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Energy use and greenhouse gas emissions of air-source heat pump and innovative ground-source air heat pump in cold climate

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Abstract

This article compares climate impacts of two heat-pump systems for domestic heating, that is, energy consumption for space heating of a residential building. Using a life-cycle approach, the study compared the energy and greenhouse gas emissions of direct electric heating, a conventional air-source heat pump, and a novel ground-source air heat pump innovated by a citizen user, to assess if such user innovation holds benefit. The energy use of the heat pumps was modelled at six temperature intervals based on duration curves of outdoor temperature. Additionally, two heat pump end-of-life scenarios were analyzed. Probabilistic uncertainty analysis was applied using a Monte Carlo simulation. The results indicated that, in ideal conditions, i.e. assuming perfect air mixing, the conventional air-source heat pump's emissions were over 40% lower and the ground-air heat pump's emissions over 70% lower,

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1. Introduction

Worldwide, the residential sector uses a large amount of energy (Saidur *et al.* 2007). Residential heating is responsible for a considerable part of household greenhouse gas emissions (Huppes *et al.* 2006). In the last decade, a great deal of interest has been raised by the potential of heat pumps to lower residential greenhouse gas emissions (e.g. Bayer *et al.*, 2012; Greening and Azapagic, 2012). Based on their source of low-grade heat, heat pumps can be roughly divided into two groups: ground-source and air-source heat pumps. Experiments and analyses have been performed, and reports drawn up, on the heating and cooling effects of different ground-source heat pump set-ups (see e.g. Bakirci, 2010; Kim *et al.* 2012; Pulat *et al.* 2009; Self *et al.* 2013). Previous studies have been performed on the environmental impacts of heat pump systems, based on the life cycle assessment methodology (LCA) (Abusoglu and Sedeeq 2013; Blom *et al.* 2010; Greening and Azapagic, 2012; Johnson 2011; Rey *et al.* 2004; Rey Martínez *et al.* 2011; Saner *et al.*, 2010; Shah *et al.* 2008). The significance of the impact of refrigerant has been discussed (see e.g. Johnson 2011, and the references therein). The electricity use has been shown to play a dominant role in the climate impacts of heat pump systems (see e.g. Saner *et al.*, 2010). Hence, the carbon intensity of electricity production plays an important role in the related emissions. Depending on the carbon intensity of the electricity, some of the studies have concluded that heat-pump systems have larger greenhouse-gas emissions than conventional heating systems, such as natural-gas-fired boilers (Blom *et al.* 2010; Abusoglu and Sedeeq 2013; Shah *et al.* 2008), while other studies have shown that use of heat pumps lowers greenhouse gas emissions (c.f. Greening and Azapagic, 2012). It is also commonly assumed that heat pumps diminish electricity consumption when installed in a house with direct electric heating. Aspects such as climate (outdoor temperature), the electricity production mix, and assumed lifespan of heating systems are important to the interpretation of the results.

It is projected that the amount of heat pump thermal energy will triple in EU member states by the year 2020 (Beurskens 2011). In Finland, there is a clear, growing trend towards heat pump installation: approximately 10% more heat pumps were sold in 2011 than in 2010, with the total cumulative number of heat pump units totaling more than 500,000 in 2012 (SULPU 2013). In Finland, total energy consumption for space heating of residential buildings accounted for around 50 TWh in 2011. Energy use by households was responsible for around 11.6 Mt of greenhouse gas emissions (GHGE) in 2009 (Nissinen *et al.* 2012). This accounts for roughly 15% of Finland's total GHGE (Statistics Finland 2011a, 2011b). Around half of Finland's population lives in detached houses, in which direct electric heating is commonly used for space heating. At national level, direct electricity heating accounts for the consumption of 4.6 TWh per year (Adato 2013).

There is a need for cost-efficient supplementary heating systems, in order to cut electricity usage, since many detached houses in Finland do not have hydronic heating systems, i.e. hot water circulating within a central heating system. During the coldest periods, wood burning stoves have traditionally been used in combination with direct electricity heating, but the number of air-source heat pumps (ASHP) has recently been increasing. However, ASHP is problematic in that it does not perform well at low outdoor temperatures (i.e. temperatures of around -15°C). A ground-source heat pump connected to a hydronic heating system does not have this problem, but in houses heated by direct electricity, the installation of such a system is responsible for a significant increase[1] in the heating system's costs.

A new type of ground-source heat pump, namely a ground-source air heat pump (GSAHP) does not require a hydronic heating system. This system integrates a conventional air-source heat pump with a heat collector (horizontal or vertical collector pipes) placed in the ground. A heat collector of this kind is also called a ground heat exchanger (GHE) or borehole exchanger (BHE). Through such a modification, the heat pump can continue to perform well at low outdoor temperatures, whereas a conventional air heat pump would suffer from significantly lowered usability with respect to space heating. It has been reported that the coefficient of performance (COP value) of heat pumps of this new is around 35% higher than that of a conventional ASHP system in a school building (Kim *et al.*, 2012), (see also Pulat *et al.* 2009 on a study of an experimental GSAHP system in Turkey). It should be noted that we are not, here, referring to ground-coupled heat exchangers, also known as earth-air heat exchangers (EAHE), or earth tubes etc. that can be also used for heating purposes (c.f. Bisoniya *et al.* 2013, Chel and Tiwari 2009). Systems of these types do not typically include compressors or chemicals, which accounts for one of their main differences to ground-source heat pump systems.

