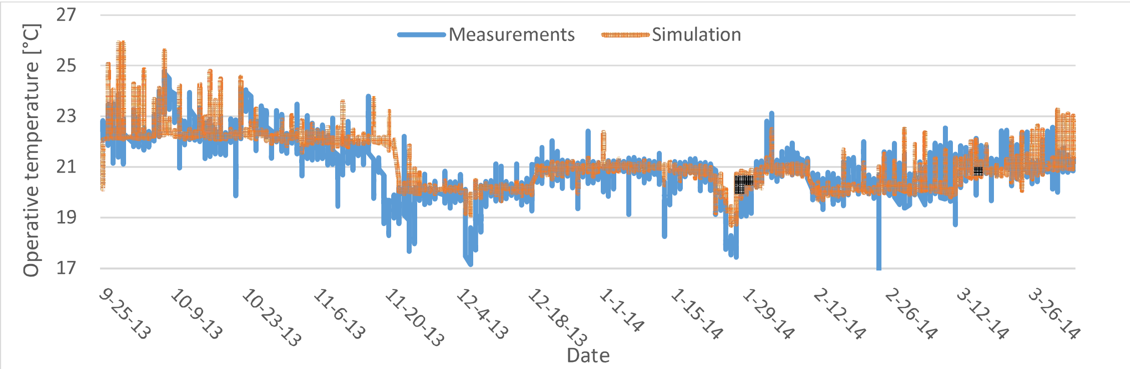

***Figure 3 -** Comparison of operative temperature at 1.1 m*

It may be seen in Figure 3 that the temperature predicted by the simulations has a tendency to be higher than the measurement in the house. The relative difference between measurements and simulation results was determined to be 2.7 % during the measurement period. It was observed that the difference was higher when there was direct sunlight present. When this contribution was higher than 100 W/m^2 the relative difference between the two temperatures increased to a value of 4.6 %.

4.2 Energy consumption

A comparison of energy consumption for the reversible heat pump and the ventilation system was conducted for the simulation and measurements. In order to obtain the energy consumption from the simulations, the power in the given time step was multiplied with the length of the time step. Furthermore, the power output from the ventilation system had to be re-calculated using Coefficient of Performance (COP) values given by the manufacturer.

The result of these calculations showed that the energy consumption in the simulations was 13.7 % lower for the heat pump and 23 % lower for the ventilation system compared to the actual energy consumption.

4.3 Improvements

Based on the measurements in the house and the simulation model, several improvement options were investigated. In order to compare the different scenarios, the thermal indoor environment is evaluated according to the operative temperature intervals given in EN 15251 (2007). The energy consumption is also taken into account when comparing the different improvements.

Previous studies have shown that a major concern regarding Fold is the high heating demand, Kazanci et al. (2014), and since the current study uses the data obtained from the heating season, the focus of the improvements had been to minimize the heating demand. Nevertheless, improvements regarding the cooling season were also investigated.

The improvements with the biggest impact regarding both thermal indoor environment and energy consumption were reducing the window area, reducing the infiltration and minimizing the thermal bridges. The results of the investigations on these three improvements are presented in Table 2, Table 3 and Table 5.

In the tables below the reference case refers to the current state of the house. The values for the reference case are obtained from the IDA ICE model.

***Table 2** – Results of reduced window area*

Window area reduction		Reference	5%	10%	15%	20%	25%	38%	58%
Indoor environment category	I	71%	74%	76%	79%	81%	84%	89%	94%
	II	27%	24%	22%	20%	17%	16%	11%	5%
	III	2%	2%	2%	1%	1%	1%	1%	0%
	IV	0%	0%	0%	0%	0%	0%	0%	0%
Energy consumption	kWh/ year	6371	6151	5943	5729	5519	5311	4742	4125

In Table 3 the values for the infiltration are given at an induced pressure of 50 Pa.

***Table 3** - Results of reduced infiltration*

Infiltration (l/s pr. m^2)		Reference (5.3)	1.5	1.0	0.5
Indoor environment category	I	71%	80%	81%	81%
	II	27%	19%	18%	18%
	III	2%	1%	1%	1%
	IV	0%	0%	0%	1%
Energy consumption	kWh/ year	6371	5552	5438	5323

In IDA ICE it is possible to define how much a building part is exposed to thermal bridges, an example is shown in Table 4. For the simulation of the effect of thermal bridges, every building part was changed corresponding to the investigated class.

***Table 4** – Example of thermal bridges in IDA ICE for external windows parameter*

Very Poor	Poor	Typical	Good	None
0,40	0,06 W/m^{2}K	0,03 W/m^{2}K	0,02 W/m^{2}K	0,00 W/m^{2}K

***Table 5** - Results of thermal bridges*

Thermal bridges		Very Poor	Poor	Typical	Good	None
Indoor environment category	I	42%	68%	79%	80%	83%
	II	29%	29%	19%	18%	16%
	III	21%	3%	1%	1%	0%
	IV	8%	1%	0%	0%	1%
Energy consumption	kWh/ year	9993	6626	5520	5359	4982

When the window areas were reduced, the amount of daylight in the house was also investigated in order to assure that the daylight levels inside the house abide the regulations of 200 lux from the Danish Building Regulations, Energistyrelsen (2014). The investigation showed that the reference case abide the regulations 93 % of the time. The case with a smallest window area abide the regulations 82 % of the time.

In Table 6 the results for different orientations of the house are shown.

***Table 6** - Results of different orientations*

Orientation		Reference	North West	West	South West	South	South East	East	North East
Indoor environment category	I	71%	69%	64%	66%	69%	69%	67%	69%
	II	27%	25%	26%	23%	22%	21%	24%	28%
	III	2%	5%	6%	6%	5%	5%	7%	3%
	IV	0%	2%	4%	4%	5%	5%	1%	0%
Energy consumption	kWh/ year	6371	7184	7870	7579	7123	7290	7203	6626

Different alternatives have also been investigated as an addition to the previously mentioned parameters:

- Automatically controlled exterior solar shading was implemented. In the current state there is no solar shading except the overhangs.
- Natural ventilation was simulated. Natural ventilation was provided from 10 % of the window area in the glazing façades that could open and it was controlled based on the temperature set-points in the house.
- Thermal mass was simulated by adding 0.1 m concrete in the walls.
- An increased embedded pipe system (EPS) area was simulated with EPS installed in the walls

The respective results of these variations are presented in Table 7.

***Table 7** - Results of other improvements*

Other parameters		Reference	Exterior solar shading	Natural ventilation	Thermal mass	Increased EPS area
Indoor environment category	I	71%	74%	74%	72%	88%
	II	27%	23%	23%	26%	11%
	III	2%	3%	3%	3%	1%
	IV	0%	0%	0%	0%	0%
Energy consumption	kWh/ year	6371	6104	6053	6192	6467

Regarding the orientation, the current orientation is the best solution both regarding thermal indoor environment and overall energy consumption. Regarding the thermal indoor environment the increased EPS area, the exterior solar shading and the natural ventilation had the biggest impact. The last two primarily made an impact in the cooling season by counteracting the cooling loads due to solar radiation.

4.4 Optimized Fold

An optimized version of the house is proposed to make it more energy efficient and improve the indoor environment. In the optimized house, the area of the glazing façades were reduced by 25 % and the U-value for the glass was lowered to $0.5\frac{W}{m^2 \cdot K}$. The infiltration was set according to the requirements for Danish houses in 2020, Energistyrelsen (2014), and the thermal bridges were reduced to make it more appropriate for a modern house. In the optimized house, there was natural ventilation when the outdoor temperature was lower than the indoor temperature to provide cooling during hot periods in the summer.

The results for thermal indoor environment and total energy consumption for the reference case and the optimized case are shown in Table 8.

Table 8 *– Results of the optimized house*

<table>
<tr><th colspan="2"></th><th>Reference</th><th>Optimized</th></tr>
<tr><td rowspan="4">Indoor environment category</td><td>I</td><td>71%</td><td>97%</td></tr>
<tr><td>II</td><td>27%</td><td>3%</td></tr>
<tr><td>III</td><td>2%</td><td>0%</td></tr>
<tr><td>IV</td><td>0%</td><td>0%</td></tr>
<tr><td>Energy consumption</td><td>kWh/ year</td><td>6371</td><td>2023</td></tr>
</table>

The optimized model significantly improved the thermal indoor environment and there was no duration when the operative temperatures were worse than the specification given in category II of EN 15251. Furthermore, the temperature never exceeded 26°C indicating that there was no overheating. The energy consumption was reduced by 68 % when the improvements were implemented.

5 Discussion

Despite being designed and constructed as a competition house, the house could still be improved in different ways. This study investigated some of the parameters that could be improved.

Various factors could have caused the discrepancies between the measurements and the simulation results. The house had been transported and it had been assembled (three times) and disassembled (two times). Hence it is likely that the infiltration and thermal bridges changed from the original values. In addition, the house had been stored in containers for several months, which could have lowered the performance of the building envelope.

By building a new house, the infiltration and the thermal bridges will be reduced. The cost of these improvements is estimated to be minimal. Most of the proposed improvements in this study are changes that would have to be made in the design-stage of the house. For example it would be troublesome to change the overall design of the glazing façades after the construction has been completed hence the practicality of the different improvements should be considered. It would however be possible to change some of the windows with insulated wall segments.

