UNIVERSITY OF TROMSØ
THE ARCTIC UNIVERSITY OF NORWAY

Study of Refrigerants for Heat Pumps in Colder Climate

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TEK-3901 Master in
Technology and Safety in the High North

A thesis submitted for the degree of
Master of Engineering
June, 2014

Preface and Acknowledgements

The work and simulations described in this thesis was carried out in the Faculty of Nature Science and Technology at the University of Tromsø, it was carried out from January 2014 to July 2014. The thesis is original, unpublished and independent work done by the author, N. E. Eriksen. The thesis consists of approximate of 24000 words, and 48 figures and 46 tables.

The master thesis is equal to 30 ECTS and is a final work for the master program in Technology and Safety in the High North conducted at the University of Tromsø. The aim of the thesis is to do an in-depth knowledge within an area which is relevant for the master program. The learning goal for the thesis is to develop and improve the research work, planning, writing and processing of information.

I am grateful for the help and discussions that I have gotten from my supervisor Hassan A. Khawaja. I am thankful for the teaching and conversations/discussions I have had with Professor J. Barabady during the two years of the master program. I want to thank Yves Ladam at Kuldeteknikk AS in Tromsø and Dr. Jørn Stene at COWI Trondheim for helping me with information and their knowledge about the use of heat pumps. I want to thank my friend Trond W. Kåven for proof reading my master thesis.
I want to specially thank my son, Nils Even, for reminding and motivating me to complete the master program and master thesis.

Tromsø, 1st of June 2014

Nils Eivind Eriksen

Abstract

The aim for the thesis is to look at the performance of natural refrigerants in heat pumps while operating in a colder climate. The advantage of using natural refrigerants is that they have significantly lower global warming potential (GWP) in comparison to the synthetic refrigerants i.e. hydro fluorocarbons (HFC) and hydro chlorofluorocarbons (HCFC). Natural refrigerants are becoming common to use as a refrigerant in domestic heat pumps however still most household heat pumps use synthetic refrigerants. Due to the harmful impact of synthetic refrigerants, EU regulations are limiting the use of synthetic refrigerants in the future appliances. Similar steps are being taken by other countries around the globe.

The motivation behind this work is to assess the energy requirement in the use of heat pump in colder climate conditions using natural and synthetic refrigerants. For the study, climatic data from Karasjok (69°28′55″N 25°6′18″E) is taken into consideration (eKLIMA, 2014). Karasjok is located in Finnmark, county of Norway. The average temperature in Karasjok over the year can be as low as -1.45 ºC (from the stats of year 2012).

The heat pump analysis was performed on three natural refrigrants, R744 (carbondioxide – CO_2), R717 (ammonia – NH_3), R290 (propane – C_3H_8) and one synthetic refrigerant R410A (a 50/50 mixture of HFCs: R32 (difluoromethane – CH_2F_2) and R125 (pentafluoroethane – CHF_2CF_3)). R290 and R744 have better performance in colder climatic condition within different analysis segments in single stage as well as two stages heat pump cycles. This shows that natural refrigerants are able to replace the synthetic refrigerants when considering the performance in the colder climate. Analysis also showed that a single stage R290 heat pump is able to cover up to 90% of the heating and hot water supply need on annual basis (data taken for years 2012 and 2013).

The study was conducted in CoolPack©, MS Excel® sheet (Kolsaker, 2013) and log p-h diagrams. The analysis of refrigerants close to transcritical pressure was performed in CoolPack©. The analysis of two-stage refrigeration cycle was performed using CoolPack© and MS Excel® (Kolsaker, 2013). The results were compared using log p-h diagrams.

The conclusion of the study is that with the use of natural refrigerants, it is possible to save energy. In addition, natural refrigerants have far lower environmental impact than its synthetic counterparts. Therefore, it is suitable to use natural refrigerants in replacement of synthetic refrigerants.

Keyword: CO_2 heat pump, cold climate, colder climate, natural refrigerants, electricity saving

Glossary

CFC	Chlorofluorocarbon
Comp.	Compressor
Cond.	Condenser
Conventional refrigerant	Refrigerants that is used nowadays (2013/2014)
CoolP.	CoolPack©
COP	Coefficient Of Performance
Diff.	Difference
DWH	Domestic Water Heating
El.	Electricity
Etc.	Et cetera – and so on
FMEA	Failure Mode and Effect Analysis
Gas c.	Gas cooler
GWP	Global Warming Potential
HCFC	Hydro Chlorofluorocarbons
HFC	Hydro Fluorocarbons
i.e.	That is
Log.	Log p-h diagram
MAC	Mobile Air-Conditions
MS Excel®	Microsoft Excel®
ODP	Ozone Depletion Potential
R134a	1,1,1,2-Tetrafluoroethane – CH_2FCF_3
R22	Chlorodifluoromethane – $CHClF_2$
R290	Propane – C_3H_8
R404A	(a 44/52/4 mixture of HFC: R125 (Pentafluoroethane – CF3CHF2), R143A (Trifluoroethane – CF3CH3) and R134a (1,1,1,2-Tetrafluoroethane – CH_2FCF_3))
R407C	(a 23/25/52 mixture of R32 (Methylene Fluoride – CH_2F_2), R125 (pentafluoroethane – CHF_2CF_3) and R134a (1,1,1,2-Tetrafluoroethane – CH_2FCF_3))
R410A	A 50/50 mixture of R32 (difluoromethane – CH_2F_2) and R125 (pentafluoroethane – CHF_2CF_3)
R717	Ammonia - NH_3

R744	Carbon dioxide - CO_2
SPF	Seasonal Performance Factor
Subcritical	Pressure/temperature is below the critical point/value of the refrigerant
Synthetic refrigerant	A refrigerant that is a mixture of two or more chemical refrigerants
TA	Air temperature
TAM	Average measured air temperature
TAN	Minimum measured air temperature
TAX	Maximum measured air temperature
Transcritical	Pressure/temperature is above the critical point/value of the refrigerant
U-value	Conventional heat transfer coefficient (per square meter)

Nomenclature

π	Pressure ratio
ºC	Degree Celsius
ΔT_{SH}	Evaporators superheat in K
Bar	Unit of pressure
h_1 or h_A	Enthalpy (kJ/kg) values from log p-h diagram and/or MS Excel®
K	Kelvin
kg/s	Kilogram per second
kJ	Kilo Joule
kJ/m^3	Kilo Joule per cubic meter
kW	Kilowatt
kWh	Kilowatt hour
$P_{compressor}$	Effect (kW) of the compressor
$P_{gas\ cooler/condenser}$	Effect (kW) of the gas cooler/condenser
P_{HT}	Effect (kW) of the high pressure compressor
P_{LT}	Effect (kW) of the low pressure compressor
p_m	Effective pressure
m^2	Square meter
$\dot{m}_C$	Mass flow (kg/s) in state C in figure 4.13
$\dot{m}_{evaporator}$	Mass flow (kg/s) in the evaporator
$\dot{m}_{gas\ cooler/condenser}$	Mass flow (kg/s) in the gas cooler/condenser

$\dot{m}_{R744}$	Mass flow (kg/s) for the current refrigerants
x_F	Liquid percent in state F in figure 4.13
W/K	Heat loss
W/m^2*K	Conventional heat transfer coefficient (per square meter)
Wh/m^3*K	Airs heating capacity

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1. Introduction

This thesis presents a feasibility study of replacing conventional refrigerant with natural refrigerants as carbon dioxide (CO_2 – R744) in domestic heat pump system. In this study, the heat pump will be used for heating the house and the domestic water supply with the assumption that the heat pump may not cover complete heating requirement in the house.

In this work, performance of natural refrigerant heat pumps is investigated for different climatic conditions that occur in Karasjok (Finnmark, Norway). The presented study is based on numerical simulations, where various analysis software are employed such as CoolPack© and MS Excel®.

The temperature variations at Karasjok are known form collected data (eKLIMA, 2014). The data consists of maximum, minimum, average and normal temperatures on daily basis. An MS Excel® sheet is also provided which summarises the domestic heating needs with climatic variations.

1.1 Background

Heat pumps have been used domestically since late 1990's. The most common type of refrigerants used in these heat pumps are hydro chlorofluorocarbons (HCFCs) and hydro fluorocarbons (HFCs). HCFCs and HFCs are high performance refrigerants however due to their environmental impact, strict regulations are being imposed by regulatory authorities in their usage (European Union (EU), US Environmental Protection Agency (EPA), Environmental Agency of Japan, Environment Canada, etc.). In the consequence, HCFCs have already been eliminated from the market in 2010. Further, EU council has placed a regulatory initiative under Kyoto Protocol to replace HFCs with refrigerants of lower global warming potential (GWP) by 2020 (Eikevik, 2013). The potential candidates for replacement of HFCs are natural refrigerants such as CO_2, Ammonia (NH_3 – R717) and petroleum products such as Propane (C_3H_8 – R290). These choices are based on their lower GWP (table 1.1), however there are other issues associated in their practical usage. For example, R717 is a toxic gas with strong pungent smell. Similarly, petroleum based refrigerants are highly flammable and potentially a fire hazard. In comparison, R744 is neither toxic nor flammable. It is well suited for use as a replacement of HFCs (Nekså, 2000).

Table 1.1: GWP, flammable and toxicity for different refrigerants (Stene, 2013) (edited)

Refrigerant	GWP	Flammable	Toxicity
R404A	3800	No	No
R407C	1700	No	No
R410A	2000	No	No
R134a	1300	No	No
R717 (Ammonia – NH_3)	0	No	Yes
R290 (Propane – C_3H_8)	3	Yes	No
R744 (Carbon dioxide – CO_2)	0 (1)	No	No

The use of R744 refrigerant in heat pumps systems is not new. There are several manufacturers around the globe especially in Asia (Japan) that are producing R744 based heat pumps for the domestic usage. These manufacturers are on the front end of the technology on R744 based heat pumps (Fernandez, Hwang and Radermacher, 2009).

R744 heat pumps have been developed and patented in Norway in the late 1980s (Eikevik, 2013) however since this technology has not been popular due to its low coefficient of performance (COP) in comparison to conventional refrigerants. R744 heat pumps have been used for domestic purposes in Norway, however the results have been far from success because of low energy savings (Stene, 2014). In a colder climate such as in Karasjok, the temperatures can get as low as -30 ˚C annually and some years even as low as -40 or -50 ˚C. In such harsh conditions normal heat pumps can stop working because of the low COP and heat delivered. Most of the heat pumps are designed to stop working when the outside temperature is between -15 to -20 ˚C (Varmepumpeinfo, 2010).

1.2 Motivation

HCFC and HFC refrigerants have high ozone depletion potential (ODP) and GWP. In 1974, Rowland and Molina published a report that their research showed that chlorine and bromine discharged move into the stratosphere and destroy the ozone. This resulted in the Montreal Protocol in 1987 where it was decided that chlorofluorocarbon (CFC) and HCFC refrigerants would be eliminated from the marked within 2010 (Nekså, 2000) and (Yitai, Zhongyan and Hua, 2013). The refrigerants that are replacing CFC and HCFC are HFC with low GWP and natural refrigerants. Natural refrigerants that will become more popular to use in the future because it is cheap to produce, evaporation heat is high, has no ODP, low GWP, etc. (Johannessen, 2006). The GWP for the some of the HFC refrigerants are 1000-3800 times higher than for the natural refrigerants (table 1.1), R410A (a 50/50 mixture of HFCs: R32 (difluoromethane – CH_2F_2) and R125 (pentafluoroethane – CHF_2CF_3)) has a GWP of 2000.

CO_2 is a gas that receives all the attention because it is related to all of the emissions. However, methane and nitrous oxide have far larger impact on the climate in the same quantity. CO_2 used in heat pumps has no impact on the GWP if it is retrieved from industrial emissions (Austin and Sumathy, 2011).

1.3 Current challenges

Challenges related to use of natural refrigerants in heat pumps in a colder climate:

- System pressure above 80 bar for R744
- Efficiency during low outside temperature/Use during low outside temperatures
- Unknown technology for the common house owner
- Low annual outside temperature
- Air source heat pump has a large decrease in COP as outside temperature decreases

1.4 Research challenges

The main research challenges in this thesis are:

- Can a R744 heat pump work sufficient in a colder climate?

- Is it possible to achieve a COP above 2 for the natural refrigerants at a low outside temperature?
- Is the one stage heat pump better than a two stage heat pump during use in low outside temperature?
- How large can the electricity saving be by using R744, R714, R290 and R410A at low temperature?
- How much of the current electricity used for heating and domestic water heating can the heat pumps cover?

1.5 Limitations

The limitations in this thesis are as follows:

- The CO_2 heat pump will only work in transcritical pressure
- Presented work is done numerical (simulation) and analytic (theoretical)
- Material fatigue is not considered in this project
- Only R744, R717, R290 and R410A refrigerants are used
- Air humidity is not considered
- The R744 heat pump is used for hydronic heating, up to 60% of the heating demand

2. Literature review

Heat pumps have been popular to install in houses for over a decade in Norway, this is because it is possible to reduce the electricity consumption in the household while cheeping the same inside temperature. A heat pump has a very easy structure and consists of four main parts; compressor, gas cooler/condenser, throttling valve and evaporator. Heat pumps are the only heating technology that has a COP that is larger than one and a heat pump is the end-user because it transforms the electricity to heating of air or water (Hakkaki-Fard, Aidoun and Ouzzane, 2013). R744 is a refrigerant that can become more popular to use as a refrigerant in heat pumps because of its good properties to heat water and because of the low GWP. The outlet temperature of the water can be relative high compared to conventional refrigerants, this gives the R744 heat pump a high COP when used for heating of water. The great temperature span (from input to output) means that the use of R744 heat pump can reduce the use of electricity up to 75%, compared with electric water heaters and gas water heaters (Yitai, Zhongyan and Hua, 2013).

2.1 Heat pump theory

2.1.1 Conventional heat pump

The energy consumption for heating is relative large in a household, this can be reduced by using a more effective energy source like a heat pump, this technology can cover heating needs for houses, office buildings or industry. A general picture of the energy consumption of a heat pump can be seen in figure 2.1. The electric energy (compressor) should stand for 1/4 or less of the heating of the refrigerant and the rest (3/4) of the energy should come from the environment (evaporator), the heat release occur in the gas cooler/condenser (Stene, 1997).

