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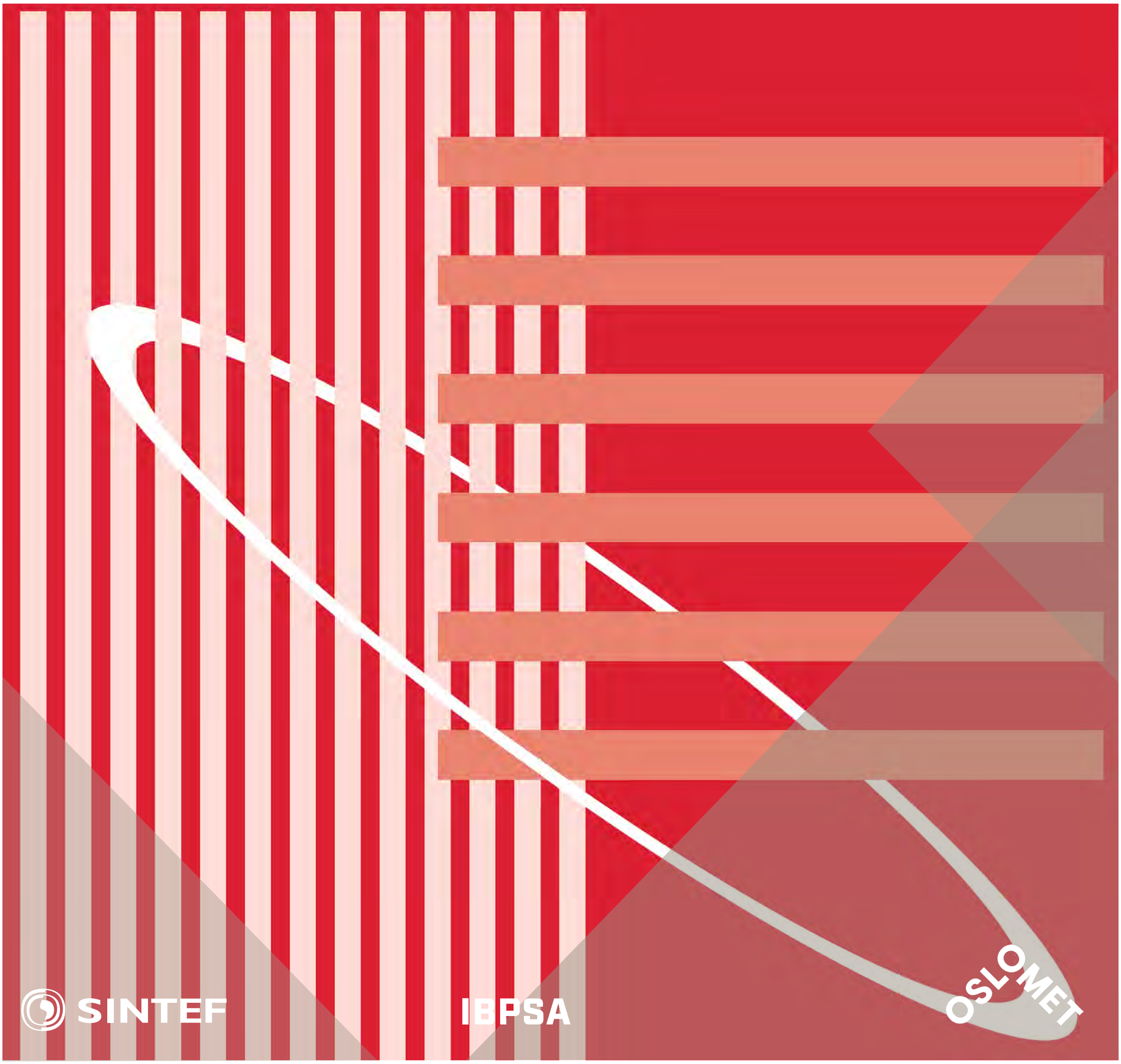

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Quasi-Dynamic Modelling of DC Operated Ground-Source Heat Pump

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Abstract

The performance of a conventional ground-source heat pump (GSHP) has been measured in the laboratory with alternating current (AC) and direct current (DC) operation using the standardised points from EN14511:2018. The results from these measurements have been used to modify a variable speed heat pump model in IDA Indoor Climate and Energy (ICE) and the annual performance of AC and DC operation have been simulated for an entire year's operation at two geographical locations in Sweden. Results show that the energy savings with DC operation from laboratory measurements span between 1.4–5.2% and when simulating the performance for an entire year's operation, the energy savings vary between 2.5–3.4%. Furthermore, the energy savings from the simulations have been compared to the bin method described in EN14825:2018.

