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Optimal dimensioning power of GSHP with district heating in an educational building

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Abstract. European Union (EU) have set great goals to reduce carbon dioxide (CO_2) emissions and mitigate global warming trend. In this context, energy performance of buildings should be improved by enhancing the use of renewable energy sources in heating and cooling. For that reason, a hybrid energy system where a ground source heat pump (GSHP) integrated with borehole thermal energy storage (BTES) system is used together with district heating (DH) has become increasingly popular in Finland during the last years. In those hybrid GSHP systems, GSHP is used as a primary energy source and DH is used for supplementing energy during peak heating load period in wintertime. While in summertime, the borehole field is used for free cooling and DH is providing heating energy for domestic hot water. In this study, a large educational building complex in Finland applying a hybrid energy system consisting of a GSHP and DH was modelled in IDA ICE 4.8. Three simulation cases were studied to analyze the effects of the GSHP power ratio on the whole system energy consumption and CO_2 emissions. The results show the total CO_2 emission obtained the minimum when the GSHP heating power ratio was 50%. However, compared to 100% district heating solution, 25% power ratio of GSHP in the hybrid system can already realize 50% reduction of CO_2 emissions as the total cooling demand was fully satisfied by borehole free cooling and 98% DH consumption was reduced.

Keywords. Ground source heat pump, district heating, dimensioning power.
DOI: https://doi.org/10.34641/clima.2022.115

1 Introduction

Currently, European Union (EU) has set two stage targets to diminish carbon dioxide (CO_2) emissions and mitigate the global warming trend: to reduce emissions by 40% by 2030, compared to the 1990 level [1], and to achieve climate neutrality by 2050 [2]. In the EU countries, the building sector accounts for around 40% of the energy consumption and consequent 36% of the CO_2 emissions [3]. Energy performance of new and old buildings should be improved by increasing utilization of renewable energy techniques in the building energy system.

One of the prevalent renewable energy techniques is a ground source heat pump (GSHP) coupled with a borehole thermal energy storage (BTES) system. The hybrid GSHP system where the GSHP integrated with district heating (DH) as a backup heating source has become increasingly popular in Finland during the last years. Hybrid GSHP systems are helpful in several applications including design strategy of demand side management [4] and mitigation of underground thermal imbalance [5]. In the design of the hybrid GSHP system, the GSHP power ratio is an important parameter which needs to be determined properly. However, based on the author's best knowledge, the optimization of the dimensioning power of GSHP based on energy consumption and CO_2 emissions in a hybrid GSHP system with auxiliary DH was not well investigated.

This study aims to investigate the effects of the GSHP power ratio of a hybrid GSHP system connected with DH on building energy consumption and CO_2 emissions of an educational building complex in Finland. The energy simulations of three studied cases with different GSHP power ratio were performed in IDA ICE 4.8. The electricity and DH energy consumptions and the corresponding CO_2 emissions were analyzed for the studied cases. The GSHP power ratio for the optimal CO_2 reduction was suggested.

2 Method

2.1 Building description

The studied building is a large educational building located in Aalto University Campus in Espoo, Finland (shown in Fig. 1). The building was completed by September 2018 and was open to the public since January 2019. The building is a 4/5 storey building with multiple types of spaces including educational area, office, restaurants, gym, workshop, computer rooms, shopping area and a metro station. The heated net floor area of the building is 39670 m^2. Currently, the building is equipped with a ground source heat pump system and connected to a district heating network. The ground source heat pump is coupled with a borehole field containing 74 boreholes with average depth of 310 m.

Fig. 1 - Studied building in Aalto University Campus, Espoo, Finland.

In this study, the building geometry was simplified as a rectangular single storey building model (shown in Fig. 2(a)), which had been calibrated to describe the energy consumption of the building in the previous study [6]. The building space was divided into five zones with room height of 4.6m (shown in Fig. 2(b)). The eventual geometry of each zone was magnified by the zone multiplier of 15.3 to obtain the real net floor area of 39670 m^2.