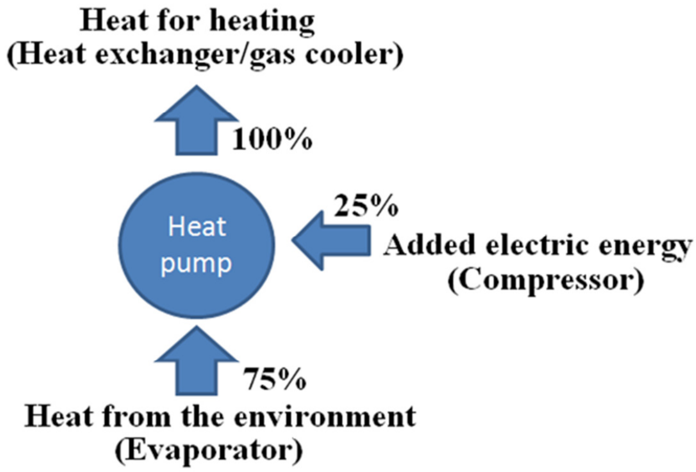

Figure 2.1: How much each part of the heat pump contributes to achieve a good performance

Figure 2.2 show a heat pump cycle in a log p-h diagram. The different stages in the log p-h diagram can be interpreted as (Ndla, n. d.):

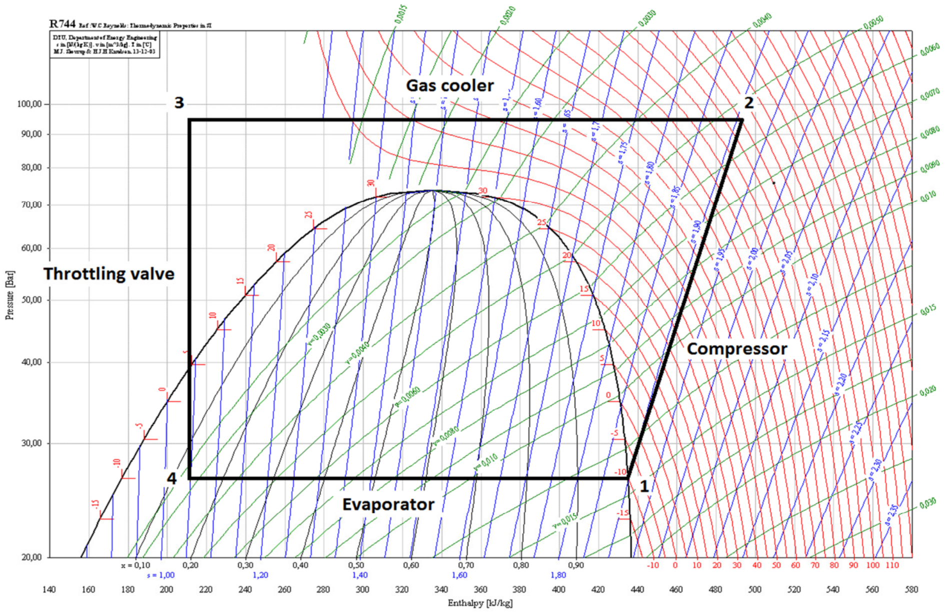

Figure 2.2: Log p-h diagram for a R744 heat pump

- **4-1 – evaporator**

  The pressure drop over the valve needs to be sufficient enough to ensure that the refrigerant liquid boils. The heat from the environment needs to be higher than the refrigerant temperature (heat goes from hot to colder), the refrigerant will evaporate and the energy is stored in the refrigerant gas.

- **1-2 – compressor**

  The refrigerant comes from the evaporator in gas phase, pressure and temperature increases because the refrigerant is being compressed. The output temperature needs to be sufficient for the heat pumps purpose.

- **2-3 – gas cooler/condenser**

  The refrigerant releases the heat (energy) to the house and/or water heating. The refrigerant goes from gas to liquid phase the pressure and temperature is constant (R744 stays in gas phase throughout this process, the gas temperature only decreases).

- **3-4 – throttling valve**

  The refrigerant is pressed through the valve to reduce the temperature and pressure. The valve is adjustable, it is adjusted in a way so the refrigerant reaches the given evaporation temperature (for R744 it also ensures correct mass flow through the system).

2.1.2 Colder climate influence on heat pumps

Heating of a house is directly influenced by the outside temperature, thereby the climate have a directly influence on the produced effect from the heat pump. Norway has four main climates; inland, fjord, south/west coast and north coast. Annual temperature vary throughout these climates. The north cost and the inland climate gives the best opportunities for energy saving because of the low annual temperature and long heating season. When considering the climate in Karasjok (inland), the temperatures can in periods be relative low. This can affects the heat pumps operating time, when the temperatures are low there can be a challenge for the heat pumps to achieve a sure plus effect from the heat pumps. When using a heat pump as the main heat source it is important to have electric ovens or fossil fuel (wood, gas or oil) as a second heat source, if the heat pump fails or the outside temperature becomes lower than the heat pump is designed for. During a year, a heat pump should cover up to 70% of the heating need (Stene, 1997) and (Aftenposten, 2013).

In a colder climate there may be an increase of failures, this can be investigated by making a risk matrix. Risk matrix is a widespread, graphical and easy tool for evaluating risk. It shows the probability for an event occurring and the consequence of the event, thereby probability x consequence = Risk. It is mainly used for determine the size of the risk and whether or not this is sufficient controlled. There are two dimensions to a risk matrix, it looks at the severity and the likelihood of an unwanted event is. This creates a matrix. When combining the probability and consequence, it will give a place in the risk matrix (CGE, 2014). Table 2.1 shows classification of consequences that may occur in a colder climate, the failure modes and the degree of consequence that the failure modes can have. By implementing the failure mode in table 2.1 and the frequency of the failure mode, it is possible to implement the failure mode in the right cell in table 2.2. An example of this can be that the damage on compressor, from collected values from Folksam (2009) this occurs relatively frequent and the consequence can be relative major. Thereby, it will be on the red area in the risk matrix. This can be interpreted that the quality of the compressor is low, not designed for the climate, poor quality on the components in the compressor, etc.

Table 2.1: Classification of the consequences of failure modes that may occur in a colder climate (Stene, 2014) and (Folksam, 2009)

Classification of consequences					
Failure modes / **Degree of consequences**	**Outside fan failure**	**Damage on compressor**	**Problem with de-icing on the evaporator**	**Icing of sump on outdoor unit***	**Icing/snow on outdoor unit***
Catastrophic	Fan engine fails	Compressor fails to operate	Ice throughout the evaporator	Ice inside the unit	Unit is covered with ice/snow
Critical	Fan stops	Low efficiency in the compressor	Much ice on the evaporator	Ice up to the unit	Half the unit is covered by ice/snow
Major	Damage on fan	Major efficiency reduction in the compressor	Ice on half of the evaporator	Large ice accumulation on the sump	Ice/snow on the unit
Minor	Low fan speed	Compressor has a minor efficiency reduction	Some ice on the evaporator	Ice accumulation on the sump	Some ice/snow on the unit
None	Fan operates when needed	Compressor operates ass it should	No ice on the evaporator	No ice on sump	No snow/ice on the unit

* Heat pump

Table 2.2: Risk matrix for failures modes that may occur in a colder climate (Folksam, 2009) and (CGE, 2014)

Probability / Consequences	Very unlikely	Remote	Occasional	Probable	Frequent
Catastrophic					
Critical					
Major					**Damage on compressor**
Minor					
None					

Classification of probability	
Probability	**Frequency***
Very unlikely	0 to 49 times in 1 year
Remote	Between 50 to 99 times in 1 year
Occasional	Between 100 to 149 times in 1 year
Probable	Between 150 to 199 times in 1 year
Frequent	More than 200 times in 1 year
Colour	**Description**
Red	The risk is unacceptable and measures need to be taken to reduce the risk
Yellow	The risk needs to be assessed and measures needs to be considered
Green	The risk is acceptable

*file: House calculation – Heat pump failure/Appendix: A

For a more detailed analyses of the failure modes mentioned in table 2.1 it is possible to use the Failure Mode and Effect Analysis (FMEA). FMEA is a methodology to analyse a system to identify potential problems. FMEA helps in identifying the failures in the system before they occurs and henceforth assist in avoiding them. FMEA has been used by Rausland (2005), by using a tabling method. In this method, potential failures are detected, ranked in their severity and occurrences rate. FMEA have been conducted to investigate some of the failures that may happen in the components of the heat pump due to colder climate are given in appendix: B.

2.2 R744 heat pump

R744 heat pumps are best suited for use where it is a large demand for hot water, like in: hotels, houses with hydronic heating, cleaners, hospital, restaurants, etc. When installing a R744 heat pump in a house it needs to have a relative large demand for domestic water heating (DWH), it should constitute 50% or more of the annual heating need for the house. The house should also have a low temperature (up to 35 ºC) hydronic heating system in the house. Thereby R744 heat pumps are best suited for use in low energy houses or passive houses (Stene, 2014).

2.2.1 Properties of R744

When designing a heat pump system it is important to design the components after the refrigerants that are being used. The properties of R744 are rather different form the conventional refrigerants on the marked and there are two factors that are special for R744; it has a low critical temperature and the high working pressure (Austin and Sumathy, 2011). R744 uses a much higher pressure than the conventional refrigerants, the volumetric refrigerant capacity (22.545 kJ/m^3 at 0 ºC) is 3 – 10 times higher than CFC, HCFC and HFC refrigerants. The critical pressure is 73.8 bar and the critical temperature is 31.1 ºC (figure 2.3). It is not possible for the R744 to transfer heat above the critical temperature by condensation as in conventional heat pump. This is due to that the R744 has a heat release at transcritical pressure, the R744 gas does not condensate and it just gets cooled down. Hence, it has gas cooler, not condenser. As seen in figure 2.3 the triple point for R744 are -56.6 ºC and 5.2 bar, at this stage the R744 goes over in solid form (dry ice) (Kim, Pettersen and Bullard, 2003) and (Stene, 2012).

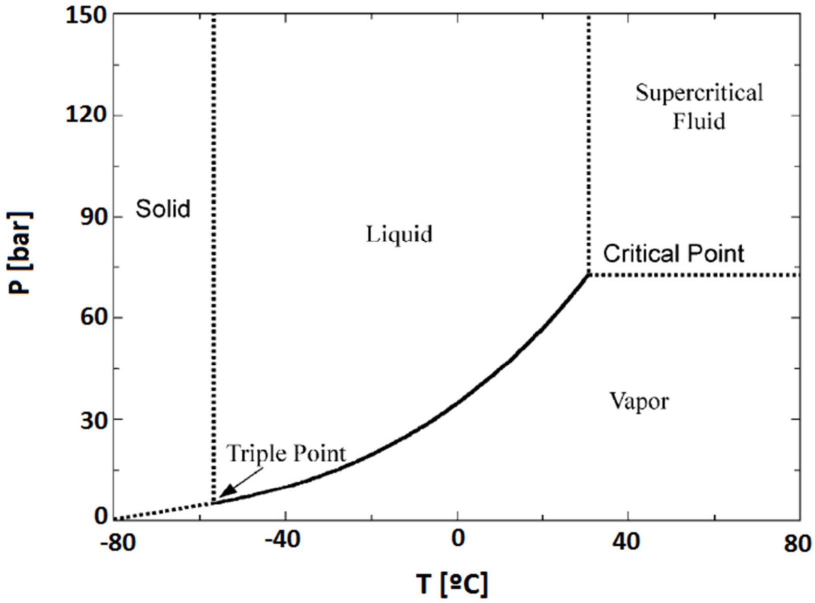

Figure 2.3: Phase diagram for R744 (Kim, Pettersen and Bullard, 2003) (edited)

Liquid R744 is colure less and moves very easily and the thermodynamics properties are very good (Store Norske Leksikon, n. d.). The gas pressure is higher for R744 compared to conventional refrigerants, thereby the temperature change associated with pressure drop will be smaller. The temperature change for R744 can be 4 – 10 times smaller than for conventional refrigerants. The density ratio for R744 is a lot smaller compared to conventional refrigerants, the ratio can be 6 – 9 times smaller (Kim, Pettersen and Bullard, 2003). The pressure that the R744 is working with can be up to 15 times higher than what conventional refrigerants work with. If the refrigerants R134a (1,1,1,2-Tetrafluoroethane – CH_2FCF_3) and R717 are working with a pressure round 10 bar, R744 works with a pressure of 100 bar when working within the same specifications. Some of the benefits with a high pressure are; relative high gas density, high volumetric heating capacity and that it is a relative lower mass flow of R744 compared with HFCs. To achieve the same amount of heating as conventional refrigerants, R744 allows for use of smaller components in the heat pump system and because of the low viscosity (Stene, 2012) and (Austin and Sumathy, 2011). When comparing a R744 and a R22 (Chlorodifluoromethane – $CHClF_2$) system with an effect of 7 kW, the R22 system has a volume of 11.4 liters and the R744 system has a volume of only 4.2 liters. Figure 2.4 show the difference in pipeline diameter between a HFC and R744 heat pump, all the components in a R744 heat pump system are smaller. The diameter of the pipes can be reduced by 60 – 70% compared to a HFC system. Thereby R744 heat pump systems weigh less, even if the walls of the components are thicker due to the high pressure in the transcritical process (Stene, 2013) and (Kim, Pettersen and Bullard, 2003).

