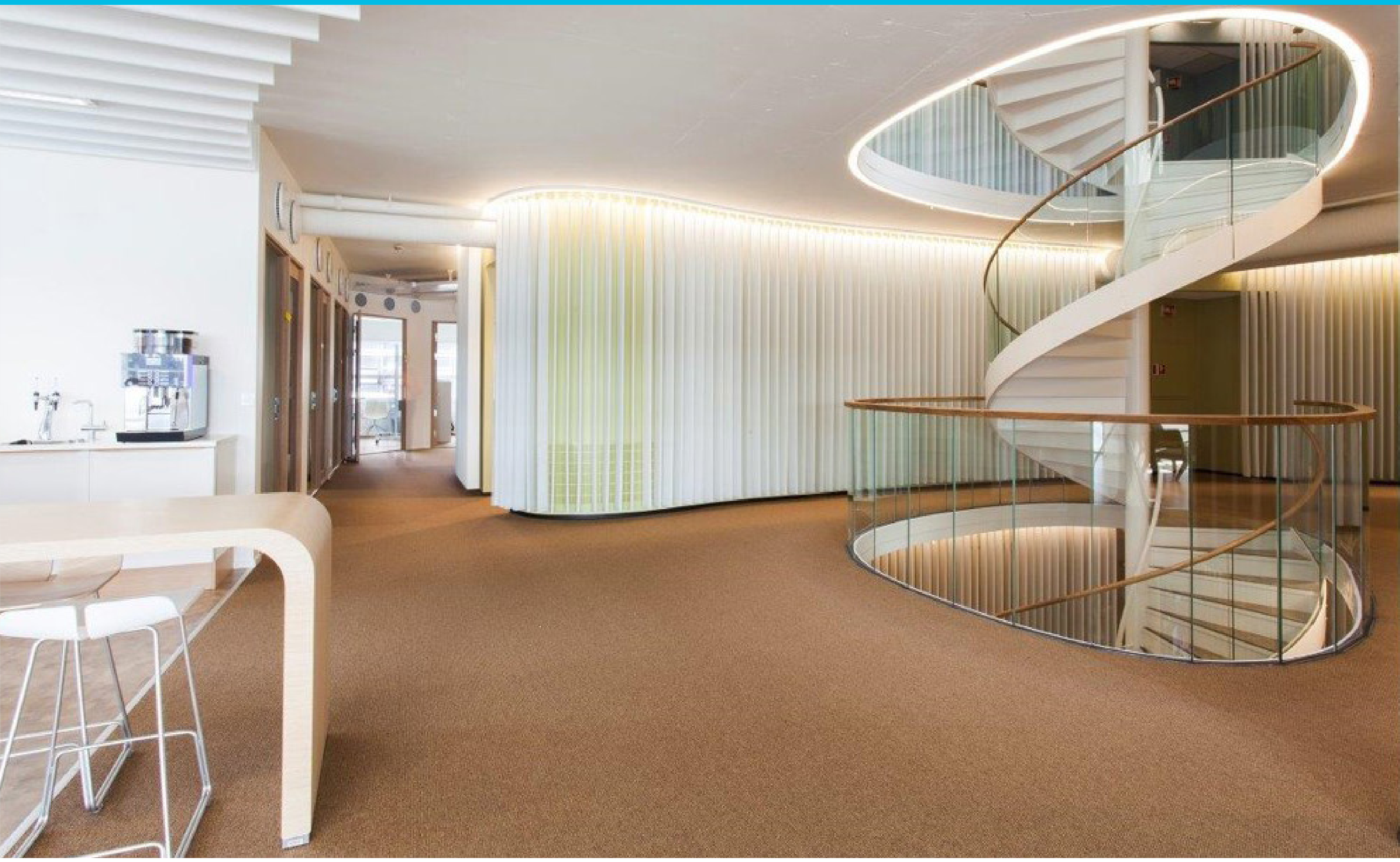
ZEB Project report 27 – 2016

Jørn Stene and Maria Justo Alonso

Field Measurements – Heat Pump Systems in NZEB

ZEB The Research Centre on Zero Emission Buildings



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Jørn Stene[1] [3] and Maria Justo Alonso [2]
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This report is also a part of IEA HPP Annex 40, which is a corporate research project on heat pump application in Nearly Zero Energy Buildings. The project is part of the Heat Pump Programme (HPP) of the International Energy Agency (IEA).

Abstract

The IEA Heat Pump Programme (HPP) Annex 40 "Heat Pump Concepts for Nearly Zero Energy Buildings" deals with the application of heat pumps as core component of the HVAC system for Nearly or Net Zero energy buildings (NZEB). Annex 40 has been structured in four tasks which comprise the following activities:

Task 1 – State-of-the-art analysis
The Task 1 is to give an overview on NZEB on the national level of the participating countries. In more detail, the political framework in terms of NZEB (e.g. building codes, legislation, definitions of NZEB), the state of market introduction, and applied technologies both on the building envelope and the building HVAC system shall be characterised. The compiled technical concepts shall be analysed regarding the heat pump application. Moreover, technologies shall be classified in a technology matrix and evaluated regarding specific advantages of single technologies for dedicated applications such as new buildings, retrofit, office, residential, etc. Technologies shall also be considered regarding different aspects of the definitions, e.g. characteristics regarding load match and grid interaction, the necessity of a grid connection, or the capability to integrate local storage.

Task 2 – Assessment of system technology
Task 2 is dedicated to identify promising concepts for the further development of system configurations and in-depth analysis of technologies and system configurations suitable for different applications in NZEB. Concepts shall be optimised by simulations regarding design, integration options, and control, but also regarding further aspects like self-consumption of energy, load match, and grid interaction, which shall be considered in Task 4. Evaluation is made based on energy performance and cost.

Task 3 – Technology development and field monitoring
Task 3 is dedicated to technology developments on the component and system level as well to gather field experiences of system solutions in **field monitoring projects**. Marketable and prototype technologies could be lab-tested or investigated in field monitoring. Task 3 is accomplished in parallel to Task 2.

Task 4 – Integration of NZEB into the energy system
Task 4 is also to be accomplished in parallel and deals with the integration of NZEB into the energy system. An NZEB should be designed in a way not to produce additional stress for the grid. In this respect, load match profiles and grid interaction shall be investigated in order to maximize self-consumption and minimize grid interaction. Thus, local storage options shall be evaluated. On the other hand, the ability of NZEB to react to signals from the grid (“smart grid”), i.e. demand response technology options shall be examined. Heat pumps are also a unique system in this respect due to the option of a transformation of electrical surplus to storable heating or cooling energy, as well in simultaneity, and due to the connection to source and sink systems which may be used as short-term storage like the ground. Control issues play an important role for these investigations, as well.

This report gives the results for the field monitoring project (**Task 3**) for **NORWAY**.

The Norwegian activities in IEA HPP Annex 40 are organized and carried out by SINTEF Building and Infrastructure (http://www.sintef.no/home/building-and-infrastructure), while COWI AS (www.cowi.no) and NTNU (http://www.ntnu.edu) are subcontracting partners. The project is funded by the governmental organization Enova SF (www.enova.no) and the Norwegian Research Centre on Zero Emission Buildings, ZEB (www.zeb.no).

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1. CO_2 Heat Pump Water Heaters in Block of Flats

1.1 Introduction

Tveita Borettslag (housing cooperative) in Oslo, Norway, comprises three blocks of flats with 819 flats. The buildings, which were erected 1967-69, were heavily renovated in 2011 in order to improve the thermal comfort for the residents and save energy. Among other things, the oil-fired boilers for each block were replaced with two independent heat pump systems – one heat pump that supplies heat to low-temperature radiators (space heating heat pump) and one heat pump for hot water heating (heat pump water heater).

Figure 1 Three block of flats at Tveita Borettslag (housing cooperative) with totally 819 flats.

The 3 heat pump water heaters (HPWH) use carbon dioxide (CO_2, R744) as the working fluid. This technology has a number of advantages:

- CO_2 heat pump water heaters represent one of the most energy efficient technologies for hot water heating and can even outperform solar heating systems with flat plate or vacuum tube solar collectors. The Coefficient of Performance (COP) is at least 20-30 % higher than the COP of the most energy efficient standard heat pump water heaters on the market.
- CO_2 heat pump water heaters can supply hot water at 60-90 °C. Hence, there is no need for reheating of water with electric immersion heaters, boilers, or district heating. This may have a considerable impact on the total annual energy saving of the system.
- CO_2 is a so-called "natural refrigerant" and does not give a net contribution to the greenhouse effect in case of unintentional leakages since the Global Warming Potential is zero (GWP=0) when used as a refrigerant. Conventional HFC refrigerants have a GWP-value between 1300 and 2000. CO_2 is also non-toxic and non-flammable (fire retardant), which means that it is a very safe refrigerant.

The CO_2 heat pump system at Tveita Borettslag was the first large-capacity CO_2 heat pump water heater (HPWH) system to be installed in Norway.

A Master student at The Norwegian University of Science and Technology (NTNU) in Trondheim has analysed and monitored one of the CO_2 HPWH (Borge, 2014). Although the buildings at Tveita Borettslag do not comply with the passive house or NZEB standard, the project is included in this Annex 40 report since CO_2 HPWH represent a very energy efficient, eco-friendly, and promising technology for the next generation of NZEB with a considerable hot water demand, i.e. block of flats, apartment buildings, hospitals, nursing homes, and sport centres. CO_2 HPWH can even be interesting in NZEB office buildings since the annual heating demand for a passive house office building in Norway typically constitute 20 to 25 % of the total annual heating demand. (Alonso et al., 2013).

1.2 The CO2 Heat Pump Cycle

1.2.1 Important fluid properties

Carbon dioxide (CO_2, R744) is a non-flammable, non-toxic working fluid (refrigerant) for heat pumps that neither contributes to ozone depletion (ODP=0) nor global warming (GWP=0) when used as a working fluid. CO_2 has a number of unique characteristics/properties:

- High operating pressure – small compressor volume
  - The critical pressure (pc) of CO_2 is as high as 73.8 bar, and the operating pressure in CO_2 heat pump systems is typically 5 to 10 times higher than that of plants using HydroFluoroCarbon working fluids (HFCs). This has a major impact on the pressure rating and design of pipelines and components.
  - Due to the relatively high enthalpy of evaporation (Δh_E) for CO_2 the required mass flow rate (kg/s) in CO_2 heat pumps is typically 5-15 % lower than that of HFC systems. On the other hand, the high operating pressure leads to an extremely high vapour density at the compressor inlet. Due to the combination of moderate mass flow rate and low vapour density, the required volume for CO_2-compressors is typically 4-10 times lower than that of HFC compressors.

- Excellent temperature fit in the gas cooler – high outlet water temperatures
  - The critical temperature (t_c) for CO_2 is as low as 31.1 °C. Since the set-point temperature for heat pump water heaters typically range from 50 to 70 °C, heat must be rejected at super-critical pressure (p < 73.8 bar) by cooling of high-pressure CO_2 vapour in a gas cooler (i.e. no condensation of the fluid). In a heat pump water heater the temperature drop (glide) for the CO_2 vapour is typically 50 to 70 °C, Figure 2.
  - A CO_2 heat pump water heater will operate in a so-called transcritical cycle with heat absorption at sub-critical pressure in the evaporator and heat rejection at super-critical pressure in the gas cooler, Figure 2.
  - Due to the relatively high inlet CO_2 vapour temperature in the gas cooler, counter-flow heat exchange and considerable temperature glide for both the CO_2 and the water, CO_2 heat pump water heaters can supply high-temperature water (60-90 °C), Figure 2. This eliminates the requirement for supplementary heating and increases the total COP for the hot water system.
- High compressor efficiency and superior heat transfer efficiency
  - Due to the low pressure ratio (π), the compressor efficiencies for CO_2 compressors will be higher than that of HFC compressors
  - The relatively flat saturation pressure curve ($\delta t/\delta p$) leads to relatively high optimum flow velocities in CO_2 heat pump systems. Together with the excellent heat transfer properties of CO_2 (low viscosity, high cp-value, high thermal conductivity etc.), CO_2 heat exchangers (evaporator, gas cooler etc.) obtain superior heat transfer efficiency.
- Small dimensions of pipelines and components
  - The moderate mass flow rate in combination with the relatively flat saturation pressure curve, $\delta t/\delta p$ leads to higher optimum flow velocities and smaller dimensions of components and pipelines than that of heat pump systems using HFCs, R290 and ammonia

1.2.2 High Coefficient of Performance (COP)

Laboratory testing, field measurements, and computer simulations have proved that CO_2 heat pump water heaters achieve minimum 20 % higher Coefficient of Performance (COP) than HFC heat pump water heaters. The main reasons are:

- Heat rejection at supercritical pressure resulting in good temperature fit between the CO_2 and the water in the counter-flow gas cooler. Low average temperature (tm) during heat rejection leads to high cycle COP.
- Superior heat transfer properties resulting in relatively low LMTD in the evaporator and relatively low average temperature difference in the gas cooler.
- High compressor efficiencies lead to relatively low input power for operation of the CO_2 heat pump water heater cycle.
- The capability of supplying high-temperature DHW (60-90ºC), which eliminates the requirement for supplementary (auxiliary) heating.

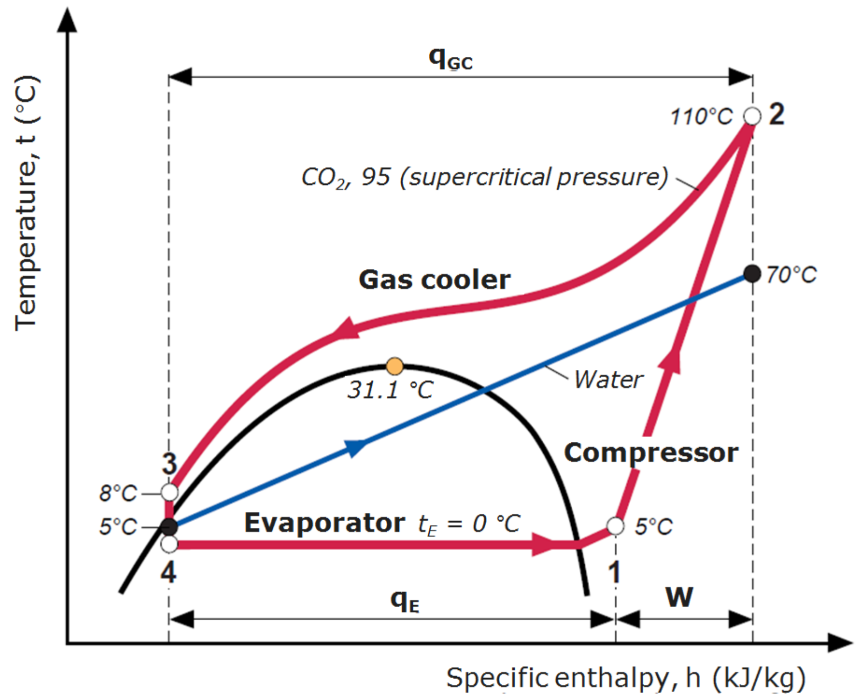


Figure 2 *Principle sketch of the CO_2 heat pump water heater cycle in a Temperature-Enthalpy diagram. In the example, CO_2 vapour is cooled down from 110 °C to 8 °C while the city water is heated from 5 °C to 70 °C. The evaporation temperature is 0 °C.*

Figure 3 shows a principle sketch of a CO_2 heat pump water heater connected to a single-shell hot water tank. The pump has variable speed drive, and during charging of the tank the rpm is adjusted according to the required outlet water temperature from the gas cooler.

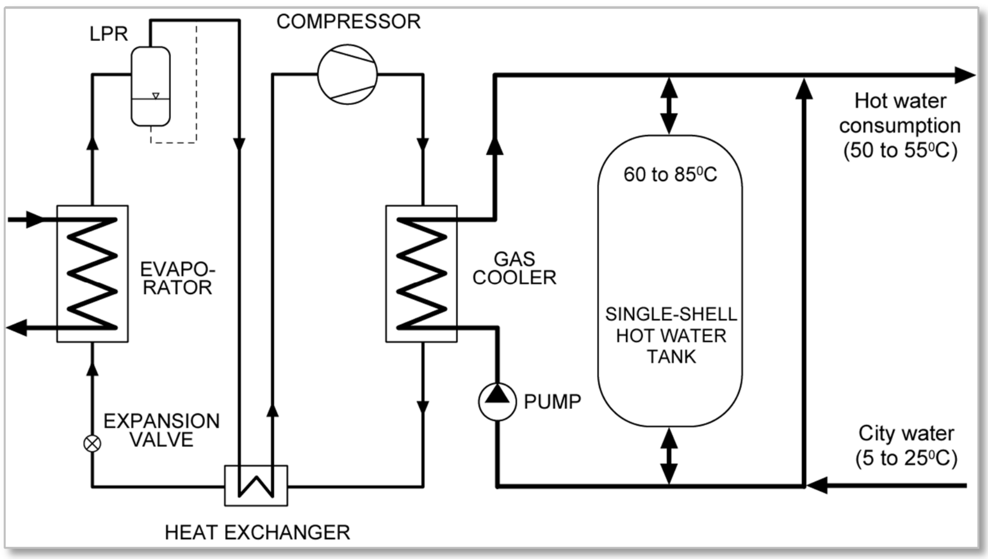


Figure 3 *Principle sketch of a CO_2 heat pump water heater system with a single-shell DHW tank and a variable speed drive (VSD) pump.*

More information about high-efficiency CO_2 heat pump water heaters can be found in Stene (2009) and Alonso/Stene (2009).

1.3 The Heat Pump Installation at Tveita Borettslag

1.3.1 Annual Heating Demand for DHW

Tveita Borettslag comprise tree identical block of flats with 3 x 273 apartments. A CO_2 heat pump water heater system covers the entire hot water demand for each of the buildings. The energy and power demands for heating of domestic hot water (DHW) for each block of flats are as follows (measured values):

- Annual heating demand – 600.000 kWh/year, approx. 2200 kWh/year for each flat
- Average power demand – 100 kW, approx. 360 W for each flat

1.3.2 Overall System Design

The CO_2 heat pump water heater system has the following sub-systems:

A. Heat source system – exhaust ventilation air – *Section 1.3.3*
B. CO_2 heat pump unit – *Section 1.3.4*
C. Hot water system – *Section 1.3.5*

Figure 4 shows the overall design for the domestic hot water (DHW) heating system in each block of flats at Tveita borettslag (Kuldeteknisk, 2014).