The differences between the actual HVAC system and the modelled HVAC system could also be a source of discrepancy in the results.

6 Conclusion

This study used measurements to validate simulation results. It was found that the average relative difference in operative temperature between simulations and measurements were 3 %, and the relative difference in energy consumption for the heat pump was 14 % and 23 % for the ventilation system. The simulations were used to investigate the effects of several parameters such as window area reduction, infiltration, thermal bridges, orientation, exterior solar shading, natural ventilation, thermal mass and increased embedded pipe system area on energy consumption and thermal indoor environment.

The simulations showed that the most important improvement was to reduce the heating demand of the building therefore the most effective improvements were reduction of window area, infiltration and thermal bridges. An improved house design based on the simulations is obtained. The improved house performed better in both thermal indoor environment and energy consumption than the reference case. The duration in indoor environment category I of EN 15251 was increased from 71 % to 97 % and the yearly energy consumption was reduced by 68 % compared to the reference building.

7 Acknowledgements

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Beyond nZEB: Experimental investigation of the thermal indoor environment and energy performance of a single-family house designed for plus-energy targets

A detached, one-story, single family house in Denmark was operated with different heating and cooling strategies for one year. The strategies compared during the heating season were floor heating without ventilation, floor heating supplemented by warm air heating (ventilation system), and floor heating with heat recovery from exhaust air. During the cooling season, the house was cooled by floor cooling and was ventilated mechanically.

Air and globe (operative, when applicable) temperatures at different heights at a central location were recorded. The thermal indoor environment, local thermal discomfort and overheating were evaluated based on EN 15251:2007, EN ISO 7730:2005 and DS 469:2013, respectively. Energy performance was evaluated based on the energy production and HVAC system energy use.

The thermal indoor environment during the heating season was satisfactory but it was not possible to reach the intended operative temperature when the outside temperatures were very low. During the cooling season, the cooling demand was high and overheating was a problem.

Although the house was designed as a plus-energy house, it did not perform as one under the Danish climate conditions. It would be possible to decrease the heating and cooling demand during the design phase through careful consideration of parameters such as the orientation, glazing area, solar shading, and thermal mass. With a lower demand, plus-energy levels can be achieved even with the minimum contribution from the energy producing components.

Introduction

Due to the depletion of fossil fuels and due to the remarkable global effects of greenhouse gas emissions, energy efficiency measures are being implemented in a variety of sectors that use significant amounts of energy. The buildings sector is one of these and a broad range of research activities are being carried out to find ways to decrease the energy consumption of buildings. The main driver behind these efforts is the massive energy requirements of buildings; buildings are responsible for 40% of the energy consumption in the member states of the European Union (European Commission 2010).

Building energy codes are becoming tighter and nearly zero-energy building (nZEB) levels are dictated for new buildings by 2020 in the European Union (European Commission 2010). A further goal is to design plus-energy houses, i.e. houses that produce more energy from renewable energy resources than they import from external resources in a given year, according to the definition given by the European Commission (2009). These trends are reflected in different initiatives such as the Passive House movement (Passivhaus Institut 2015; the International Passive House Association 2015) and recently in the Active House Alliance (2015). The idea of plus-energy houses is also being promoted with competitions such as Solar Decathlon (2012), where multidisciplinary teams from universities compete to design, build and operate plus-energy houses. Plus-energy houses could have a significant role in the energy system in a number of ways: they can compensate for the old buildings that are too expensive to upgrade to nZEB levels, and they can act as small power plants in the energy system.

When striving for energy efficiency in buildings, it should not be overlooked that people spend most of their time indoors (Olesen and Seelen 1993). Buildings are built for people to live in and to have a comfortable, healthy and productive indoor environment, not to save energy. Thus, energy savings should not be achieved at the cost of occupant thermal discomfort. Instead, both of these goals should be achieved simultaneously.

The international focus on the residential sector is increasing (ASHRAE 2014) and there have been a number of surveys that monitored energy performance and indoor environment in passive houses (Schnieders and Hermelink 2006; Larsen et al. 2012; Brunsgaard et al. 2012). One commonly encountered problem in low-energy and passive houses is overheating (too high temperatures). Overheating has been reported from different countries such as Denmark by Larsen and Jensen (2011), from Sweden by Janson (2010) and Rohdin et al. (2014), and from Finland by Holopainen et al. (2015). Maivel et al. (2015) also reported overheating in new apartment buildings in Estonia. Some of the main reasons of overheating are large glazing areas, poor or lack of solar shading, lack of ventilation (Larsen 2011), lack of thermal mass, and lack of adequate modeling tools in the design phase (Phillips and Levin 2015).

In addition to overheating, varying room temperatures (Rohdin et al. 2014; Holopainen et al. 2015), too low air temperatures in winter, stuffiness and poor air quality, and too low floor surface temperatures in winter (Rohdin et al. 2014) are some of the other problems that were encountered in low-energy and passive houses. A discussion of research needs regarding the indoor environmental quality in low-energy houses can be found in Phillips and Levin (2015).

There are still unsolved issues and problems to be addressed regarding low-energy, passive or plus-energy houses and designers and engineers can therefore benefit from research on the design, construction and operation of such buildings. In this context, a detached, one-story, single family house, which was initially designed to be a plus-energy house (Kazanci et al. 2014), was operated from 26/09/2013 to 1/10/2014 to compare a number of different heating and cooling strategies (Kazanci and Olesen 2014a; Kazanci and Olesen 2015a). The thermal indoor environment and energy performance of the house were monitored during this period.

The house was designed for, and participated in the competition Solar Decathlon Europe 2012 in Spain, and since then it has been used as a full-scale experimental facility. During the experiments reported here it was unoccupied and the internal heat gains were simulated by means of heated dummies.

The performance of the different heating and cooling strategies was evaluated in terms of the resulting thermal indoor environment, local thermal discomfort (vertical air temperature difference between head and ankles) and overheating, according to EN 15251 (European Committee for Standardization 2007), EN ISO 7730 (European Committee for Standardization 2005) and DS 469 (Danish Standards 2013), respectively. The energy production and the energy consumption of the heating, cooling and ventilation systems of the house were used to evaluate its energy performance.

Details of the house

Construction details

The test house was a single family, detached, one-story house with a floor area of 66.2 m^2 and a conditioned volume of 213 m^3. The house was constructed from pre-fabricated wooden elements that were made from layers of laminated veneer lumber boards, which in combination with I beams in between formed the structural elements. The house was insulated with a combination of 200 mm mineral wool (between the boards of the structural elements) and 80 mm compressed stone wool fibers (40 mm on each side, outside the boards of the structural elements). A drawing of the structural element is given in Kazanci et al. (2014). The walls, roof and floor structures were formed by installing prefabricated elements in a sequential order and the joints were sealed. The North and South glazing façades were inserted later and the joints between the glazing frame and the house structure were sealed. The house was supported on 200-300 mm concrete blocks and the space between the ground and the house's floor structure was covered which created a crawl-space below the house.

Inside the house, there was a single space with a high and inclined ceiling, which combined kitchen, living room and bedroom. The technical room was completely insulated from the main indoor space, and had a separate entrance. The wall between the technical room and the indoor space was insulated with the same level of insulation as the envelope. The glazing façades were partly shaded by the roof overhangs. No solar shading was installed in the house except for the skylight window. All windows had a solar transmission of 0.3. The largest glazing façade was oriented to the North with a 19° turn towards the West. Figure 1 shows the exterior views of the house.

FIGURE 1

The surface areas and thermal properties of the envelope are given in Table 1.

TABLE 1

Details of the heating, cooling, and ventilation system

The sensible heating and cooling of the house relied on the low temperature heating and high temperature cooling principle via the hydronic radiant system in the floor. The system was a dry radiant system, consisting of a piping grid installed in the wooden layer. The details of the floor system were: chipboard elements with aluminum heat conducting profiles (thickness 0.3 mm and length 0.17 m), PE-X pipe, 17x2.0 mm. Pipe spacing was 0.2 m. A wooden floor covering was used with a thickness of 14 mm and a thermal conductivity of 0.13 W/mK. The available floor area for the embedded pipe system installation was 45 m^2, which is 68% of the total floor area. The design flow rates in the heating and cooling modes were 619 kg/h and 336 kg/h, respectively. The flow rates were calculated according to EN 15377-2 (European Committee for Standardization 2008). Figure 2 shows the details of the floor heating and cooling system used in the house.

FIGURE 2

The heat source and sink of the house for space heating and cooling was outdoor air, using a reversible air-to-brine heat pump. The minimum and maximum cooling capacities and the nominal power input in the cooling mode were 4.01, 7.1, and 2.95 kW, respectively. The minimum and maximum heating capacities and the nominal power input in the heating mode were 4.09, 7.75, and 2.83 kW, respectively.

A flat-plate heat exchanger was installed between the hydronic radiant system of the house and the air-to-brine heat pump. The pipes between the heat exchanger and the heat pump were filled with an anti-freeze mixture (40% ethylene glycol) to avoid frost damage during winter.

