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Integration of a heat exchanger on the supply air with the ground-source heat pump in a passive house – case study

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Abstract. A ground-source heat pump (GSHP) system designed to cooperate with a low temperature heating system and hot water production was installed in a new passive house nearby the city of Poznan (Poland). The paper describes an integration of the GSHP system with a heat exchanger on the supply airflow in order to recover the waste heat from outside (fresh) ventilation air and store it in the ground during summer, as well as to heat the supply air in winter what prevents the recuperator from freezing. Analytical calculations to evaluate the energy, economy and environmental benefits of supplying the low heat reservoir with heat from fresh air were made. Chilled supply air compensates the heat gains from building and is exhausted, simultaneously overcoming a part of the cooling loads by using mechanical ventilation system. Heat taken in the heat exchanger by the glycol mixture is used in the evaporator of the heat pump, which produces hot water with increased energy efficiency – increased coefficient of performance (COP). After that glycol mixture is supplied to the ground. In the case of stagnation of the heat pump, glycol mixture exiting the heat exchanger is directed to the ground, where part of the delivered waste heat is stored. The results of calculation experiment indicated that heat recovered from the supply ventilation air during summer is increasing the Seasonal Performance Factor (SPF) of the heating system in winter, consequently significant energy savings can be achieved, what is a part of the idea of the Sustainable Development. Based on the simulations it has also been demonstrated that the proposed technical solution has a high application potential for micro and macro scale installations using GSHP.

1. Introduction

One of the most important challenges that the 21. Century has to take is to stop the Global Warming, that is mainly caused by the rapid CO_2 emission. According to the International Energy Agency the global average building energy usage per person must decrease by 10% in the next 8 years to meet the assumptions of the 2°C Scenario (2DS), which is still less restrictive as the Under 2°C Scenario accepted while the Climate Agreement in Paris [1]. The heating and hot water production demands are responsible for approximately 80% of building stock final energy use and unfortunately are mostly covered by energy generated by burning fossil fuels [1, 2].

Ground-source heat pump (GSHP) system for heating and cooling is an energy efficient and innovative technology, that use the ground as a low heat source or low-heat reservoir. GSHP is able to produce heat with the Coefficient of Performance (COP) usually in the range between 3 and 3.8 [2] but often the COP is higher – 3.6÷4.4 [3]. GSHP are recommended to integrate with low-temperature heating systems.

However in recent years in many papers the decrease of the seasonal heat pumps efficiency was reported [3, 4]. K. Allaerts et al. [4] reports that a GSHP in heating dominated school building in Belgium would cause a drop in the average ground temperature from 12.3°C at the start of GSHP operation, to 5.5°C after 15 years. The improvement of the energy efficiency of GSHP system is necessary and it can be achieved by supplying the ground with waste energy from extracted air with satisfying effect. S. Zhou et al. [5] refers that also in cooling dominated areas the effectiveness of the GSHP decrease with time, because the soil temperature is rising while the heat pump works in the cooling mode. Integration of a cooling tower with the GSHP system can withstand those changes and ensure a long term efficiency of the system. It can be concluded that the main cause of the drop in efficiency is the use of the ground as a heat source, when in reality it is more like a heat storage, so any GSHP system should be designed considering the imbalance of cooling and heating loads and in case of significant differences - technical solutions should be undertaken to ensure constant performance of the GSHP in the lifetime cycle.

The examples cited are only a few of many possible technical solutions to maintain the advantages of the GSHP systems, but many of them are expensive ways, that reduces the benefits of using a heat pump. Hence it is important to develop solutions that will be such effective as cheap.

2. The hybrid ground source heat pump system (HGSHP)

In this paper the performance evaluation of a GSHP system combined with low temperature heating and hot water production in a passive house nearby city of Poznan (Poland) is described. A low-cost measure which improves the energy efficiency of GSHP system is taken under investigation.

2.1. Building description

The considered residential passive building is a single-family two-storey house, which is designed for 4 persons. It is located in the 2nd climate zone according to the Polish Standard PN-76/B-03420. The heating consumption coefficient doesn't exceed 15 kWh/(m^2·a) and is fully covered by using renewable energy sources.

