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Energy and Exergy Performances of Air-Based vs. Water-Based Heating and Cooling Systems: A Case Study of a Single-Family House

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ABSTRACT

Air-based and water-based indoor terminal units can be used to heat and cool indoor spaces. The choice of terminal unit has direct effects on occupant thermal comfort, and on system performance through energy and exergy inputs to heating and cooling plants and through energy and exergy inputs to auxiliary components (pumps and fans).

Energy and exergy performances of a floor heating and cooling system were compared to an air heating and cooling system using different heating and cooling plants. The whole chains of exergy flows were studied, and the importance of auxiliary components in whole system performance was studied.

The radiant floor heating and cooling system performed better than the air-based system in terms of energy and exergy input to the heating/cooling plant. In heating, floor heating required 15% and 17% smaller exergy input compared to warm-air heating, when coupled to a boiler and to a heat pump, respectively. In cooling, by using a floor cooling system coupled with the ground, the overall exergy consumption can be reduced by 78% compared to an air cooling system and 80% compared to a floor cooling system coupled with an air-to-water heat pump.

Water-based systems required significantly smaller exergy input to the auxiliary components compared to air-based systems; 68% less in heating and 53% less in cooling, indicating a clear benefit for the water-based system over the air-based system.

Pump and fan powers become comparable to heating and cooling loads when considered in exergy terms. The exergy input to auxiliary components should be minimized to fully benefit from the water-based low temperature heating and high temperature cooling systems, and in general in heating and cooling systems.

INTRODUCTION

Heating and cooling systems can be considered to consist of three main parts; heating/cooling plant, distribution, and heat emission/removal. These parts should be considered as a whole, since heating and cooling systems have direct and significant effects on occupant comfort in indoor spaces (thermal and acoustic), energy performance of buildings, and on the energy flows and greenhouse gas emissions on a global scale. A holistic design approach should ensure that occupants are satisfied with the indoor environment, and that this is achieved with the lowest possible energy use.

Different heat emission and removal systems (terminal units) are available to condition indoor spaces, mostly

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being air-based or water-based systems, depending on the heat emission or removal medium used. Previous studies evaluated the performance of air-based and water-based heating and cooling systems for office buildings (Sastry and Rumsey 2014; Fabrizio et al. 2012; Imanari et al. 1999), but so far there has only been little focus on residential buildings and dwellings regarding cooling systems and their exergy performance.

Several studies have documented that energy analysis alone is not sufficient to completely understand energy use (Dovjak et al. 2010; Yildiz and Gungor 2009; Shukuya 1994). In addition to the energy analysis, exergy analysis enables us to compare the effects of working temperatures and qualities of different energy sources and flows.

Energy and exergy performances of air-based and water-based heating and cooling systems were compared in this study using a single-family house. Floor heating and cooling systems were compared to warm-air heating and air cooling systems using different heating and cooling plants. Whole chains of exergy flows were studied, and particular attention was paid to the auxiliary components (pumps and fans), in terms of energy and exergy, to emphasize their importance in the whole system performance.

DETAILS OF THE CASE STUDY

The studied house was assumed to be in Copenhagen, Denmark. Construction details, description and details of the heating, cooling and ventilation systems of the actual house are given in Kazanci et al. (2016a) and Kazanci and Olesen (2014).

Description of the Cases

It was assumed that the house was heated or cooled with different space heating and cooling systems: three heating and four cooling systems. Figure 1 shows the schematic drawings of these systems: floor heating connected to a boiler (FH-B), floor heating connected to an air-to-water heat pump (FH-HP), warm-air heating with heat recovery on the exhaust air (WAH), floor cooling connected to an air-to-water heat pump (FC-HP), floor cooling connected to a ground heat exchanger (FC-GHEX), air cooling using outdoor air as the intake (AC-OA) or using cooler air from crawl-space (AC-CS).