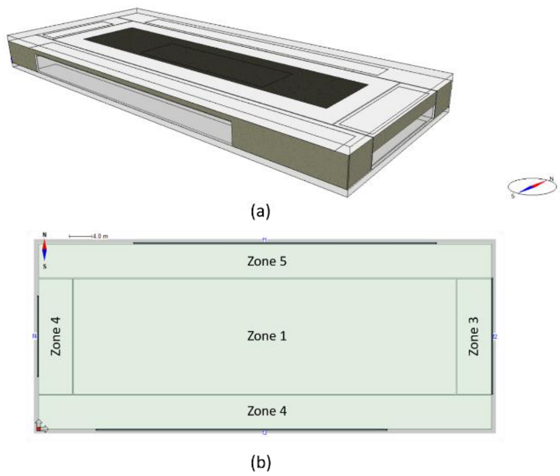

Fig. 2 - The geometry of the building model.

The U-values of the external wall, the roof and the base floor are 0.17 W/m^{2}K, 0.09 W/m^{2}K and 0.18 W/m^{2}K respectively. Each external wall has one window with the U-value of 0.6 W/m^{2}K, solar heat transmittance (g-value) of 0.49 and direct solar transmittance (ST) of 0.41. The windows are equipped with solar shading by blinds between outer panes, which are drawn down when solar radiation level is above 100W/m^2. Moreover, the windows are set to be always closed. The average infiltration air flow rate of the building is 0.045 m^3/hm^2.

The main internal heat gains of the building are from occupants, lighting, and equipment [7]. The internal heat gains used in this study are listed in Table 1. The average occupancy profile is shown in Fig. 3. The average lighting and equipment usage profiles are both set the same as the average occupancy profile.

Tab. 1- Internal heat gains.

Internal heat gains from occupants, lighting and equipment	
Occupants	7378 occupants in the building, which is equal to average occupancy density 0.186 1/m^2 with activity level of 1 met, clothing level of 0.85 ± 0.25 clo
Lighting	Average gain 8.7 W/m^2, internal gain from lighting equals to 16.7 kWh/(m^2, a)
Equipment	Average gain 5.1 W/m^2, internal gain from equipment equals to 11.2 kWh/(m^2, a)

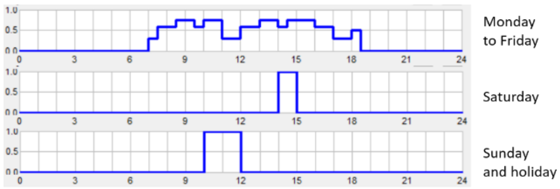

Fig. 3 - The average occupancy profiles. The hours of the day are shown on the horizontal axis and the occupancy is shown on the vertical axis (0 = 0% occupancy, 1 = 100% occupancy).

2.2 Description of heating, cooling and ventilation systems

In order to investigate the effects of different share of GSHP in the total heat power demand of the building, Case 1 using DH for heating and electric air-cooled chiller for cooling is defined as the reference case in this study. The DH substation annual efficiency is 97%. The coefficient of performance (COP) of the air-cooled chiller is 3.0 at the rating conditions (35/7°C). A simplified schematic of the plant model and connections to heating and cooling networks is shown in Fig. 4.

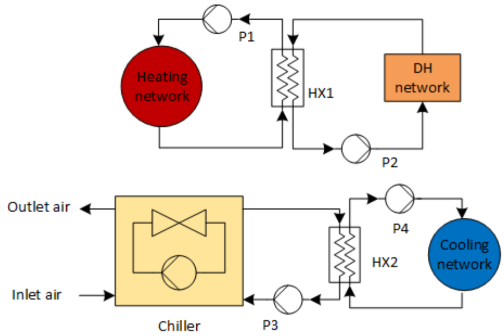

Fig. 4 - Simplified schematic of the plant model and connection in the reference case (Case 1).

In the reference case, the heating set-point temperature is set as 21°C. The heating is delivered through three different circuits: space heating, air handling unit (AHU) heating and domestic hot water (DHW). The space heating is carried out by a hydronic radiator heating system with dimensioning temperatures of 50/35°C. For the space heating, the supply water temperature is controlled according to the control curve shown in Fig. 5. For the inlet water of a reheat coil of the air handling unit, the constant temperature set point of 60 °C is implemented.

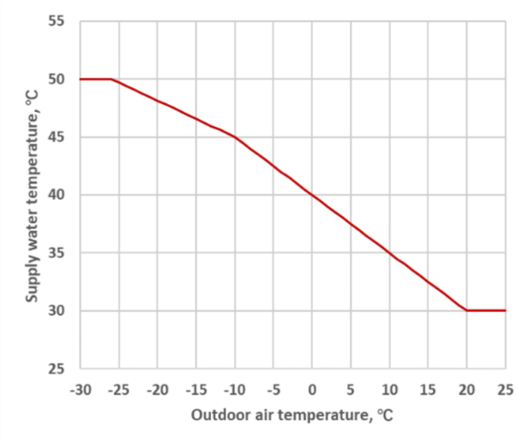

Fig. 5 - Control curve of supply water temperature of radiant panels.

