Thesis for the degree of Master of Science

Theoretical Comparative Case Study of two Geothermal Heat Pumps. One with Capacity Control and one with on/off Control.

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Student report

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Titel: Theoretical Comparative Case Study of two Geothermal Heat Pumps. One with Capacity Control and one with on/off Control.

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Synopsis

Hensikten med dette prosjektet var å foreta en teoretisk studie av to måter å regulere et varmeanlegg med en varmepumpe med stempelcompressor. De to typene av regulering var av/på og frekvensstyring. Det ble programmert en modell av en varmepumpe og et varmeanlegg i EES (Engineering Equation Solver). Modellen hadde timebaserte effekter for en fyringssesong som inndata. Disse dataene var hentet fra to modeller av ulike hus. Ett sett var fra Simien og det andre fra BIEF.

EES modellen ble programmert til å beregne tur- og returtemperatur for varmeanlegget basert på en fast romtemperatur , golvkonstruksjonen med varmemotstand og inndata effektene.

Tur- og returtemperaturene fra varmeanlegg ble brukt til å beregne condenseringstemperatur for varmepumpeprosessen. Varmepumpeprosessen ble modellert med underkjøling, overheting, isentrpisk- og leveringsvirkningsgrad, gangtid for kompressor med tilleg for tid for oppkjøring av condenseringstemperatur, spisslast når kondensator kapasitet var mindre enn behov og motor virkningsgrader avhengig av rpm. Av/på varmepumpene ble modellert med et konstant kompressorvolum. De frekvensstyrte varmepumpene ble modellert med kompressor som regulerte omdreining pr minutt slik at innsugd volumstrøm utførte en massestrøm av kuldemedie ved gitt entalpidifferanse ovr kondensator som var lik inndata for ønsket varmeeffekt.

Fordampertemperaturen ble satt til en konstant verdi.

EES beregnet så timesverdier som for eksempel kondensatorytelse, el-forbruk til spisslast og el-motor, trykkforhold i kompressor og timesverdi for varmefaktor. Varmefaktoren ble akkumulert til en Årsvarmefaktor.

Det ble utført beregninger med både on/off og frekvensstyrt varmepumpe på alle modelldata av husene. Resultatene ble laget til grafer for å visualisere og sammenligne resultatene for å finne de beste ytelsene med tanke på SPF.

Resultatene viste at frekvensstyrte varmpeumper hadde den høyeste årsvarmefaktoren.

Abstract

The purpose of this project has been to do a theoretical study of two types of controlling the performance of a heating system based on a heat pump with reciprocating compressor. The two types where On/Off and Frequency Control. A model of a refrigeration process and a heating system for a house were programmed in EES (Engineering Equation Solver). The model used hourly heating demand capacities for the period of a heating season as input data. These data were extracted from two types of models, one set of data were from Simien and the other set of data from BIEF sampling program.

The EES model was programmed to calculate Supply- and Return temperatures for heating system based on a fixed indoor temperature, the construction of the floor heating system with the heat resistance and the input capacities from house models.

The supply/return temperature for the floor heating system was used to calculate condensing temperatures for the refrigeration process. The refrigeration process were modeled with sub cooling, super heating, isentropic and compressor delivery efficiencies, run time for compressor adjusted for ramp time, additional heater when condenser capacity was insufficient and el-motor efficiencies depending on the rpm. The on/off heat pumps were modeled with a compressor with fixed volume. The frequency controlled heat pump was modeled with a compressor that adjusted the rpm in order to obtain the swept volume for performing a mass flow of refrigerant at given enthalpy difference of condenser equal to input data for heating capacity.

The evaporating temperature was set to a fixed level.

The EES calculated then results like hourly data like condenser capacity, electrical consumption for motor and heater, pressure ratio and hourly Performance Factor. The hourly Performance Factor was accumulated for calculating Seasonal Performance Factor.

The modeling of on/off and frequency heat pump was conducted on all of the house model data. The results were made into graphs for visualizing and compared for which had the better performance regarding SPF.

The results showed that the frequency controlled heat pumps had the highest SPF. The results also indicated that if longer heating season with same DOT capacity the on/off heat pump increased the performance compared to an inverter heat pump.

Preface

This thesis was written as part of the Master degree within the field of Indoor Environmental Engineering at Aalborg University 2011.

In this Thesis a model of a heat pump with two different types of adjusting energy performance have been analyzed. The analysis have been performed with data from a theoretical house with theoretical user profiles and a house with sampled data on consumption and the user patterns derived by making a theoretical model of the house.

I would like to thank my employer Multiconsult as and my colleagues in Drammen for their patience and support in me writing this thesis. Thanks to Rasmus Lund Jensen and Mette Havgaard Vorre for guiding me through this, and to Thomas Helmer Pedersen for helping me with modeling in EES.

Johan I. Friisvold

Aalborg 20. June 2001

Nomenclature

A	Area (m^2)
LCC	Life Cycle Cost
COP	Coefficient of performance (-)
SPF	Seasonal Performance Factor
C	Heat capacity (J/Ks)
Cp	Specific heat capacity (kJ/kgK)
d	diameter(m)
DOT	Design Outdoor Temperature
f	Frequency(Hz)
h	Specific enthalpy(J/kg)
l	Length(m)
$m^{\cdot}$	mass flow refrigerant(kg/s)
$m^{\cdot}$	mass flow water(kg/s)
$m^{\cdot}$	mass flow brine(kg/s)
p	pressure(bar)
p1	Condensing pressure(bar)
p2	Evaporation pressure(bar)
Q	Thermal energy(kWh)or(J)
Φ	Thermal capacity (W)
t	temperature(C)
T	Temperature(K)
tp	Temperature high pressure compressor (C)
U	Thermal transmittance(W/m2K)
$V^{\cdot}$	Volume flow rate
V	Volume(m^3)
W	Power(el)(W)
X	vapour quality(-)

α	Heat transfer coefficient(W/M2K)
Δ(d)	Differense
η	Efficiency(-)
λ	Pressure drop coefficient, or
δ	Thickness of material(m)
ρ	Density(kg/m^3)
v	Velocity(m(s)
λ	Compressor delivery rate

Table of Contents

1 Purpose

The purpose of this work is to find out if it is a larger potential of reducing the share og exergi (electricity) used for heating houses using an inverter controlled heat pump rather than an on/off one which is the common way today to control geothermal based heat pumps. At the same time determining the impact internal head loads has on design size of heat pumps for a number of user profiles

This is done by feeding data from modeled houses compensated for user profiles into models of the two types of heat pumps in EES. The models convert power demand into temperatures and use the temperature to reflect the refrigeration process in a heat pump. The SPF for each case is calculated and the result analyzed.

2 Background for studying heat pump SPF (Seasonal Performance Factor)

The reason for studying SPF in heat pumps is to establish knowledge on which systems that has the least electricity input in order to utilize such a system for reducing non-renewable energy sources. This chapter tries to place heat pumps in a larger picture and tries to narrow down which type of buildings and user profiles the heat pump in this work is chosen for.

On an overall view the earth's population is growing (1) and most likely also the use of energy together with it. A large number of people in the world think it is necessary to reduce the use of non-renewable energy (i. e .fossil fuels) in order to reduce pollution and the potential of Global Warming. (2) Reducing use of fossil fuels can be done by replacing it with renewable energy such as from geothermal heat pumps. EU parliament has issued a directive on increase of share of renewable by 11,5% from 2005 to 2020 for EU countries (also EØS and EFTA countries). (3) One of the main reasons for increasing renewable energy use for EU is to ensure the supply of energy within EU.

2.1 Energy "production" today. (Transformation?)

In the last century oil and other fossil fuels has been a very important energy source. The fact that oil reserves won't last forever is making many world leaders to focus on ways to replace fossil fuels as energy source.
Energy is derived from several sources and amounts as listed (2004) below.

Oil 34.3%
Gas 20.9%
Coal 25.1%
Nuclear 6.5%
Combustible renewable and renewable waste 10,6%
Hydro 2,2%
Geothermal 0.414%
Solar 0.039%
Wind 0.064%
Tide 0.0004%
Non-renew. Waste 0.2%
Other** 0.5%

IEA Energy Statistics (4)

If oil, gas and coal is to be replaced by renewable energy there will e a need to minimize energy use from high quality sources such as electricity and use as much as possible low quality energy (solar and geothermal)

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2.2 Energy consumption today and predicted. Potential of replacing with renewable source.

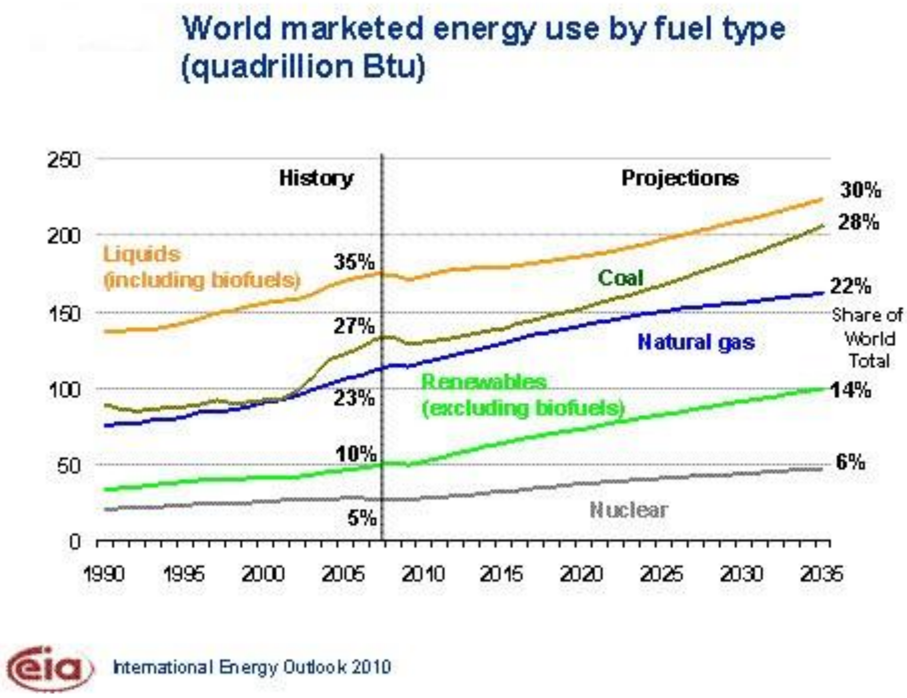

Fig 01 shows the predicted increase of use of energy split on each source. (5)

The energy use per sector was in 2007 according to IEA like this:

Industry (agriculture, mining, manufacturing, and construction)	51%
Transport	27%
Residential	14%

Commercial sector (service & institutional sector) 7%

This shows that total residential and commercial sector energy use estimates to about 21%. The amount used for heating and hot tap water varies depending on altitude, culture, size house and so on. The potential for increasing the share of renewable (geothermal) energy should be substantial since production only amounts to bit more than 0,4% (ref chapter 2.1). (6)

3 Categories of buildings and locations for making models.

Buildings can be split up into sub groups under "Industry, Residential" and Commercial" from chapter 2.2 in order to systemize user patterns and building envelope. The various parameters for categorizing are such as:

- Location, thermal (longitude, face-direction, obstruction, m above sea level,)
- Geographical and density(common energy source)),
- Size (no floors, perimeter relative to sq meter/volume),
- Type of usage (living house, elderly home, office, factory, cabin, warehouse etc),
- Time of usage (all year, daytime etc – comes with type of house),
- Type of building materials (heavy/light), amount of sq meter windows,
- Type of heat distribution (air-borne, water borne, direct el). Construction type/degree of insulation, sun shading.
-

3.1.1 Sub group Living house

Living houses can again be divided into groups like

- Residential houses,
- blocks,
- row-houses,
- Bungalows (one floor).

4 Chosen building in this work for analysis of SPF.

4.1 Main data

In this work it is chosen a to-floor wooden residential house located in the middle of Norway and one Bungalow located in Denmark.

Age approximately 30 year old

Hydronic system for energy distribution

Brine/water heat pump

Bore hole for energy source

4.2 Building envelope and user impact on energy use.

4.2.1 Danish house

The building is modeled and energy consumption calculated. General data on energy/power from area is obtained and incorporated to show energy/power profile of building with occupants.

4.2.2 Norwegian house

A fictitious house was modeled in Simien (7) and added/subtracted estimated user pattern regarding internal heat loads and consumption.

4.3 Ratio heating/hot water residential.

4.3.1 In general

One can assume that the amount of energy used to compensate for heat loss through building envelope is a function of day degrees and inside temperature. This is taken care of in models. Hot water consumption data are usually presented as an amount per habitant per day, or a percentage of total energy consumption for house/household. A common way of calculating the amount of water consumption in households is to have f .i. 40liter as basis and add 35 liter per household member. (7) Another method is using 60 liter a day flat.

4.4 Energy carrier

The most suitable energy carrier for heating a building using heat pump is a hydronic system (opposite to a airborne heating system). The energy can then easily be distributed from a central location in the house and there are less problems with dry air, noise and distribution of pollution (if 100% fresh air is used the energy consumption pre-heating this will result in higher energy demand). Additional peak/backup power can also easily be implemented in water borne system. Floor heating is often preferred in a heat pump powered system because of low supply temperature due to large emitting area which is one of conditions for achieving a high COP. Fan-coils can also provide the same power at similar water-temperatures while radiators normally need higher temperatures. (9)

4.5 User profiles of habitants.

A user profile for occupants is implemented in energy/power consumption of the houses in order to determine the impact the variations has on sizing of the heat pump power.

1. Young couple with small kids, 2Y2y. (Ref enclosure B)

5 Larger share of renewable energy by use of heat pumps.

5.1 Less losses with less consumption

Heat pumps extract energy from a source with lower temperature level and lift it up to usable level by supplying compressor work. The COP (Coefficient of Performance) reflects the efficiency of a heat pump by making a ratio of usable power out per power input. (10)

The most common energy sources for compressors are electricity. Electricity is converted either from renewable source or from fossil or nuclear source. Losses occur when energy is extracted from a source and moved to a higher quality than its surroundings because no processes are 100% efficient. The losses in electricity production and consumption can be listed as:

- Production losses hydro power plant, 80-95%
- Production losses thermal power plant, 35-50%
- Production losses thermal power plant with co-utilization 50-80%
- Transformer losses and Transport losses, 5-10%
- End user losses, 0-30%

If a geothermal heat pump extracts 2 times the input from renewable source and alternative heating is electricity the decrease in production when coming from a thermal plant with η_{total} of 0,5 can be 1/o,5 = 2 instead of 3/o,5 = 6

When fossil fuels are to be replaced there will be a demand on renewable energy.

As most of the hydro plant possibilities in the world have been exploited and many of the renewable energy sources for electricity are somewhat intermittent it makes sense to utilize the energy needed where needed by extracting geothermal by a heat pump. This also goes together with EU's directive on increasing self-sufficiency regarding energy supply inside EU region.

5.2 Private economy

Using heat pumps in houses can also be a cost-effective investment providing the conditions are right.

It is in the house owner's interest to minimize energy consumption for heating the house in order to minimize expenses regarding heating. Differentiated energy prices for day and night could also result in viable solutions for production of energy at night at low price and consumption during day when electricity was utilized as high quality energy.

Differentiating on el and temp energy price

5.3 Literature survey.

When doing a search on previous work executed on analyzing SPF in heat pumps for buildings with hydronic system including habitant's user profile none was found for Scandinavian climate. It was found work on combined HP and AC for Chinese climate by C.K.Lee (11) where the results indicated a minimum reduction of 27% of energy input. A work on performance of all-variable speed chiller plants by Hartman (12) concluded with maximum COP at 50-70% of rated capacity. L.Zhao, L.L.Zhao, Q.Zhang and G.L.Ding are in their work concluding that given a fixed size on the heat exchanger it is a correlation between COP and flow rate in an exchanger (13). Fredrik Karlson in his D.Ph theses investigated the SPF of a theoretical model using an equation for the outdoor conditions and in-line equipment data obtained from manufacturers and state of the art equipment (14). He concluded with an increased SPF of 10% for a variable speed controlled ground source heat pump. Furthermore the result showed that the brine circulator capacity is of high importance regarding energy efficiency of the heat pump system.

6 Hypothesis

If an inverter for controlling the condensing temperature in a heat pump lower the Pc/Pe ratio then the SPF will be larger in such a heat pump.

7 Methodology

This is a (Theoretical) Comparative analytic case study of two types of heat pumps. On-off and frequency controlled. The purpose was to find out wheatear an inverter controlled heat pump has a significant higher SPF than an on/off controlled one. In order to make a realistic comparison calculations were performed of a house with user profile covering a life-span with a model of a on/off and an inverter controlled heat pump for determining the SPF and analyze the findings.

First a literature survey was conducted. The findings were that studies have been conducted on hydronic systems, inverter heat pumps, heat dissipation building envelope, user profiles of habitants of houses. (ref to relevant work) No work was found that took into consideration energy data (energy demand envelope, internal energy load and losses) of a house over the period of a heat demand season and using these data to feed into a heat pump system.

The data on energy demand, heat gain and energy loss due to habitants are gathered from a Danish house in Vejle.

EES modeling program is used to simulate the refrigeration and heating system process and find energy consumption and energy balance in heating system and heat reservoir.

8 Theory

Calculating the SPF for a heat pump supplying energy to a house involves a wide range of calculations. The theory explained here is limited to the equations for heating- and brine system and the refrigeration process used for modeling the systems.

The heat demand input data are extracted from building models.

8.1 Listing of areas that are involving calculation in order of determining COP and SPF

1. Heat load equipment and habitants implemented in heat demand data.
2. Supply temperature floor heating based on heat loss building and temperature resistance through floor structure.
3. Temperature from heat pump based on temperature supply temperature and temperature in storage tank.
4. Determining energy/power ratio for heat pump. Sizing of heat pump.
5. Condensing temperature in heat pump based on supply temperature and exchanger temperature loss.
6. Isentropic efficiency of compressor
7. Volumetric efficiency of compressor
8. Energy extracted from heat source based on η Carnot compressor and mass flow refrigerant in heat pump.

8.1.1 Floor heating - Water borne.

A hydronic system is designed for distributing capacity/energy from a source to a consumer. Consumer here is a house consisting of rooms.

In order to compensate for heat loss through the envelope of a building by using water borne system, there are mainly three different systems that can be used. These are:

- Fan Coils
- Radiators
- Floor heating – (Wall and Ceiling)

Which system to choose depends on for instance if it is a new build or old house, type of heat source, numbers of floors and so on. Floor heating is rated as the better one for comfort and high COP if heat pump is used. This system will be described in this work. (11) Down side is that it is slow to control.

For a heat pump installation it is essential to design a system with as low as possible supply temperature as well as as high source temperature as possible. In this way the geothermal HP system operates with the least input of electricity – hence the COP is high. This implies that the exchangers (refrigerant/water and water/air(brine)) on both sides have low dT to minimize temperature losses in order to minimize the "lift". The "lift" refers to the pressure ratio for the refrigerant cycle.

Floor hydronic systems consistent of pipes and heat distributers typical made of Al (in light material floors) and just pipes in concrete floors. The top of the floor acts like a heat exchanger towards the room with both thermal radiation and convective heat transfer. The heat is transferred with a combination of convection and radiation depending on the difference of floor and surrounding temperatures .For comfort measures the floor temperature should be within certain limits – typically below 26°C (12)on the floor. The lower temperature of the floor depends of the felt-heat number the top material of the floor. Floor with a high number extracts more heat from the foot and it feels cold to stand on (barefooted). (17)

In order to move energy from water to rooms there has to be a dT between circulating water and room air.

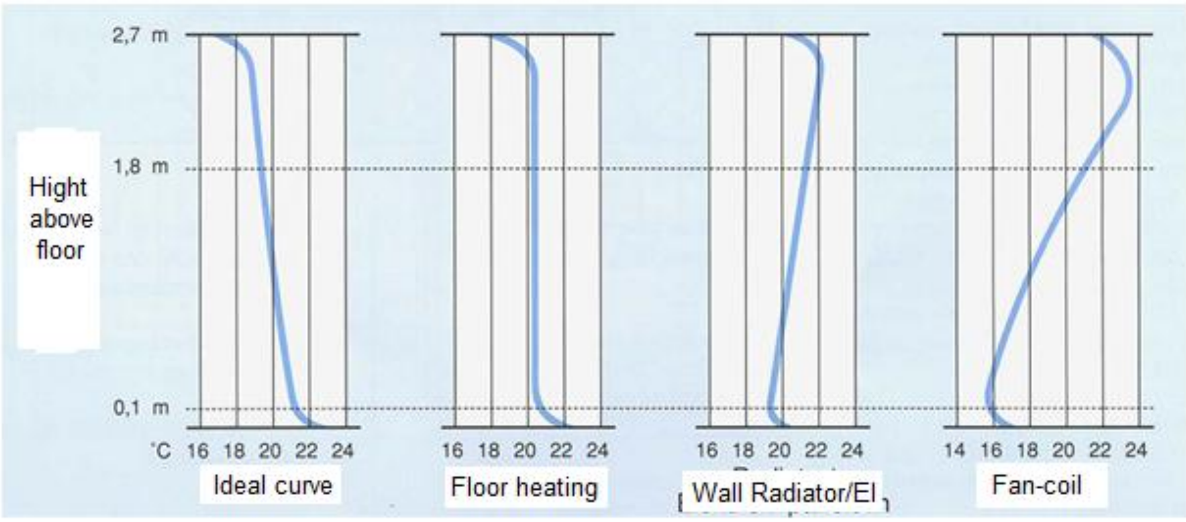

Fig 02 Illustration of four temperature gradients given by four systems of heating. As indicated a floor heating system is closest to ideal temperature gradient. (13)

8.1.1.1 Floor – room dT

The temperature of the floor (and thereby the power from the floor) is calculated based on the need of power to obtain chosen room temperature and is a combination of thermal radiation and convective heat transfer. The empiric formula for power floor heating with regards to dT floor room . (11) is found to be:

$$\Phi = 8{,}92 * dT^{41{,}1} \qquad \text{(W)} \qquad [1.1]$$

8.1.1.2 Floor construction – dT one dimensional.

To find the mean circulating water temperature for the floor the temperature resistance between floor surface and water pipes is to be calculated. Below equation calculates the temperature on next layer when power and *t* start is known.

$$d\Phi = \frac{\lambda}{s} * A * (T_2 - T_1) \quad \text{(W)} \qquad [1.2]$$

8.1.1.3 Floor water pipes

Equation for conduction through cylinders calculates power from internal to external radius $r_{1\ \text{and}}\ r_2$, for the length denoted as and l with the temperature difference between the inner and outer expressed as $t_1 - t_2$. Heat transfer coefficient α and thermal transmittance λ is incorporated in the equation.

$$\Phi = (t_1 - t_2) * \frac{\pi}{\frac{1}{\alpha * r\_1} * + \frac{1}{2*\lambda} * \ln\left(\frac{r2}{r1}\right)} \quad \text{(W)} \qquad [1.3]$$

The area A of the heat flux is 2πrl.

