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Published in: E3S Web of Conferences

DOI: 10.1051/e3sconf/202017216012

Published: 30/06/2020

Document Version Publisher's PDF, also known as Version of record

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Please cite the original version:

Fadejev, J., Simson, R., Kesti, J., & Kurnitski, J. (2020). Measured and simulated energy performance of OLK NZEB with heat pump and energy piles in Hämeenlinna. *E3S Web of Conferences*, *172*, Article 16012. https://doi.org/10.1051/e3sconf/202017216012

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Measured and simulated energy performance of OLK NZEB with heat pump and energy piles in Hämeenlinna

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Abstract. In this work, measured energy use of the building space heating, ventilation supply air heating, appliances and lighting is compared against simulated energy use modelled in IDA ICE. As built energy need and detailed measured input data is applied in building model calibration procedure. Calibrated building model energy performance is studied in both measured and test reference year climate conditions. Previously modelled as built plant automation and implemented control logics are compared against measured. Geothermal plant in this study consists of heat pump, solar collectors, boreholes and energy piles. Heat pump SCOP estimated by post processing according to heat pump manufacturer's performance map is compared against measured SCOP on the monthly basis. Opinion on actual plant operation is given and energy performance improvement potential is quantified. Important parameters for successful building model calibration are presented. Building compliance with Finland NZEB requirements are assessed. The results show good match with measured energy use after the model calibration.

1 Introduction

According to European Parliament directive 2010/31/EU [1] all new buildings built from January 2021 are to comply with nearly zero-energy buildings (NZEB) national requirements. NZEB requirements for public buildings are already in force. Meeting NZEB requirements considers application of renewable energy sources such as geothermal and solar in the design. Geothermal energy utilization is mainly performed with a ground source heat pump (GSHP) and according to a review on worldwide application of geothermal energy [2] total installed worldwide GSHPs capacity has grown 2.15 times in the period of 2005 to 2010 and 45% from 2010 to 2015, application of GSHP is registered in 82 countries around the globe.

Annual GSHP SCOP values up to 4.5 and overall geothermal plant SCOP values up to 3.9 (including control and distribution losses) were obtained based on measured performance of actual GSHP installations [3-4]. In most cases, operation of a heat pump is accompanied by the unbalanced geothermal energy extraction/injection that leads to a significant loss in long-term operation performance [5]. To maintain stable long-term operation of GSHP plant and improve geothermal energy yield along with seasonal coefficient of performance (SCOP), a source of thermal storage to be considered in the plant design. Reda [6] studied the benefits of solar thermal storage numerically in a GSHP plant with a borehole field type ground heat exchanger (GHE), where application of solar thermal storage

helped in improvement of GSHP plant SCOP from 1.6 to 3.0. Allaerts et al. [7] has modelled the performance of a GSHP plant with dual borehole field and active air source storage in TRNSYS, where cooling tower i.e. dry cooler was applied as a thermal storage source. According to results of such thermal storage application, overall size of borehole field was reduced by 47% compared to the same capacity single borehole field plant without thermal storage.

GSHP plant performance is depended on the type of GHE considered in the plant design. Typical closed loop GHEs are classified by the position of installation horizontal and vertical. Horizontal GHE is generally cheaper to install compared to vertical GHE, however requires more land area for the installation. In buildings with limited land area, vertical GHE in form of a borehole reaching up to 400 meters in depth might be a solution instead of horizontal GHE installation. Though, drilling very deep boreholes might be not only very expensive, but also drilling depth might be limited by the government regulations in the region of interest. In this case, field of multiple shorter boreholes (not exceeding the drilling depth limit) spaced at known distance to each other might be considered as a GHE alternative.

In buildings with pile foundations, installation of heat exchange piping into foundations piles enables the foundation piles to perform as a ground heat exchanger similarly to previously described field of boreholes. Geothermal pile foundations are known also as geothermal energy piles [8]. As the installation of heat exchange piping into foundation pile compared to the

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drilling of a new borehole is much cheaper, energy piles tend to be a very cost effective GHE solution. As the layout of energy piles is generally defined by the foundation plan, thermal interferences between closely located adjacent piles appear. Thermal interferences may also appear in field of boreholes, depending on the spacing between them. Sizing and assessment of borehole field or energy piles performance is generally carried out with help of numerical modelling regarding which more detailed aspects are described in previously conducted study by Fadejev and Kurnitski [9].

From the perspective of thermal storage application, not all types of GHEs would benefit from a thermal storage due to varying thermal losses intensity, GHE storage capacity and peak heat extraction/rejection rates. To consolidate previous statement, assuming that the same exact amount of heat is stored in a single borehole GHE compared to the same amount of heat stored in a GHE consisting of multiple boreholes and total length of single borehole is equal to the sum of multiple boreholes, field of multiple boreholes would be capable of extracting more heat compared to a single pile due to rejected storage heat of boreholes located in the centre of the field still can be utilized by the boreholes located at the edges of borehole field in the process of storage heat dissipation.