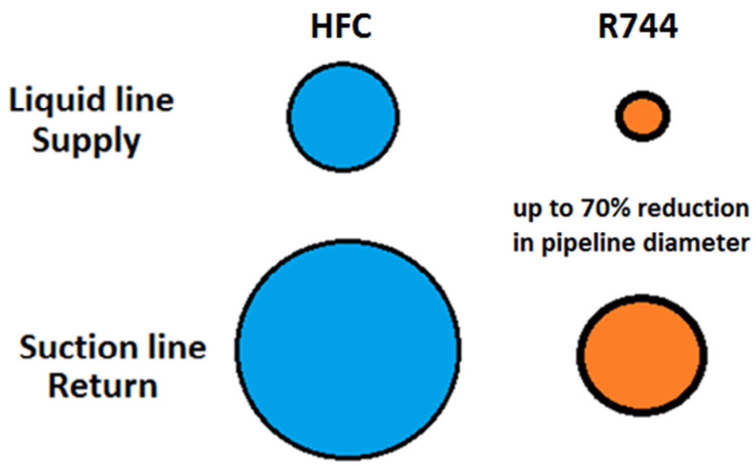

Figure 2.4: Difference in pipe dimension in a HFC and R744 system (Eikevik, 2013) (edited)

2.2.2 Main components

2.2.2.1 Compressor

Because of the thermodynamic characteristics of R744, the transcritical cycle operates at a high pressure. Figure 2.5 compares the mean effective pressure (p_m) and pressure ratio (π) for a R744 and R134a compressor with the same cooling capacity at 0 ˚C. R744 has a lower pressure ratio and a low displacement compared to R134a. Transcritical compressors need to have thicker walls to handle the high operational pressure. The volumetric capacity is large, thereby the compressor is smaller compared to HFC compressors. The pressure can is some cases come up to 150 bar (Reulens, 2009) and (Kim, Pettersen and Bullard, 2003).

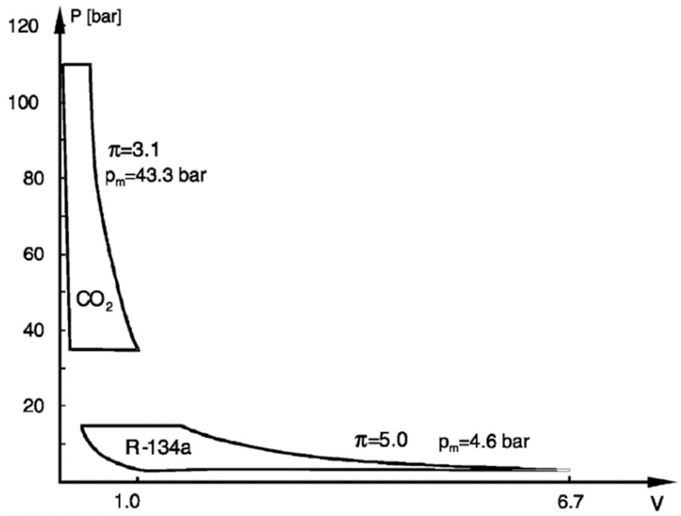

Figure 2.5: Pressure/volume diagram that compares R744 and R134a compressor capacity (Kim, Pettersen and Bullard, 2003)

The high difference in suction and discharge pressure in the compressor can lead to leakage and pressure loss; in this case the pressure difference between suction and discharge pressure can in some cases reach 150bar. As a consequence there can be leakage in the compressor. However, with a proper design the leakage can be reduced significantly. A reciprocating compressor is best suited for R744 and its high pressure, this is because it is easier to improve the sealing of the cylinder with a sealing ring. The high difference between suction and discharge pressure results in high stress on drive mechanisms and bearings, this can be reduced by using a two stage compressor or oil free bearings (Reulens, 2009).

The compressor can be used in one or two stage compression, one stage is the most common in a R744 system. By applying two stage compressions it is possible to save energy if the compression work is large (Kim, Pettersen and Bullard, 2003).

Main types of compressors that are designed for the high pressures that occur in transcritical R744 heat pumps (Reulens, 2009):

- Reciprocating (piston) compressor
  - Hermetic reciprocating compressors
  - Semi-hermetic reciprocating compressors
- Rotary compressor

2.2.2.2 Gas cooler

Gas coolers are basic component in a refrigerant system and are very important for achieving good energy efficiency and heat transfer. There is not a great difference between heat exchanger used in subcritical cycles and in transcritical cycles, the main difference is that it needs to be resist a much higher operational pressure.

The traditional finned coils with copper tubes and continues aluminium finnes area the most used and widespread gas coolers. The success is related to good cost/performance ratio, high reliability and a flexibility of design. However, to achieve an even better heat transfer the use of flat multiport minichannel (figure 2.6) gas cooler will increase the area on the refrigerant side with three and the pressure drop will be reduces (Reulens, 2009) and (Kim, Pettersen and Bullard, 2003).

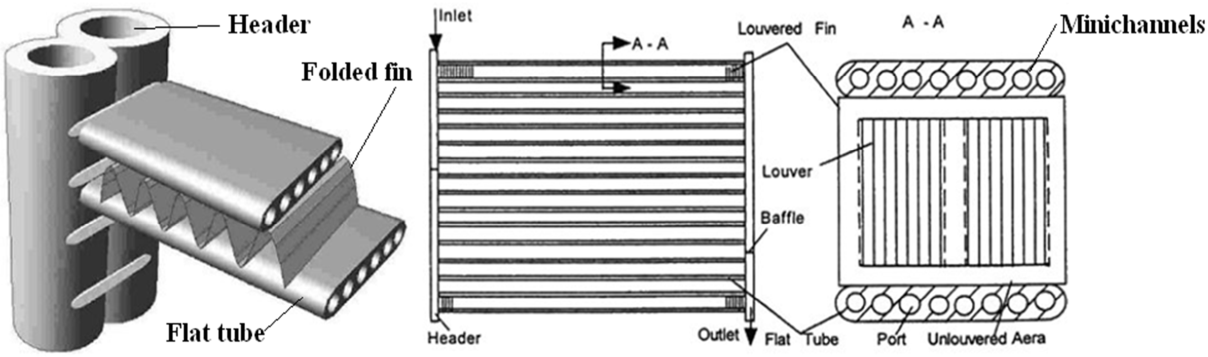

Figure 2.6: A possible design for a flat multiport minichannel gas cooler (Reulens, 2009) (edited)

2.2.2.3 Throttling valve

The throttling valve in a subcritical cycle has a different role than in a transcritical cycle, in a transcritical system it acts as a mass flow rate controller to ensure a balance of performance between the compressor, gas cooler and evaporator. It is therefore important that the throttling valve is properly sized if the cycle shall operate correctly and without affecting the performance of the heat pump.

Back pressure valve is the most used throttling valve used in transcritical cycles. In this valve the position of the stem is controlled by the upstream pressure that is in contact with an adjustable spring: the actions of the valve keeps a constant pressure at the gas cooler outlet and the valve reacts to an increase of the upstream pressure by increasing the flow. This valve is able to control the upper cycle pressure. The most used solution is showed in figure 2.7, it shows a liquid receiver after the evaporator to protect the compressor. This arrangement provides flooded conditions in the evaporator with a certain mass flow rate of refrigerant. Thereby this is continuously evaporated inside the liquid receiver to produce cooling of the dense gas before the throttling valve. The liquid receiver has another function, i.e. it acts as storage for allowing the charge transfer due to variations of the void fraction inside the components of the refrigerant system (Reulens, 2009).

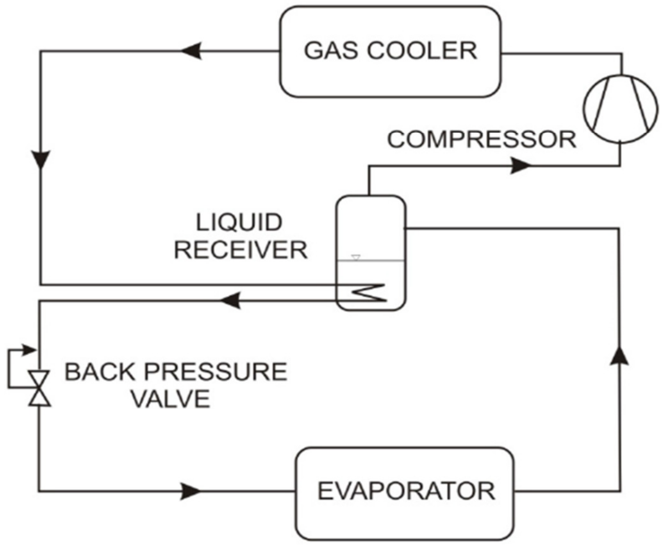

Figure 2.7: Back pressure valve enable to keep flooded in the evaporator (Reulens, 2009)

2.2.2.4 Evaporator

Multiport minichannel finned tube in aluminium have been used for many years in heat exchangers, mainly in the automotive industry. There has been much interest in using this in transcritical R744 cycles. These heat exchangers uses flat tubes through profiles and fines are fixed to the plain side of the tube. The fines are usually formed in a V form between two contiguous tubes, the fines are often louvered so they achieve a high heat transfer coefficient at moderate air velocity. The different features of minichannel heat exchanger are illustrated in figure 2.8. The different profiles on the inside of the contiguous tubes give different rates of heat transfer (Reulens, 2009).

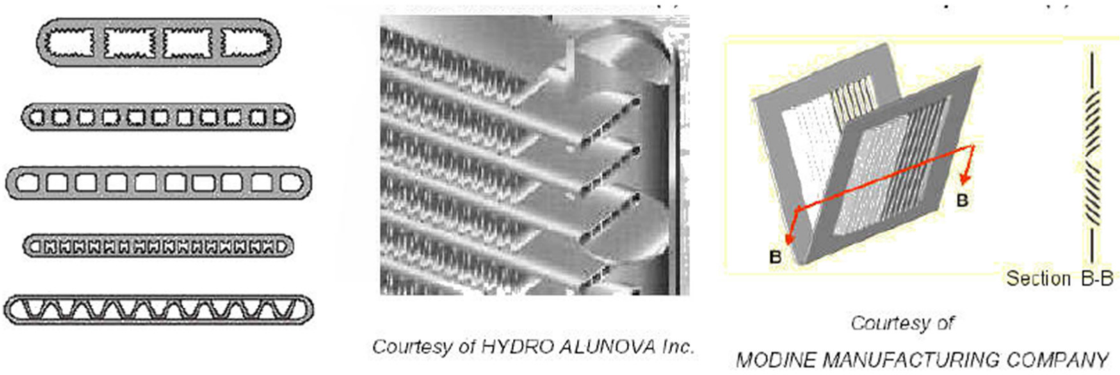

Figure 2.8: Illustration of different minichannel ports in heat exchangers (Reulens, 2009)

2.2.3 HFC emissions

The most common refrigerants used in heat pumps to day uses are HFC, these refrigerants have a large GWP compared to the natural refrigerants (R290, R717 and R744). The HFC refrigerant has a GWP that are 1000 – 3800 times larger than for the natural refrigerants (table 1.1). Figure 2.9 shows that the releases of R134a to the atmosphere are increasing annually. More use of R134a is the same as more release of emissions to the atmosphere. One of the largest contributor to the emission of R134a refrigerant are mobile air-conditions (MAC) in cars, a German studies showed that the annual emission of R134a was 10.2% of MAC system in the car (Reulens, 2009).

By using natural refrigerant it is possible to reduce the greenhouse gas emissions, it is a small step in the right direction to reduce the emissions to the atmosphere.

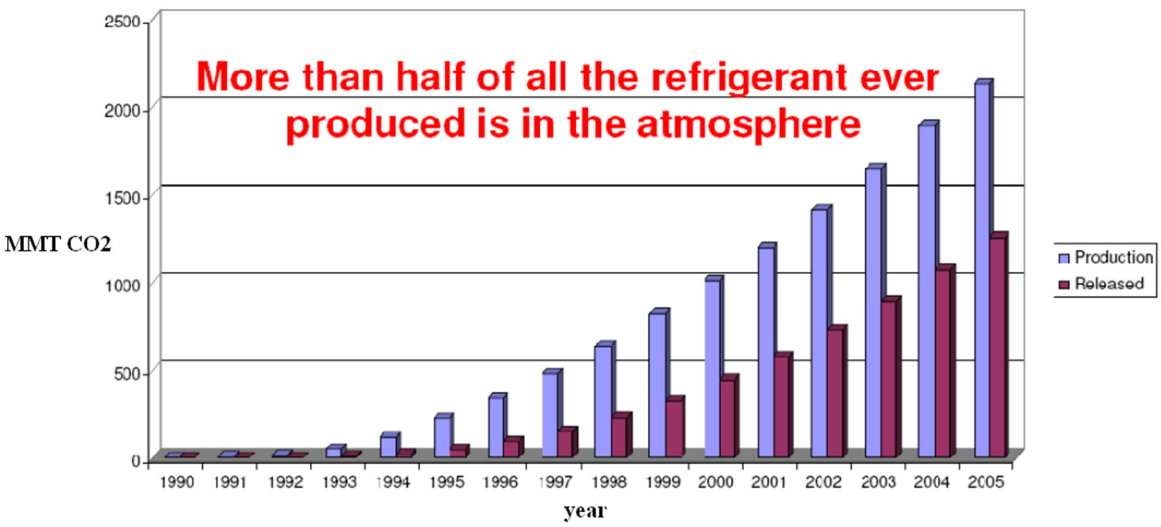

Figure 2.9: Global production and release of R134a in million Metric tons of R744 equivalent (Reulens, 2009)

2.3 Safety issues related to R744

2.3.1 Air

R744 is not flammable, it is used as a fire-fighting medium or a fire-preventing medium (Kim, Pettersen and Bullard, 2003). R744 is not considered as a toxic gas, the natural concentration of the gas in the air is 0.038% (Stene, 2013). With a the concentration of 3% the breading rate will increase, at 4% it will be an immediate danger for life and 10% is the lowest reported lethal concentration of R744 in the air per volume (Kim, Pettersen and Bullard, 2003). R744 will displace the air in low laying areas or basements because it is 2.2 times heavier than air at 20 ℃ (Stene, 2013). If the heat pump is installed at a place where there is a possibility that the R744 will displace the air it should be installed a gas alarm, similar to those installed in caravans.

2.3.2 Compressed air

The pressure in a R744 system can be up to 15 times higher than for conventional refrigerants, this makes this type of heat pumps more "scary" for the common man. Because of the high pressure in the system it is important to have sufficient safety measures to prohibit that this high pressure will do damage to properties or persons. Some of the safety measures can be (Kim, Pettersen and Bullard, 2003):

- Use of a blowout disk on the high and low pressure side. Release pressure on high side (gas cooler) can be 160 bar and on the low pressure side (evaporator) can be 90 bar.
- Components in the system needs to be design to handle a pressure that are at least two times the pressure of the blowout disks, in both high and low pressure side. The best would be that the components handle three times the blowout disk pressure.
- There shall be conducted proof of integrity shall be carried out against; bursting, failure by fatigue and failure by vibration.