A mixing station, which linked the radiant floor heating and cooling system with the heat source and sink, and a controller adjusted the flow and supply temperature to the floor loops. The radiant system was controlled based on the operative temperature set-point that was inserted on a room thermostat (a matt gray half-sphere) in 0.5 K intervals and on the relative humidity inside the house to avoid condensation during summer.

The house was ventilated mechanically by an air handling unit (AHU). The mechanical ventilation was only used to provide fresh air into the house since the main sensible heating and cooling terminal of the house was the radiant system. This also made it possible to have lower airflow rates compared to a system where space heating and cooling is mainly obtained by an air system (Olesen 2012). The design ventilation rate was 0.5 ach (the Danish Ministry of Economic and Business Affairs 2010). The intake air was taken from the crawl-space.

Passive and active heat recovery options were available in the AHU. The passive heat recovery was obtained by means of a cross-flow heat exchanger and this passive heat recovery system had an efficiency of 85% (sensible heat). By-pass was possible depending on the intake air temperature. The active heat recovery was achieved by means of a reversible air-to-water heat pump that was coupled to the domestic hot water tank. The AHU could supply fresh air at a flow rate of up to 320 m^3/h at 100 Pa. Humidification of the supply air was not possible due to the limitations of the AHU. The two air supply diffusers can be seen on the technical room wall in Figure 3.

FIGURE 3

The house was designed as a plus-energy house and the roof of the house was covered with photovoltaic/thermal (PV/T) panels. PV/T panels convert the incoming solar radiation into electricity (PV) and thermal energy (T). The electricity was generated by mono-crystalline cells and the total cell area was 50.8 m^2. The house was connected to the grid and there was no storage of electrical energy. The installed nominal power was cut down to 9.2 kWp by two inverters. The thermal part of the PV/T panels was not operational until the summer of 2014 and only limited data are available from this period; therefore they are not further described in this study. Detailed information regarding the performance of the thermal part and its effects on the electricity production of the PV cells were reported by Kazanci et al. (2014).

Further details of the components and the system can be found in Kazanci et al. (2014), Skrupskelis and Kazanci (2012), Kazanci and Olesen (2014b).

Materials and methods

The house was located in Bjerringbro, Denmark and it was used as a full-scale experimental facility where the thermal indoor environment and energy performance of the house were monitored for a full year, from 26/9/2013 to 1/10/2014.

Experimental set-up

The house was unoccupied during the measurement period but the occupancy and equipment schedules (internal heat gains) were simulated by means of heated dummies. Each dummy was a circular aluminum duct, with a diameter of 220 mm and with a height of 1 m. It had closed ends and an electrical heating element (wire) was installed on the internal surfaces of the duct, with an adjustable heat output up to 180 W (Skrupskelis and Kazanci 2012).

The occupancy and equipment schedules were adjusted with timers. Two dummies were used to simulate occupants (the dummies had the same surface temperatures as a person would have) at 1.2 met (ON from 17:00 to 08:00 on weekdays and from 17:00 to 12:00 on weekends), one dummy (equipment #1, 120 W, 1.8 W/m^2) was always ON to simulate the house appliances that are always in operation, the fourth dummy (equipment #2, 180 W, 2.7 W/m^2) was used to simulate the house appliances that are in use only when the occupants are present and the fifth dummy was used to represent additional lights (180 W, 2.7 W/m^2, ON from 06:00 to 08:00 and from 17:00 to 23:00 until 27th of May 2014, and after this date, ON from 20:00 to 23:00 every day). The house had ceiling mounted lights ON from 21:00 to 23:00 every day (140 W, 2.1 W/m^2). Additionally, there was a data logger and a computer (80 W, 1.2 W/m^2), and a fridge (30 W, 0.4 W/m^2) which were always ON.

Measurements and measuring equipment

A number of physical parameters, energy consumption and production were measured and recorded. The temperatures (air and globe) were measured at heights of 0.1 m, 0.6 m, 1.1 m, 1.7 m, 2.2 m, 2.7 m, 3.2 m and 3.7 m at a central location in the occupied zone following EN 13779 (European Committee for Standardization 2007). The measurements above the occupied zone were taken to evaluate the effects of thermal stratification based on the different heating and cooling strategies. The stratification is particularly important for this house because of its high and inclined ceiling. The thermal stratification from the floor to the ceiling was used as an indicator of the performance of the heating strategy regarding heat loss from the conditioned space to the outdoors.

Globe temperatures were measured with a gray globe sensor, 40 mm in diameter. This sensor has the same relative influence of air- and mean radiant temperature as on a person (Simone et al. 2007) and, thus, at 0.6 m and 1.1 m heights will represent the operative temperature of a sedentary or a standing person, respectively. The air temperature sensor was shielded by a metal cylinder to avoid heat exchange by radiation. Both the globe and air temperature sensors have ±0.3°C accuracy in the measurement range of 10-40°C (Simone et al. 2013). The output from the sensors was logged by a portable data logger.

A panoramic view of the interior of the house, the measurement location and the sensors used for the measurements may be seen in Figure 3.

The energy consumptions of the air-to-brine heat pump, mixing station, and the controller of the radiant system were measured with wattmeters. The energy consumption of the AHU and energy production of the house (from the PV/T panels) were measured through a branch circuit power meter (BCPM). The wattmeters that were used to measure the consumption of the mixing station and the controller of the radiant system had an accuracy of ±2% ±2 W. The wattmeter that was used to measure the consumption of the air-to-brine heat pump had an accuracy of 3%. The BCPM's accuracy was 3% of the reading.

A full specification of the parameters measured and the measuring equipment can be found in Kazanci and Olesen (2014b).

Experimental settings

Heating season

Different heating strategies were compared during the heating season. In the beginning of the heating season, floor heating was controlled according to different operative temperature set-points (without any ventilation). Afterwards, floor heating was supplemented by warm air heating from the ventilation system, and during the last part of the heating season, the ventilation system was only used to provide fresh air (with passive heat recovery from the exhaust air) while floor heating was providing the required space heating. The building code in Denmark requires that each habitable room and the dwelling as a whole must have a fresh air supply and individual room temperature control (the Danish Ministry of Economic and Business Affairs 2010). The design ventilation rate was 0.5 ach.

The most important boundary conditions for these strategies in the heating season are given in Table 2 (FH: floor heating, HR: heat recovery, HRPH: heat recovery and pre-heating, corresponding to warm air heating). The numbers in the abbreviations are related to the indoor temperature set-points.

TABLE 2

Cooling season

The HVAC system was operated similarly during the cooling season. The house was cooled by floor cooling and was ventilated with the mechanical ventilation system (with passive heat recovery from the exhaust air). Different operative temperature set-points and different ventilation rates were tested. Internal solar shading covering 20 m^2 (manually operated) was installed on the North façade on 30/07/2014 and it was used in the fully down position until the end of the experiments.

The house was not cooled from 20/06/2014 to 23/06/2014 (the floor cooling and the AHU were OFF), to allow repairs to be made to the HVAC system.

The most important boundary conditions for the strategies used in the cooling season are given in Table 3 (FH: floor heating, CS: cooling season, FC: floor cooling, HV: higher ventilation rate, S: solar shading). The numbers in the abbreviations are related to the indoor temperature set-points.

TABLE 3

Results and discussion

Heating season

The performance of different heating strategies was evaluated based on the indoor environment category achieved according to EN 15251 (European Committee for Standardization 2007), and the measured temperature stratification. The categories are given according to EN 15251 (European Committee for Standardization 2007) for sedentary activity (1.2 met) and clothing of 1.0 clo. The indoor environment categories achieved with different heating strategies are given in Table 4.

TABLE 4

Figure 4 shows the operative temperature at 0.6 m height and the external air temperature during the heating season.

FIGURE 4

It may be seen from Table 4 and Figure 4 that even though different heating strategies were used, the overall performance regarding the indoor environment was satisfactory, i.e. 80% of the time in Category 2 according to EN 15251 (European Committee for Standardization 2007). It may also be seen that there were periods when the indoor environment was outside Category 3: for 2% of the time it was in Category 4.

The results show that it was possible to keep the indoor operative temperature close to the set-point, although the systems struggled to achieve this when the outside temperatures were below -5°C (for 2% of the time only Category 4 was achieved). In addition to the increased heating demand, one possible explanation for this is that both the air-to-brine heat pump and the AHU were affected by the lower outside air temperatures.

The operative temperature set-point of 20°C was too low. This is because even though the ventilation system would be heating the indoor space, the floor heating system did not start the water circulation in the loops until the operative temperature had dropped below 20°C. This resulted in several periods with room temperatures below 20°C.

Vertical air temperature difference between head and ankle levels (for sedentary occupants, 1.1 m and 0.1 m above the floor, respectively) at the measurement location was evaluated according to EN ISO 7730 (European Committee for Standardization 2005), as an indicator of local thermal discomfort. The average temperature differences as a function of the heating strategy are given in Table 5 and the temperature difference during the entire heating season may be seen in Figure 5.