Introduction

The recent market developments of solar photovoltaic (PV) and batteries for residential buildings have had an exponential growth the last couple of years (IEA (2019); Ralon et al. (2017)). As PV and batteries, as well as most household appliances are natively operated on DC, DC distribution in buildings has lately gained interest since conversion losses can be avoided compared to traditional AC topologies.

There are many attempts in literature to estimate the energy savings when switching from AC to DC distribution in buildings (Vossos et al. (2014); Glasgo et al. (2016); Seo et al. (2011); Denkenberger et al. (2012); Hofer et al. (2017)), however findings from these studies differ substantially, varying between 1.5–25.0% in saving potential depending on the choice of reference case, types of appliances included (and their efficiency gains), and system studied. Further, findings in literature on DC savings for individual systems/appliances are often taken as a steady-state value without considering the dynamic behaviour of the appliances. Ryu et al. (2015); Lucía et al. (2013); Weiss et al. (2015); Fregosi et al. (2015) and Kakigano et al. (2010), have shown energy savings from DC operation for individual household appliances and HVAC-components in the span 1.5–9%, which is strongly reflected in the final energy savings for the system. Differences in energy savings are evident when studying different types of appliances but suggests that more research is needed to pin-down the savings more precisely. Also, often a constant efficiency saving is assumed without regards to its varying operation.

Conventional heat pumps are operated using DC at the final stage, where the supplied AC is rectified inside the heat pump using an AC/DC conversion step. Since these conversions are subject to losses, it is desirable to minimise these by feeding the heat pump directly with DC. A demonstration of a DC operated heat pump is presented in Huang et al. (2019) for a DC micro grid in Sweden, coupled with direct DC generation from PV, DC storage in a battery and direct DC loads. As this study focuses on the micro grid energy performance, no comparison was made for the DC operated heat pump's performance.

Ollas (2020) and Gerber et al. (2019) have suggested topologies for an AC and DC system respectively and presents the potential energy savings for DC, especially when renewable energy (RE) and battery storage are included. Missing in these studies are however measured energy gains from DC operation of individual appliances.

In this paper the annual performance of an AC and DC operated GSHP for space heating is quantified using laboratory measurements of an AC and DC operated GSHP. The laboratory measurements are done according to the steady-state operating points specified in EN14511:2018 to show the energy gains achieved with direct DC operation. The results from these steady-state points are then used to simulate the annual performance of the GSHP for a nearly-zero energy building (nZEB) at two locations in Sweden. The simulations are done using IDA ICE and a modified model for a variable speed heat pump.

Theory

A heat pump uses electrical input to generate a heating/cooling output through a cycle of evaporation and condensation of a refrigerant. In this study, a variable speed compressor heat pump is studied. The compressor speed is controlled to give a specific supply temperature of the water in the heat distribution system based on the measured outdoor temperature, i.e. colder outdoor temperatures gives higher supply temperatures. A variable speed heat pump is designed to operate in a specific frequency interval, meaning that at low part loads, the compressor cannot reduce the speed enough and will instead operate in an ON/OFF mode.

Since the heating/cooling[1] output is higher than the required power input, conventional theory about efficiency cannot be applied. Instead, the Coefficient of Performance (COP) is used to evaluate the performance, and it is defined as

$$COP(t) = \frac{Q_{heat}(t)}{W_{power}(t)} \tag{1}$$

where Q_{heat} and W_{power} are the heat output and power input respectively at time t. The COP is often related to a specific operating point at steady-state conditions. A more relevant performance factor is the Seasonal Coefficient of Performance (SCOP) which describes the average performance over a defined time period, considering only the space heat generation. From time-series data, the SCOP can be found, similar to (1), by analysing the heat generation and power usage over a defined time-period, as

$$SCOP = \sum_{t_1}^{t_2} \frac{Q_{heat}(t)}{W_{power}(t)} \tag{2}$$

where Q_{heat} and W_{power} are the sum of heat delivered and electrical energy usage over the time period t_1–t_2.