The purpose of the present study is to identify and evaluate the savings in end-use energy and reductions in greenhouse-gas emissions from different energy-related innovations in Finnish households. This paper focuses on the quantification of potential reductions when using a new type of heat pump, i.e. a ground-source air heat pump. Such an approach is compared to a conventional air-source heat pump and direct electric heating. The conventional heat pump was chosen as a point of comparison, as it represents the most

[1] According to the ground-source heat pump installer company Senera Inc., the total switching cost of a conventional GSHP system is €20,000, in case of a detached house (120 m^2) with an existing hydronic central heating system (e.g. conversion from oil-fired heating). If hydronic heating is not in place, the additional cost is approximately €12,000 (+60%).

common set-up in this geographical and housing context; currently about 500 000 such devices have been installed in Finland (SULPU, 2013). Detached houses with direct electricity heating are considered, since the installation of both conventional air-source heat pumps and the new type of ground-source air heat pump is relevant to such dwellings.

2. Materials and Methods

2.1. Heat pump systems

In this paper, two types of heat pump system are considered. The first is a conventional air-source heat pump (air-to-air) that extracts low-grade heat from the air and converts it into high-grade heat for space heating. This system includes an outdoor fan that is used to pull in outdoor air from which heat is extracted. Such heat is then used to evaporate the refrigerant in the evaporator. Next, the gaseous refrigerant is compressed, raising its pressure and temperature. This high-grade heat is then transferred to the air and distributed around the indoor space using another fan. After the refrigerant has cooled and condensed, it passes through an expansion valve, in which the pressure drops and the cycle restarts from the beginning. A schematic diagram of the heat pump is given in Fig. 1(a). Because a heat pump of this kind does not require any drilling for heat extraction, it is easy to install. Due to technical limitations, the heat pump cannot provide all of the energy needed for space heating during very cold periods. During such times, supplementary heat provided by electric radiators is needed.

The other type of heat pump considered here is an innovative heat pump, a ground-source air heat pump comprising a vertical heat collector (also termed a GHE) placed in the ground, a heat pump unit and a fan. The heat pump unit consists of an evaporator, a compressor and a condenser. The heat pump system is schematically illustrated in Fig 1(b). In the heat collector, an antifreeze solution circulates and transfers the collected low-grade heat from the ground to the refrigerant. Furthermore, the heat pump unit includes a refrigerant-to-air heat exchanger. This transfers the heat to the air, which is then blown into the space with the aid of a fan. The operation principle is analogous to conventional ASHP as explained above, except for the source from which low-grade heat is extracted. The main difference to a typical ground-source heat pump is the distribution of high-grade heat: a ground-source air heat pump uses a fan to transfer the heat to the indoor air, whereas in the case of a ground-source heat pump, the heat is typically transferred to the water by a refrigerant-to-water heat exchanger, and distributed with the aid of an under-floor heating system, or radiators on the walls. Accordingly, GSAHP does not require installation of under-floor pipes or any other pipework that would distribute heat indoors via water. However, the heat collector of GSAHP requires drilling or digging, depending on the style of installation. Collector designs can be divided in two types by piping orientation: horizontal and vertical designs. A ground source air heat pump has been commercialized by Jääsähkö Inc. Finland, as the culmination of 113 modifications and innovations by users of heat pump technologies in Finland (Hyysalo *et al.* 2013a, 2013b).

[insert Figure 1 here]

Figure 1 Schematics of studied ground-source heat pump system: (a) air-source heat pump system (ASHP), (b) ground-source air heat pump system (GSAHP).

2.2. Theoretical framework

To assess the greenhouse-gas emissions of heat-pump systems (i.e. carbon footprint), the principles of life cycle assessment (LCA) were employed. The aim of this study is to compare and describe the relevant physical flows of systems for space heating. Thus, an attributional LCA approach was used.

The scope of the calculations was limited to encompass only the life cycle of the heat pump, and the construction of the building and the installation of the original electric heating system were excluded. The heat pumps were assumed to be installed as retrofit. This means that, in the case of GSAHP, account is taken of installation work for a heat collector in accordance with Greening and Azapagic (2012). Both systems have an indoor unit, including a fan that blows hot air within a certain space. This indoor unit is rather easy to install and requires no extensive work. Due to the minimal installation work required for the ASHP system (both the indoor and outdoor unit), the installation process was excluded from the assessment. The aim was to compare the baseline situation, i.e. direct electric heating, with two heat pump scenarios. Because the use phase has been identified as the biggest contributor to climate impacts in previous studies (see e.g. Greening and Azapagic, 2012), the use phase was analyzed in more detail than the manufacturing and end-of-life phases.