In cold climate regions, where indoor climate conditions are generally ensured with heating, operation of GSHP plant during the heating season cools down the ground surrounding GHE. Installing a "free cooling" heat exchanger between the ground loop and cooling system buffer tank allows to partly cover buildings cooling demand via direct "free cooling".

Considering all abovementioned benefits of GSHP plant, it appears to be very attractive heat source option in NZEB design. Especially in regions, where no heat sources such as district heating with low primary energy conversion factors are available and only electricity energy source is present.

The present study is the continuation of previously conducted research [10] on topic of design and energy performance modelling of geothermal heat pump plant in commercial hall-type building OLK NZEB located in Häämenlinna, Finland. This study covers the analysis of OLK NZEB measured energy use for the period of 01.02.18-01.31.19 along with free cooling impact on indoor climate and geothermal heat pump plant energy performance assessment. Simulated energy use of case 14 from [10] corresponding to energy use of as built initial design case is compared against measured energy use of room unit heat, air handling unit (AHU) heating coil heat, lighting and equipment electricity. As measured outdoor climate conditions and actual building use differ from test reference year (TRY) climate [12] and initial design building use, a building model calibration was conducted in IDA ICE applying detailed hourly based measured data and as built documentation parameters. Building model is being calibrated on the monthly basis and is further applied in TRY climate to assess the impact on building energy performance and quantify the modelling accuracy. Calibrated building model allows further research in terms of coupling it with detailed modelled geothermal plant model and assessment of different parameters impact such as indoor temperature setpoints, AHU setpoints on building energy performance. Measured case conditions are compared against initial design intent and suggestions regarding improving the energy performance are provided. Building model calibration procedure is described in detail and suggestions on required measured parameters by building monitoring/logging system are given to allow successful building model calibration in IDA ICE.

Further results provide an insight on geothermal heat pump plant seasonal coefficients of performance (SCOP) for simulated and measured cases. Energy performance values (EPV) of each case are presented, compliance with Finland NZEB requirements is assessed. Measured heat pump SCOP is compared against one estimated by post processing according to heat pump manufacturer's performance map. Opinion regarding geothermal heat pump plant operation and suggestions on improving its energy performance with quantified expected performance increase are presented.

2 Methods

Measured data for period of 01.02.18-01.31.19 with a hour timestep resolution was obtained from OLK NZEB building monitoring/logging system, processed, then analyzed in Excel. For building model calibration procedure, measured data was converted to input files that comply with IDA ICE simulation environment. The modelling in IDA ICE was performed in advanced level interface, where user can manually edit connections between model components, edit and log model specific parameters, observe models code. A detailed OLK NZEB building model was prepared in IDA ICE based on the as built documentation with accountancy for available measured input data. Building model calibration was performed on the monthly basis with the goal to achieve perfect fit against measured AHU heat and room unit heat, while outdoor climate conditions correspond to measured climate, indoor air temperatures



Fig. 1. (a) Initial design model in IDA ICE. (b) OLK NZEB in Hämeenlinna. (c) Building calibration model in IDA ICE

and AHU setpoints are are defined as hourly based measured data, internal gains are AHU operation schedules are modified due to not being either measured and/or include additional separately cooled equipment electricity use. Simplified heat pump SCOP calculation model based on the actual heat pump performance map and measured evaporator/condenser inlet/outlet temperatures was completed in Excel mainly using second-degree polynomial equations.

2.1 Building model data

Compared to initial design model on Fig. 1 (a), a more detailed room-based model of OLK NZEB depicted on Fig. 1 (c) has been modelled in IDA ICE based on as built documentation for building model calibration procedure. Table 1 presents a detailed overview of general parameters describing the building model.

Table 1. Building descriptive parameters

Descriptive parameter	Value
Location	Finland
Net floor area, m ²	1496.5
External walls area, $U = 0.16 \text{ W/(m^2 K)}, \text{ m}^2$	1201
Roof area, $U = 0.12 \text{ W/(m2 K)}, \text{ m}^2$	1467
External floor area, $U = 0.14 \text{ W/(m2 K)}$, m2	1496.5
Windows area, SHGC = 0.33 , U = 0.79 W/(m ² K), m ²	158
External doors, $U = 1.0 \text{ W/(m^2 K)}, \text{ m}^2$	67
Initial design heating set point, °C	18
Initial design cooling set point, °C	25
AHUs heat recovery, % (TK01/TK02)	75/78
Measured air tightness, $m^3/m^2 h$	0.76 @50 Pa
Heating/cooling room units	radiant panels
Heat load design temperature, °C	-26
Design heat load, kW	84
Heat pump capacity, kW	40

Thermal bridges in calibration model were defined according to calculated values obtained during the design process and further presented on Fig. 2.



Fig. 2. Thermal bridges values of building calibration model

It is worth to note, that thermal bridges were modelled as internal due to the room-based calibration model.