TABLE 5

FIGURE 5

It may be seen from Table 5 and Figure 5 that the vertical air temperature difference was the highest for the cases where the floor heating was supplemented by warm air heating from the ventilation system. For each heating strategy and for the overall heating season, the average temperature difference was less than 2 K indicating that the requirements of Category A were met according to EN ISO 7730 (European Committee for Standardization 2005) at the measurement location.

The thermal stratification is an inevitable physical phenomenon and it can be used to analyze the indoor environment created by different heating strategies. The thermal stratification is important for occupant thermal comfort (due to local thermal discomfort) and for heat loss from the building. A high temperature gradient will increase the energy consumption and local thermal discomfort (Müller et al. 2013). In Table 6, average air temperatures at selected heights are given based on the heating strategy.

TABLE 6

Figure 6 shows the air temperature differences between the selected heights for the heating season.

FIGURE 6

The results shown in Table 6 and Figure 6 indicate that the thermal stratification inside the house was greatest when the floor heating was supplemented by warm air heating from the ventilation system. On average, the temperature difference between the highest (3.7 m) and lowest (0.1 m) measurement points was 2.8 K when the floor heating was supplemented by warm air heating while in other cases it was between 0.7 K and 0.9 K. Figure 6 shows a clear increase in the temperature difference (thermal stratification) between the highest and lowest points when the floor heating was supplemented by warm air heating, and the temperature difference reached almost 4 K in some periods.

Because of the lower density of the warm supply air compared to the room air, the supply air tends to flow along the ceiling and not to mix well with the room air. Due to this phenomenon and the thermal stratification, in the cases where the floor heating was supplemented by warm air heating, the space above the occupied zone was being heated. This increases the heat loss from the indoor space and especially where there are glass façades with higher U-values compared to the external walls. Increased thermal stratification is a phenomenon to avoid (unless an underfloor air distribution or a displacement ventilation system is used); especially in a house with a high and tilted ceiling, as in this test house, so it is recommended to use a radiant floor heating system in spaces with high ceilings.

Cooling season

The performance of different cooling strategies was evaluated based on the indoor environment categories given in EN 15251 (European Committee for Standardization 2007) for sedentary activity (1.2 met) and clothing of 0.5 clo. In addition, the hours above 26°C and 27°C were calculated following DS 469 (Danish Standards 2013). According to DS 469 (Danish Standards 2013), 26°C should not be exceeded for longer than 100 hours during the occupied period and 27°C should not be exceeded for longer than 25 hours. Even though these specifications are given for offices, meeting rooms, and shops, it is considered to be applicable also for residential buildings, and according to DS 469 (Danish Standards 2013), mechanical cooling would normally not be installed in residential buildings in Denmark.

The indoor environment categories achieved, and the hours above 26°C and 27°C as a function of the cooling strategy are given in Table 7, and the operative temperature and external air temperature during the cooling season are given in Figure 7.

TABLE 7

FIGURE 7

The house performed worse in the cooling season than in the heating season in terms of providing a satisfactory thermal indoor environment; for 57% of the time the operative temperature was in Category 2 and for 19% of the time it was outside the recommended categories in EN 15251 (European Committee for Standardization 2007). This occurred mainly in the transition periods (i.e. May and September) and due to overheating, which was a problem during the cooling season, except in August and September. The hours above 26°C and 27°C exceeded the values recommended in DS 469 (Danish Standards 2013). Decreasing the operative temperature set-point and increasing the ventilation rate helped to address the increased cooling load, but with a higher energy consumption. This is mainly due to the longer operation of the floor cooling and to increased cooling effect of the supply air.

The results show that even though the floor system was in heating mode during most of May, which is a part of the transition period, floor cooling could have been activated in the second half of May, which would have reduced the hours above 26°C and 27°C, and improved the indoor environment.

The main reasons of overheating were the large glazing façades including the lack of solar shading, the orientation of the house and the lack of thermal mass to buffer sudden thermal loads. In the current location of the house, direct solar radiation from the South façade was not a problem, because of the orientation and longer overhang on the South façade. Most of the overheating hours were in the late afternoon (i.e. from 18:00 until sunset), when there was direct solar gain through the North façade.

The vertical air temperature difference between head and ankles was evaluated according to EN ISO 7730 (European Committee for Standardization 2005) as an indicator of local thermal discomfort. The average temperature differences as a function of the cooling strategy are shown in Table 8 and the temperature difference during the cooling season may be seen in Figure 8.

TABLE 8

FIGURE 8

For each cooling strategy and on average, the vertical air temperature difference was lower than 2 K indicating that the requirement for Category A was met at the measurement location, according to EN ISO 7730 (European Committee for Standardization 2005). The high values of fluctuation may be attributed to direct solar radiation on the sensor at 1.1 m height.

Thermal stratification at the measurement location was also evaluated. The average temperature at chosen heights as a function of the cooling strategy and the temperature difference between the lowest and highest measurements points are given in Table 9. The air temperature difference between the selected heights over the cooling season is shown in Figure 9.

TABLE 9

FIGURE 9

It may be seen from Table 9 and Figure 9 that there was a natural pattern of thermal stratification and average values were slightly higher compared to the values obtained in the heating season (except for floor heating supplemented by warm air heating). This effect could be explained by the floor cooling. The sudden increases in the temperature difference could be due to direct solar radiation on the sensors.

In this study, operative temperature was used as an indicator of the thermal indoor environment, and vertical air temperature difference between head and ankles was used as an indicator for local thermal discomfort, but human thermal comfort is also affected by other factors such as floor surface temperature, radiant temperature asymmetry and draught (European Committee for Standardization 2005). All of these factors would have to be considered before any definitive conclusion on occupant thermal comfort can be drawn.

Energy performance

The energy performance of the house was evaluated by considering the energy produced by the PV/T panels on the roof and the energy used by the HVAC system. The monthly electricity production from the PV/T panels is given in Table 10.

TABLE 10

The annual electricity production from the PV/T panels (4043.9 kWh) was lower than had been predicted by the simulations, 7434.3 kWh (Kazanci et al. 2014). This difference could have occurred for several reasons: it was observed during the competition period in Solar Decathlon Europe 2012 (17th-28th of September 2012) that the output of the PV/T panels was lower than the expected values (Kazanci and Olesen 2014b), the climate could have differed from the weather files used in simulations and the location was different. In addition to these factors, some of the PV/T panels were damaged during disassembly/assembly and transportation of the house, and finally, some trees around the house threw shadows on the PV/T panels.

For the HVAC system's energy consumption, the air-to-brine heat pump, mixing station, controller of the radiant system, and AHU were considered. During the heating season, the set-point of the heat pump was 35°C until 21/11/2013 and after this date, it was 40°C. The set-point of the heat pump was changed to 15°C on 27/05/2014. The energy consumption of the components for each strategy is given in Table 11.

Heating degree days (HDD) and cooling degree days (CDD) were calculated for each case using a base temperature of 17°C and 23°C, respectively. Figure 10 shows the total average energy consumption of the cases together with the calculated degree days following the methodology described by Quayle and Diaz (1980).

TABLE 11

FIGURE 10

In Table 11, the heat pump energy consumption includes the heat pump cycle consumption, an integrated pump and the heat pump control system. The mixing station's consumption includes the circulation pump of the radiant floor heating system, a motorized mixing valve and the control unit. The consumption of the controller of the radiant system includes a control unit and an actuator at the manifold for each of the four loops (to open or close the loops). The AHU's consumption includes fans (supply and exhaust), control system, by-pass damper, internal heat pump cycle (active heat recovery) and its related equipment.

The results show that the energy consumption increased markedly when the warm air heating (FH21-HRPH and FH20-HRPH) was in operation, and these strategies struggled to provide the intended thermal indoor environment despite the increased energy consumption. The energy consumption during the cases FH20 and FH21 were close to each other, but a better thermal indoor environment was achieved with FH21. The last two cases in the heating season, FH21-HR and FH20-HR, have lower energy consumption and achieved a better thermal indoor environment compared to the cases with the same set-points without ventilation (FH21 and FH20). The FH22 strategy had the lowest energy consumption (although it had the highest operative temperature set-point) and the most satisfactory thermal indoor environment, although this was partly due to the relatively high external air temperatures during this period.

During the cooling season, the increased ventilation rate and lowered operative temperature set-point increased the energy consumption. This was expected, due to higher power input to the fans in the AHU and longer operation time of the pump in the floor cooling system. The increased energy consumption contributes to a better thermal indoor environment, but other strategies should be employed to reduce the cooling demand by means of energy efficient measures (e.g. lower ventilation rates when the house is unoccupied, natural ventilation when the outside conditions are suitable, decreased glazing area, solar shading, a better orientation of the house and so forth). The effects of different building and HVAC system improvements on the energy consumption and thermal indoor environment were parametrically studied and reported in Andersen et al. (2014).

It would be possible to increase the energy performance of the house by means of simple modifications, e.g. 29% (1051 kWh, 15.9 kWh/m^2 conditioned floor area) of the heat pump's consumption was due to the brine pump that is integrated into the heat pump and it was always ON due to the internal control algorithm of the heat pump. This energy consumption can be decreased by synchronizing the mixing station pump and this pump (as there is no storage in between).

The overall energy performance of the house during the measurement period is summarized in Table 12. In the overall energy balance, positive values indicate energy surplus while negative values indicate energy deficit, i.e. the house used more energy than it produced on an annual basis.