A theoretical-one-room building that has three occupants, a floor area of about 150 m^2, and an indoor temperature of +21°C was studied. The air mixing and the heat distribution are thought to be ideal in the building, since the room layout is assumed to be very simple. In practice, some extra heating would be needed to maintain the temperature at +21°C all around the building (notice that air heat pumps heat and circulate the existing air in the building). The thermal and other technical properties of the property are collated in Tab. 1, determined from the building code in Finland (Ministry of Environment, 2008). The functional unit was heating the above-described detached house for one year in Finland. Only the global warming potential indicator was included in the LCA, and thus only the carbon footprint (and no other environmental impacts) was determined in the life-cycle impact assessment.

Table 1 Technical details of the studied property (Ministry of Environment, 2008).

	Floor	Roof	Outer walls	Windows	Doors
Thermal transmittance [W/m^2K]	0.24	0.15	0.24	1.4	1.4
Area [m^2]	147	147	113	24.5	8.2

The energy performance and demand of the building were calculated for four different climate conditions in Finland. In the calculation the passive gains, such as thermal gains from the sun, people and use of appliances were taken into account. The mildest climate is in the coastal and Southern part of the country (climate zone I) whereas the coldest climate is in the North (climate zone IV). The annual average outdoor temperatures in zone I and IV are 5.3°C and -0.4°C, respectively. The average outdoor temperatures during the coldest month of the year lie in zone I and IV at -4.5°C (February) and -13.06°C (January), respectively. (Ministry of Environment, 2011)

The operation of the heat pumps was modelled at six temperature intervals based on duration curves of temperature. For the energy usage calculation Eskola *et al.* (2012) was followed. All the relevant equations are described in the supplementary section of this article. The basic idea of the method is to calculate the heating energy need, heat provided by the heat pump, and the supplementary direct electric heating needed at the selected six temperature intervals. Energy for space heating generated by a heat pump and the pump's electricity consumption were calculated using the heating demand of the hypothetical house and the temperature-dependent coefficient of performance of the heat pump. The duration curves of temperature in different climate zones were according to the Finnish building code (part D3) (Ministry of Environment, 2011).

2.3. Used data

The conventional air-source heat pump was assumed to be performing as indicated in tests conducted by the Swedish Energy Agency (2008). In the case of the GSAHP, performance tests have not been done, so the COP factors were obtained from the manufacturer. The air-source heat pump was assumed to operate in an on-off manner, whereas the ground-source air heat pump was known to operate with an inverter. The relevant technical details of the heat pumps are presented in Tab. 2.

Table 2 Technical specifications of the studied heat pumps. Data from tests of Swedish Energy Agency (2008) and ground-source air heat pump producer Jääsähkö Inc. (Salmela 2012).

	Air-source heat pump	Ground-source air heat pump
Refrigerant	R-134a, 4.7 kg	R-410, 1.2 kg
COP (Coefficient of performance, with 100% power)	2.2 at -15 °C 2.6 at -7 °C 2.8 at 2 °C 3.2 at 7 °C	3.45
COP (50% power)	n.a.	4.7
Power out [kW]	3.1 at -15 °C 3.5 at -7 °C 4.0 at 2 °C 4.9 at 7 °C	5.0 (with full power) 2.5 (with 50% power)

The carbon footprint of the raw material acquisition, manufacturing and transportation of the ASHP system was calculated similarly to Greening and Azapagic (2012). The end-of-life phase evaluation was based on Finnish conditions, i.e. it was assumed that the heat pump system is treated in a process that produces about 700 kg CO_2e/tonne of waste, and that ethylene glycol is combusted, generating around 1.4 kg CO_2e/kg of waste.

The materials of the GSAHP were assumed to follow Greening and Azapagic (2012) except for the refrigerant. The amount of refrigerant in the GSHP system was estimated by the Finnish manufacturer, and the refrigerant in the GSAHP was 1.2 kg of R-410A, which has a GWP100 value of 2088 (Solomon *et al.*, 2007). The vertical heat collector set up is a more probable installation form than the horizontal, because most houses do not have the big yard needed for the horizontal setup. According to Azapagic and Greening (2012), digging for the horizontal collector is slightly more energy intensive than drilling the bore hole, and the pipework is longer in horizontal setup than in vertical setup. In vertical design, it was assumed that pipework is 300 m long and the collector is located in a 300 m deep borehole (see Greening and Azapagic, 2012). The assumed life-time of both heat pumps was 20 years, as in the study by Greening and Azapagic (2012). The compressor unit was assumed to be the restricting component of the system life-time, as other components such as piping, fans, etc. have longer life-times. The findings of Greening and Azapagic (2012) and the Ecoinvent database (2010) were used as a source of data for transportation of components, drilling equipment, and heat pumps.