2.2 Input data and modelling

This section describes the measured input data applied in building model calibration along with individual components modelling methods. Measured data contained missing periods that were filled with interpolated data between them. For this reason, some data spikes might occur on figures presented in the results section of this paper.

Ambient climate conditions were defined in IDA ICE climate file according to measured outdoor air temperature, relative humidity, wind direction, wind velocity, direct and diffusive solar radiation.

2.2.1 Heating system modelling

There are two secondary heating systems in OLK NZEB calibration model – floor heating and radiant ceiling heating panels. Both were modelled as ideal heaters. Floor heating was modelled with design power of 40 W/m2. Radiant heating ceiling panels total design power of ca 40 kW was spread across the building model in locations according to design documentation.

2.2.2 Air handling units modelling

There are two main air handling units installed in OLK NZEB – TK01 and TK02, both equipped with rotary heat exchanger and water heating coils. In actual installation, supply air temperature setpoint is controlled according to exhaust air temperature value. However, this feature was neglected in modelling due to the fact that measured supply air temperatures with a hour timestep resolution were applied (different for each AHU) to achieve match with measured AHU heat value. AHUs technical parameters were obtained from design and commissioning documentation, while initial operational schedules were presented by OLK NZEB staff, they are discussed in results section.

AHU TK01 serviced high hall-type rooms, while TK02 all other rooms. AHU TK02 operated according to design airflow of 0.7 m3/s, while TK01 operated at 1 m3/s (operation at part load of design airflow). Part load was accounted with coefficient of 0.625 to AHU airflow since actual design airflow is 1.6 m3/s. Such modelling approach was applied to properly calculate fans electricity consumption. For this reason, exact fans pressure and efficiency values were setup to obtain design documentation specific fan power (SFP).

2.2.3 Internal gains modelling

Building internal gains consist of following components – occupancy gains, lighting gains, equipment gains and solar gains. Solar gains are calculated in IDA ICE based on climate description and building geometry/envelope properties. Occupancy, lighting and equipment gains were described according to measured and estimated data.

In OLK NZEB, lighting and equipment consumptions were not measured separately. On the

room basis there were four electricity measuring points – Total annual consumption of these zones for the period of 01.02.2018- 31.1.2019 was ca 91.3 MWh/a.

In OLK NZEB, there is heavy machinery installed in one of measuring points. This equipment is also being cooled by separate active cooling system and for this reason cannot be accounted as internal gains in a building heat balance. As the machinery electricity consumption was not measured separately, that internal gains had to be then estimated. Method of estimation was proposed by building constructor, as the installed total lighting power of 8.71 kW was known, it was proposed to sum up the total consumption of that measuring point on an hourly basis and limit maximal consumption value to 8.71 kW, account only for working hours and without weekends. This resulted initial electricity consumption of 71 MWh/a being cut down to ca 23.9 MWh/a due to exclusion of heavy machinery electricity consumption. Measured electricity consumption for other measuring points was left as it is resulting in ca 20.2 MWh/a. Final internal heat gain for lighting and equipment applied in first calibration case was 44.1 MWh/a.

As the measured electricity consumption was available in hour resolution, three control signal input files were created based on measured and estimated data. The data was first sorted into the correct order to match IDA ICE date structure. Measured data starting date in excel was 00:00 01.02.18 which corresponds to hour 744 in IDA input file. Input files were created accounting for previous. IDA ICE model zones were further grouped into three separate categories to allow lighting signal being controlled according to input files data.

Occupancy was modelled according to estimated data presented by OLK NZEB staff. Total of 10 occupants are accounted in the modelling that are spread across the building heated spaces. Two different occupancy profiles were prepared based on that data.

2.2.4 Cargo gates opening modelling

Total of three cargo gates opening phase were measured. However, measured data appeared to be illogical and was neglected in the modelling. Though, in last calibration case presented in results section, some estimated cargo gates opening was applied.

2.2.5 Indoor air temperature setpoints modelling

In OLK NZEB, indoor air temperature along with actual setpoint at point of time in particular room were measured and exported from building monitoring system in one-hour resolution for 11 rooms accordingly. Successful attempts were made to account for either measured indoor air temperature in each room, as well as measured setpoint in each room. However, due to the wide variations in indoor air temperature measured results, it was decided to calculate hourly based building average weighted temperature to use it as a heating system setpoint input in all building calibration model zones. Building weighted average temperature (BWAT) was calculated based on specific heat loss of each room and its measured indoor air temperature.