Methodology

In this section, the case study, being a single-family nZEB, is introduced together with the method used for the heat pump steady-state measurements, and finally the heat pump and building models are described, and key performance indexes are defined.

Case Study – nZEB

The single-family nZEB in this study was developed and commissioned within the project "New Energy Efficient Demonstration for Buildings" (NEED4B), and have the specifications given in Table 1. The specified working interval for the GSHP means that the compressor can adjust its speed to deliver 1.5–6 kW of heat output. For a heating demand below 1.5 kW, the compressor will start to work in ON/OFF mode. If the supply temperature of the heating system is too low at maximum compressor speed the backup heater will start to generate more heat to the system, to cover the demand deficit. For the building in this case study the system is more or less monovalent, and the use of the backup heater is limited. For the modelling of the annual heat pump performance, a nZEB is used and the building model is adopted from Chen and Markusson (2018) and adjusted with a mechanical supply and exhaust ventilation with a heat exchanger. In the referred article, a comparison was made with measured performance from Ylmén and Persson (2017), and the results showed good coherence between simulated and measured performance.

Heat Pump Laboratory Measurements

A conventional GSHP have been tested in a laboratory setup using the standardised operating cycles from EN14511:2018 (CEN (2018a)), at both AC and DC operation, in order to quantify the energy savings from the latter. For the DC measurements, the rectification stage, done inside the heat pump, has been removed and the heat pump is supplied directly with 380 VDC[2].

Measurements are made at four operating frequencies of the compressor, spanning between 30–118 Hz, and at two operating modes of brine and water temperatures (0/35°C and 0/55°C), according to the operating points defined in EN14511-2:2018. Where the former is typically used for low-temperature underfloor space heating and the latter for DHW production or space heating using a radiator system.

Building & Heat Pump Modelling

Using the measured data for the heat pump and data for the building from the case study, the heat pump performance have been simulated for an entire year's operation in an nZEB for two locations in Sweden: Malmö (south) and Luleå (north). Figure 1 shows a duration curve of the outdoor temperatures at the two simulated locations[3].

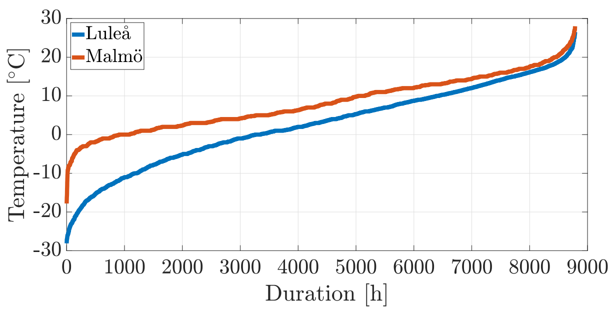

Figure 1: Duration curve of the outdoor temperatures for the two simulated locations: Luleå and Malmö.

A model of the single-family house is created in IDA ICE version 4.8. The house model has a total floor area of 155 m^2 and is modelled with an occupancy level of two adults and two children to mimic a

[1] In this study, only heating is required and thus, cooling is left out from here on out.

[2] The relays where operated on AC, but these power levels remained below 7 W.

[3] The weather data are taken from the ASHRAE IWEC 2 database, available in the IDA ICE program, and are used in the simulation of the building energy usage.

Table 1: Technical specification used in the modelling of the studied single-family nZEB.

Floor area (2 floors)	155 m^2
Average heat transfer coefficient, U_{avg}	0.20 $W/m^2/K$
Heating system	GSHP with a floor heating system
Ventilation	Balanced ventilation system with heat recovery
Ventilation flow rate – supply and exhaust	60 and 66 l/s respectively
Heating demand at −22°C	3.5 kW
GSHP working interval	1.5–6.0 kW
Air tightness	0.2 $l/s/m^2$ external surface with a pressure difference of 50 pa
Rotatory heat exchanger efficiency	82%

normal-sized Swedish single-family house. Different occupancy schedules are applied to the residents, and the house is divided into 13 zones in the modelling. An internal heat load (i.e., heat generated by lighting and other equipment) of 30 kWh/m^2 is used, suggested by SVEBY (Levin et al. (2009)) as a standard value in Sweden for residential building energy simulations. The weather data files for Malmö and Luleå are available in the IDA software, and are derived from integrated surface hourly weather data, originally archived at the national climate data centre (Equa Simulation Technology AB (1999)).