The domestic hot water (DHW) consumption is defined by two main features, the level of consumption and the consumption profile. The annual heat energy consumption level of DHW is 11 kWh/(m², a). The daily DHW consumption profile is shown in Fig. 6. The supply water temperature of the DHW is 55 °C.

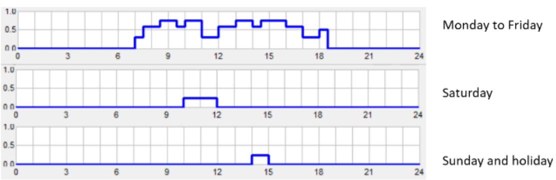

Fig. 6 - The DHW consumption profile. The hours of the day are shown on the horizontal axis and the usage rate is shown on the vertical axis (0 = 0% usage rate, 1 = 100% usage rate).

The cooling is distributed through space cooling and AHU cooling in the building. The space cooling is carried out by a hydronic cooling panel system with the supply water temperature of 15°C. The set-point temperature of space cooling is 25°C. The AHU cooling supply water temperature is set as 5°C constantly.

The building has a mechanical balanced ventilation system with heat recovery. The heat recovery is applied in the AHU with heat exchanger efficiency of 73% on supply air side. In addition, the AHU is equipped with defrost protection on the heat exchanger. The defrost protection is controlled according to the exhaust air temperature after the heat exchanger. When the exhaust air temperature falls below -5°C, the supply air bypasses the heat exchanger.

The ventilation system is a constant air volume system with schedule control, while night ventilation is not applied in this study. The constant supply and exhaust air flow rates are both defined as 2.3 L/(s, m²) during occupied time and 0.76 L/(s, m²) during unoccupied time. The fan operation rates are controlled according to fan operation schedule (shown in Fig. 7). The supply air temperature is controlled by the return air temperature according to the curve shown in Fig. 8.

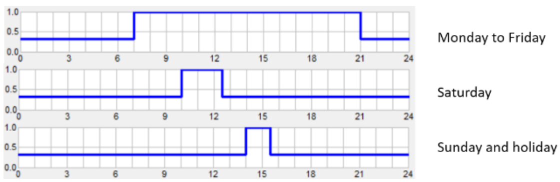

Fig. 7 - The fan operation schedules. The hours of the day are shown on the x-axis and the operation rate of ventilation system is shown on the y-axis (0 = 0% fan operation rate, 1 = 100% fan operation rate).

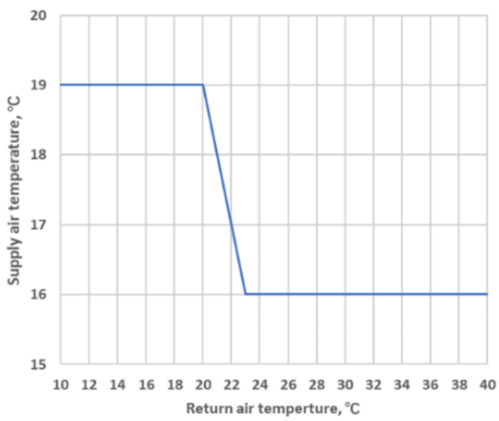

Fig. 8 - Control curve of supply air temperature of AHU.

2.3 Definition of simulated cases

Apart from the reference case (Case 1), two studied cases applying different GSHP power ratios are simulated to investigate the effects of the GSHP power ratio on the whole energy system performance. In Case 1, the total heating power is covered by the DH, and the total cooling power is provided by the electric air-cooled chiller. In Cases 2-3, a certain part of the total heating power demand of the building is covered by the GSHP, and the rest of the heating power is provided by the DH. Besides, the total cooling power in Cases 2-3 is provided by free cooling from the BTES instead of electric air-cooled chiller. The detailed definitions of these three cases are listed in Table 2. The simulation period of all three cases is year 2019.

Tab. 2 - Definitions of the three simulated cases.