TABLE 12

Operation and performance

The results from the measurement period show that the control of heating and cooling system through set-points and the interaction of the radiant floor heating and cooling system with the ventilation system have considerable effects on the thermal indoor environment and on the energy performance. The optimal system combination is when the floor heating and cooling system is emitting or removing the necessary heat and the ventilation system is only used to provide the necessary supply of fresh air without actively heating or cooling the intake air. This would simplify the system operation since only one system will heat or cool the indoor space, and lower airflow rates can be used since the only task of the ventilation system will be to provide fresh air.

The heating and cooling set-points should be carefully selected to avoid periods with too low or too high indoor temperatures. This requires choosing set-points so that the radiant floor heating and cooling system has enough time to provide the necessary indoor conditions. It should also be noted that a higher indoor temperature set-point in the heating season would result in a higher energy use because of the longer operation time of the heating system, and because of the increased operating temperatures in the heating system which affect the heat pump performance. A similar effect will be observed in the cooling season with a lower indoor temperature set-point.

The fresh air intake for the AHU was under the house and this proved to be a beneficial approach; it was observed that the temperature below the house was warmer than the external air temperature during winter and colder than the external air temperature during summer, so it buffered the variations in the external air temperature to a certain extent. Figure 11 shows the external and intake air temperatures.

FIGURE 11

Although this application was beneficial for this location, it may not always be appropriate (e.g. a radon barrier might be needed in locations where radon is a concern). A definitive conclusion on the benefits of this approach must consider the effects of heat loss to this space in comparison to heat loss to the ground.

Throughout the 12-month operation of the house, the heating and cooling systems were active with respective set-points also during the transition periods (i.e. May and September) but it is not practical to provide constant heating or cooling during the transition periods, therefore the heating and cooling system operation and the switchover between these modes require careful consideration. Operation of the systems needs to be improved to avoid unnecessary heating and cooling in the transition periods.

A recent study analyzed the horizontal temperature distribution in the present house and found that the radiant floor cooling created a uniform thermal indoor environment in the cooling season, despite the large glazing façades (Kazanci and Olesen 2015b).

Earlier studies with different simulation software (Skrupskelis and Kazanci 2012; Andersen et al. 2014) showed that the current orientation of the house was optimal in terms of thermal indoor environment and energy performance, but based on the present results, a reversed orientation, i.e. the façade with the longer overhangs towards the North, would have been more energy effective, as it would decrease the solar heat gain during the cooling season and increase the solar heat gain during the heating season.

Previous studies (Kazanci et al. 2014; Skrupskelis and Kazanci 2012; Andersen et al. 2014) showed that the large glazing façades (including the lack of solar shading) of the house resulted in a high heating and cooling demand and this drastically decreased the energy performance of the house. This was confirmed by the experiments; the currently installed heating and cooling systems of the house struggled to achieve a comfortable thermal indoor environment during the cold periods and overheating was a significant problem during the cooling season.

The results show that the house would have benefited from a higher thermal mass to buffer the sudden thermal loads, especially during the periods in cooling season when there was direct solar gain and during the transition periods. This confirms a previous simulation study (Andersen et al. 2014) which showed that the house would benefit from increased thermal mass, in terms of energy performance and thermal indoor environment.

During the heating season, when the outside temperatures were below -5°C, the heating system of the house struggled to reach the operative temperature set-points. An evident reason for this was the increased heating demand. The air-to-brine heat pump and the AHU were also affected by the lower external air temperatures. The initial design of the heating and cooling system incorporated a ground heat exchanger to obtain free cooling during summer and a coupled heat pump for the heating season, as the heat sink and source of the house (Kazanci and Olesen 2014b). This would have been beneficial, since the performance of a ground-coupled heat pump would not have been affected significantly by the varying external air temperatures. Also, it would have been beneficial during the cooling season by significantly decreasing the energy consumption for cooling (Skrupskelis and Kazanci 2012).

The house was initially designed as a plus-energy house and shown by previous simulation studies that it performs as one on a yearly basis (Kazanci et al. 2014), but the measurement results show that it did not produce any energy surplus under the climate conditions in Denmark (Table 12). This is mainly due to the lower electrical output from the PV/T panels (4044 kWh) than estimated in the design conditions (7434 kWh), high heating and cooling energy consumption (4613 kWh, 69.7 kWh/m^2 conditioned floor area), changes in the house's heating and cooling systems resulting in higher energy consumption than the design values, differences in the simulations and in the actual components of the house.

The results presented in this study are not influenced by the users since there were no users and the heat gains were controlled, therefore the results are purely related with the design and operation of the house and its systems.

During the experiments, it was not possible to make changes in the building envelope (except for the installation of the internal solar shading), so the modifications were on the heating, cooling and ventilation system operation. The design of the current building diverged from the ideal design of an energy efficient nZEB where the heating and cooling demands of the house would have been minimized in the design phase. The results of this study show that it is not enough and it is not energy efficient to address the heating and cooling loads through adjusting set-points and adjusting the flow rates of air and water.

Although the results reported in this study are for a particular house, the results regarding the systems and their operation have implications on a broader scale. In order to achieve high energy performance together with a comfortable thermal indoor environment, the initial step is to reduce the heating and cooling demands of the house as much as possible during the design and this should be done in a holistic way where orientation, shading, thermal mass, air-tightness, thermal bridges, etc. are considered simultaneously. This is crucial since the demand determines the final energy consumption. After it is assured that the heating and cooling demands are minimized, the resulting demand should be addressed with the most energy efficient and environmentally friendly heating and cooling strategies. The radiant low temperature heating and high temperature cooling systems enable the integration of natural heat sources and sinks (ground, night-time radiative cooling, solar, sea-water, etc.) into the heating and cooling systems and they also provide a draught-free and uniform thermal indoor environment, therefore they are a promising means of achieving high energy performance without sacrificing occupant thermal comfort.

Once it is ensured that the demand of the building (including plug loads) is minimized, the most energy efficient heating and cooling systems are chosen, and the control of these systems are optimized, then it would be possible to reach plus-energy targets even with a minimal contribution from the energy producing components in future buildings.

Conclusion and future research

A detached, one-story, single family house designed for plus-energy performance was operated for one year. During this period different heating and cooling strategies were compared and the energy performance of the house and its thermal indoor environment were monitored. The main conclusions are as follows.

- During the heating season, it was possible to provide the intended operative temperature inside the occupied zone except for the periods when the external air temperatures were below -5°C.
- Radiant floor heating combined with heat recovery from ventilation was the optimal heating strategy as it provided a uniform temperature distribution within the space and decreased the heat losses due to thermal stratification.
- The performance of the house in terms of maintaining a comfortable thermal indoor environment was worse in the cooling season than in the heating season. Overheating was a significant problem, and the main reasons for this were the large glazing façades, the orientation of the house, the lack of solar shading, and the lack of sufficient thermal mass to buffer the sudden thermal loads.
- The operation of the heating and cooling system during the transition periods was problematic and this affected the thermal indoor environment and energy performance negatively.
- The house had a high heating and cooling demand that could easily have been reduced at the design phase. Although, it might be possible to address the excessive heating and cooling loads by adjusting set-points, water and airflow rates, these would result in increased energy consumption,

as in the present study. It is crucial to minimize the demand before attempting to satisfy it in the most energy efficient way.

- Although the house was designed as a plus-energy house, it did not perform as one under the climatic conditions of Denmark. This was mainly due to the electrical output from the PV/T panels being lower than had been assumed in the simulations, but also to the unnecessarily high energy consumption of the house and the fact that the heating and cooling system differed from that of the initial design. It would be possible to increase the energy performance of the house by making simple modifications to the operating strategies and to the architectural design of the house. It would then be possible to achieve plus-energy levels with the same active (energy producing) components.

The present study resulted in possible future investigations including:

- Space heating and night radiative cooling possibilities with PV/T panels;
- Consideration of several plus-energy houses' role in the energy system;
- Optimization of the operation of heating and cooling system during transition periods. Can enough thermal mass let the house run without active heating or cooling during transition periods and how to evaluate the thermal comfort during these periods?
- Possibility of using phase change materials (PCM), either passively or actively, to increase the effective thermal mass of the building and its effects on the thermal indoor environment and energy performance;
- Benefits of using a predictive control algorithm to control the heating and cooling systems of the house, especially during the transition periods.

Although every building is different, there are certain general rules to be followed during the design and operation phases. The main findings of this study, together with these possible future investigations, could benefit the design and operation of future nZEB and plus-energy houses and could help to improve the thermal indoor conditions without increasing the energy use.