2.3.3 Explosion

Explosion is the sudden release of energy stored in a system or component. This can occur if there is a malfunction in operations, construction weakness, safety failure, etc. If an explosion occurs, there is rapid release of high pressure gas (in the case of heat pumps, it is high pressure refrigerant R744) and possibility of parts fragments flying around. Such incident can have serious implication on safety of property and personnel. The extent of damage is dependent on components volume, operating pressure and refrigerants.

The energy released in explosion varies between refrigerants. For example, R22 releases ½ of the energy as R744 at room temperature, but at a temperature of 150 ºC the explosion energy is 2 times higher for R22. However, energy released in a MAC from R134a is approximately the same as R744 ranges around 80 kJ. The amount of energy rises significantly if refrigerant is flammable such as R290 (46,000 kJ) (Kim, Pettersen and Bullard, 2003).

More information about general safety in relation to heat pumps use in cold climate is given in chapter 2.1.2.

2.4 Choosing of refrigerant and mass flow

2.4.1 Refrigerant

The reason for choosing R744, R717 and R290 is because they are natural refrigerants that will become more popular in the future and because of the low GWP, and because HFC refrigerants with a GWP above 150 will be phased out in the future (Eikevik, 2013). These natural refrigerants has a GWP of maximum 3, this is relatively low compared with HFC refrigerants like R410A that has a GWP of 2000 (table 1.1). The reason for using a HFC in this thesis is to compare it to the natural refrigerants. R410A has the highest COP at low temperature when comparing it with R404A (a 44/52/4 mixture of HFC: R125, R143A (Trifluoroethane – CF3CH3) and R134a), R407C (a 23/25/52 mixture of HFC: R32, R125 and R134a) and R134a (table 2.3). Thereby R410A will be the best suited to be compared with the natural refrigerants (file: Thesis-one stage – Kolsaker/Appendix: A).

Table 2.3: COP (including heating of domestic water) comparison between several synthetic refrigerants by using MS Excel® sheet (Kolsaker, 2013)

Evaporation temperature [ºC]	R410A COP	R404A COP	R407C COP	R134a COP
15	3.61	3.09	3.56	3.43
10	3.43	2.95	3.36	3.25
5	3.26	2.82	3.18	3.09

0	3.10	2.70	3.02	2.95
-5	2.95	2.59	2.87	2.81
-10	2.81	2.48	2.73	2.69
-15	2.68	2.38	2.60	2.57
-20	2.57	2.29	2.49	2.46
-25	2.46	2.20	2.38	2.36
-30	2.35	2.11	2.28	2.27
-35	2.26	2.03	2.19	2.18
-40	2.17	1.96	2.10	2.10

2.4.2 Mass flow

The R744 refrigerant has been calculated from the "CO_2 heat pump" MS Excel® sheet for the pre-master thesis work (file: CO_2 heat pump/Appendix: A). In the MS Excel® sheet the R744 is designed to heat up 320 l/day for heating of domestic water and 2000 l/day for heating of hydronic heating (mass flow of 0.0233 kg/s for R744).

The mass flow from MS Excel® was implemented in CoolPack© to find the heating capacity of the gas cooler (figure 2.10), this value is 6.73 kW. This value is implemented in the condenser value in CoolPack© for the current refrigerants to find the mass flow. The results are then; R717 are 0.0048 kg/s, R290 are 0.018 kg/s and for R410A are 0.0293 kg/s (file: Mass flow – R744, R717, R410A and R290/Appendix: A).

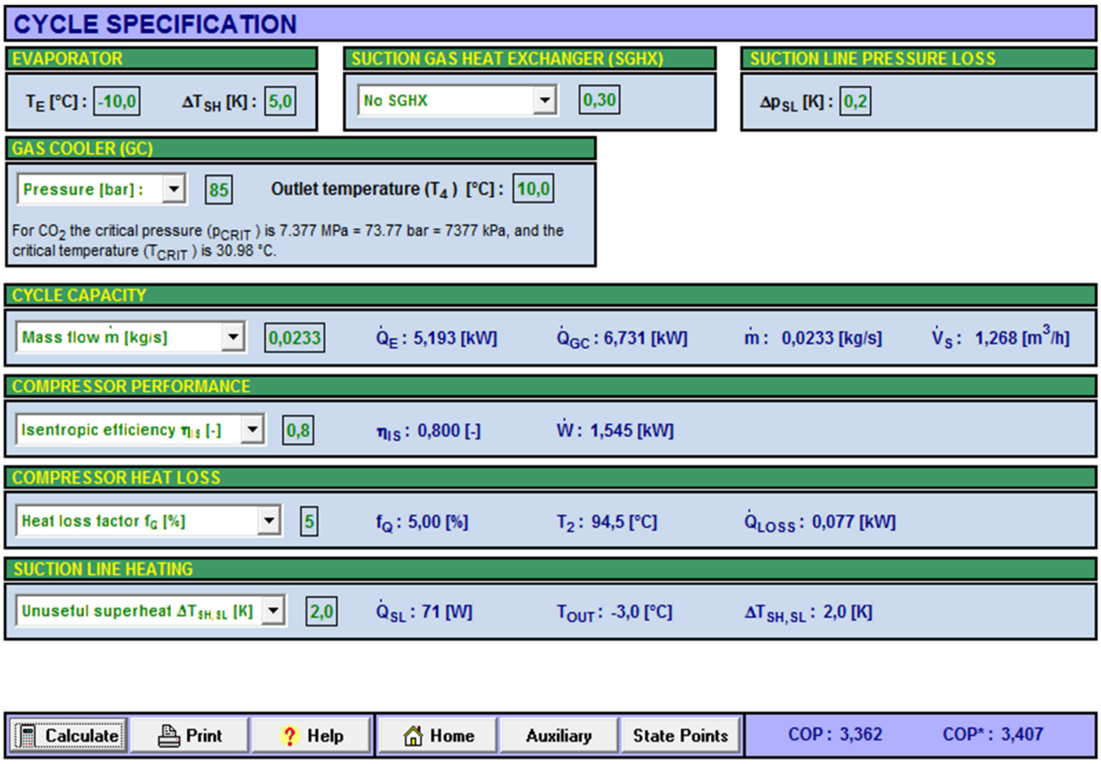

Figure 2.10: Example of calculation of mass flow in CoolPack©

3. Methodology

The manufacturers of heat pumps are continuous improving the heat pump technology so they can achieve a higher COP and that the heat pumps would be able to operate at an even lower outside temperature than today. The most important data regarding heat pump use in colder climate is the size of the house, heat loss, outside temperature and the domestic water usage.

3.1 Data analysis of outside temperatures

Temperature data have been collected from eKLIMA (2014) (file: Temperatures in Karasjok 2012 – 2013/Appendix: A). The temperatures data from January 2012 to December 2013 have been collected. Temperature data from eKLIMA (2014) are in table 3.1. The air temperature (TA) was measured daily; at 01.00 (TA 01), 07.00 (TA 07), 13.00 (TA 13) and 19.00 (TA 19). Table 3.1 also show the average (TAM), the maximum (TAX) and the minimum (TAN) temperatures recorded each day. The data below the recorded temperatures consists of the lowest recorded temperature (Lowest), the highest recorded temperature (Highest), the average temperature (Average) and the normal temperature for each month, shown in table 3.1 with red scripture.

Table 3.1: A shortened (days: 1.-7. and 27.-29.) table of data collected from eKLIMA (2014)

KARASJOK-MARKANNJARGA February 2012							
Day	TA 01	TA 07	TA 13	TA 19	TAM	TAX	TAN
1	-18,5	-22,7	-18,6	-21,2	-20,5	-18,3	-29,8
2	-20,5	-23,4	-24,9	-23,9	-23,4	-19,7	-27,9
3	-23,9	-25,3	-21,7	-19,7	-23,2	-19,7	-27,1
4	-25	-25,6	-27	-29,5	-28,1	-19,6	-34
5	-34,9	-26,4	-28,4	-29,5	-30,1	-26	-35,9
6	-34	-36,9	-35,4	-39,2	-36,5	-29,5	-39,2
7	-34,1	-35,1	-31,8	-29,8	-31,9	-29,8	-39,4
---	---	---	---	---	---	---	---
27	-20,8	-17,3	-11,9	-10,7	-13,8	-10,7	-21,1
28	-7,1	-6,3	-4,8	-4,2	-5,6	-3,4	-10,9
29	-12	-17,4	-5,8	-16,3	-13	-4,2	-18

Numbers	29	29	29	29	29	29	29
Lowest	-34,9	-36,9	-35,4	-39,2	-36,5	-29,8	**-39,4**
Date	5	6	6	6	6	7	7
Highest	0,9	-1,5	-1,2	-1,2	-1,1	**1,6**	-6
Date	11	11	11	11	11	11	19
Sum							
Average					**-16,4**	-10,6	-23,6
Normal					-15,4		
Deviation					-1		

3.1.1 Lowest temperatures

To make this graph the lowest measured temperature each month in 2012 and 2013 was selected and implemented (figure 3.1). As the graph indicates the lowest measured temperature in this period was -41.3 ℃ in January 2013 and the highest measured temperature was 1.1 ℃ in June 2012. Lowest measured temperature in Norway occurred in Karasjok in 1886, the measured temperature was -51.4 ℃ (Meteorologisk institutt, 2013a).

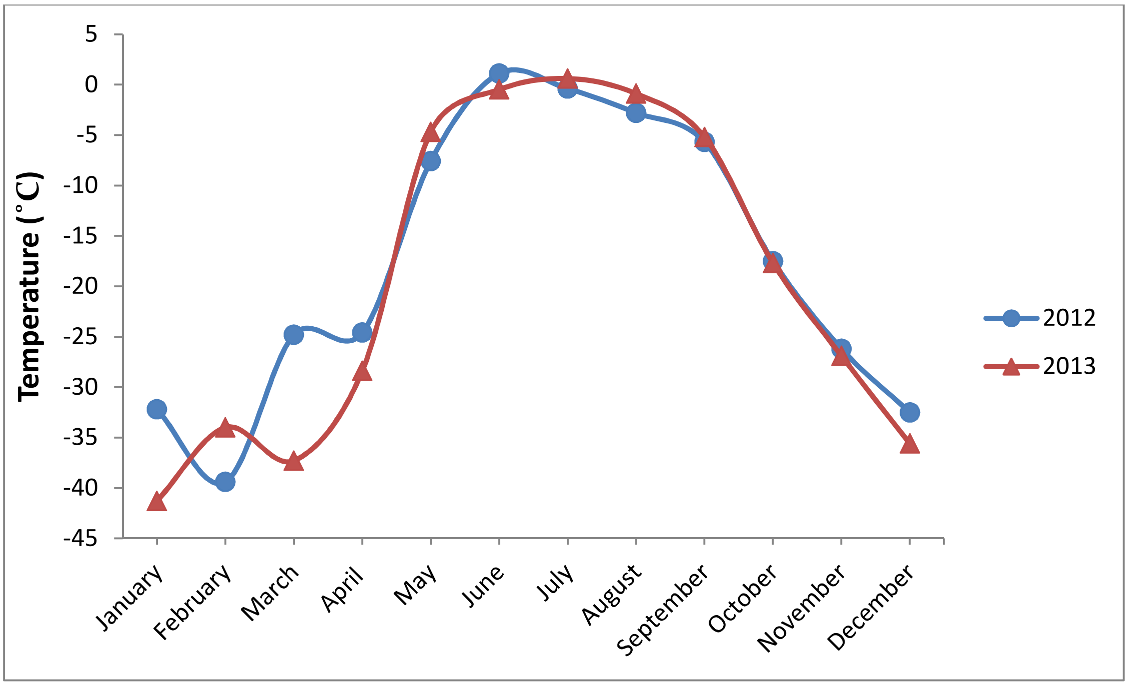

Figure 3.1: Lowest measured temperature per month in Karasjok in 2012 and 2013

3.1.2 Average temperature

Figure 3.2 shows the average temperatures measured each month in 2012 and 2013. As seen in the graph the months with the lowest average measured temperature were February, December of 2012 and March 2013 with a temperature of -16.4 ℃, -15.8 ℃ and -14.2 ℃. This gives a good indication of which temperatures the heat pump needs to be able to work within. The annual average temperature in Karasjok is usually blow 0 ℃, it is just some few years where the temperature is above 0 ℃ (Meteorologisk institutt, 2013b).

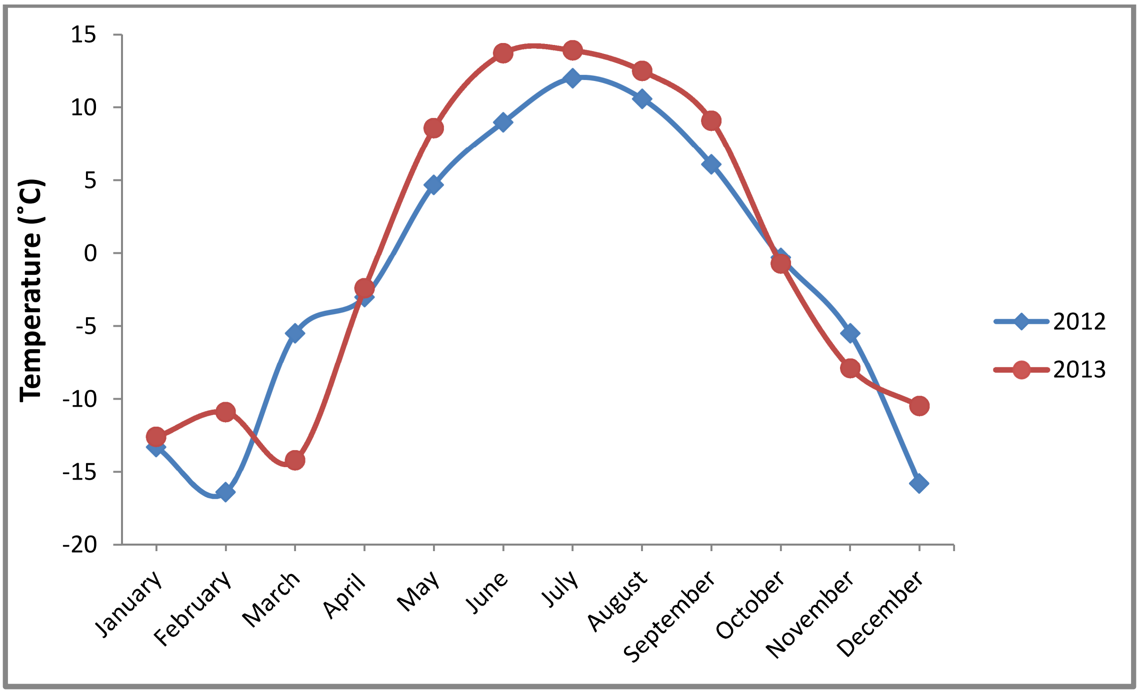

Figure 3.2: Average temperature each month in Karasjok in 2012 and 2013

3.1.3 Normal temperature

The normal temperature is the average temperature that has been registered over a certain 30 year period, the graph in figure 3.3 shows the period from 1961 – 1990 in Karasjok. This is done to avoid that extreme weather shall impact the value and corrupt the normal temperature value (Meteorologisk institutt, 2010). The normal temperature gives a respectable indication of which temperatures the heat pump need to work with throughout its lifetime. Heat pumps are able to operate up to 12 years and sometimes even longer, depending on performed maintenance and operation condition (Aftenposten, 2013).