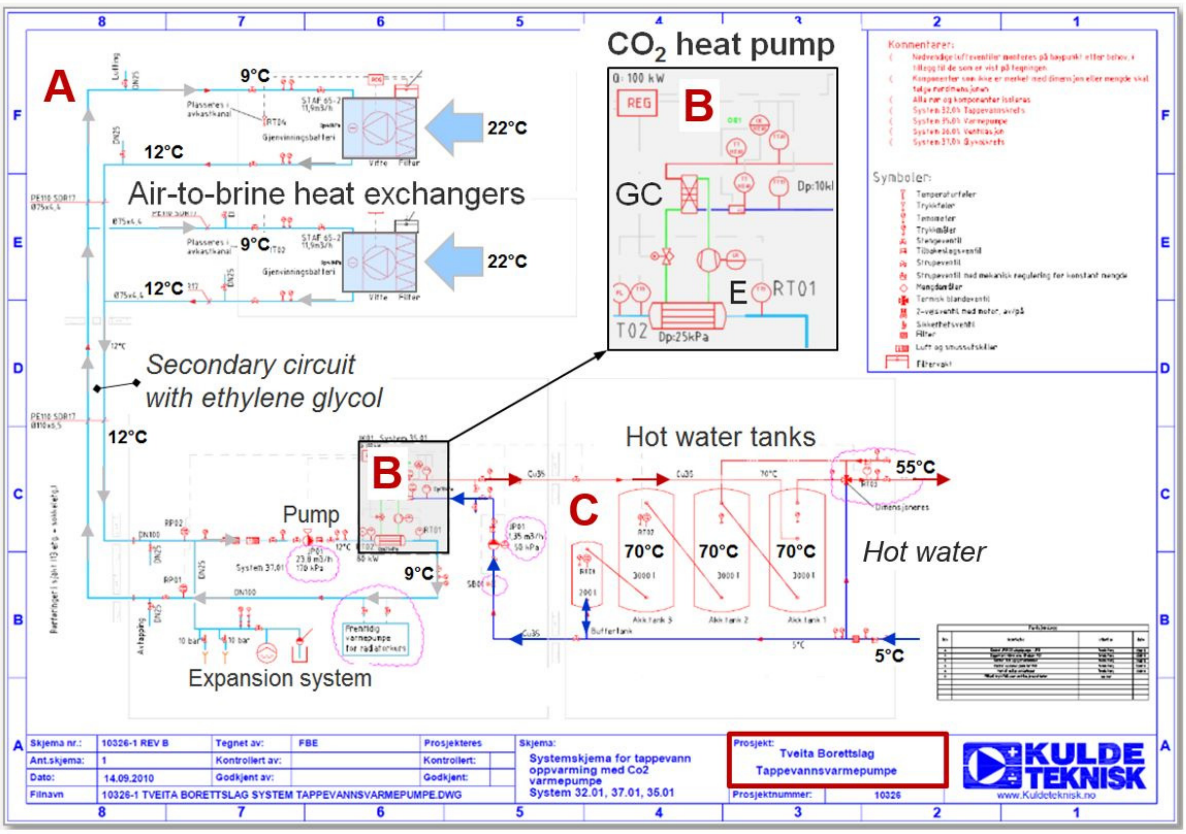


Figure 4 Overall design of the DHW heating system at Tveita Borettslag (Kuldeteknisk, 2014).

1.3.3 Heat Source – Exhaust Air from the Ventilation System

Although the building envelope and technical installations, including the heating system, were heavily renovated, the existing exhaust air ventilation systems were not replaced with balanced ventilation systems due to excessive costs. Exhaust air at a nearly constant temperature of approx. 22 °C from two exhaust ducts at the roof-top was therefore considered to be the most cost-efficient heat source for the heat pump system. By utilizing the exhaust air as heat source for CO_2 heat pump water heaters, the ventilation loss has been greatly reduced. Due to the large available air flow rate, the exhaust air has also been used as heat source for heat pump systems that cover the space heating demand in each of the three buildings.

The heat source system for each block of flats comprises:

- 2 x brine-to-air heat exchangers for heat recovery from exhaust air
- Secondary (closed loop) system with pumps, expansion system, brine treatment system, deaerater, and various valves – charged with anti-freeze fluid (ethylene glycol)

The secondary system is connected to the evaporator for the CO_2 heat pump water heater unit. At design conditions the inlet/outlet brine temperatures for the brine-to-air heat exchangers are *12/9 °C*. In a block of flats of passive house standard with high-efficiency heat recovery from a balanced ventilation system, boreholes in bedrock (ground-source) or ambient air (air-source) are the most relevant heat sources for a CO_2 heat pump water heater.

1.3.4 The CO_2 Heat Pump Unit

The single-stage CO_2 heat pump unit from Green & Cool (model Pacific BO 1115 HT) has the following main components:

- Evaporator – plate-and-shell heat exchanger (flooded type – the CO_2 liquid reservoir also serves as a low-pressure, i.e. a CO_2 liquid storage for high-side pressure control)
- Liquid separator
- Compressor – reciprocating (piston), intermittent operation (on/off)
- Gas cooler – 2 x plate heat exchangers connected in series, counter-flow operation
- Suction gas heat exchanger – plate heat exchanger
- Expansion valve – electronic back-pressure valve
- Oil return system

The nominal heating capacity is 100 kW at 12/9 °C on the evaporator side and 5/70 °C on the gas cooler side. Figure 5 shows the design and main pipeline dimensions for the CO_2 heat pump unit (Borge, 2014).

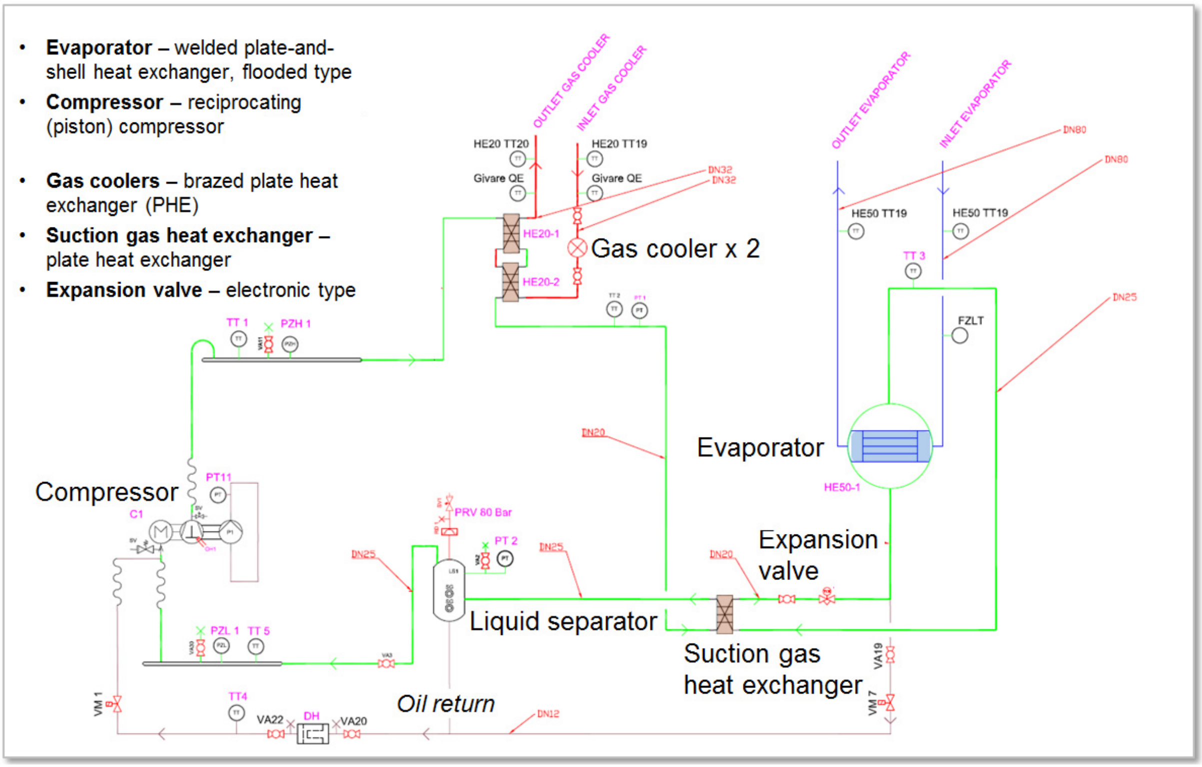


Figure 5 Main components and pipeline dimensions for the 100 kW CO_2 heat pump water heater unit at Tveita Borettslag (Borge, 2014).

The CO_2 heat pump unit operates with a constant *100 bar* high-side pressure in the gas cooler, maintained by the electronic back-pressure valve. 100 bar is more or less the optimum gas cooler pressure at the prevailing operating conditions, which leads to the highest possible Coefficient of Performance (COP) for the heat pump. The set-point for the outlet water temperature from the gas cooler is *73 °C*. The compressor is switched on when the temperature in DHW storage tank 13 (*Figure 6*) drops below *60 °C*, and the compressor is switched off when the inlet water temperature in the gas cooler exceeds *20 °C*.

1.3.5 The Hot Water System

The hot water system for each block of flats comprises 13 x 400 litres hot water storage tanks connected in series, electric heaters for reheating of DHW in the recirculation pipeline, expansion system, and various valves, *Figure 6* and *Figure 7*.

A 400 litre tank has a smaller diameter and cross sectional area than 650 and 1000 litres tanks, resulting in less heat conduction between hot and cold water during tapping and charging of the tanks. Heat conduction will increase the average inlet water temperature to the gas cooler and consequently reduce the COP for the CO_2 heat pump water heater.

The storage tanks have a relatively small tube diameter on inlet/outlet (OD 32 mm), and diffusers have not been installed in order to reduce the water velocity. At maximum water flow rate during tapping mode the water velocity through the tanks becomes quite high, resulting in mixing of water at different temperature levels. This in turn increases the average inlet water temperature to the gas cooler and reduces the COP of the CO_2 heat pump.

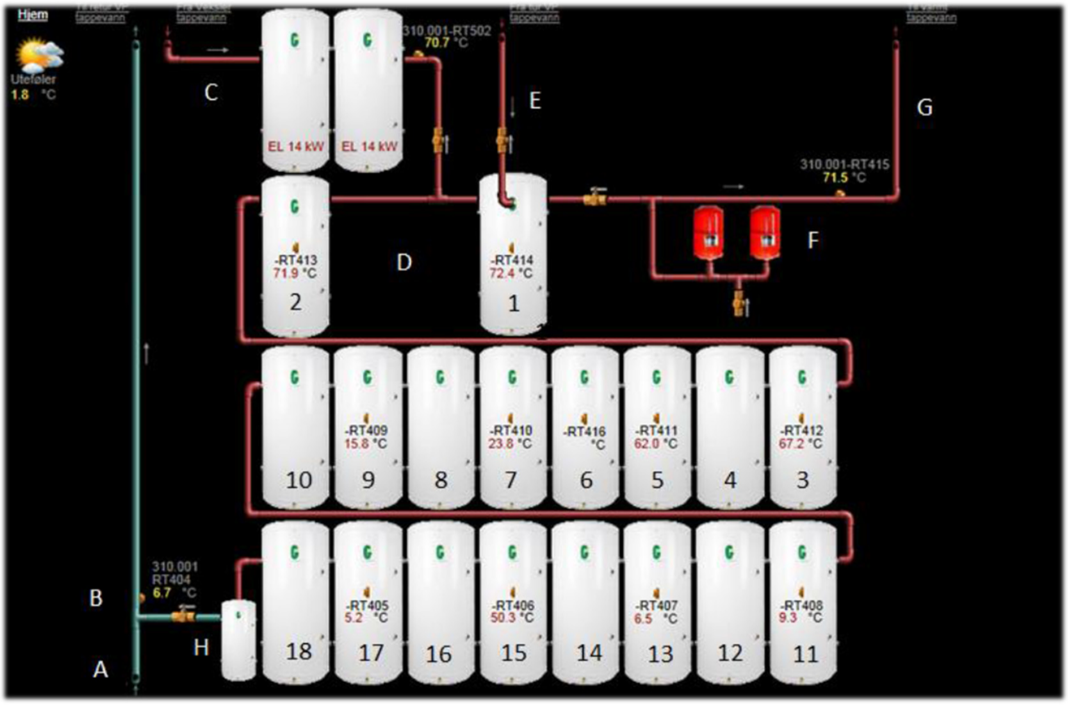


Figure 6	DHW system design. A – City water, B – Temperature sensor, C – Return from DHW recirculation system to electric heaters (reheater), D – DHW storage tanks, E – Hot water from CO_2 heat pump (73 °C), F – Expansion system, G – Hot water distribution to flats, H – Buffer tank (Borge, 2014).

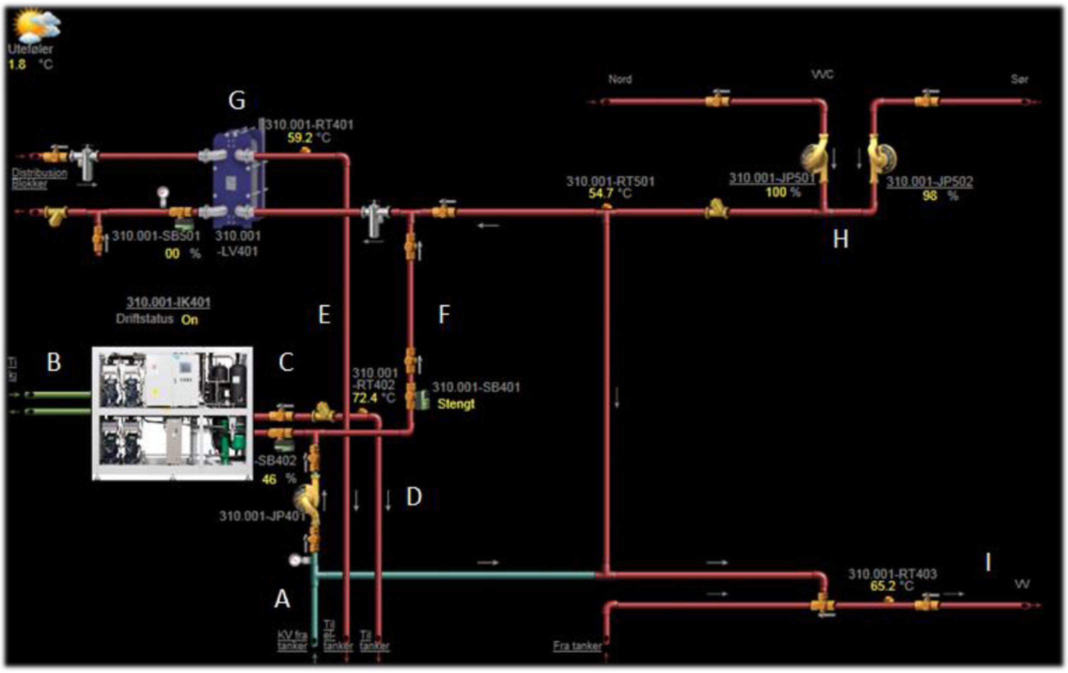


Figure 7	DHW system design. A – city water (5-10 °C), B – Evaporator side, CO_2 heat pump, C – Gas cooler side, CO_2 heat pump (approx. 73 °C), D – Hot water from gas cooler to storage tanks, E – Return from recirculation system to two storage tanks with electric heaters, F – City water, G – Back-up heat exchanger, H – Return pipeline for recirculation system, I – Supply pipeline for recirculation system (Borge, 2014).

1.3.6 Monitoring

The CO_2 heat pump water heater is equipped with a monitoring system from Green & Cool. *Table 1* shows the measured water and brine temperatures, CO_2 temperatures (evaporator, gas cooler), CO_2 pressures (evaporator, gas cooler) as well as the heating capacity for the gas cooler (Borge, 2014). The measuring uncertainty for the brine temperatures is ±0.5 °C, while the uncertainty ranges from ±0.5-1.0 for the water and CO_2 temperatures. The measuring uncertainty for the CO_2 pressure is ±0.5 bar for the evaporator (low pressure level, 30-50 bar) and ±1.0 bar for the gas cooler (high pressure level, 100 bar).

Table 1 Measuring points for the CO_2 heat pump water heater unit (Borge, 2014)

Sensor	Measurement	Measuring uncertainty
TT8	Evaporator – ethylene glycol – inlet temperature (°C)	±0.5 °C
TT9	Evaporator – ethylene glycol – outlet temperature (°C)	±0.5 °C
TT3	Evaporator – CO_2 – inlet temperature (°C)	±1.0 °C
TT26	Evaporator – CO_2 – outlet temperature (°C)	±1.0 °C
PT2	Evaporator – pressure (bar)	±0.5 bar
TT5	Compressor – CO_2 – inlet temperature (°C)	±1.0 °C
TT1	Compressor – CO_2 – outlet temperature (°C)	±1.0 °C
TT19	Gas cooler – water – inlet temperature (°C)	±1.0 °C
TT20	Gas cooler – water – inlet temperature (°C)	±1.0 °C
TT2	Gas cooler – CO_2 – outlet temperature (°C)	±1.0 °C
PT1	Gas cooler – pressure (bar)	±1.0 bar
E1	Gas cooler – heating capacity (kW) – energy meter	*

Figure 8 shows the monitoring display for the CO_2 heat pump water heater (Borge, 2014).