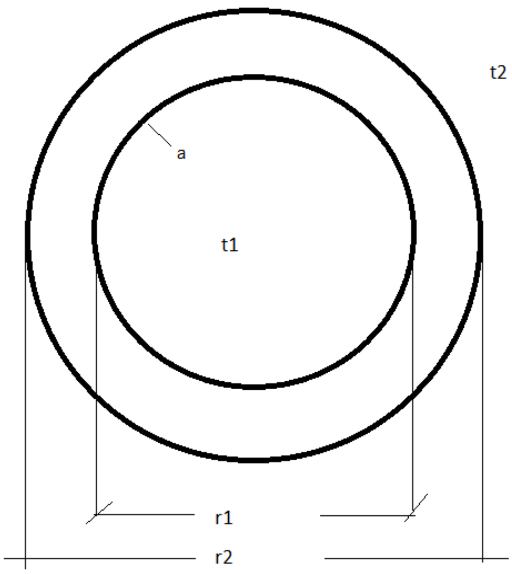

Fig 03. Sketch of floor water pipe. T1 water temp, T2 heat distributor

8.1.2 Hydronic system floor.

8.1.2.1 Heat transfer coefficient.

When transferring power by circulating fluid the c_p is usually fixed. The most common substance to use as circulating fluid is water with an inhibitor to prevent organic growth and/or mixed with antifreeze. This will alter the c_p.

For controlling power transferred this leaves the flow and/or dT_{fluid} as control parameters. In order to transfer power efficiently from fluid to pipe wall the flow should be turbulent hence Reynolds number above laminar flow. When designing a floor heating system it is choices to be made according to what is most cost effective regarding investment and LCC regarding pipe sizes. Ø17 mm PEX for floor heating systems are often used.

Recommendations from Uponor/Wirsbo for 17 mm PEX (15) is $flow_{min}$ at 0,2 m/s and $flow_{standard}$ at 0,45 m/s for 17 mm PEX.

In order to establish a heat transfer coefficient for use in equation [1.3] between water inside PEX and wall PEX the Reynolds number can be expressed like this:

$$\Phi Re_{17} = v_{fluid} * d_{pipe} / \mathrm{v}_{fluid} \quad (-) \qquad [1.4]$$

When chosen $velocity_{water}$ is 0,2 m/s, $viscosity_{water}$ is 0,85 E^{-6} and $Øpipe_{inside}$ is 13mm the Re amounts to 3058 which is above limit of laminar flow at 3000 (16) Heat transfer coefficients turbulent flow water are for velocities between 0,5 and 5,5 m/s 1700 – 23000 W/m^2K. (17) The minimum velocity in floor heating pipes should then be above turbulent flow in order to achieve sufficient heat transfer number.

8.1.2.2 D T fluid floor impact on COP

In order to achieve a low supply temperature from heat pump (high COP) the dT $fluid_{floor}$ should be relatively small. This implies high velocity and/or short pipes for each zone in the house. As mentioned under [1.4] the minimum should be above turbulent.

dT for compensating heat loss house.

$$\Phi = m^{\cdot}_{circulating\ water} * cp * (T_{supply\ temp\ floor} - T_{return\ temp\ floor}) \text{ (W)} \quad [1.5]$$

And

$$\left(T_{supply\ temp\ floor} - T_{return\ temp\ floor}\right)\ can\ be\ about\ 3 - 10\ K\ at\ DOT \qquad [1.6]$$

This will be a compromise between power input for $fluid_{circulation}$ and achieved lower condensing temperature COP of heat pump (and investment). COP calculation is given by equation:

$$COP = \frac{\Sigma\Phi_{delivered}}{\Sigma P_{input}} \quad (\text{-}) \qquad [1.7]$$

Were $\Sigma\Phi_{delivered}$ is Φcondensing, $\Phi_{pump\ energy\ to\ water/rine}$, and $\Sigma\Phi_{input}$ is $P_{input\ compressor}$ and $P_{input\ pumps}$ (+P for control devices). (18)

If dT fluid floor is increased the Tsupply is increased and LMTD condenser will decrease. To achieve same power output the Tcondenser must be increased and then the COP decreases.

If dT fluid floor is increased the flow fluid decreases hence same power. Calculation for pump power is

$$\Phi_{pump} = m^{\cdot}_{circulating\ fluid} * \rho * \Sigma dp_{hydronic\_system} \quad (\text{W}) \qquad [1.8]$$

Were dp is pressure loss in circulation system floor.

$$\Sigma dp_{hydronic\ \ system} = dp_{components} + dp_{piping} + dp_{\xi} \quad (\text{p}) \qquad [1.9]$$

To optimize the system regarding COP on P pump the equation

$$COP = \frac{P_{cond} + P_{pump} * \eta}{P_{comp} + P_{pump}} \quad (\text{-}) \qquad [1.10]$$

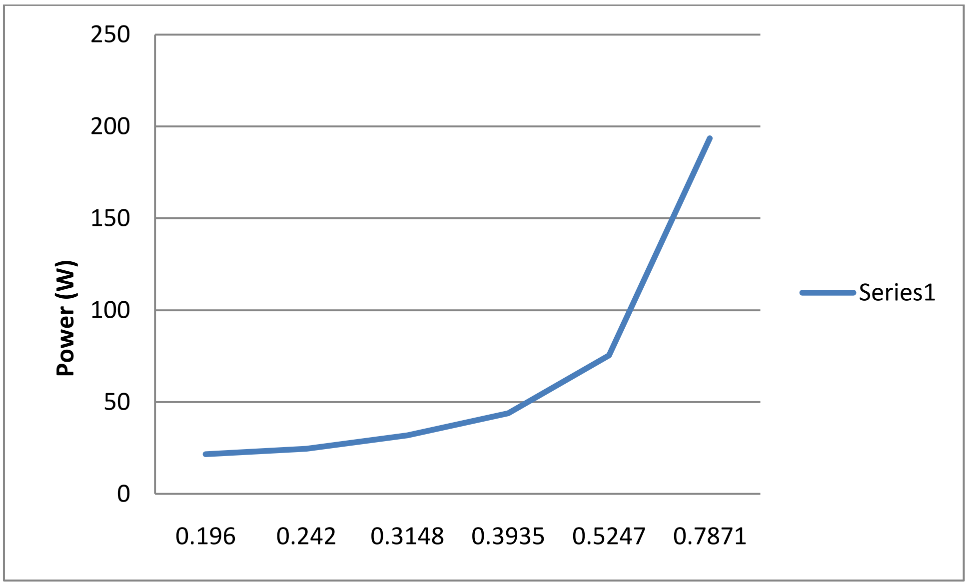

Fig 04. Illustrating the impact the velocity in PEX 17 has on system curve total. X-axis giving flow in m/s.

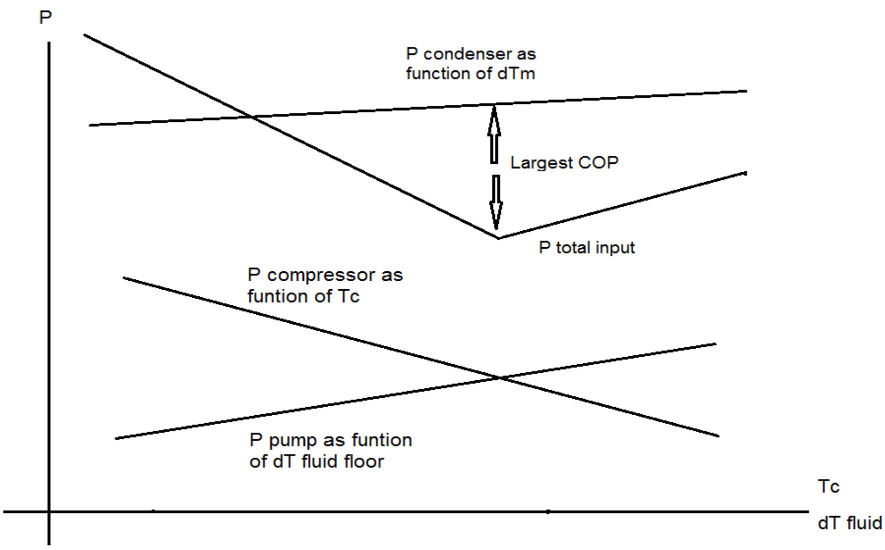

Fig 05. Illustration of connection between condensing temperature and fluid flow.

High velocity leads to high dp pipes which results in more energy for circulation pump. It is possible to have variable flow in order to reduce pump work at for instance 2/3 power demand and lower. Reduction shouldn't affect the heat transfer number in such way that supply temperature from heat pump need to increase.

8.1.2.3 Condenser calculations

Condenser logarithmic temperature difference is given by equation:

$$LMTD = \frac{dTa - dTb}{\ln\left(\frac{dTa}{dTb}\right)} \quad \text{(K)} \qquad [1.11]$$

And

$$\Phi = U * A * \mathrm{LMTD} \quad \text{(W)} \qquad [1.12]$$

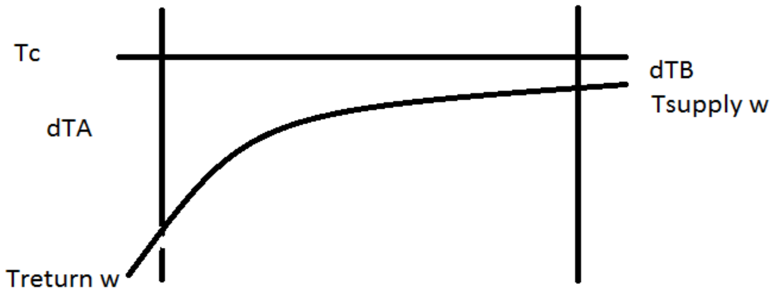

Fig 06. Sketch illustrating temperatures in a heat exchanger.

Fig 06 is showing a exchanger with condensation on the hot side.

If given the same Tcondensing the LMTD will decrease if dTA increases and dTB decreases due to increase in dT fluid floor.

It becomes evident that when LMTD decreases the U or A in condenser must increase in order to obtain same power transfer to fluid. A is somehow difficult to alter in a running plant but U can vary due to changes in mass flow rate through condenser.

8.1.3 Rest time compressor and Volume storage tank.

The compressor in heat pump with on/off control is normally set to stop when achieved temperature level set point. Set point can be temperature needed for power the demand or higher level for storing energy. “Off” time compressor is needed to ensure sufficient lubrication to avoid damage of the compressor. To ensure energy availability at “off time” a buffer tank or accumulator is often implemented. The energy availability is then ensured at “off time”.

A common limitation for time between start of compressors is 10 minutes. (19)

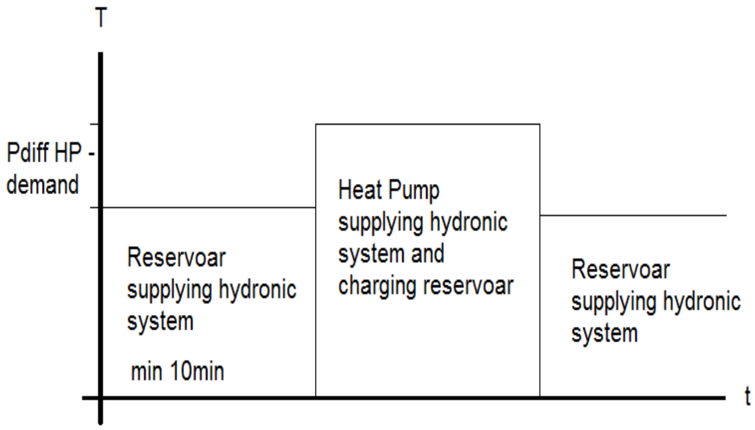

Fig 07. Charge and discharge of buffer tank – Php larger than Pdemand.

$$Q_{buffer} = \Phi_{\mathrm{hp}} * \mathrm{t}_{\mathrm{rest}} \quad \text{(Wh)} \qquad [1.13]$$

Conditions for t design and subsequently maximum Q_{buffer} is when P_{demand} is near P_{hp}.

The accumulating substance can be a fluid or a combination with a phase-shift substance for reduction of storage volume.

8.1.4 Refrigeration Process

8.1.4.1 Carnot cycle

The Carnot cycle is a reference to measure theoretical or practical efficiencies of heat engines. The cycle is a reversible process and performed with an ideal gas that follows two isotropy's (S1 and S2) and two isotherms (Tc and To). The temperature is in K and entropy in kJ/kgK. The reservoir is defined by (20)

$$T_o * (S_2 - S_1) \qquad \text{(kJ/kg)} \qquad [1.14]$$

The work applied to move this amount to level defined by

$$(T_c - T_o) * (S_2 - S_1) \qquad \text{(kJ/kg)} \qquad [1.15]$$

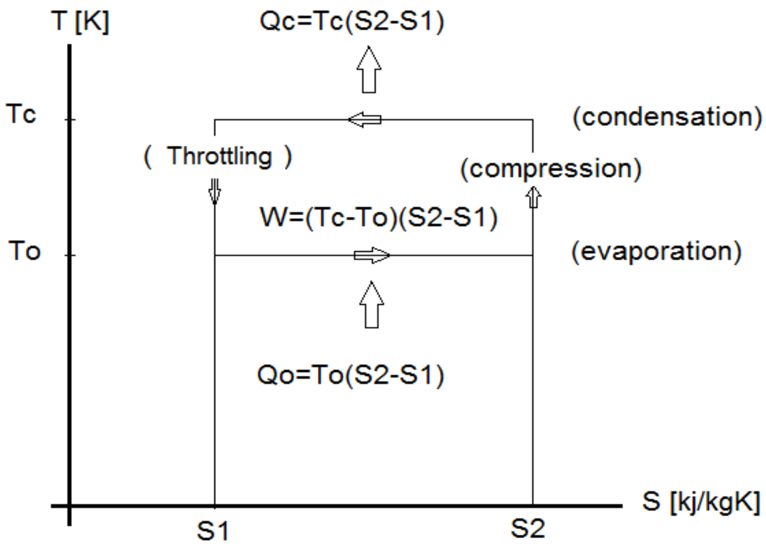

Fig 08 Carnot process in T-S diagram

Theoretical Carnot factor for heat pump where the Qo is the heat reservoir and the Qc is the heat sink. W is theoretical work added to lift reservoir up to Qc level..

$$\varepsilon_c = \frac{Q_c}{W} = \frac{T\_(S2-S1)}{(T_c-T_o)(S_2-S_{1)})} = \frac{T_k}{T_k-T_o} \quad (-) \qquad [1.16]$$

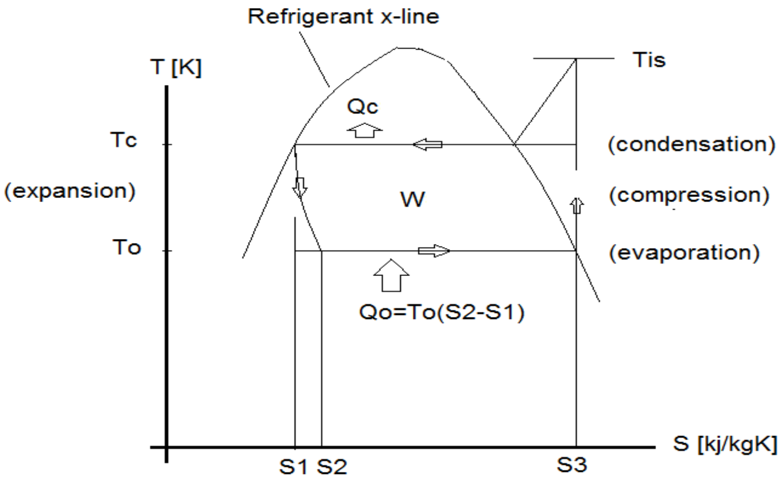

Fig 09 Theoretical refrigeration process in T-S diagram

In fig xx the theoretical refrigeration cycle takes the energy Qo from level To and lifts it up to level Tc by adding W at compression. The refrigerant is then throttled from Tc to To along a enthalpy bar..

8.1.4.2 Refrigeration cycle in logP-h diagram

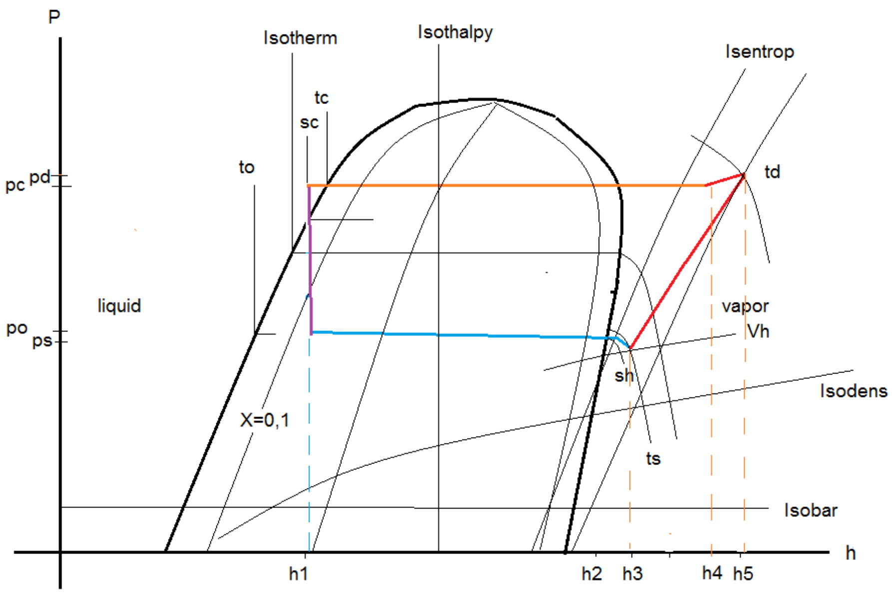

Fig 10 Sketch of Refrigeration process in logP-h diagram

Describing a refrigeration cycle is often done in a logarithmic pressure-enthalpy diagram. The refrigeration process lifts capacity/energy from a level *to* which is evaporating temperature, to *tc* which is condensing temperature. These to levels are also defined by po and pc.

Depending if it is a azeotrope or zoetrope refrigerant the evaporating and condensing occurs at isotherm conditions - or with a glide. The refrigerant changes between liquid and vapor conditions. The refrigerant evaporates from liquid to vapor in the evaporator, defined between enthalpy states h1 and h2 in above sketch. The refrigerant changes from x=0 which is pure liquid to x=1 at h2 which is pure vapor. The opposite is happening in the condenser. Depending on temperatures and the heat exchanger properties the refrigerant can achieve sub cooling which is temperature below pure liquid conditions. The opposite happens at the outlet of the evaporator where super heat occurs.

The level of which the energy is transported to is below the condensing temperature and depends on the dT in the heat exchanger. Same for extracting energy, the evaporating temperature is lower than heat source hence energy can only move from a higher to a lower level.

Evaporator specific capacity is dh between h2 and h1. Condenser specific capacity is between dh h4 and h1. Compressor specific work is dh between h5 and h2.
The more sub cooling that is achieved the higher the evaporator capacity turns out. Subsequently the lower level the temperature needs to be delivered at (Tc) the smaller share of compressor work applied.

The condition of the gas entering the compressor has a high impact on the performance of the compressor and the process. If the vapor has a low super heat or even degree of moist the delivery efficiency will be reduced significantly due to loss from swept volume. Some refrigerants has properties that makes it worth wile to exchange heat between the suction gas and liquid. This applies to R-134a. For R-22 and ammonia the opposite is happening.

The equation for compressor work can be shown like this:

$$P_e = \frac{P_t}{\eta_{is}} = \frac{\dot{m}_r(h_{is}-h_s)}{\eta_{is}} = \frac{\dot{V}_{induced}(h_{is}-h_s)}{V_{swept}*\eta_{is}} = \frac{\dot{V}_{induced}*\lambda*(h_{is}-h_s)}{V_{swept}*\eta_{is}} \text{ (W)} \qquad [1.17]$$

Factors that increase compressor power consumption are increasing h_{is}-h_{swept} because ratio p_c/p_o increases and that η_{is} decrease.