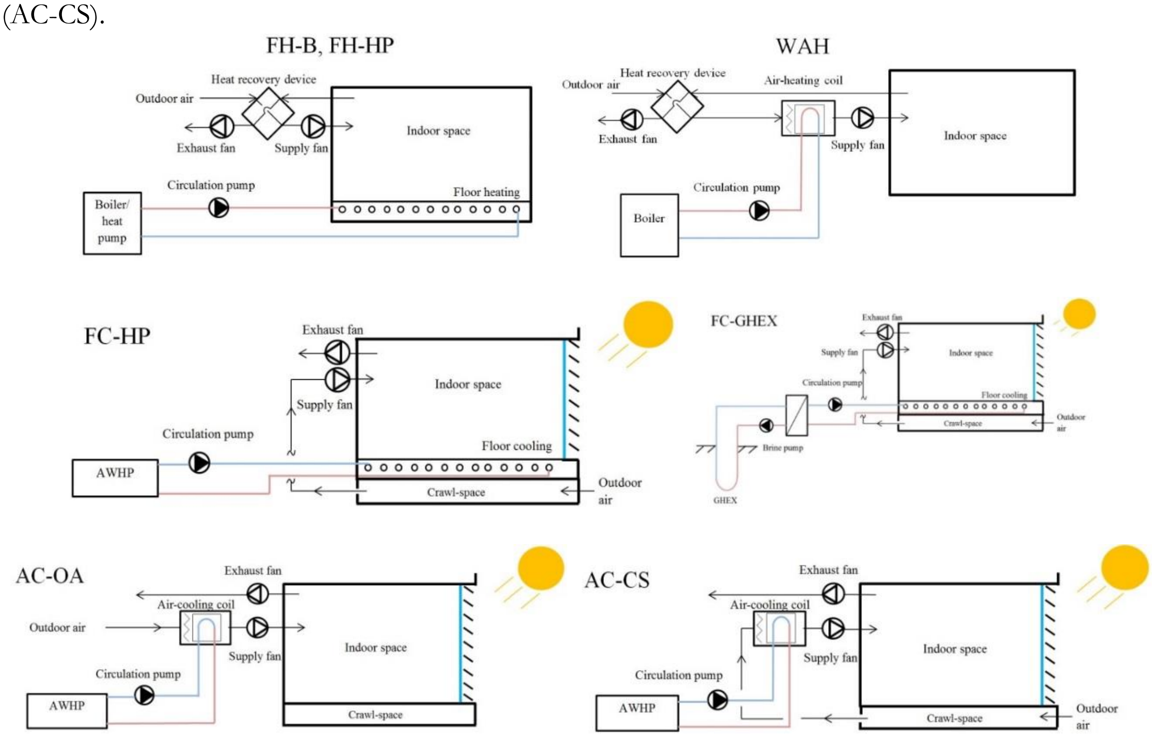

Figure 1 Schematic drawings of the studied systems (Kazanci et al. 2016a, 2016b)

Determination of the Design Heating and Cooling Loads

The calculations were carried out under steady state conditions and the outdoor temperatures were assumed to be -5°C (23°F) and 30°C (86°F) in winter and summer, respectively. The indoor temperatures (air and mean radiant temperatures) were 20°C (68°F) and 26°C (78.8°F) in winter and summer, respectively.

The internal heat gain was assumed to be 4.5 W/m^2 (1.4 Btu/hft^2) which represents two persons at 1.2 met and other household equipment. No solar heat gains were considered in heating season. In cooling season (July, noon), assumed direct solar radiation on the South and West directions were 390 (123.6) and 149 W/m^2 (47.2 Btu/hft^2), respectively, and the diffuse solar radiation was 32 W/m^2 (10.1 Btu/hft^2) (Hansen et al. 1997). The shading coefficient of the external solar shading was 0.1 (blinds, 45° inclination, light colored) (Hansen et al. 1997).

For floor heating and cooling cases, the ventilation rate was 0.5 ach. For all cases, the infiltration rate was 0.2 ach. For floor heating cases, supply air temperature was 16.3°C (61.3°F) after the heat recovery and in warm-air heating case the supply air temperature was 35°C (95°F), limited by the building code in Denmark. In cooling cases, the intake air was 21.3°C (70.3°F) when taken from the crawl-space and this was the supply air temperature in floor cooling cases, supply air temperature was 14°C (57.2°F) for air cooling cases. Resulting space heating loads were 2180 W (32.9 W/m^2 [10.4 Btu/hft^2]) for floor heating cases, and 2048 W (30.9 W/m^2 [9.8 Btu/hft^2]) for warm-air heating case. Space cooling loads were 876 W (13.2 W/m^2 [4.2 Btu/hft^2]) for floor cooling cases, and 1042 W (15.7 W/m^2 [5.0 Btu/hft^2]) for air cooling cases.