Case	GSHP power ratio	DH power ratio
Case 1	0	100%
Case 2	25%	75%
Case 3	50%	50%

In Cases 2-3, the COP of the simulated GSHP is assumed to be 4.3 at the rating conditions (0/35°C). The borehole numbers are calculated by applying the same ratio of the total maximum heating power of GSHPs and the total length of boreholes (32.2 W/m) in all studied cases as in real the building. The borehole depths are all the same as that in real boreholes (320m).

The GSHP system is used in both heating and cooling seasons. In heating season, the GSHP satisfies the basic heating demand. When the GSHP is unable to produce enough heat for the building, the DH is used as an auxiliary heat source. The condenser side of the GHSP and the DH are both connected to a heating network via a 5m^3 hot water storage tank. In cooling season, the cooling is produced by free cooling from the borehole field. The borehole field is connected to the cooling network via a 3m^3 cold water storage tank.

Fig. 9. shows the simplified schematic of the plant model and connection of heating and cooling networks in Cases 2-3. In heating mode, the GSHP serves as the primary heating source for the heating and DHW generation. The pump P5 is off work during heating season. The GSHP and the pumps P1 and P2 receive permission to start up or shut down according to a temperature set point of the hot water storage tank (HT). The hot water storage tank is used as a buffer tank for storing the hot water generated by GSHP and mitigating temperature variation. If the GSHP cannot produce enough heating energy, the pump P3 will start up and the DH will be used to heat up the water in the hot water storage tank. In cooling mode, the DH is not used as there are not much heating demands in the building. The pump P3 is off work during heating season. The DHW is produced only by the GSHP. Besides, the pump P5 starts up and the BTES produces all cooling energy. The cold water storage tank (CT) serves as a buffer tank to mitigate the temperature variation of the cooling water from the BTES.

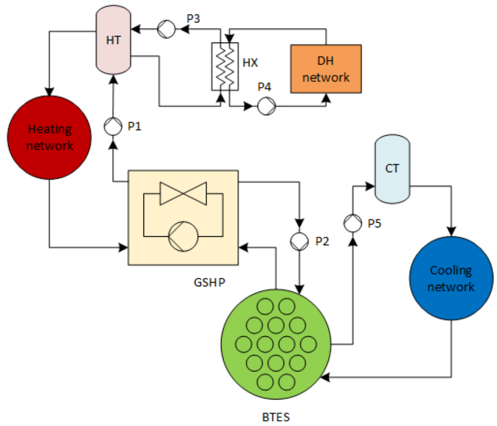

Fig. 9 - Simplified schematic of the plant model and connection in Cases 2-3.

In Cases 2-3, the heat distribution system consists of space heating, AHU heating and DHW in which the parameters are set as the same to those in Case 1. The cool distribution system consists of AHU and space cooling which both adopt the inlet cooling water temperature of 15°C. In the AHU, the liquid side temperature rise of the cooling coil is changed from 5°C to 1°C to simulate a cooling coil with a larger heat transfer area than the default cooling coil used in Case 1. The other parameters of the AHU cooling and space cooling circuits are set as the same to those in Case 1.

2.4 *Weather data and simulation tool*

According to the classification of Finnish climate zones, the studied building in Espoo is located in climate zone 1, which is the southernmost climate zone in Finland [8].

In this study, the dimensioning heating demand of the building is simulated without internal heat gains under the dimensioning outdoor temperature (-26°C) of southern Finland. The dimensioning cooling demand of the building is simulated with 100% internal heat gains using synthetic weather data of ASHRAE. The design day is chosen according to cumulative frequency of 1 % and the maximum dry-bulb and wet-bulb temperatures are 25.8°C and 17.6°C respectively.

The annual energy simulation uses hourly weather data from Helsinki-Vantaa test reference year (TRY2012). The TRY is developed according to the weather data observations of 30 years (1980-2009) measured by the weather station of the Finnish Meteorological Institute at the Helsinki-Vantaa airport [9]. The annual average temperature in Helsinki-Vantaa region is +5.4 °C. The average number of degree days is 3952Kd under the indoor temperature of 17 °C.

The annual CO_2 emissions due to energy use are calculated by using average Finnish emission factors for electricity and district heating presented in Hirvonen et al. [10]. The CO_2 emissions factor for electricity and district heating are 96 kgCO_2/MWh and 137 $kgCO_2/MWh$ respectively.