2.4. Scenarios

The baseline scenario was defined as a building with direct electricity heating, in other words no heat pump is installed and used for heating. Because the refrigerants used in the heat-pumps at hand (R-134a and R-410) are known to have high potential for global warming (Solomon, 2007), the end-of-life stage was examined according to two scenarios: best and worst-case scenarios. Because the number of installed heat pump units is steadily growing, there is a corresponding, potential risk that emissions will grow, if heat pump users are negligent in their handling of refrigerants. Two extreme scenarios were studied in order to quantify the order of magnitude of refrigerant management's impact during the retirement of the heat pump systems. In both scenarios, the climate impacts of heat pumps were compared: of a conventional air-source heat pump in one case, and of an innovative ground-source air heat pump in the other. In scenario one, the waste treatment was assumed to be done properly, assuming no refrigerant leakage during the retirement of the system. Scenario two also considered both heat pumps, but assumes 100% leakage of the refrigerant during the retirement of the heat pump. This would correspond to a very negligent user who does not follow the appropriate waste-treatment procedure for heat pumps. In the sense of end-of-life treatment, scenario one corresponds to the best case and scenario two to the worst case. In reality, user behavior and waste handling are somewhere between these two extremes during the retirement of heat pumps.

2.5. Sensitivity Analysis

The risks and conditions embodied in the given results can be better understood when a sensitivity and uncertainty analysis is done (Saltelli *et al.* 2008). An uncertainty analysis was used to study parameter uncertainty. Parameter uncertainty and variability were propagated with Monte Carlo simulations, using the Simulacíon 4.0 add-in of MS Excel.

A Monte Carlo simulation with 5,000 iterations was used. All input variables were assumed to be independent of each other. In all, 32 and 44 sources of parametric uncertainty were included in the simulation for ASHP and GSAHP, respectively. In environment and ecology, the continuous log-normal distribution is widely used and it was considered to correspond to the distribution of these parameters too. Thus, all uncertainties were accounted for, based on a continuous log-normal distribution with a positive range for random variables, and parameterized using the median and the uncertainty factor. The 95% confidence intervals for the distributions were obtained using a relatively common approach, by multiplying and dividing the median by the uncertainty factor (UF) (c.f. Mattila *et al.* 2012, Frischknecht *et al*. 2005).

Because a pre-defined example building was studied, the areas and thermal transmittances of building elements were assumed to have no uncertainty. Electricity production was included with the variation reported by Statistics Finland by using a benefit allocation method in 2000-2010, resulting in a value of 205-350 g CO_2/kWh (Statistics Finland, 2012). The uncertainty of R-410a production was according to Johnson (2011). Expert judgment was used to define the rest of the uncertainty factors. The masses of refrigerants were estimated to have an uncertainty factor of 1.2. Similar uncertainty factors (1.1-1.3) were used for the materials used in the manufacturing the heat pumps. The ship transportation, mass of ethylene glycol and its waste management were included with a higher uncertainty factor (UF about 1.5). The emission factor of electronic waste management was included with very high uncertainty (UF=2.5). The leakage percentage of refrigerants was let to vary between 0.7% and 1.4%.

3. Results

3.1. Energy and emission results

Fig. 2 shows the GHGE results of space heating for the studied systems for scenario one (best case). The figure presents the emissions accounted for one year, with the total life time of the heat pump systems being 20 years. From the figure, it is seen that emissions of direct electricity using radiators to heat the indoor air are higher than emissions of heat-pump systems in every climate zone. In addition, the ground-source air heat pump has much lower emissions than the air-source heat pump in every zone. Furthermore, it can be clearly seen from Fig. 2 that the emissions increase from the first climate zone to the fourth. This is an obvious result, as the climatic conditions become colder and more heating is needed towards the North. One should also note that the difference in emissions between the conventional and the new type of heat pump grows when moving from a milder climate zone to a colder one. This is explained by the fact that in climate zones III and IV, there are more days when the COP of the heat pump drops and the ASHP cannot function very well. In other words, in colder zones, there are fewer days during which the temperature is between 7-12°C, the generally optimal temperature range for ASHP operation.

In agreement with prior studies, the electricity use can be clearly seen as the major contributor to the emissions in every system and climate zone. The electricity use includes the emissions of electricity used by the heat pump as well as the supplementary electricity for heating with radiators. The heating with radiators is needed when the heat pump does not provide all the energy needed for space heating due to technical limitations, such as in very cold periods. The materials' production and end-of-life stage makes only a minor contribution to the emissions in this scenario.

The results of scenario two (worst case) are presented in Fig. 3. In this scenario, during the retirement of the heat pump, we assumed complete leakage of the refrigerant in the heat-pump appliance. The impact of leakage is related to the amount lost and the relative GWP value of the refrigerant, and is now clearly seen in the results; the emissions of the end-of-life phase increased the total carbon footprint by around 10% compared to Fig. 2. The figure also shows the fact that the impact of leakage in the ASHP system is bigger, as the system contains more refrigerant compared to the studied GSAHP.

In Fig. 4, the average output power of the GSAHP system as a function of time in climate zones I and IV (mildest and coldest climate zones) is plotted. The figure also shows the total power needed for space heating. The average power has been calculated at six temperature intervals, and the hours have been sorted with respect to temperature intervals in ascending order. This means that the coldest periods (and hours) are the first hours, shown left, whereas the warmer interval and hours are towards the right on the x-axis. The last temperature interval is the warmest (temperature above +12°C) and thus no heating is required, so heating power is zero. From Fig. 4, we see that the total power required for space heating is greater than the average heating power of the GSAHP during the first 1700 hours in zone IV. This means that the heat pump cannot produce enough heat in the two coldest temperature intervals. Therefore, supplementary heating is needed in order to satisfy the energy demand for space heating. This supplementary heating energy produced by direct electric heating, is marked in the figure with colored rectangles for zone IV. In zone I, there is also a need for supplementary heating in the two coldest temperature intervals. However, the length of the coldest period is shorter in zone I compared to zone IV, so the amount of supplementary energy is clearly smaller than in zone IV.

