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Computational and experimental performance analysis of a novel method for heating of domestic hot water with a ground source heat pump system

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Abstract

The paper presents a novel method developed for energy efficient heating of domestic hot water (DHW) with a ground source heat pump (GSHP) system. The method is based on step-based heating of DHW, where the DHW is gradually heated from the inlet temperature of domestic cold water to the target temperature of DHW using a specifically designed GSHP system concept. The developed method was tested and validated using experimental and computational performance analyses and the energy efficiency, operation and potential limitations of the developed application were also studied in existing apartment buildings. To demonstrate the efficiency of the developed application, the performance of the new concept was compared to the performance of a conventionally used GSHP system. The results demonstrated that the developed GSHP concept delivered up to 45–50% improvement in energy efficiency of the DHW heating process over the conventional GSHP application. The measured Seasonal Performance Factors were 2.5–2.6 for the conventional application and as high as 3.7–3.8 for the developed application, when DHW was heated from 7.5 °C to 55 °C. The case study demonstrated that the developed concept is also well applicable in existing buildings, but there is still room for improvement.

32 **Keywords – ground source heat pump; geothermal energy; energy efficiency; domestic hot water; smart control;**
33 **renewable energy source; apartment building**

34 **1. Introduction**

35 The recast Energy Performance of Buildings -directive (EPBD) requires that all new buildings must be nearly zero-
36 energy buildings (nZEBs) by the end of 2020 [1]. While the detailed country-specific definitions of nZEBs are left to the
37 Member States of the European Union, the recast EPBD requires nZEBs to be buildings with high energy performance
38 and that renewable energy sources (RES) are effectively utilized to cover the primary energy (PE) demand of nZEBs [1].
39 Recent studies demonstrate that heat pump systems, especially a ground source heat pump (GSHP) system, are one of
40 the best applications to utilize RES and to improve the primary energy performance of buildings cost-effectively, de-
41 pending on the Member State-specific PE factors [2-7]. The popularity of heat pump installations is also increasing in
42 deep renovations of buildings, as the heat pump systems typically provide a cost-effective alternative to significantly
43 improve the energy efficiency of a building over the more conventional renovation measures related to improving the
44 energy efficiency of the building envelope [2,3,8,9].

45 Heating of domestic hot water (DHW) accounts for a significant share of the overall energy consumption in residen-
46 tial buildings. According to a study conducted by Yao and Steemers [10], up to 20% of the overall domestic energy is
47 used for DHW purposes in the United Kingdom. The DHW consumption of residential buildings accounts for as high as
48 72% of the total DHW consumption [11]. A literature review conducted by Ahmed et al. [12] indicated that the DHW
49 consumption due to occupants fluctuates significantly from country to country, but an average consumption is typically
50 30–70 L/person/day. Ahmed et al. [12] also concluded that the season of the year affects the DHW consumption rate so
51 that more DHW is consumed during the winter time and less during the summer time, respectively. According to their
52 study, the specific DHW consumption of smaller apartments with 1–2 occupants is approximately 1.5 times higher than
53 the specific DHW consumption of larger apartments in Finnish residential apartment buildings.

54 In many European countries, a significant share of the population lives in multi-family apartment buildings, where
55 the heating of DHW can account for more than 50% of the overall heating energy demand of the building, depending on
56 the age, size, location and energy performance of the building [5,13-16]. In Nordic countries, the average DHW con-
57 sumption rate is approximately 35 L/person/day with the highest consumption rate occurring in Finland (43
58 L/person/day) and the lowest occurring in Denmark (20 L/person/day) [12]. For comparison, Swan et al. [17] have de-
59 termined that the average DHW consumption of typical Canadian households is approximately 67 L/person/day. This

60 indicates that energy efficient heating of DHW has a significant impact on the overall energy performance of future
61 nearly zero-energy apartment buildings, where the energy efficiencies of the building envelope and technical systems
62 of the building are already maximized or at a high level.

63 Several studies have investigated the performance, energy saving potential and operation of ground source heat
64 pump systems in different climate conditions and concluded that the GSHP system is typically a highly efficient energy
65 production technology in many different applications and building types, when cost-effectiveness, energy efficiency and
66 environmental impact aspects are taken into account [18-24]. While the technology development of the GSHP systems
67 has improved the energy performance, operational characteristics, reliability and controllability of the systems, the
68 overall energy efficiency of a GSHP system is still highly dependent on the temperature levels of both the evaporator
69 and the condenser. It has been concluded in numerous previous studies that energy efficient heating of DHW is the
70 major challenge related to the performance of the GSHP systems especially in cold climate conditions, as the tempera-
71 ture requirement of DHW is typically 55–60 °C to prevent legionella growth [2,5,8,18,25-27]. The coefficient of perfor-
72 mance (COP) of conventional GSHP systems is typically in the range of 2.2–3.0, when DHW with the aforementioned
73 temperature requirement of 55-60 °C is produced and the inlet temperature of domestic cold water is 5–10 °C, respec-
74 tively.

75 The literature review related to ground source heat pump applications in energy efficient and nearly zero-energy
76 residential buildings indicates that cost-efficient, operational and implementable applications are required to improve
77 the energy efficiency of DHW heating with the GSHP systems, as the heating of DHW accounts for a significant share of
78 the overall energy consumption in new, deeply renovated and future nearly zero-energy residential buildings. The liter-
79 ature review also indicates that there is a limited number of studies related to investigating and improving the efficien-
80 cy of heat pump system applications operating in multi-family residential buildings. In addition, majority of the previous
81 studies [28-33] have focused on studying solar assisted ground source heat pump (SAGSHP) applications, which may not
82 be the cost-optimal GSHP system application in many of the apartment building cases due to the main limitations relat-
83 ed to the solar thermal systems, such as a limited roof area in large high-rise apartment buildings and increased tech-
84 nical space requirements for installation of accumulators or heat storages.

85 This study presents a novel method developed for energy efficient heating of DHW with a GSHP system in multi-
86 family apartment buildings without the need for auxiliary heating systems or supporting renewable energy production
87 systems, such as a solar thermal collector system. The performance of the developed method is studied and validated
88 by computational and experimental analyses and also by conducting on-site field studies in existing apartment buildings

located in the cold climate of Finland. The efficiency of the developed method is also compared to the efficiency of a conventional method applied for heating of DHW with a GSHP system to determine the improvements in energy performance. As a result of the computational validation analysis, a dynamic simulation model is developed to simulate the performance and operation of a GSHP system utilizing the developed DHW heating method. The simulation model can be further developed to optimize the performance of GSHP systems according to local electricity spot-prices, case-specific DHW consumption profiles or cost-optimal demand response strategies. The objectives of this study are:

- to study and validate the performance of a novel method developed for energy efficient heating of DHW with a GSHP system in multi-family apartment buildings;
- to compare the efficiency of the developed method to the efficiency of a conventional and typically used application for heating of DHW with a GSHP system;
- to determine the difference in energy efficiency between the developed and the conventional GSHP DHW heating applications;
- to develop and validate a dynamic simulation model for performance assessment and optimization of GSHP systems utilizing the developed method;
- to study the performance, functionality and potential limitations of the developed method in existing apartment buildings located in cold climate conditions.