Building description:

- total usefull floor area with adjustable temperature/total usable volume $A_G = 148.8m^2/V_G = 361.9m^3$
- energy demands for heating and ventilation 1 632.4 kWh/a
- specific energy demand for heating and ventilation 10.97 kWh/(m^2·a)
- designed demand of heat generation power for heating and ventilation $Q_{C.O.} = 3.1$ kW
- designed demand of heat generation power for hot water production $Q_{C.W.U.} = 3.5$ kW
- air tightness of the building $n_{50} = 0.6\ h^{-1}$
- heat transfer coefficient of the external walls / of the roof $U_w = 0.15$ W/(m^2·K) / $U_r = 0.08$ W/(m^2·K)
- high insulated windows capable to gain solar energy in the heating season with external blinds, to protect from overheating in cooling season: U= 0.8 W/(m^2K), U_g= 0.5 W/(m^2K), g_{win}= 0.7, g_{sum}= 0.35

2.2. HVAC system

2.2.1. Heating system. The heating system of the building consists of a ground source heat pump of heating capacity Q_g equal to 3.5 kW (B0/W35) and the COP is 4.2 (B0/W35). As the thermofluid distribution medium within the heat pump the refrigerant R407C is used. The GSHP system cooperate with a floor heating system. The heating water temperatures for designed conditions are t_{sup}/t_{ret} = 30/25°C. The heat pump is integrated with a hot water storage tank of a volume of 200 dm^3. The existing heating system works with the priority of domestic hot water production. As a supporting measure, the heat pump is equipped with an electric heater to support the water heating during peak hours.

Characteristics of heat power, energy consumption and effectiveness of energy processing (COP) of the heat pump for different supply water temperatures $t_{sup\ Heat}$ = 30°C and $t_{sup\ HW}$ = 50°C depending on the low-heat reservoir temperature are shown in Fig.1. The heat pump works in the first mode while overcoming heat losses, whereas to hot water production the parameters are higher.

For the purpose of calculations, it was assumed that:
- The supply water temperature $t_{sup\ Heat}$ in the heating system is constant during the year and it is equal to 30°C,
- The daily hot water demand doesn't exceed 35 dm^3/person.
- The hot water is heated from the temperature of 10°C to the desirable temperature t_{HW}= 45°C,
- A daily heat dissipation of 0.6 kWh/d while heat distribution and 1.8 kWh/d, because of the idle time, is assumed,
- There is no circulation of hot water,
- Heat dissipation from the pipelines in the low-heat reservoir was not considered,
- Heat gains by-produced by the circulation pumps and they influence of glycol mixture heating have been omitted,
- The overcooling of the refrigerant in the evaporator was assumed on the value of 3K and the overheating in the condenser was set as 5K.

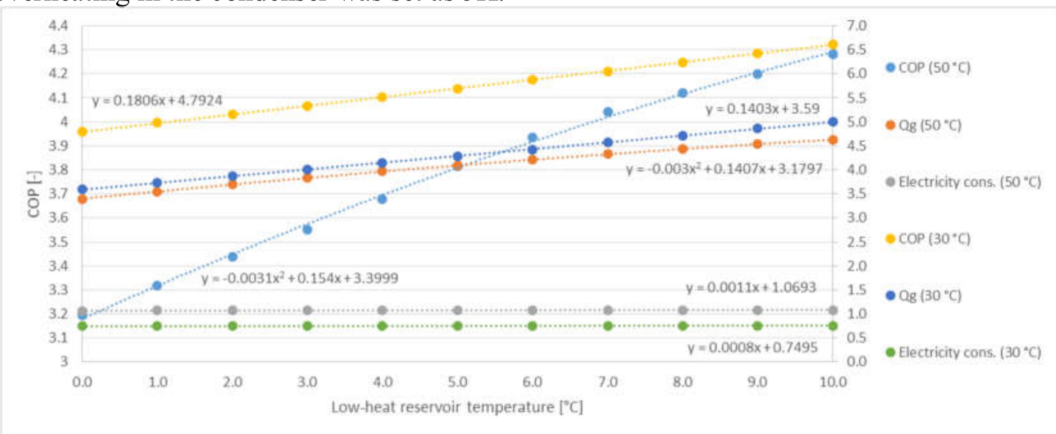

Figure 1. Parameters of the heat pump in the relationship on the temperature of the low-heat reservoir

2.2.2. Low-heat reservoir. The low heat-reservoir for the heat pump is a horizontal ground heat exchanger (HGHE) consisting of two loops – every loop is 100 m long – connected with a collector. The depth of the HGHE is 2.0 m below the ground level. As the thermofluid distribution medium is used a 35% ethylene glycol mixture. The pipeline was situated on a natural soil with sand priming and filling and covered by the original soil with extreme caution.