[insert Figure 2 here]

Figure 2 Greenhouse gas emissions (GHGE) of space heating in the example house in different climate zones for scenario one. Three different heating systems are presented: direct electricity (DE), air-source heat pump (ASHP), and ground-source air heat pump (GSAHP).

[insert Figure 3 here]

Figure 3 Greenhouse gas emissions for scenario two, assuming total leakage of the refrigerant during the retirement of the heat pumps. Abbreviations are as in Fig. 2

[insert Figure 4 here]

Figure 4 Average heating power as a function of time in climate zones I and IV. The required total power for space heating is marked with a continuous line for zone IV, and a dashed line for zone I. The average heating power of the ground-source air heat pump is marked with a dotted line (zone IV) or a dash-dotted line (zone I). See the text for details.

3.2. Results of the uncertainty analysis

For clarity, the results only for the mildest climate zone (I) are shown here. The analysis is similar for other climate zones but the energy needed for heating increases when moving to colder regions. Figure 5 shows the carbon footprint distributions for the two studied heat pump systems according to the Monte Carlo simulation. The carbon footprint results show overlap (Fig. 5 a). The distribution of the ASHP is broader and flatter than that of the GSAHP. It is clearly more probable that the carbon footprint of GSAHP is smaller than of ASHP; the median value for GSAHP is about 1100 kg CO_2e, whereas for ASHP it is about 1800 kg CO_2e (Fig. 5 b).

The contribution of each input variable was assessed with Spearman's rank correlations (Spearman's rho) between the input variable and the carbon footprint. The top five rank correlations of the heat pump systems system are shown in Tab. 3. The most influential parameters are related to the electricity, whereas all other parameters have a very low rho value (<<1). The amount of electricity used by the heat pump is mainly dependent on the indoor temperature and the COP of the heat pump. In other words, by decreasing the required indoor temperature and by increasing the COP value of the system, the electricity use is minimized.

[insert Figure 5 here]

Figure 5 Carbon footprint distributions of a ground-source air heat pump (GSAHP) and a conventional air-source heat pump (ASHP), climate zone I (a); the cumulative distributions are shown in (b).

Table 3: Spearman rank correlations between the five most influential variables and the carbon footprint of studied heat pump systems.

Ground-source air heat pump		Air-source heat pump	
Input parameter	Spearman's rank correlation (ρ)	Input parameter	Spearman's rank correlation (ρ)
Emission factor of Finnish electricity	0.76	Emission factor of Finnish electricity	0.77
Amount of used electricity	0.60	Amount of used electricity	0.59
Emission factor for copper production	0.034	Amount of natural gas in production stage	0.027
Amount of ethylene glycol	0.028	Van transportation distance	0.020
Amount of bentonite	0.027	Distance to installation site	0.019

4. Discussion

The results indicate that a GSAHP cuts the electricity use and emissions more than a conventional ASHP compared to direct electricity heating. The clear benefit of a GSAHP is that it extracts heat from the ground, where the temperature does not drop as significantly as the outdoor air temperature during winter. The benefits of a ground-source air heat pump relative to a conventional air source heat pump increase the colder the building location is. The electricity consumption in Finland is highest in winter, and the share of fossil fuels in the electricity production is large. In winter, the emission factor for electricity has typically a high value, and at the same time ASHP fails to function. This means that GSAHP system has clearly more potential to reduce greenhouse gas emissions especially during wintertime.

The invention of GSAHP is evidence of citizen-originated solutions to find adequate solutions for locations and markets that are not considered by mass manufacturers as relevant or lucrative (von Hippel, 2005). In Finland alone 113 inventions or modifications were made to heat pump equipment between 2005-2012 (Hyysalo *et al.* 2013). Adapting heating technologies to the particularities of the local climate and building conditions, such as the combination of large direct electricity heated housing stock and cold climate in Finland, can thus be considered a warranted facet of climate change mitigation efforts worth attention.

The calculations presented in this paper include some simplifications. For instance, the room layout of the considered house was assumed to be very simple and the distribution of heat indoors is assumed to be ideal. In practice, some extra heating of the make-up air is needed. Additionally, it is challenging to define the emission factor for electricity (see e.g. Soimakallio *et al.* 2011). Moreover, the actual energy mix for electricity production is very complex and involves not only national but also Nordic energy production (Kopsakangas-Savolainen and Svento 2012).

It is worth noticing that the rebound effect has been reported to affect the total reduction potential of energy measures in households (Gram-Hanssen *et al.* 2012). The heat pump system can be used for cooling during summer, for instance. However, Finnish summers are mild, and there are fewer than 30 days per year with potential cooling needs, and even then temperatures are almost never above 30^{o}C. In these conditions, energy consumption of an air-source heat pump for cooling remains modest, and the ground-source air heat pump even more, due to cool ground around the collector circuit. Although the assessment of the effects of the rebound effect was beyond the scope of this paper, the rebound effect has to be taken into account when making long-term planning for sustainable housing. Either way, rebound effects should not be overplayed or used as an excuse, as they can be small (Gillingham *et al.* 2013).