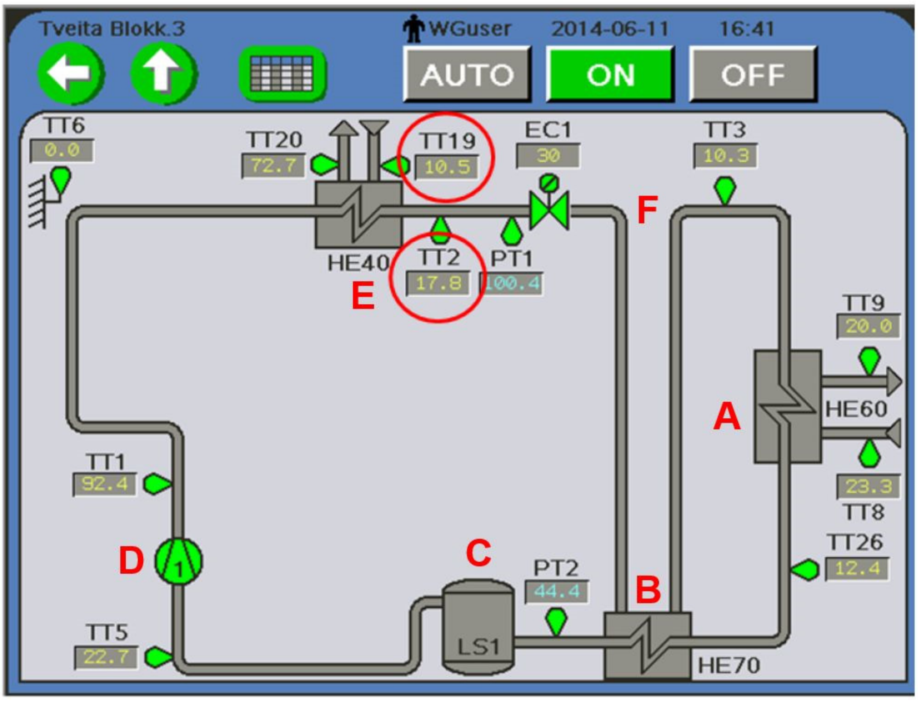


Figure 8 Monitoring display for the CO_2 heat pump water heater. TT – Temperature sensors, PT – Pressure sensors, A – Evaporator, B – Suction gas heat exchanger, C – Liquid separator, D – Compressor, E – Gas cooler, F – Electronic expansion valve (Borge, 2014).

Figure 9 shows the monitoring display for the thermal energy meter (heat meter) for the gas cooler, comprising a volume flow meter and temperature sensors (Borge, 2014).

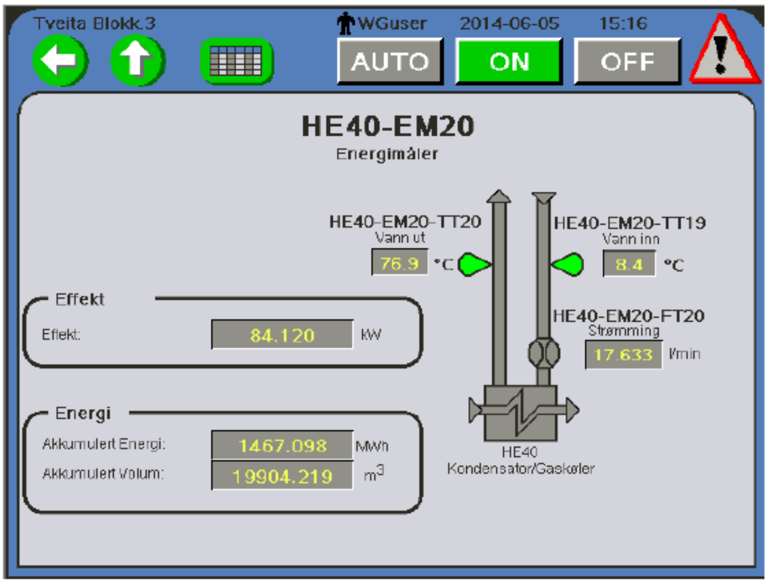


Figure 9 Monitoring display for the thermal energy meter for the gas cooler (Borge, 2014).

1.3.7 Measuring results

1.3.7.1 Gas Cooler Performance and Heating Capacity

Figure 10 shows the measured gas cooler pressure and outlet water temperature from the gas cooler during operation and standstill. When the CO_2 heat pump supplies heat to the DHW system, the gas cooler pressure is approx. 100 bar while the outlet water temperature is approx. 73 °C.

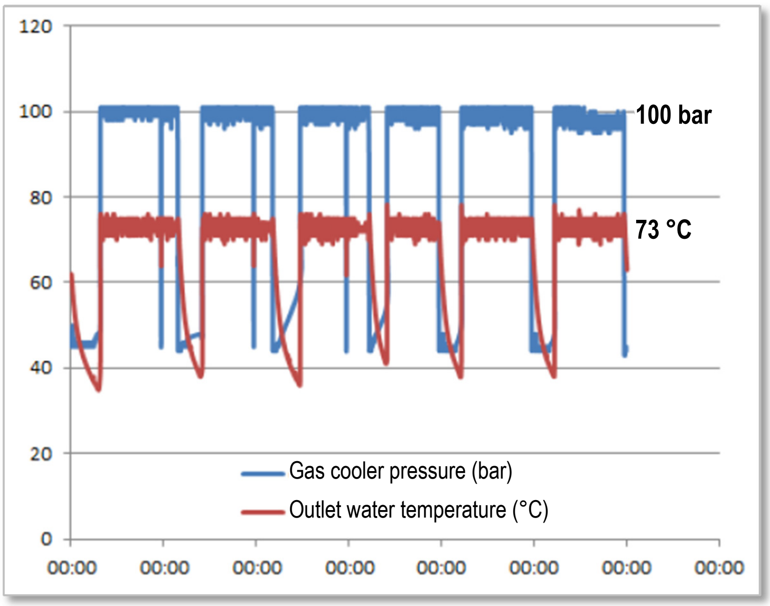


Figure 10 Measured CO_2 pressure and outlet water temperature for the gas cooler (Borge, 2014).

Figure 11 shows the gas cooler heating capacity during one year of operation. The reason for the variations in the heating capacity is mainly due to the fact that a 280 kW space heating heat pump is connected to the same heat source (secondary) system. During summer operation there is no space heating demand, and the CO_2 heat pump water heater achieves the designed heating capacity of approx. 100-110 kW. As soon as the other heat pump starts to operate, the temperature level for the brine in the secondary system drops, resulting in a lower evaporation temperature for the CO_2 heat pump and consequently a lower heating capacity. As a general rule, the heating capacity is reduced by about 3 % for each °C drop in evaporation temperature. The lowest measured heating capacity is about 85 kW.

The average heating capacity during one year of operation is approx. *90-95 kW*.

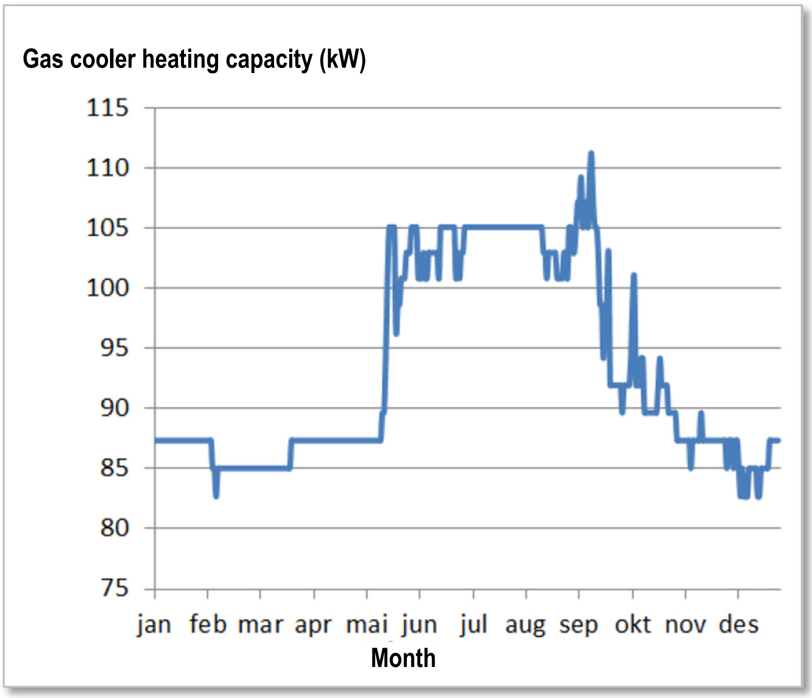


Figure 11 Heating capacity of the CO_2 heat pump during one year of operation (Borge, 2014)

1.3.7.2 Seasonal Performance Factor (SPF)

Due to limitations in the measuring equipment and the monitoring system it has not been able to make an exact calculation of the Seasonal Performance Factor (SPF) or Seasonal COP (SCOP) for the CO_2 heat pump water heater system.

- The annual heat supply (kWh/year) from the CO_2 heat pump water heater had to be estimated since the monitoring system does not store historical data for the thermal energy meter for a longer period of time. The annual heat supply for DHW heating was estimated to be *1,775,000 kWh/year.*
- The compressor was not equipped with a power meter. Hence, the input power and energy to the compressor was estimated by using the computer programme CoolPack with measured input values for evaporation temperature and pressure, gas cooler pressure, outlet CO_2 temperature from the gas cooler, and isentropic efficiency (η_{is}) for the compressor (relatively constant value). The annual input energy for the compressor was estimated to be *395,000 kWh/year*.
- The estimated annual energy saving was approx. *1,360,000 kWh/year*.

- The estimated average COP (SCOP or SPF) for the CO_2 heat pump water heater was approx. **SCOP = 4.4**. This corresponds to an annual energy saving of roughly 75 % compared to a direct electric heating system (electric immersion heaters).
    - The estimated SPF does <u>not</u> include energy use for pumps in the heat source and DHW systems, heat supply from electric heaters in the recirculation pipeline system or heat losses from DHW pipelines and storage tanks.

Figure 12 100 kW CO_2 heat pump water at Tveita Borettslag (Green&Cool, 2014).

- A 350 kW CO_2 heat pump water heater (HPWH) installed in a hospital in Tromsø (2014 – combined liquid chiller and heat pump system) and a 30 kW HPWH installed in a block of flats in Trondheim (2014 – exhaust air heat pump system) have also been monitored. The measurements have proved that at similar temperature levels for the heat source system (5-8 °C) and hot water system (70-75 °C) an **SCOP in the order of 4** can be achieved when including the electric energy input for the pumps for the heat source system and the hot water system.

1.3.8 Improvements of the CO_2 Heat Pump Water Heater System

The performance of the CO_2 heat pump water heater system at Tveita housing cooperative (block of flats) can be improved by implementing the following measures:

- <u>Control system</u>
    - *Current system* – CO_2 gas cooler is operated constantly at 100 bar
    - *Recommendation* – Install controller for optimization of the gas cooler pressure according to the operating condition (variable heat source temperature)
    - *Benefit* – Increases the seasonal COP of the CO_2 heat pump (SCOP) since maximum COP is achieved at all operating conditions.
- <u>Domestic hot water storage tanks</u>
    - *Current system* – The DHW accumulation system comprises 18 x 400 litres single-shell hot water storage tanks - small diameter tubes at inlet/outlet and no diffusers at the inside of the tanks to reduce water velocity.
    - *Recommendation* – Replace the existing accumulation system with a smaller number of larger DHW tanks, if possible tall and slim tanks with large height/diameter ratio, large diameter inlet/outlet piping and diffusers at inlet/outlet, in order to minimize water velocity and mixing of hot/cold water.
    - *Benefit* – Reduces mixing of hot and cold water in the tanks (reduces exergy loss) due to lower water velocities inside the tanks. This will in turn increase the seasonal COP of the CO_2 heat pump (SCOP) due to lower average water temperature at the gas cooler inlet.

- Pumps
    - *Current system* – Pumps with constant rpm. The water flow rates, including the flow rate through the CO_2 gas cooler, are controlled by control valves.
    - *Recommendation* – Replace pumps with variable speed drive (VSD) pumps, incl. the pump for the CO_2 gas cooler, and remove all control valves.
    - *Benefit* – Lowers the annual energy consumption for pumps and thereby increases the total energy saving of the CO_2 heat pump water heater system.
- Metering and monitoring system
    - *Current system* – Simplified metering and monitoring system. No real time data logging/sampling.
    - *Recommendation* – Extend the metering system with more temperature sensors in the CO_2 and water circuits as well as to install more flow meters in the water circuit as well as power meters for the compressors. Replace existing monitoring system with a more advanced monitoring system with real time logging/sampling and accumulative/historic data storage.
    - *Benefit* – Enables direct calculation of COP for the CO_2 heat pump water heaters. Improves the quality of the monitoring system which in turn gives the operating personnel the best possible tool for optimized operation.
- Insulation of pipelines
    - *Current system* – Old building with uninsulated hot water pipeline system
    - *Recommendation* – Install new pipeline system which is insulated according to the Norwegian standard NS12828.
    - *Benefit* – Reduces heat losses from DHW pipelines to a minimum and increases overall system energy efficiency since less heat is required for reheating of DHW in the recirculation system.

1.4 Conclusions – Recommendations with regards to Design and Operation

CO_2 heat pump water heaters (HPWH) represent a very energy efficient, eco-friendly, and promising technology for the next generation of NZEB with a DHW demand, i.e. block of flats, apartment buildings, hospitals, nursing homes, sport centres, etc. CO_2 HPWH can even be interesting in NZEB office buildings since the annual heating demand for a passive house office building in Norway typically constitute 20-25 % of the total annual heating demand.

- CO_2 (R744) has favourable environmental properties: GWP_{CO2}=0 when CO_2 is used as a working fluid – non-toxic and non-flammable fluid – the high operating pressure (< 100 bar) does not constitute an increased risk compared to HFC systems due to the limited volume of tubes and vessels (i.e. p·V is in the same order of magnitude).
- CO_2 heat pump water heaters (HPWH) can provide domestic hot water at **60-90 °C** and still achieve a high COP due to the optimized cycle for DHW heating.

- Important features regarding heating capacity, components, design and operation:
  - Heating capacity
    - Should equal the average thermal power demand for DHW heating over a 24 hour operating period + 25-30 %. The peak load DHW demand is covered by the DHW stored in the accumulation tanks.
  - Standard components for CO_2 HPWH units:
    - Compressor – intermittent operation (on/off control)
    - Gas cooler – rejects heat to the DHW
      - The heat exchanger should have a long thermal length. The temperature approach (Δt_A), i.e. the temperature difference between the CO_2 and the inlet city water at the gas cooler inlet, should be 4-6 °C in the design point
      - Counter-flow operation is crucial for minimum Δt_A (maximum cool down of the CO_2 gas) and maximum COP
      - The outlet DHW temperature from the gas cooler (t_{DHW}) should be controlled by a variable speed drive (VSD) pump
    - Suction gas heat exchanger – superheats the compressor suction gas
      - Prevents liquid droplets from entering the compressor
      - Reduces the optimum gas cooler pressure due to increased CO_2 gas temperature at the gas cooler inlet
      - Increases the COP by max. 5 %
    - Expansion valve (back-pressure valve) – controls the gas cooler pressure together with the low-pressure receiver (liquid CO_2 reservoir)
      - Optimum gas cooler pressure control is recommended in all systems with variable city water temperature and/or variable heat source temperature. The higher the DHW set-point temperature, the higher the optimum gas cooler pressure.
      - Ejector – the expansion valve can be replaced by an ejector that increases the suction pressure for the compressor by typically 4-8 bar, and with that it increases the COP
    - Evaporator
    - Low-pressure receiver (LPR) – liquid CO_2 reservoir for gas cooler pressure control together with the expansion valve
      - Equipped with integrated oil return system
  - DHW storage tanks (accumulation tanks) – covers DHW peak demands
    - Designed according to the DHW peak load demand, leading to typically 18-20 operating hours for the CO_2 HPWH
    - Single-shell tanks connected in series
    - The tanks should preferably be tall and slim in order to minimize conductive heat transfer between hot and cold water inside the tanks during tapping (DHW draw off) and thermal charging

- The inlet/outlet water pipelines should have adequate diffusers in order to minimize the water velocity and consequent mixing of hot and cold water. The inlet/outlet tubes should have sufficiently large diameters.
- A separate storage tank for the DHW recirculation system may be used. The tank should have an electric heater for DHW reheating.

o DHW pipelines

- Adequate insulation of all DHW pipelines in order to minimize heat loss from the recirculation system and energy use for reheating of DHW

Figure 13 shows, as an example, a CO_2 heat pump water heater (HPWH) system designed for e.g. an NZEB block of flats, an apartment building or a nursery home. The DHW system comprises an air-source, water-source or brine-source CO_2 HPWH unit, four DHW storage tanks connected in series, a high-efficiency DHW pump with variable speed drive, integrated electric immersion heaters as back-up in DHW storage tank no. 4, and a separate DHW tank (5) with an integrated electric immersion heater for the DHW recirculation system.

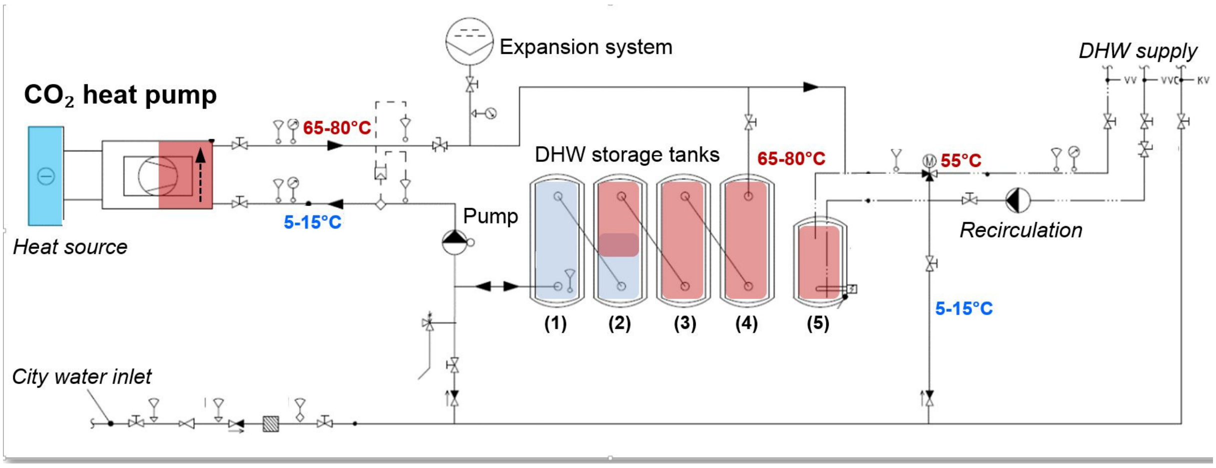


Figure 13 Example – A CO_2 heat pump water heater (HPWH) installation with a CO_2 heat pump unit, DHW storage tanks, and a pump for water circulation through the gas cooler.