2. Methods

2.1. Heating of DHW with geothermal heat pump systems

2.1.1 Thermodynamic principle

This study focuses on developing a new method for energy efficient heating of DHW with a GSHP system in multi-family apartment buildings. The method can also be utilized in non-residential buildings and with other heat pump types in addition to the GSHP, e.g. air-to-water, exhaust air and hybrid heat pumps. The method is based on step-based heating of DHW, where the heating of DHW is started from the inlet temperature of domestic cold water (DCW), e.g. 5-10 °C. In the step-based heating method, the domestic cold water is gradually heated to the target temperature of DHW, e.g. 55 °C, whereas in majority of the conventional DHW heating applications utilizing a GSHP system, the domestic cold water is heated to the required temperature of DHW in a separate heat storage tank or in a combined heat storage tank used for space (and ventilation) heating and DHW. The conventional heating method requires that the heat storage tank used for heating of DHW must be at a high temperature, e.g. 60-65 °C, at all times to prevent le-

117 gionella growth [34], which reduces the COP of the heat pump system significantly compared to a situation, where the
 118 domestic cold water is heated to the required temperature of DHW in several steps.

119 The basic thermodynamic principle of a five-step heating process of DHW using the R410A refrigerant, a commonly
 120 used refrigerant in modern commercially manufactured GSHP systems, is presented in Fig. 1 [35]. The data presented in
 121 Fig. 1 is calculated using the thermodynamic properties of the R410A refrigerant. The heating of DHW from 5 °C to 55 °C
 122 is divided into 5 steps, which improves the overall COP of the heating process, as the condensing pressure of R410A is
 123 significantly lower at low condensing temperatures and increases exponentially, when the condensing temperature
 124 increases (see Fig. 1). By dividing the heating of DHW into multiple steps, the average condensing pressure of the over-
 125 all heating process decreases, thus improving the energy performance of the heat pump system significantly compared
 126 to the situation, where the heat pump system maintains 60-65 °C temperatures in the heat storage tank at all times in
 127 order to supply 55 °C DHW to occupants. The average condensing pressure of the five-step heating process of DHW is
 128 approximately 22 bar using the R410A refrigerant (Fig. 1), whereas the average condensing pressure of the convention-
 129 al one-step heating process of DHW is approximately 42 bar with the R410A refrigerant. To further demonstrate the
 130 thermodynamic effectiveness of the step-based heating process of DHW, the calculated indicative COP-values of each
 131 step are also presented in Fig. 1 in a scenario, where the evaporating temperature is -5 °C, superheat of the refrigerant
 132 in the evaporator is 3 K and the isentropic efficiency of the compressor is 75% [35].

133 **Fig. 1.** Basic thermodynamic principle of a five-step heating process of DHW using the R410A refrigerant. The average condensing
 134 pressure of each step is presented as a function of the condensing temperature (evaporating temperature is -5 °C, superheat of the
 135 refrigerant in the evaporator is 3 K and the isentropic efficiency of the compressor is 75%) [35].
 136

137

138   The five-step heating process of DHW presented in Fig. 1 is an example to demonstrate the basic thermodynamic
139 principle and the effectiveness of the step-based heating method in a specific case, where domestic hot water is heated
140 from 5 °C to 55 °C. In general, the optimum number of heating cycles should be selected according to the heat pump
141 and thermal storage applications used in the implementation. Majority of the commercially manufactured heat pump
142 units have been optimized to operate using specific temperature differences for the heat exchangers of both the evap-
143 orator and the condenser. A typical temperature difference between the inlet and outlet brine of the evaporator is 3–5
144 K, whereas a typical temperature difference between the inlet and outlet water of the condenser is 6–8 K. This means
145 that a typical commercially manufactured heat pump unit can increase the temperature of the heating water (conden-
146 ser side) by approximately 7 °C during one heating cycle. Therefore, the actual number of heating cycles using conven-
147 tional commercial heat pump units is approximately 7-8, if DHW is heated from 5 °C to 55 °C and the stratification phe-
148 nomena of water in the heat storage tank are also taken into account.

149

150 2.1.2 Operation of conventional GSHP system

151   The typical configurations of the conventional GSHP system applications (variable and fixed condensing operation)
152 are presented in Figs. 2 and 3. According to multiple on-site field measurements conducted by the authors, the overall
153 energy efficiency of both applications is similar, when heating of DHW is discussed. Both of the applications presented
154 in Figs. 2 and 3 utilize one step-based heating of DHW, where the DCW is heated from the inlet temperature directly to
155 the required temperature of DHW in the heat storage tank. This requires that the temperature of the heat storage tank
156 must be above 55 °C (typically 60-65 °C) at all times.

157

158 **Fig. 2.** Typical configuration of a ground source heat pump system equipped with an exchange valve and variable condensing opera-
 159 tion.
 160

161 **Fig. 3.** Typical configuration of a ground source heat pump system equipped with a refrigerant superheat exchanger and fixed con-
 162 densing operation.
 163
 164

165 2.1.3 Operation of GSHP system with step-based heating of DHW

166 The developed configuration of a GSHP system utilizing the step-based heating of DHW is presented in Fig. 4. The
 167 main challenge related to the step-based heating of DHW is the temperature requirement of DHW. To maximize the
 168 energy efficiency improving benefits of the method, the heating of DHW should be started from the inlet temperature
 169 of domestic cold water, which is typically 5–10 °C, depending on both the season of the year and the location of the
 170 building. However, 55 °C DHW should be available to occupants at the same time and it is challenging to meet both of
 171 these criteria simultaneously. For this reason additional heat storage tanks and more advanced control of the heating
 172 process are required (see Fig. 4).

Fig. 4. Configuration of a ground source heat pump system equipped with step-based heating of DHW, refrigerant superheat exchanger and variable condensing operation.

The operation principle of the GSHP system utilizing the step-based heating of DHW is as follows (Fig. 4):

- the GSHP system starts to heat tank 1, which is at the temperature of DCW (7 °C) in the beginning of the heating process. Exchange valves 1-3 are controlled as follows: valve 1: connection *a* is open and *b* is closed, valve 2: connection *a* is closed and *b* is open, valve 3: connection *a* is closed and *b* is open. The temperature of tank 2 is approximately 50-55 °C, the temperature of tank 3 is approximately 55-60 °C and the temperature of tank 4 is approximately 60-67 °C. Tanks 2-4 are connected in series and used to deliver DHW to occupants during the heating process of tank 1;
- tank 1 is gradually heated from the initial temperature of 7 °C to the target temperature of 50-55 °C using exchange valve 0 (connection *a* is closed and *b* is open) and the heat exchanger. The GSHP system is equipped with variable condensing operation and the DHW is heated using the step-based heating process with 7-8 steps (see Fig. 1 for the thermodynamic principle of a five-step heating process), as the temperature increase of DHW is approximately 7 °C in each step. When the condensing temperature is high enough, the superheat circuit is started and the superheat energy is discharged to tanks 3 and 4, according to the temperature levels of both the superheat and tanks 3-4. The superheat circuit in tanks 3 and 4 is connected in series using heating coils so that tank 4 is heated first and tank 3 is heated second;
- tanks 2-4 are discharged during the heating process of tank 1 by delivering DHW to occupants. When the temperature of tank 2 has decreased to the temperature of DCW (7 °C), the heating of DHW is switched to tank 2 and the heating process is started from the beginning as with tank 1. The control of exchange

valves 1-3 is switched as follows: valve 1: connection *a* is open and *b* is closed, valve 2: connection *a* is open and *b* is closed, valve 3: connection *a* is open and *b* is closed. During the discharge period of tank 2, tank 1 is heated to the target temperature of 50-55 °C;

- during the discharge of tank 2, the temperature of the tank is decreased from the initial temperature of 50-55 °C to the temperature of DCW (7 °C). However, the required temperature of DHW is maintained at all times, as the temperatures of tanks 3 and 4 are higher (55-60 °C for tank 3 and 60-67 °C for tank 4) and all tanks 2 (or tank 1 when the heating process is switched), 3 and 4 are connected in series. In addition, the superheat received from heating of tank 1 is also used to heat tanks 3 and 4 during the discharge period of tank 2, when the temperature level of the superheat circuit is high enough.