Factors that reduces compressor power is when λ decreases and that v_{dens} decreases ($m^3/kg_{refrigerant}$ increases). This result in reduction for both V $flow_{swept}$ and mass $flow_{ref}$

Refrigerants

Refrigerants can be divided into natural ones like CO_2, NH_3, and C_3H_8 which has low impact on the Global Warming Potential and low ozone depletion potential and CFC which are banned, HCFC which are temperedly drop in blends, and HFC which are defined the least damaging on the environment. Some of the HCFC’s and HFC’s are blends with glide. (21) (R22 has similar properties as R290) -11)

8.1.4.3 Efficiencies and losses compressor.

Isentropic efficiency. (ɳis)

The compressor work is defined between h5 and h3. (Ref Fig __) Theoretical ideal compressor work is defined between h4 and h3 along an isentrop. Isentropic efficiency (ɳis) is defined as (h4-h3)/(h5-h3). Because of not ideal compression the added compressor work is transformed to energy in the refrigerant and results in higher Pt and compressor work as convection loss from compressor outside.

Delivery efficiency (λ)

The delivery efficiency can be described as the sum of several losses. This is

- loss from back flow from clearance room,
- loss from expansion of swept volume,
- loss from impellent pressure (dp for inducing flow), (loss from throttle).

Clearance room loss cylinders

The volume loss occurs between 3 and 4 (shown in the fig below) when a small amount of compressed gas at discharge pressure expands back into the cylinder before inlet valve opens.

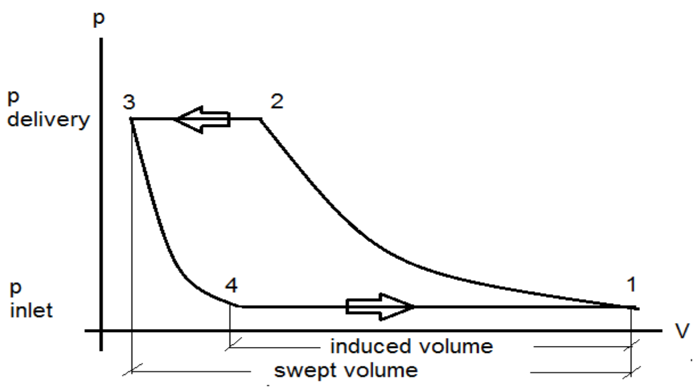

Fig 11. Volumetric efficiency of a reciprocating compressor shown in a P-V diagram.

Expansion of swept volume loss

This is loss when refrigerant vapor from evaporator expands inside compressor because of rise in temperature from hot surfaces into and inside cylinder wall. This is causing a decrease in mass flow refrigerant comparison to Vh suction.

Swept volume loss is defined by what ratio vapor at suction condition the compressor manages to discharge at pdischarge and consist of induced volume per swept volume.

Impellent pressure loss.

To be able to move a substance a pressure difference is required. The compressor has to have a lower suction pressure than evaporation pressure and a higher discharge pressure than condensing pressure in order to operate as a pump.

Volumetric and isentropic efficiencies are manufacturer data or can be measured during operation of the compressor.

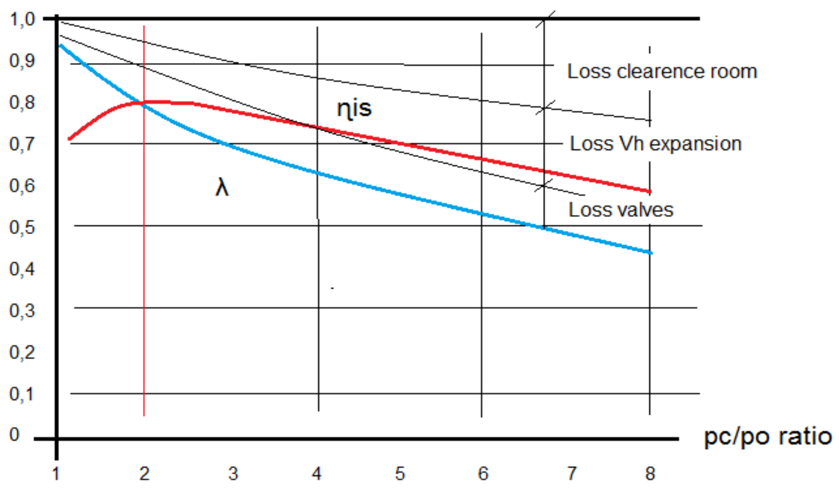

Fig 12. Volumetric and isentropic efficiencies in chart.

8.1.4.4 HP capacity regarding DOT conditions

The reason for investing in a heat pump is normally to save money over a period of time. The investment and running costs split into the life span of the investment should give a lower energy price than the alternative. The optimum is the size of heat pump that gives the lowest price per kWh delivered.

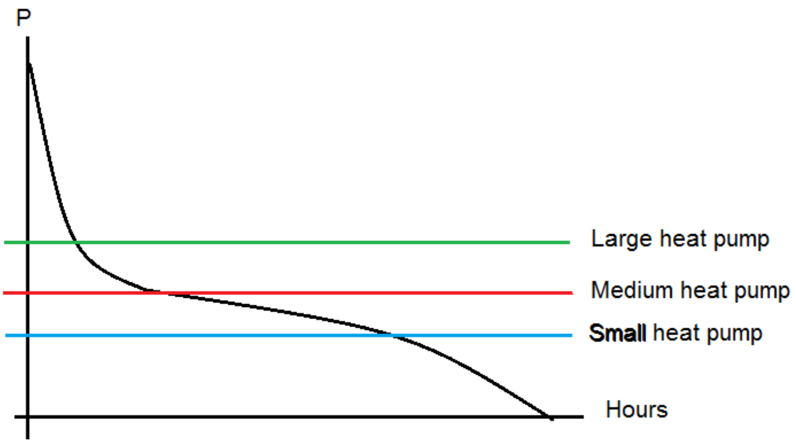

Fig 13 Sketch of energy coverage of a small, medium and large heat pump of a typical Nordic climate with long heat demand season and some days with peak demand.

A large heat pump is here just above 50% of heat capacity for DOT and a small one about 25% of DOT heat capacity.

Usually a larger size has a higher price, Fig xx shows some parameters that are important when deciding the size of the heat pump.

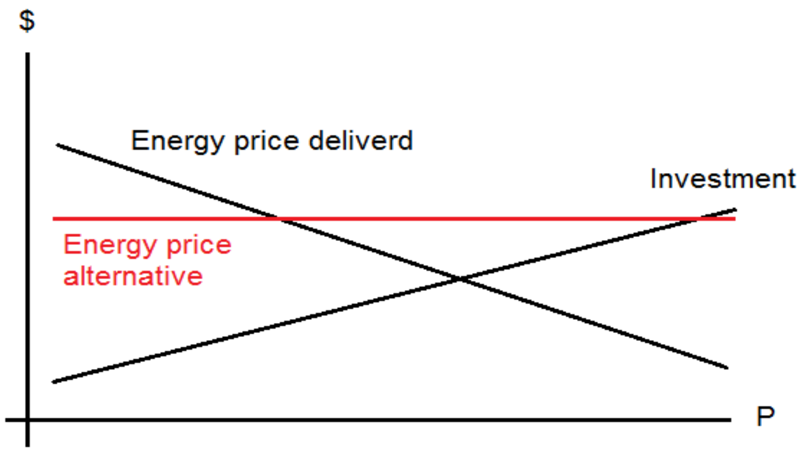

Fig 14.   Diagram on three parameters that decides size of heat pump.

8.1.4.5 Compressor power consumption and efficiencies.

In order to perform the “lifting” work the compressor has to overcome a number of losses. The theoretical power consumption is

$$P_{tcompressor} = m^{\cdot}_{ref} * (h_{is} - h_s) \quad (kW) \qquad [1.18]$$

The efficiencies can be listed as follows:

- η_i – (indicated) = Pt/Pi. These are the losses described as λ number and often shown in diagrams. These include pressure drop in refrigeration circuit.
- η_m – (mechanical) = Pi/Pe. This is the losses to do with mechanical and fluid friction inside compressor. For estimation can equation: $P_{m\ loss} = f_m * V^{\cdot}_s$ be used. F_m = 50-90 kWs/m^3
- η_{is} – (sum I and m) = $\eta_i * \eta_m$
- η_{tr} – (transmission) = Pe/Pel. This will be zero for hermetic and semi hermetic compressors that has a common shaft for electro motor and compressor.
- η_{el} – (electrical, heat) = The losses occurring due to mechanical, temperature and field losses. Varies depending on degree of load on motor, and motor size. And if motor is incorporated as part of compressor.
- η_{inv} – (electrical, heat) = Losses due to electrical circuits in inverter altering voltage and frequency.

$\eta_{total} = \eta_i * \eta_m * \eta_{is} * \eta_{tr} * \eta_{el}$ ($* \eta_{inv}$ if used) (-) [1.19]

8.1.4.6 Circulation pumps power consumption and efficiencies.

In order to perform the circulation in the hydronic system and the brine system the pumps has to overcome a number of losses. The theoretical power consumption is

$$P_{tpump} = V^{\cdot} * dp(pipe\ wfitiings + valves + equipment) \quad (kW) \qquad [1.20]$$

The efficiencies can be listed as follows:

- η_i – (indicated) = Pt/Pi. These are the losses described as λ number and often shown in diagrams
- η_m – (mechanical) = Pi/Pe. This is the losses to do with mechanical and fluid friction inside pump.
- η_{tr} – (transmission) = Pe/Pel. This will be zero for in-line pumps that has a common shaft for electro motor and impeller.
- η_{el} – (electrical, heat) = the losses occurring due to mechanical, temperature and field losses. Varies depending on degree of load on motor and motor size. And if motor is incorporated as part of compressor.
- η_{inv} – (electrical, heat) = Losses due to electrical circuits in inverter altering voltage and frequency.

$\eta_{total} = \eta_i * \eta_m * \eta_{is} * \eta_{tr} * \eta_{el}$ $(* \eta_{inv}$ if used) [1.21]

Rotation speed of electric motor and compressor.

The rotation per minute of the electric motor (AC) is dependent on the frequency of the AC, number of poles of the motor and slip like this:

$$Rpm = \frac{120*f}{p} * (1 - Slip\ rate) \quad \text{(r/min)} \qquad [1.22]$$

Were

f is frequency

p = number of poles in el motor

Slip rate is the difference between synchronous rotation to the AC frequency and the actual speed caused by needed difference in location of the field and the rotor in order to move the rotor. The more load on the motor the more power and torque needed in the field. The maximum slip-rate is at maximum load of motor/maximum current input. (26)

If the motor and compressor are direct coupled the speed of the compressor equals the electric motor.

Compressor volumes

The equation [1.22] describing the relation between rpm and compressor data to amount of swept volume.

$$\dot{V} compressor = \frac{pi*d^2*s*z*n}{4*60} \quad (m^3/s) \qquad [1.23]$$

Were

D is diameter cylinder

S is length of piston movement

Z is number of cylinders

N numbers of cycles per minute

Induced volume is

$$\dot{V} induced = \frac{\dot{V} compressor}{\lambda} \quad (m^3/s) \qquad [1.24]$$

8.1.4.7 Calculating power consumption and energy consumption

Required power for el motors compressor and pumps are then:

$$P_{compressor} = m^{\cdot}_{ref} * (h_{is} - h_s)/\eta\text{total} \quad (kW) \qquad [1.25]$$

$$P_{pump} = V^{\cdot} * dp(pipe\ wfitiings + valves + equipment)/\eta\text{total} \quad (kW) \quad [1.26]$$

The hydronic system in a house usually circulates all the time, while the brine system usually is circulating when the compressor is running plus a time before start of compressor to ensure stable conditions.

$$Q_{compressor} = P_{compressor} * t_{compressor} \quad (kWh) \qquad [1.27]$$

$$Q_{brinepupmp} = Q_{\_brinepupmp} * t_{compressor} \quad (kWh) \qquad [1.28]$$

$$Q_{hydronicpump} = Q_{\_hydronicpump} * 24 * 365 \quad (kWh) \qquad [1.29]$$

Pumps were calculated like this:

$$dP = \frac{0{,}316 * Re^{-0{,}25} * 0{,}5 * \rho * v^2}{d_{pipe}} \quad \text{(Pa)} \qquad [1.30]$$

And Rey number where calculated like this

$$Re = \frac{v * d_{pipe}}{visk} \qquad (-) \qquad [1.31]$$

Velocity fluid was chosen for Re number above turbulent flow in order to achieve good heat transfer. Viscosity for water and brine respectively.

8.1.4.8 Hydronic system in Logp-h chart

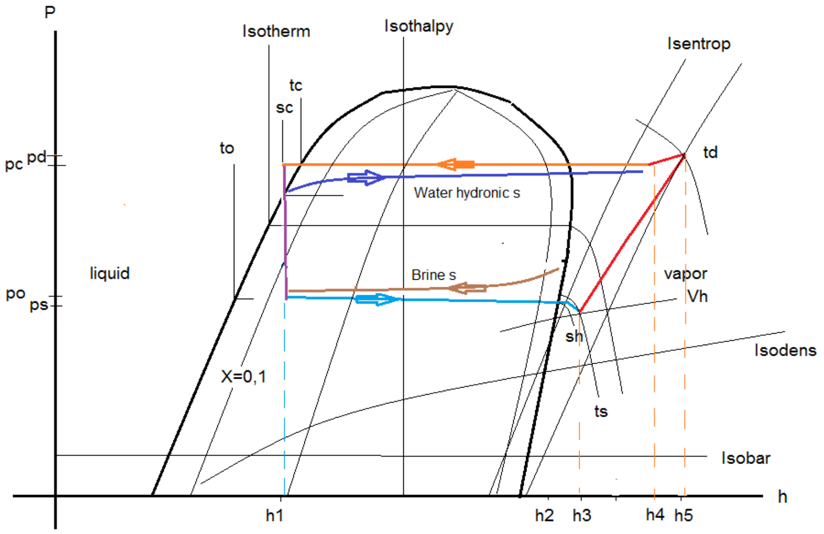

Fig 15. Hydronic floor system and brine collector system indicated in logp-h diagram

Compressor work is defined as.

$$P_{compressor} = m^{\cdot}{}_{ref} * (h_4 - h_3)/\eta\text{total} \quad (kW) \qquad [1.32]$$

The shaft power will be

$$P_{shaft} = m^{\cdot}{}_{ref} * (h_5 - h_3)/\eta\text{total} \quad (kW) \qquad [1.33]$$

COP

The COP for a heat pump can be shown like this:

$$COPheating = \frac{\Phi heat\ utelized}{Pinput} \qquad (-) \qquad [1.34]$$

where

- Φheat utelized is heat from condenser, compressor, pumps and so forth that goes towards heating the location intended.
- P_{input} is power for keeping the system running, like compressor work and pump work (21)
- COP is a coefficient calculated by power inputs that gives capacity values.

SPF

The SPF for a heat pump can be shown in the same way but with Q instead of P.

$$SPFheating = \frac{Qheat\ utelized}{Qinput} \qquad (-) \qquad [1.35]$$

where

- Qheat utelized is heat from condenser, compressor, pumps and so forth that goes towards heating the location intended.
- Q_{input} is energy for keeping the system running, like compressor energy and pump energy (21)
- SPF is accumulated values over a period of time. A season usually lasts from when the outside temperature goes down to 13-15 °C in the autumn till the temperature reaches the same in the spring.

8.1.4.9 Heating system sketch.

The system includes the equipment shown in Fig 16bwith exception of the collector and the evaporator which were taken out. The system boundary is shown as dotted line.

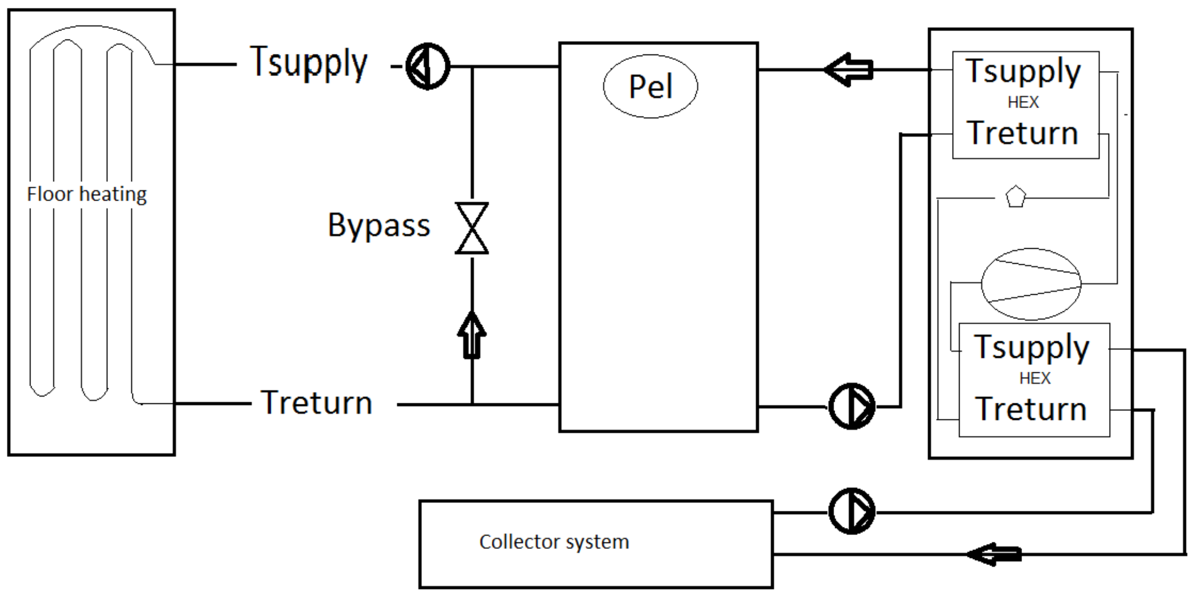

Fig 16   Schematic diagram of hydronic system, buffer tank, heat pump and collector system

Bore hole

A bore hole is supposed to give the Geothermal renewable energy input to the heat pump. The energy is transported from the hole to the heat pump by a fluid in a pipe or ground water is pumped up and returned after energy has been extracted. Design data for a bore hole is approximate 30-40 W/m usable hole (with water – and capacity varies greatly depending on type of rock) when using a collector loop. If pumping up ground water tests has to be performed to determine energy that can be extracted.

Temperature at 10-15 meter depth is often 1-2 degrees higher than ground water temperature. At 200 meter depth it is about 3-6 degree higher.

Heat conductivity for rock varies around 2-4,5 W/mK   (Norway). (22)

In order to minimize the amount of refrigerant in the process a secondary medium is often used for transporting the energy from source to the heat pump. Opposite a system with refrigerant is extracting the energy directly from the source is called a DX-system (direct expansion). A collector system has one more temperature exchange than a DX-system and consequently a higher temperature loss.

Collector system is often a PE pipe with a mix of water/antifreeze making c_p. The Heat exchanger/evaporator in HP is normally a X-plate exchanger for domestic heat pumps. These are small in volume and efficient, down side is they are hard to clean. If the circulating fluid carries pollution this can and will cause fouling of the exchanger.

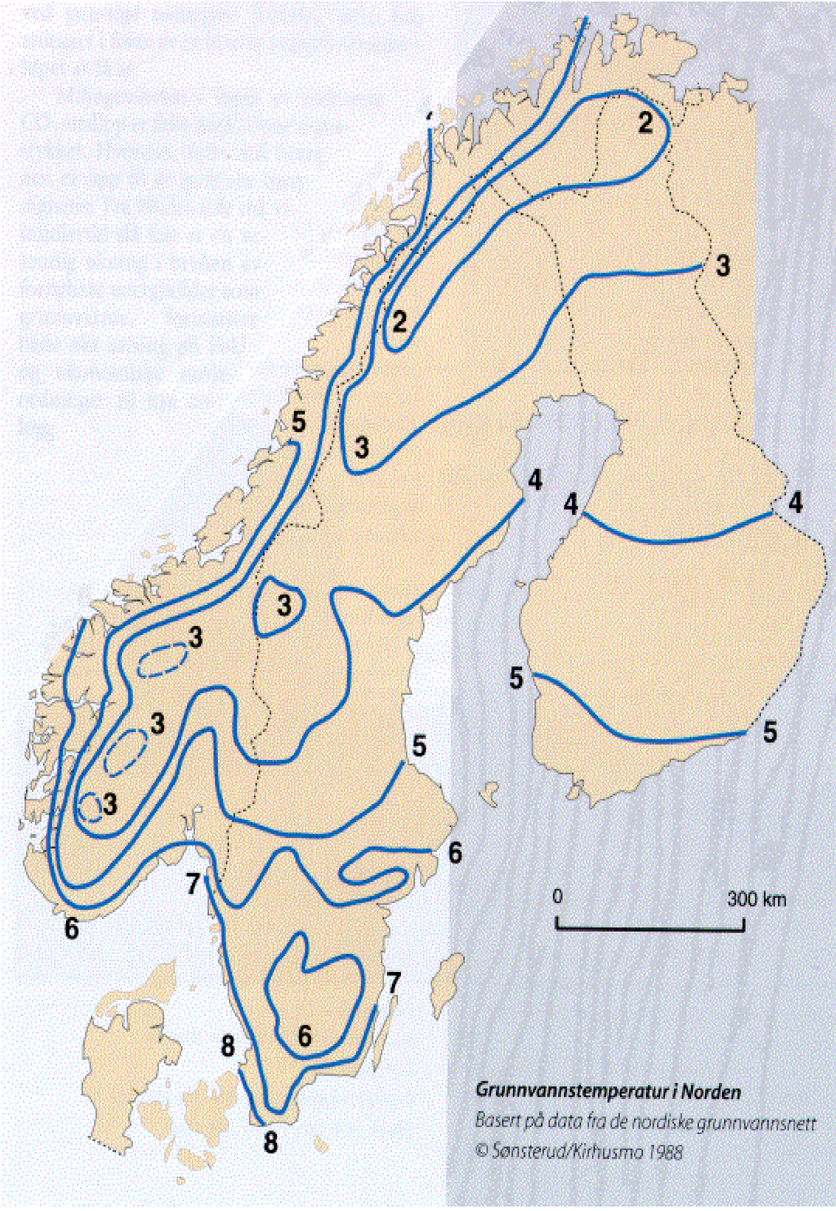

Fig 17. Map of the ground water temperature in Norway (29)

9 Modelling the hydronic- and refrigeration systems for on/off and inverter controlled Heat Pump.

For comparing the two types of heat pumps, models were made in EES to simulate performance over one heat demand season. The result of the calculations in EES shows the SPF for one HP with on/off control and one with inverter control. The belief is that an inverter controlled one is more adapted to the performance hence higher SPF.