The simulation work of this study is performed by IDA Indoor Climate and Energy (IDA ICE) 4.8 software [11]. The software has been chosen as it is long-developed and reliably applicable in simulating building energy consumption, plant performance, indoor air quality and indoor thermal comfort. It has a detailed and dynamic multi-zone simulation application with variable time step. The IDA ICE has been validated in several studies [12,13], which provides sufficient reasons for using this tool in this study.

3 Results

The results of the power simulations show the maximum heating and cooling demands of the simulated building are 2978 kW and 1685 kW respectively in the design outdoor conditions. Based on these, the dimensioning heating and cooling power are rounded up to the next nearest 100 kW. Therefore, the total heating power is 3000 kW, and the total cooling power is 1700 kW. According to the definition of Cases 1-3, the dimensioning GSHP heating power, DH power and the number of boreholes are listed in Table 3.

Tab. 3 - Dimensioning GSHP heating power, DH power and the number of boreholes in studied cases.

Case	GSHP heating power (kW)	DH power (kW)	Number of boreholes
Case 1	0	3000	0
Case 2	750	2250	70
Case 3	1500	1500	140

Fig. 10 shows the district heating power duration curve for Cases 1-3. Compared to Case 1, the peak load DH power was reduced half in Case 2. However, the total DH duration time also decreased significantly from 6400 h to 580 h. In Case 3 where the GSHP power ratio increasing to 50%, the peak load DH power was decreased to 160 kW, and the total DH duration time was reduced to only 10 h.

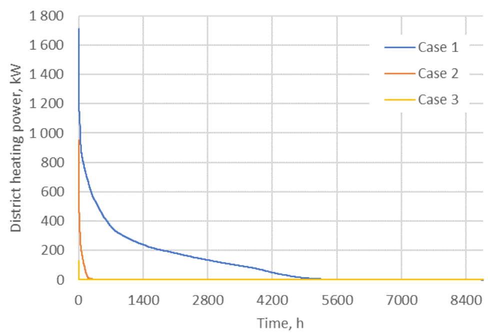

Fig. 10 – District heating power duration curves.

Fig. 11 shows the GSHP heating power duration curve for Cases 2 and 3. In Cases 2 and 3, the peak load GSHP power both reach their dimensioning GSHP heating power. However, from Case 2 to Case 3, the peak load duration time was reduced from 360 h to 8 h while the total GSHP power duration time was the same in these two cases.

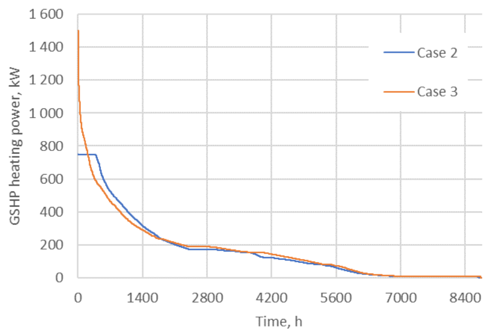

Fig. 11 – GSHP heating power duration curves.

Fig. 12 shows the cooling power duration curve for Cases 1-3. Compared to Case 1, the peak load cooling power and the total cooling duration time were slightly reduced in Cases 2. In Cases 3, the cooling power duration curve almost overlaps that of Case 2, which indicates the borehole free cooling totally satisfied the cooling demand in these two cases.

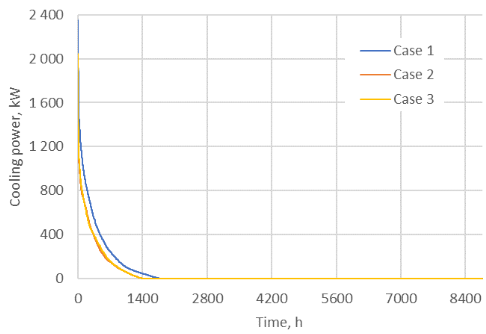

Fig. 12 – Borehole free cooling power duration curves.

Table 4 shows the annual heating and cooling energies of three simulated cases. The result shows the changes of GSHP power ratio slightly affected the produced heating and cooling energy. The difference of heating and cooling energy between the GSHP cases (Cases 2-3) and the reference case (Case 1) can be attributed to the differences of temperature levels.