Determination of Systems' Operation Parameters

The required floor surface temperature was 24.7°C (76.5°F) for floor heating cases corresponding to a specific heat output of 48.4 W/m^2-floor heating area (15.3 Btu/hft^2). The heat output and surface temperatures were calculated according to EN 1264-2 (2008) and Babiak et al. (2009). The mass flow rate was 469 kg/h (1034 lb/h) (EN 1264-3 2009). Supply and return water temperatures were 33°C (91.4°F) and 29°C (84.2°F), respectively, and corresponding heat pump COP was 2.63.

The heating rate necessary for bringing the air after the heat recovery device to the supply air temperature of 35°C (95°F) was 2559 W (8732 Btu/h). This heat was provided by the boiler and the supply and return water temperatures to the air-heating coil were 50°C (122°F) and 39°C (102.2°F), respectively (ASHRAE 2000). Corresponding supply air flow rate was 410 m^3/h (1.9 ach [241 cfm]), and water mass flow rate in the air-heating coil was 201 kg/h (443 lb/h).

For floor cooling cases, a floor surface temperature of 23.2°C (73.8°F) was required corresponding to 19.5 W/m2-cooled floor area (6.2 Btu/hft^2). The required supply and return water temperatures were 18.6°C (65.5°F) and 21.6°C (71°F), respectively, corresponding to a heat pump COP of 3.42. The required mass flow rate in the floor loops was 250 kg/h (551 lb/h).

The rate of heat to be removed from the intake air was 1389 W (4740 Btu/h) and 634 W (2163 Btu/h) for AC-OA and AC-CS, respectively. The required ventilation rate was 1.2 ach. The water flow rate in the air-cooling coil was 238 kg/h (525 lb/h) and 109 kg/h (240 lb/h) for AC-OA and AC-CS, respectively. The supply and return water temperatures to and from the air-cooling coil were 7°C (44.6°F) and 12°C (53.6°F), respectively, and the corresponding heat pump COP was 2.79.

In FC-GHEX, a single U-tube vertical heat exchanger was assumed to be coupled to the floor cooling system. 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 (47°F) (Kazanci et al. 2014). The incoming and outgoing liquid temperatures to and from the borehole were 17°C (62.6°F) and 13°C (55.4°F), respectively, with a mass flow rate of 208 kg/h (459 lb/h). Further details of this ground heat exchanger are given in (Skrupskelis and Kazanci 2012).

For the determination of the pump and fan powers, actual components installed in the house were used (Kazanci and Olesen 2014). Table 1 shows a summary of the pump and fan powers.

Table 1. Summary of Pump, Fan, and Total Power for Studied Cases

Case	E_{pump}, W (Btu/h)	E_{fans}, W (Btu/h)	E_{total}, W (Btu/h)
FH-B & FH-HP	27.5 (93.8)	67.9 (231.7)	95.4 (325.5)
WAH	25.0 (85.3)	273.0 (931.5)	298.0 (1017)
FC-HP & FC-GHEX*	25.2 (86.0)	67.9 (231.7)	93.1 (317.7)
AC-OA	25.0 (85.3)	173.6 (592.3)	198.6 (677.7)
AC-CS	23.0 (78.5)	173.6 (592.3)	196.6 (670.8)

*: 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 CALCULATION METHODOLOGY

Exergy performance of the systems was analyzed following the methodology described by Shukuya (2013). Only heating approach is described. Details of the cooling calculation methodology are given in Kazanci et al. (2016b).

In the most general form under steady-state conditions, exergy balance can be written as follows.

$$[Exergy\ input] - [Exergy\ consumed] = [Exergy\ output] \quad (1)$$

where [Exergy consumed] = [Entropy generated] · To. Heating exergy load, $X_{heating}$ [W], (Shukuya 1994) is defined as

$$X_{heating} = Q_{heating}\left(1 - \frac{T_o}{T_i}\right) \quad (2)$$

where $Q_{heating}$ is space heating load [W], T_o is outdoor (environmental) temperature [K] and T_i is indoor temperature [K]. Once the heating exergy load is known, the exergy supplied to the indoor space from floor heating, $X_{FH,out}$ [W], and through warm-air, $\Delta X_{WAH,out}$ [W], can be determined with Eqs. (3) and (4), respectively.

$$X_{FH,out} = Q_{heating}\left(1 - \frac{T_o}{T_{S,FH}}\right) \quad (3)$$

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

where $T_{S,FH}$ is average temperature of the heated floor surface [K], V_{sa} is volumetric flow rate of supply air [m^3/s], c_a is specific heat capacity of air [J/kgK], ρ_a is air density [kg/m^3] and T_{sa} is temperature of the supply air [K].