9.1 Model of on/off controlled Heat Pump.

This chapter explains the method used for making a model for calculating the SPF for an on/off HP based on input data. The numbers used to explain method are referring to one of the house models.

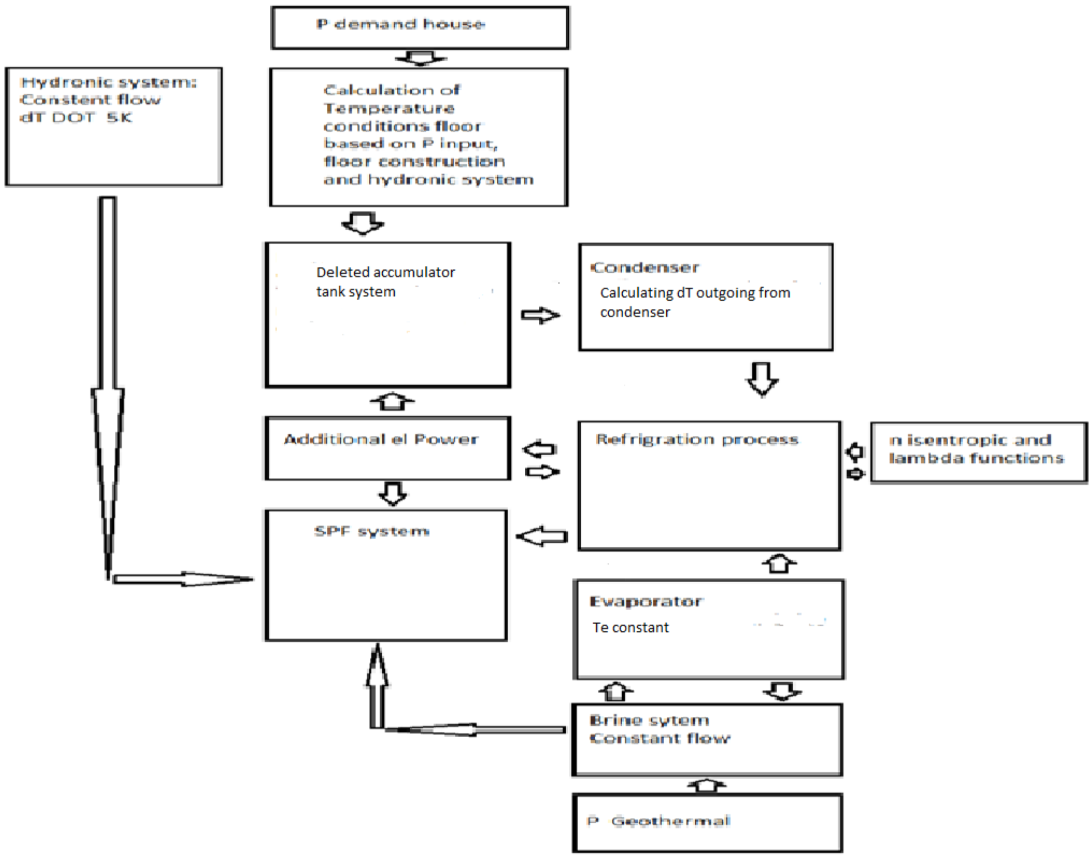

Fig 1.1 Block diagram showing relations between modules in program-

The method above has been revised and some changes have had to be made along the process. The initial idea was to implement a storage- or buffer tank for the heat pump to fill with temperature according to demand temperature from the floor hydronic system. This method implied a calculation at small time intervals in order to determine when the tank needed filling or heat pump needed to stop. This method had to be changed because of insufficient computer capacity to perform the program calculations 8760 hours by a few second intervals. The alternative method used was to calculate one hour individually and sum the results.

The other initial idea that was changed was the calculation of condensing temperature and evaporating temperature. The initial idea was to calculate these based on capacities from input data. The plan was to model the heat exchangers and use temperatures and capacities from the refrigeration model as input data.

For me this turned out to be impossible to model in EES when the condenser capacity didn't reflect the demand capacity from input parameters (as when Heat Pump had intermittent operation) then the Condenser capacity and LMTD was unknown and the Tcondenser remained unknown. The same problem was experienced trying to calculate Tevaporater. The Pevaporator was unknown when Tevaporator was unknown and opposite way round.

The best solution of trying to determine the Tc and Te was to make some simple functions that described the temperature changes due to capacity and temperature changes of the hydronic system. These are explained I respective chapters.

System boundaries of model.

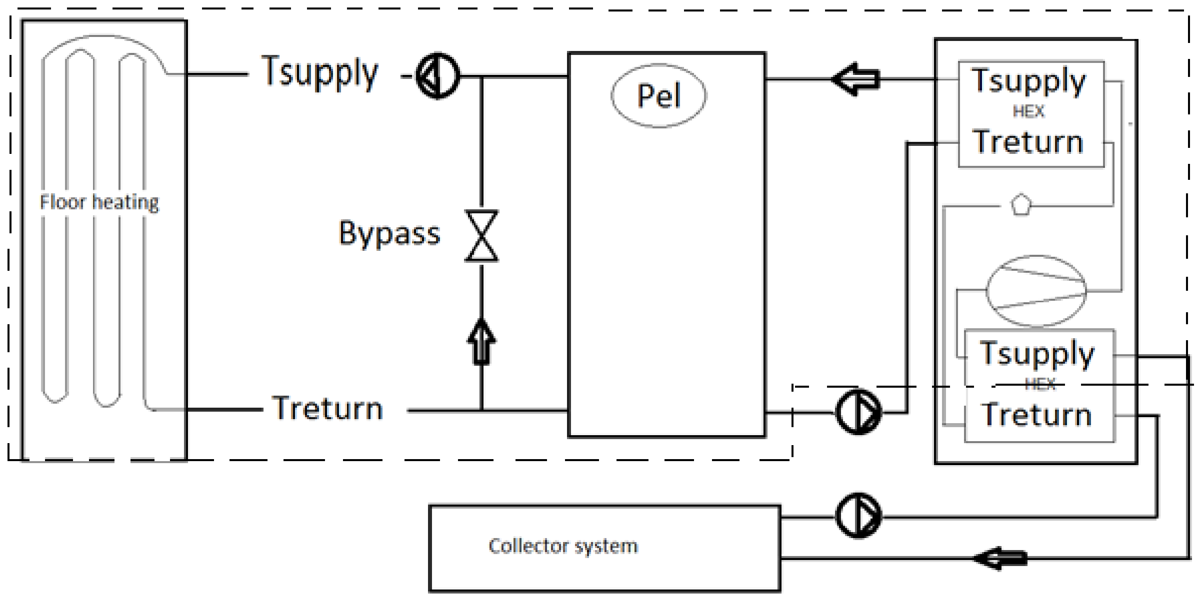

Fig 1.2..Heating system as modeled. The by-pass was taken out of the model when not calculating time fill and time charge for buffer tank. The Inverter controlled heat pump was designed for variations around 50% capacity and was not equipped with el-heater.

Floor heating

Input data was the power P needed to obtain room temperature in house at set point. Area for floor heating was reduced with a factor because of areas not covered with piping in the model such as stairs, kitchen cupboards and inside walls. Heat capacity per m^2 was found:

$$\Phi = \frac{\Phi input}{Anet} house \qquad (\text{kW/m}^2) \qquad [2.1]$$

An empiric equation were used to find temperature on floor depending on room temperature: The capacity from 2.1 was fed into this one

$$\Phi = 8{,}92 * dT^{41{,}1} \qquad (\text{W}) \qquad [2.2]$$

Were dT is temperature difference between air and floor. Room temperature in this model corresponds with the one of the house model. 21°C

$$Tfloor = Troom + dT \qquad (\text{C}) \qquad [2.3]$$

The temperature on top of the floor together with the power demand was used to calculate the temperature differences through the floor construction to the water temperature needed. For the floor boards, membrane and aluminum heat distributor equation 2.2 were used. It was assumed that Al distributors covered the same area as floor heating. .Aluminum distributors' k-value was given by Uponor/Wirsbo to 220 W/mK

$$dT = 1m2 * \Phi * \frac{k}{s} \qquad (\text{C}) \qquad [2.4]$$

For calculating temperature increase through PEX pipe an equation for radiant heat loss in cylinders were used.

Meter pipe/m^2 net floor was chosen to 4 meter so the capacity for 1 meter pipe was then 10W/m which was used in equation 2.5

$$\Phi = \pi / \frac{T1 - T2}{\frac{1}{\alpha * D_i} + \frac{1}{2 * \lambda} * \ln\left(\frac{D_O}{D_i}\right) + 1/(\alpha * D_O)} \qquad (\text{W}) \qquad [2.5]$$

Di is internal radius and Do external radius
T_1 is internal temperature and T_1 is external temperature. PEX pipe of Ø17mm was used. U value for PolypropyleneX PEX pipe was given by Uponor/Wirsbo to 0.35 W/mK

Some simplifications were made, the Al distributers were assumed to conduct temperature without loss angular to the heat flux in order to distribute evenly upwards from all of the area calculated.

Now the temperature demand dependent on capacity for heating the house had been found.

The flow of hydronic system was fixed and the dT_{water} was then made to change due to increase or decrease of the power supply/demand.
Temperature difference water at DOT were decided to 5K and mass flow water were calculated based on this.

$$\Phi = \dot{m} * cp * dT \qquad \text{(kW)} \qquad [26]$$

Temperature difference water was then split in two on each side of the mean water temperature for Tsupply and Treturn found in 2.5 like this:

$$Tsupply = Tmean + 0{,}5 * \Phi demand/\dot{m}water * cp \quad \text{(C)} \qquad [2.7]$$

$$Treturn = Tsuply - \Phi demand/\dot{m}water * cp \qquad \text{(C)} \qquad [2.8]$$

The volume flow of water were calculated by using ρ_w for 35°C at 994kg/m^3.

Tsupply - Tcondensing

For intermittent operation of refrigeration process there is a need for storage of energy used when compressor rests. In order to increase storage capabilities at lower supply temperature demand when power factor was relative high the tank temperature was set to 1.44K (varies depending on HP capacity and demand capacity) above mean water temperature at heat pump design point. In this way the compressor got longer rest time than with temperature tank at supply temperature line.

The on/off pump was modeled to deliver water of Ttank level up to hp design point. Below this point the heat pump has higher capacity due to lower pressure ratio of the compressor. Above design point the capacity becomes less than design capacity and decreases. Above design point a function was made in order to make the supply temperature from the heat pump follow the supply temperature for hydronic system. This was done to reflect the decreasing capacity delivered by the heat pump because of increasing return temperature for hydronic system.

The LMTD of the condenser was found by making a model of a heat exchanger and using the numbers from there in the LMTD equation like this

$$\Phi_o = U_e * A_e * LMTD \qquad \text{(W)} \qquad [2.9]$$

The numbers used for the heat exchanger model was A=0.85m^2, α_water =1800W/m^{2}K, α_refrigerant=2500 W/m^{2}K, α, λ=55 and δ=0.0015m in

$$U = \frac{1}{\alpha_w} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{ref}} \qquad \text{(W/mK)} \qquad [2.10]$$

When this method didn't work the function below was for On/Off HP made in order to reflect the change in dT between outgoing water from condenser and the condensing refrigerant. The capacity from the On/Off HP was increasing when decrease in supply temperature demand. This will be reflected in the LMTD and the dT on the supply water and refrigerant. The dT out was chosen to have less span than the LMTD. The Twater which is the mean water temperature was chosen as basis of the function. A temperature dependent on dT at HP design point where added on as the minimum dT at minimum capacity.

$$Tc = Twater + 1{,}44 + 2 \qquad \text{(C)} \qquad [2.11]$$

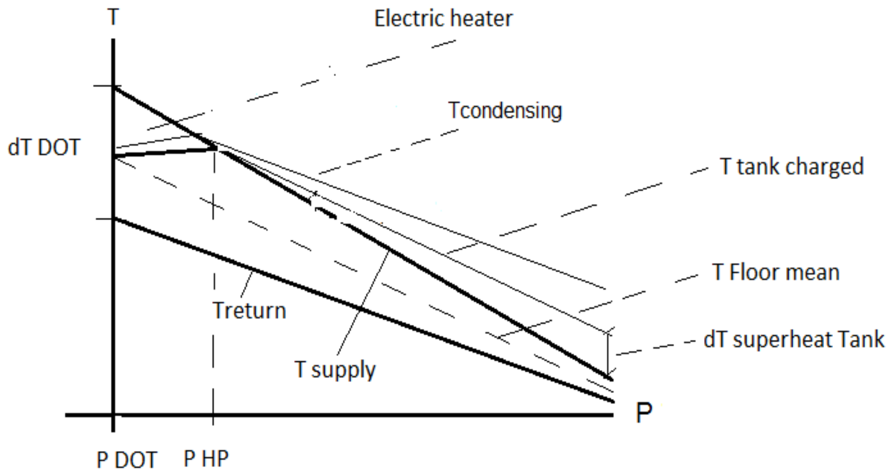

Fig 1.3. Diagram showing relations between Tmean, Tsupply,Ttank with superheat and Tcondensing (LMTD =Tcondensing-Ttank).

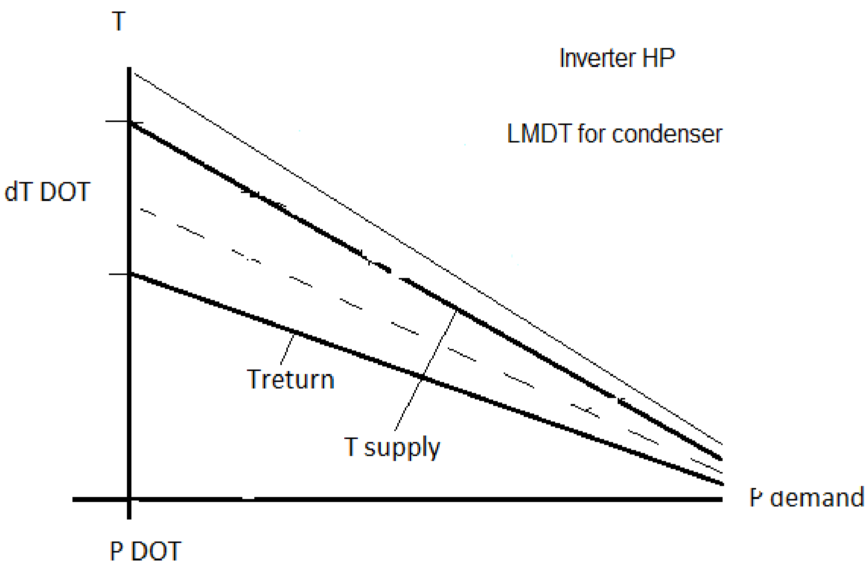

Fig 1.4. Tcondensing and Tsupply reflects the LMDT between the two as increasing capacity when increasing temperature.

The function used for determining Tc in Inverter heat pump was one that increased the difference (only slightly because higher flow rate in condenser makes for a lower LMTD) by increased supply temperature to the floor. Here it wasn't any peak capacity – the limitations regarding capacity were the compressor rotation and tp of the compressor.

The equation (3.1) is substituting the LMTD calculation in order to reflect the temperature difference at the output of the condenser for the inverter HP. The dT between water and refrigerant is increasing when increase in capacity delivered from the HP.

$$Tc = Ttank^{1.015} \qquad (C) \qquad [2.12]$$

Energy - Capacity ratio

Here the heat pump was chosen for a capacity of 5kW at Tc 37,5°C and Te at -2°C which are the conditions for giving off this power according to the Simien program where the hose were modeled in..

P DOT house in Kongsberg Norway from Simien model was 8,6 kW

Here P heat pump is equal to less than 60% of P_{DOT}. This energy production covers about 80% of Q_{house} with electricity input of about 20% of that. This was estimated by simplifying the curve for day-degrees at Kongsberg.

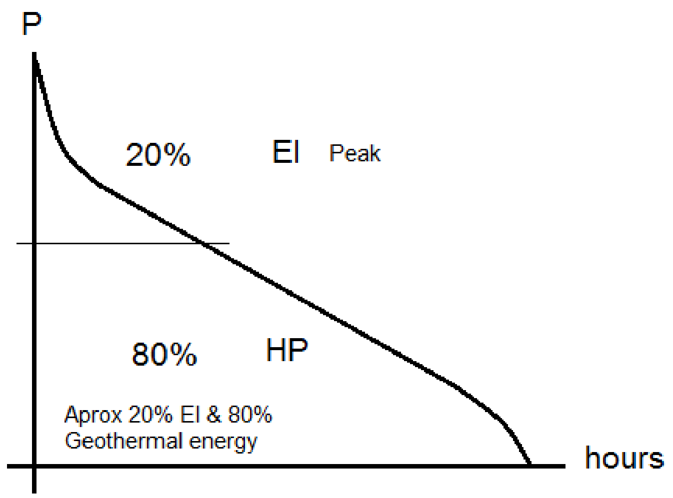

Fig 1.5. Sketch of energy ratio from El and Hp at 57% heat pump capacity.

After determining the capacity of the HP Q_{tank} was found with

$$Q = \Phi HP * t/3600 \qquad (kWh) \qquad [2.13]$$

Where *t* is in seconds was 12min by 60 sec/min, 720s.

HP design capacity was calculated as a function of energy/power ratio. An integral of the energy demand during heating season was made and estimation on Power HP versus P DOT were made

The equation used for finding net volume tank is:

$$Vtank = \frac{Qtank * 3600}{((dT_{floor}) * cp)} \qquad (m^3) \qquad [2.14]$$

Floor dT at heat pump design condition were 2,87K. (1.44K above mean temperature which were used to calculate Ttank level and by that Tsupply from heat pump at conditions below HP design.

<u>Time charging and discharging buffer tank</u>

This method was changed as mentioned in the beginning of the modeling section of the report. Jump to heat pump process.

Time for charging the tank starts when bottom temp is equal to Treturn from floor hydronic system. At this condition the HP has a power output and the floor system has a power input. Between zero P demand and HP Pdesign the difference of these two will be the power for charging the tank. Like this:

$$t\ fill = (Vtank * cp * \rho * (Ttank - Treturn))/(\Phi cond - (Phouse) \quad (s) \quad [2.15]$$

(*t* from here is used for calculating Q_{input} in SPF)

When charging tank with Twater (mean temperature) + 0,5 x dTwater floor HPdesign the run- and rest time compressor will increase and thereby less losses for startups. But Tcondensing will be higher and then lower the COP.

Discharging or emptying of tank with superheat involves mixing Treturn with Tsupply floor for conditions with lower capacity demand than HPdesign. The equation used for this is:

$$t_{empty} = Vtank/(V^{\cdot}{}_{water_{floor}} - V^{\cdot}water_{bypass}) \quad \text{(s)} \qquad [2.16]$$

Volume flow bypass was calculated like this:

$$V^{\cdot}water\_bypass = Tsupply_{floor} * V^{\cdot}floor/Ttank \quad (\text{m}^3/\text{s}) \qquad [2.17]$$

Fig xx Mean floor temperatures and T buffer tank when charged.

Heat Pump process

Efficiencies' compressor

Isentropic efficiency and λ compressor was made into functions according to "German rule" based on the pressure ratio $\frac{Pc}{Pe}$ and a typically R22 compressor efficiency diagram. (24)
The functions adjust the compressor performance according to pressure ratio Pc/Pe. The "German Rule" is basically to make a straight line out of the section of the diagram that is needed.

$$\lambda = 0.065 * \frac{P_c}{P_e} + 0.9 \qquad (-) \qquad [2.18]$$

$$\eta_{is} = 0.06 * \frac{P_c}{P_e} + 0.9 \qquad (-) \qquad [2.19]$$

These two efficiencies were used for both types of heat pumps.

Efficiency start compressor

After a "rest" period the refrigeration cycle has leveled with the Phigh and Plow at the same level. When starting a cycle of producing heat it takes time before Phigh has reached Tcondenser and the HP is producing heat. The time estimated to reach production for the on/off pump was made to

$$t = 6 + \frac{P_c}{P_e} * 2 \qquad \text{(s)} \qquad [2.20]$$

which is added on to each cycle in the SPF calculations where the Pcondeser capacity is lower than Pdemand. Otherwise the compressor is set to run for the whole hour. Ref Karlson p 23). The time is chosen to be about half of the real time since the model calculates full power for el motor from start while in field the compressor would start at a low load and increase up to full power. For the Inverter pump the extra time were neglected due to very few stops and starts since the compressor capacity were adjusted according to load.

Refrigeration cycle

The chosen refrigerant is R-290
A reciprocating compressor was chosen.

On/off

Input data for HP process is Tcondensing temperature level calculated previously in model. Tcondensing reflects the needed temperature for transferring the power through to room. Volume flow compressor swept is fixed but induced volume varies due to λ from Pc/Pe ratio. At HPdesign the capacity reflects the demand in power and temperature. At lower demands the heat pump has higher capacity and need the buffer tank to cycle. At larger demands the heat pump has not sufficient capacity and needs additional power from El heater.