Other characteristics of the low-heat reservoir and other important assumptions:

- Soil	wet sand (geotechnical research)
- Soil density / soil conductivity	ρ_s = 2166 kg/m^3 / λ_s = 2.1 W/(m·K)
- Depth of groundwater	1.5 m below the ground level
- Constant volume flow of the glycol mixture	$\dot{V}_g$ = 0.35 dm^3/s
- The internal dimension / wall thickness of the pipe and the pipe material	d/e = 32/3.7 mm, PE, SDR17
- Absolute roughness of the pipe	k = 0.001 m
- Pipe conductivity	λ_p = 0.45 W/(m·K)
- Pressure drop of the working fluid of the GHE	Δp = 25 kPa
- Average electricity demand of the circulation pump PO-PP	$P_{E\ PO\text{-}PP}$ = 40 W
- Average electricity demand of the circulation pump PO-PW	$P_{E\ PO\text{-}PW}$ = 20 W

2.2.3. Ventilation system. The considered building is equiped with a supply and exhaust ventilation system with a nominal air flow designed as $\dot{V}_s/\dot{V}_e$ = 300/300 m^3/h with a high-efficient heat recuperator, that works with an annual average temperature recovery efficiency Φ_T = 88%. The air-handling unit (AHU) is localized in the technical room and has an automation system that allows the air-flow rate to

be controlled and also time control. In the summer mode the air bypasses the recuperator. The AHU is also equipped with a multi-step electric heater with electric power of 1.5 kW, which is used to pre-heat the fresh air to the temperature of -1°C during cold winter days.
Other characteristics of the ventilation system and other important assumptions:
- The air-flow rate within the cooling season (from 1 May to 30 September) 300 m^3/h
- The air-flow rate within the heating season (from 1 October to 30 April) 150 m^3/h
- The efficiency of the heat transfer in the electric heater 95%
- The air-flow is constant in the day/night period,
- In the cooling season, when the external air temperature $t_e > 16°C$, the AHU works using the by-pass (without recuperation),

2.2.4. Integration of the heating and mechanical ventilation systems. To improve the energy efficiency of the GSHP system and trying to obviate the need of fresh air pre-heating in the electrical heater it was decided to integrate the GSHP with the ventilation system using an air-to-glycol heat exchanger on the supply air side. This solution also has the ability to cool the supply air-flow during cooling season and provide the ground with the collected heat. The designed air-to-glycol heat exchanger is a three-row cooler with plate fins with an interspace of 2.5 mm. It is equipped with a stainless steel drip tray and a condensate drain connection. An overview of the HGSHP system is given by Fig. 2.
The design parameters of the air-to-glycol heat exchanger (designed for cooling application):
- Nominal air-flow rate $\dot{V}_s = 300\ m^3/h$
- Constant volume flow of the glycol mixture $\dot{V}_g = 0.35\ dm^3/s$
- Temperature of the fluid supplying the low-heat reservoir $t_s = 10°C$
- Temperature / relative humidity of the air entering the cooling coil $t_{a,e} = 35°C$ / $\varphi_{a,e} = 45\%$
- Temperature / relative humidity of the air leaving the cooling coil $t_{a,l} = 16{,}5°C$ / $\varphi_{a,l} = 92\%$
- Air pressure drop by flowing through the cooling coil $\Delta p_a = 5$ Pa
- Glycol mixture pressure drop by flowing through the cooling coil $\Delta p_g = 50$ Pa
- Nominal cooling power of the heat exchanger $\dot{Q}_C = 3{,}0$ kW
- Average electricity demand of the circulating pump PO-01 $P_{E\ PO\ 01} = 40$ W

Other properties of the air-to-glycol heat exchanger and assumptions:
- Control of the flow of the glycol mixture through the cooling coil is done using a three-way valve - quality control
- The heat exchanger works in heating mode only when the external air temperature is lower than -1°C and heats the air-flow to the temperature of -1°C. In case that the heat exchanger is not capable to heat the supply air to this level, the electrical heater supports its work.
- In the cooling mode the heat exchanger cools the air-flow to the temperature of 16°C only when the outdoor temperature exceeds 16°C.

In other cases the glycol mixture flow from the GHE by-passes the air-to-glycol heat exchanger and flows directly to the heat pump.