The energy demand simulated in IDA does not include air-rising losses due to opening of windows and external doors. Instead, only losses due to transmission and ventilation are compensated for. The air-rising losses are assumed to be 4 kwh/m^2 per year according to Levin et al. (2009).

A balanced ventilation system with a rotatory heat exchanger is used, with a supply and exhaust flow rate of 60 and 66 l/s respectively. The efficiency of the heat exchanger is set to 82%, according to the manufacturer's specifications. The house is heated by a floor heating system that is directly connected to the heat pump, which means that no storage tank is used. The water supply temperature in the floor heating system is specified in the range 20–35°C and varies with the outdoor temperature. Figure 2 shows the simulated heating system supply temperature, t_{supply}, as a function of outdoor temperature, $t_{outdoor}$ together with the set-points for the GSHP operation. The house is modelled with the constraint to keep an indoor temperature of at least 21°C.

As the difference between the AC and DC operated heat pump with regards to COP, is small, it is difficult to use an ordinary regression model to capture these differences. Instead, a model based on tables of measurement data is used, where the values are normalised by the Carnot COP, COP_{Carnot}. The output at the condenser, Q_c, is given in a table as a function of the compressor speed (in Hz) for the two tested points 0/35 and 0/55°C. Interpolation between the points is made by assuming a continuous derivative. The COP is handled similarly, but instead of giving

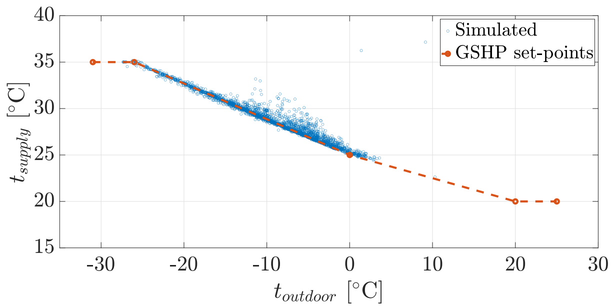

Figure 2: Simulated supply temperature for the heating system, t_{supply}, as a function of outdoor temperature, $t_{outdoor}$ together with the GSHP operating set-points.

the COP in the table, a ratio is calculated as

$$n_c(t) = \frac{COP(t)}{COP_{Carnot}} \tag{3}$$

The COP used in the simulation, $COP(t)$, is the interpolated value of $n_c(t)$ multiplied by the Carnot COP, COP_{Carnot}. The compressor and evaporator power, $P_c(t)$ and $Q_e(t)$ respectively, are then calculated as

$$P_c(t) = \frac{Q_c(t)}{n_c(t) \cdot COP_{Carnot}} \tag{4}$$

$$Q_e(t) = Q_c(t) - P_c(t) \tag{5}$$

Based on discussions with the GSHP manufacturer, the COP value was assumed constant for compressor frequencies below the lowest test point, e.g. 30 Hz.

The heat pump is controlled by a PI-controller. The output is converted to a desired frequency between 18–120 Hz. The floor heating system have a large thermal mass and limits the need for a tank. The PI controller has a tacking time of 30 seconds that limits the wind-up.