While the operation of the phased heating of DHW requires more control equipment than the conventional GSHP system, the smart control strategy is well implementable using conventional building management software. According to the previous on-site field studies conducted in existing buildings by the authors, the capacity of the superheat is approximately 10-15% of the nominal capacity of the heat pump unit, e.g. 5-7 kW with a 50 kW GSHP unit. Previous field measurements also demonstrate that as high as 90-95 °C outlet temperatures are achievable from the superheat circuit, when the outlet temperature of the condenser is 57-60 °C.

2.2. Experimental analysis

2.2.1 Performance assessment of the studied GSHP systems

The energy performance of the studied GSHP systems was assessed by conducting measurements related to the energy efficiency of the systems in DHW heating process. The space heating was excluded from the measurements. Furthermore, the heating of DHW circulation system was also excluded from the testing setup. The installation principles, which were used in the measurements, of both the conventional and the developed heat pump systems are presented in Fig. 5. The main components and measurements related to the testing procedure are also presented in Fig. 5.

$$SPF_{total} = \frac{Q_{heat,total}}{Q_{electricity,HP+aux.}} \quad (1)$$

228

229   where: $Q_{heat,total}$ is the total heating energy produced by the heat pump system during the measurement period, in-
230 cluding condensate and superheat, kWh; $Q_{electricity,HP+aux.}$ is the total electrical energy consumed by the heat pump sys-
231 tem and auxiliary equipment, such as circulation pumps and control equipment, kWh. However, the auxiliary heating
232 systems, such as electrical heating coils or back up heaters, were not included in the calculation of the SPF, as they were
233 not included in the testing setup (see Fig. 5). In the measurements of the study, the SPF presents the overall perfor-
234 mance of the studied heat pump applications in heating of DHW.

235   The measurements were conducted at the heat pump measurement and test facilities of a Finnish heat pump
236 manufacturer Gebwell Oy. The tested heat pump unit was a Gebwell Taurus 90, which has two scroll-type compressors,
237 electronic expansion valve and a total heating capacity of 90 kW [37]. According to the manufacturer, the COP-values
238 (measured according to the EN 14511 standard [38,39]) of the tested heat pump unit are 4.5 (0/35 °C) and 3.2 (0/50 °C),
239 respectively [37]. Only one of the two compressors was used in the test measurements resulting in a 45 kW nominal
240 heating power output of DHW, as this scenario was determined to represent the actual average operating conditions of
241 the tested heat pump unit better, when monitored operation of installed GSHP systems is discussed.

242   While the testing procedure and measurement arrangements (Fig. 5) do not accurately meet the requirements of
243 part 2 (test conditions) and part 3 (test methods) of the EN 14511 standard [38,39], the results of the measurements
244 give accurate information about the performance of both of the systems in dynamic operating conditions. In addition,
245 the measurement results of both of the studied systems are comparable with each other, which was the main objective
246 of the experimental analysis. The main measurement equipment used in the experimental analysis are presented in
247 Table 1. All the measurement sensors and equipment were calibrated before the performance tests. The recording
248 resolutions of energy meters were 10 kWh for produced condensate and superheat energies and 1 kWh for consumed
249 electrical energy. The measurements included a total of three 16-hour measurement cycles for both of the studied heat
250 pump applications. The selected measurement cycle was determined to represent an average daily DHW consumption
251 profile of typical Finnish residential apartment buildings.

252

253 **Table 1.** The main measurement equipment used in the experimental analysis.

Measurement	Meter	Tolerance according to the manufacturer
Temperature sensors	Pt500 (Class A)	$\pm 0.19\text{ }^{\circ}\text{C}$ (20 °C): $\pm 1.0\%$ $\pm 0.37\text{ }^{\circ}\text{C}$ (110 °C): $\pm 0.3\%$
Produced energy, condensate circuit	Kamstrup Multical 601	$\pm 0.7\text{--}0.9\%$
Produced energy, superheat circuit	Kamstrup Multical 601	$\pm 0.7\text{--}0.9\%$

Consumed electrical energy	Carlo Gavazzi EM340	±1%
Water flow rate, condensate circuit	Kamstrup Ultraflow 54 (DN50)	±1.0–1.5%
Water flow rate, superheat circuit	Kamstrup Ultraflow 54 (DN25)	±1.0–1.5%
Water flow rate, DHW	Kamstrup Ultraflow 24 (DN25)	±2% (temperature range 0.1–30 °C) ±3% (temperature range 30–90 °C)

2.2.2 DHW consumption profile

The daily DHW consumption was selected according to the power output of the tested heat pump unit to represent a typical application, where the heat pump unit would operate in actual case apartment buildings. The tested heat pump unit is normally used in applications, where the number of apartments is approximately 50. The average number of occupants was assumed to be 1.5 occupants/apartment resulting in a total number of 75 occupants. The specific DHW consumption of an occupant was assumed to be 56 dm³/occupant/day, which shows good agreement with the recent DHW consumption-related studies and results in a 4200 dm³ total daily consumption of DHW [11,12,17,40]. The DHW consumption profile used in the measurements is presented in Table 2. The profile was determined to represent an average DHW consumption profile of a Finnish apartment building. The supply temperature requirement of DHW was selected to be 55 °C in the measurements according to the current Finnish building regulations [41]. The inlet temperature of domestic cold water varied according to the time of the day, but an average DCW inlet temperature was approximately 7.5 °C in the measurements, respectively.

Table 2. The daily consumption profile of DHW used in the measurements.

Number of consumption cycle	Time of day	Duration of consumption cycle [s]	Flow rate [dm ³ /s]	Total flow rate of consumption cycle [dm ³]
1	6:00	567	0.3	170
2	6:30	567	0.3	170
3	7:00	300	0.8	240
4	7:30	567	0.3	170
5	8:00	567	0.3	170
6	8:30	567	0.3	170
7	9:00	567	0.3	170
8	10:00	525	0.3	157.5
9	11:00	525	0.3	157.5
10	12:00	525	0.3	157.5
11	13:00	525	0.3	157.5
12	14:00	525	0.3	157.5
13	15:00	525	0.3	157.5
14	16:00	525	0.3	157.5
15	17:00	525	0.3	157.5
16	18:00	960	0.3	288
17	19:00	960	0.3	288

18	20:00	960	0.3	288
19	20:30	300	0.8	240
20	21:00	960	0.3	288
21	22:00	960	0.3	288
Total consumption of DHW, dm³				4200

2.3. Computational analysis

2.3.1 Simulation method

The computational analysis was conducted using the IDA Indoor Climate and Energy 4.7 (IDA-ICE) simulation tool and the Early Stage Building Optimization (ESBO) -plant model, which is integrated in the latest versions of IDA-ICE and can be used to simulate the on-site renewable energy production systems as a part of dynamic energy simulation of buildings. IDA-ICE is a versatile dynamic simulation tool for simulation of energy consumption, indoor air quality, thermal comfort conditions and on-site renewable energy sources, e.g. different heat pump systems and solar-based energy production systems, in complex multi-zone buildings [42-48]. The reliability, accuracy and performance of IDA-ICE has also been validated in numerous previous studies by measurements, field tests and comparisons to other validated simulation tools [42-48].