In reality, the energy usage in households is very much dependent on the behavior of the occupants. First of all, the indoor temperature has an impact on the thermal transmittance and thus on the required energy for space heating. Secondly, in order to let the heat distribute in the best possible way, the correct placement and installation of the heat pump is important. In addition, the proper maintenance of the heat pump also affects the system's energy consumption.

The sensitivity analysis showed that the uncertainty of emission factor of electricity and the overall electricity are the most important variables explaining the emission results (high correlation value). As the emission factor of electricity had the highest correlation for both studied heat pump setups, the carbon footprint distributions differ mainly due to the electricity use. In Finnish conditions with a cold winter, a conventional heat pump (ASHP) uses more electricity during a year than a GSAHP in general.

The refrigerants used in heat-pump equipment are typically so-called fluorinated greenhouse gases (F-gases). It is a known fact that F-gases have high global warming potential (GWP) values in general. Although in the European Union there are regulations in place that aim at reducing emissions of F-gases, there is a risk of remarkable climate impact as leakages may occur during installation, servicing and disposal of heat pump equipment.

This study focused on greenhouse-gas emissions and climate impacts, and found that the operation phase was a major contributor to the impacts. This fact holds for many other environmental impacts, too (see e.g. Greening and Azapagic, 2012).

5. Conclusions

In this study, the level of greenhouse gas emissions during the life cycle of two heat pump systems was determined: that of a conventional air-source heat pump and that of an innovative ground-source air heat pump. The systems' performance was analyzed and compared to direct electric heating in four climate zones. According to our results, there were clear differences in greenhouse-gas emissions in the studied systems. Theoretically, the ground-source air heat pump system had the best performance in this sense. This is because the ground-source air heat pump has higher coefficient of performance and uses less electricity especially in low outdoor temperatures. The benefits of the ground-source air heat pump relative to the conventional air-source heat pump increase the colder the building location is. The life-cycle approach was seen to be important when looking at the end-of-life treatment of the heat pumps, as total leakage of the refrigerant increased the emissions by around 10% during the system life-time, compared with proper waste management during the retirement of the system. A probabilistic sensitivity analysis demonstrated that the most influential parameters are related to electricity.

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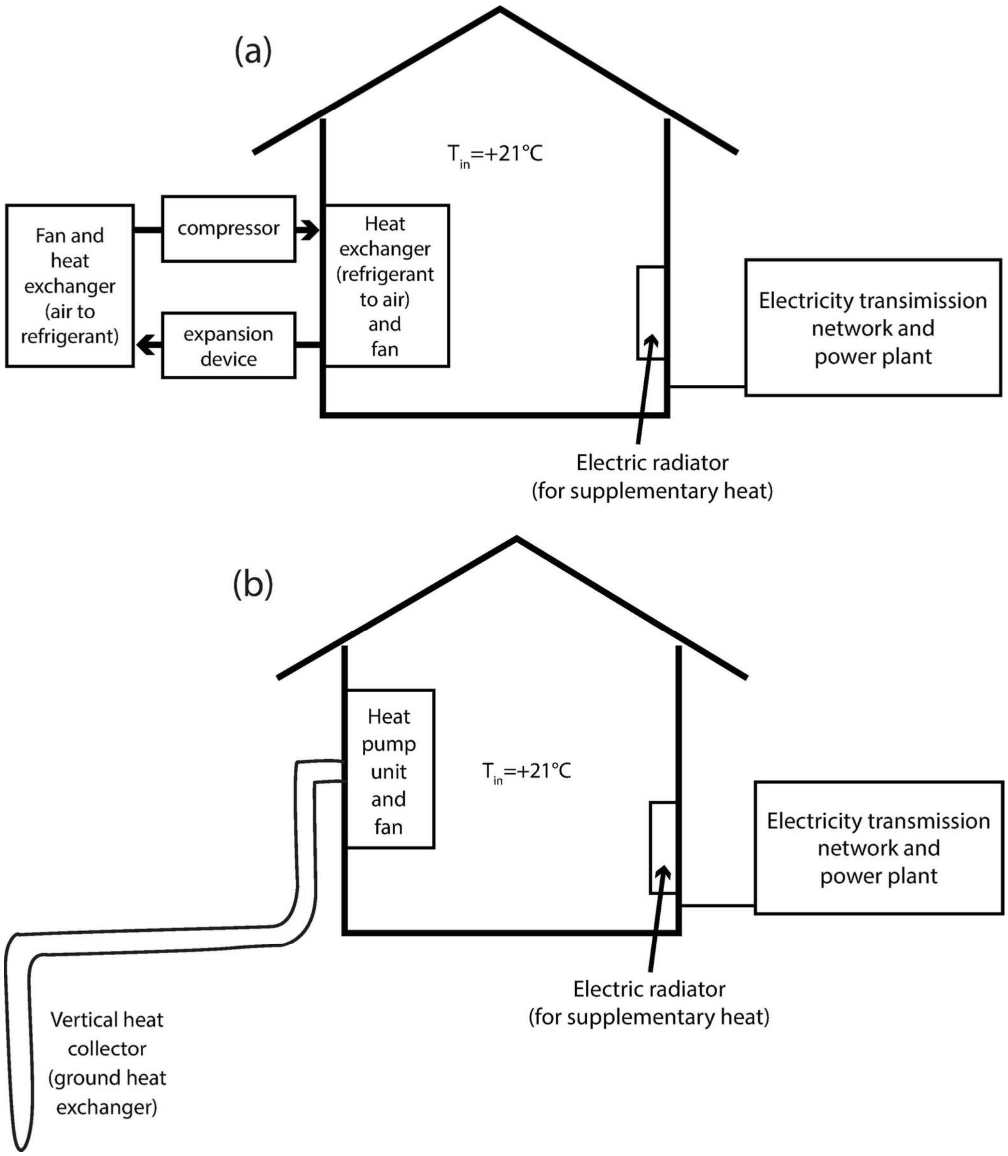