Overall control strategy:

- During the DHW tapping (draw off) periods, cold city water at 5-15 °C flows into the bottom of DHW tank 1 whereas the same amount of DHW flows at 65-80 °C from the top of tank 4 through the mixing valve to the various tapping sites (max. 55 °C).
- The CO_2 heat pump will normally run during the DHW tapping (draw off) periods.
- A variable speed pump circulates the cold city water through the counter-flow gas cooler. The gas cooler heats the water to the set-point temperature (65-80 °C) before it flows into DHW tank 4.
- When the tapping (draw off) period has ceased, the CO_2 heat pump will be running as long as the water temperature at the bottom of DHW tank 1 is lower than the set-point temperature (65-80 °C).
- The gas cooler pressure for the CO2 HPWH should be continuously optimized in order maximize the COP for the heat pump at varying heat source temperature and inlet city water temperature in the gas cooler.

2. Heat Pump System in a New Office Building

2.1 Introduction

Miljøhuset GK is the headquarters of the contracting company GK Norge AS. The building, which was opened in 2012, is situated in Oslo and has a total heated area of *13,650 m²* including a repair shop and storage rooms. The building was constructed according to the Norwegian passive house standard NS3701 (Class A) and was certified as *"Very good"* according to the BREEAM-Nor classification system (http://ngbc.no/english).

The building has *no hydronic heat distribution system*, and space heating and space cooling are solely provided by heating and cooling of the ventilation air. An air-source heat pump system covers the entire space cooling demand and the base load for the space heating. Electric heaters are used as peak load for heating. A separate chiller covers the process cooling demand, i.e. cooling of server stations and telecom equipment.

Figure 14 Miljøhuset GK – a 13,650 m² office building in Oslo of passive house standard. The building was certified as "Very good" according to the BREEAM-Nor classification system.

2.2 System for Thermal Energy Supply

2.2.1 The Energy Plant – Heating and Cooling

The heating and cooling demands for the passive house building include heating of ventilation air, domestic hot water (DHW) heating, snow melting, and process cooling. The energy plant comprises the following sub-systems:

- Reversible heat pump/liquid chiller system – space heating/cooling, snow melting
  - **320 kW** heating cap. (winter mode) – **500 kW** cooling cap. (summer mode)
- Electro boiler (*200 kW*) – peak load and back-up for space heating and snow melting
- Electric immersion heaters – DHW heating (integrated coils in the storage tanks)
- Electric heaters – *200 W* peak load for space heating (located in the rooms)
- Liquid chiller (*25 kW*) – process cooling (server/telecom equipment), heat recovery
- Dry cooler – rejection of excess heat from the liquid chiller circuit

Figure 15 shows a simplified sketch of the energy plant at Miljøhuset GK incl. the reversible heat pump and liquid chiller units, electro boiler, electric immersion heaters (DHW system), liquid chiller for process cooling, heat exchangers (A, B), and dry cooler (Orvik 2014/15).

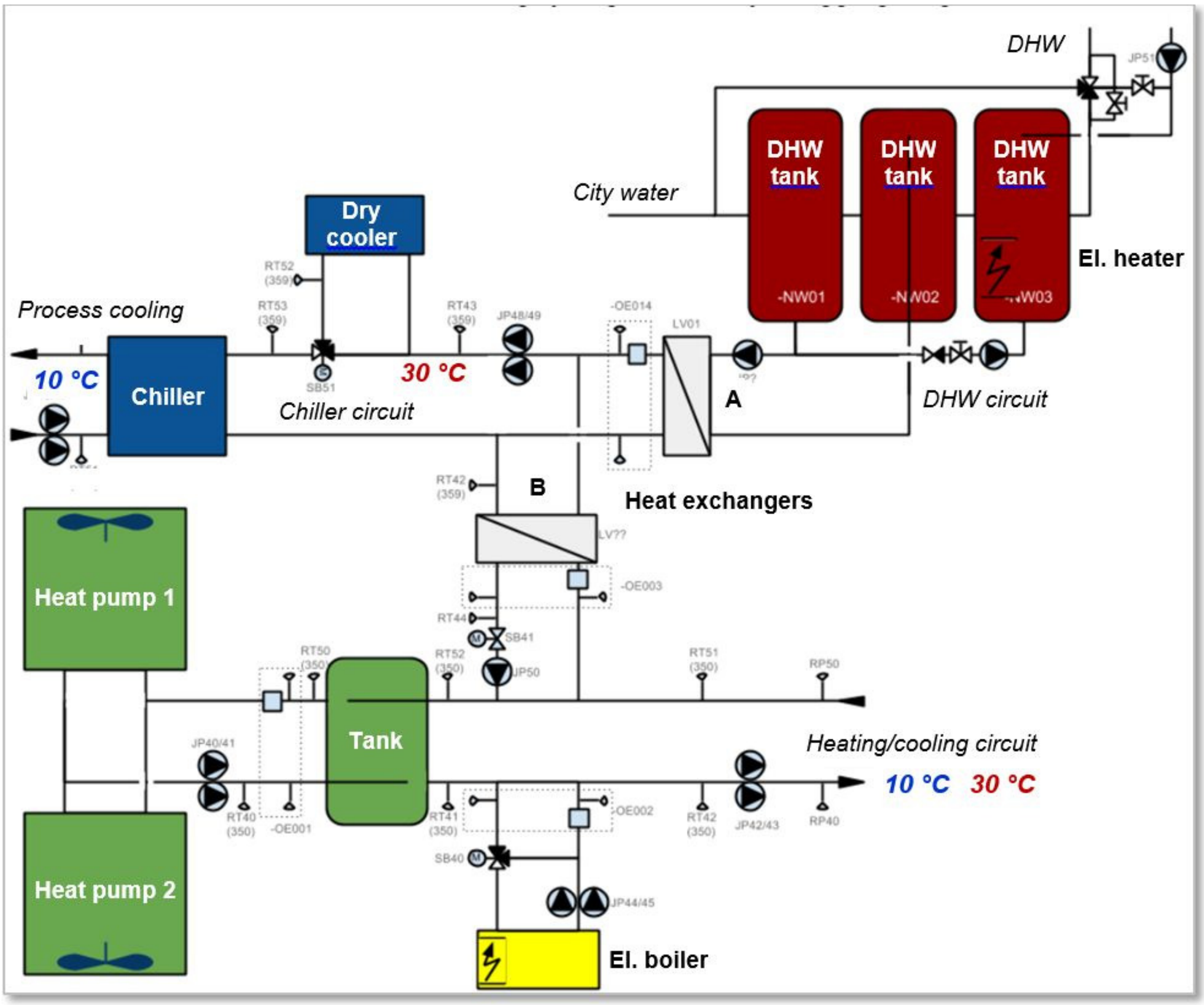


Figure 15 Simplified sketch of the energy plant for heating/cooling at Miljøhuset GK (Orvik, 2014/15).

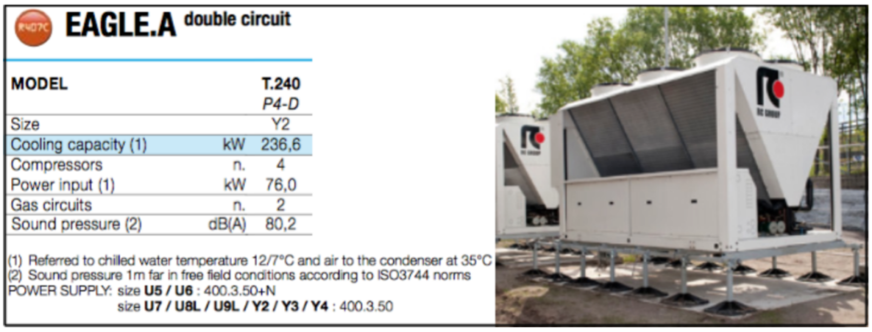
The standard heat pump/chiller units and the electro boiler are connected to a common closed-loop pipeline system for heating and cooling supplying warm/chilled liquid to heater/cooling batteries in 6 different ventilation units. The distribution system is charged with 30 % ethylene glycol in order to avoid freezing in the outdoor heat exchangers (evaporators) during winter operation. The distribution temperatures at overall heating mode and overall cooling mode are approx. *30 °C* and *10 °C*, respectively.

The condenser heat from the liquid chiller for process cooling is utilized for preheating of domestic hot water (DHW) and space heating, whenever possible. Two plate heat exchangers (A, B) separate the chiller circuit from the DHW circuit and the space heating circuit. Surplus condenser heat from the chiller is rejected to the ambient by means of a dry cooler system.

The DHW system comprises three storage tanks in serial connection, and electric immersion heaters reheat the preheated water to about 60 °C.

The heat pump installation includes two identical *reversible air-source units* that either provide heating or cooling to the common distribution system for heating/cooling:

- Type: Standard air-to-brine heat pump/chiller unit
- Heating/cooling capacity: **250 kW** (+7/35 °C), **160 kW** (-15/35 °C)
- Working fluid: R410A – 2 independent circuits per unit
- Compressors: 4 scroll compressors, intermittent operation
- Stop temperature: -12 °C



EAGLE.A double circuit

MODEL		T.240 P4-D
Size		Y2
Cooling capacity (1)	kW	236,6
Compressors	n.	4
Power input (1)	kW	76,0
Gas circuits	n.	2
Sound pressure (2)	dB(A)	80,2

(1) Referred to chilled water temperature 12/7°C and air to the condenser at 35°C
(2) Sound pressure 1m far in free field conditions according to ISO3744 norms
POWER SUPPLY: size **U5 / U6** : 400.3.50+N
size **U7 / U8L / U9L / Y2 / Y3 / Y4** : 400.3.50

Figure 16 Specification for one of the R410A air-source heat pump unit (Orvik, 2014/15).

2.2.2 Distribution System for Space Heating and Space Cooling

The building has no hydronic heat distribution system, and space heating and space cooling are solely provided by heating and cooling of the ventilation air. According to GK Norge AS, the ventilation efficiency and with that the indoor air quality is maintained by using special inlet valves that control the air distribution at varying air flow rates (VAV system) and varying air temperatures. *Figure 17* shows a simplified sketch of one of the 6 ventilation units equipped with a combined heating and cooling battery ("Heat exchanger") that provides either heating or cooling of the ventilation air.

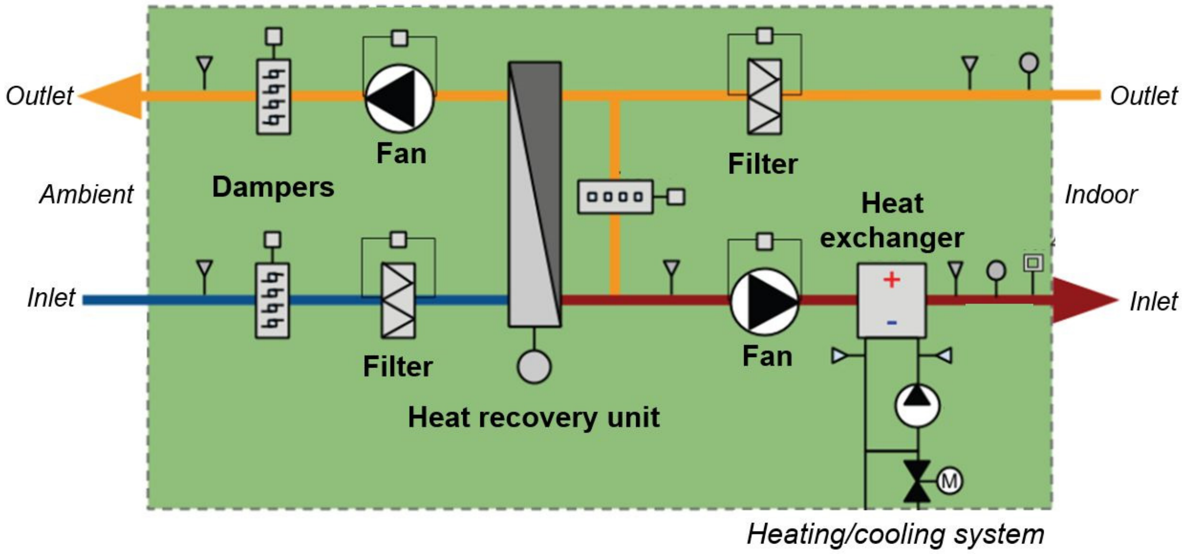


Figure 17 Simplified sketch of one of the 6 ventilation units comprising inlet and outlet ventilation ducts, heat recovery unit (rotating wheel), fans, heating and cooling battery ("Heat exchanger"), filters and dampers (Orvik, 2014/15).

2.3 Monitoring and Power/Energy Analysis

A student at The Norwegian University of Science and Technology (NTNU) in Trondheim has, through his Project and Master work, monitored and carried out an in-depth analysis of the heating and cooling systems (Orvik, 2014/15).

The entire heating and cooling system at Miljøbygget GK is connected to a building energy management system (BEMS), *Figure 18*.

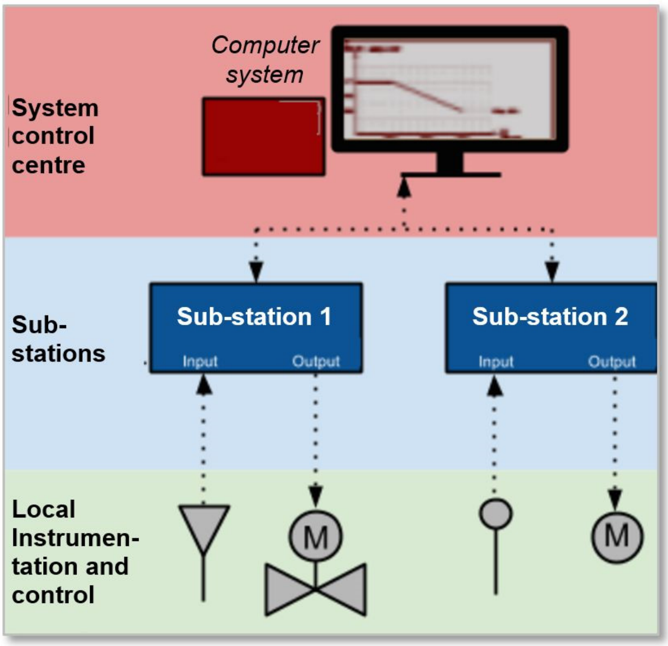


Figure 18 Principle illustration of the building energy management system (Orvik, 2014/15).

The instrumentation for the heating and cooling system includes temperature sensors, pressure sensors, volume flow meters, valve position elements, capacity elements, etc. This enables detailed monitoring of the status and performance for the components and sub-systems including the heat pump units. *Figure 19* shows an example of the user interface for the building energy management system (BEMS) – heating and cooling system.

2.3.1 Measurements

The power and energy measurements for Miljøhuset GK have been carried out at system level only. *Table 2* shows the measured thermal energy supply (heating, cooling) for the different sub-systems as well as the average, annual COP (SCOP) for the reversible heat pump/chiller and liquid chiller in 2013 (Orvik, 2014/15).

Table 2 The measured energy balance and SCOPs in 2013 (Orvik, 2014/15).

	Heating	%	Cooling	%	SCOP
Heat pump/chiller	125 MWh/year	42 %	75 MWh/year	26 %	2.6
Liquid chiller	110 MWh/year	36 %	216 MWh/year	74 %	4.0
Electro boiler	15 MWh/year	5 %			
Electric heater, DHW	52 MWh/year	17 %			
TOTAL	*302 MWh/year*	*100 %*	*291 MWh/year*	*100 %*	

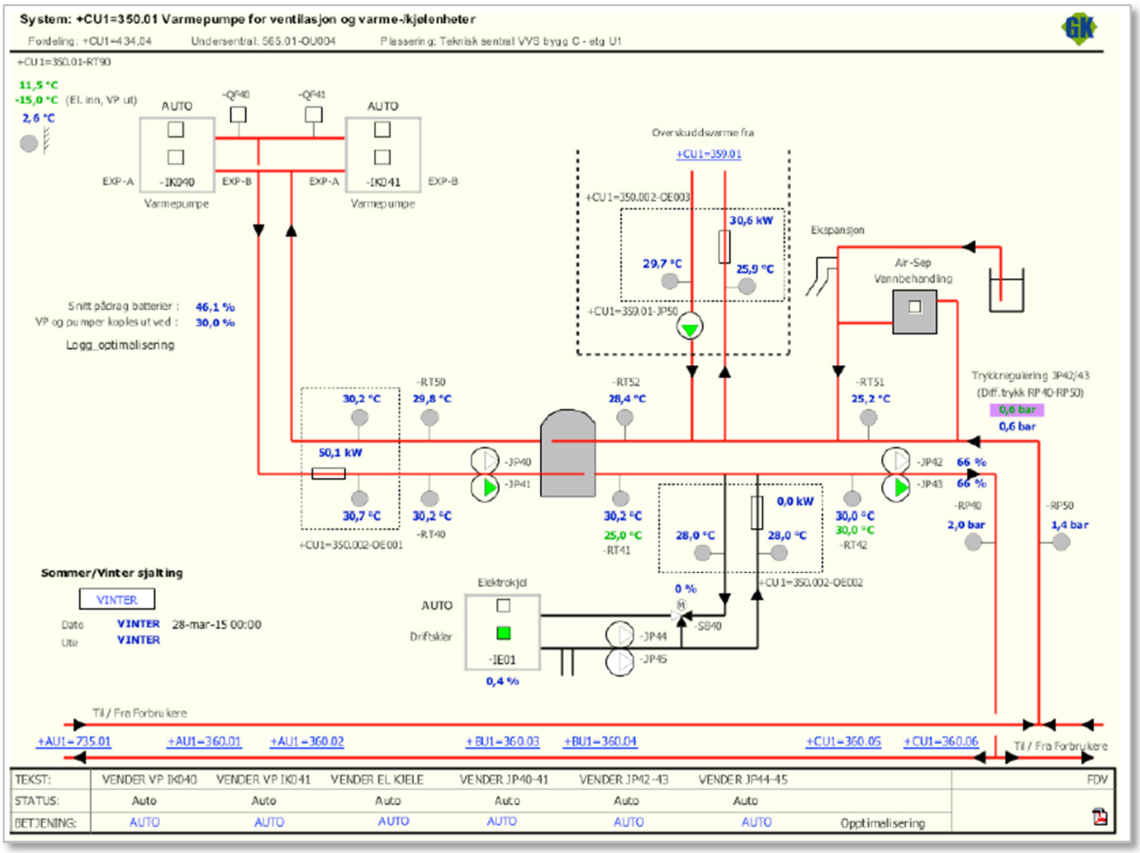


Figure 19 *Example of user interface for the BEMS – heating and cooling system (Orvik, 2014/15).*

Figure 20 shows the measured total electric input energy (kWh/year), the measured thermal energy supply – heating, cooling (kWh/year) with and without snow melting and process cooling as well as the calculated thermal energy demands (kWh/year) for 2013 (Orvik, 2014).