The studied heat pump applications were modelled according to Fig. 5 using the ESBO Plant model and appropriate methodology and simplifications. The simulated heat pump unit was calibrated according to the technical performance specifications of the Gebwell Taurus 90 heat pump unit using a detailed calibration methodology developed in previous studies by the authors [49,50]. The authors have also validated the integration of IDA-ICE and ESBO Plant model as an accurate simulation method to be used in simulations of modern variable condensing heat pump systems [50]. The performance of the simulated heat pump model was calibrated to correspond to the performance data of the Gebwell Taurus 90 heat pump unit. The calibration results were as follows:

- operating conditions 0/35 °C (EN 14511): simulated COP 4.50, measured COP (by the manufacturer): 4.5;
- operating conditions 0/50 °C (EN 14511): simulated COP 3.19, measured COP (by the manufacturer): 3.2.

2.3.2 Studied simulation cases

The speed and consistency of the control strategy used in the simulation model are essential to accurately simulate the step-based heating of DHW presented in Fig 5. In order to ensure that 55 °C DHW is available to occupants at all times and to maximize the energy efficiency of the heat pump system simultaneously, the heating of DCW must be

294 started at the optimum time and the step-based heating process of DHW must be optimally matched to the DHW con-
295 sumption profile. The daily consumption profile of DHW used in the simulations was modelled according to Table 2. To
296 study the effect of different control strategies and other essential factors on the energy efficiency and functionality of
297 the developed step-based heating of DHW, the following simulation cases were investigated:

- 298 • loose control strategy: the temperature of DHW is allowed to vary and slightly decrease from the 55 °C re-
299 quirement for short periods of time, the maximum variation of the temperature is set to 2.5 °C resulting in
300 a temperature set point range of 52.5 °C (minimum) to 55.0 °C (maximum);
- 301 • regular control strategy: the temperature of DHW is not allowed to vary and is maintained at the 55 °C
302 (temperature set point range 54.8–55.0 °C) requirement level at all times, but the heat storage tanks of
303 DHW are not overheated in advance according to the predicted DHW consumption profile to ensure that
304 the temperature of DHW supplied to occupants is never below 55 °C;
- 305 • tight control strategy: the temperature of DHW is not allowed to vary and is maintained at the 55 °C (tem-
306 perature set point range 54.8–55.0 °C) requirement level at all times. The heat storage tanks of DHW are
307 also optimally overheated, avoiding unnecessary overheating, in advance according to the predicted DHW
308 consumption profile to ensure that the temperature of DHW supplied to occupants is never below 55 °C;
- 309 • more efficient heat exchanger for heating of DHW (see Fig. 5): the minimum temperature difference be-
310 tween the heated and cooled liquid flows (pinch point) of the 54 kW counter-flow heat exchanger used in
311 the measurements was approximately 3.5 K. An additional case was studied, where the minimum temper-
312 ature difference was decreased to 2.0 K by using a larger and more efficient counter-flow heat exchanger
313 to determine the improvements in energy performance of the developed GSHP system.

314 The conventional heat pump system was modelled according to Fig. 5 and simulated by using a relatively tight con-
315 trol strategy that corresponded to the control strategy used in the measurements of the experimental analysis.

316

317 *2.4. Performance analysis in existing buildings*

318 *2.4.1 Studied case buildings*

319 The developed GSHP system concept utilizing the step-based heating process of DHW has been installed in over 20
320 Finnish multi-family apartment buildings since 03/2016, when the concept was first introduced. The case buildings
321 include both new and existing apartment buildings built during different decades. The energy efficiency and operation

322 of the installed systems have been monitored in all of the buildings. In general, the overall efficiency of the GSHP sys-
323 tem decreases in existing apartment buildings, where the temperature level of the original water radiator heating sys-
324 tem of room spaces is typically high (supply water dimensioning temperature typically 60–80 °C), and the share of ener-
325 gy consumption related to heating of DHW is lower from the overall heating energy consumption of the building due to
326 the high energy consumption of room spaces. For this reason a new apartment building was selected as a case study
327 building to better demonstrate the effectiveness of the developed concept.

328 The studied case apartment buildings consist of four individual apartment buildings located in Espoo, Finland. The
329 total number of apartments included in the buildings is 140 and all of them are occupied. The average size of the
330 apartments is approximately 40 m². The buildings were completed in December 2016 and they have been in use for
331 approximately 6 months. The total heated floor area of the case buildings is 5 660 m² and the total heated volume is 16
332 600 m³, respectively. The thermal transmittances of the external structures are:

- 333 • external walls: 0.15 W/(m²K);
- 334 • roofs: 0.08 W/(m²K);
- 335 • base floors (connected to the ground): 0.15 W/(m²K);
- 336 • windows: 0.8 W/(m²K);
- 337 • external doors: 1.0 W/(m²K).

338 Furthermore, the measured air-tightness (the q₅₀-value) of the building envelope is 1.2 m³/(m² h) in the studied
339 case buildings, which is a typical value for new energy efficient apartment buildings, as the reference air-tightness value
340 of the building envelope is 2.0 m³/(m² h) for new buildings according to the current Finnish building regulations [51].
341 The site plan, main geometry and features of the case apartment buildings are presented in Fig. 6.

Site plan of the studied buildings (5 661m²)

343 **Fig. 6.** The site plan (left) and main geometry (right) of the studied case apartment buildings.
344

345 *2.4.2 Technical systems of the case buildings*

346 The main heating system of the studied case buildings consists of two Gebwell Taurus 90 heat pump units and two
347 four-tank heat storage combinations (see Fig. 5) used for step-based heating of DHW. The auxiliary heating system of
348 the buildings is electricity. The heat distribution system of the case buildings is hydronic floor heating system with 40/35
349 °C dimensioning temperatures. The temperature set point of DHW is 55 °C and the designing temperatures of the DHW
350 circulation system are 55/50 °C with a 0.13 dm³/s water flow rate. The ventilation system of the buildings is decentral-
351 ized (individual air handling units in each apartment) supply and exhaust air ventilation system with energy efficient
352 (temperature efficiency $\eta_T = 80\%$) heat recovery system and electric preheating coils for supply air. The case apartment
353 buildings are also equipped with building-specific sewage water heat recovery systems, where the heat included in the
354 sewage water is collected from each apartment building. The nominal heat recovery capacity of each sewage water
355 recovery unit is approximately 4 kW resulting in a total capacity of 16 kW. The recovered heat is utilized in the brine
356 collector system of the GSHP system by installing the heat recovery system in parallel with the boreholes. The GSHP
357 system includes 16 boreholes with each borehole being 285 meters deep. The installation of the GSHP system setup is
358 presented in Fig. 7.

Fig. 7. The installation of the GSHP system setup including two 90 kW GSHP units and two four-tank heat storage combinations for step-based heating of DHW. Phased charging and discharging of tanks 1 and 2 are controlled by 3 exchange valves (below left).

3. Results

3.1. Experimental analysis

The performance measurements of the experimental analysis included a total of three 16-hour measurement cycles for both of the studied heat pump applications. The average temperatures of both the DHW and the DHW heat storage tanks for the conventional and new developed GSHP systems are presented in Fig. 8. The temperatures presented in Fig. 8 represent the measured average temperatures of one 16-hour measurement cycle. The average inlet temperature of domestic cold water was approximately 7.5 °C in all of the measurement cycles. The average inlet temperature of brine was approximately 1.6 °C for the novel GSHP system and approximately 2.5 °C for the conventional

371 heat pump system. The difference in the average brine inlet temperatures was occurring due to the measurement ar-
 372 rangements.