Key Performance Indicators

The heating demand for the building is simulated in IDA ICE by setting a desired indoor temperature. The required heat pump electrical energy to maintain that indoor climate is calculated as

$$E_{power} = \int_{t_1}^{t_2} W_{power}(t)dt \tag{6}$$

where W_{power} is the heat pump input power, and the energy usage is evaluated for an entire year's operation (t_1–t_2). Similarly, the generated heating energy is calculated as

$$E_{heat} = \int_{t_1}^{t_2} Q_{heat}(t)dt \quad (7)$$

From EN14825:2018 (CEN (2018b)), the SCOP is calculated by dividing the evaluated time period into a number of hours at different outdoor temperatures ("bins") for a specified climate[4], to reflect the variations in the heating period. For each of these bins, a COP value is calculated and is then used to calculate the SCOP value for the annual operation. For the time-series data given from the simulations in IDA ICE, the SCOP calculation is adopted from Zottl et al. (2011) and system boundary "SPF_{H3}" without the inclusion of heat and electricity for domestic hot water production, Q_{W_hp} and E_{HW_hp}, and electrical energy use of the brine/well pump, $E_{S\_fan/pump}$. Thus, the SCOP is given from the IDA simulations as

$$SCOP_3 = \frac{Q_{H\_hp} + Q_{HW\_bu}}{E_{HW\_hp} + E_{HW\_bu}} \quad (8)$$

where Q_{H_hp}, Q_{HW_bu} are the heat energy delivered for space heating from the heat pump and electrical back-up heater respectively, and E_{HW_hp} and E_{HW_bu} the electrical energy usage for the heat pump (for space heating) and electrical back-up heater respectively.

For the SCOP comparison the bin method from EN14825:2018 is applied to the results from the laboratory measurements of the GSHP. The bin-method is based on the calculation method for a low temperature system in a "cold climate" in EN14825:2018, using some modifications. The main modification is the replacement of the cold reference climate, with climate data for Luleå and Malmö[5]. Another modification, compared to the standard, is that the building's energy signature is based on the IDA ICE simulations with a heating demand of 3.5 kW at −22 °C. The heating demand is assumed to decreases linearly down to 0 kW at +6 °C, see Figure 3 for a visualisation of the simulated heating demand as a function of outdoor temperature. Thereby the building simulated has a heating demand to +6°C, instead of +16°C as defined in EN14825:2018.

Results

Here, results are presented first for the laboratory measurements of the AC and DC operated heat pump, using the points specified in EN14511:2018, and then these measured performance sets are incorporated into the heat pump model to model the performances for two geographical locations. The latter

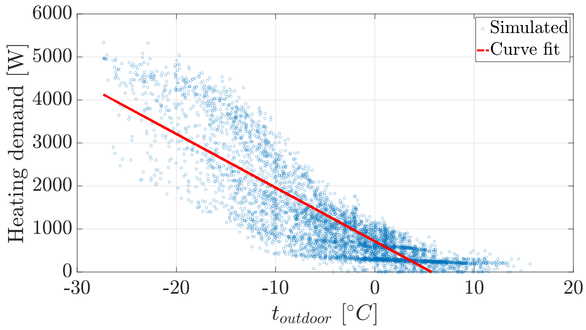

Figure 3: Energy signature for the modelled nearly-zero energy building showing the measured heating demand at difference outdoor temperature, $t_{outdoor}$, and a linearization of the dependency.

to quantify the energy savings from DC operation, also with regards to the GSHP's operating condition. A comparison of the simulated energy savings is also made using the bin method defined in EN14825:2018.

Heat Pump Measurements – AC & DC Operation

Results from the laboratory measurements of AC and DC operation of the heat pump are given in Table 2 for different operating frequencies and modes of operation. Results are shown for heat output (to the system) and input power (electrical). The resulting COP values are also given in Figure 4 at different compressor frequencies and brine/water temperatures. Noteworthy from these results is that the measured COP is higher for all cases with DC operation, due to the eliminated rectification loss, and that the gains are higher for the low-loading, i.e. low operating frequencies.

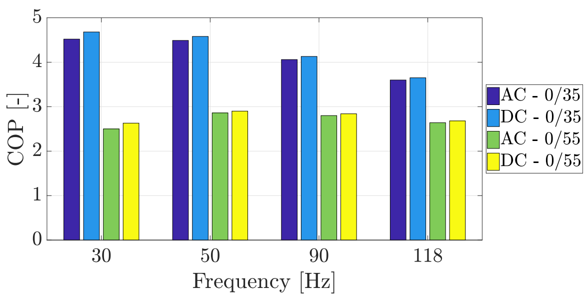

Figure 4: Measured Coefficient of performance (COP) from laboratory measurements at different compressor frequencies and brine/water temperatures (0/35 and 0/55).