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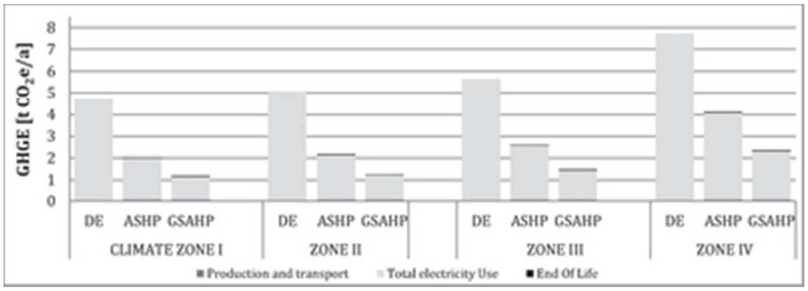

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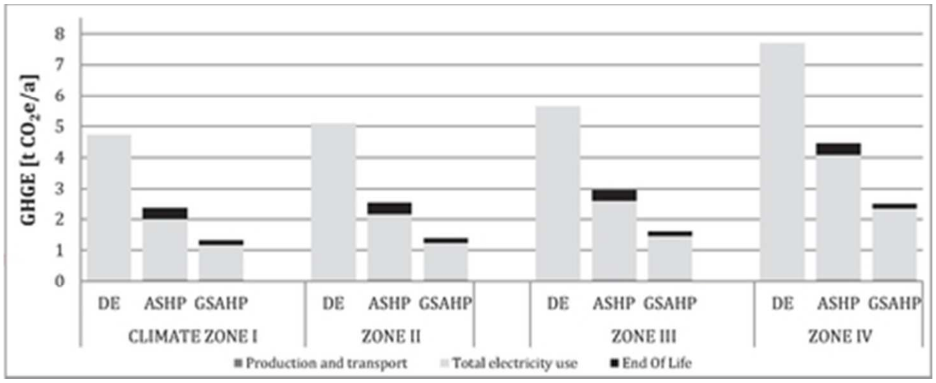

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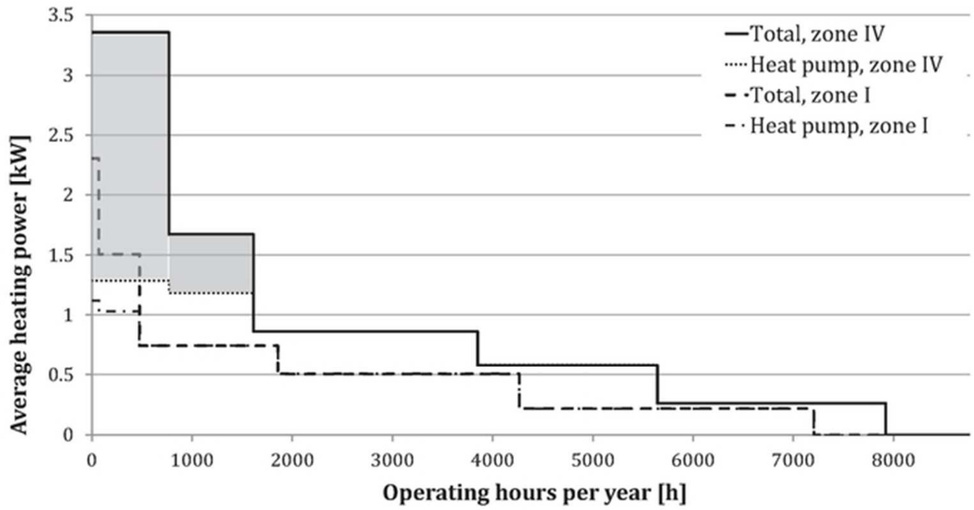