During a year the temperature difference in Karasjok can be relative large, it has been measured to be up to 83.3 ℃ (Meteorologisk institutt, 2013a).

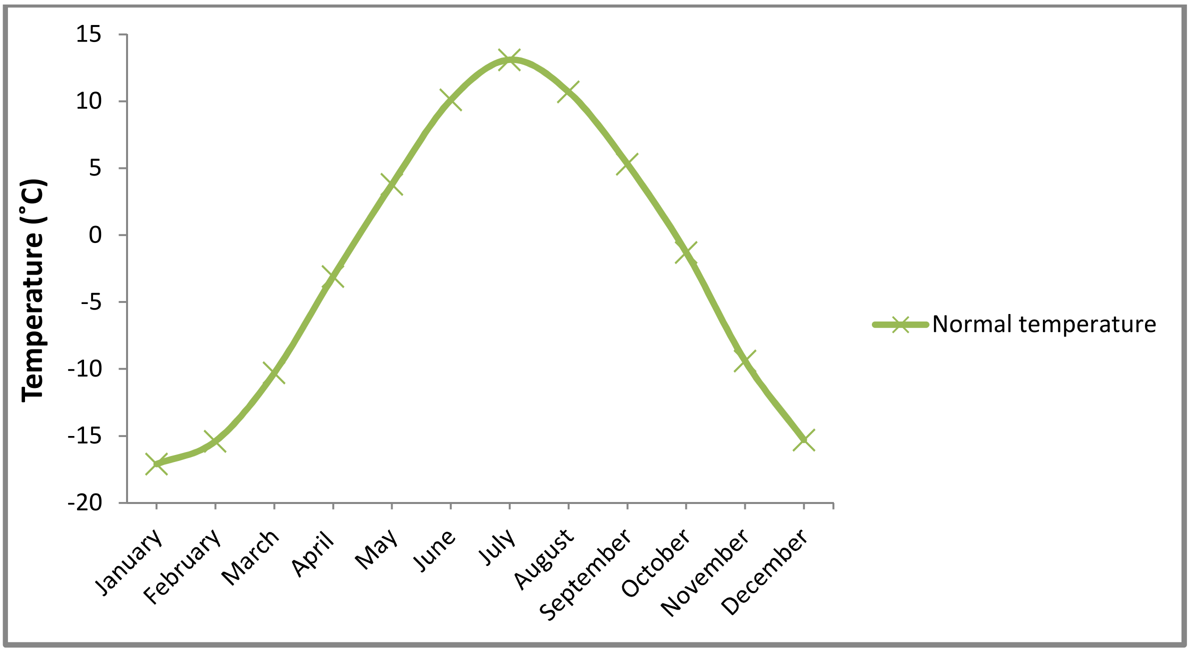

Figure 3.3: Annually normal temperature in Karasjok

3.1.4 Overview of the low temperatures

Figure 3.4 and 3.5 show how many days in 2012 and 2013 the temperature was below -20 ºC. The different columns indicate the various temperatures and how many days the temperature lasted. In 2012 it was registered 12 days and 2013 it was registered 19 days of temperatures below -30 ºC. Temperatures registered between -25 ºC and -30 ºC was in 2012 20 days and in 2013 it was 23 days. The spreading of the different temperatures and how many days it lasts varies from year to year.

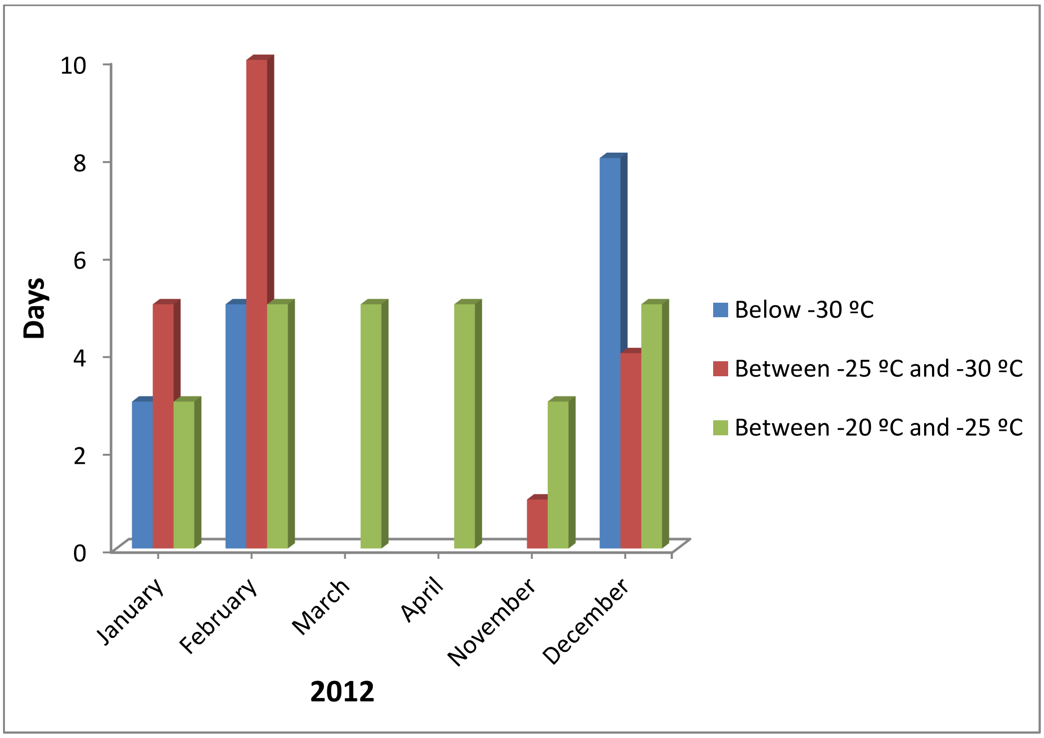

Figure 3.4: Registered temperatures below -20 ºC in 2012

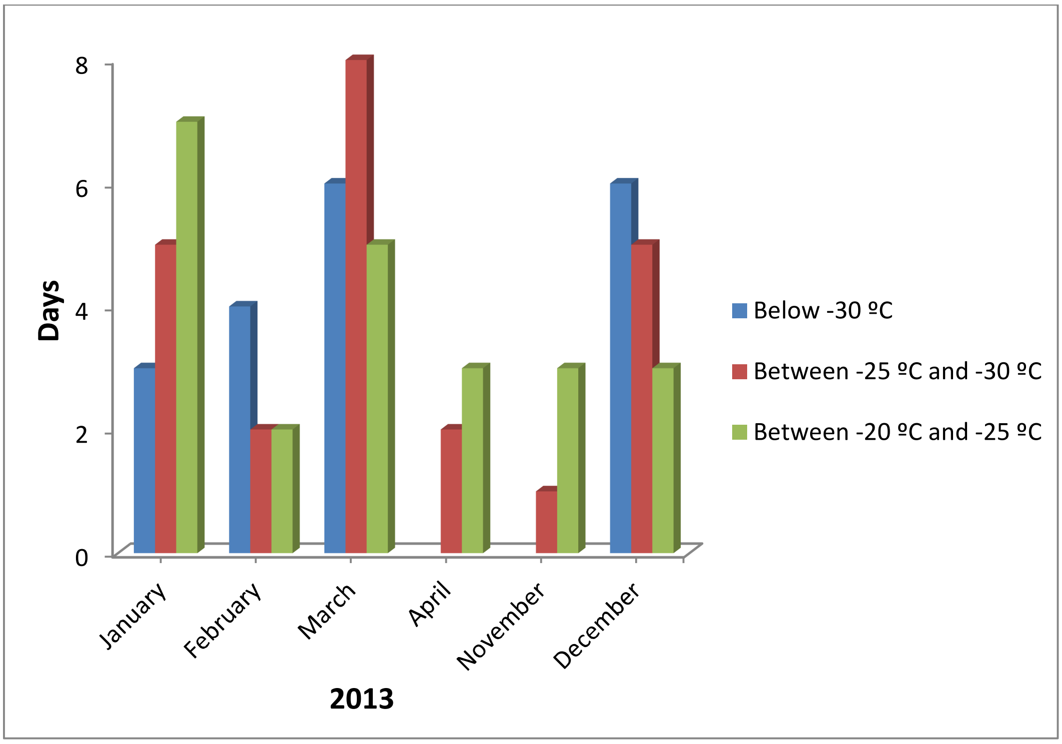

Figure 3.5: Registered temperatures below -20 ºC in 2013

3.2 Electric consumption

The electricity consumption in the house in Karasjok in 2012 and 2013 is presented in figure 3.6 (file: House calculation – Electric use now/Appendix: A). When looking at the electricity consumption in 2012 the highest consumption was in the winter time and the lowest was in June to September, this is what can be expected in a household. In 2013 the peak in the consumption in March, this can be because low temperature and/or more use of domestic hot water.

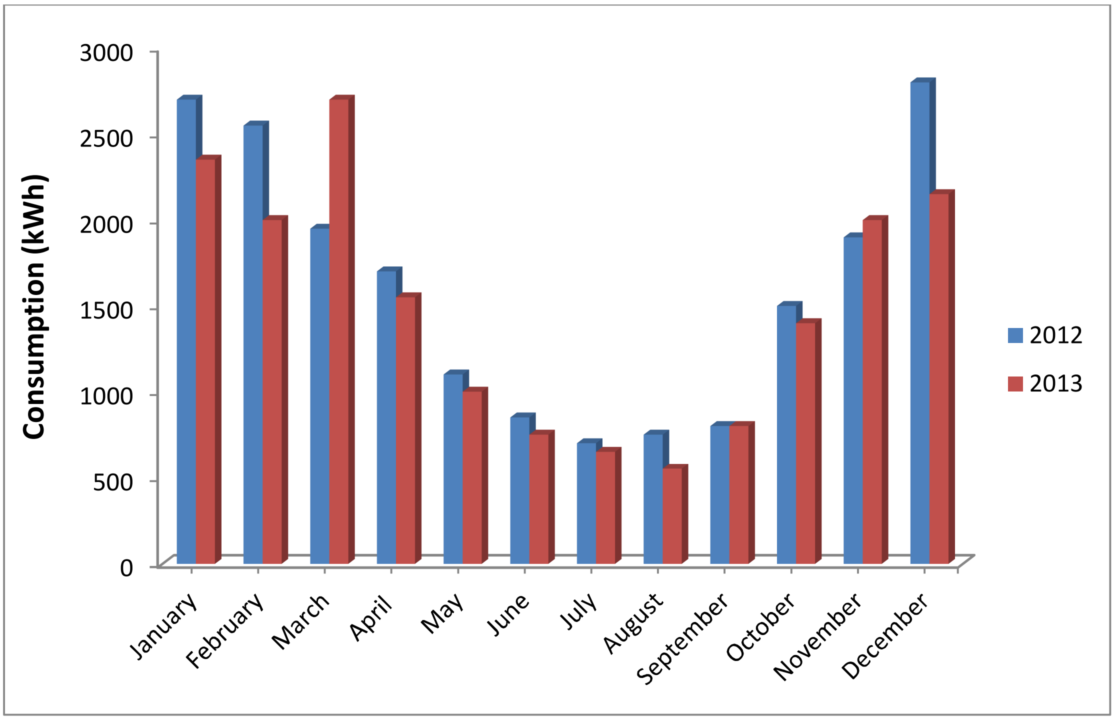

Figure 3.6: Annually electricity consumption (Luostejok Kraftlag, 2014)

Figure 3.7 shows the electricity consumption and the registered average temperature each month in 2012 (figure 3.2). The annual consumption in 2012 was 19300 kWh. The consumption and temperature follow each other, except in February. This irregularity can be caused by more usage of electricity for heating that fossil fuel in February.