The theoretical calculations of the maximum heating/cooling demands (kW) and the annual heating/cooling demands (kWh/year) were carried out prior to the building project, and the results were used as a basis for classification of the building according to the Norwegian passive house standard ("Passive house Class A") and according to BREEM-Nor ("Very Good"). The Norwegian building simulation programme "Simien", which is based on the Norwegian standard NS3031, was used for the simulations. The simulations were based on a normal reference year for the climate and standardised values for periods of use for the building incl. ventilation systems during weekdays and weekends, power demand for electrical appliances, presence of people, etc. Process cooling and recovery of condenser heat for e.g. space heating and preheating of DHW were not included in the simulations.

Figure 21 shows the measured power duration demand curves for heating (heating + snow melting) and cooling (process cooling + space cooling) for 2013 (Orvik, 2014/15).

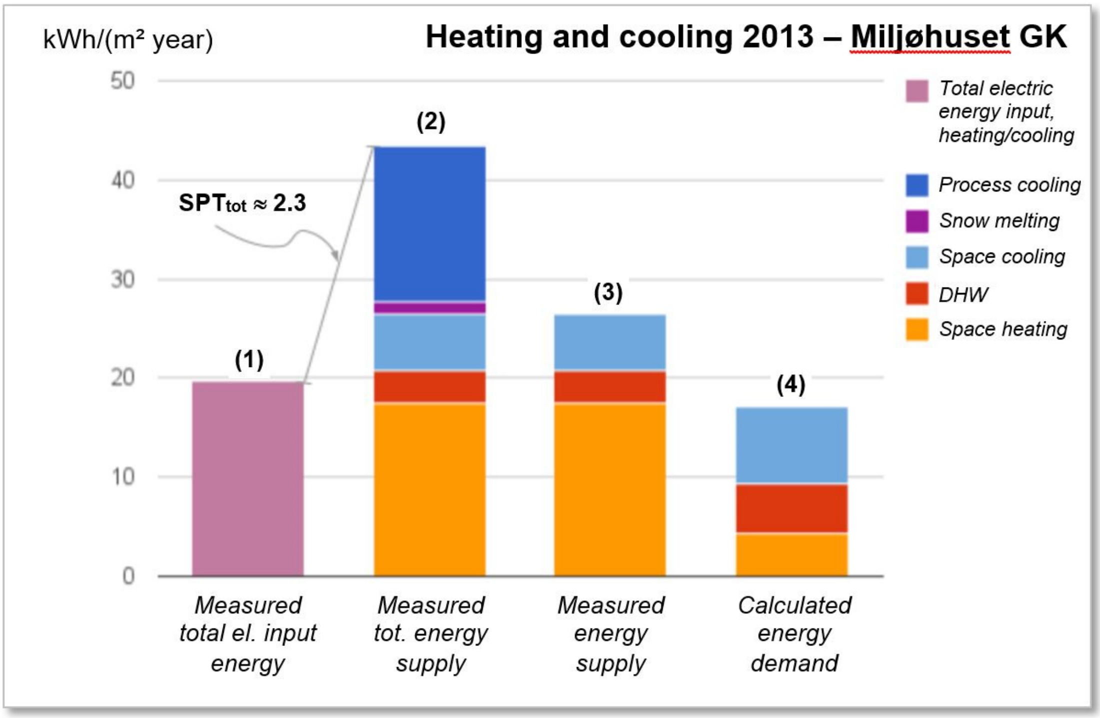


Figure 20 *The measured total electric input energy to the various sub-systems (1), measured thermal energy supply (2, 3), and calculated thermal energy demand, Simien (4). In (3) and (4) process cooling and snow melting are not included (Orvik, 2014/15).*

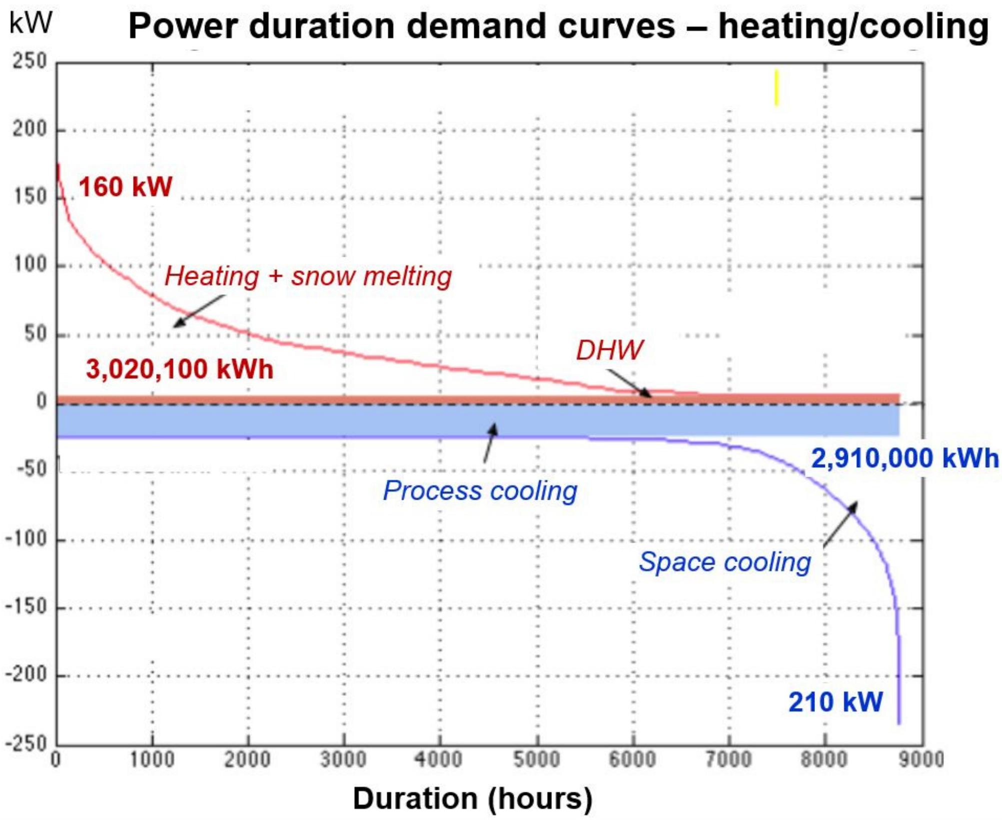


Figure 21 *The measured power duration demand curves for heating (heating + snow melting) and cooling (process cooling + space cooling) for 2013 (Orvik, 2014/15).*

Table 3 *Calculated and measured maximum power and annual energy demands for heating and cooling. The measured demands for space heating and heating of ventilation air have been corrected according to a "normal" (average) year (Orvik, 2014/15).*

	CALCULATED	MEASURED	DEVIATION COMMENT
Space heating and heating of ventilation air	4.2 kWh/(m²year)	**18 kWh/(m²year)**	+ 330 % – extremely large deviation
	22.7 W/m²	**12.5 W/m²**	- 45 %
DHW heating	5.0 kWh/(m²year)	**3.5 kWh/(m²year)**	- 30 %
Space cooling	7.8 kWh/(m²year)	**5.5 kWh/(m²year)**	- 30%
	26.4 W/m²	**15.4 W/m²**	- 60%
Heat recovery from process cooling (25 kW)	*	**35 %** of the total heating demand	Covers **35 %** of the total heating demand
SPF_{tot} – reversible heat pump and liquid chiller	*	**2.3**	Heating and cooling – incl. pump/fan energy

Table 3 shows the calculated and measured maximum power demands (kW) and annual specific energy demands (kWh/m²år) for Miljøbygget GK. The measured power and energy demands for space heating and heating of ventilation air have been subjected to a "normal year correction" since the simulations in Simien were based on average ambient air temperatures over a 30 year period (Orvik, 2014/15).

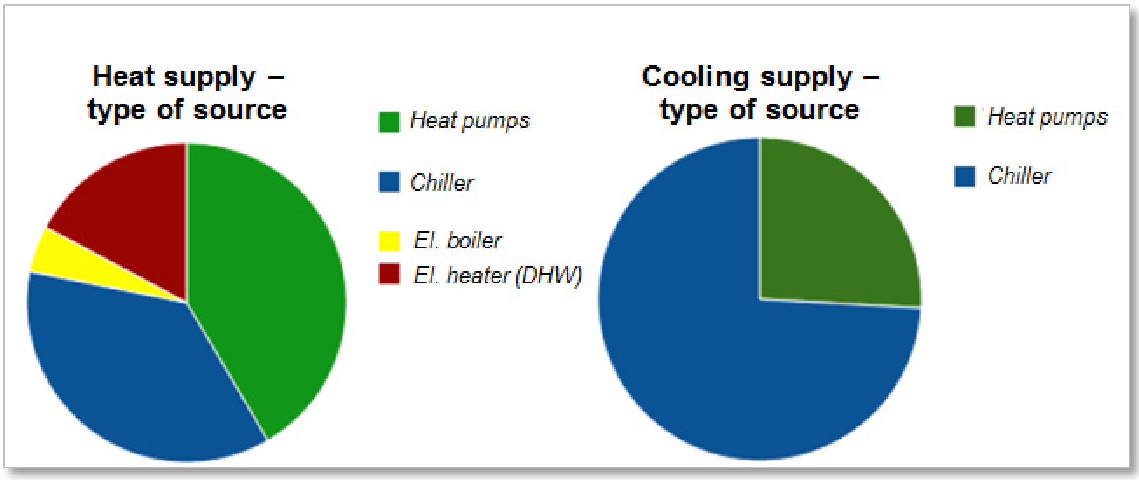


Figure 22 *The measured annual heat supply and cooling supply for 2013 categorized according to type of source (Orvik, 2014/15).*

2.3.2 Results and Discussion – Improvement of the Existing Systems

The results and discussion are based on Orvik (2014/15) – measurements for 2013/14:

The data processing system needed to be tuned/calibrated before all the heating and cooling demands and the input electric energy could be measured correctly.

2.3.2.1 Heating and cooling Demands

1. Space/ventilation air heating – The measured specific heating demand was about 4 times higher than the calculated value, i.e. *18 kWh/(m²year)* vs. 4.2 kWh/(m²year). The measured maximum heating demand for space heating and heating of ventilation air (*170 kW, 12 W/m²*) was almost *50 % lower* than the calculated value (310 kW).
   - The deviation was due to a larger heat loss from the building envelope (higher U-value), lower internal heat loads, lower efficiency for the ventilation heat recovery units, and longer operation time for the ventilation system than calculated.
   - More accurate calculations of the heating demand would have resulted in a considerably lower capacity for the combined heat pump/chiller system and with that lower investment costs and improved average COP – see *Section 2.3.2.2* .
2. DHW – The domestic hot water (DHW) demand accounted for about *15 %* of the total heating demand of the building, i.e. *3.5 kWh/(m²year)*
3. Space cooling – The measured specific space cooling demand was about 30 % lower than the calculated value, i.e. *5.5 kWh/(m²year)* vs. 7.8 kWh/(m²year). The deviation may be due to lower internal heat loads. The measured maximum space cooling demand (*210 kW, 15 W/m²*) was about *40 % lower* than the calculated value (360 kW).
   - More accurate calculations of the space cooling demand would have resulted in a considerably lower capacity for the combined heat pump/chiller system and with that lower investment costs and improved average COP – see *Section 2.3.2.2* .
4. Process cooling – The measured specific annual process cooling demand accounted for as much as *75 %* of the total annual cooling demand for the building, i.e. *16 kWh/(m²year)*. Consequently, the process cooling demand was almost identical to the specific heating demand, i.e. *18 kWh/(m²year)*.

2.3.2.2 Heating and Cooling Systems – Energy Supply and Energy Use

1. Heat pump/chiller – The combined air-to-brine heat pump and liquid chiller units were designed for a total heating capacity of *500 kW* at -7/35 °C and a total cooling capacity of *500 kW* at 10/35 °C. The measured maximum heating and cooling capacities in 2013 were 170 kW and 210 kW, respectively – i.e. considerably lower.

   The measured Seasonal Performance Factor (SCOP) for the air-to-brine heat pump and liquid chiller units including pumps and fans ranged from *2.2 to 2.4* (2013/2014).
   - *The heat pump/chiller units are considerably oversized* due to the inaccurate thermal power calculations. If they had been designed for a lower capacity, this would have resulted in a lower investment costs (better profitability) and lower annual energy use due to improved SPF.
   - The heat pump and liquid chiller units are *not optimized* for operation in the Nordic (cold) climate. A better design with e.g. large fin distance in the evaporator, a more advanced defrosting system, and variable speed drive piston or screw compressors would have contributed to a higher SPF.

2. Liquid chiller – The condenser heat from the liquid chiller for process cooling covered about *35 %* of the total heating demand in the building (space heating, DHW heating):

   - Heat recovery from process cooling equipment can be an important heat source and should always be taken into account when designing the heating and cooling systems for office buildings of passive house or NZEB standard.

   - The cooling capacity of the liquid chiller for process cooling was considerably lower than that of the heat pump/chillers for space cooling, i.e. *25 kW* vs. 500 kW, but covered a much larger cooling demand due to very long operating time.

3. Electro boiler – The electro boiler (peak load) covered about *5 %* of the total annual heating demand. The boiler was only used at low ambient air temperatures when the air-source heat pump units were switched off (t_{amb} < -12 °C to -15 °C).

4. Electric immersion heater – The measured heat supply from the electric immersion heaters in the DHW system was about *52 MWh/year*, which is actually higher than the annual DHW heating demand.

   - There was no net recovery of condenser heat from the liquid process chiller to the DHW system, and about 5 MWh/year heat was actually rejected from the DHW system to the ambient air via the dry cooler. I.e. the electric immersion heaters in the DHW system covered the entire heating demand. The heat recovery system has a serious malfunction and will be redesigned.

5. Auxiliary equipment – The measured total annual electricity consumption for pumps, dry cooler system with fans, and other auxiliary equipment accounted for about *25 %* of the total electricity use for the heating and cooling system.

   - The IE2 pumps should be replaced with IE3 pumps with variable speed drive

2.3.2.3 Other Aspects

1. Air-source vs. ground-source heat pumps and liquid chillers – An air-source heat pump and liquid chiller system were selected due to the low heating demand and low the temperature level in the heat distribution system, and a ground-source system was regarded as unprofitable. However, the measurements showed that the annual heating demand was about *240,000 kWh/year*, i.e. 4 times higher than the theoretical calculations. Ground-source heat pumps and chiller systems have much higher investment costs than air-source systems, but they have the following advantages.

   - The heating capacity of the heat pump is independent of the ambient air temperature, and the heat pump supplies heat even at low ambient air temperature which is especially important for passive house buildings in cold climates. The energy coverage factor will be *20-30 %-points* higher than that of air-source systems.

   - Due to a higher average source temperature, the SCOP (SPF) will be higher.

   - A large share of the annual cooling demand can be covered by free cooling (i.e. renewable cooling) from the borehole system.

   - The systems have much less operational problems, lower maintenance costs, and no noise from the heat source system.

   - At least 40 years lifetime for the ground-source system (boreholes, BHE etc.). Considerably longer lifetime for the heat pump and chiller units, typically 20-25 years vs. maximum 10 years for air-source systems in Nordic climate.

2. Indoor climate – Preliminary measurements carried out by SINTEF Building and Infrastructure have demonstrated that the users are *very satisfied with the indoor climate* (i.e. the air quality and the thermal comfort).

   - The project has demonstrated that the entire space heating demand in this passive house office building can be covered by heating of ventilation air. This excludes the traditional hydronic heat distribution system with radiators.
   - Due to the elevated air temperature during the heating period, it is essential that the ventilation system is able to maintain adequate ventilation efficiency at varying air flow rates (VAV system) and varying inlet air temperature.

3. Supply water temperature vs. COP – The max. supply water temp. in the heat distribution system is as low as *30-35 °C*, which is very favourable for an air-source heat pump system since excessive temp. lifts at low ambient air temperatures are avoided.

   - Moderate temperature lifts reduce overall wear and tear and increase the lifetime of the compressors and the air-cooled heat exchangers.
   - A measured SCOP (SPF) for heating and cooling of *2.3* is relatively low when taking into account the low distribution temperature. More optimised design and operation will improve the system performance (ref. point 4 and 5).
   - The supply temperature cannot be lower than 30 °C since the heat pump is also being used for snow melting. The snow melting system has been designed for a supply temperature of 30 °C.

4. Brine pumps – The twin pump installation between heat pump/chiller units and the accumulation tank (ref. *Figure 15*) is running at constant speed the entire year. This results in a higher annual energy demand for the pump and a reduction in the SCOP for the heat pump/chiller units compared to a system with variable volume flow rate.