2.3 Geothermal heat pump plant description

OLK NZEB geothermal heat pump plant fundamental scheme is presented on Fig. 3. Plant design considers option to separate energy piles loop via closing motorized valve (V-3) during the summer thermal storage cycle from the boreholes loop, in order to allow boreholes to provide "free cooling" while at the same time energy piles are being loaded with heat from source of thermal storage. In order to prevent the formation of the ice in the ground and possible frost heave, geothermal loops brine outlet temperature should not drop below 0...-1 °C. Therefore, circulation pumps (V-2 and V-3) in each loop will stop when measured (T2 and T3) brine outlet temperature drops below the set point of 0 °C. Condenser side of the heat pump is connected to a hot buffer tank, in which heat carrier temperature is maintained according to a supply schedule temperature that is dependent on outdoor air temperature value with its maximal value of supply side +50 °C at design outdoor air temperature conditions of -26 °C. Heat pump is capable of operation whenever the temperature in one of the loops is above the set point of 0 °C. On the contrary, heat pump stops its operation when there is no flow in the system (both loops are below the set point).



Fig. 3. Geothermal heat pump plant fundamental scheme

With this control logics all the available geothermal energy will be absorbed.

Whenever the cooling cycle starts, there is no heat demand in the system and heat pump will not operate. As the heat pump is inactive, energy piles loop should be separated by e.g. three-way valve from evaporator circuit. In this case only boreholes (heat wells) are active and flow in their circuit goes through "free cooling" heat exchanger.

For each of two loops there is a separate thermal storage. In case of boreholes (heat wells), the required amount of heat is supplied during their free cooling operation.

Solar collector (with/without buffer tank) is applied as a thermal storage source in energy piles loop. Thermal storage source is connected via heat exchanger to energy piles loop inlet pipe. Whenever the heap pump is inactive, energy piles loop should be separated by e.g. three-way valve and design flow is maintained in energy piles, which are loaded with heat separately from energy wells.

Solar thermal storage is controlled according to a temperature difference (ΔT) set point logics, where two temperatures are measured and desired value of ΔT is maintained. In solar thermal storage loop $\Delta T = 6K$. Measured temperatures in solar thermal storage loops on Fig. 3 are T4A and T4B. Whenever T4A temperature value is higher than 6K of T4B temperature value, pump P-4 starts it operation until the temperature of T4A reaches the desired $\Delta T = 6K$. Control of "Free cooling" loop operates by the logics "when beneficial" i.e. pump P-1 starts it operation whenever temperature T1B is higher than T1A.

3 Results and discussion

Results presented in this section are divided into two subsections – Section 3.1 describes a building model calibration results and Section 3.2 presents energy performance analysis, respectively.

3.1 Building model calibration results

Fig. 4 presents the results of building model calibration of OLK NZEB in IDA-ICE on the monthly basis for a building operation period of 01.02.18-01.31.19. Three out of four cases were simulated, while one out of four corresponds to actual measured data obtained via building monitoring system during the prior mentioned measuring period. Each case consists of three delivered energy components - room units delivered heat, air handling unit (AHU) heating coil delivered heat and lights/equipment delivered electricity i.e. building internal heat gains. In measured case, lighting/equipment energy component (green coloured on Fig.4) additionally contains heavy machinery electricity use since both lights and equipment electricity are measured together on the room basis and no exclusive separation for each consumer source exists. Besides, heavy machinery has a separate cooling system installed and therefore heavy machinery electricity use does not contribute as an internal heat gain in the building heat balance.

Results of Case 1 on Fig. 4 correspond to a simulation with initial settings where AHU operational profiles where defined according to Building Owner (HAMK) proposal and lights/electricity internal gains where modified to meet maximal internal lights load in order to exclude heavy machinery from building heat balance (see Section 2.2). In Case 1, simulated room unit heat resulted in 41.4 MWh/a and AHU heat was 18.9 MWh/a which is ca 35% smaller compared to 63.3 MWh/a of measured room unit heat and ca 19% less than 23.3 MWh/a of measured AHU heat respectively. In terms of month by month analysis, in heating period of Feb -May simulated AHU heat was exceeding measured one, while in heating period of Sept - Jan simulated AHU heat falls behind the measured dramatically. AHU operational profile proposed by HAMK in Case 1 in AHU TK01 was slightly shorter (64 hours per week) than in AHU TK02 (77 hours per week), while each repeated from month to month for the whole simulated period for both AHUs (see Table 2). AHUs modelled supply temperature corresponded to the measured one while modelled heat recovery performance corresponded to design documentation.



Fig. 4. Building model calibration results

However, as there significant measured and simulated AHU heat difference exists, it can be assumed that Case 1 AHU operational profiles proposed by HAMK do not match measured case scenario. Case 2 was generated specifically based on the assumption above, where initial AHU profiles were modified month by month to meet measured AHU heat on the monthly basis.