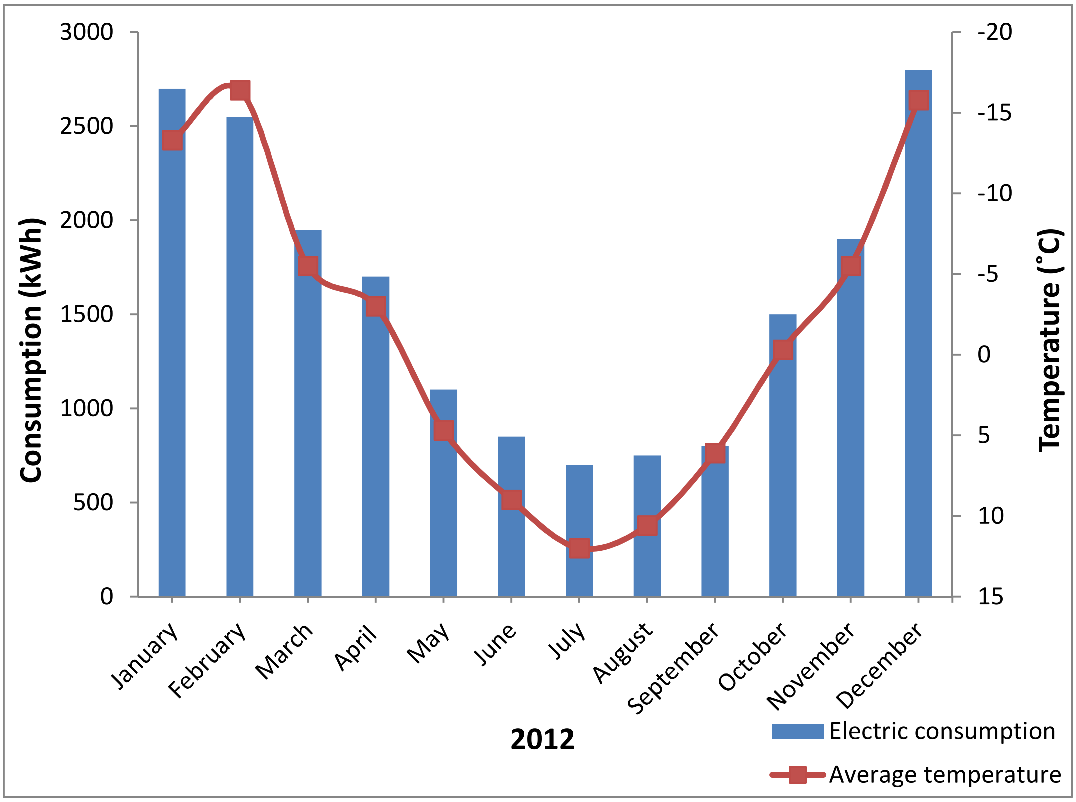

Figure 3.7: Electricity consumption and average temperature in 2012

In figure 3.8 the graph shows the electricity consumption and the registered average temperature each month in 2013 (figure 3.2). The annual consumption in 2013 was 17900 kWh, which is less than in 2012. The lower consumption can be because of a higher average temperature in 2013. The electricity consumption and temperature follow each other throughout the graph ass it should.

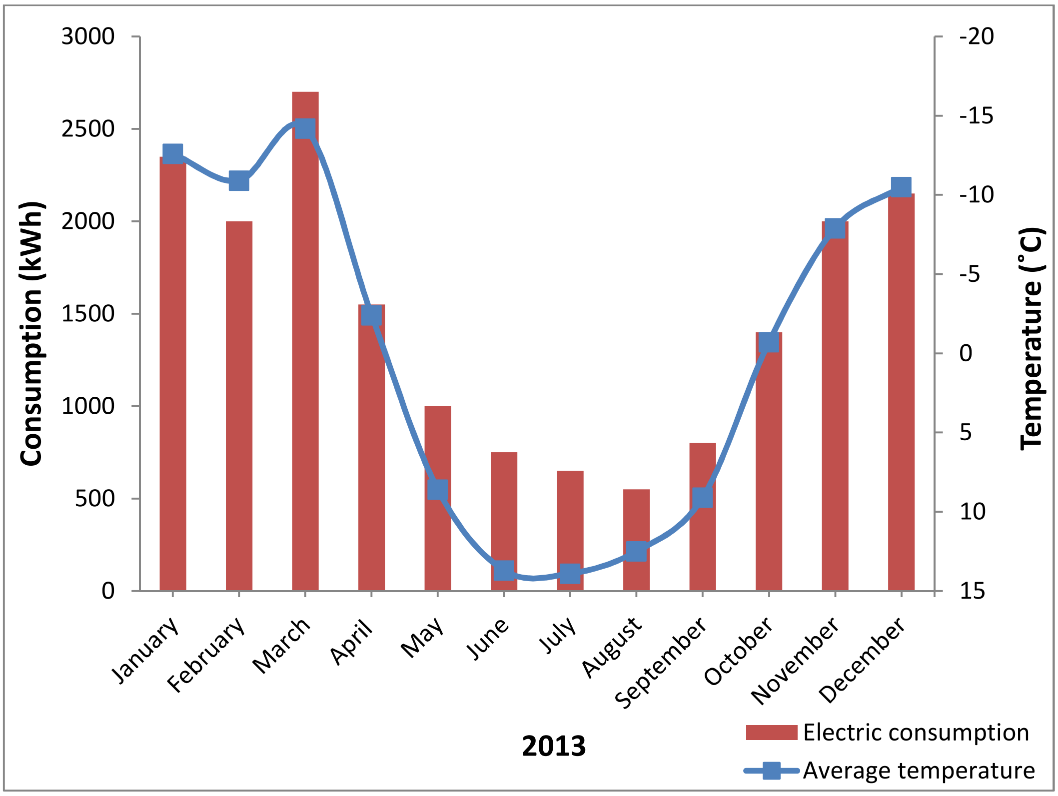

Figure 3.8: Electricity consumption and average temperature in 2013

3.3 Simulations

The two simulation alternatives that have been used are the simulation program CoolPack© and MS Excel® sheet (Kolsaker, 2013), in CoolPack© the log p-h diagram have also been used frequently. By using these alternatives it is easy to confirm the results that it generated in CoolPack© and MS Excel® sheet (Kolsaker, 2013). During the simulations in CoolPack© and MS Excel® sheet (Kolsaker, 2013) the only values that was changed manually during each simulation was the evaporation temperature, this was done in both the single stage and two stage compressions. Thereby the values that that changed during the simulations was the gas cooler/condenser effects, compressor effect and the discharge temperature from the compressor.

MS Excel® sheet (Kolsaker, 2013) had only simulation in single stage compression, hence it was need to build a two stage compression simulation in this MS Excel® sheet. The two stage simulation was compared with a log p-h diagram simulation and the difference in COP was 7.72% (table 3.2) (file: Intermediate and Kolsaker – Confirm Kolsaker/Appendix: A).

Table 3.2: Verification of the two stage compression in MS Excel® sheet (Kolsaker, 2013)

	CoolPack©	Log p-h diagram	Excel® (Kolsaker, 2013)	CoolPack© versus Excel® (Kolsaker, 2013) [%]	Log p-h diagram versus Excel® (Kolsaker, 2013) [%]
Compressor 1	1.40 kW	1.68 kW	1.68 kW	16.90	0.14
Compressor 2	1.30 kW	0.78 kW	1.09 kW	18.76	28.74
Condenser	10.1 kW	8.24 kW	8.56 kW	18.00	3.70
COP	3.74	3.34	3.08	17.67	7.72

3.4 House calculations

To calculate the heat loss in the current house in Karasjok, there has been used exact value (table 3.3). The total heat loss was calculated to be 264.81 W/K, hence the annual energy needed to cover the heat loss is 27181.95 kWh/year (table 3.3). This gives a heating need for 166.86 m^2/year. The details of these results have been produced in file: House calculation – Heat loss/Appendix: A.

Table 3.3: Detailed calculation of heat loss in the house (VVSforum, 2014), (Enova, 2013), (Sintef, 2009a) and (Sintef, 2009b)

Windows	Area [m^2]	U-value [W/m^2*K]	[W/K]
6	0,4	2,4	5,76
2	1,8	2,4	8,64
5	1,08	2,4	12,96
4	1,08	2,4	10,37
Doors			
1	2,1	2,4	5,04
2	1,9	2,4	9,12
Walls			
South side	18,48	0,3	5,54
North side	16,98	0,3	5,09
East side	28,83	0,3	8,65
Vest side	24,67	0,3	7,40

Foundation			
South side	17,83	0,8	14,26
North side	17,43	0,8	13,94
East side	17,43	0,8	13,94
Vest side	18,23	0,8	14,58
Floor			
1	63,5	0,3	19,05
Roof			
2	55,25	0,2	22,10
Thermal bridges			
	162,9	0,05	8,15
Infiltration		[Wh/m^3*K]	
441,95	154,68	0,33	51,05
Ventilation		[Wh/m^3*K]	
	88,39	0,33	29,17
Total heat loss			**264.81**
Energy use to cover heat loss			**27181.95 kWh/year**

The result from table 3.3 makes it possible to calculate the heating need in the house at a chosen outside and inside temperature. Table 3.4 show these calculations (file: House calculation – Heat loss/Appendix: A).

Table 3.4: Energy need at different outside temperatures for heating of the house (VVSforum, 2014)

Energy need at different temperatures		
Outside temperature [°C]	**Inside temperature [°C]**	**Energy need at this temperature difference [kW]**
15	22	1,85
10	22	3,18
5	22	4,50
0	22	5,83

-5	22	7,15
-10	22	8,47
-15	22	9,80
-20	22	11,12
-25	22	12,45
-30	22	13,77
-35	22	15,09

4. Results

In this chapter there will be presented information and results about one stage and two stage compression heat pump, how to calculate COP (equation 1 – 3 and 5 – 7) and how the seasonal performance factor (SPF) is calculated (equation 4). Produced results from simulations in CoolPack©, log p-h diagram and MS Excel® sheet (Kolsaker, 2013) are presented in the graphs and tables. There will also be presented results about how much electricity it is possible to save by using a heat pump instead of only electricity, and how much of the heating and DWH the heat pump is able to cover at low outside temperatures.

4.1 One stage compression

The graphs and tables show how much power the compressor need at different temperatures, how much effect the gas cooler/condenser can produce at the different evaporation temperatures and how the COP decrease as the temperature decreases for the current refrigerants. CoolPack©, log p-h diagram and MS Excel® sheet (Kolsaker, 2013) has been used to calculate the different values.

The calculation of COP can be done in two different ways, equation 1 uses the h-values collected from log p-h diagram and equation 2 uses the effect of compressor and gas cooler/condenser. Figure 4.1 show were the h-value it reed form a log p-h diagram (1-2-3-4). The figure also show that if the R744 heat pump only produces DWH to point 3' the COP will become rather low and if it produces DWH to point 3'' the COP will be even better. Thereby the COP will be largest if it can produce hot water to point 3 in figure 4.1.

$$COP = \frac{h_2 - h_3}{h_2 - h_1} = \frac{(473 - 216)kJ/kg}{(473 - 420)kJ/kg} = 4.85 \qquad (1)$$

$$COP = \frac{P_{gas\ cooler/condenser}}{P_{Compressor}} = \frac{7\ kW}{1.5\ kW} = 4.67 \qquad (2)$$

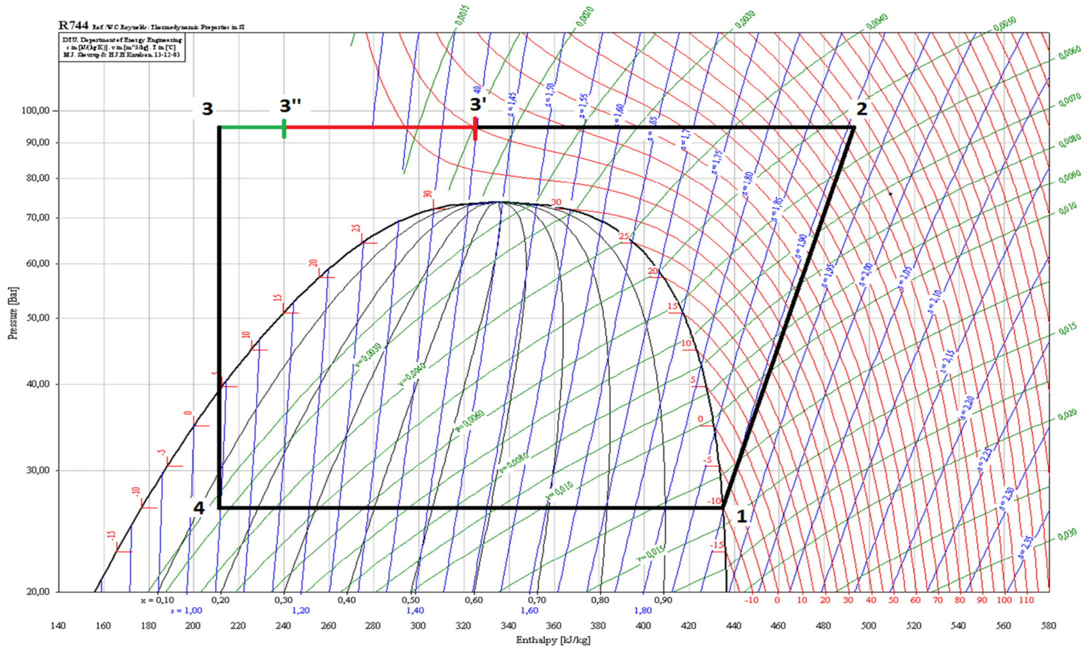

Figure 4.1: Were to read the h-value (1-2-3-4)

Because the R744 heat pump shall heat up the house and the domestic water, the other refrigerant (R290, R410A and R717) needs to have implemented a 2 kW electric heater that is used for DWH in the total COP calculation (equation 3).

$$COP = \frac{(7 + 2)\ kW}{(1.5 + 2)\ kW} = 2.57 \tag{3}$$

Because of the low difference between log p-h diagram and MS Excel® sheet (Kolsaker, 2013), only an average of 0.23% and 0.27% for R290 without and with DWH included in the COP (table 4.1) (file: Thesis-One stage – Percent difference/Apendix: A). Hence, it is not necessary to have both log p-h diagram and the MS Excel® sheet (Kolsaker, 2013) in chapter 4.1.1 – 4.1.4.