   - The existing twin-pumps will be replaced, and separate variable speed drive pumps will be installed in each of the heat pump/chillers circuits. The volume flow rate will be controlled according to the heating/cooling capacity of each unit.

5. Accumulation tank – The accumulation tank between the heat pump/chiller units and the heating and cooling system (ref. *Figure 15*) does not work properly due to mixing of hot and cold brine flows inside the tank (temperature/exergy loss). This results in a lower average COP for the heat pump and chiller units:

   - The accumulation tank will be redesigned/ optimized. By using heat pump/ chillers with variable speed compressors the accumulation tank is no longer required.

2.4 Conclusions – Recommendations with regards to Design and Operation

Air-source heat pump and liquid chiller systems represent an interesting alternative in *office buildings of passive house or NZEB standard* due to moderate investment costs and acceptable operating characteristics. However, the latter requires:

- Calculations of thermal power demands for heating and cooling
    - Use sufficiently accurate computer models for power calculations (heating, cooling) in order to prevent oversizing of the heat pump and chiller units.
- Design and operation of air-source heat pump and liquid chillers in Nordic climate
    - Temperature level in hydronic distribution systems for heating and cooling
        - Low-temperature heat distribution systems with maximum 30 to 40 °C supply temperature maximizes the energy coverage factor and the COP in heating mode and reduces wear and tear for the compressors
        - High-temperature cooling systems with minimum 12 to 18 °C supply temperature maximizes the COP in cooling mode
    - Variable speed drive compressors, e.g. piston, screw or scroll
        - Adapts the heating or cooling capacity to the actual demands in the building – no need for accumulation tanks in the distribution system
    - Optimized design of the air-cooler heat exchanger (evaporator, condenser)
        - Large heat transfer surface leads to less frosting due to higher surface temperature and consequent less annual energy use for defrosting
        - Large fin distance (4-6 mm) leads to lower defrosting frequency for the coils and consequent less annual energy use for defrosting
    - Optimized defrosting system with high-quality demand control
        - Hot gas defrosting – standard reversed cycle
        - Defrosting with heat from a sub-cooler (novel system design)
    - Fans with ultra-low energy consumption
        - Advanced fans with energy efficient EC motors
- Surplus heat from liquid chiller for process cooling (servers, telecom equipment etc.)
    - Should be utilized as much as possible at a moderate temperature level
    - Utilize a desuperheater or an oil cooler (screw compressors) for DHW heating at a sufficiently high temperature level (60-70 °C)
- Auxiliary system
    - Use variable speed IE3 or IE4 pumps in order to minimize parasitic losses

3. Heat Pump System in a Renovated Office Building

3.1 Introduction

Powerhouse Kjørbo consists of two office buildings in Sandvika outside Oslo. The buildings from 1985 were refurbished to passive house standard (class A) in 2013-2014, and the total heated area is about 5,200 m². The buildings are planned with a net annual energy production during the year (plus energy building) due to the 200,000 kWh/year electricity generation from a large number of photovoltaic (PV) panels on the roof. Powerhouse Kjørbo has been certified as *"Outstanding"* according to the BREEAM-Nor classification system and is one of the pilot projects supported by the Norwegian Research Centre on Zero Emission Buildings (www.zeb.no). Important partners in the project have been the architectural company Snøhetta, the contractor Skanska, the aluminium company Hydro, the real estate firm Entra Eiendom, and the Norwegian environmental organization Zero. Powerhouse Kjørbo has been described as *"the most environmentally friendly office building in the world"*.

Figure 23 Powerhouse Kjørbo – a 5,200 m² office building in Sandvika refurbished and upgraded to passive house standard. The building was BREEAM-Nor certified as "Outstanding".

In addition to a high-quality air-tight building envelope with low average U-value, VAV ventilation system with low SFP factor and high-efficiency heat recovery, utilization of daylight, demand controlled lighting systems (LED), and efficient solar shading, the buildings are equipped with an *energy efficient heating and cooling system* with heat pumps.

3.2 Heating and Cooling Demands

Table 4 shows the calculated power and annual energy demands for space heating, heating of ventilation air, domestic hot water (DHW) heating, space cooling, and computer cooling.

Table 4 Calculated heating and cooling demands at Powerhouse Kjørbo (Nordang, 2014/15).

Demand	Power demand	Specific power demand	Annual energy demand	Specific annual energy demand
Space/vent. air heating	52.0 kW	10.0 W/m²	98,800 kWh/y	19.1 kWh/(m²y)
DHW heating	2.8 kW	0.55 W/m²	24,800 kWh/y	4.8 (kWh/(m²y)
Total – heating	**54.8 kW**	**10.6 W/m²**	**123,600 kWh/y**	**23.8 kWh/(m²y)**
Space cooling	54.0 kW	10.4 W/m²	9,500 kWh/y	1.8 kWh/(m²y)
Computer cooling	10.0 kW	1.9 W/m²	88,000 kWh/y	16.9 kWh/(m²y)
Total – cooling	**64 kW**	**12.3 W/m²**	**97,500 kWh/y**	**18.7 kWh/(m²y)**

3.3 The Thermal Energy System

3.3.1 Equipment and System Design

The heating and cooling system at Powerhouse Kjørbo comprises:

- *Ground heat source/sink system – 10 boreholes* in bedrock, each 225 m deep, borehole heat exchangers, BHE (single U-tubes, PE100, turbulence collectors) and PE supply/return pipelines charged with 25 % ethanol (brine). The estimated free cooling from the boreholes was 40-50 kW at 12/17 °C supply/return brine temperature.
- *A standard brine-to-water heat pump and liquid chiller unit* (SH-HP) for space heating, heating of ventilation air, and space cooling. Two 900 litre accumulation tanks are placed between the heat pump unit and the heat distribution system.
- *A standard brine-to-water heat pump* for domestic hot water heating (DHW-HP). The heat pump is connected to two 600 litre storage tanks in the DHW system.
- *A district heating heat exchanger unit* for peak load and back-up. The heat exchanger is connected in series after the condenser for the SH-HP heat pump unit
- A computer and pure water cooling system
- *A centralized heat distribution system* – hydronic closed-loop pipeline system with 2 x 900 litre accumulation tanks connected to radiators and heater batteries in the air-handling/ventilation units – design temperatures *50/40 °C* and *50/25 °C*, resp. A separate heat exchanger is installed between the heat distribution system and the ground-source system for rejection of excess heat when the pump is operating in the cooling mode
- *A centralized cooling distribution system* – centralized hydronic closed-loop pipeline system with chilled water connected to cooler batteries in the air-handling (ventilation units)

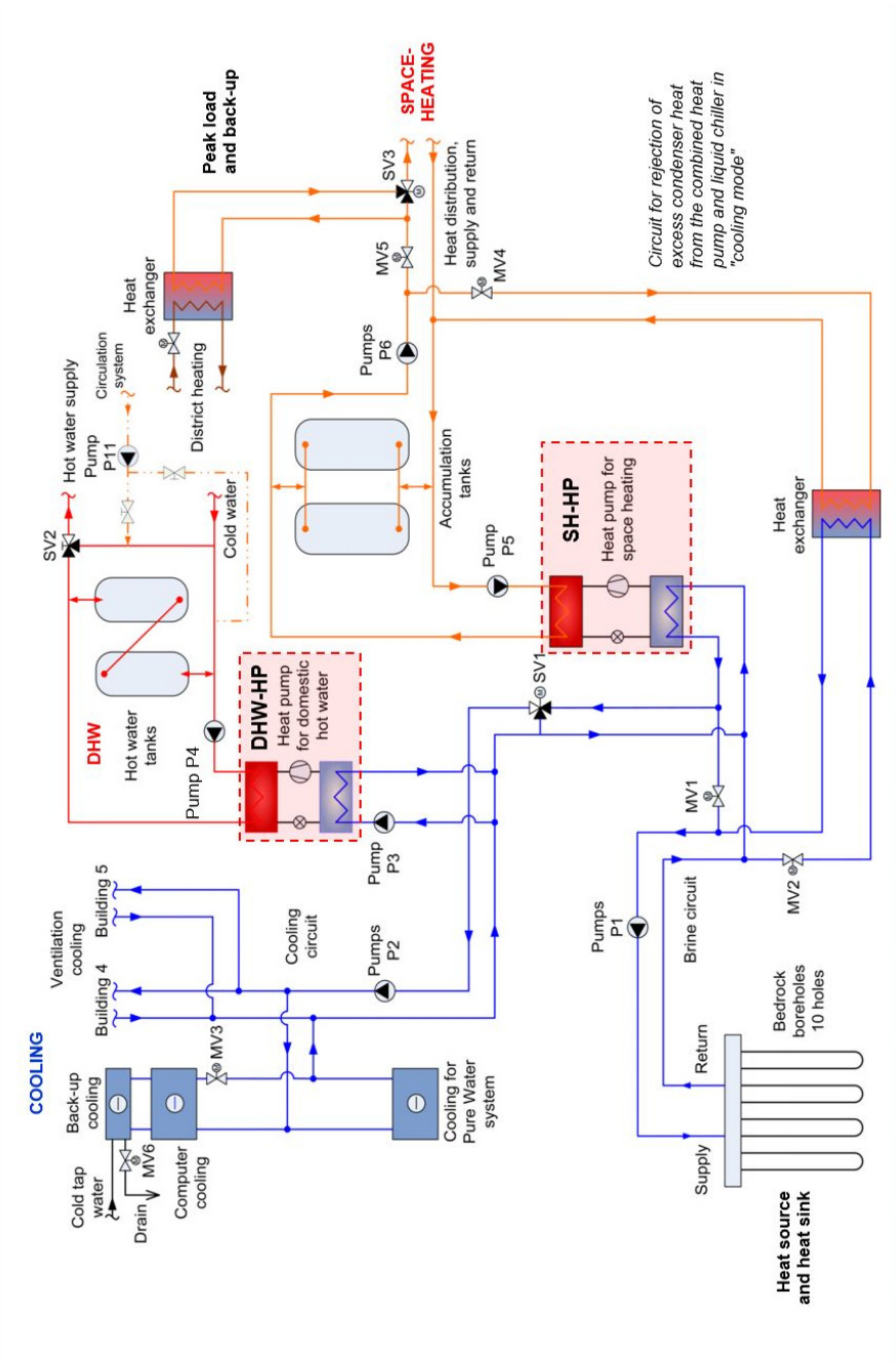


Figure 24 Simplified sketch of the thermal energy system – heat pump and liquid chiller, DHW heat pump and district heating heat exchanger – for space heating, heating of ventilation air, domestic hot water (DHW) heating, space cooling, and process cooling at Powerhouse Kjørbo (Nordang, 2014/15).

3.3.2 Heat Pump and Liquid Chiller Units

3.3.2.1 Heat Pump and Liquid Chiller

The heat pump unit for space heating, heating of ventilation air and back-up space cooling (SH-HP) was designed to cover the gross power demand for heating since this gave additional points in the BREEAM-NOR evaluation due to zero NO_X emissions from the heating system. District heating is only used as back-up. Specifications:

- Type: Standard brine-to-water heat pump/chiller unit
- Heating capacity: **65 kW** at 0/45 °C
- Working fluid: R410A
- Compressors: 2 scroll compressors, intermittent (on/off) operation
Max. 3 start/stops per hour
- Expansion valve: Electronic type
- Max. outlet water temp.: 65 °C
- COP: 4.2 at 0/35 °C – data from manufacturer
3.4 at 0/45 °C – data from manufacturer

Figure 25 The heat pump and liquid chiller unit at Powerhouse Kjørbo for space heating, heating of ventilation air, and back-up space cooling (Nordang, 2014/15).

The supply water temperature in the heat distribution system is controlled according to an *ambient air compensation curve* (control curve). This means that the supply temperature from the heat pump is reduced when the ambient temperature (i.e. the space heating demand) increases and vice versa. This maximizes the COP for the heat pump.

3.3.2.2 Heat Pump Water Heater

The heat pump water heater is a standard R407C brine-to-water heat pump unit. The DHW is heated to only *50-60 °C* due to the application of a chlorine dioxide (ClO_2) disinfection system at the city water inlet. Due to the relatively low annual DHW demand, a CO_2 heat pump water heater was considered to be too expensive. However, CO_2 heat pumps have become more cost-effective and are now competitive to standard HFC heat pumps due to much higher maximum supply temperature (60-90 °C), higher COP and longer projected lifetime.

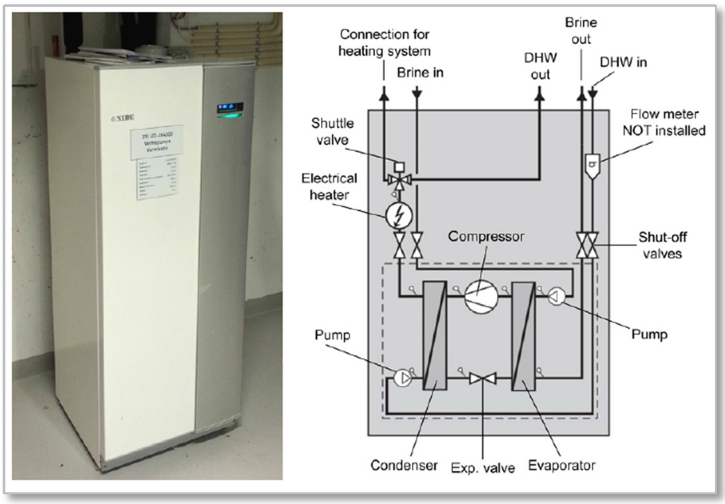


Figure 26 The heat pump water heater at Powerhouse Kjørbo (Nordang, 2014/15).

Specifications for the heat pump water heater:

- Type: Standard R407C brine-to-water heat pump unit
- Heating capacity: **8.5 kW** at 0/45 °C – residential unit
- Working fluid: R407C
- Compressor: 1 piston compressor, intermittent (on/off) operation
Max. 3 start/stops per hour
- Expansion valve: Thermostatic type
- Max. outlet water temp.: 65 °C
- COP: 4.8 at 0/35 °C – data from manufacturer
3.8 at 0/45 °C – data from manufacturer

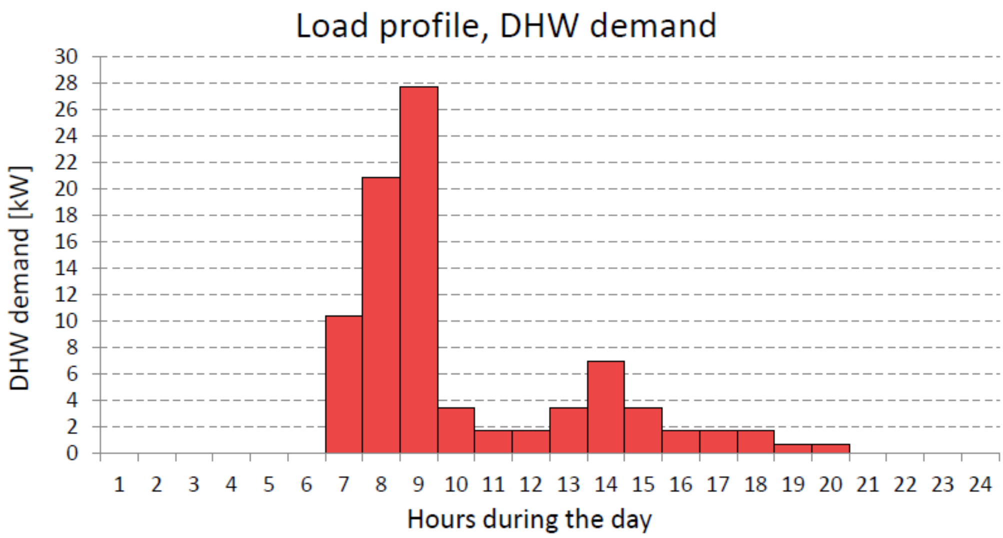


Figure 27 Estimated load profile for the DHW demand (Nordang, 2014/15).

3.3.3 Operating Modes

The heat pump and liquid chiller system is operated in "Heating Mode" or "Cooling Mode":
Heating mode – Ref. *Figure 28* – The space and DHW heating demands are the dominating thermal loads. There is no space cooling demand but a small process cooling demand.