Table 2. AHUs working hours results

	AHU operational time						
Month	Case 1		Case 2		Case 3		
Month	hours/week						
	TK01	TK02	TK01	TK02	TK01	TK02	
February	64	77	35	59	40	58	
March	64	77	35	38	35	38	
April	64	77	30	28	28	25	
May	64	77	55	64	54	60	
June	64	77	64	93	64	93	
July	64	77	63	98	63	101	
August	64	77	160	168	116	140	
September	64	77	110	94	65	71	
October	64	77	64	78	64	66	
November	64	77	64	85	64	70	
December	64	77	68	89	63	76	
January	64	77	105	86	100	83	
Average	64	77	71	82	63	73	

As a result of calibration Case 2 (see Fig. 4), perfect match in simulated AHU heat of 23.3 MWh/a compared to measured 23.3 MWh was obtained in simulation with AHUs operational hours presented in Table 2. On the other hand, simulated room unit heat resulted in 41.1 MWh/a that produces a difference of -35% compared to the measured room unit heat i.e. practically same result as in Case 1. Assuming that building envelope thermodynamic properties were defined in coherence with building as built design documentation and indoor air temperatures setpoints were defined according to measured data with an hour resolution, significant simulated and measured room unit heat difference can be imposed by either inaccurate overestimated internal heat gains i.e. lights/equipment electricity use and/or additional sources of heat loss. Former can be explained by the lack of separate heavy machinery electricity use monitoring which led to initial guess regarding the lights/equipment internal gain from measured 90.9 MWh/a (heavy machinery included) to 44 MWh/a (heavy machinery excluded) i.e. ca -52% in first two simulated cases. Latter can be explained by the systematic cargo doors opening which behaviour is logged but due to the monitoring system failure could not be included in the modelling of first two cases. Additionally, some exhaust fans exist in laboratory part of the building, but their operation is not separately monitored and was neglected in the modelling. To further meet simulated room unit heat with the measured for completing the building model calibration procedure, either lights/equipment electricity use should be decreased and/or additional sources of heat loss should be implemented in the model. Case 3 was generated based on prior mentioned.

In Case 3 on Fig. 4 for a period of Feb - July lights/equipment electricity input data was scaled on the monthly basis which resulted in simulated room unit heat meeting measured. In same case for a period of Aug -Jan lights/equipment electricity was left unscaled and this input corresponded to Case 1 and Case 2, while cargo gates opening was implemented in the model which daily opening durations presented in Table 3. As a result of such modifications, internal gains for lights/equipment resulted in 37.3 MWh/a compared to 44 MWh/a of initial guess, while cargo doors were opened on average for 15 minutes per day within the year. In Case 3, additional modifications to AHU operational profiles of Case 2 were also conducted (see Table 2), to obtain perfect match in both simulated room unit heat and AHU heat.

Table 3. Cargo gates opening results

	Cargo gates opening				
Month	Case 1	Case 2	Case 3		
	min/day				
February	0	0	0		
March	0	0	0		
April	0	0	0		
May	0	0	0		
June	0	0	0		
July	0	0	0		
August	0	0	20		
September	0	0	43		
October	0	0	32		
November	0	0	42		
December	0	0	30		
January	0	0	12		
Average	0	0	15		

As mentioned in Section 2.2.5, simulations were performed by applying measured building weighted average temperature (solid red in left Fig. 5) as a heating system setpoint for each zone in the calibration IDA ICE building model. Building weighted average (BWA) temperature was calculated based on each zone specific heat loss and its measured hourly indoor air temperature. Fig. 5 (left) compares measured BWA temperature initial design indoor temperature against from simulations conducted in [10]. During the heating period of Feb - May and Oct - Jan measured BWA temperature was on average ca 1.54 °C higher than initial design indoor temperature. From the perspective of room overheating analysis, measured BWA temperature exceeded cooling setpoint of 25 °C during working hours of 08:00-17:00 for 135 °Ch. However, only rooms serviced by radiant ceiling panels are cooled via the connection to geothermal "free cooling" heat exchanger. Measured indoor air temperature in cooled room Hall 103 is depicted on Fig. 2 in yellow and peaked at +26.3 °C. In cooled room Hall 103 indoor air temperature exceeded cooling setpoint for 22 °Ch during working hours of the measuring period. According to Finnish and Estonian regulations, indoor air temperature should not exceed 100 °Ch during the summer period i.e.



Fig. 5 (left) Measured and design indoor air temperatures. (right) Measured and design AHU temperatures

01.06 - 31.08 in test reference year (TRY) climate. Fig. 5 (right) compares measured AHUs TK01/TK02 supply air temperatures applied in simulations against initial design AHU supply temperature. On average, within the heating period, measured supply air temperature was ca 1 °C higher than initial design AHU supply air temperature in case of both AHUs. As built design lacks "free cooling" connection to AHUs and for this reason measured supply air temperature is much higher during cooling period than initial design supply AHU temperature.