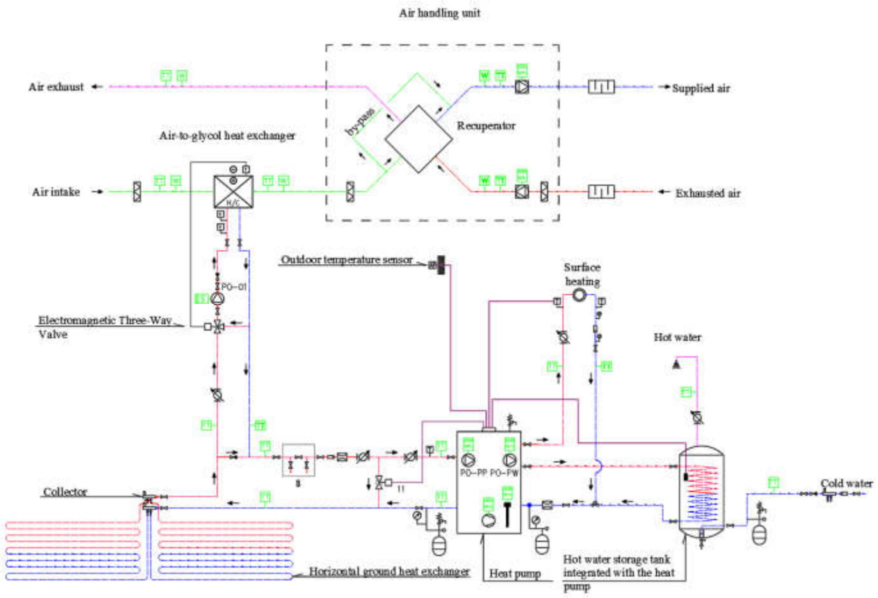

Figure 2. Ground source heat pump combined with additional air-to-glycol heat exchanger in supply air channel.

2.3. Operation strategy

The hybrid ground source heat pump system works in three different operation modes, depending on temporarily approaching hot water production and heating or cooling demands with a significant influence of outdoor air parameters.

2.3.1. Operation mode I – GSHP operates only for heating and hot water production. A default operation mode of the GSHP system. The low-heat reservoir is used to overcome heat gains in the building, what includes also hot water production. In case of appearing of heat demands on the side of the heat sink, the heat pump turns on and draws heat from the low-heat reservoir using the PO-PP circulation pump and transform it by the heat pumps compressor.

2.3.2. Operation mode II - Ground heat exchanger co-operates with the ventilation system. This operation mode is a common solution made for pre-heating or pre-cooling of ventilation air. It prevent efficiently the recuperator from freezing in winter and receive heat gains from the building by cooling the supplied air in summer. If there are heating or cooling demands of the fresh air, the circulation pump PO-01 turns on and moves the working fluid which receive the heat gains or overcome the heat losses.

2.3.3. Operation mode III - Integration of the GSHP and the air-to-glycol heat exchanger. Combination of the GSHP system and the heat exchanger on the supply ventilation air forms a hybrid ground source heat pump system. This modification make possible to use the ground as a heat source and heat sink at the same time for different operations like hot water producing (heat pump) and air cooling (air-to-glycol heat exchanger).

When the outdoor air temperature is lower than -1°C, the HGSHP works in the operation sub-mode IIIa as follows. The working fluid medium leaving the GHE firstly flows through the air-to-glycol HE, where the heat transfer between glycol and ventilation air takes place– it warms the air and gets colder. After this operation the working fluid enters the heat pumps evaporator, where it also gives the heat away. At the end the glycol mixture comes back to the GHE, where it receives heat from the ground. Next the cycle is repeated.

In cooling season, when the outdoor air conditions are exceeding the 16°C level, the HGSHP switches to the operation sub-mode IIIb. The glycol-mixture, that leaves the GHE is directed to the air-to-glycol HE, where it is heated, because of the overtaking of heat loads from the ventilation air. After this, the heated working medium gives the heat away in the evaporator of the heat pump and returns to the GHE where it is respectively cooled.

3. Calculations

Calculations were made considering hourly intervals and using the author's code written in MS Excel.

3.1. Theoretical model

The analysis of the operation of the integrated HGSHP system based on a theoretical heat exchange model. It describes a thermodynamically closed system which consist of the heat transfer to or from the 35% ethylene glycol mixture in three areas:

1) The ground – heat exchange with the surroundings of the ground heat exchanger,
2) Air in the ventilation system – heat exchange in the glycol heat exchanger between glycol and supply air,
3) R407C refrigerant – heat exchange in the evaporator of the heat pump (direct evaporation of the R407C in the heat pump internal circuit)

Depending on the variant, the operation of individual elements or integration of all parts of the system working at the same time was considered. The thermodynamically properties of water and air were approximated using appropriate equations [6]. Parameters of glycol were read from the tables [7].