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Figures:

Figure 1. *Exterior views of the house, seen from North-West (left) and South-West (right)*

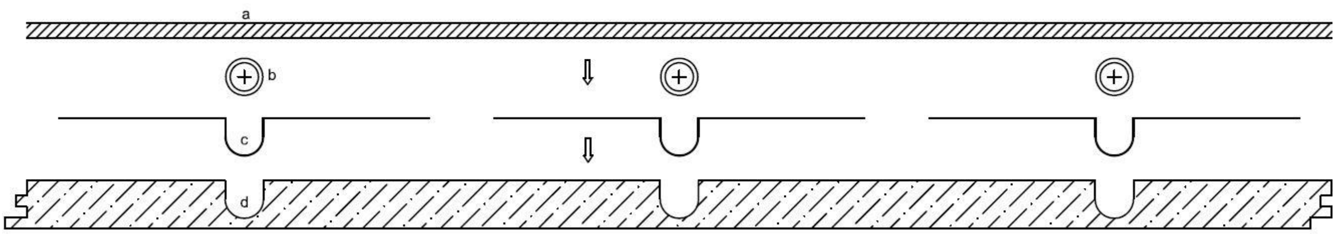

Figure 2. *a) Floor covering b) Pipe, 17x2.0 mm c) Heat distribution plate, 0.3 mm, d) Under-floor plate, 22 mm*

Figure 3. *Panoramic view of the interior (left), the measurement location (middle) and the globe and air temperature sensors (right)*

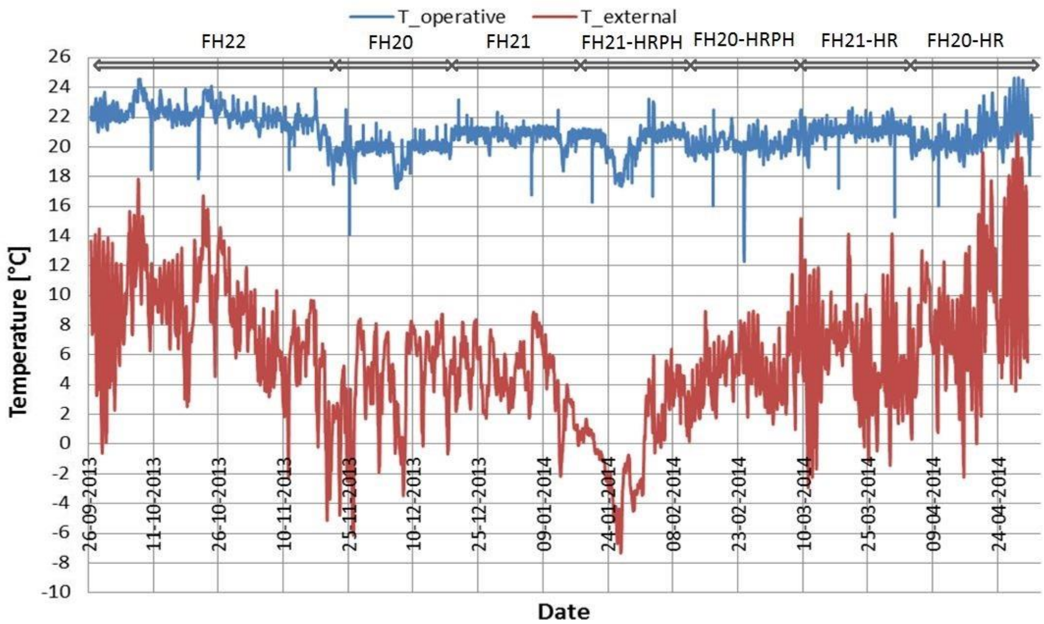

__Figure 4.__ Operative temperature and external air temperature during the heating season

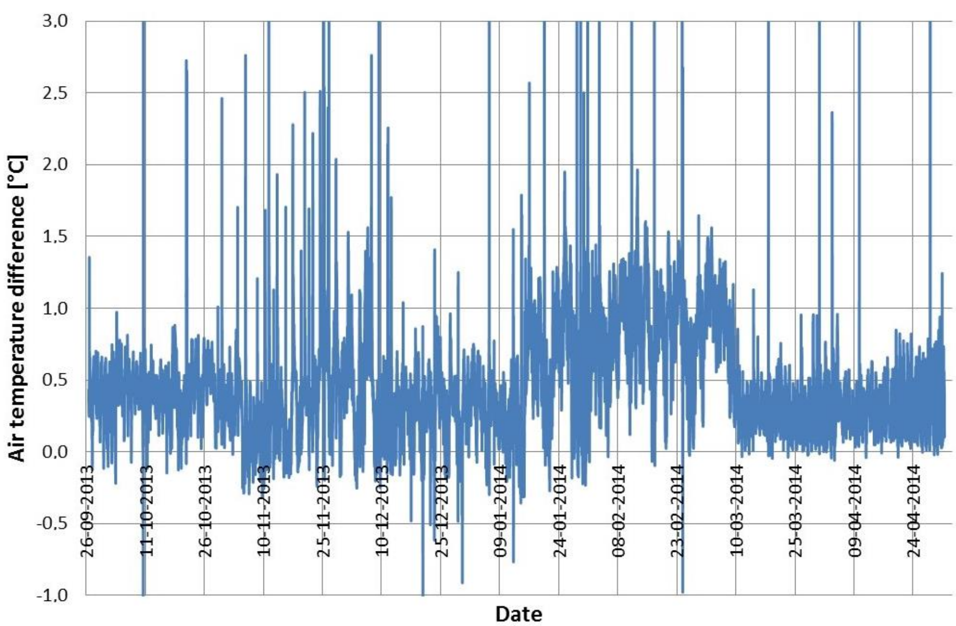

__Figure 5.__ Vertical air temperature difference between head and ankles during heating season

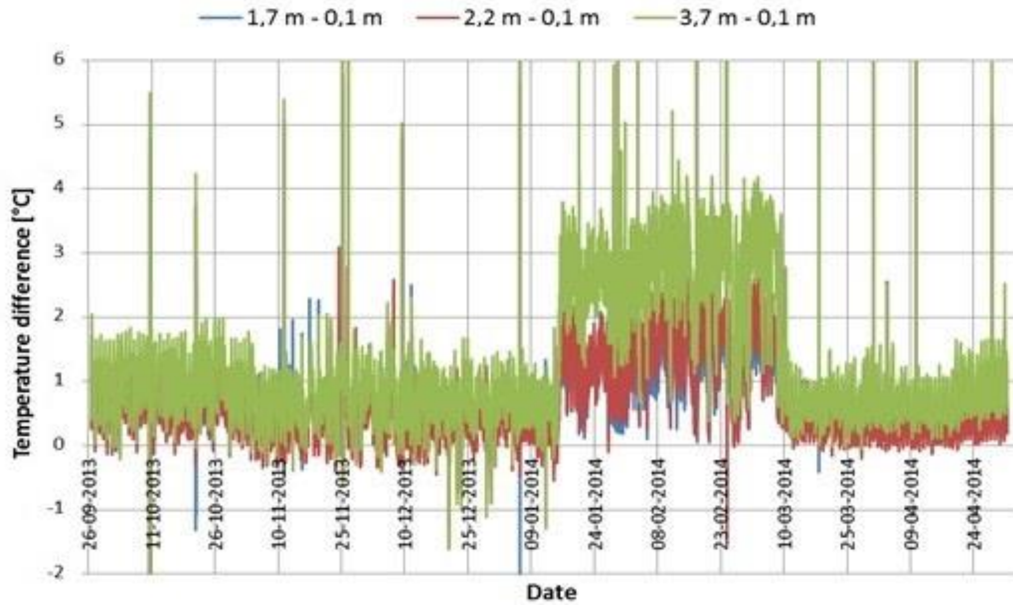

Figure 6. *Air temperature difference between the selected heights, heating season*

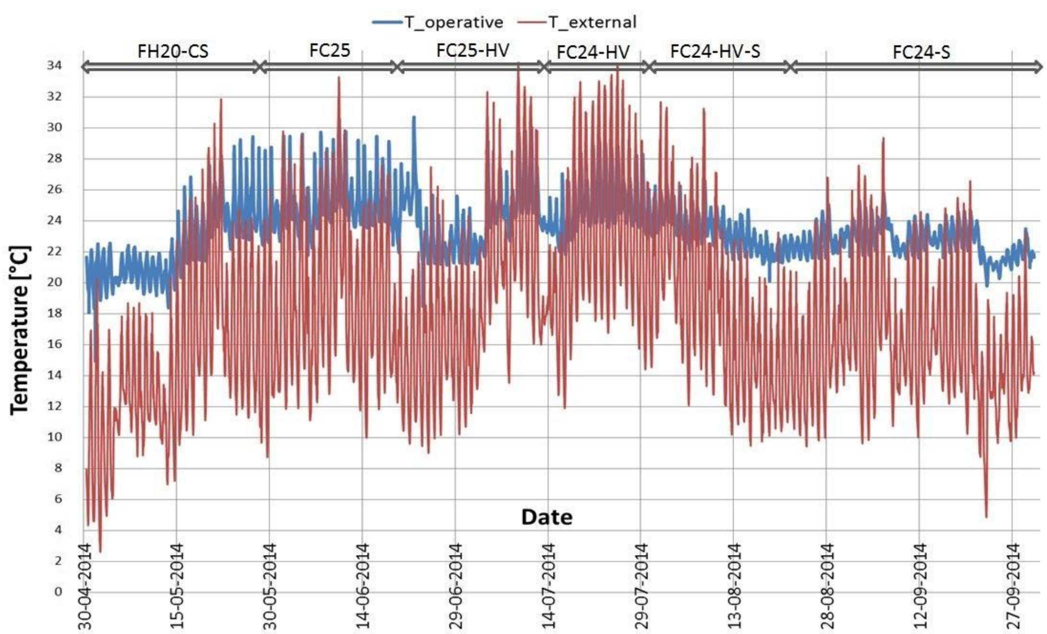

Figure 7. *Operative temperature and external air temperature during the cooling season*