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

The exergy consumption in the floor structure, X_c [W], is obtained from the exergy balance as

$$\Delta X_W - X_c = X_{FH,out} \quad (5)$$

$$\Delta X_w = X_{w,supply} - X_{w,return} \quad (6)$$

where ΔX_w is the difference in the rate of exergy between the supply and return water flows [W]. The exergy of the supply and return water flows, X_w [W], are calculated as

$$X_w = V_w c_w \rho_w \left\{(T_w - T_o) - T_o \ln\frac{T_w}{T_o}\right\} \quad (7)$$

where V_w is volumetric flow rate of water [m^3/s], c_w is specific heat capacity of water [J/kgK], ρ_w is density of water [kg/m^3] and T_w is temperature of water [K]. Eq. (7) can also be used to calculate the exergy of the air flows by replacing the necessary flow, physical parameters, and temperatures of water with those of air.

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

$$\Delta X_{w,coil} - X_c = \Delta X_a \quad (8)$$

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

$$\Delta X_a = X_{a,out} - X_{a,in} \quad (10)$$

where ΔX_a is the difference in the rate of exergy between the air leaving ($X_{a,out}$) and the air entering the air-heating coil ($X_{a,in}$) [W].

It was assumed that the natural gas fired condensing boiler had an efficiency, η_{boiler}, of 90% (Shukuya 2013), (Kilkis 2012). The ratio of the chemical exergy to the higher heating value of natural gas, r, was taken as 0.93 (Shukuya 2013). The exergy input to the natural gas fired boiler, $X_{in,boiler}$ [W], is calculated using Eq. (11).

$$X_{in,boiler} = \frac{Q_{boiler}}{\eta_{boiler}} r \quad (11)$$

where Q_{boiler} is the amount of heat to be provided by the boiler [W].

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, η_{TOT}, was 0.35 (Shukuya 2013). The exergy input to the power plant by natural gas, $X_{in,power\ plant}$ [W], is calculated as follows.

$$E_{HP} = \frac{Q_{heating}}{COP} \quad (12)$$

$$X_{in,power\ plant} = \frac{E_{HP}}{\eta_{TOT}} r \quad (13)$$

Exergy input required at the power plant for the pump and fans is also calculated using Eq. (13) by replacing the electricity input to the heat pump (E_{HP}) with respective pump power (E_{pump}) and fan power (E_{fans}). E_{HP} is replaced with E_{total} in Eq. (13) to calculate the necessary exergy input to the power plant for auxiliary components ($X_{in,pp,aux}$, [W]).

RESULTS AND DISCUSSION

Thermal Exergy Flows

Figure 2 shows the whole chains of exergy flows for heating and cooling cases.