Heat exchange between the ground and the glycol mixture in the ground heat exchanger can be calculated as follows [8]:

$$\dot{Q}_{GR} = L \cdot (\vartheta_1 - \vartheta_2) \cdot [R'_{tot} \cdot \ln(\vartheta_1 \cdot \vartheta_2^{-1})]^{-1} \qquad \text{(Eq.1.)}$$

Where: L [m] – the lenght of the ground source heat exchanger; ϑ_1 [K] – the temperature difference between the fluid entering the heat exchanger and the ground; ϑ_2 [K] – the temperature difference between the fluid leaving the heat exchanger and the ground; R'_{tot} $[m \cdot K \cdot W^{-1}]$ - replacement thermal resistance,

Heating and dry cooling operations of fresh ventilated air in the glycol heat exchanger are given by following equations [7].

$$\begin{cases} \dot{Q} = \dot{m}_a \cdot (h_{1a} - h_{2a}) \\ \dot{Q} = \dot{m}_g \cdot (h_{1g} - h_{2g}) \\ \dot{Q} = U \cdot A \cdot \Delta t_{log} \end{cases} \qquad \text{(Eq.2.)}$$

Where: $\dot{m}_a, \dot{m}_g$ [kg/s] – the mass flow of air, glycol mixture; $h_{1a}, h_{2a}, h_{1g}, h_{2g}$ [kJ/kg]– the specific enthalpy of air entering and leaving the HE and glycol mixture entering and leaving the HE; U $[W/(m^2 \cdot K)]$ – the heat transfer coefficient of the heat exchanger, A $[m^2]$ – the effective area of the heat exchanger, Δt_{log} [K] – the logarithmical temperature difference between the airflow and the flow of glycol mixture.

Heat exchange within the heat pump in the evaporator between the glycol mixture and the refrigerator R407C can be derived as follows:

$$\dot{Q} = \dot{Q}_h - \dot{E}_{el} \qquad \text{(Eq.3.)}$$

Where: $\dot{Q}_h$ [kW] - heat flow generated by the heat pump [$\dot{Q}_h$=f(COP)]; $\dot{E}_{el}$ [kW] – energy demand used in the heat pump [$\dot{E}_{el}$=f(COP)].

The relation of the energy conversion efficiency (COP) from the temperature of the low heat reservoir for assumed temperatures of the upper heat source was obtained from the heat pump manufacturer who empirically prepared the data (as shown in Fig. 1).

3.2. Simulations

Simulations were prepared using meteorological data of the Poznan City taken from ministerial resources [13], which are considering typical meteorological years and statistical climatic data compiled on the basis of the standard EN-ISO 15927:4. They take into account the dry and wet bulb temperature variations, relative humidity, moisture content and radiation intensity for different orientations and the angle of incidence for every hour of the year.

To calculate the heating and cooling demands daily usage profile of devices and daily attendance profile of residents were made. The following quantities were assumed:
- Sensible and latent heat gains from the occupants,
- Heat gains from the lightening system,
- Heat gains from devices,
- Heat gains or heat dissipation through the building fabric,
- Heat gains from solar radiation through transparent building compartments,
- Heat gains or heat dissipation trough the fresh air infiltration in the cracks,
- Efficiency of using the above heat gains.

The simulation model was created on the basis of heat exchange equations shown in Section 3.1. Due to the fact, that the calculations were carried out in MS Excel application some simplifying assumptions were made.

It was assumed that the heat exchange between the soil and the glycol mixture in the ground heat exchanger affect the soil temperature only when the heat flow $\dot{Q}_{GR}$ is bigger than the heat flow that can be distributed $\dot{Q}_{dstr}$ which depends on the heat conduction capacity H_S [W/K] of the ground. H_S is calculated as the product of soil heat conductivity λ_s, length of the GHE L_{GHE} and the logarithmical temperature difference between the soil and glycol $\Delta t_{log,S\text{-}Gl}$ in the GHE [Eq. 4.,5.,6.]. In such a case, the soil distribute heat according to the conduction capacity and the surplus heat flow $\dot{Q}$' heats or cools the ground proportionally to the soil conductivity. If the soil temperature changes due to the heat flow from the GHE, the ground strive for equalization of the boundary layer soil temperature $t_{S,B}$ and the infinite undisturbed soil temperature $t_{S,P\&W}$. Otherwise, if the heat flow is lower than the heat conduction capacity of the soil, the heat flow is fully absorbed and distributed by the ground and the soil temperature remain undisturbed. The heat flow $\dot{Q}_{GR}$ is calculated according to Sunden [8] assuming the N value equal the external radius of the pipe of the GHE. The reference temperature of the ground at the depth of 2 m is calculated with high precision according to the methodology mentioned by Popiel, Wojtkowiak [9,10,11] for the lawn covered soil. The boundary layer soil temperature for the n-th hour is derived by the Eq.7.