3.2 Energy performance analysis results

Results of measured and simulated OLK NZEB energy performance for a period of 01.02.18 – 31.01.19 are presented in Table 4. Total number of presented cases is four. First one corresponds to a reference initial design case from [10]. Second case describes actual measured energy performance. Third case presents calibrated model energy performance with heat pump operation modelled in Excel according to installed heat pump performance map and measured evaporator/condenser

	Case	Initial design (simulated SCOP)	Measured data (actual SCOP)	Calibrated model (Excel SCOP)	Calibrated model in TRY climate (Excel SCOP)
	Units	Specific annual energy consumption per floor area (kWh/m ² a)			
ng	Delivered room unit heat	32.1	42.3	42.3	41.8
ildi	Delivered AHU heat	9.5	15.5	15.5	15.5
Bu	Delivered DHW heat	4.1	3.6	3.6	3.6
	Top-up heating	2.8	14.9	0.3	0.3
	Heat pump compressor	9.2	20.1	13.0	12.9
	Cooling electricity	0.0	0.0	0.0	0.0
ţ	Fans electricity	9.2	9.5 ¹	9.5	9.5
lan	Pump electricity	2.0	2.0^{1}	2.0^{1}	2.0^{1}
4	Lighting and equipment electricity	13.0	24.9	24.9	24.9
	DHW electricity	1.5	4.5	4.5	4.5
	EPV ²	45(64)	91(129)	65(92)	65(92)
	Units	Seasonal coefficient of performance value (SCOP)			
	Heat pump SCOP	4.68	2.88	4.46	4.46
	Whole plant heating SCOP	3.28	1.97	3.80	3.80
	Whole plant SCOP with DHW	2.95	1.56	3.45	3.45

 Table 4. Measured and simulated OLK NZEB annual energy performance results

¹Pumps/fans electricity in measured case are estimated values. Measured data is only available as total AC equipment. Automation and BMS electricity were deducted. ²Energy performance value (EPV) is calculated with electricity primary energy factor of 1.2 and 1.7 in parentheses (valid for buildings constructed before 01.01.2018)

inlet/outlet fluid temperatures i.e. inlet to heat pump evaporator from geothermal heat exchanger and outlet from heat pump condenser to hot tanks. Fourth case describes calibrated model energy performance in TRY climate conditions with applied Excel calculated SCOP from the previous case. Energy usage components are depicted as delivered energy values that account for efficiencies and distribution losses of the heating/cooling system's generation and consumption side.

Compared to initial design case, measured heat consumption of hydronic heating system (room unit heat in Table 4) and AHU heating coil turned out to be ca

32% and ca 63% higher than in design case, respectively. On the other hand, domestic hot water (DHW) heat is ca 12% lower than in initial design case. Indoor climate conditions in measured case were less favorable in terms of heat consumption due to 1.54 °C higher average heating system setpoint temperature and 1 °C higher average AHU supply air temperature within the heating period. Initial design AHUs heat recovery temperature efficiency is slightly better i.e. n = 0.8 compared to as built AHUs n = 0.78. From the perspective of internal gains, measured case appliances (lighting and equipment electricity in Table 4) delivered energy of 24.9 kWh/m²a is ca 17.8 MWh/a i.e. ca 92% higher compared to reference case. Nevertheless, decrease in heat consumption due to higher appliance's internal gains is significantly reduced by the additional heat losses through cargo gate opening and lower overall solar radiation in measured case compared to reference. In reference case opening of cargo gates was not modelled. As mentioned in Section 3.1, cargo gates opening in measured case is a rough estimation due to malfunction of monitoring system and lack of appropriate separation in the logging/monitoring of lights/equipment/cooled heavy machinery electricity consumption which also led to rough estimation of measured appliances electricity. Impact of cargo gates opening on OLK NZEB heating need in calibration case is ca 32%. Impact of climate conditions on the heating need can be quantified by comparing measured case room unit heat of 42.3 kWh/m²a against the calibration case in TRY climate room unit heat of 41.9 kWh/m²a which yields a difference of ca 1.2%. On the other hand, comparison of measured and reference climate data resulted in measured degree days of 3803 °Cd against reference degree days of 3661 °Cd at balance point temperature of +15 °C. Additionally, the sum of measured diffuse and direct solar radiation is ca 2.8% i.e 23.4 MWh/a smaller in measured climate in comparison to TRY climate case within the heating period. There is a good agreement in almost matching fans electricity consumption of measured 9.5 kWh/m²a against reference 9.2 kWh/m²a. However, specific fan power (SFP) in measured case is slightly lower and on average AHUs operation duration was ca 13% higher compared to reference. Measured and simulated delivered room unit and AHUs heat are also presented in form of 24h moving averages on Fig. 7 (left) resulting in good agreement.

Results of seasonal coefficient of performance (SCOP) in Table 4 are divided into three categories – heat pump SCOP only accounts for AHU and room unit heat, whole plant heating SCOP also considers top-up heating and pumps electricity, the last one accounts additionally for domestic hot water (DHW) heat.