373

374

375 **Fig. 8.** Average outlet temperatures of DHW with the conventional (above left) and new developed (above right) GSHP systems.
 376 Average temperatures of the DHW heat storage tanks with the conventional (below left) and new developed (below right) GSHP
 377 systems.

378

379 While the targeted daily DHW consumption was 4200 dm³ for both of the studied applications, the measured aver-
 380 age DHW consumptions of the measurement cycles were:

381

- conventional GSHP system application: 4394 dm³/day;

382

- new developed GSHP system application: 4288 dm³/day.

383

384 The average cumulative heating energy produced and the average cumulative electrical energy consumed by the
 385 studied heat pump systems during a 16-hour measurement cycle are presented in Fig. 9. As presented in Fig. 9, the
 386 conventional heat pump system produced more energy than the new system due to the fact that approximately 2.5%
 387 more DHW was heated by the system. The main reason for the higher DHW consumption was the lower pressure loss
 388 of the DHW system in the conventional heat pump system configuration. When DHW was consumed according to the
 389 time periods presented in Table 2, e.g. 525 or 960 seconds, higher water volume was flowing through the conventional
 390 heat pump configuration due to the lower pressure loss of the DHW connection principle (see Fig. 5). In addition, more
 energy was also produced due to higher accumulator heat losses. Furthermore, the temperature of DHW was 1–4.5 °C

391 lower than the 55 °C temperature requirement with the developed heat pump system during several consumption
 392 cycles of the average 16-hour measurement cycle (see Fig. 8). An energy efficient control strategy (charging and dis-
 393 charging cycles of tanks 1 and 2) of the developed GSHP system was used in the measurements of the experimental
 394 analysis, which resulted in the situation that the target temperature of DHW was not achieved in all of the 21 consump-
 395 tion cycles. The main results related to the energy efficiency of the studied heat pump systems are presented in Table 3.
 396

397
 398 **Fig. 9.** The average heating energies produced and electrical energies consumed by the conventional (left) and new developed (right)
 399 heat pump systems in laboratory measurements.
 400

401 **Table 3.** Measured energy flows and calculated Seasonal Performance Factors of the studied heat pump systems.

Heat pump system	$Q_{\text{condenser}}$ [kWh]	$Q_{\text{superheat}}$ [kWh]	$Q_{\text{heat,total}}$ [kWh]	$Q_{\text{electricity,HP+aux.}}$ [kWh]	SPF
Conventional	260	-	260	103	2.52
Novel step-based	210	30	240	64	3.75

402
 403 The new developed system also provides significant improvements in the dynamic operation of the heat pump
 404 unit. The average number of start-ups of the compressor decreased by approximately 40% with the new system com-
 405 pared to the conventional system. In addition, the measured average condensing pressures were approximately 28%
 406 lower with the new system. These factors reduce the stress and improve the longevity of the compressor. The average
 407 operating times of the compressors and the average evaporating and condensing pressures for both of the studied heat
 408 pump applications are presented in Fig. 10.

409 According to the measurement results of the test cycles, the repeatability of the results was good with both of the
 410 studied heat pump systems. The average difference in the measured parameters was 1–3% between the three meas-
 411 urement cycles. Furthermore, the temperature trends, operating times and the performance data of the measurement
 412 cycles followed the trends, data and features presented in Figs. 8-10 and Table 3 accurately with both of the studied
 413 systems. The control strategies of the systems were remained unchanged between the measurement cycles to deter-

mine if similar results were achievable from several cycles and to exclude any potential cycle-specific measurement errors or operation problems.

Fig. 10. Average operating times of the compressors for the conventional (above left) and new developed (above right) GSHP systems. Average evaporating and condensing pressures for the conventional (below left) and new developed (below right) GSHP systems.

3.2. Computational analysis

The total DHW consumption was selected to be 4288 dm³/day according to the laboratory measurements in all of the studied simulation cases. The inlet temperatures of both brine and domestic cold water used in the simulation models were also adjusted to correspond to the temperature data of the experimental analysis. The simulated temperatures of DHW in different simulation cases are presented in Fig. 11. The simulated temperatures of the DHW storage tanks for the step-based heat pump concept are presented in Fig. 12, respectively. Fig. 12 presents the temperatures of the DHW storage tanks from the regular control strategy simulation case. As in the experimental analysis, the temperature set point of the DHW heat storage tank was 60 °C in the conventional heat pump system simulation cases.

In general, the simulation results showed good agreement with the measured performance and temperature data of the experimental analysis. As demonstrated in Figs. 11 and 12, the control strategy of the step-based heating process of DHW has impact on both the outlet temperature of DHW and the energy performance of the heat pump system. As

the priority of the DHW system is to deliver 55 °C water to occupants, the control strategy should be adjusted and optimized so that the temperature requirement is achieved in all circumstances with as high energy efficiency as possible.

Fig. 11. Simulated outlet temperatures of DHW with the conventional heat pump system (above left) and new developed GSHP system utilizing regular (above right), tight (below left) and loose (below right) control strategies.

The temperature profiles of tanks 1-4 with the loose and tight control strategies were similar with the temperature profile of the regular control strategy presented in Fig. 12. The main difference was that the temperatures of the tanks were a little higher with the tight control strategy and a little lower with the loose control strategy, respectively. The heat pump system also responded either faster (tight strategy) or slower (loose strategy) to the reduction in temperatures of the tanks in order to maintain sufficiently high temperatures to supply 55 °C DHW throughout the simulation period. Tanks 1 (or tank 2 when the heating process is switched), 3 and 4 are connected in series so that the outlet water of tank 1 (or 2) is the inlet water of tank 3 and the outlet water of tank 3 is the inlet water of tank 4, respectively. The outlet water of tank 4 is domestic hot water supplied to the DHW system. There is also a 3-way control valve after tank 4 (see Fig. 5) to mix domestic cold water to the outlet water of tank 4, if needed, so that the supply temperature of DHW is 55 °C.

Fig. 12. Simulated temperatures of DHW storage tanks 1 (above left), 2 (above right), 3 (below left) and 4 (below right) with the new developed GSHP system utilizing the regular control strategy.

The last simulation case studied a scenario, where a more efficient heat exchanger is used for heating of DHW (see Fig. 5). The minimum temperature difference (pinch point) of the counter-flow heat exchanger was reduced to 2.0 K from the original 3.5 K, which resulted in an average of 1.5 °C lower inlet and outlet temperatures (heating/condenser side) of the heat pump unit. According to preliminary estimations based on previous studies and literature, the authors estimated that the overall efficiency of the heat pump unit should increase by approximately 4% due to the lower inlet and outlet temperature levels. The additional simulation case using the calibrated heat pump simulation model was conducted to confirm the estimation and to provide further information on the potential improvements in energy efficiency and operation of the developed step-based heat pump system concept. The simulation results utilizing the more efficient heat exchanger showed improvements in energy performance of the heat pump system with all the studied control strategies. According to the simulation results, the average improvement in the energy performance of the

465 developed system was approximately +2.1%, when the 1.5 °C lower inlet and outlet temperature levels were applied
 466 and other parameters remained unchanged in the simulations.

467 A summary of the simulation results related to the energy efficiency of the studied heat pump systems and control
 468 strategies is presented in Table 4. The heat losses of the DHW storage tanks were also simulated and taken into account
 469 in the calculation of the overall energy performance of the systems. The average thermal transmittance of the tank
 470 coatings, including the top and bottom parts of the tanks, was 0.2 W/(m² K) and the room air temperature, where the
 471 storage tanks are located, was set to 21.0 °C in all of the simulation models.

472

473 **Table 4.** Simulated energy flows and calculated Seasonal Performance Factors of the studied heat pump systems.