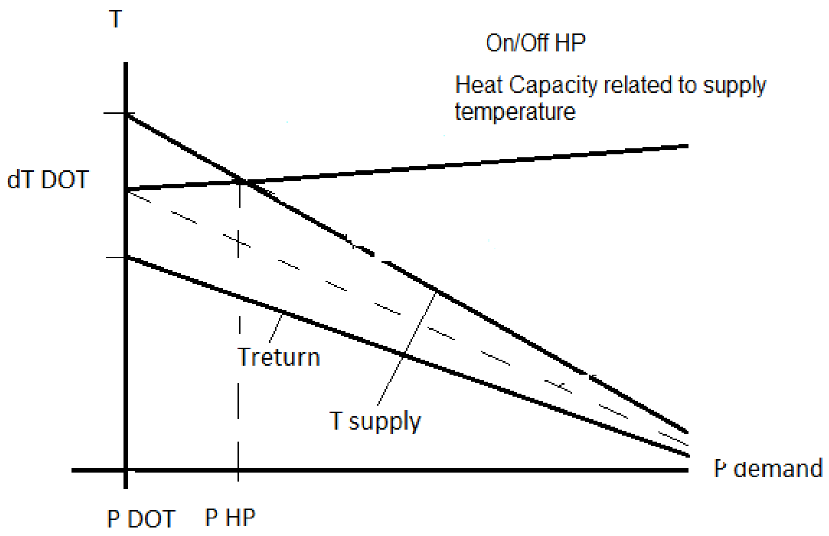

Fig 1.6.. Capacity on/off heat pump dependent on supply temperature level.

The El power for additional heater was modeled as the difference between demand and Pcondenser.

Inverter

Input data for Inverter HP process is P demand house at Tcondensing temperature level calculated. Tcondensing reflects the needed temperature for transferring the power through to room. . Volume flow compressor swept is variable between upper and lower limit. Here upper limit is set to P house at DOT in order to cover all of the heat demand without additional heater. Below lower limit demands the heat pump has higher capacity and need the buffer tank for intermittent operation. This may be optional and the best solution to operate the compressor down to zero load in this model. The LMDT

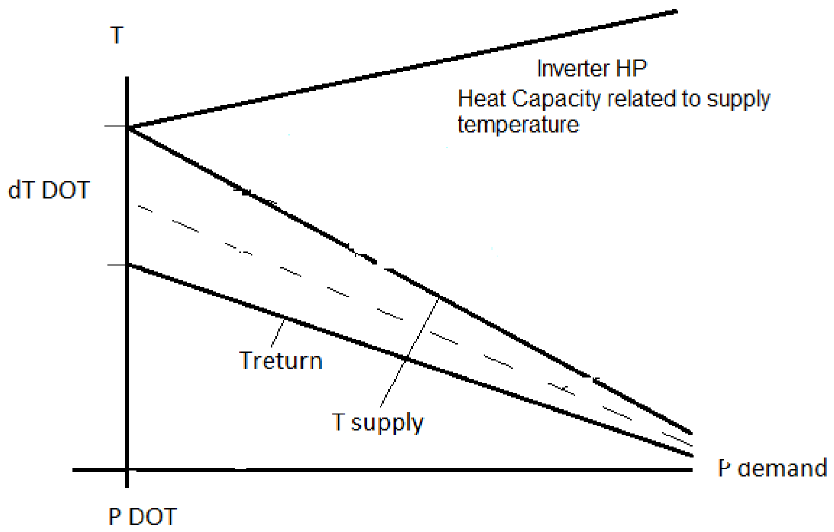

Fig 1.7. Inverter HP capacity dependent on supply temperature level.

Tevaporating is the resulting temperature level after calculating temperature difference at transferred capacity from rock to evaporating refrigerant with the given evaporator capacity. This calculation is performed with Trock at a constant temperature. Tevaporating changes due to needed capacity to extract from the rock if constant flow on brine or it is constant with flow brine as the vaieable. It was used two different equations in order to find the dT between rock temperature and refrigerant. The LMTD-equation was used for dTm through condenser and equation on transferred power through cylindrical objects to determine the dT from rock to brine. It was only modeled on a single collector pipe in bore hole in the collector model. The bore hole was looked at as an inner pipe with brine as flowing fluid and with heat transfer coefficient. The bore hole filled with water was looked at as a solid without heat transfer coefficients.

$$\Phi_o = U_e * A_e * LMTD \qquad (K) \qquad [2.21]$$

The real problem occured when connecting these models together with the refrigeration model. This was found for me to be impossible at given circumstances and a function related to the supply temperature was implemented to reflect the change in evaporator temperature according to change in heat pump capacity. This made for many adjustments according to variations in evaporator capacity. The final solution in order to have a degree of similar conditions was to set the evaporator temperature to a fixed Te. Variable brine flow with constant dT was modeled on a set of calculations to see the impact this had on the SPF comparison to fixed energy consumption and fixed Te.

Superheat and sub cooling has been chosen to 5 and 3K. The pressure difference inside the refrigeration circuit and thereby the power for moving the refrigerant around is not incorporated in the model here.

Defining refrigeration process.

The refrigerant cycle is defined by 7 conditions.

- Sate 1 is suction into compressor after 5K superheat
- State 2 is discharge out of compressor
- State 3 is x=1(vapor) in condenser
- State 4 is x=0(liquid) in condenser
- State5 is sub cooling by 3K and start of expansion
- State 6 is after expansion and inlet to evaporator
- State 7 is x=1(vapor) in evaporator

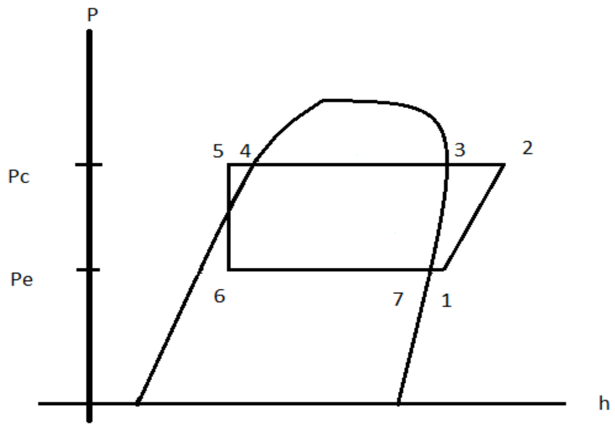

Fig 1.8. Logarithmic P-h diagram with defined states for refrigeration cycle

The EES program has the refrigerant states incorporated. The inputs are condensing and evaporating temperatures, super heat and sub cooling, and efficiency of compressor. The delivery- and isentropic efficiencies were here a function of the pressure ratio Pc/Pe. The 7 states defined here are cross references from known states and builds up to define the whole process.
When the cycle is defined the capacity is chosen by the mass flow of refrigerant in the system.

On/off
HP design power for house in Norway was calculated for 5kW. Volume compressor cylinder was fixed for given mass flow at the design point 5kW with related supply temperature for hydronic system and corresponding evaporator temperature. Because of the λ-delivery coefficient the compressor will have higher capacity at smaller temperature lift and opposite.

Inverter.
The same compressor volume was used for the inverter compressor. (Variations on this was also modeled to compare results. The mass flow refrigerant by enthalpy difference was made to be equal the heat capacity needed for the hydronic system at any condition. At low capacity the Tc was made constant in some models.

$$(T_{tank} - T_{return}) * m^{\cdot}_{w} * cw = \mathrm{m}^{\cdot}\_\mathrm{ref} * (\mathrm{h2} - \mathrm{h5}) \qquad \text{(W)} \qquad [2.22]$$

The rpm and frequency were calculated by the needed swept volume flow of the compressor correspondingly.

Additional electric heater for Tsupply

When power demand for house exceeds HP capacity an electric heater is adding capacity in order to supply the temperature according to Tsupply set point. The supply and return temperature curve can be illustrated like this based on floor hydronic system calculations:

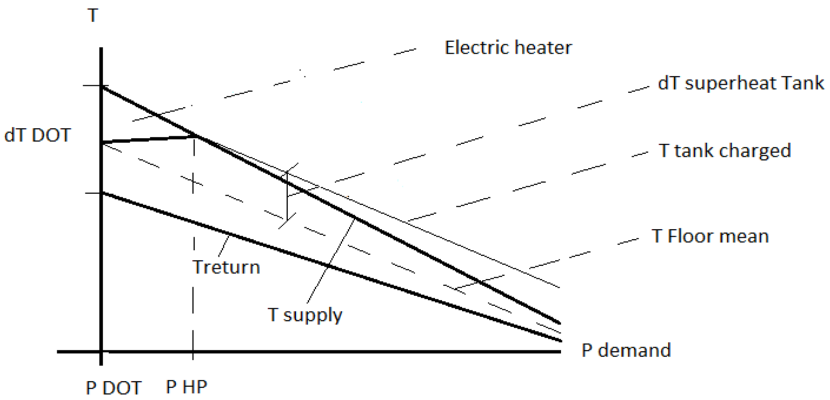

Fig 1.9. Supply and return temperature diagram with on/off HP design showing additional el heater.

When the HP reaches design conditions the El heater will take Tsupply up to needed value. At the same time the Treturn from hydronic system becomes higher than HPdesign and HP power is decreasing as shown in declining curve at point Tsupply and P HP.

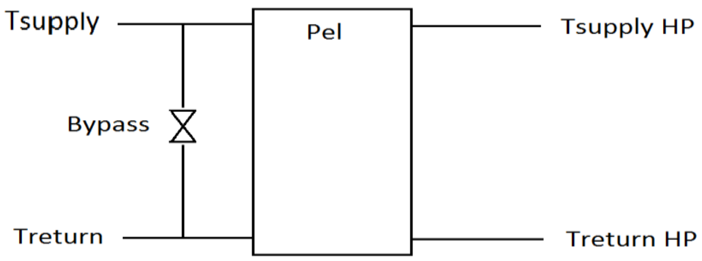

Fig 1.10. Sketch of buffer tank system with by-pass and additional el Power.

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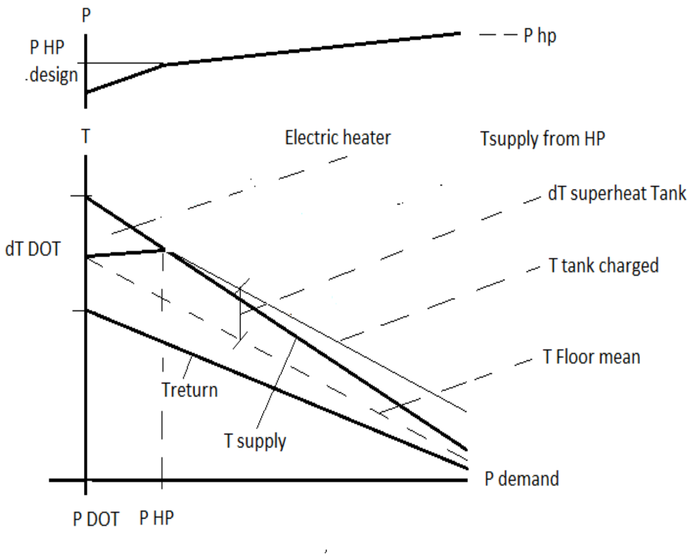

Fig 1.11. Capacity of heat pump relative to supply temperature.

The on/off heat pump will have decreasing heating capacity as the supply temperature increases as shown in fig 1.12.

Evaporator

The evaporator was modeled with an Area of 0,55 m^2, α_{brine} chosen to 2000W/m^2K and α_{ref} to 2000 W/m^2K. These numbers were chosen from DANVAK (23)
Wall heat exchanger was chosen to 15mm and heat conduction to 55 W/mK for steel.

$$LMTD = \frac{1}{\alpha_w} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{ref}} \qquad \text{(C)} \qquad [2.23]$$

$$\Phi_o = U_e * A_e * LMTD \qquad \text{(W)} \qquad [2.24]$$

Collector

Collector system was modeled with PE40 pipe of 200 meter length and inner diameter of 38,5 mm. The k-value of the PE was chosen to 0.4. A solution of water/ethyl glycol 70/30 was chosen for making brine. The $c_{p\ was}$ found to be 3,4kJ/kgK (24). Viscosity was chosen to be 5 mPas for temperatures between 0 and 10 below.

After not succeeding with the constant flow model the temperature difference of the brine was set to 4K and mass flow brine was calculated from evaporator capacity at estimated -2°C as design.

The supply and return brine temperature was defined by using the mean temperature and dividing the dTbrine on each side.

When the programming of this didn't succeed the Te was set to -2 °C. The bore hole equation below was also not used other than to check the temperature differences available when setting the Te.

Bore hole

Temperature difference rock – brine is modeled with basis in equation for cylinder. It is presumed an energy flux from rock same as hp model requires for evaporator. The diameter on the rock was chosen to 0.17 meter and temperature rock is chosen to 8°C.

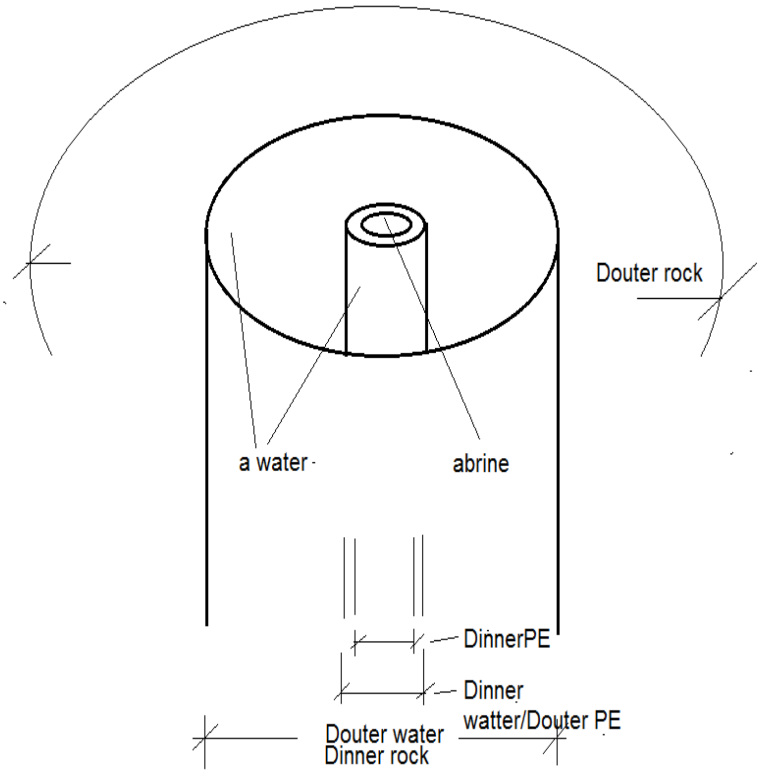

Fig 1.12. Sketch of bore hole as modeled.

The equation for Usystem collector value – it was not used heat transfer coefficients on either sides of the water inside the hole:

$$\mathrm{U_{brinesys}} = \pi\left(\frac{1}{\mathrm{a_{brine1}*D_{brine}}} + \frac{1}{2*\lambda_{PE}} * \ln\left(\frac{D_{PEo}}{D_{PEi}}\right) + \frac{1}{2*\lambda_{water}} * \ln\left(\frac{D_{PEo}}{D\_{Ri}}\right) + \frac{1}{2*\lambda_r} * \ln\left(\frac{D_{Ro}}{D_{Ri}}\right)\right.$$

(W/mK) [2.25]

$$\Phi_{collector} = U_{collector} * l_{pipe} * dT \qquad \text{(kWh)} \qquad [2.26]$$

9.2 Model input data.

For Norwegian conditions a two story theoretical house was modeled in Simien and situated at Kongsberg. Energy consumption for heat demand building was 18342 kWh total a year. Heated floor area was 196 m^2. Mean temperature year was 4,7 °C and DOT was-21°C. No ventilation modeled. Two different user profiles where added (subtracted) to the house model in order to investigate the differences this had on the heat pump operation. The Simien calculated power consumption for the house on an hourly basis.

A theoretical hourly user profile for habitants of the house was made based on heat loads pr person and heat loads for equipment. It was made two profiles, one for middle of the week and one for weekends. These profiles where then subtracted from to the consumption data of the house and a net power consumption where found. These data where fed into the model for calculation of the SPF.

For house in Denmark data from a generic model was used. These data were also on an hourly basis. The data came from HUSE 6172. Case 1 and Case 140.

9.3 Model of inverter controlled Heat Pump.

This chapter explains the method used to calculate the SPF for an inverter HP based on the same input data as previous chapter. HP compressor volume flow design is the same as for the on/off HP. Frequency range as for an air/air HP with approximately same power. (27)

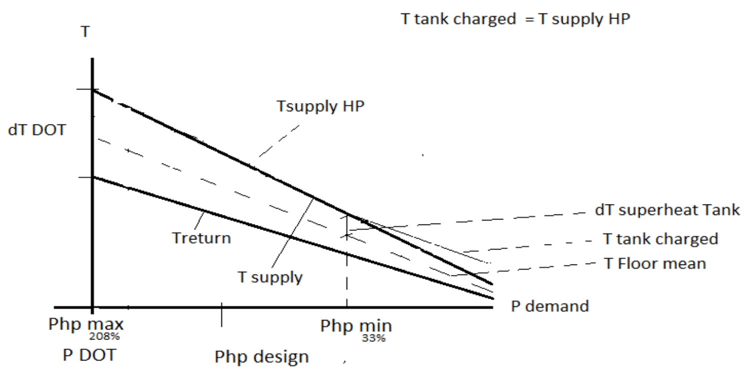

Fig 1.13. Supply and return temperature diagram with inverter HP design.

The range was based on 100% capacity at 50Hz and HP design. At DOT conditions the compressor would have doubled the capacity and rpm would be more in order to overcome losses in compressor. No additional electric heater was modeled together with inverter HP's.

$$V^{\cdot}compressor = \frac{pi * d^2 * s * z * n}{4 * 60} \quad (m^3/s) \qquad [2.27]$$

Were

D is diameter cylinder

S is length of piston movement

Z is number of cylinders

N numbers of cycles per minute

$$Rpm = \frac{120 * f}{p} \quad (r/min) \qquad [2.28]$$

Were

f is frequency

p = number of poles in el motor

10 Results from Models

The models where grouped in two, one for the Norwegian models and one for the Danish models. The input data (hourly capacities and indoor temperature of 21°C in order to calculate temperature resistance) where fed into the EES model and chosen output data from equations made in the program was transferred to an XL sheet for making charts. Each model calculation took around 3 minutes to perform.

10.1 Huse 6172 Case 1

This house/profile was located in Denmark and had a maximum heating demand capacity of 7,5 kW at DOT and total energy consumption a year of 17.276,5 kWh. The figure below shows collected data on 6870 hours basis. The data was sorted on the criteria of highest capacity first and then decreasing numbers. The SPF was calculated with Brine Pump only on energy input side. Additional heater, Hydronic pump and Compressor were used on both sides.

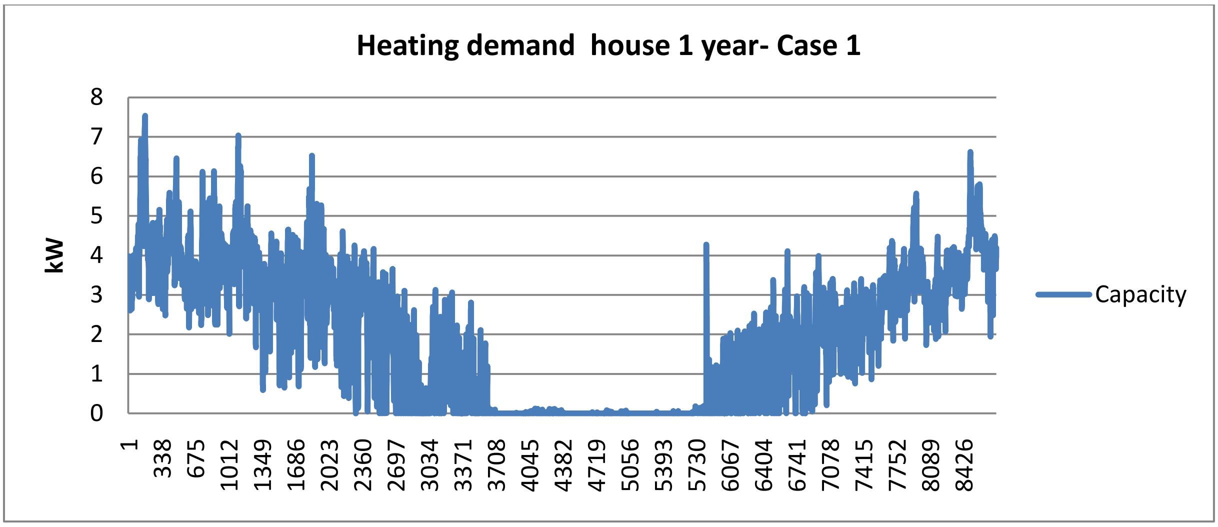

Fig 2.1. Collected data on heating capacities from Case 1

To analyze performance a number of varieties in heat pump sizes were made in order to extract possible patterns.

	Size Capacity HP of House	Design Capacity	Design Capacity HP	Vh compressor	Q Reneable and elHP kWh
Case 1	50%	7,532	3,766	0,02288	12847-4529
Case 1	70%	7,532	5,27	0,0275	13624-3790
Case 1	100%	7,532	7,532	0,0339	13592-3869
Case 1	Inverter	7,531	3,766	0,02192	14002-3330

Table 2.1. Capacities of Heat Pumps selected.

The Inverter Heat Pump was selected with the same compressor volume as the On/Off one for 50% capacity. The On/Off Heat Pumps capacities were sized at their design point. (For instance the 50% Heat pump compressor capacity was sized at 50% of house heat demand and corresponding water temperatures and Tc).