Table 5. measured and calculated COP results heating

Month	Measured cond.outlet	Measured evap.inlet	Calculated COP	Measured COP
Feb	46	7	4.82	1.26
Mart	49	5	4.19	2.01
April	48	12	4.68	1.99
May	55	13	3.53	1.52
June	55	15	3.24	0.94
July	55	17	2.90	0.26
August	55	15	3.24	0.78
Sept	55	12	3.80	1.77
Oct	54	9	3.86	2.32
Nov	54	8	4.12	2.73
Dec	50	4	4.21	2.26
Jan	48	3	4.16	2.67
Average	52	10	3.90	1.71

According to results, geothermal heat pump plant in measured case underperformed dramatically resulting in overall plant SCOP of 1.56 which compared to simulated initial design whole plant SCOP of 2.95 is ca 47% lower than was expected. Without accountancy for top-up heating, heat pump SCOP in measured case resulted in 2.88 compared to initial design 4.68 underperforming by 38%. Top-up heating electricity value represents the energy consumption at point when ON/OFF heat pump was not able to meet building heat demand (due to evaporator entering temperature reached 0 °C limit) and top-up heating would provide additional energy to keep temperature in hot buffer tank according to desired setpoint. In measured case top-up heating electricity



Fig. 6. Measured heat pump plant performance



Fig. 7. (left) Measured and calibrated delivered heat. (right) Measured and design plant supply temperatures

consumption of 14.9 kWh/m²a is ca five times higher than in initial design case 2.8 kWh/m²a. Poor measured geothermal plant performance can be also observed on the monthly basis in Table 4, where in July measured average plant COP equals to 0.26. Low measured overall plant SCOP is caused by improper geothermal plant operation due to wrong control algorithms and/or faulty plant automation system, where top-up heating dominates as a heat source instead of heat pump. This can be observed on Fig. 6 where e.g. in the beginning of February (0 - 200 h) evaporator inlet fluid temperature was ca +12.5 °C i.e. supply fluid from energy piles and boreholes loop to heat pump was within the heat pump operation range, yet measured heat pump compressor power ranged in 0...1 kW barely operating, while top-up heating was operating with power of ca 20...55 kW. Same can be also observed in the beginning of April (1400 - 1500 h) and November (6700 - 7000 h). Also, during the cooling period (2500 - 5000 h), when mostly DHW consumption is present and geothermal loop fluid temperatures are within +10...+18 °C range, top-up heating is still operating, while according to initial design it should not. According to measured results, plant clearly operates not in coherence with initial design intent. To quantify the potential of geothermal plant in measured case conditions, heat pump performance was modelled in Excel with second-degree polynomial equations at an hourly time step based on the actual heat pump performance map data and measured evaporator inlet and condenser outlet fluid temperatures. This simplified modelling approach has known limitations and assumptions, where all thermodynamic processes in soil and geothermal heat exchanger (GHX) are neglected i.e. GHX and heat source are assumed to be infinite, heat pump evaporator inlet temperature corresponds to measured one and is not influenced by the operation of modelled heat pump. Hypothetically, results of this case roughly correspond to highest achievable SCOP at measured GHX temperature conditions based on the heat pump performance map data and measured secondary side temperatures. Whole plant SCOP with DHW modelled in Excel based on the measured data resulted in 3.45 i.e. 17% higher compared to simulated initial design case of 2.95 and ca 2.2 times higher compared to actual measured SCOP. It is worth to note, that actual

installed heat pump model differs from one simulated in initial design case. However, in initial design case [10] SCOP was obtained as a result of detailed numerical simulation, which is far more accurate than simplified SCOP estimation approach applied in this study. Nevertheless, due to poor measured plant performance it was decided that a simplified SCOP estimation would be enough to quantify the possible best performance of actual as built plant.

Additional discrepancies in measured plant operation compared to initial design case can be observed on Fig. 7 (right), where according to initial design intent both AHU and radiant ceiling panels secondary side supply fluid temperatures were meant to be maintained according to a heating curve presented on Fig. 8.



Initial design heating curve at -26 °C outdoor air temperature value requires the supply side temperature to be equal to its maximal value of +50 °C decreasing with outdoor air temperature decline down to +20 °C when outdoor air temperature reaches a value of +20 °C. However, according to Fig. 7, measured AHUs supply fluid temperature is not dependent on the heating curve, while radiant ceiling panels supply temperature shows signs of dependency still being higher than designed. Latter might be explained by radiant ceiling panels not being capable of maintaining the desired setpoint (see Fig. 5 left) temperature, which in turn might result in

plant supply temperature increase. This negatively impacts the heat pump COP decreasing overall plant energy performance. According to heat pump performance modelling in Excel, installed heat pump with rating conditions capacity of ca 40 kW is capable of meeting ca 99.5% of heat demand in measured climate conditions.