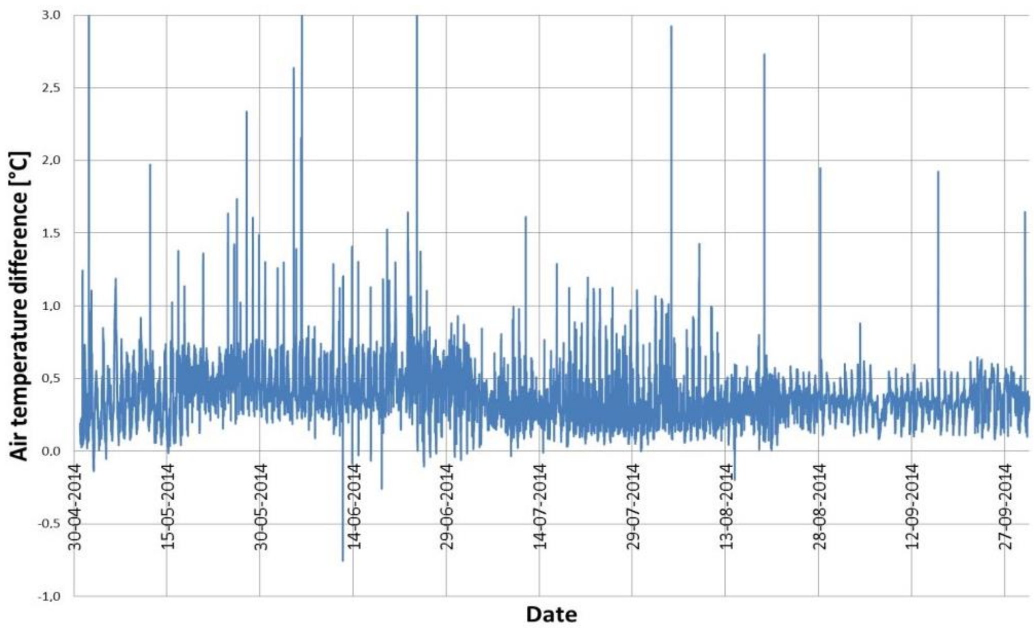

Figure 8. *Vertical air temperature difference between head and ankles during cooling season*

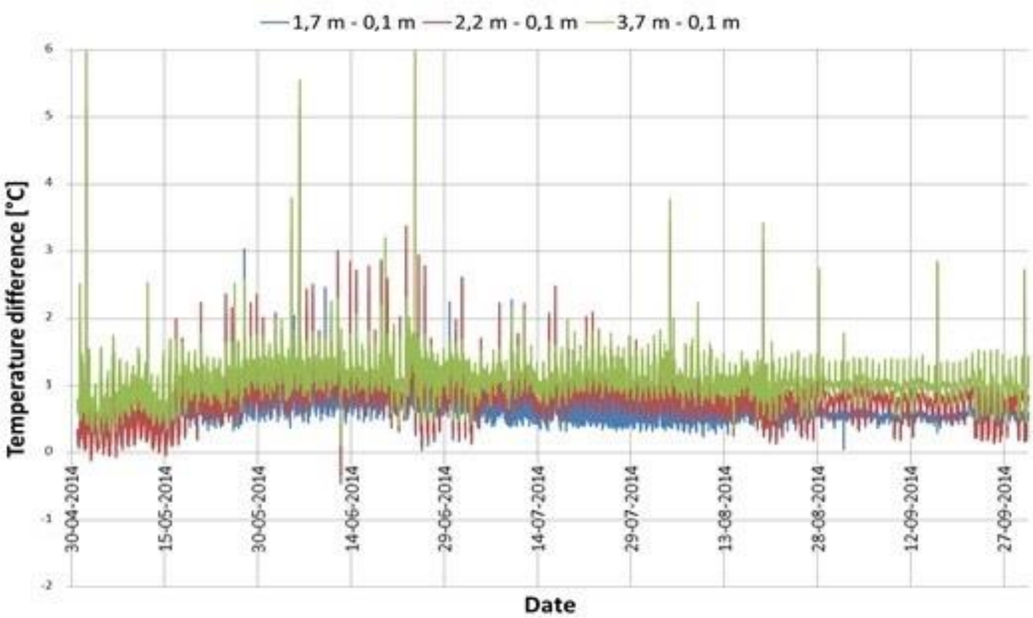

Figure 9. *Air temperature difference between the selected heights, cooling season*

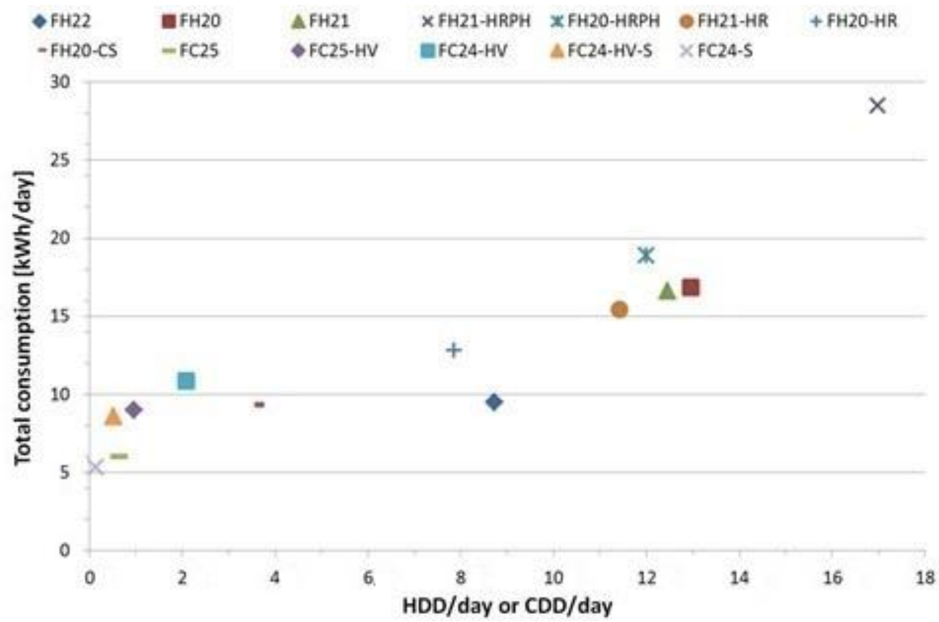

Figure 10. *Energy consumption per day versus heating or cooling degree days per day*

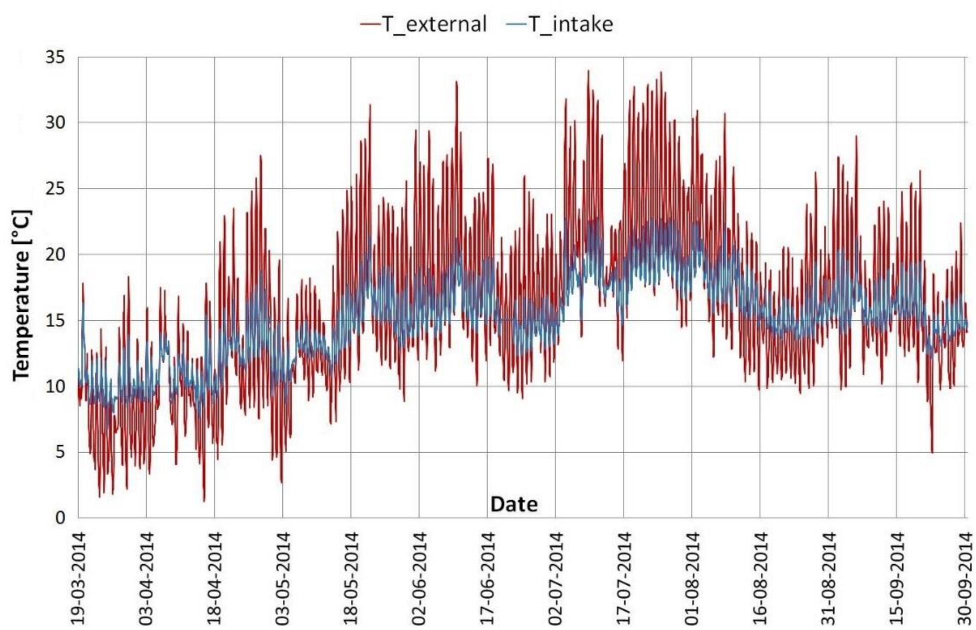

Figure 11. *The external air temperature and the intake air temperature to the AHU*

Tables:

Table 1. *Thermal properties of the envelope*

	North	South	East	West	Floor	Ceiling
Walls, Area, [m^2]	-	-	37.2	19.3	66.2	53
Walls, U-value, [W/m^{2}K]	-	-	0.09	0.09	0.09	0.09
Windows, Area, [m^2]	36.7	21.8	-	-	-	0.74
Windows, U-value, [W/m^{2}K]	1.04	1.04	-	-	-	1.04

Table 2. *Periods and experimental settings of the different cases, heating season*

Period	*Average external air temperature [°C]*	*Floor heating set-point [°C]*	*Ventilation*	*Case abbreviation*
26th of Sep to 21st of Nov	8.2	22	Off	FH22
21st of Nov to 18th of Dec	4.0	20	Off	FH20
18th of Dec to 16th of Jan	4.6	21	Off	FH21
16th of Jan to 10th of Feb	0.0	21	On, heat recovery and pre-heating**	FH21-HRPH
10th of Feb to 10th of Mar	5.0	20	On, heat recovery and pre-heating**	FH20-HRPH
10th of Mar to 3rd of Apr	5.5	21	On, heat recovery	FH21-HR
3rd of Apr to 1st of May*	9.0	20	On, heat recovery	FH20-HR

*: The dummies simulating the occupants and a dummy (equipment #2) were OFF during this experimental period.