- Motor valves MV2 and MV4 are closed (no rejection of excess heat to the boreholes)
- The variable speed pump P1 in the ground-source system is controlled to maintain a 4-5 °C temperature difference between the supply and return brine flows of the boreholes
- The heating capacity of the space heating heat pump (SH-HP) is controlled to meet the set-point temperature in the heat distribution system (max. *50 °C*). The capacity is sufficient to cover the entire space heating demand in the building, and district heating is only used as back-up (valve MV10 opens and SV3 controls the supply temperature).
- When there is a DHW demand, brine pump P3, water pump P4 and the heat pump water heater (DHW-HP) are turned on. DHW-HP has only one compressor (on/off operation)
- Shunt valve SV1 in the brine circuit is used to control the inlet brine temperature to the process cooling system (approx. 12 °C, constant value)

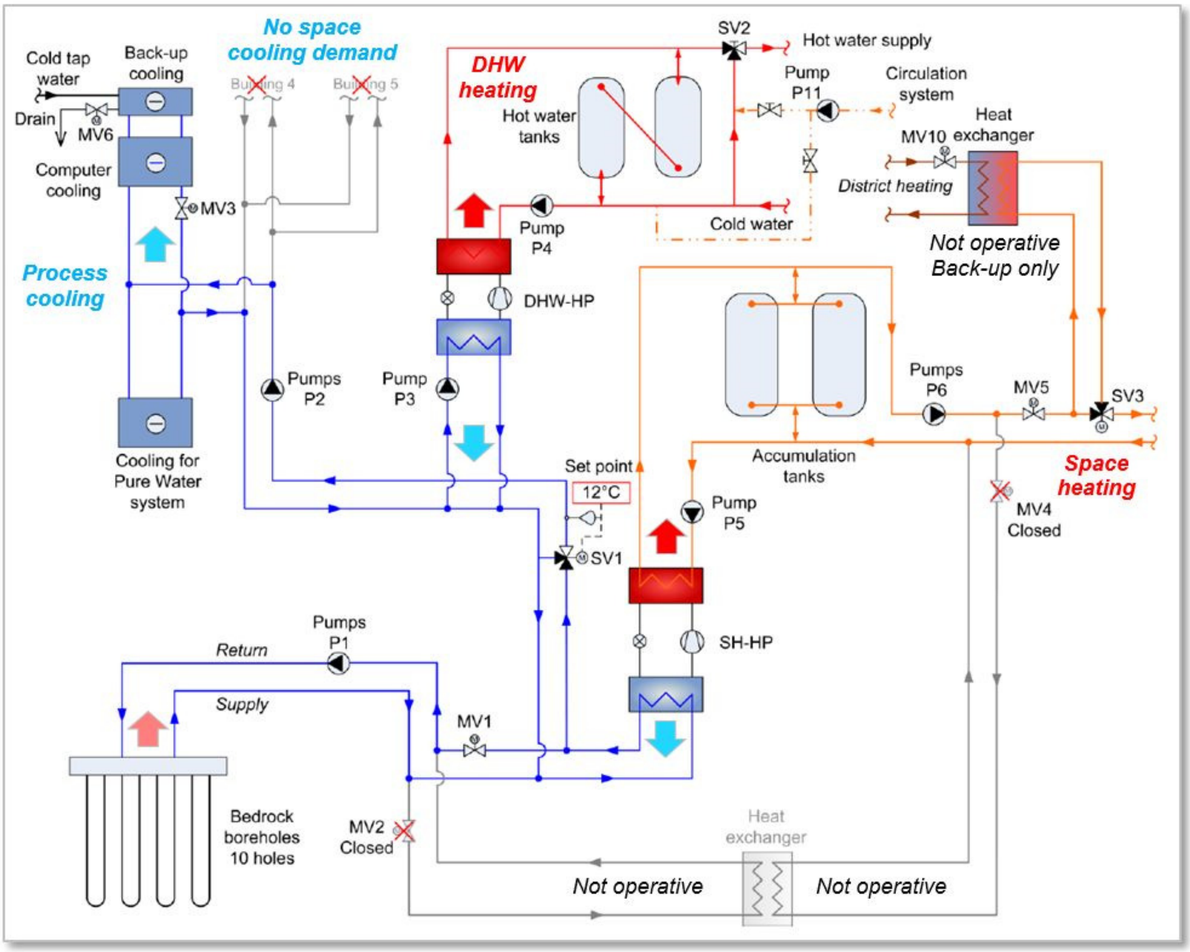


Figure 28 The thermal energy system operating in "Heating Mode – heat pump mode". Both space heating and DHW heating – process cooling but no space cooling (Nordang, 2014/15).

Cooling mode – The demand for process cooling and space cooling are the dominating thermal loads. There is no space heating demand but a DHW heating demand.

- In "Cooling Mode – free cooling only" (*Figure 29*) where the cold brine from the borehole system covers the entire space cooling and process cooling demands (free cooling). SH-HP is switched off while the DHW-HP covers the DHW heating demand.
- In "Cooling mode – liquid chiller operation" (*Figure 30*) the brine temperature from the boreholes is not sufficiently low to cover the cooling demands.
    - Heat pump SH-HP is switched to "liquid chiller operation" (24 hour time delay)
    - Motor valves MV1 and MV5 are closed, motor valves MV2 and MV4 are opened, and pumps P5 and P6 are running. The excess condenser heat from the liquid chiller is rejected via the plate heat exchanger to the borehole system.
    - In order to avoid frequent start/stop of the compressor, the cooling capacity is controlled according to the return temperature in the cooling system (max. 17 °C)

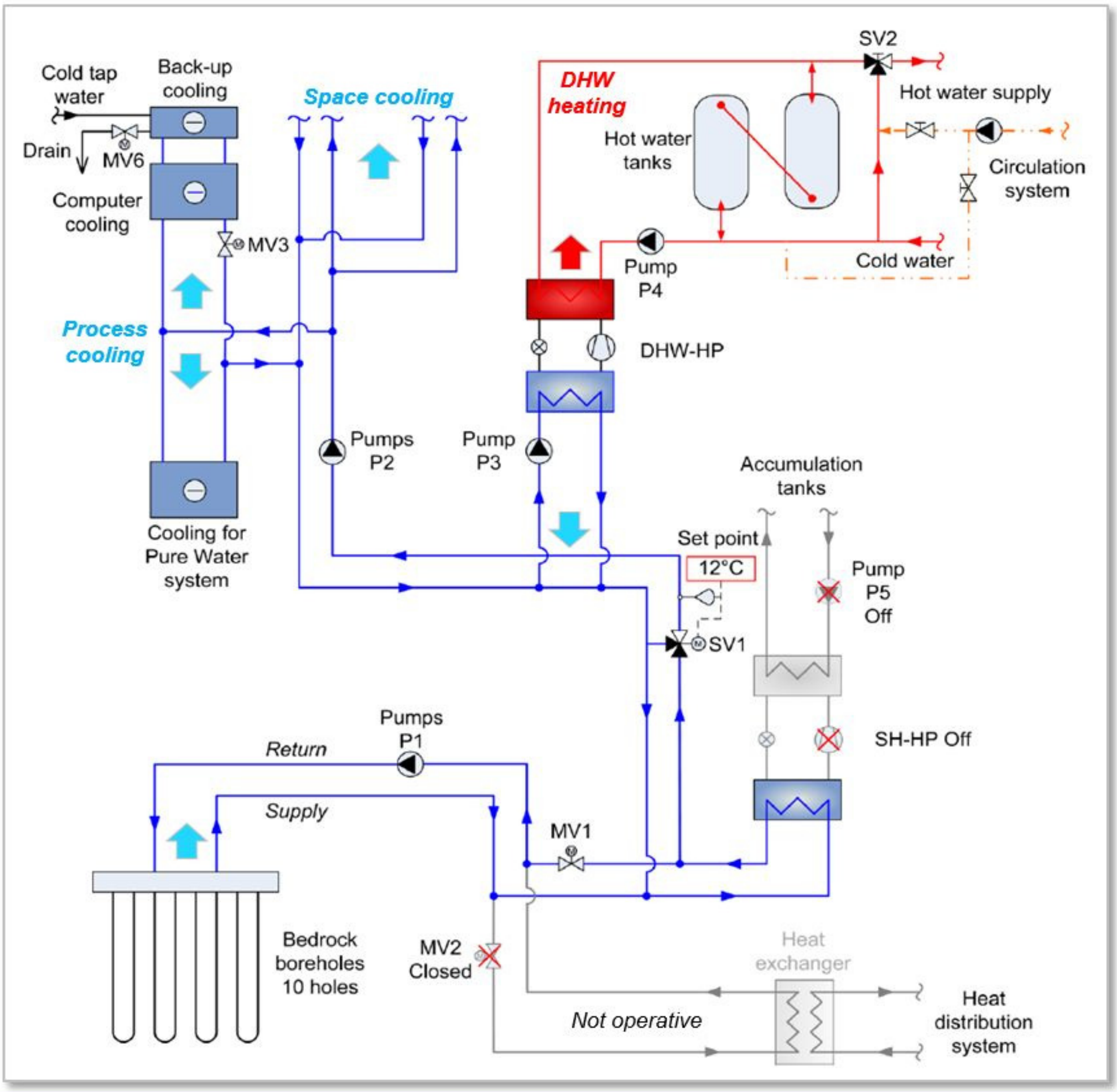


Figure 29 The thermal energy system operating in "Cooling Mode – free cooling only". DHW heating but no space heating – both process cooling and space cooling (Nordang, 2014/15).

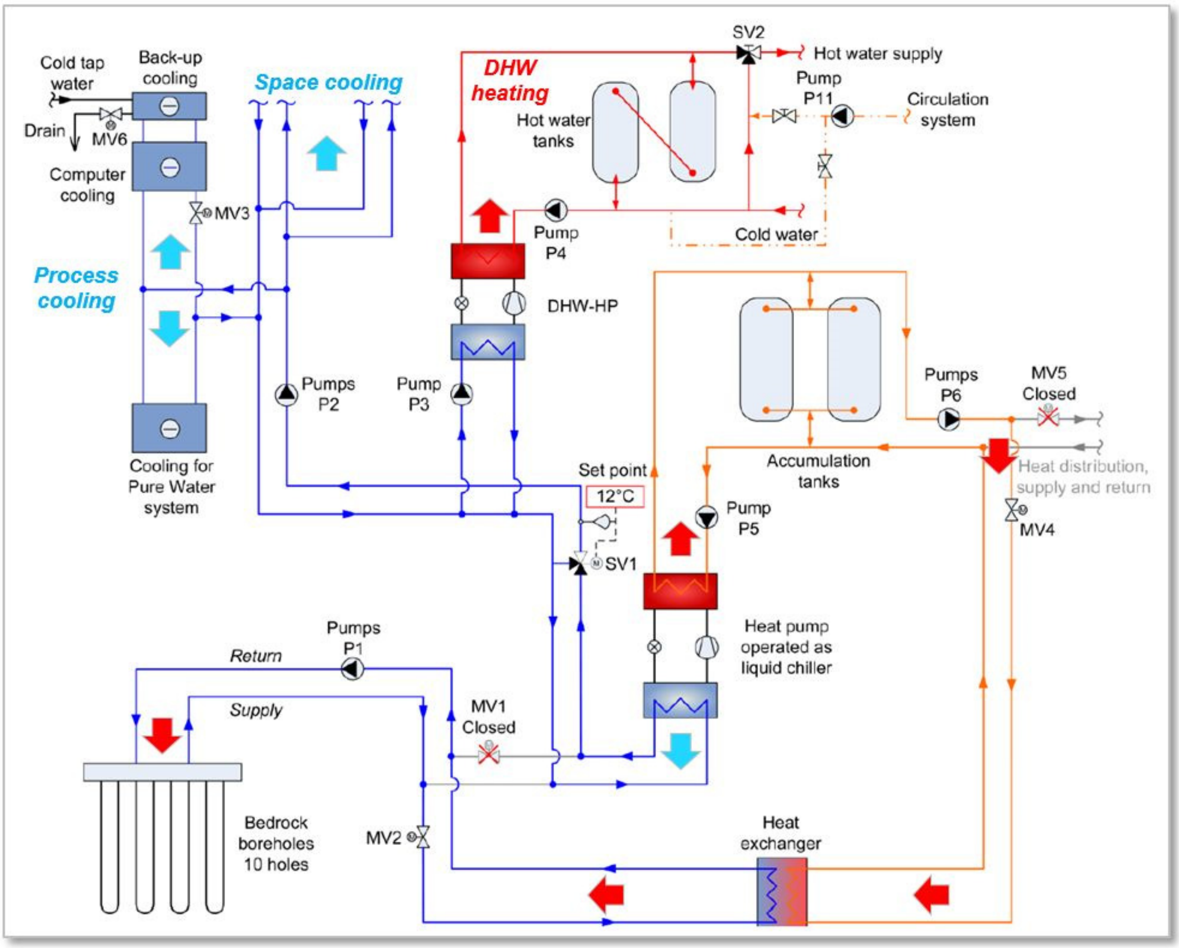


Figure 30 The thermal energy system operating in "Cooling Mode – liquid chiller operation". DHW heating but no space heating – process cooling and space cooling (Nordang, 2014/15).

3.4 Measurements and Analysis

A student at The Norwegian University of Science and Technology (NTNU) in Trondheim has, through his Project and Master work, monitored and carried out an in-depth analysis of the heating and cooling systems (Nordang, 2014/15).

Figure 31 on the following page provides an overview of the temperature sensors, pressure sensors, electric power/energy meters, and thermal power/energy meters which are installed in the borehole (brine) circuit, the DHW system, the space heating system incl. the district heating circuit (back-up), and the circuits for space cooling (cooling of ventilation air) and process cooling. All the sensors are linked to a centralized control and monitoring system.

Figure 32 shows, as an example, a screen-shot from the extensive monitoring system.

3.4.1 Heating and Cooling Demands

Figure 33 shows the calculated and measured power demand duration curves for heating (space heating and heating of ventilation air) and cooling (process cooling and cooling of ventilation air). The heating demand for DHW is not included (Nordang, 2014/15).

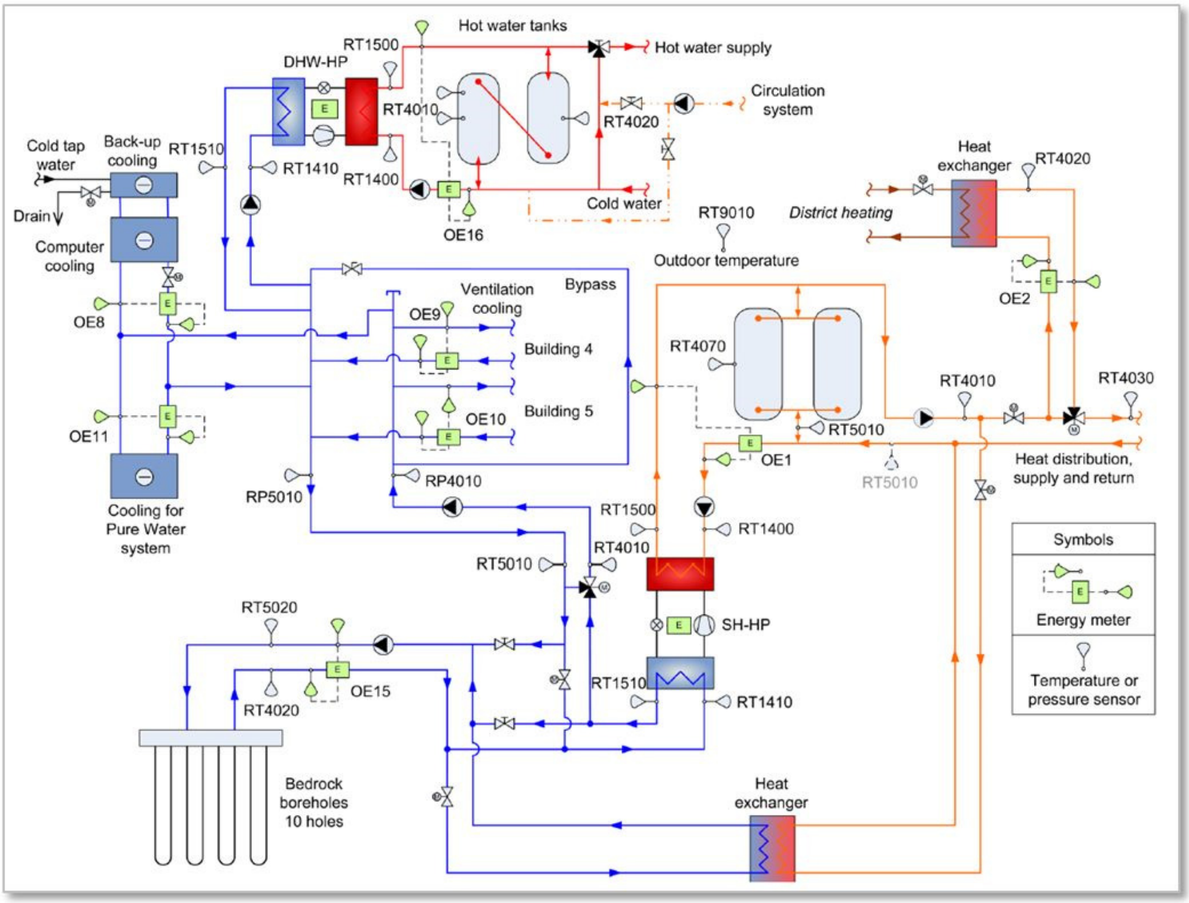


Figure 31 Instrumentation for the thermal energy plant with temperature sensors, pressure sensors, electrical power/energy meters, and thermal power/energy meters (Nordang, 2014/15).

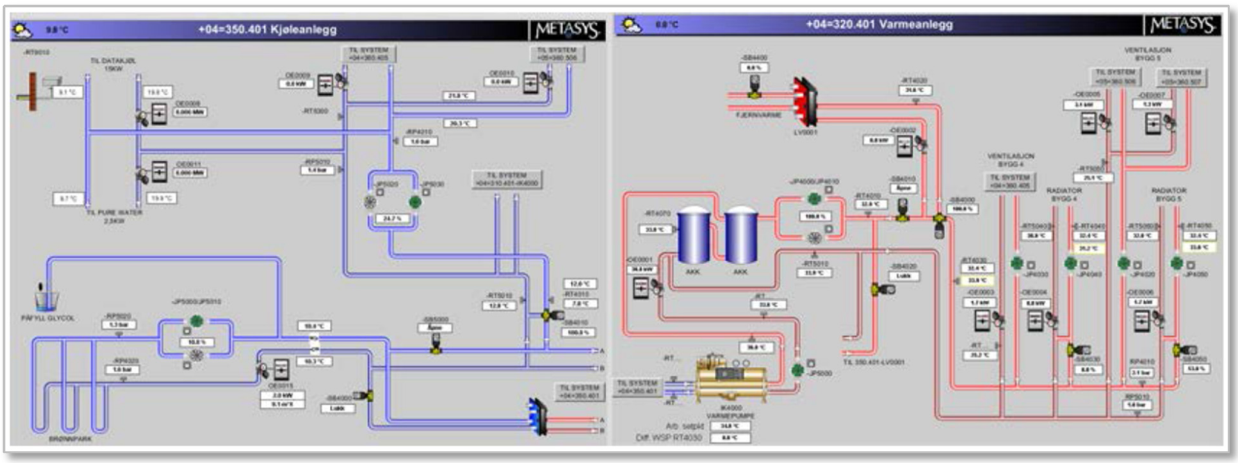


Figure 32 Example of screen shot from the monitoring system (Nordang, 2014).

Table 5 shows the calculated and measured maximum power demands (kW) and annual specific energy demands (kWh/m²år) for Powerhouse Kjørbo (Nordang, 2014/15). The measured power and energy demands for space heating and heating of ventilation air have been subjected to a "normal year correction" since the simulations in Simien were based on average ambient air temperatures over a 30 year period (Nordang, 2014/15).