Table 4.1: Comparison of R290 COP between log p-h diagram and MS Excel® sheet (Kolsaker, 2013)

R290	**Log p-h diagram**		**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**COP**	**COP with DWH**	**COP**	**COP with DWH**	**COP diff. [%]**	**COP with DWH diff. [%]**
15	3.75	17.21	3.77	17.28	0.53	0.41
10	3.53	12.30	3.55	12.36	0.56	0.49
5	3.34	9.55	3.35	9.58	0.30	0.31
0	3.16	7.77	3.17	7.80	0.32	0.38
-5	3.00	6.55	3.01	6.56	0.33	0.15
-10	2.85	5.64	2.85	5.65	0.00	0.18
-15	2.71	4.95	2.72	4.96	0.37	0.20
-20	2.58	4.40	2.59	4.41	0.39	0.23
-25	2.47	3.96	2.47	3.97	0.00	0.25
-30	2.36	3.60	2.36	3.60	0.00	0.00
-35	2.26	3.29	2.26	3.30	0.00	0.30
-40	2.17	3.03	2.17	3.04	0.00	0.33

4.1.1 R744

The fixed values that have been used:

$\dot{m}_{R744}$ – 0.0233 kg/s

Heat loss factor – 5%

Isentropic efficiency of the compressor – 80%

Gas cooler pressure – 85 bar

Gas cooler output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

From the graph in figure 4.2 the cross point is almost at -10 ºC, here the COP is 4.27 and the compressor effect is 1.58 kW. The highest COP is 9.63 and the highest compressor effect is 3.20 kW. At the highest COP the gas cooler has an effect of 5.52 kW and at the highest compressor effect the condenser has an effect of 8.24 kW.

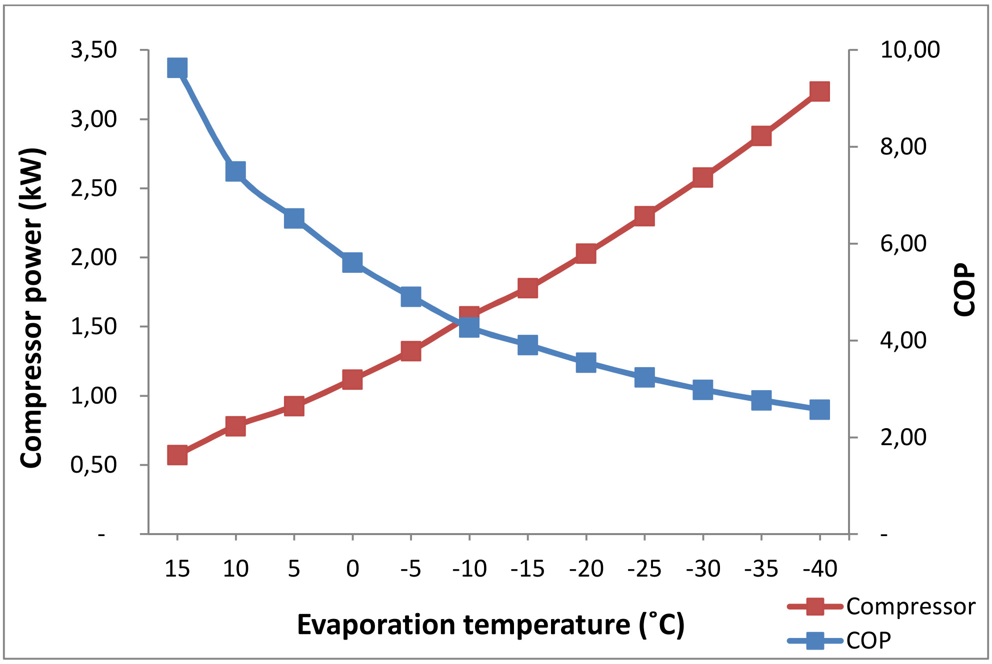

Figure 4.2: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using R744 in CoolPack©

Table 4.2 show the effects of the compressor and gas cooler, and the COP value from simulations in CoolPack© and log p-h diagram. The difference between the CoolPack© and the log p-h diagram results becomes lower ass the evaporation temperature reduces. The highest difference in COP is 18.25% and the average difference is 4.56%. The log p-h diagram has the highest COP for all the evaporation temperatures. At an evaporation temperature of -40 ºC the COP is 2.59, compressor effect is 3.26 kW and the gas cooler effect is 8.45 kW.

Table 4.2: Values from CoolPack© and log p-h diagram for R744

	CoolPack©				**Log p-h diagram**			
Evaporation temperature [ºC]	**Comp. effect [kW]**	**Gas c. effect [kW]**	**COP**		**Comp. effect [kW]**	**Gas c. effect [kW]**	**COP**	**COP diff. [%]**
15	0.57	5.52	9.63		0.43	5.05	11.78	18.25
10	0.78	5.86	7.49		0.64	5.35	8.41	10.94
5	0.93	6.05	6.53		0.79	5.68	7.20	9.31
0	1.12	6.28	5.61		0.95	5.82	6.12	8.33
-5	1.33	6.50	4.91		1.18	6.12	5.20	5.58

-10	1.58	6.73	4.27		1.44	6.40	4.45	4.04
-15	1.78	6.96	3.91		1.70	6.70	3.94	0.76
-20	2.03	7.20	3.55		1.98	7.03	3.55	0.00
-25	2.30	7.45	3.24		2.35	7.44	3.16	-2.53
-30	2.58	7.70	2.98		2.66	7.80	2.94	-1.36
-35	2.88	7.97	2.77		2.87	8.03	2.80	1.07
-40	3.20	8.24	2.58		3.26	8.45	2.59	0.39

4.1.2 R717

The fixed values that have been used:

$\dot{m}_{R717}$ – 0.0048 kg/s

Heat loss factor – 5%

Isentropic efficiency of the compressor – 70%

Condensation temperature – 28 ºC

Condensation output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

From the graph in figure 4.3 the cross point is slightly above -5 ºC, here the COP is 6.27 and the compressor effect is 1.10 kW. The highest COP is 15.75 and the highest compressor effect is 3.00 kW. At the highest COP the condenser has an effect of 6.30 kW and at the highest compressor effect the condenser has an effect of 8.83 kW.

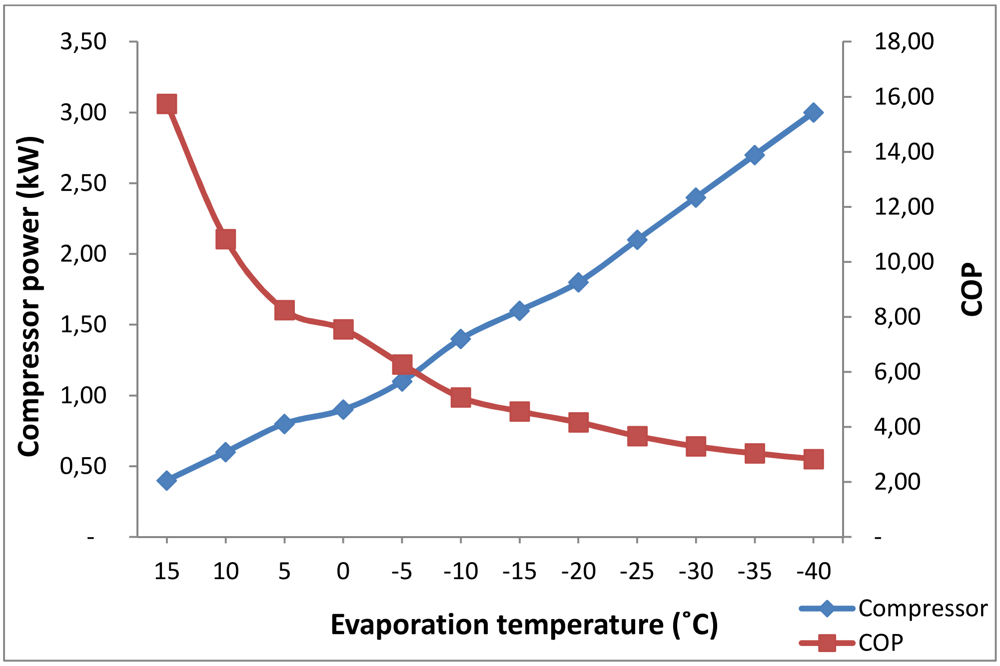

Figure 4.3: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using R717 in CoolPack©

Table 4.3 show the effects of the compressor and condenser, and the COP results from simulations in CoolPack© and log p-h diagram. The largest difference is at an evaporation temperature of 5 ºC, the difference is 10.52% and the average difference is 4.42%. The log p-h diagram simulation gave the highest COP, at an evaporation temperature of -40 ºC the COP is 2.91, compressor effect is 2.92 kW and the condensation effect is 8.52 kW.

Table 4.3: Values from CoolPack© and log p-h diagram for R717

	CoolPack©				**Log p-h diagram**			
Evaporation temperature [ºC]	**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
15	0.40	6.30	15.75		0.38	6.31	16.61	5.18
10	0.60	6.50	10.83		0.54	6.45	11.89	8.92
5	0.80	6.60	8.25		0.72	6.60	9.22	10.52
0	0.90	6.80	7.56		0.90	6.76	7.51	-0.67
-5	1.10	6.90	6.27		1.10	6.92	6.32	0.79
-10	1.40	7.10	5.07		1.31	7.11	5.44	6.80
-15	1.60	7.30	4.56		1.53	7.30	4.77	4.40

-20	1.80	7.50	4.17		1.77	7.51	4.24	1.65
-25	2.10	7.70	3.67		2.03	7.73	3.82	3.93
-30	2.40	7.90	3.29		2.30	7.97	3.46	4.91
-35	2.70	8.20	3.04		2.60	8.24	3.16	3.80
-40	3.00	8.50	2.83		2.92	8.52	2.91	2.75

When implementing the DWH in the COP calculation, this gives a relatively lower COP results ass seen in figure 4.4 and table 4.4. The COP is 3.49 at an evaporation temperature of 15 ºC and at an evaporation temperature of -40 ºC the COP is 2.14 in the log p-h diagram simulation (table 4.4).

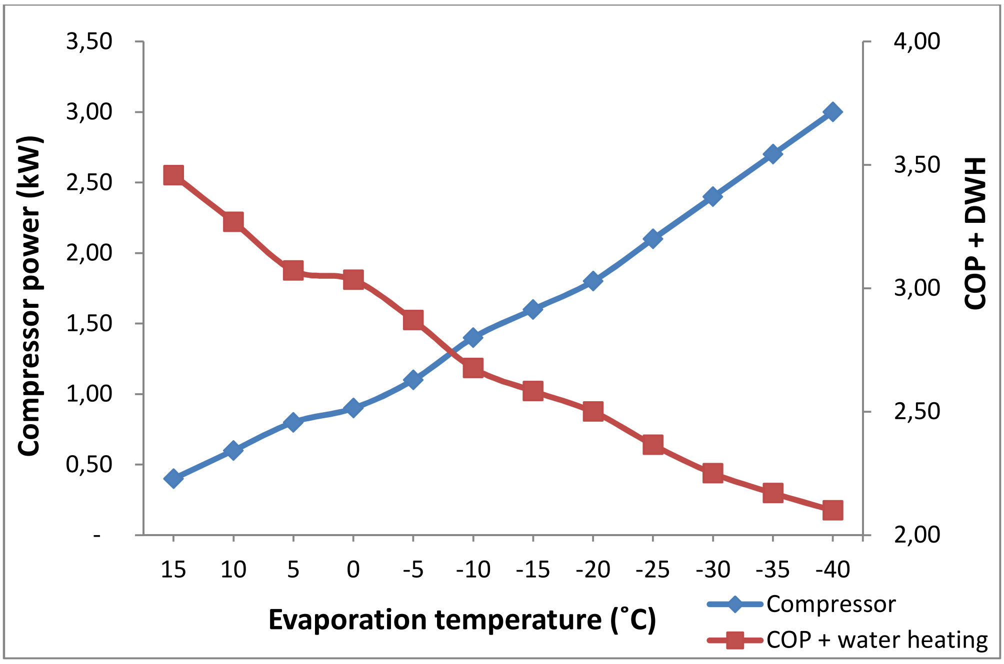

Figure 4.4: Compressor effect (kW) and COP + DWH plotted against evaporation temperature (ºC) using R717 in CoolPack©

Table 4.4: COP difference when DWH is implemented

Evaporation temperature [ºC]	CoolPack© COP	Log. COP		COP diff. CoolP. and log.
15	3.46	3.49		0.86
10	3.27	3.32		1.51
5	3.07	3.17		3.15

0	3.03	3.02		-0.33
-5	2.87	2.88		0.35
-10	2.68	2.75		2.55
-15	2.58	2.63		1.90
-20	2.50	2.52		0.79
-25	2.37	2.42		2.07
-30	2.25	2.32		3.02
-35	2.17	2.22		2.25
-40	2.10	2.14		1.87

4.1.3 R290

The fixed values that have been used:

$\dot{m}_{R290}$ – 0.018 kg/s

Heat loss factor – 5%

Isentropic efficiency of the compressor – 70%

Condensation temperature – 28 ℃

Condensation output temperature – 10 ℃

Evaporators superheat (ΔT_{SH}) – 2 K

From the graph in figure 4.5 the cross point is at -5 ℃, here the COP is 6.17 and the compressor effect is 1.20 kW. The highest COP is 17.75 and the highest compressor effect is 2.80 kW. At the highest COP the condenser has an effect of 7.95 kW and at the highest compressor effect the condenser has an effect of 9.11 kW.

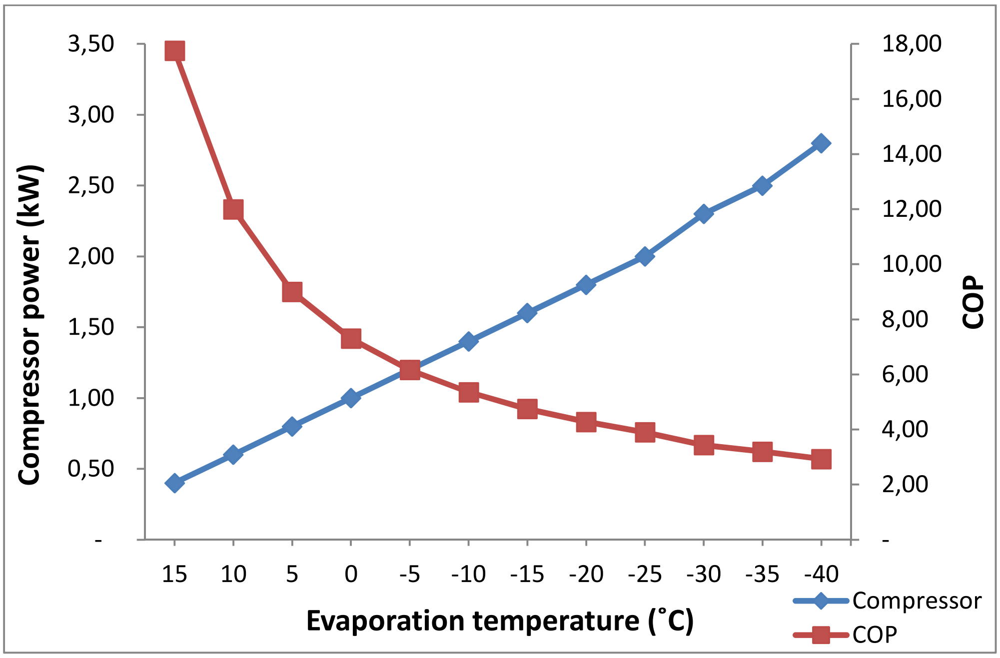

Figure 4.5: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using R290 in CoolPack©

Table 4.5 show the effects of the compressor and condenser, and the COP results from simulations in CoolPack© and log p-h diagram. The largest difference is at an evaporation temperature of 0 ºC, the difference is 6.05% and the average difference is 3.93%. The log p-h diagram simulation gave the highest COP, at an evaporation temperature of -40 ºC the COP is 3.03, compressor effect is 2.71 kW and the condensation effect is 8.23 kW.