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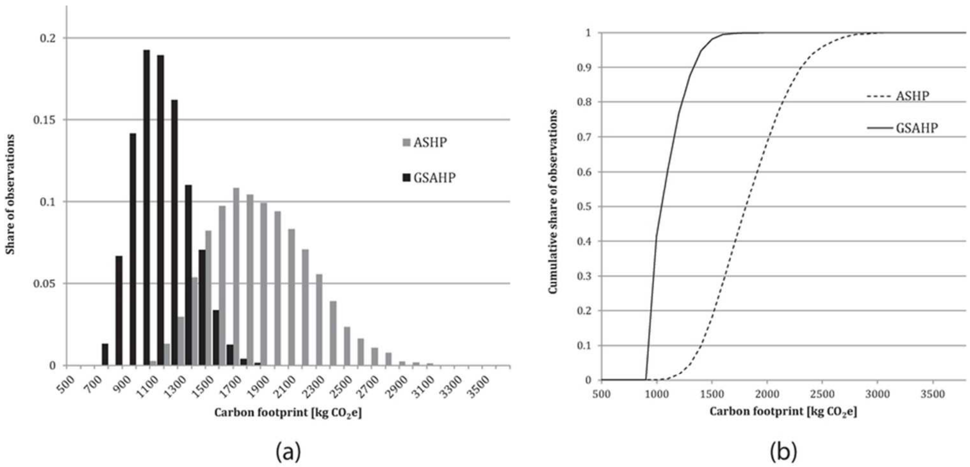

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Summary

This supporting information provides more details about the energy calculation method used for determining the electricity use of the heat pumps and supplementary electric heating. This description is based on the Finnish report by Eskola *et al.* (2012).

In the method, temperatures are divided into six intervals, see supporting Tab. S1. However, in the coldest climate zones III and IV, the lower boundaries are selected to be -29°C and -35°C, respectively. The Eqs. (S1-S6) are applied for each temperature interval in order to calculate the energy use.

Table S1. Selected temperatures for intervals [°C].

Operating temperature of the heat pump, outdoor temperature (T_{out})	-20	-15	-7	2	7	20
Lower boundary of the temperature interval (T_{lb})	-22	-18	-11	-2	4	14
Upper boundary of the temperature interval (T_{ub})	-18	-11	-2	4	14	28

The weighting coefficient for space heating (k_i) in the interval i is calculated as follows:

$$k_i = (DH_{ub}{}^i - DH_{lb}{}^i) / DH_{hp}, \qquad \text{(S1)}$$

where $DH_{ub}{}^i$ [°Ch] is the cumulative degree hour value in upper boundary of the interval i, $DH_{lb}{}^i$ [°Ch] is the cumulative degree hour value in the lower boundary of the interval i, and DH_{hp}[°Ch] is the degree hour value for the heating period (threshold value for heating is +12 °C). All degree hour values are obtained from table compiled for Finland for the different climate zones (I-IV) than can be found in the report by Eskola *et al.* (2012).

The required space heating is calculated for each interval by using the weighting coefficient k_i as follows:

$$Q_i = k_i Q_{tot}, \qquad \text{(S2)}$$

where Q_i is the required heating energy in the interval i, and Q_{tot} is the total required heating energy during the heating period.

For inverter type of heat pump, the part power (β_i) in the interval i can be expressed as:

$$\beta_i = Q_i / (\varphi_i \, t_i), \qquad \text{(S3)}$$

where φ_i [kW] is the output power of the heat pump in interval i, and t_i [h] is the number of operating hours of the heat pump in interval i (i.e. the duration [h] of the temperature interval i).

We assume that there is no limiting temperature for the operation of the heat pump. However, when temperature decreases the performance of the heat pump decreases as well (COP decreases). We assume that the heat pump operates also during cold periods but the output heat does not provide all the required heat, especially below the design point. The supplementary heating energy needed for the interval i (Q_{sh}^i) is calculated as follows:

$$Q_{sh}^i = k_{sh}^i Q_i, \qquad \text{(S4)}$$

where the share of the supplementary heating (k_{sh}^i) is calculated by using Eqs. (S5-S7):

$$k_{sh}^i = [DH_{dp} - (T_{in} - T_{dp}) N_{dp}] / DH_{ub}^i, \qquad \text{(S5)}$$

where DH_{dp} is the cumulative degree hour value at the design temperature point [°Ch], T_{in} [°C] is the indoor temperature, T_{dp} [°C] is the temperature of design point, and N_{dp} [h] is the cumulative number of hours at the design point.

$$k_{sh}^i = (DH_{dp} - DH_{lb}^i) - (T_{in} - T_{dp})(N_{dp} - N_{lb}^i) / (DH_{ub}^i - DH_{lb}^i), \qquad \text{(S6)}$$

where N_{lb}^i [h] is the cumulative number of hours at lower boundary of the temperature interval i.

$$k_{sh}^i = 1 - (T_{in} - T_{dp})(N_{ub}^i - N_{lb}^i) / (DH_{ub}^i - DH_{lb}^i), \qquad \text{(S7)}$$

Eq. (S5) is used when design temperature is in the lowest temperature interval. If design temperature is not in the lowest temperature interval, Eqs. (S6) and (S7) are used. Eq. (S6) is used in the interval, in which the design temperature is. Eq. (S7) is used in those intervals in which the design temperature is higher than the upper boundary of an interval.

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