$$\dot{Q}_{dstr} = H_s \cdot \Delta t_{log,S-Gl}\ [W] \quad \text{(Eq.4.)}$$
$$H_s = 2 \cdot L_{GHE} \cdot \lambda_s\ [W/K] \quad \text{(Eq.5.)}$$
$$\dot{Q}' = \dot{Q}_{GR} + \dot{Q}_{dstr}\ [W] \quad \text{(Eq.6.)}$$
$$t_{S,B,n} = t_{S,B,n-1} + \dot{Q}' \cdot H_s^{-1} + \Delta t_{log,S,B-S,P\&W}\ [°C] \quad \text{(Eq.7.)}$$

The effective area of the air-to-glycol heat exchanger A_{HE} = 5,53 m^2 is given by the manufacturer. To avoid iterative calculations for the air-to glycol HE, logarithmical temperature difference Δt_{log} was replaced by an arithmetical temperature difference Δt_{art} using the correction factor k, such that:

$$\Delta t_{log} = k \cdot \Delta t_{art}\ [K] \quad \text{(Eq.8.)}$$

The characteristic parameters of heat transfer in the air-glycol heat exchanger in the form of the ratio kUA [W/K] in case of dry cooling and heating and kUA" for the wet cooling function were determined on the basis of averaged results of 40 measurement points, obtained using the calculation application provided by the manufacturer [14].

The main point of the simulation is to examine and compare the effects of integrated (or individual) operation of the HGSHP system on changes in working medium temperature in the whole year. Calculations and simulations were performed for three variants of the operating mode (see point 2.3).

3.3. Results and discussion

Graphs that illustrate hourly [Fig. 3] and average daily glycol mixture temperatures [Fig. 4] in three characteristic points of the HGSHP system were created. Those temperatures are compared with the

undisturbed soil temperature represented by the line $t_{S,P\&W}$ and the boundary layer soil temperature shown using the line $t_{S,B}$. The symbols from the Fig. 3 and Fig. 4 are explained below:
TT-01 – glycol mixture leaving the HGHE temperature
TT-02 – glycol mixture leaving the air-to-glycol HE and/or crossing through the by-pass temperature
TT-03 – glycol mixture leaving the evaporator and/or crossing through the by-pass temperature
$t_{S,P\&W}$ – undisturbed soil temperature
$t_{S,B}$ – ground temperature influenced by the heat exchange in HGHE