10.1.1 Case 1 50% On/Off Heat Pump

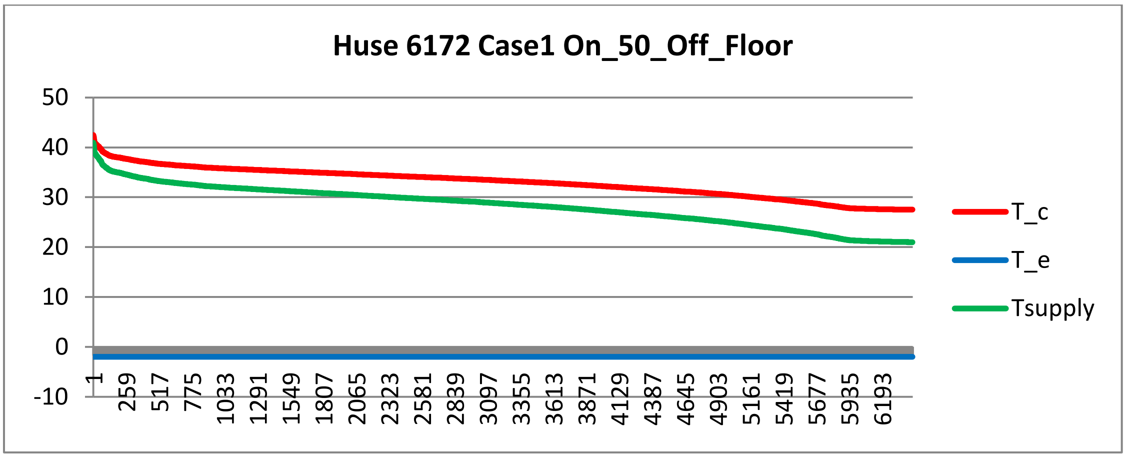

Fig 2.2. Heat pump that covers 50% demand. Temperatures.

In fig xx the temperatures for evaporating, condensing and supply water are shown. The increasing temperature difference between supply and condensing reflects the increase in capacity of the Heat Pump at decreasing temperature differences between heat source and heat sink. At minimum capacity the effect of the fixed temperature can be seen.

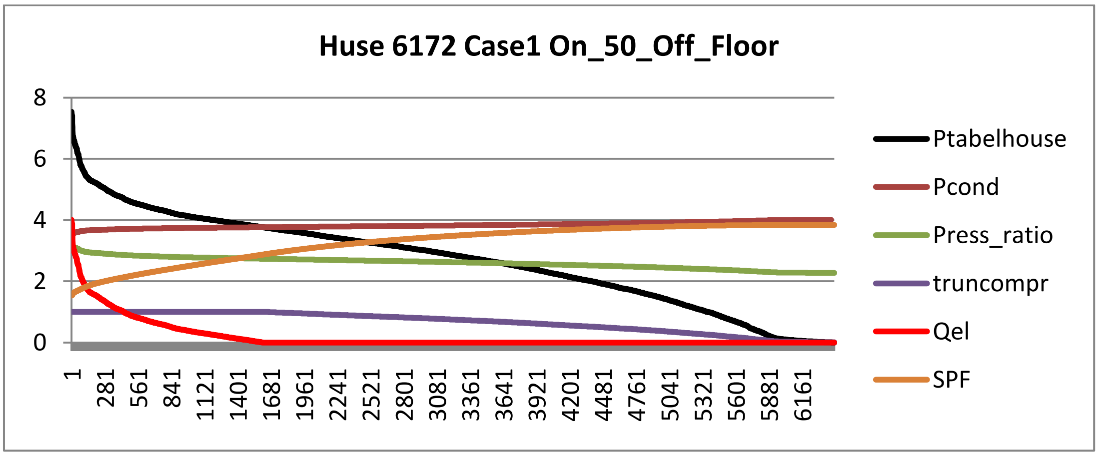

Fig 2.3. Heat pump that covers 50% demand.

In figure xx relations between capacities and ratios are shown. (Qel should be Pel). Here can be seen that the condenser capacity crosses the demand capacity (Ptabelhouse from input data) at 50% of Ptabelhouse maximum capacity. At higher demand the additional el heater covers the difference. Looking at the Condenser capacity from left to the right one can see that there is decreasing capacity on the whole length. This is because after additional heater supplies needed supply temperature the return temperature is still increasing and causing a higher condensing temperature and higher pressure ratio. The on-time for the compressor decrease from 1/1 to fractions of the hour depending on how much energy is needed within that hour. The reductions in on-time for the compressor are influenced both of the decrease in heat demand capacity and the increase in condenser capacity due to decrease in temperature difference between source and sink. This is seen along the whole heating period on the pressure ratio.

10.1.2 Case 1 70% On/Off Heat Pump

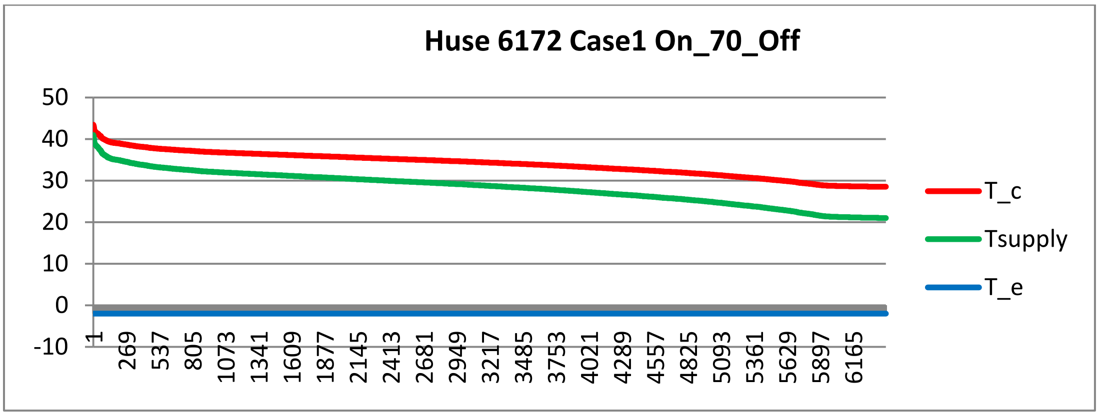

Fig 2.4. Heat pump that covers 70% demand.

In Fig shows similar pattern as for the 50% Heat Pump. The condensing temperature related to the supply temperature are here slightly increased due to higher capacity performance of the heat pump.

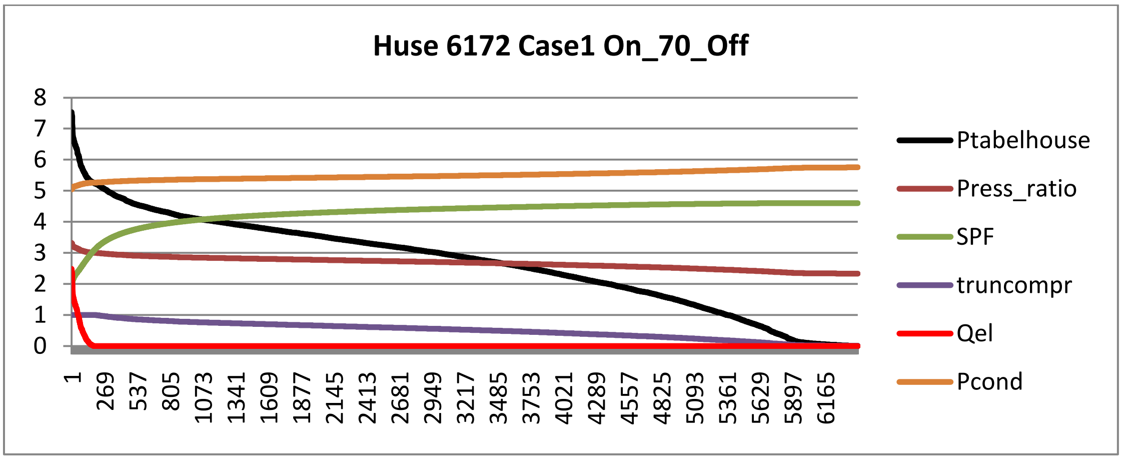

Fig 2.5. On/Off Heat pump that covers 70% demand.

Fig XX with a heat pump of 70% capacity coverage shows a higher resulting SPF than the 50% one. The compressor has intermittent operation over the whole heating season except for 179 hours

10.1.3 Case 1 100% On/Off Heat Pump

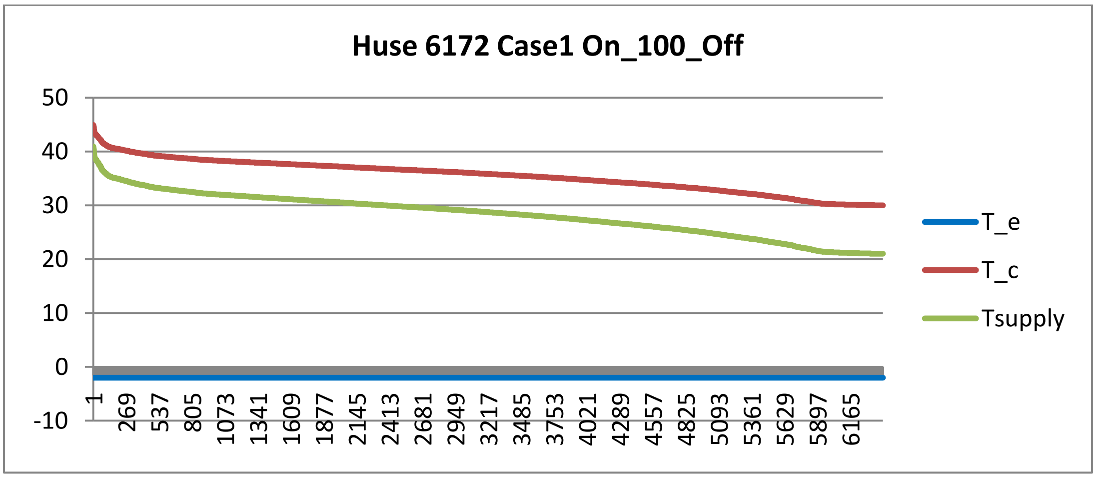

Fig 2.6. Heat pump that covers 100% demand. Temeratures.

Fig xx shows same pattern as for the previous Heat Pump models. The condensing temperature is here higher than for the 70% due to higher capacity of the Heat Pump.

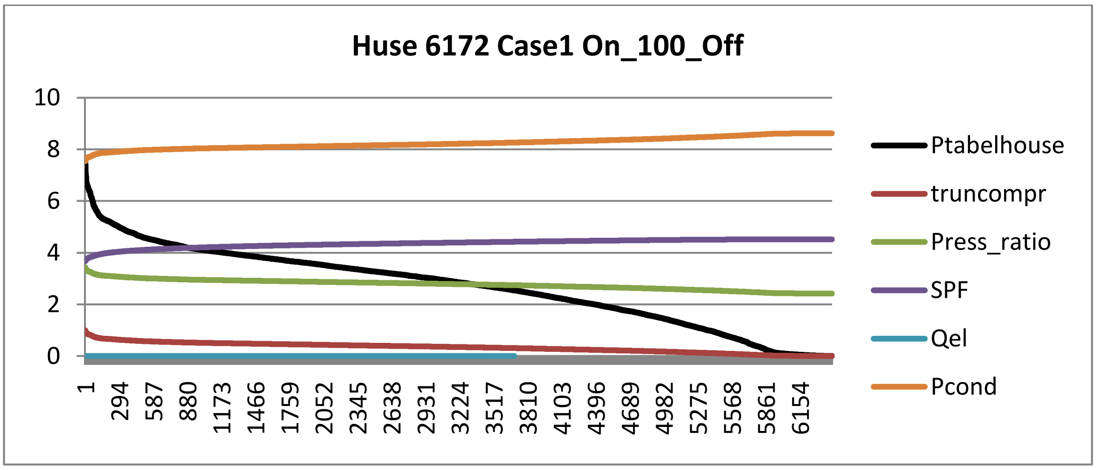

Fig 2.7. Heat pump that covers 100% demand.

The Heat Pump is here designed for 100% coverage of the demand. Because of decreasing temperature lift the capacity of the condenser is here reaching above 8,5 kW. The SPF is slightly higher than for the 70% Heat Pump. The run time for compressor is only 1/1 at the initial stage. With intermittent operation there will be more start/stop losses than for continuous operation.

10.1.4 Case 1 Inverter Heat Pump

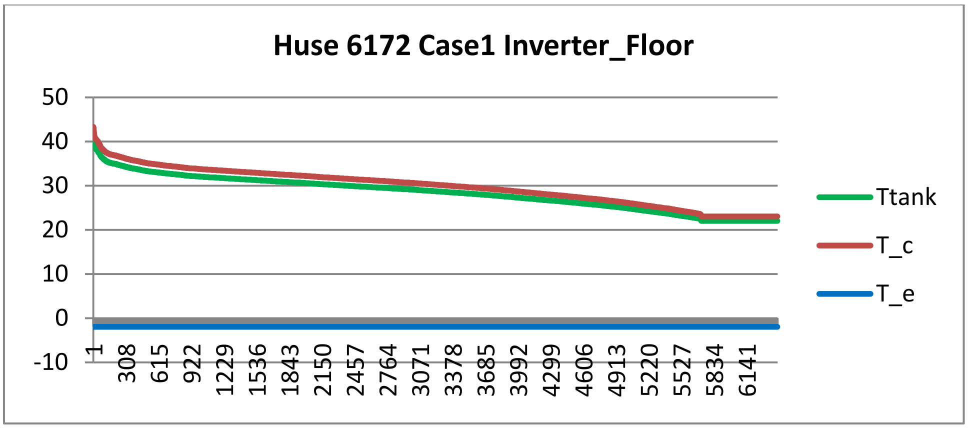

Fig 2.8. Heat pump model with frequency controller that covers 100% demand. Temperatures.

Here the temperature difference is opposite to the On/Off models because the Heat Pump capacity is following the demand there is only need for a temperature difference between condensing and water to ensure sufficient capacity transference. Here the temperature difference is increasing when increasing demand due to need of higher capacity transferred.

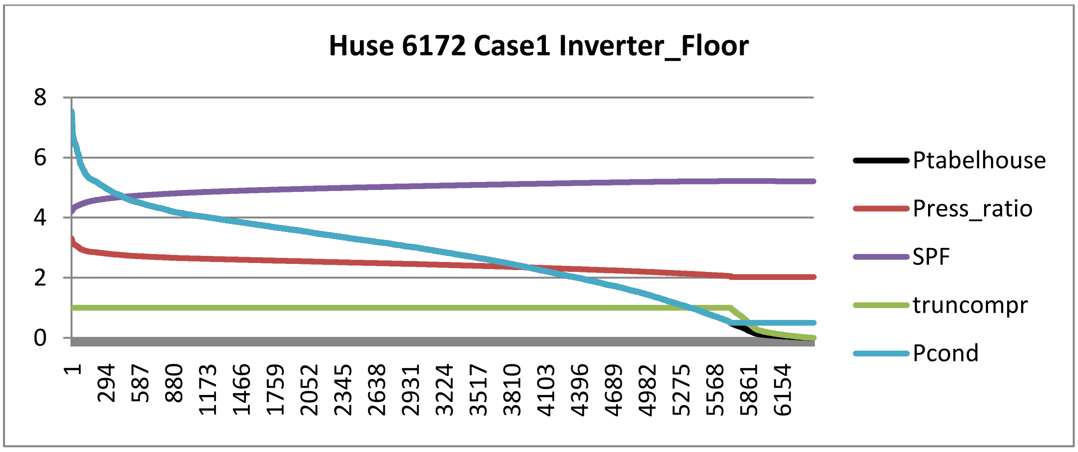

Fig 2.9. Heat pump model with frequency controller that covers 100% demand.

The demand curve (Ptabelhoues) is here similar to condenser capacity except at minimum capacity when the temperature in accumulator tank was fixed in order to ensure a minimum revolutions on the compressor. (The Ptabelhouse curve is covered by the Pcond line except at minimum capacity). Here also it is seen that the SPF relates to the Pressure ratio as an inverse function.

10.1.5 Case 1 Comparing SPF's

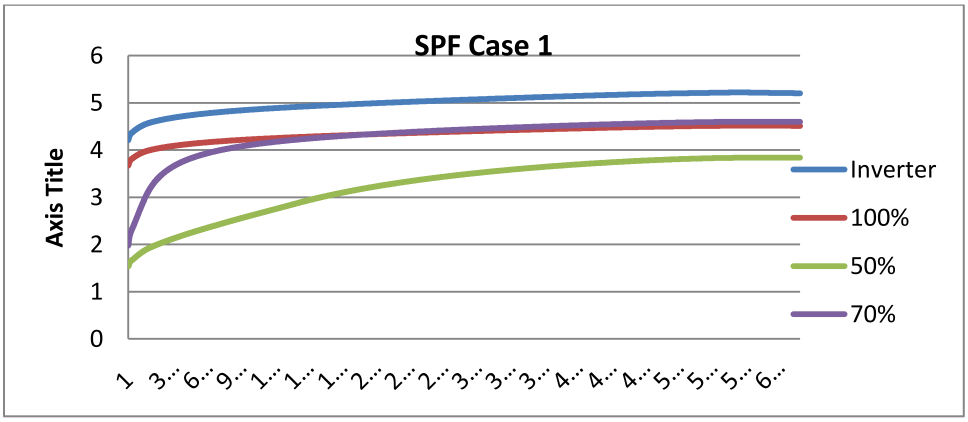

Fig 2.10. Seasonal Performance Factors from the Case1 Heat Pump models.

When comparing the SPF's in Case 1 shown in figure xx the Inverter Heat Pump has the highest SPF followed by 100% and 70%. The 70% is overtaking the 100% Heat Pump due to less start/stop losses.

10.2 Huse 6172 Case 140

This house/profile was located in Denmark and had a maximum heating demand capacity of 2,747 kW at DOT and total energy consumption a year of 2.759,2 kWh. The figure below shows collected data on 6870 hours basis. The data was sorted on the criteria of highest capacity first and then decreasing numbers. The SPF was calculated with Brine Pump only on energy input side. Additional heater, Hydronic pump and Compressor were used on both sides.

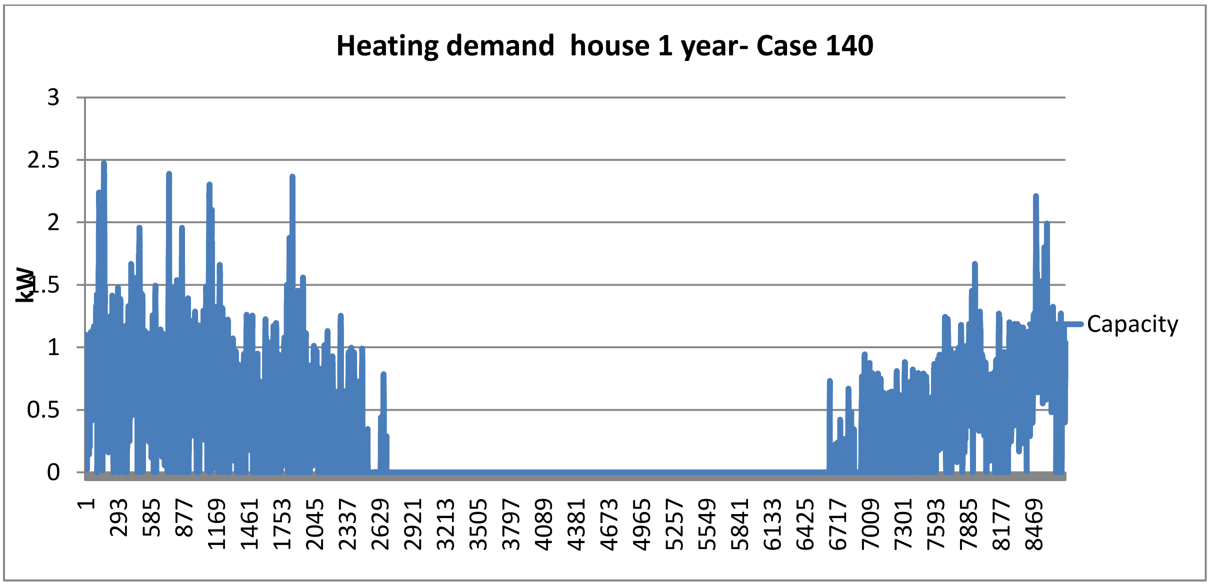

Fig 2.11. Collected data on heating capacities from Case 140

	Size Capacity HP of House	Design Capacity	Design Capacity HP	Vh compressor	Q reneable and elHP kWh
Case 140	50%	2,474	1,237	0,0129	2215-562
Case 140	100%	2,474	2,474	0,01845	2269-520
Case 140	Inverter	2,474	1,237	0,0129	2346-422

Tabel 2.2. Capacities and results of Heat Pumps selected.