**: Heat recovery refers to the passive heat recovery and pre-heating refers to the active heat recovery (warm air heating) in AHU. The supply air temperature was between 30 to 34°C, except for the periods with low outside air temperatures when it dropped to 27°C.

Table 3. *Periods and experimental settings of the different cases, cooling season*

Period	*Average external air temperature [°C]*	*Floor cooling set-point [°C]*	*Ventilation type and ventilation rate*	*Solar shading*	*Case abbreviation*
1st of May to 27th of May*	14.7	20**	Heat recovery, 0.5 ach	No	FH20-CS
27th of May to 19th of June	18.7	25	Heat recovery, 0.5 ach	No	FC25
19th of June to 13th of July	18.7	25	Heat recovery, 0.8 ach	No	FC25-HV
13th of July to 30th of July	22.7	24	Heat recovery, 0.8 ach	No	FC24-HV
30th of July to 21st of Aug	18.1	24	Heat recovery, 0.8 ach	Yes	FC24-HV-S
21st of Aug to 1st of Oct	16.0	24	Heat recovery, 0.5 ach	Yes	FC24-S

*: The dummies simulating the occupants and a dummy (equipment #2) were OFF during this experimental period.

**: Floor system was in heating mode, transition period.

Table 4. *The category of indoor environment based on operative temperature at 0.6 m height, heating season*

Indoor environment category/case	***FH22***	***FH20***	***FH21***	***FH21-HRPH***	***FH20-HRPH***	***FH21-HR***	***FH20-HR***	***Total, average***
Category 1 (21.0-25.0°C)	92%	2%	37%	22%	11%	67%	35%	45%
Category 2 (20.0-25.0°C)	97%	44%	92%	72%	61%	98%	77%	80%
Category 3 (18.0-25.0°C)	100%	95%	100%	93%	99%	100%	100%	98%
Category 4*	0%	5%	0%	7%	1%	0%	0%	2%

*: Category 4 represents the values outside Categories 1, 2, and 3.

Table 5. *Time-averaged vertical air temperature difference between head and ankles, heating season**

Case	***FH22***	***FH20***	***FH21***	***FH21-HRPH***	***FH20-HRPH***	***FH21-HR***	***FH20-HR***	***Total, average***
Temperature difference [°C]	0.4	0.4	0.3	0.7	0.9	0.3	0.3	0.5

*: Vertical air temperature between head and ankles should be less than 2 K, in order to be within the limits of Category A of EN ISO 7730.

Table 6. *Time-averaged air temperature at the selected heights and the difference between highest and lowest measurement points, heating season*

Height/case	***FH22***	***FH20***	***FH21***	***FH21-HRPH***	***FH20-HRPH***	***FH21-HR***	***FH20-HR***	***Total, average***
0.1 m [°C]	21.7	19.2	20.3	19.5	19.5	20.8	20.4	20.4
1.7 m [°C]	22.3	19.7	20.7	20.7	20.7	21.2	20.9	21.1
2.2 m [°C]	22.3	19.6	20.6	20.9	21.0	21.1	20.8	21.1
3.7 m [°C]	22.6	20.0	21.0	22.3	22.3	21.5	21.2	21.7
Temperature difference between 3.7 m and 0.1 m [°C]	0.9	0.8	0.8	2.8	2.8	0.7	0.8	1.3

Table 7. *The category of indoor environment based on operative temperature at 0.6 m height, cooling season*

Indoor environment category/case	***FH20-CS***	***FC25***	***FC25-HV***	***FC24-HV***	***FC24-HV-S***	***FC24-S***	***Total, average***
Category 1 (23.5-25.5°C)	52%	56%	36%	54%	39%	22%	41%
Category 2 (23.0-26.0°C)	73%	72%	49%	72%	58%	36%	57%
Category 3 (22.0-27.0°C)	87%	87%	75%	91%	84%	72%	81%
Category 4	13%	13%	25%	9%	16%	28%	19%
Hours above 26°C	48	129	79	87	7	0	350*
Hours above 27°C	19	71	38	34	0	0	162*

*: Although the overheating hours cannot be directly added for the different cooling strategies, their total is given to indicate the duration of overheating during the cooling season.

Table 8. *Time-averaged vertical air temperature difference between head and ankles, cooling season*

Case	***FH20-CS***	***FC25***	***FC25-HV***	***FC24-HV***	***FC24-HV-S***	***FC24-S***	***Total, average***
Temperature difference [°C]	0.4	0.5	0.4	0.3	0.3	0.3	0.4

Table 9. *Time-averaged air temperature at the selected heights and the difference between highest and lowest measurement points, cooling season*

Height/case	***FH20-CS***	***FC25***	***FC25-HV***	***FC24-HV***	***FC24-HV-S***	***FC24-S***	***Total, average***
0.1 m [°C]	21.7	24.7	23.6	24.7	23.1	22.3	23.2
1.7 m [°C]	22.3	25.6	24.4	25.3	23.6	22.9	23.8
2.2 m [°C]	22.3	25.8	24.6	25.5	23.8	23.1	24.0
3.7 m [°C]	22.7	26.0	24.8	25.7	24.1	23.3	24.2
Temperature difference between 3.7 m and 0.1 m [°C]	1.0	1.3	1.2	1.1	1.1	1.0	1.1

Table 10. *Monthly electricity production from the PV/T panels (month/year)* [kWh]*

10/13	**11/13**	**12/13**	**01/14**	**02/14**	**03/14**	**04/14**	**05/14**	**06/14**	**07/14**	**08/14**	**09/14**	**Sum**
214.6	143.0	61.9	59.7	168.8	310.2	507.3	621.1	665.1	519.7	405.0	308.1	4043.9

*: Electricity production from the PV/T panels between 26-30/09/13 was 59.3 kWh.

Table 11. *Energy consumption of the HVAC system components*

Case	***HDD***	***CDD***	***Heat pump [kWh]***	***Mixing station [kWh]***	***Controller, radiant system [kWh]***	***AHU [kWh]***	***Total [kWh]***	***Total, average [kWh/day]***
FH22	496	-	518.9	15.5	5.3	0.0	539.7	9.5
FH20	350	-	438.3	11.3	3.8	0.0	453.4	16.8
FH21	361	-	460.6	15.0	5.0	0.0	480.6	16.6
FH21-HRPH	425	-	463.2	10.4	3.5	236.6	713.7	28.5
FH20-HRPH	337	-	307.2	2.2	0.8	221.2	531.4	18.9
FH21-HR	275	-	321.8	7.2	2.5	39.3	370.7	15.4
FH20-HR	220	-	304.6	4.3	1.5	47.5	358.0	12.8
FH20-CS	97	-	206.4	1.3	0.4	42.6	250.7	9.3
FC25	-	15	95.4	5.4	1.8	36.0	138.7	6.0
FC25-HV	-	20	110.1	3.2	1.4	75.2	189.8	9.0
FC24-HV	-	36	114.8	6.7	2.0	60.9	184.3	10.8
FC24-HV-S	-	12	105.6	3.8	1.2	78.8	189.3	8.6
FC24-S	-	6	145.9	0.6	0.2	65.7	212.4	5.3
Total	-	-	3592.9	86.7	29.3	903.8	4612.7	-

Table 12. *Overall energy balance of the house (yearly values)*

Heat pump [kWh/m^2]	54.3
Mixing station [kWh/m^2]	1.3
Controller, radiant system [kWh/m^2]	0.4
AHU [kWh/m^2]	13.7
Energy consumption of the HVAC system, total [kWh] and per floor area [kWh/m^2]	4613 / 69.7
Electricity produced by the PV/T panels [kWh]	4044
Overall energy balance (production-consumption) [kWh]	-569

Heating and cooling systems should be evaluated holistically to ensure an optimal indoor environment and energy efficiency simultaneously. Measurements, simulations, and thermodynamic tools were used to compare the performance of different space conditioning systems.
All of the system analysis tools (energy, exergy, and entransy) showed that low temperature heating and high temperature cooling systems (a hydronic floor heating and cooling system in this study) had the highest performance among the compared systems.
Although a single-family house was used for the evaluations, the developed calculation methods can be applied to other buildings and systems.

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