Table 5 shows the purchased energies of three simulated cases. Although there was no cooling electricity consumption in Cases 2-3, meaning the borehole free cooling satisfied the total cooling demand of the building, the annual total electricity consumptions in Cases 2-3 were still more than that in Case 1. Compared to Case 1, the annual total electricity consumption in Case 2 was 9% higher, as the electricity used for the GSHP and pumps in heating season was more than that used for chiller and pumps in cooling season. Besides, from Case 2 to

Tab. 4 - Breakdown of annual heating and cooling energy for Case 1 (ref), Case 2 (25% of heat covered by GSHP), Case 3 (50% of heat covered by GSHP).

Case	Heating and cooling (kWh/m²,a)							Comparison	
	Space heating	Space cooling	AHU heating	AHU cooling	DHW	Total Cooling	Total Heating	Relative difference of total cooling	Relative difference of total heating
Case 1(ref)	10.8	4.3	14.6	9.3	13.1	13.6	38.6	-	-
Case 2	10.7	4.7	14.6	6.3	13.0	11.0	38.3	-19 %	-0.6 %
Case 3	10.7	4.4	14.6	6.9	13.1	11.3	38.4	-17 %	-0.3 %

Tab. 5 - Breakdown of purchased energies for Case 1 (ref), Case 2 (25% of heat covered by GSHP), Case 3 (50% of heat covered by GSHP).

Case	Electricity (kWh/m²,a)							DH (kWh /m²,a)	Comparison	
	Light-ing	Equip-ment	Electric cooling	GSHP heatin g	Fans	Pumps	Total electr-icity	Total DH	Relative difference of total electricity	Relative differenc e of total DH
Case 1 (ref)	16.7	11.2	4.5	0	15.2	0.05	47.7	39.8	-	-
Case 2	16.7	11.2	0	8.9	15.2	0.15	52.2	0.9	9 %	-98 %
Case 3	16.7	11.2	0	9.4	15.2	0.14	52.7	0	11 %	-100 %

Tab. 6 - Annual CO_2 emissions of Case 1 (ref), Case 2 (25% of heat covered by GSHP), Case 3 (50% of heat covered by GSHP).

Case	CO_2 emissions (ton/a)			Comparison
	Electricity CO_2	DH CO_2	Total CO_2	Relative difference of total CO_2 emissions (%)
Case 1 (ref)	182	216	398	-
Case 2	199	5.0	204	-50 %
Case 3	201	0.1	201	-50 %

Case 3, as the GSHP heating power ratio increases, the electricity used for GSHP heating increased, which mainly contributed to the increase in the total electricity consumption. On the other side, the DH consumption in Case 2 decreased by 98% compared to Case 1. The DH consumption in Case 2 was only 0.9 kWh/(m², a). In Case 3 where the GSHP power ratio increased to 50%, there was even no DH consumption, which indicates the backup heating could be not needed anymore.

Table 6 shows the annual CO_2 emissions of the three simulated cases. With the increase of GSHP heating power ratio, the CO_2 emissions of electricity consumption increased, and the CO_2 emissions of DH consumption decreased. Compared to Case 1, the CO_2 emissions of electricity consumption increased significantly in Case 2. However, as the GSHP ratio increased evenly from Case 2 to Case 3, the CO_2 emissions of electricity consumption changed marginally. On the other side, the CO_2 emissions of DH consumption had a dramatical decrease in Case 2 compared to Case 1. From Case 2 to Case 3, the CO_2 emissions of DH consumption were almost eliminated. The total CO_2 emissions in Cases 2 and 3 both realized 50% reduction. However, from the economical point of view, the 25% GSHP power ratio would be more appropriate for the system design, as the investment could be less on the borehole field.

4 Conclusions

This study investigated the effects of the GSHP power ratio on the total energy consumption and CO_2 emissions of a hybrid GSHP system of an educational building in Finland. The simulation of three studied cases were performed in IDA ICE 4.8. The summary of the results of simulation is listed as following.

- Compared to 100% DH heating, the 25%

GSHP power ratio increase the heating electricity consumption insignificantly, but it can reduce the DH energy consumption by 98%. Besides, the 25% GSHP power ratio can totally satisfy the cooling demand of the building by borehole free cooling. Based on these, the 25% GSHP power ratio can reduce 50% CO_2 emissions.

- As the GSHP power ratio increased more than 25%, the CO_2 emissions remained almost unchanged as the GSHP could meet the total heating demand without using the DH as a backup heat source. Based on economic considerations, the 25% GSHP power ratio was sufficient for the hybrid GSHP system. This result provides a useful reference for designs of hybrid GSHP system.

Acknowledgement

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