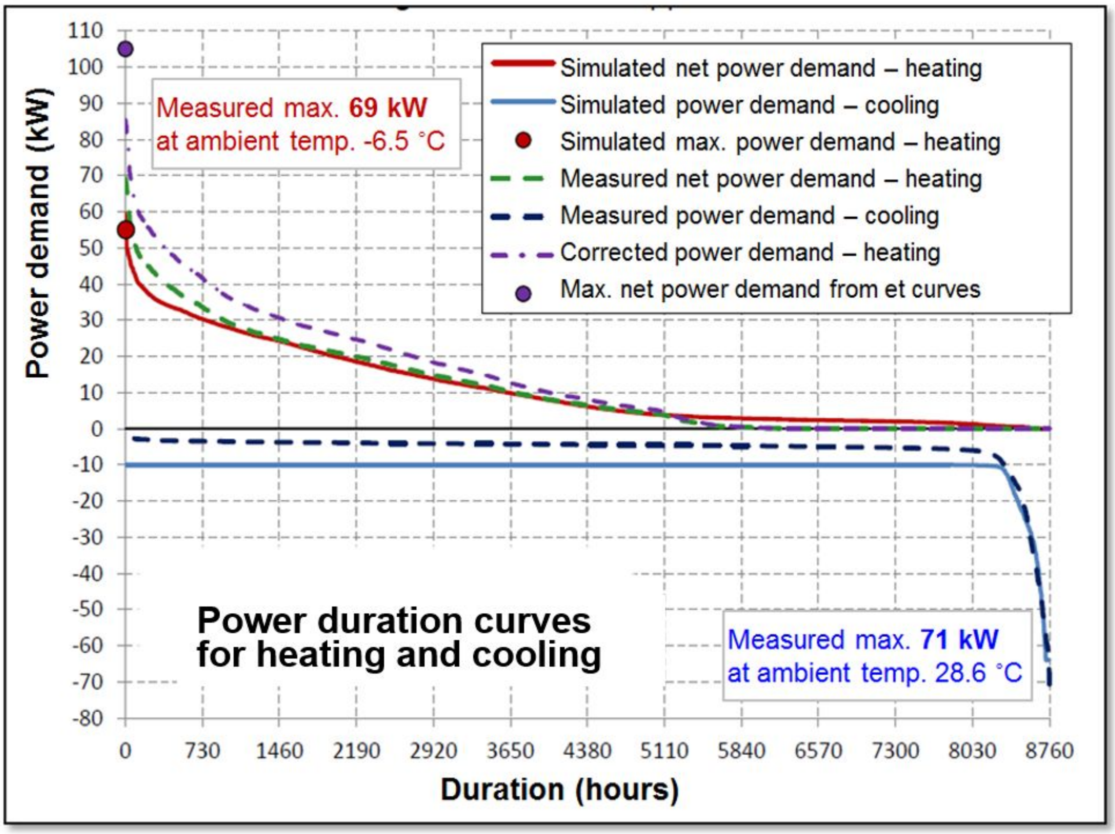


Figure 33 Calculated and measured power demand duration curves for heating (space heating + heating of ventilation air) and cooling (process cooling + cooling of ventilation air). The DHW heating demand is not included. The curve "Corrected power demand – heating" has been corrected according to a normal (average) year (Nordang, 2014/15).

Table 5 Calculated and measured maximum power and annual energy demands for heating and cooling. The measured demands for space heating and heating of ventilation air have been corrected according to a normal (average) year (Nordang, 2014/15).

	CALCULATED	MEASURED	DEVIATION COMMENT
Space heating and heating of ventilation air	19.1 kWh/(m²year)	**24.6 kWh/(m²year)**	+ 29 %
	10.0 W/m²	**14.4 W/m²**	+ 37 %
DHW heating	4.8 kWh/(m²year)	**1.9 kWh/(m²year)**	- 60 %
Space cooling	1.8 kWh/(m²year)	**2.0 kWh/(m²year)**	+ 11 %
	10.4 W/m²	**13.5 W/m²**	+ 29%
SPF – heat pump for space and vent. air heating	*	**3.9**	Incl. pump energy for the borehole system
SPF – heat pump water heater (DHW heating)	*	**2.9**	Incl. pump energy

- Space heating and heating of ventilation air
    - *Heating, maximum power demand* – The measured max. power demand for space heating and heating of ventilation air was *37 % higher* than the calculated value when the measured data were converted to a "normal" (mean) year. The deviation was mainly due to a lower thermal efficiency for the heat recovery heat exchangers in the ventilation systems.
    - *Heating, annual energy demand* – The measured annual energy demand for space heating and heating of ventilation air was *29% higher* than the calculated value when the measured data were converted to a normal (mean) year. The deviation was mainly due to a lower thermal efficiency for the heat recovery heat exchangers in the ventilation systems.
        - The entire annual heating demand was covered by the SH heat pump unit
- DHW heating
    - The annual DHW heating demand was overestimated by approx. *60 %*. I.e. the DHW user pattern for this particular office building departed considerably from the standard (normative) value in the Norwegian building code NS3031.
- Space cooling (cooling of ventilation air)
    - *Cooling, maximum power demand* – The measured maximum power demand for space cooling was *29 %* higher than the calculated value. The deviation is mainly owing to the extremely warm summer in 2014.
    - *Cooling, annual energy demand* – The measured annual energy demand for space cooling was *11 %* higher than the calculated value. The deviation is mainly owing to the extremely warm summer in 2014.
        - The entire cooling demand was covered by the ground source system, i.e. free cooling from the 10 boreholes (total borehole length approx. 2,250 m)

The calculated annual energy demands for heating and cooling are applied for analysis and pre-selection of the overall concepts for thermal energy supply of the building, e.g. an air-source vs. a ground-source heat pump system. On the other hand, the calculated maximum power demands for heating and cooling are used when dimensioning the heating and cooling units and other main components in the heating and cooling system.

"Miljøbygget GK" (*Section 2*) was simulated in the Norwegian building simulation programme *Simien* as a single-zone building, i.e. a simplified calculation. It turned out that the annual energy demand for space heating and heating of ventilation air was greatly underestimated, while the calculated maximum power demands for heating and cooling were about *50 % higher* than the measured values. The latter led to an oversized and expensive heat pump and liquid chiller system with relatively poor part load efficiency.

"Powerhouse Kjørbo" was also simulated in Simien, but as a multi-zone building with more detailed input and analysis. Despite this fact, a relatively large deviation between the calculated and measured power and energy demands for heating and cooling was observed.

3.4.2 The Ground-Source Heat Pump System

Two ground-source heat pump units are installed at Powerhouse Kjørbo:

- A standard 64 kW (0/45 °C) brine-to-water heat pump and liquid chiller unit for space heating and heating of ventilation air – back up for space cooling (not used so far)
- A standard 8.6 kW (0/45 °C) brine-to-water heat pump for DHW heating

3.4.2.1 The Borehole System

The heat pump units are connected to a common ground-source system comprising 10 boreholes, each 225 m deep. The borehole system was designed to cover the entire space and process cooling demand in the building (65 kW) by free cooling at 12/17 °C supply/return temperature in the distribution system. I.e. the outlet brine temperature from the boreholes cannot exceed the required set-point temperature in the cooling system. In standard ground-source heat pump systems the heat pump is utilized as a liquid chiller that covers the peak load space cooling demand in the building, and the excess condenser heat is rejected to the boreholes at a temperature level between 25-30 °C. The conventional system design requires fewer boreholes than a system based entirely on free cooling, but the annual energy consumption will be slightly higher due to occasional chiller operation during the summer.

The ground-source simulation programme Earth Energy Designer (EED) was used to calculate the average brine temperatures and thermal energy balance for the borehole system during several years of operation. *Figure 34* shows the simulated and measured mean brine temperatures at max. power (capacity) and part load operation in heating and cooling mode.

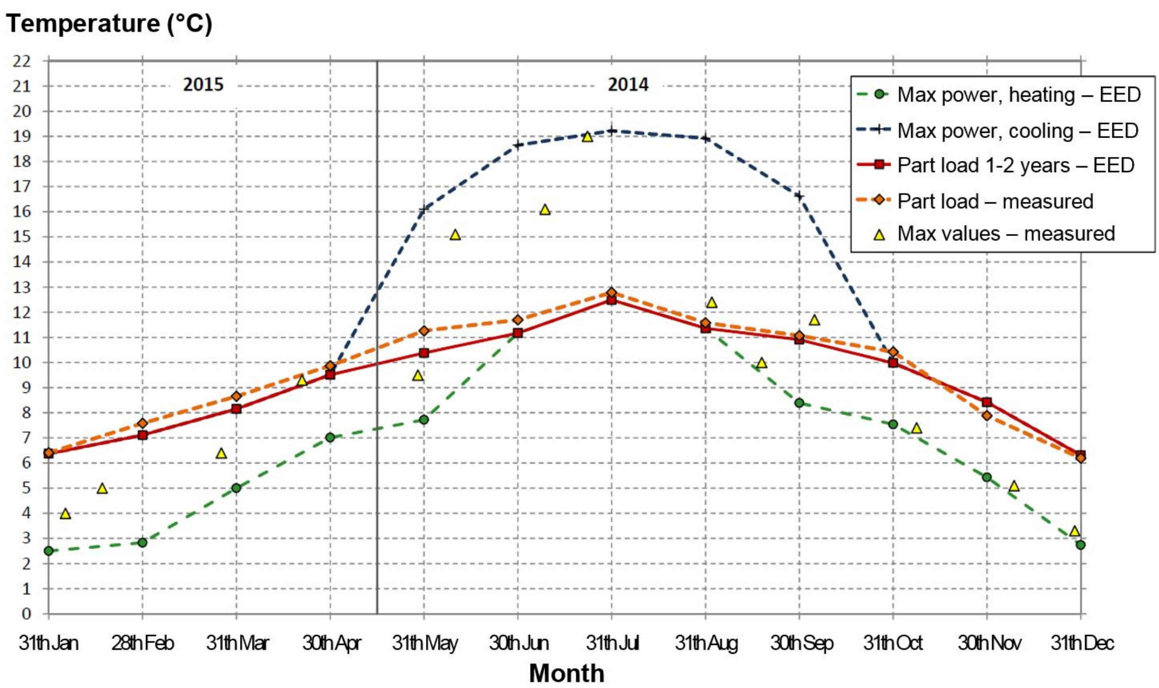


Figure 34 Simulated (EED) and measured average brine temperatures for the ground-source (bore-hole) system at max. power and part load in heating and cooling mode (Nordang, 2015).

- The measured values corresponds rather well with the simulated values, which demonstrates that EED is a suitable tool for borehole simulations as long as the input parameters have sufficient accuracy

- The measured average brine temperature during the heating season (space heating and heating of ventilation air) ranges from about 3 to 10 °C. The relatively high temperature level provided excellent operating conditions for the heat pump units.
- The measured minimum mean temperature during heating mode was as high as 3 °C. The reason for the relatively high minimum temperature in heating mode was that the borehole system was designed to cover the peak load cooling demand with free cooling, since this gives an extra point in the BREEAM evaluation.
- The measured maximum average brine temperature in cooling mode was approx. 19 °C, which obviously was sufficient to cover the maximum cooling demand since the heat pump unit never has been operated as a chiller. This indicates that the cooling system during high load periods has been operating at relatively high distribution temperatures.
- If standard design rules for the boreholes system had been applied, the number of boreholes could have been be reduced from 10 to 5-6, thus reducing the investment costs by approx. 40-50 %

3.4.2.2 The Heat Pump Units

- <u>Space Heating and Heating of Ventilation Air</u>
    - The measured Seasonal Performance (SPF, SCOP) for the heat pump unit was about **3.9** **(**including the annual input energy to the pumps in the borehole system), and the energy coverage factor was approx. 100 %. Consequently, *the net energy saving for the SH heat pump unit was 75 %*.
    - The most important factors leading to the high SPF was the application of a low-temperature heat distribution system (50/40 °C) and the oversized ground-source system with a relatively high average brine temperature
- <u>DHW heating</u>
    - The measured SPF for the DHW heat pump unit was about **2.9** including the annual energy input to the pumps. Since the energy coverage factor was approx. 100 % this corresponds to *a net energy saving of 65 %*.
    - The relatively high COP was caused by fluctuations in the outlet temperature from the DHW heat pump between 25 and 60 °C with an average of 45 °C. The standard legionella-safe DHW storage temperature in Norway is 65 to 70 °C. The DHW temperature at Powerhouse Kjørbo is lower since the DHW system has been equipped with a chlorine dioxide cleaning system.

3.4.3 Improvement of the Existing Heating and Cooling Systems

Based on the operational experience and measurements from the prevailing heating and cooling system at Powerhouse Kjørbo, some recommendations regarding system redesign and alternative operational strategies have been suggested:

- <u>Combined heat pump and liquid chiller</u> – The current ground source system was designed to cover the entire cooling demand by free cooling from the boreholes in order to gain extra points in the BREEAM evaluation. However, this is not a cost-efficient design. The heat pump system should have been designed as a conventional ground-source system. In such a design, the SH heat pump unit is utilized as a liquid chiller at peak load cooling demands, and the excess condenser heat is rejected to the boreholes at e.g. 35/30 °C. This design would have halved the required number of boreholes, i.e. a reduction from 10 to 5 boreholes (65,000 € reduced costs). The recommended mutual distance between the holes is 15-20 m.

- Natural working fluids – Both heat pump units are using HFC working fluids (R410A, R407C), which are controlled by the F-gas Directive since they are strong greenhouse gases. In order to improve the eco-friendly profile of this unique refurbished building, the 65 kW SH could have used propane (R290) as working fluid while the 8.6 kW DHW heat pump could have used CO_2 (R744).
- CO_2 heat pump water heater – The measurements revealed that the DHW system does not work properly since the DHW temperature fluctuates between 20 and 60 °C, depending on the momentary DHW consumption at the tapping sites. During DHW heating and thermal charging of the hot water tanks, the water must be recirculated through the heat pump condenser several times since the heat pump is only able to heat the water about 10 °C. The operational problems would have been completely solved by installing a CO_2 heat pump water system as described in *Section 1.4*.
- Variable speed drive compressors – The 65 kW SH heat pump is equipped with 2 scroll-compressors with intermittent control (on/off control), which means that the lowest heating capacity is approx. 30 kW. The measurements revealed that the heat pump unit did not operate satisfactorily due to undesirable temperature fluctuations in the hydronic systems at low heat loads since one compressor is frequently switched on/off. This problem would have been completely resolved by installing a SH heat pump unit with variable speed drive (VSD) compressors.
- Recovery of process heat – In the current configuration the excess heat from computer cooling is transferred directly to the boreholes and not utilized by the SH and DHW heat pumps. The computer cooling system should have been connected in series before the heat pump evaporators in order to maximize the heat source temp.

3.5 Conclusions – Recommendations regarding Design and Operation

Ground-source heat pump and liquid chiller systems represent an interesting alternative in *office buildings of passive house or NZEB standard* due to excellent performance and large energy savings, large flexibility with regard to heating and cooling, as well as long lifetime. In order to attain successful installations, the following parameters are of great importance.

- Calculations – design of the heat pump/chiller system and profitability analysis
  - Use sufficiently accurate computer models in order to obtain correct sizing of the heat pump and chiller units as well as the ground-source system (boreholes)
  - Carry out a profitability/sensitivity analysis for the entire thermal system and the different sub-systems including the heat pumps and the heat source system. Due to the relatively low heating and cooling demands in passive house buildings and NZEBs, it is crucial to keep the costs for the heating and cooling systems including the heat pump/chiller at a moderate level.
- Ground-source concept – Free cooling from the borehole system should be utilized as much as possible. At high space cooling loads the heat pump units should be utilized as liquid chillers, where the capacity is controlled according to the cooling demand in the building. The heat pump should be able to cover the max. cooling load. During liquid chiller operation the excess condenser heat is rejected to the boreholes.

- Peak load space cooling covered by the heat pump is less energy efficient than a system with free cooling only, but the number of boreholes and thereby the costs for the ground-source system can be considerably reduced
    - The borehole system should use high-efficiency IE3 or IE4 pumps to minimize energy consumption (parasitic losses)

- Natural working fluids – Heat pumps and liquid chillers should preferably use so-called natural working fluids, i.e. propane (R290), CO_2 (R744) and ammonia (R717)
    - Natural working fluids are, in contrary to the HFCs and HFOs, 100 % eco-friendly fluids since they have zero or negligible GWP-value and do not have any negative impact on the global environment during manufacturing of the fluids or in case of unintentional leakages from the units

- Efficient compressors – The heat pump and liquid chillers should be equipped with energy efficient compressors with good part load qualities, preferably piston or screw compressors with variable speed drive or piston compressors with cylinder unloading
    - Variable speed drive compressors achieve very high part load efficiency and adapts the heating and cooling capacity to the actual demands in the building, i.e. there is no need for accumulation tanks in the distribution systems

- Optimum distribution systems – The building should have low-temperature hydronic distribution systems for heating (max. 45 to 50 °C) and high-temperature distribution systems for cooling (min. 12 to 16 °C)
    - Optimum temperature levels in the distribution systems for heating and cooling maximizes the Coefficient of Performance (COP) for the heat pump and liquid chiller units in heating and cooling mode

- Utilization of surplus heat – Surplus heat from separate liquid chillers for process cooling (servers, telecom equipment etc.) should be utilized as much as possible to cover heating demands in the building
    - DHW heating at a legionalla-safe temperature (65-70 °C) should be covered by a desuperheater or an oil cooler (screw compressors) since DHW can be heated to a sufficiently high temperature without increasing the condensation temperature. If no surplus heat is available, a CO_2 heat pump water heater should be used.

- Optimized instrumentation and monitoring system – The instrumentation should enable measurements of important temperatures as well as electric and thermal power and energy demands for the heating and cooling system. All sensors should be checked and calibrated before start-up, and the monitoring system should be tuned. Standardized graphs will simplify the monitoring of the heating and cooling system.
    - Advanced monitoring systems are of great importance to obtain optimized operation and minimum energy use for the heating and cooling systems and to detect operational problems at system and component level at an early stage.

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Sør-Trøndelag fylkeskommune
www.stfk.no

www.zeb.no