Simulated Heat Pump Performance

The measured performance from Table 2 has been incorporated into the heat pump model, described in the methodology section, to simulate one year's operation for the nZEB case study at two geographical locations: Malmö[6] and Luleå[7]. The simulated energy savings, with regards to a reduced energy demand, span between 2.5–3.4% for DC operation in Luleå

[4]In EN14825:2018, three reference climates are given – Helsinki ("cold"), Strasbourg ("average") and Athens ("warmer").

[5]The outdoor temperature data is taken from Meteonorm version 6.1.

[6]**Longitude:** 55.609534 — **Latitude:** 13.002925

[7]**Longitude:** 65.588774 — **Latitude:** 22.156974

Table 2: Results from laboratory measurements of an AC and DC operated heat pump according to the operating points defined in EN14511:2018.

		Unit	Frequency [Hz] 30		50		90		118	
Common data	Brine in/Water out	°C	0/35	0/55	0/35	0/55	0/35	0/55	0/35	0/55
	Brine out	°C	-3	-3	-3	-3	-3	-3	-3	-3
	Water in	°C	30	47	30	47	30	47	30	47
AC operation	Heat output	W	1796	1389	3063	2632	5961	5305	8057	7444
	Power input	W	397	556	682	921	1470	1895	2241	2825
DC operation	Heat output	W	1810	1426	3053	2616	5964	5283	8029	7379
	Power input	W	387	542	667	902	1443	1857	2198	2755
	COP gain	%	**3.5**	**5.2**	**2.0**	**1.4**	**1.7**	**1.4**	**1.4**	**1.5**

and Malmö respectively. Based on the simulated performance of the heat pumps operation made in IDA ICE, a duration curve of the compressor's frequencies is shown for both locations in Figure 5. Noticeable is that the heat pump is operated at the lower range of its frequency interval during a vast majority of the time and thus in the upper span of the energy savings according to Table 2. Furthermore, the difference between AC and DC operating frequency is small. The heat pump is well oversized for the simulated building and for working in space heating mode only, and thus operates frequently in an ON/OFF stage, especially prominent for the Malmö case.

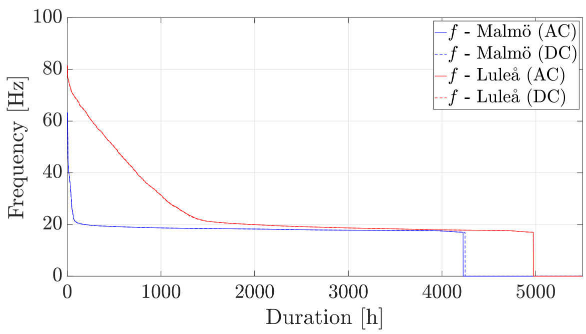

Figure 5: Registered compressor frequencies for AC and DC operated heat pump in Luleå and Malmö from the IDA simulation.

As a comparison, the energy savings for the two locations have also been calculated for AC and DC operation using the bin method described in EN14825:2018 using the outdoor temperature recordings from Meteonorm 6.1 and the measured heat pump performance from Table 2. The resulting SCOP savings are shown in Table 3 together with the modelled results from IDA ICE, which shows good coherence.

Conclusion

Results from laboratory measurements of a GSHP show performance gains with DC operation, in terms of an increased COP, in the range 1.4–5.2% for the measured operating cycles defined in EN14511:2018.

Table 3: Comparison of SCOP savings from IDA simulation and the bin method for the two geographical locations.

Method	*Unit*	*Malmö*	*Luleå*
Energy savings – IDA	%	3.4	2.5
Energy savings – bin	%	3.3	2.8

The energy savings also differ depending on the operation point, with higher gains for the lower operating frequencies. When these results are applied and simulated for an entire year's operation, the energy savings, in terms of a reduced electricity demand, span between 2.5–3.4% for the two simulated locations in Sweden. The energy savings from DC operation are larger for Malmö due to a more frequent operation in the lower frequency interval, where the largest savings are noted from the laboratory measurements.

Furthermore, the energy savings for space heating from the simulated results in IDA ICE and the bin method, according to EN14825:2018, shows good coherence.

Acknowledgment

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