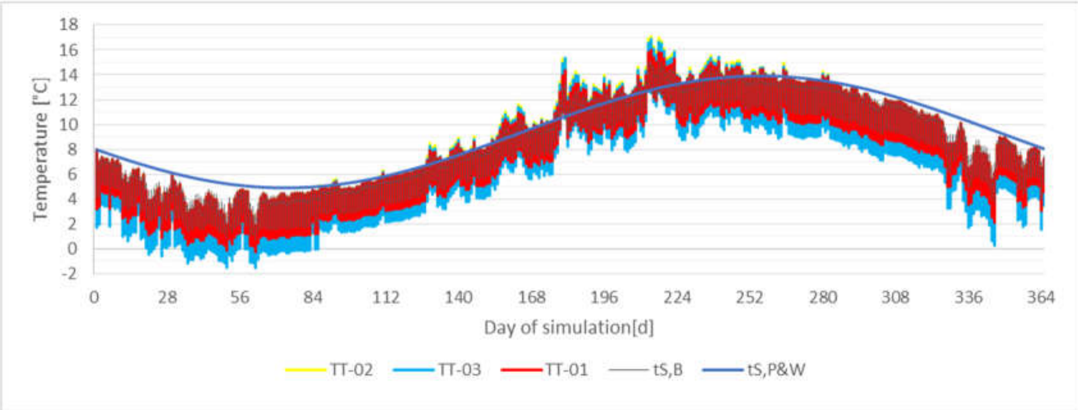

Figure 3. Average ground and working fluid temperatures – hourly analysis

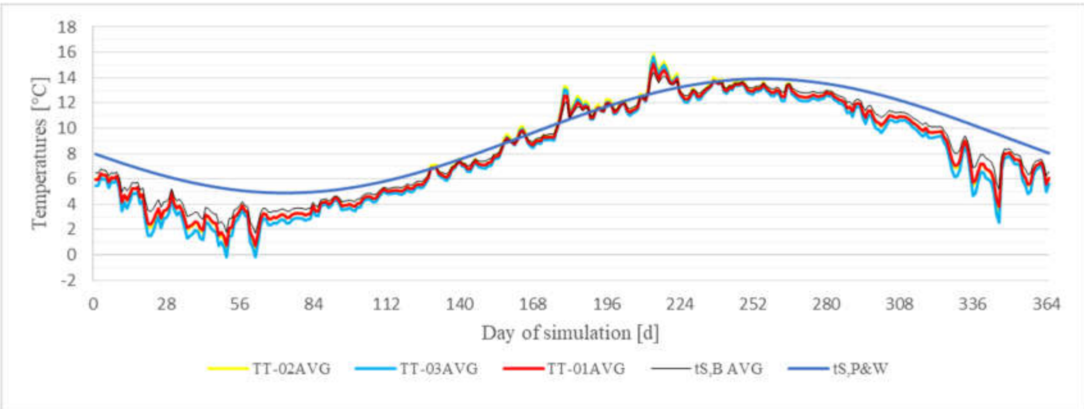

Figure 4. Average ground and working fluid temperatures – daily analysis

The simulation showed that the TT-03 temperature annual change is in the range between -1.5°C and 16,83°C, while the TT-01 temperature includes values from -0.24°C to 15.96°C.

Table 1. Amount of heat taken / given in the parts of the GSHP system

Results for the heat exchange in the GHE			
	HGSHP	**GSHP + el. heater**	
Amount of heat taken from the ground	3 770.6	3 984.2	kWh/a
Amount of heat received by the ground	336.3	0.0	kWh/a
Max. unitary heat flux received from the ground	2.52	2.19	kW
Max. unitary heat flux provided to the ground	1.31	0.00	kW
Results for the heat exchange in the air-to-glycol HE			
	HGHSP	**GSHP+ el. heater**	
Amount of heat directed to the HE	235.7	0.0	kWh/a
Amount of heat transferred by the electric heater	0.0	235.7	kWh/a
Amount of heat delivered by the HE	796.1	0.0	kWh/a
Max. unitary heat flux provided to the HE	0.73	0.00	kW
Max. unitary heat flux received from the HE	1.95	0.00	kW
Results for the heat pump working for heating and hot water producing.			
	HGSHP	**GSHP + el. heater**	
Heat demands for hot water production (h.w.p.)	2 957.62	2 957.62	kWh/a
Heat demands for heating and ventilation (h&v)	1 616.04	1 616.04	kWh/a
Amount of heat taken from the ground for h.w.p.	2 234.93	2 232.16	kWh/a
Amount of heat taken from the ground for h&v	1 332.29	1 336.35	kWh/a
Amount of electricity used by the compressor for h.w.p.	722.68	725.45	kWh/a
Amount of electricity used by the compressor for h&v	283.75	279.69	kWh/a
Amount of electricity used by the electric pre-heater	0.0	248.1	kWh/a
Max unit. heat flux taken from the ground by the heat pump for h.w.p.	4.03	3.85	kW
Max unit. heat flux taken from the ground by the heat pump for h&v	1.20	1.22	kW
Seasonal efficiency of the heat pump for h.w.p. ($SCOP_{HWP}$)	4.09	4.08	-
Seasonal efficiency of the heat pump for h.w.p. during summer season	4.37	4.29	-
Seasonal efficiency of the heat pump for h.w.p. during winter season	3.92	3.94	-
Seasonal efficiency of the heat pump for h&v ($SCOP_{H\&V}$)	5.70	5.78	-
Seasonal performance factor ($SPF_{H\&V+HWP+EH}$)	4.54	3.65	-

Results of calculations are shown in Tab.1. In this table a comparison between the energy amounts and heat fluxes in installations in the considered building with the HGSHP system and in a similar basic GSHP system without an air-to-glycol HE with using electric heating of ventilated air in winter were made.