Simulated heat pump system	Q _{DHW} [kWh]	Q _{losses,storage} [kWh]	Q _{heat,total} [kWh]	Q _{electricity,HP+aux.} [kWh]	SPF
Conventional system	236.9	1.4	238.3	92.8	2.57
New step-based system					
- loose control strategy	235.6	1.0	236.6	62.5	3.79
- regular control strategy	236.5	1.0	237.5	63.4	3.75
- tight control strategy	236.9	1.1	238.0	64.7	3.68
New step-based system equipped with more efficient heat exchanger					
- loose control strategy	235.6	0.9	236.5	61.3	3.86
- regular control strategy	236.5	1.0	237.5	62.0	3.83
- tight control strategy	236.9	1.0	237.9	63.3	3.76

474

475 The primary simulation model calibration was carried out according to the COP-values of the selected Gebwell Tau-
 476 rus 90 heat pump unit. Further investigations were also conducted to validate, if the operation of the developed simula-
 477 tion model is accurate and realistic compared to the measured performance of the actual heat pump system studied in
 478 the experimental analysis. According to the measurement results of the experimental analysis, the novel step-based
 479 GSHP system produced approximately 240 kWh of heating energy during the measurement cycles and consumed ap-
 480 proximately 64 kWh of electrical energy during the cycles (see Table 3), respectively. This results in a 3.75 SPF of the
 481 system. The simulated system produced approximately 238 kWh of heating energy and consumed approximately 63
 482 kWh of electrical energy with the regular control strategy, resulting in a 3.75 SPF of the system as well. From the energy
 483 performance point of view, the simulation model of the developed GSHP system shows very good agreement with the
 484 measured performance of the system. The formation of produced heating energy and consumed electrical energy dur-
 485 ing the simulation period is also similar with the measured energy performance data presented in Fig. 9.

486 The temperature data of DHW is also similar between the simulated system and the measured actual system (see
 487 Figs. 8 and 11). However, a little variation between the simulated and measured systems can be found from the tem-

488 perature data of the DHW storage tanks 1-4. When the temperature data of the DHW storage tanks is compared during
489 the 16-hour measurement cycles, the following differences can be identified:

- 490 • the average temperature of tank 1: 45.3 °C (experimental analysis), 37.4 °C (computational analysis);
- 491 • the average temperature of tank 2: 48.1 °C (experimental analysis), 43.9 °C (computational analysis);
- 492 • the average temperature of tank 3: 52.4 °C (experimental analysis), 52.7 °C (computational analysis);
- 493 • the average temperature of tank 4: 56.3 °C (experimental analysis), 57.2 °C (computational analysis).

494 In addition to the differences in the average temperatures of the DHW tanks, some differences can also be identi-
495 fied in the operation of the simulated and the measured GSHP systems. The charging and discharging cycles between
496 tanks 1 and 2 are switched more frequently with the simulated system than with the actual system. In fact, the simulat-
497 ed system switches between tanks 1 and 2 in almost every DHW consumption cycle (see Fig. 12), whereas the actual
498 system operates with tank 1 for the majority of the measurement cycles (see Fig. 8). This is also the main reason why
499 the average temperatures of tanks 1 and 2 are lower with the simulated system compared to the measured tempera-
500 ture data of the actual system. The average temperatures of tanks 3 and 4 are much closer to the measured tempera-
501 ture data of the experimental analysis, respectively. There are two main factors explaining the difference in the opera-
502 tion of the simulated and the actual GSHP systems:

- 503 • the temperature of tank 1 is allowed to decrease a little bit lower during the discharging period in the
504 simulated system. The temperature of tank 1 is decreased to approximately 10-13 °C in majority of the
505 DHW consumption cycles with the simulated system, whereas the temperature of tank 1 is decreased to
506 approximately 18 °C with the measured system of the experimental analysis;
- 507 • the operation of heat pump units equipped with a separate superheat circuit cannot be accurately mod-
508 elled with the current version of IDA ICE. While the modelling of heat pump systems is accurate, when en-
509 ergy performance and basic operation of the systems are discussed, simulation of systems producing su-
510 perheat at 65-75 °C temperature level and condensate at 50-55 °C temperature level simultaneously is not
511 possible by using the default brine-to-water heat pump models included in IDA ICE. The heat pump mod-
512 els can be calibrated to correspond to the performance and operation of high temperature heat pump ap-
513 plications, but different temperature levels cannot be produced simultaneously in the simulation. This in-
514 evitably leads to the situation, where the control actions of the simulated heat pump systems are faster,
515 e.g. switching the charging and discharging cycles between tanks 1-4 with the developed GSHP system ap-
516 plication, than the control actions of the corresponding actual heat pump systems.

517

518

3.3. Performance analysis in existing buildings

519 The heat pump system has been operating for a little over 6 months since December 2016 and the system has both
 520 annual and cumulative energy metering for produced heating energy and consumed electrical energy. The energy me-
 521 tering consists of the same calibrated Kamstrup measurement equipment as presented in Table 1. All 140 apartments
 522 included in the studied case buildings have been fully occupied since the beginning of April 2017. The measured cumu-
 523 lative performance data of the GSHP system operating in the case buildings is presented in Tables 5 and 6. Table 5 pre-
 524 sents the cumulative performance data and Seasonal Performance Factors on 20.4.2017 and Table 6 presents the cor-
 525 responding data on 16.6.2017. The SPF of the heat pump units and the overall performance of the system were calcu-
 526 lated using Eq. (1) and the definition and system boundaries described in section 2.2.1.

527

528 **Table 5.** Measured cumulative energy flows and calculated Seasonal Performance Factors of the GSHP system operating in the stud-
 529 ied case buildings on 20.4.2017.

Heat pump unit	$Q_{\text{condenser}}$ [MWh]	$Q_{\text{superheat}}$ [MWh]	$Q_{\text{heat,total}}$ [MWh]	$Q_{\text{electricity,HP+aux.}}$ [MWh]	SPF
GSHP 1	267.2	7.2	274.4	50.3	5.46
GSHP 2	95.4	7.8	103.2	33.9	3.04
Overall performance of the system	362.6	15.0	377.6	84.2	4.48

530

531 **Table 6.** Measured cumulative energy flows and calculated Seasonal Performance Factors of the GSHP system operating in the stud-
 532 ied case buildings on 16.6.2017.

Heat pump unit	$Q_{\text{condenser}}$ [MWh]	$Q_{\text{superheat}}$ [MWh]	$Q_{\text{heat,total}}$ [MWh]	$Q_{\text{electricity,HP+aux.}}$ [MWh]	SPF
GSHP 1	313.4	11.1	324.5	62.7	5.18
GSHP 2	125.6	13.2	138.8	44.8	3.10
Overall performance of the system	439.0	24.3	463.3	107.5	4.31

533

534 The energy performance of the system is carefully monitored and continuous updates are conducted to optimize
 535 both the efficiency and overall operation of the system, as the real dynamic operation of a large scale GSHP system with
 536 over 100 apartments differs from a carefully monitored laboratory testing arrangements. Tables 5 and 6 indicate that
 537 heat pump unit 2 has been primarily utilized for heating of DHW and heat pump unit 1 has been primarily utilized for
 538 heating of room spaces via the hydronic floor heating system. The control strategy for heating of room spaces has been
 539 implemented so that heat pump unit 2 will start to heat the room spaces, if the full capacity of heat pump unit 1 is not
 540 sufficient to maintain the required supply water temperature of the floor heating system. However, as the heat pump
 541 units have been connected in parallel, both of them are required to heat DHW, when higher consumption peaks occur.