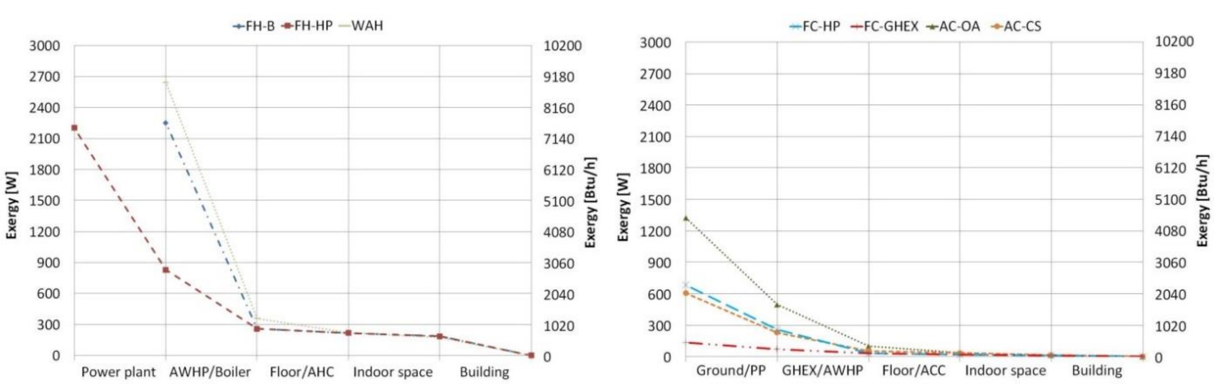

Figure 2 Whole chains of exergy flows for heating (left) and cooling (right) cases (AWHP: air-to-water heat pump, AHC: air-heating coil, PP: power plant, GHEX: ground heat exchanger, ACC: air-cooling coil)

The results show that even though warm-air heating has a slightly lower heating exergy load (Table 2), floor heating system requires 15% smaller exergy input when coupled to a boiler and 17% smaller exergy input when coupled to a heat pump, compared to the warm-air heating system (values were obtained by using the exergy input to the boiler, $X_{in,boiler}$, and to the power plant, $X_{in,power\ plant}$, with the values given in Figure 2). This is caused by the difference in the required temperature values, and, thus, different exergy inputs and consumptions in the system components. The heating results also show that the exergy performance of the whole system can be further improved by better matching the low-exergy demand of the low temperature floor heating system.

The results of the cooling season analyses in Figure 2 show that when the intake air to the air handling unit is outdoor air, a large amount of exergy is needed (more than double of that when the CS is used) to cool this air to the supply air temperature. A previous study showed that a floor cooling system coupled to an air-to-water heat pump would require 28% smaller exergy input to the power plant than AC-OA (Kazanci et al. 2016b).

Figure 2 shows that AC-CS requires 76 W (259 Btu/h) less exergy input to the power plant than FC-HP but requires 275 W (938 Btu/h) more exergy input to the power plant for the auxiliary components (Figure 3), and has larger exergy consumption than FC-HP in the rest of the system components. The performance of the floor cooling system could be improved by having higher conversion efficiency in the power plant, using on-site generated renewable electricity or increasing COP through a better heat pump, but it is also possible to use another cooling source than a heat pump and this was studied in FC-GHEX case.

It is possible to significantly reduce the overall exergy input and consumption by coupling the high temperature floor cooling system with a GHEX; the overall exergy consumption can be reduced by 78% compared to AC-CS and 80% compared to FC-HP, when both the exergy consumed in the ground and the brine pump are taken into account.

The results of the cases, in which the cooler air from the crawl-space and a GHEX were used, emphasize the advantages and importance of integrating naturally available heat sinks (and sources) into the heating and cooling systems in buildings and the low temperature heating-high temperature cooling radiant systems make this possible.

Exergy Input to Auxiliary Components

Figure 3 shows the pump power, fan power, their total, and the necessary exergy input to the power plant for auxiliary components.

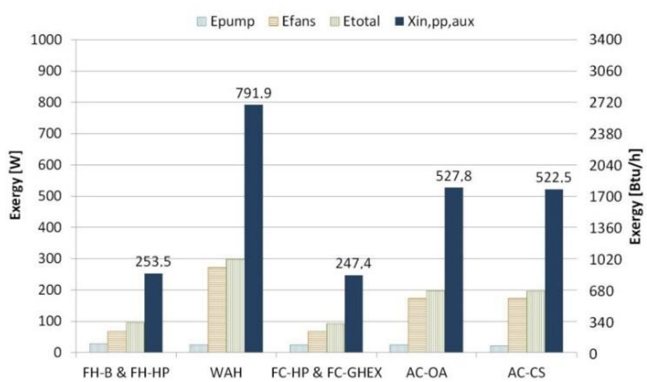

Figure 3 Exergy input to the auxiliary components and to the power plant

Water-based systems require significantly less exergy input to the auxiliary components; 68% less in heating and 53% less in cooling compared to air-based systems. This difference is mainly because, in floor heating and cooling systems, the heat emission system is water-based and the ventilation system is only used to provide the necessary amount of fresh air, on the other hand, in air heating and cooling systems, high air-flow rates are needed to address the heating and cooling loads. These system behaviors indicate a clear benefit for water-based systems over the air-based systems.