10.2.1 Case 140 50% On/Off Heat Pump

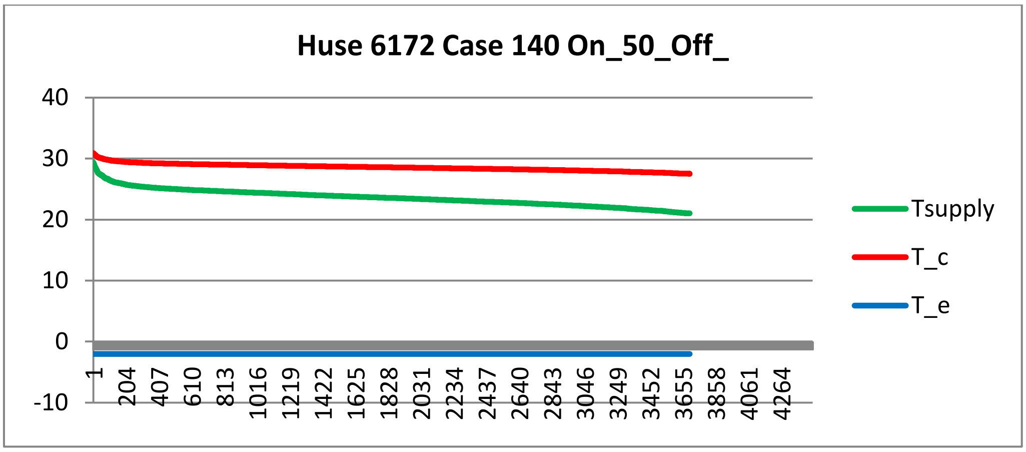

Fig 2.12. Heat pump model with On/Off HP that covers 50% demand. Temperatures.

The temperature difference between Tc and Te is increasing with decreasing demand in fig 2.12

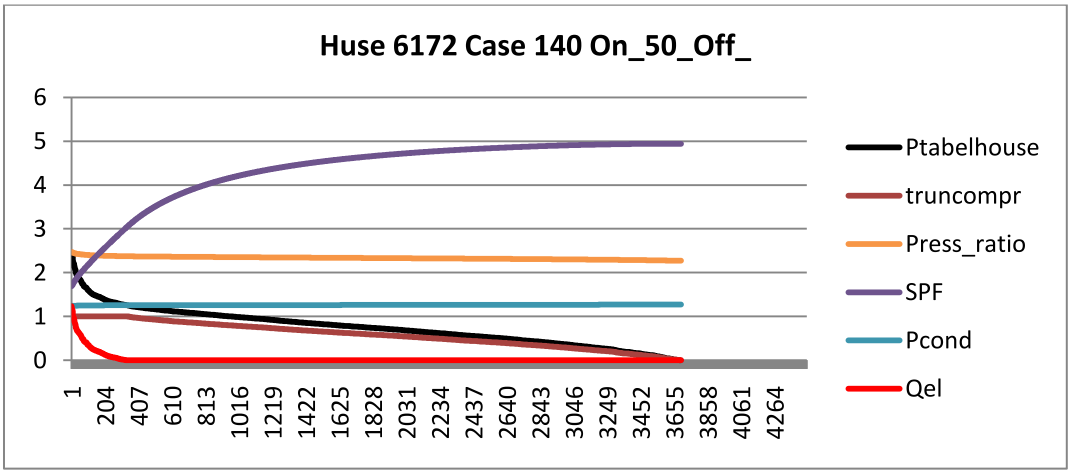

Fig 2.13. Heat pump model with On/Off HP that covers 50% demand.

At 2,47 kW DOT design capacity for hose a 50% heat pump has a low span in temperature lift and the SPF will be affected by this. Very few hours that needs additional heating capacity.

10.2.2 Case 140 100% Heat Pump

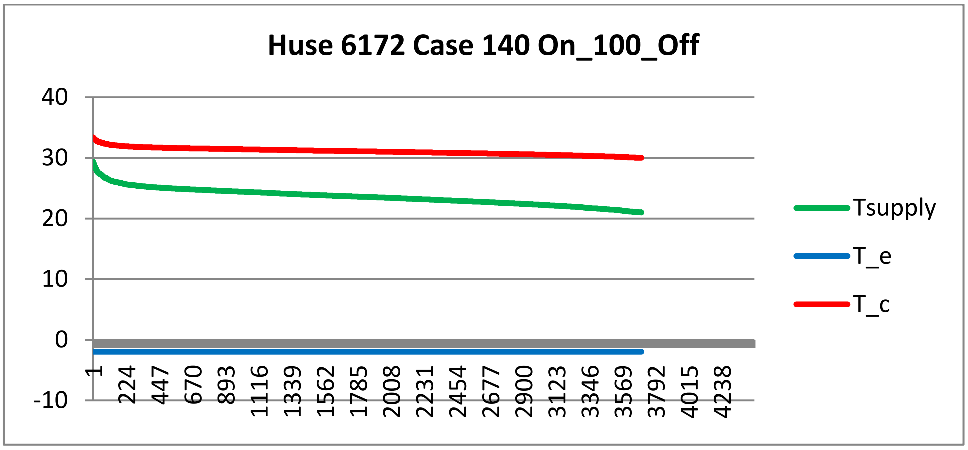

Fig 2.14. Heat pump model with frequency controller that covers 100% demand. Temperatures.

The temperature differences in Fig 2.14 between Tc and Te are for the 100% heat pump higher than for the lower capacity.

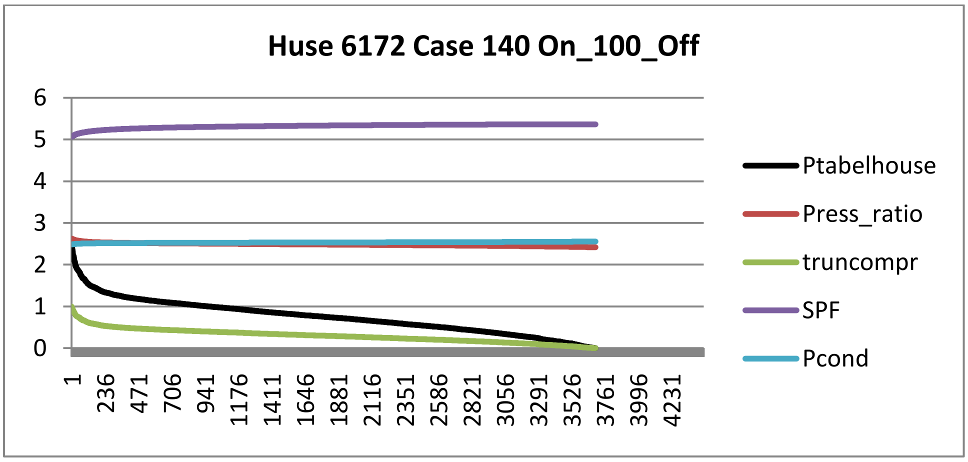

Fig 2.15. Heat pump model On/Off that covers 100% demand. Temperatures.

In Fig XX and xx it can be seen that when the condensing pressure is at about a constant level and the evaporating temperature also is at a constant level the pressure ration and SPF also stay nearly constant.

10.2.3 Case 140 Inverter Heat Pump

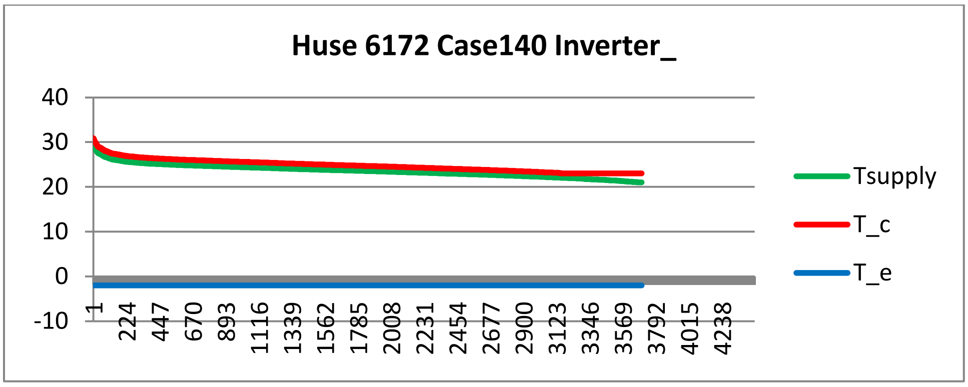

Fig 2.16. Heat pump model with Inverter that covers 100% demand. Temperatures.

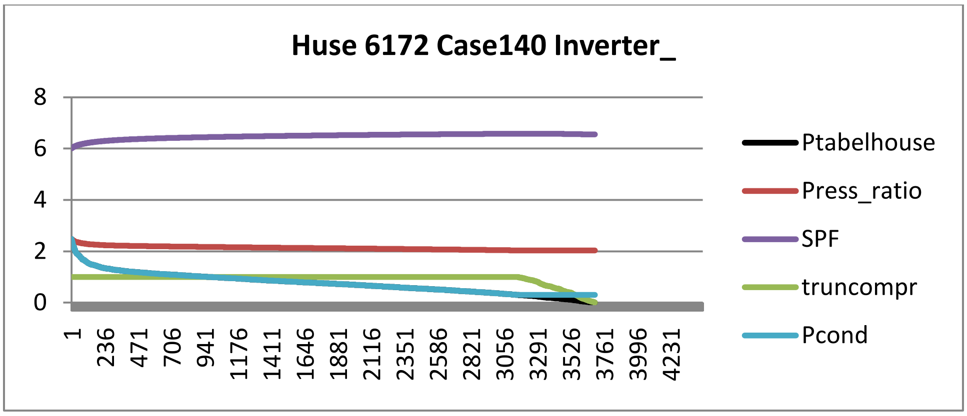

Fig 2.17. Heat pump model with Inverter that covers 100% demand.

In Fig XX the condenser capacity line is covering the input Ptablehouse except at the end were condensing capacity is kept at a minimum level.

10.2.4 Case 140 Comparing SPF's

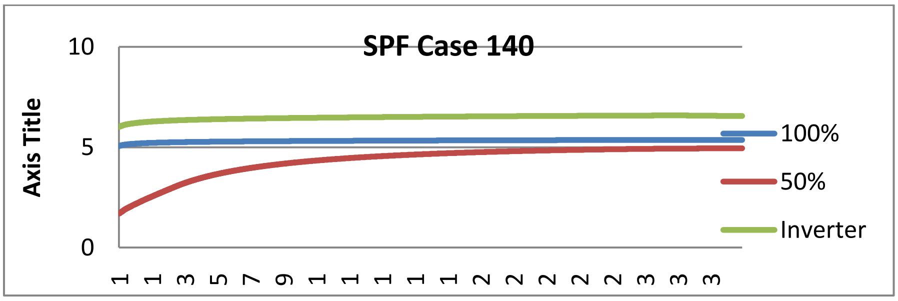

Fig 2.18. Case 140 Heat pump model s SPF

10.3 House Kongsberg Comparing user house and profiles for SPF

The house located in Kongsberg Norway has a heating demand of 8,7 kW at DOT conditions. The Energy demand was found to be 18342 kWh spread over 4459 hours in the year. The method for selecting operation conditions for the Heat Pumps was the same as for previous Case 1 and 140.

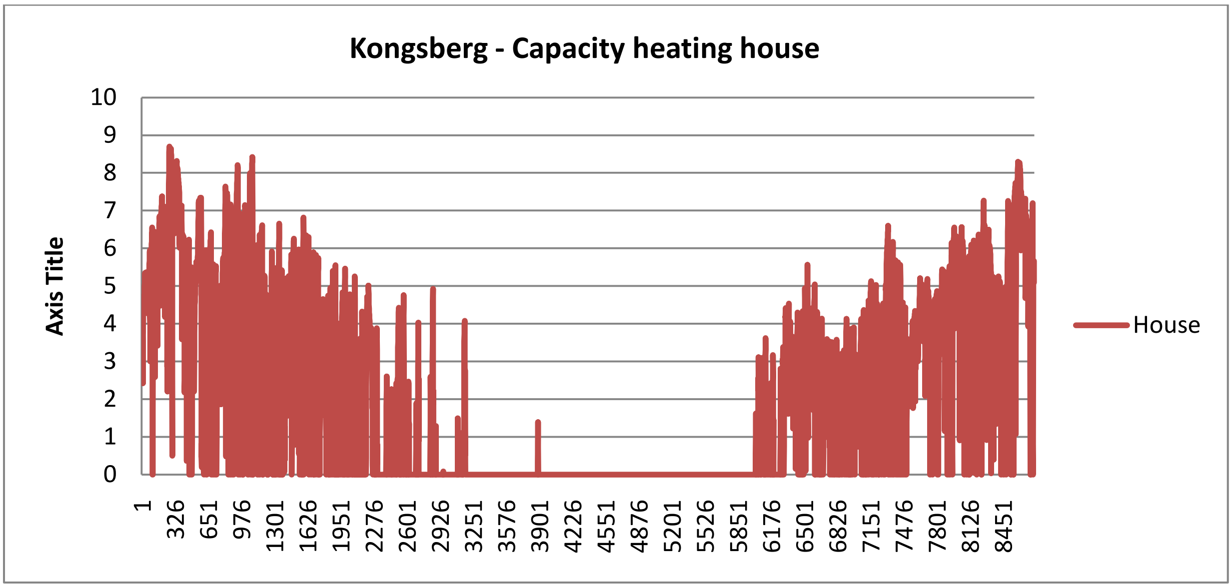

Fig 2.19. Heating capacity demand for house without occupants.

	Size Capacity HP of House	Design kW Capacity	Design kW Capacity HP	Vh m^3/s compressor	Q reneable and el HP kWh
House	50% O/o	8,7	4,3	0,001745	11703 -6730
House	65% O/o	8,7	5,6	0,0023	13562 -4894
House	100% O/o	8,7	8,7	0,003906	14252 -4260
House	40% Invert	8,7	3,48	0,0011	14491 -3920
House	50% Invert	8,7	4,3	0,001745	14394 -4017
House	65% Invert	8,7	5,6	0,0023	14295 -4116

Table 2.3. Capacities and result for selected for Heat Pump models.

10.3.1 Kongsberg – Capacity data from House model - 50% HP capacity

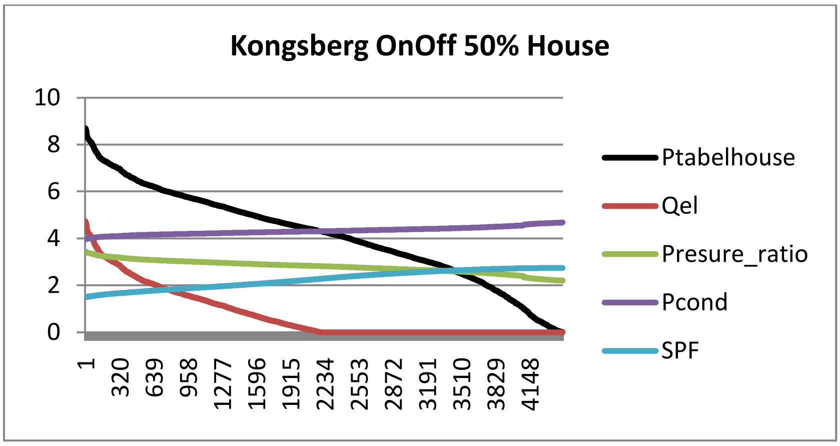

Fig 2.20. Heat pump model On/Off that covers 50% demand.

Here in Fig XX can be seen that the duration of additional heat last for approximately halve of the heating season. This affects the SPF in the way that it increases almost with at same degree the whole period.

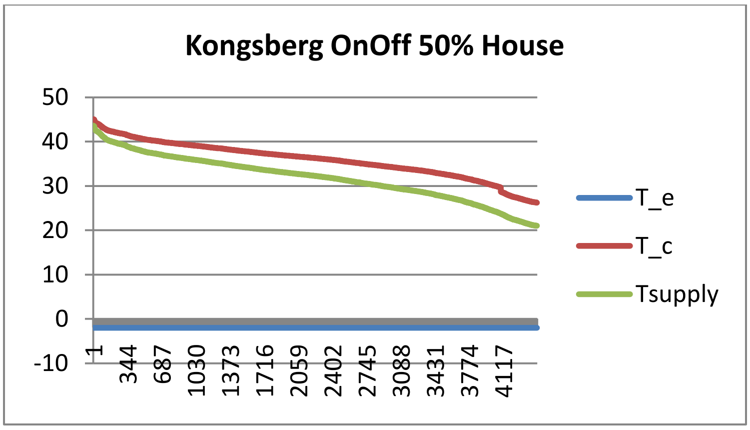

Fig 2.21. Heat pump model On/Off that covers 50% demand. Temperatures.

Again in Fig xx the increasing temperature difference when decrease in temperature level.

10.3.2 Kongsberg – Capacity data from house model - 65% HP capacity

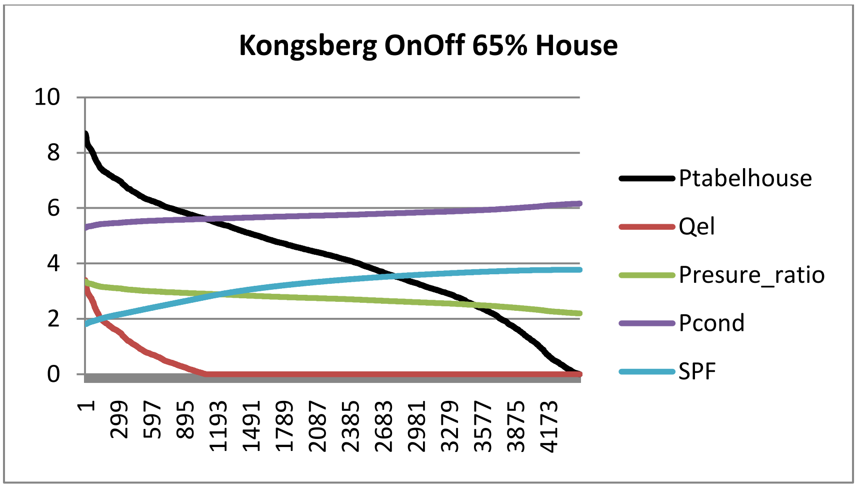

Fig 2.22. Heat pump model On/Off that covers 65% demand.

In Fig xx the SPF curve is showing a leveling tendency because of the reducing affect of the additional el heater.

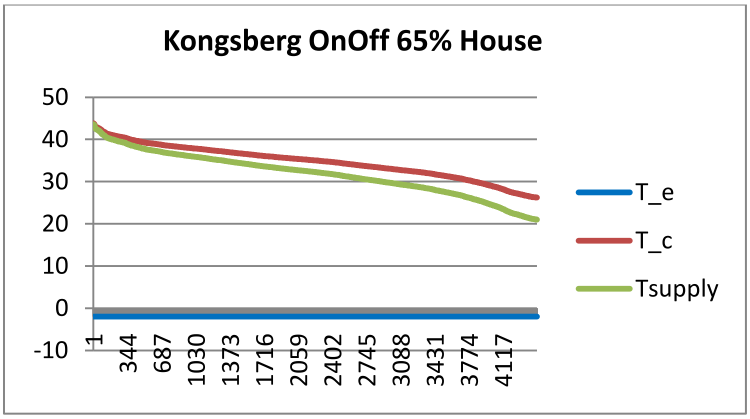

Fig 2.23. Heat pump model On/Off that covers 65% demand. Temperatures.

Again in Fig xx the increasing temperature difference when decrease in temperature level.

10.3.3 Kongsberg – Capacity data from house model - 100% HP capacity

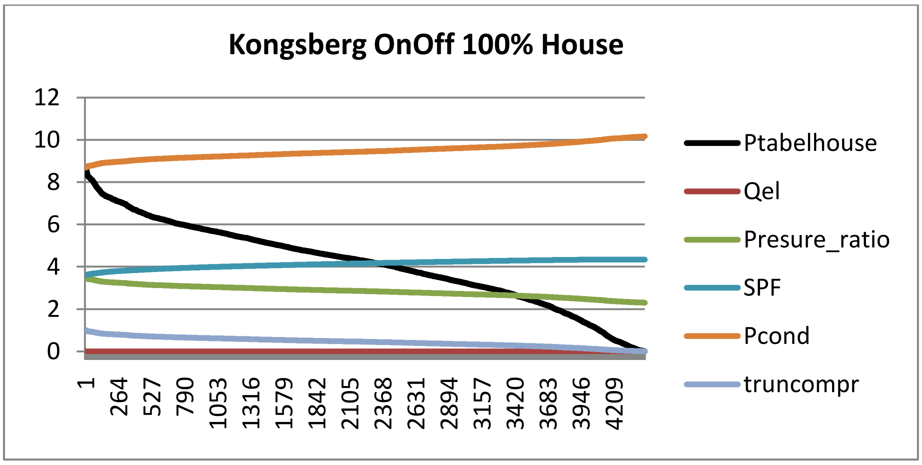

Fig 2.24. Heat pump model On/Off that covers 100% demand.

When no additional heater is implemented the SPF starts at a higher level than for Heat Pump with lower capacity. This results in a higher resulting SPF.

10.3.4 Kongsberg – Capacity data from house model - 100% HP capacity

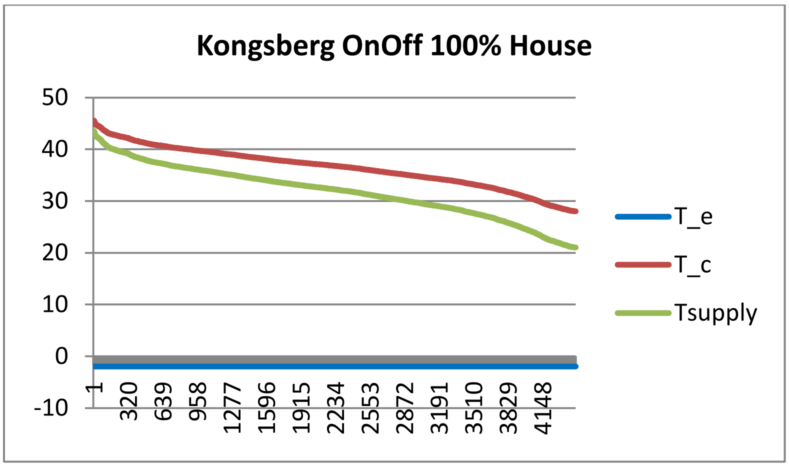

Fig 2.25. Heat pump model On/Off that covers 100% demand. Temperatures.