542 Furthermore, the heating priority of the heat pump units can also be switched for autumn so that heat pump unit 1 will
543 be prioritized for heating of DHW and heat pump unit 2 for heating of room spaces to balance the annual operating
544 times of the units.

545 As presented in Tables 5 and 6, the full potential of the developed system has not been reached yet, when the
546 measured cumulative performance data is compared to the measured performance data of the experimental analysis.
547 The main reason for this is that the installed GSHP system has been prioritized to heat DHW so that there is always
548 DHW available for the occupants and that the 55 °C temperature requirement of the current Finnish building regula-
549 tions is also achieved in all circumstances. However, the performance of the studied system will likely improve over
550 time when the charging and discharging cycles of the step-based heating tanks have been optimized, as it takes time to
551 determine a stable and continuous DHW consumption profile in new multi-family apartment buildings with a large
552 number of apartments, where previous DHW consumption data is not available. As the results of the experimental
553 analysis demonstrated, as high as 3.7-3.85 Seasonal Performance Factors are achievable in the heating process of DHW
554 with an optimally controlled GSHP system utilizing the step-based heating of DHW, depending on the specific case con-
555 ditions, such as temperatures of domestic cold water and brine inlet.

556 An essential factor affecting the performance of the GSHP systems in real case conditions is the DHW circulation
557 system. In the studied case buildings, the continuous heat demand of the DHW circulation system is approximately 3
558 kW with a high temperature level requirement of 55/50 °C. This means that the heat pump units must also heat tanks 3
559 and 4 of the system separately at regular intervals even though there is no actual DHW consumption, as the total vol-
560 ume of tanks 3 and 4 is 1080 dm³ for the two GSHP units meaning that the temperature of the tanks decreases relative-
561 ly fast even when there is no actual DHW consumption occurring, e.g. during night times.

562 Based on the operation of the studied GSHP system so far, the authors estimate that there is approximately a 10%
563 improvement potential in the overall energy performance of the system and approximately 12-15% energy efficiency
564 improvement potential in the heating process of DHW, after the optimum dynamic control strategy has been deter-
565 mined and fine-tuned. An important factor of the case study is also to monitor the adequacy of DHW, as one step-based
566 DHW heating concept with 4 tanks has a total volume of 1080 dm³, which is sufficient for approximately 50 apartments
567 in typical operating conditions. While there are two step-based systems installed in the case buildings, the total number
568 of apartments is 140, which means that the overall capacity of the installed systems is definitely at the upper limit. This
569 is also the main reason why the overall energy performance of the system has not reached its full potential yet, as the
570 adequacy of DHW has been ensured by maintaining a little higher temperatures in the DHW storage tanks than what

571 would be necessary after a stable DHW consumption profile has been determined and the control strategy of the step-
572 based DHW heating process has been optimized.

573 The sewage water heat recovery system installed in the studied buildings also improves the overall energy perfor-
574 mance of the GSHP system. The total nominal heat recovery capacity of the system is approximately 16 kW, but it is
575 only achieved when domestic water consumption occurs in the buildings and a significant proportion of the water con-
576 sumption is domestic hot water. This means that the energy flow from the waste water heat recovery system is not
577 continuous and it is not occurring during night times. However, as the collected heat is utilized in the brine collector
578 system of the GSHP system, where the average temperature level of the brine is typically below 5 °C, some energy can
579 also be collected from the domestic cold water, e.g. flush water of toilets, flowing through the heat recovery units. This
580 means that the overall efficiency of the sewage water heat recovery system is high, when maximum collected heat
581 energy is discussed.

582 According to the temperature measurements carried out in the GSHP system, the average brine inlet temperature
583 of the system has been approximately 4.5 °C since the commissioning of the system. While the sewage water heat re-
584 covery system definitely has some impact on the brine inlet temperature and on the overall performance of the GSHP
585 system, it is difficult to determine the precise effect of the actual heat recovery system on the overall performance of
586 the system, as the heat collected from the heat recovery units is transferred to the boreholes when the GSHP units are
587 off. Furthermore, the groundwater flow inside the boreholes also affects the temperature of the brine meaning that the
588 collected heat of the sewage water heat recovery system might not be fully utilized, as part of the collected heat is
589 transferred to the bedrock due to strong flow of groundwater. In addition, the maximum nominal capacity of the bore-
590 holes (16 x 285 m) is approximately 150 kW compared to the 16 kW nominal capacity of the sewage water heat recov-
591 ery system, meaning that the sewage water heat recovery system has a relatively low effect on the brine inlet tempera-
592 ture, if the GSHP system is operating at full capacity. However, in actual operating conditions the effect of the heat
593 recovery system on the brine inlet temperature is more significant, as the GSHP system rarely operates at full capacity.

594

595 **4. Discussion**

596 As the final nearly zero-energy building regulations of the recast EPBD come into force in 2020 and the energy per-
597 formance of the building envelope and technical systems of new buildings improves, energy efficient heating of domes-
598 tic hot water will play a significant role in improving the overall energy performance and decreasing the delivered ener-

599 gy consumption of nZEBs. As the energy efficiency of both the building envelope and building services systems im-
600 proves, heating of DHW can easily account for more than 50-60% of the total heating energy demand in new low-
601 energy multi-family apartment buildings. Furthermore, the recast EPBD requires that the reduced energy demand of
602 nZEBs should be covered to a significant extent by renewable energy sources. Typically heat pump systems, such as
603 ground source heat pump systems, are one of the best applications to utilize on-site renewable energy sources, when
604 energy performance and cost-effectiveness are discussed. To improve the efficiency of the GSHP systems especially in
605 heating of DHW, majority of the previous studies have focused on research related to solar assisted heat pumps in or-
606 der to improve the efficiency of the heat pump unit in producing the higher temperature levels required for heating of
607 DHW. However, in many cases the investment in solar thermal energy is not the most economically viable measure,
608 especially in cold climate conditions, when cost-optimality is discussed.

609 The results of this study indicate that the developed method for energy efficient heating of DHW is functional and
610 has a significant energy saving potential compared to the conventionally used GSHP system applications. The measure-
611 ment results of the experimental analysis demonstrated that up to 49% improvement in energy efficiency of the heat
612 pump system was achieved with the developed heat pump application compared to the conventionally used applica-
613 tion. While the measurement results between the conventional and new developed systems were not fully comparable
614 due to the fact that the temperatures of DHW were below the target temperature (55 °C) in a few measurement cycles
615 with the developed system, the difference in efficiency is so substantial that it can be concluded that the developed
616 system operates significantly more efficiently in heating of DHW than the conventional system. However, the average
617 temperature of the brine inlet was approximately 1.0 °C higher in the measurements with the conventional system
618 resulting in better performance, respectively. Furthermore, the target temperature of DHW can be easily achieved with
619 the step-based heating concept by increasing the temperatures of the DHW storage tanks or by further fine-tuning the
620 control parameters of the system.

621 According to the results of the experimental analysis, other essential benefits of the developed system were the
622 significantly longer operating times and lower condensing pressure levels of the studied heat pump unit resulting in
623 improved longevity and reliability of the system. The start-ups of the heat pump unit decreased by approximately 40%
624 with the step-based heating process application. As a result of the lower condensing pressure levels, the average tem-
625 perature level of the superheated gas of the refrigerant was approximately 27% lower during the measurement cycles
626 with the new developed system. While there are more DHW storage tanks and control automation included in the de-
627 veloped system, which inevitably increases the maintenance need of the system in actual case buildings, the finalized

operation principle of the system has been simplified as much as possible by using only the necessary control equipment. In addition, the developed system can be combined with all the major commercial heat pump makes and models that include separate superheat recovery of the refrigerant. The developed system can also be implemented in existing GSHP systems by installing the additional DHW storage tanks, heat exchanger and control equipment.