Global goal of OLK NZEB design and construction was to reach Finland NZEB target [11] which is 135 kWh/m²a of primary energy consumption in case of commercial hall-type building. Each case energy performance values (EPV) are presented in Table 4. Since 01.01.2018 new NZEB requirements took place in Finland [11], while initial design phase occurred in early 2014. Before year 2018 primary energy factor for electricity was 1.7 and after 1.2 in Finland. For convenience, EPV results presented in parentheses (Table 4) correspond to primary energy factor applied before year 2018. Initial design simulated EPV according to most recent primary energy factors resulted in 45 kWh/m²a exceeding the Finland NZEB target by the factor of 3. On the other hand, measured case EPV resulted in 91 kWh/m²a i.e. two times higher than initial design result. Nevertheless, measured case EPV complies with Finland NZEB commercial hall-type building requirements either with both pre and after year 2018 primary energy factors and carries official status of nearly zero-energy building. However, there is a room for improvement, as EPV in calibrated case with Excel modelled geothermal plant SCOP resulted in 65 kWh/m²a i.e. ca 29% improvement in EPV compared to measured case scenario. As already discussed in this section, operation of plant automation system should be checked and adjusted/tuned for the plant to operate in accordance with initial plant design intent/control logics.

4 Conclusion

According to the results of the present study, main goal of designing and constructing OLK NZEB - a commercial hall-type building that complies with Finland NZEB requirements has been successfully achieved resulting in measured EPV of 91 kWh/m²a i.e. ca 33% lower than the NZEB target value of 135 kWh/m²a.

Analysis of measured geothermal heat pump plant performance and modelling revealed the discrepancies in plant operation that resulted in surprisingly low overall geothermal heat pump plant SCOP of 1.56 compared to initial design expectation of 2.95. Nevertheless, measured data analysis exposed inappropriate plant operation where electrical top-up heating dominated instead of heat pump, while heat pump on the other hand had favorable operating conditions but was not operating due to most likely inappropriately adjusted automation system. As an output of this study, it is suggested for the OLK NZEB building owner to check and readjust/tune the plant automation system to meet initial design intent/control logics. According to modelling results, prior mentioned adjustments could possibly lead to decrease of EPV up to ca 29% and overall plant SCOP

increase of up to 3.45 according to simplified Excel based model. Also, in terms of heat pump COP increase, it is suggested to control AHUs secondary side supply temperature according to heating curve.

From the perspective of plant operation during cooling period, installed "free cooling" heat exchanger managed to deliver enough cool via radiant cooling panels system to observed room Hall 103 which resulted in indoor air temperature peak of +26.3 °C, while outdoor air temperature was +32 °C. According to Finnish and Estonian regulations, indoor air temperature should not exceed 100 °Ch during the summer period. In observed room Hall 103 indoor air temperature exceeded cooling setpoint for 22 °Ch, which perfectly meets the overheating regulations.

Building model calibration procedure confirmed that it is possible to reach a good agreement between measured and simulated results in IDA ICE simulation environment, which is capable of processing enormous amount of measured input data via source files and allows to perform high detail modelling. Despite the lack in separation of measured lighting/equipment/heavy machinery electricity, issues with monitoring cargo gates opening times and some missing measured data, nonmeasured AHUs airflows and fans electricity use, with some effort, assumptions and input data modifications, this study produced a calibrated monthly basis model using most of the available measured hourly based data as an input and achieved a very good agreement. For improvement of building energy performance, it is suggested to lower the indoor air temperature in building by ca 1.5 °C to meet the design intent, as measured weighted average indoor air temperature appeared to be ca 1.54 °C higher. Also, it is suggested to lower AHUs supply air temperature by 1 °C to meet the initial design intent.

This study unveiled the importance of building monitoring/logging system, especially in buildings with non-conventional custom heating/cooling plant design, as in actual operation plant may underperform dramatically. And specifically, for assuring the as designed heating/cooling plant and systems operation, it is extremely important to have an appropriate welldesigned building monitoring/logging system in coherence with building model calibration needs. Based on this study, the following list of measured parameters with hourly resolution for whole year period to be included in building monitoring/logging system for conduction of successful detailed building model calibration in IDA ICE:

• Outdoor climate data (outdoor air temperature, relative humidity, wind velocity, wind direction, direct and diffuse solar radiation);

• Indoor air temperatures, setpoints;

• Plant primary and secondary side temperatures

(including heat pump evaporator/condenser inlet/outlet, thermal storage components etc);

• Plant energy consumptions by component

(heat pump compressor/condenser/evaporator, cooling equipment, top-up heating/cooling/DHW, circulation

pumps, AHU heating/cooling coils, buffer tanks primary secondary side energies, chillers etc);

• Internal gains data by component (lights, equipment, cooled equipment);

• AHUs energy and operation by components

(supply/return air flows, fans electricity, supply/return air temperatures, air conditioning components energies/temperatures);

• Cargo gates/big-sized windows opening data.

This study will be continued with the assessment of improved/fixed OLK NZEB geothermal plant energy performance with accordance to suggestions presented in the present study.

This research was supported by Ruukki Construction Oy, Terveellinen digitalo -hanke, EAKR-hank A72991, Estonian Centre of Excellence in Zero Energy and Resource Efficient Smart Buildings and Districts, ZEBE (grant No. 2014-2020.4.01.15-0016) funded by the European Regional Development Fund and by European Commission through the H2020 project Finest Twins (grant No. 856602).

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