Table 2 shows the comparison of the energy and exergy use for auxiliary components and the space heating and cooling.

Table 2. Comparison of Energy and Exergy Use for Auxiliary Components and Space Conditioning*

Case	FH-B	FH-HP	WAH	FC-HP	FC-GHEX	AC-OA	AC-CS
$Q_{heating(cooling)}$, W (Btu/h)	2180 (7439)	2180 (7439)	2048 (6988)	876 (2989)	876 (2989)	1042 (3556)	1042 (3556)
$X_{heating(cooling)}$, W (Btu/h)	186 (635)	186 (635)	175 (597)	12 (41)	12 (41)	14 (48)	14 (48)
$E_{pump}/Q_{heating(cooling)}$, %	1	1	1	3	3	2	2
$E_{pump}/X_{heating(cooling)}$, %	15	15	14	215	215	180	165
$E_{fans}/Q_{heating(cooling)}$, %	3	3	13	8	8	17	17
$E_{fans}/X_{heating(cooling)}$, %	37	37	156	580	580	1246	1246
$E_{total}/Q_{heating(cooling)}$, %	4	4	15	11	11	19	19
$E_{total}/X_{heating(cooling)}$, %	51	51	171	795	795	1426	1411
$X_{in,pp,aux}/X_{in,boiler}$, %	11	-	30	-	-	-	-
$X_{in,pp,aux}/X_{in,power\ plant}$, %	-	12	-	36	184	40	87

*: E_{pump}, E_{fans} and E_{total} are given in Table 1. $X_{in,pp,aux}$ for different cases is given in Figure 3. $X_{in,boiler}$ and $X_{in,power\ plant}$ for different cases are given in Figure 2.

Results in Table 2 show that the pump powers seem negligible in energy terms when compared to heating and cooling energy loads, but in exergy terms they become comparable, and this is more important when the space heating and cooling exergy load is small, as in cooling cases. Fan powers are significant in energy terms when compared to space heating and cooling loads particularly in air-based systems and this effect can be seen more clearly in exergy terms.

The relative significance of auxiliary components is more evident in air-based heating and cooling systems compared to the water-based systems, though it is also important for water-based systems to minimize the pump energy use. The required exergy input to the power plant for auxiliary components is comparable to the exergy input to the boiler and to the power plant for the heat pump. When naturally available heat sinks were used (GHEX and CS) this effect became more pronounced as in FC-GHEX and AC-CS, and this clearly shows that in order to benefit truly from the "free" cooling opportunities, the natural resources should be used wisely and the auxiliary energy input to utilize these sources should be kept to a minimum.

It should be noted that optimizing a system by lowering the pressure drops (e.g. for floor systems having more loops or a larger pipe diameter) will increase the initial costs, and this issue should also be considered carefully.

CONCLUSION

Energy and exergy performances of a floor heating and cooling system were compared to an air heating and cooling system using a single-family house as a case study. The effects of different heat sources and sinks on the system performance and the exergy use of auxiliary components were also compared.

Water-based floor heating and cooling systems can address the same heating and cooling loads with smaller exergy consumption than the air-based systems. The floor heating system required 15% and 17% smaller exergy input compared to the warm-air heating, when coupled to a boiler and to a heat pump, respectively. By using a floor cooling system coupled to a ground heat exchanger, the overall exergy consumption can be reduced by 78% compared to an air cooling system and 80% compared to a floor cooling system coupled to an air-to-water heat pump.

Water-based systems required significantly less exergy input to the auxiliary components compared to air-based systems; 68% less in heating and 53% less in cooling operation.

Pump and fan powers might not seem significant when compared to space heating and cooling loads in energy terms, but they become comparable when considered in exergy terms. Their importance becomes even more evident when the actual space heating and cooling loads are small, and the required exergy input to the power plant for space heating and cooling purposes become comparable to the required exergy input for auxiliary components.

Using naturally available heat sources and sinks can bring significant exergy input reductions in the supply side. In order to truly benefit from this potential, the energy and exergy used by pumps to extract heat from or reject heat to these sources and sinks should be minimized.

For the optimal design of a heating and cooling system, all three parts of the system (generation, distribution, and emission) should be considered as a whole. Exergy approach, together with energy, can clearly show the improvement possibilities and can be used to optimize the system performance.

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