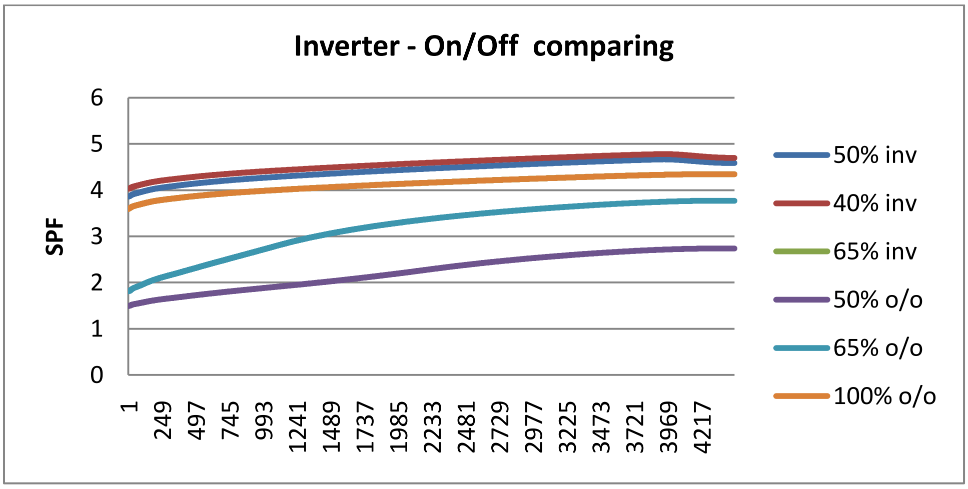

Fig 2.26. Heat pump model comparisons, On/Off – inverter.

10.4 Kongsberg – Capacity data from 2Y2y model

The user profiles made for a young couple with 2 young kids made a reduction to 15841 kWh for heating the house. The heating capacity was reduced from 8,7 kW to 8,2 kW.

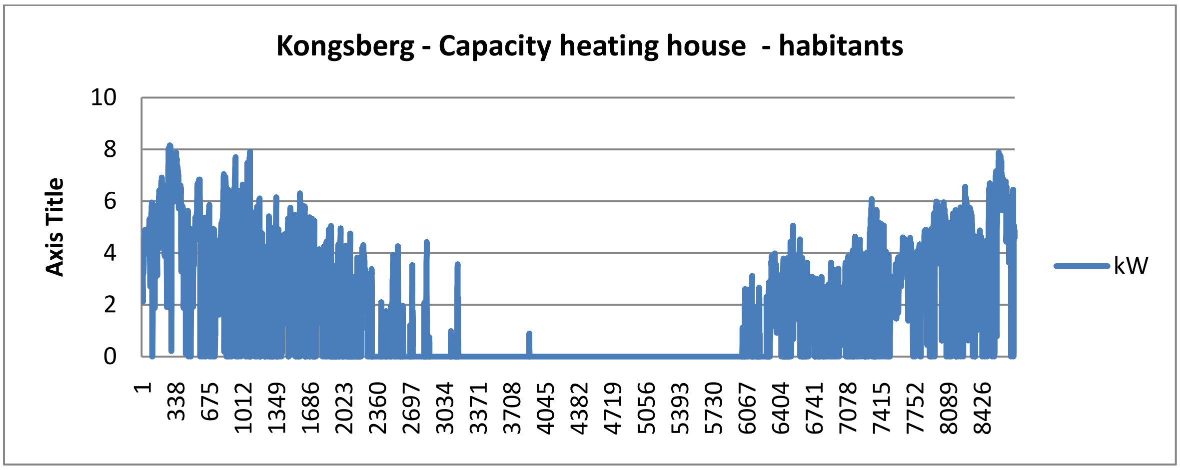

Fig 2.27. Diagram on heat demand capacities during a year.

	Size Capacity HP of House	Design kW Capacity	Design kW Capacity HP	Vh m^3/s compressor	Q reneable and el HP kWh
2Y2y	50% O/o	8,15	4	0,00156	10330 -5589
2Y2y	50% Invert	8,15	4	0,00156	12894 -3004
2Y2y	50% On/Off House HP	8.15	4.3	0,001745	10756-5168

Table 2.4. Heating capacity and results for house with occupants. Kongsberg .

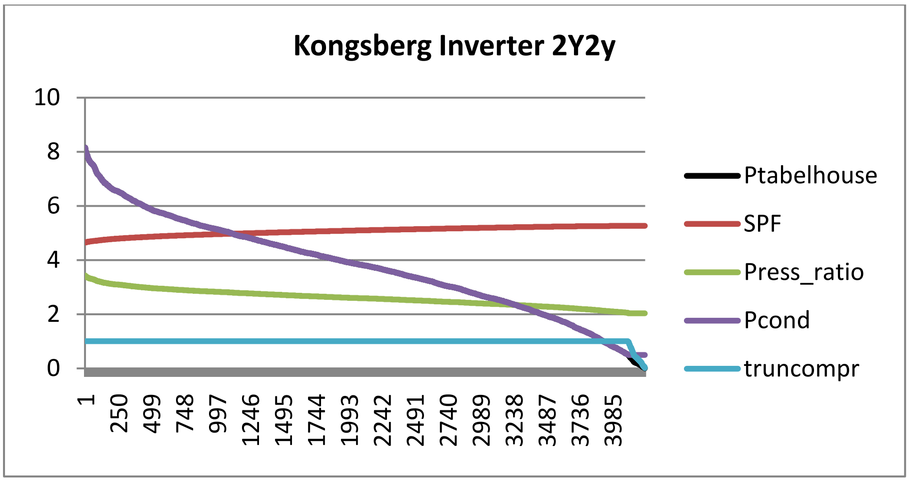

Fig 2.28. Heat pump model with 2Y2y - Inverter 50% HP capacity

10.4.1 Kongsberg – Capacity data from 2Y2y model -On/off HP capacity

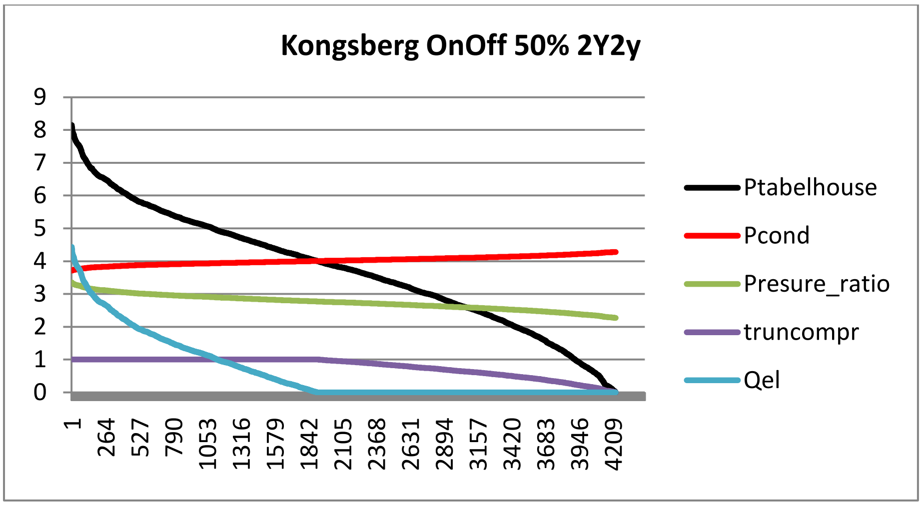

Fig 2.29. Heat pump model with On/Off that covers 50% demand.

10.4.2 Kongsberg – Capacity data - House50% with input from 2Y2y HP capacity

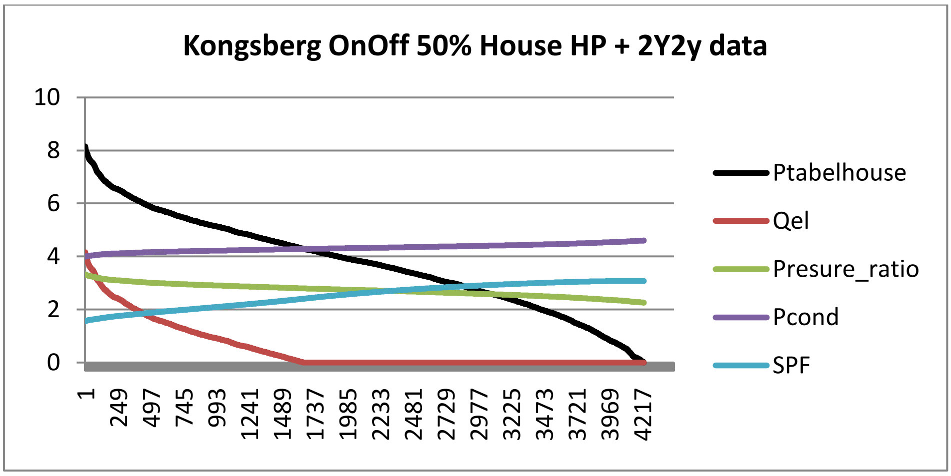

Fig 2.30. Heat pump model with On/Off that covers 50% demand.

10.4.3 Kongsberg – Capacity data - Inverter 40/50/65% HP capacity

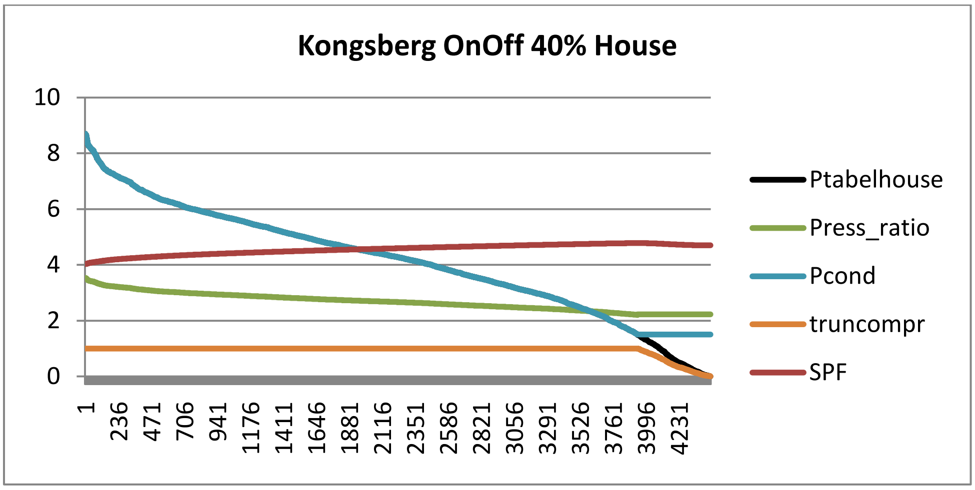

Fig 2.31. Inverter Heat Pump with 40% compressor volume comparison to On/Off Heat Pump

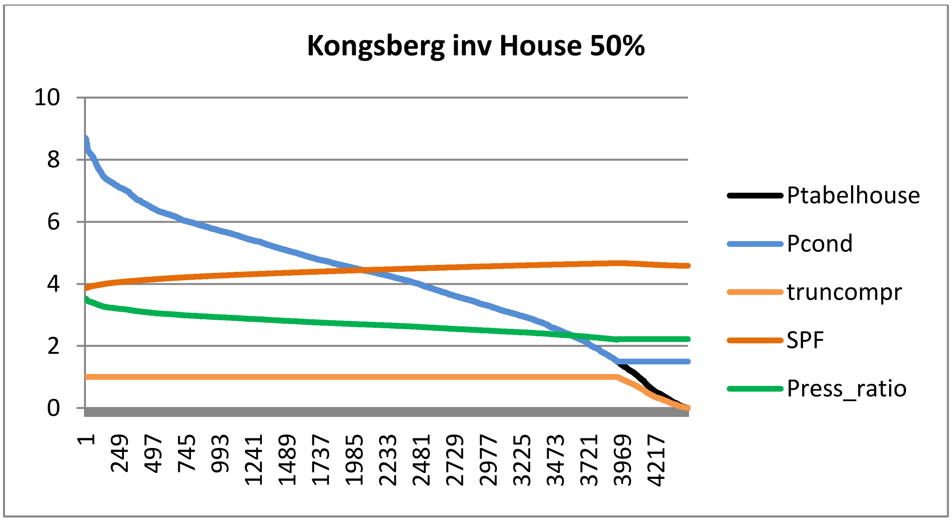

Fig 2.32. Inverter Heat Pump with 50% compressor volume comparison to On/Off Heat Pump

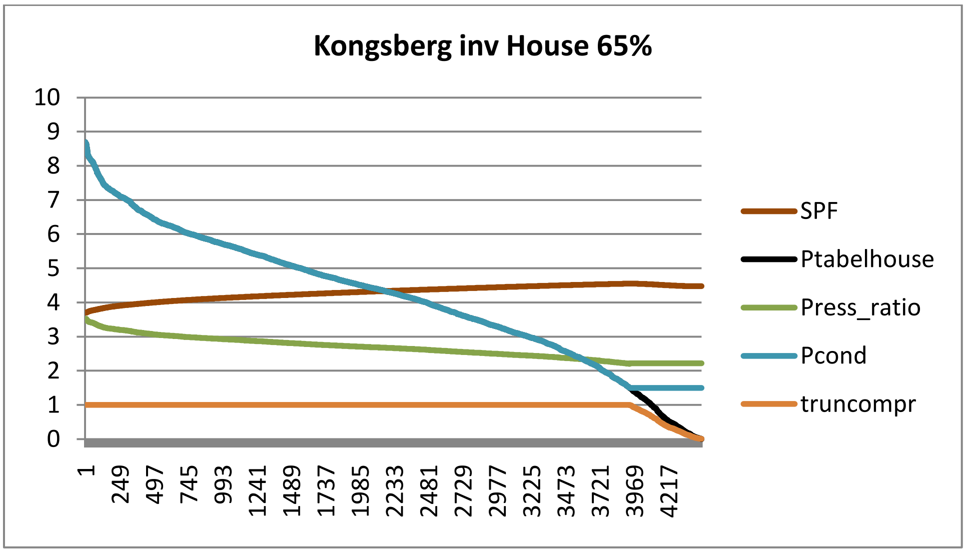

Fig 2.33. Inverter Heat Pump with 65% compressor volume comparison to On/Off Heat Pump.

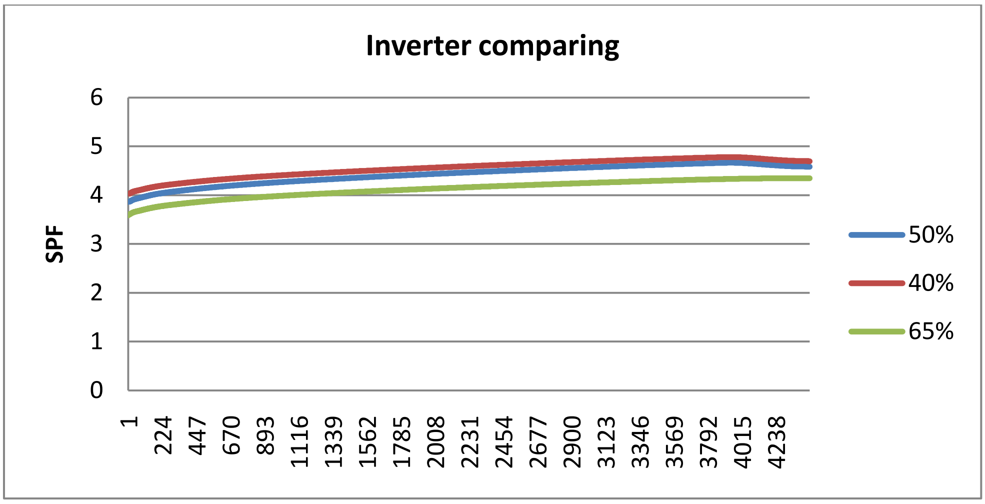

Fig 2.34. Kongsberg House capacities Heat pump model SPF.

In fig xx it shows that the smallest compressor volume gives the highest SPF when combined with variable speed.

11 Discussion with conclusion.

The purpose of this report was to make a theoretical comparison between an On/Off and an Inverter controlled heat pump. In order to do so several models with a variety of input was made. There were a few set-backs related to the inexperienced programmer as mentioned earlier. However, the two types of models describe the processes as close to reality as it was possible at this stage.

The advantages of models are that they can be used to compare processes with exactly the same given conditions.

The model of the heating system is identical for both models and gives a good description on the relation between temperature resistance in the floor above the heat pipes and the capacity transferred to the room and water temperatures. Simplifications like only one-dimensional heat transfer and equal temperature on the floor were made. The materials in the floor were chosen to illustrate a semi old house.

Calculation of the temperatures for condenser had to be simplified because of difficulties in making loops when programming the model. The condenser temperature was made as a function of the water temperature in order to reflect the outgoing temperature difference of the condenser (dT – not LMTD). The evaporator temperature was set to a fixed level.

The effect of the evaporator temperature being set at a fixed level would most likely be that the inverter heat pump has higher pressure ratio at low capacity hence lower COP, and the On/Off heat pump has a lower pressure ratio hence a higher COP.

The model of the refrigeration process was with exception of the above turning into a good tool. The model took into consideration the isentropic – and delivery efficiencies as a function of pressure ratio, and the sub cooling and superheating was considered. The start-losses were made into extra run time depending on pressure ratio. Numbers of start/stop might be higher than real because of programming only within the hour and not over a period of time decided by the energy amount in the buffer tank.

With this in mind the models can be discussed.

Case 1 – House in Denmark.

This house was equipped with three different heat pumps, 50%, 70%, 100% and an Inverter controlled one. The supplied energy ranged from 4529 kWh for the 50% to 3330 for the inverter. Surprisingly the 100% one showed higher energy consumption than the 70% one. The cause of this is probably the frequent start/stop losses incorporated in the model to reflect the time from the compressor starts until it is delivering energy at right temperature. Out of the SPF comparison in fig 2.10 it can be seen that it takes a long time for the 50% HP to increase the low Performance Factor in the beginning when the el-heater is in use.

Case 140 – House in Denmark.

Here was used a house model with low DOT capacity and low energy demand during heating season. Peak capacity reached 2,47 kW and three models were made, 50%, 100% and Inverter. Not surprisingly the Inverter pump is consuming the least energy. But at such a low capacity in heat demand the potential of saving bought energy is low as well. The variation in consumed energy is 140 kWh in favor of the Inverter and this has limited possibilities for investment for the Inverter heat pump. The inverter heat pump turned out to be using the least energy to heat the house.

Kongsberg – House model.

This house was situated in an inland climate in Norway. It was modeled three On/Off heat Pumps and three Inverter ones. 50%, 65% and 100% On/Off, and 40%, 50% and 65% Inverter. The heat demand capacity was 8.7 kW and energy demand was 18342 kWh. As shown in table 2.26 all of the Inverter sizes performed better than the On/Off ones. The 100% on/off comes very close to the poorest Inverter one. It was interesting to see that the larger the Inverter pump was sizes the lower the SPF turned out. This was investigated and the reason was found to be that the modeled efficiencies for el motor was made a function of the rpm. Since the rpm was higher for the Inverter with low compressor volume the efficiency was best here. The comparison between the heat pumps shows that the inverter pump has the highest SPF with the 100% On/Off on second.

Kongsberg – House model including occupants.

A simple user profile for a four-person family was implemented into the House heat demand model. This resulted in decreased heating capacities and consequently a reduction in energy demand over the heating period. The reduction in installed heating capacity resulted in a capacity of 8.15 kW and energy demand was 15841 kWh. The difference in heat demand capacity resulted in a reduction of installed heat pump capacity of 0.3 kW when considering 50% coverage. This would not affect the selection of size heat pump unless the capacity-step of the model was around this point. The result of using a House heat pump for the user-profile is a slightly higher SPF.

All in all this report shows that an inverter heat pump reaches a higher SPF than the on/off ones compared with at these conditions given here. The result from Kongsberg House shows that a 100% on/off comes close to the poorest inverter one. Comparing with results from Case 140 which showed more difference between the inverter and the on/off there can be reasons for believing that for larger energy demand (long heating season) the on/off heat pump can compete with an inverter one.

12 Further work/Improvements

One of the reasons for making models is to see what the consequences or performances will be when (if) before anything is build. The cost of correcting design errors increases from low cost at initial stages (planning and design) to costs that cannot be covered. For the latter the owner will have a plant that doesn't work as good as it possibly could.

The reason for making these models was to find out which had the best SPF and then analysis could be carried out regarding which one to choose regarding economy. The one with the best SPF is not necessarily the one for best economy

In order to get as good as possible information for selecting the best (economical or other criteria) heat pump it is crucial to have a model that reflects the necessary parameters that effects performance. The models made in this work are of general character and does not have specific data from suppliers. The models could be improved regarding:

- Implementation of heat exchanger for condensing that calculates Tc based Hydronic- and Refrigeration process data.
- Implementation of heat exchangers for evaporating that calculates Te based source-, and Refrigeration process data.
- Improving the heat source model for connecting to the evaporator.
- Implement alternative model for variable flow system for hydronic and brine in order to analyze which results in highest SPF.
- Implementing hot water consumption into the model based on number of occupants.
- Find a method of implementing the function of an accumulator tank in order to reduce numbers of start and stops for heat pumps. This would include making a system that utilized the layering advantages of water in a tank and controlled water flow depending on capacity both towards hydronic system and condenser.

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Enclosures:

A: Kongsberg 15.06. Results and graphs

B: Heat demand living House Kongsberg

C: Huse 6172 17.06 results graphs

D: BIEF udtræk - Hus 6172

E: EES program for Kongsberg house

F: EES program for Huse 6172