According to Tom Allen Seneca Oy, the investment cost of the developed GSHP system is approximately 5-10% higher than the investment cost of a corresponding GSHP system equipped with the conventional DHW heating operation. In an apartment building with 100 apartments, the initial investment cost of the developed system is approximately 5% higher than the investment cost of the conventional system and in an apartment building with 30 apartments, the investment cost of the developed system is approximately 10% higher, respectively. For comparison, the technical space requirement for one four-tank configuration is approximately 1.5 floor-m², whereas the technical space requirement for the corresponding conventional DHW heat storage configuration with one 1.3 m³ tank is approximately 0.9 floor-m². Other components of both the conventional and the developed GSHP systems require approximately the same amount of technical space, so the increased technical space requirement of the developed concept is approximately 0.6 m² for each four-tank configuration installed in the building. Due to the relatively complex installation principle of the developed system, the on-site installations of the system have also been minimized to ensure correct and fast installations. Each four-tank assembly comes pre-installed from the Gebwell factory so that all the main components, such as the heat exchanger, exchange and one-way valves, temperature sensors, insulations etc., and connections have been installed at the Gebwell factory. In addition, the operation of the main components has also been tested at the factory so that the on-site installation and commissioning of the systems in actual buildings is as simple and efficient as possible. In addition to the more complex and detailed installation of the developed system, it is essential to notice that the increased maintenance need of the system also requires relatively comprehensive metering and careful monitoring of the system.

The results of the computational analysis showed good agreement with the results of the experimental analysis. The developed dynamic simulation model corresponded to the actual system well, when performance, operation, controllability and accuracy are discussed. The simulated control strategies (loose, regular and tight) had effect on both the temperature of DHW and energy performance of the heat pump system demonstrating that the selected simulation tool can be used to accurately simulate dynamic operation of complex and versatile control systems. The studied control strategies also corresponded accurately to the measured temperature and performance data of the experimental analysis. The developed simulation model can also be utilized in future studies and projects, where the economic viabil-

ity of the step-based heating of DHW is required to be assessed in different building types and techno-economic environments. The simulation model can be further developed to maximize the cost-effectiveness of GSHP systems utilizing the step-based heating of DHW by optimizing the operation of the system according to electricity spot-prices and weather forecasts, or by developing cost-optimal demand response strategies for new and existing multi-family apartment buildings. While the developed simulation model was not implemented in the case study, the authors expect a good agreement between the simulated and actual performance of the system based on the results of the simulations and experimental analysis and on the results of recent studies conducted by the authors [49,50], where the performance of simulated and measured heat pump systems is compared.

According to the simulation results, using a more efficient counter-flow heat exchanger improves the energy performance of the step-based heating process by approximately 2%. However, based on the operation and measured performance of the studied heat pump unit in different operating conditions, the improvement potential is expected to be approximately 3.5–4% with the actual system. When measurement results are available from an actual test case, the simulation model can be fine-tuned and calibrated according to the actual performance data of the upgraded system. The counter-flow heat exchanger included in the studied GSHP concept of the experimental analysis has already been replaced to the more efficient heat exchanger and it will be used in all new case projects, where a GSHP system utilizing the step-based heating of DHW is installed.

According to the preliminary results of the case study, the performance of the installed GSHP system operating in real case conditions has not reached its full energy saving potential yet. A significant aspect affecting the performance of the system in actual case buildings is the dimensioning and heat demand of the DHW circulation system, which was excluded from the experimental analysis. High temperature requirement, e.g. 55/50 °C or even 58/55 °C, and continuous heat demand of the DHW circulation system decrease the performance of the heat pump unit, increase the number of start-ups of the unit and force the system to operate shorter time periods especially during night times, when there is little or no actual DHW consumption occurring in the building. If the heat demand of the DHW circulation system is e.g. 3-5 kW and the smallest power output of the heat pump unit is 30-40 kW, the operating conditions are challenging even with a properly dimensioned heat storage system. The performance and operation of the GSHP system installed in the studied case buildings will be constantly monitored and improvements in the control strategy, heating set points and other operating parameters are continuously performed in order to maximize the energy performance of the system without sacrificing the adequacy of DHW. The developed system has already been installed in over 20 multi-family apartment buildings, majority of the buildings are apartment buildings which have been renovated, since 03/2016 and

all of the installed systems have performed efficiently after the control strategy of the system has been fine-tuned and synchronized with the individual DHW consumption profile of the building. In general, this means that the DHW consumption profile of the building must first be determined and then co-optimized with the system performance of the GSHP system.

Based on the results, findings and limitations of the study, future research is also required. The rapid development of modern energy systems requires that reliable and validated simulation methods are needed to simulate the performance and operation of different renewable energy sources and complex hybrid energy systems. The developed simulation model was validated in the study demonstrating good agreement with the measured performance data of the experimental analysis, but it is recommended to be further developed for larger scale studies, where the developed system is combined with other energy production technologies, such as solar thermal collectors, district heating or gas boiler. Future research is also required to be carried out with the installed GSHP systems by carefully monitoring the performance and operation of the installed systems and by implementing machine learning features in the building automation control unit, which controls and operates the entire GSHP system concept. Based on the recent development of machine learning applications used in the building sector, implementation of modern machine learning features has a high potential to significantly improve the operation and efficiency of processes such as the phased step-based heating of DHW developed in the study. Functional machine learning applications would make the co-optimization of individual DHW consumption profiles of different buildings and the performance and operation of the GSHP systems automatic and highly efficient. Future research is also recommended to be conducted by installing and monitoring the developed GSHP systems in buildings, where both the techno-economic environment and climate conditions are different. An interesting aspect for future research is also to determine the improvements of the sewage water heat recovery system in the energy performance of the developed GSHP concept by conducting individual laboratory or field measurements. Different heat storage alternatives, such as applicable tank-in-tank solutions, could also be considered for improved energy performance and reduced heat losses in the future development of the concept.

5. Conclusions

The objective of the study was to develop a novel method for energy efficient heating of DHW with a GSHP system and to study and compare the performance and operation of the developed system to a conventional GSHP system application in multi-family apartment buildings by conducting computational and experimental performance analyses. Based on the findings of the study, the following conclusions can be drawn:

- the performance of the developed method was successfully tested and validated using both computational and experimental analyses, the results of the analyses showed good agreement with each other;
- the energy efficiency of the developed method was up to 45–50% higher in the heating process of DHW than the energy efficiency of the conventionally used method. In addition, the start-ups of the heat pump unit also decreased by approximately 40% with the developed application resulting in longer operating times and reduced stress of the compressor unit;
- the SPF of the conventional GSHP application was approximately 2.5–2.6 in the heating process of DHW, whereas the SPF of the developed GSHP application was approximately 3.7–3.8 delivering a significant improvement in energy efficiency, when the outlet temperature requirement of DHW was 55 °C and the inlet temperature of domestic cold water was 7.5 °C;
- a functional and accurate dynamic simulation model was developed and validated in the computational analysis, the developed model can be used for performance assessment and optimization of control strategies of GSHP systems utilizing the developed method;
- the case study demonstrated that the developed method is also effective and functional in actual dynamic case conditions. However, the case study also revealed potential limitations and inefficiencies related to the operation of the GSHP system, which are recommended to be addressed in the future development of the concept.

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