Obtained results of the simulation allow to formulate that the integration of an additional heat exchanger ventilation air heat exchanger with the GSHP system make possible to direct 796.1 kWh/a heat from the airflow to the glycol mixture. Almost 60% of this waste heat is used directly in the evaporator of the heat pump. The rest – 336.3 kWh/a is absorbed by the ground. The HGSHP system need to take from the soil 213.7 kWh/a less energy than the classic GSHP system. During winter the air-to-glycol heat exchanger pre-heats the airflow and obviate the need of using the electric heater. This generates savings in the value of 235.7 kWh/a heat produced from electricity. The COP of the heat pump in the HGSHP is increased during summer season by 0.08, but during winter, the COP fell by 0.02 for h.w.p. and 0.08 for heating and ventilation purposes. Summarizing the annual effects of the integration, the SPF increase from 3.65 to 4.54 what represent a 24.38% growth (considering demands for heating, ventilation and electricity for the electric heater).

3.4. Energy, economy and environmental analysis

To determine the total energy, economy and environmental benefits of using the designed HGSHP system instead the traditional GHSP system and electric air heating, the additional air cooling by the air-to-glycol HE in summer must been taken under investigation. Also the work of the supplementary circulation pump PO-01, which is necessary for the operation of the HGSHP, should be considered. Results of the analysis are shown in Tab.2. Assumptions for the energy, economy and environmental analysis:

- Average seasonal efficiency of chill production (for the non-integrated system) SEER =3.0
- Electricity unit price P_{EL} = 0.55 zł/kWh
- Total investment cost of the air-to-glycol HE installation and integration I_0 = 3000 zł
- Operation cost (without electricity) for both systems are the same
- CO_2 emission factor (data from the National Centre for Emissions Management) e_{CO2}=798 kg/MWh

Table 2. Energy, economy and environmental analysis

Energy analysis			
	HGSHP	**GSHP+el. heater+ cooling**	
Amount of electricity used by the heat pump	1 006.43	1 005.14	kWh/a
Amount of electricity used by the circulating pump PO-01	128.00	0.00	kWh/a
Amount of electricity used by the circulating pump PO-PP	44.12	43.97	kWh/a
Amount of electricity used by the circulating pump PO-PW	22.06	21.98	kWh/a
Amount of electricity used by the electric heater	0.00	248.07	kWh/a
Amount of electricity used by the cooling system	0.00	265.38	kWh/a
Total amount of electricity consumption	1 200.61	1 584.54	kWh/a
Total amount of final energy (heat and cold)	5 369.80	5 369.80	kWh/a
Seasonal Performance Factor of the heating and cooling system ($SPF_{H\&V+HWP+EH+PO+COOLING}$)	**4.47**	**3.39**	**-**
Economy and environmental analysis			
	HGSHP	**GSHP+el. heater+ cooling**	
Annual variable cost of the system operation	660.34	871.50	zł/a
Total investment costs	3 000.00	0.00	zł
Simple Payback Time	14,21		a
Total annual CO_2 emission	958.09	1 264.47	kg CO_2/a
Total CO_2 emission reduction	306.38		kg CO_2/a

Obtained results of the energy, economy and environmental analysis show that the integrated system require only 75.8% of the total annual electricity consumption used by the non-integrated system. This results in 211.16 zł/a savings. The HGSHP system is characterized by a SPF factor (considering demands for heating, cooling, ventilation and electricity for circulation pumps and electric heater) equal 4.47. Without integration, the SPF is lower by 1.08 and reaches the value of 3.39, so the integration ensure a 31.8% growth of the energy effectiveness. Environmental benefits of system integrating are clear and reach a CO_2 emission decrease about 306.4 kg in the year.

4. Conclusions

Application of a hybrid ground source heat pump system combined with the air-to-glycol heat exchanger on the supply air duct in a passive building leads to energy, economy and environmental benefits. Appropriately designed and executed low-heat reservoir and glycol heat exchanger obviate the need of using an energy intensive electric air heater, which is a common solution in traditional ventilation systems in temperate and cold climates. During summer this integration allows for pre-cooling of supply air and regeneration of the ground what finally increases the COP of the heat pump. The presented solution agrees with the idea of the Sustainable Development, because it minimize the energy consumption and reduces the CO_2 emission and, what is important, it decreases also the running costs, while the investment itself is modest. The designed innovation has a high application potential for micro and macro scale installations using GSHP.

5. Appendices

In the near future it is planned to validate the performed calculations based on collected measurement data. The presented technological system is controlled with different sensors shown in Fig.2. Incomplete data are collected from 1 January 2017.

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