Table 4.5: Values from CoolPack© and log p-h diagram for R290

	CoolPack©				Log p-h diagram			
Evaporation temperature [ºC]	**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
15	0.40	7.10	17.75		0.41	7.03	17.21	-3.14
10	0.60	7.20	12.00		0.58	7.11	12.30	2.44
5	0.80	7.20	9.00		0.75	7.19	9.55	5.76
0	1.00	7.30	7.30		0.94	7.28	7.77	6.05
-5	1.20	7.40	6.17		1.13	7.37	6.55	5.80
-10	1.40	7.50	5.36		1.32	7.46	5.64	4.96
-15	1.60	7.60	4.75		1.53	7.57	4.95	4.04

-20	1.80	7.70	4.28		1.74	7.68	4.40	2.73
-25	2.00	7.80	3.90		1.97	7.80	3.96	1.52
-30	2.30	7.90	3.43		2.20	7.93	3.60	4.72
-35	2.50	8.00	3.20		2.45	8.07	3.29	2.74
-40	2.80	8.20	2.93		2.71	8.23	3.03	3.30

Figure 4.6 and table 4.6 show the values when DWH is implemented in the calculations. The COP is 3.79 at an evaporation temperature of 15 ºC is 3.79 with CoolPack© and at an evaporation temperature of -40 ºC the COP is 2.17 in the log p-h diagram simulation (table 4.6).

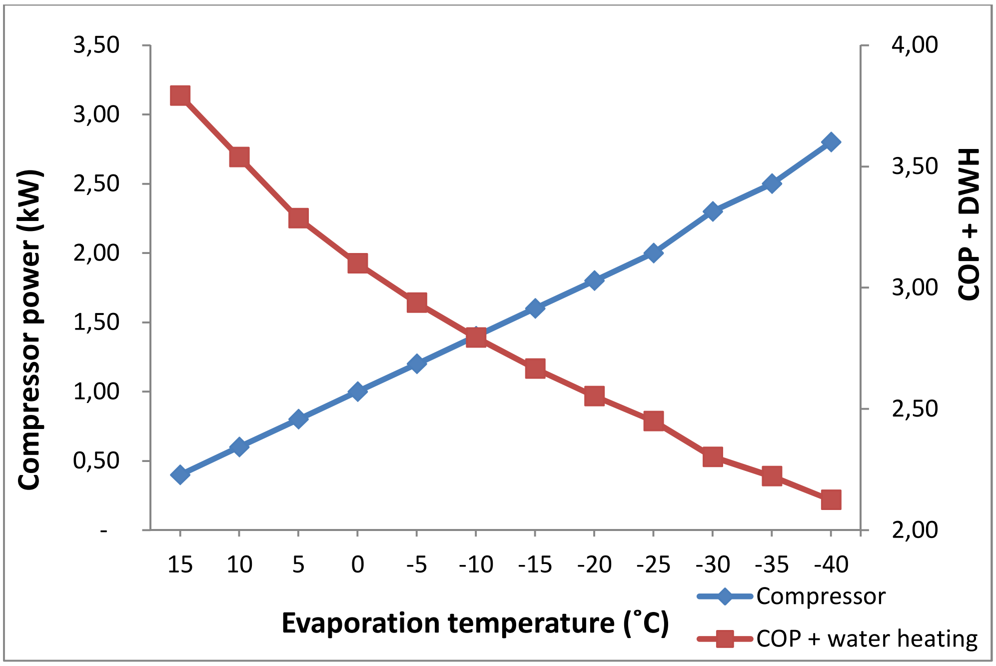

Figure 4.6: Compressor effect (kW) and COP + DWH plotted against evaporation temperature (ºC) using R290 in CoolPack©

Table 4.6: COP difference when DWH is implemented

Evaporation temperature [ºC]	CoolPack© COP	Log. COP		COP diff. CoolP. and log.
15	3.79	3.75		-1.07
10	3.54	3.53		-0.28

5	3.29	3.34		1.50
0	3.10	3.16		1.90
-5	2.94	3.00		2.00
-10	2.79	2.85		2.11
-15	2.67	2.71		1.48
-20	2.55	2.58		1.16
-25	2.45	2.47		0.81
-30	2.30	2.36		2.54
-35	2.22	2.26		1.77
-40	2.13	2.17		1.84

4.1.4 R410A

The fixed values that have been used:

$\dot{m}_{R410A}$ – 0.0293 kg/s

Heat loss factor – 5%

Isentropic efficiency of the compressor – 70%

Condensation temperature – 28 ºC

Condensation output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

From the graph in figure 4.7 the cross point is -5 ºC, here the COP is 6.00 and the compressor effect is 1.20 kW. The highest COP is 16.75 and the highest compressor effect is 2.90 kW. At the highest COP the condenser has an effect of 6.65 kW and at the highest compressor effect the condenser has an effect of 8.38 kW.

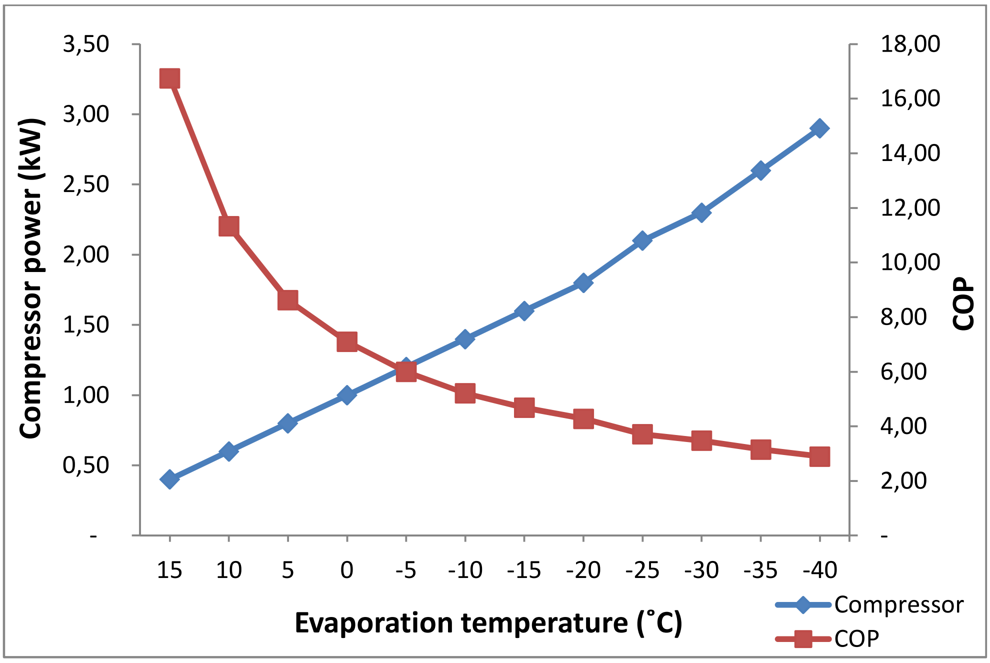

Figure 4.7: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using R410A in CoolPack©

Table 4.7 show the effects of the compressor and condenser, and the COP results from simulations in CoolPack© and log p-h diagram. The largest difference is when the evaporation temperature is 0 ºC, the difference is 6.82% and the average is 3.89%. The log p-h diagram simulation gave the highest COP, at an evaporation temperature of -40 ºC the COP is 2.97, compressor effect is 2.82 kW and the condensation effect is 8.39 kW.

Table 4.7: Values from CoolPack© and log p-h diagram for R410A

	CoolPack©				Log p-h diagram			
Evaporation temperature [ºC]	**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
15	0.40	6.70	16.75		0.40	6.73	16.88	0.77
10	0.60	6.80	11.33		0.57	6.85	12.05	5.98
5	0.80	6.90	8.63		0.74	6.97	9.37	7.90
0	1.00	7.10	7.10		0.93	7.10	7.62	6.82
-5	1.20	7.20	6.00		1.13	7.24	6.40	6.25
-10	1.40	7.30	5.21		1.34	7.37	5.52	5.62
-15	1.60	7.50	4.69		1.55	7.52	4.84	3.10

-20	1.80	7.70	4.28		1.78	7.68	4.30	0.47
-25	2.10	7.80	3.71		2.02	7.84	3.87	4.13
-30	2.30	8.00	3.48		2.28	8.01	3.52	1.14
-35	2.60	8.20	3.15		2.54	8.19	3.22	2.17
-40	2.90	8.40	2.90		2.82	8.39	2.97	2.36

Figure 4.8 and table 4.8 show the results when DWH is implemented in the calculation. The COP is 3.64 at an evaporation temperature of 15 ºC and at an evaporation temperature of -40 ºC the COP is 2.15 in the log p-h diagram simulation (table 4.8).

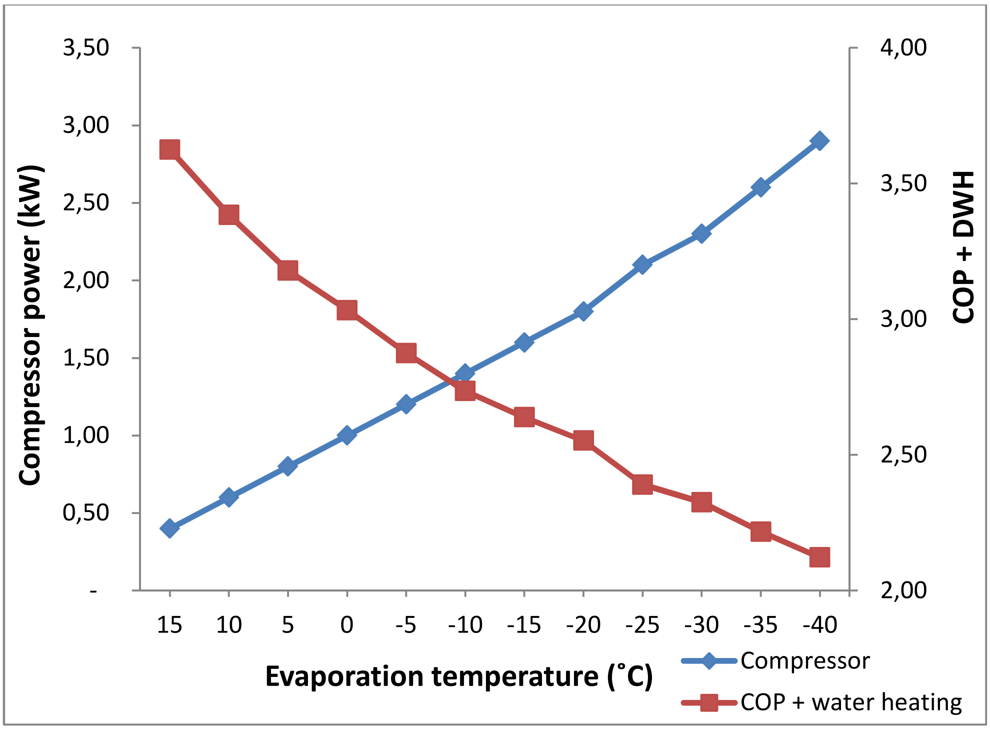

Figure 4.8: Compressor effect (kW) and COP + DWH plotted against evaporation temperature (ºC) using R410A in CoolPack©

Table 4.8: COP difference when DWH is implemented

Evaporation temperature [ºC]	CoolPack© COP	Log. COP		COP diff. CoolP. and log.
15	3.63	3.64		0.27
10	3.38	3.45		2.03
5	3.18	3.27		2.75

0	3.03	3.10		2.26
-5	2.88	2.95		2.37
-10	2.74	2.81		2.49
-15	2.64	2.68		1.49
-20	2.55	2.56		0.39
-25	2.39	2.44		2.05
-30	2.33	2.34		0.43
-35	2.22	2.24		0.89
-40	2.12	2.15		1.40

4.1.5 SPF

SPF is the measurement for the average annual COP, thereby considering the variation in the outside temperature. The formula for SPF takes to account the output effect (gas cooler/condenser) and input effect (compressor, fans, controls, de-icing) (equation 4) (Heat pump association, 2014). In the SPF calculation the inside fan, outside fan, the control and the de-icing had effects of 0.04 kW, 0.06 kW, 0.005 kW and 0.5 kW. Compressor effect for the various temperatures is implemented in the calculation. The de-icing is implemented from October to April and the fan and control are implemented throughout the year. The de-icing is used these months because the average temperature is below 0 ℃.

When doing the simulations the evaporation temperature is set to be 5 ° lower than the registered outside temperature in figure 3.3.

$$SPF = \frac{Heat\ pump\ output\ effect\ [kW]}{Heat\ pump\ input\ effects\ [kW]} \qquad (4)$$

The basic for the calculation of the SPF is the normal temperature in Karasjok (chapter 3.1.3). Figure 4.9 – 4.12 show the normal temperature, COP and the SPF for the current refrigerants (file: Thesis-One stage – SPF/Appendix: A). Figure 4.9 show that the SPF for R744 is 4.11.

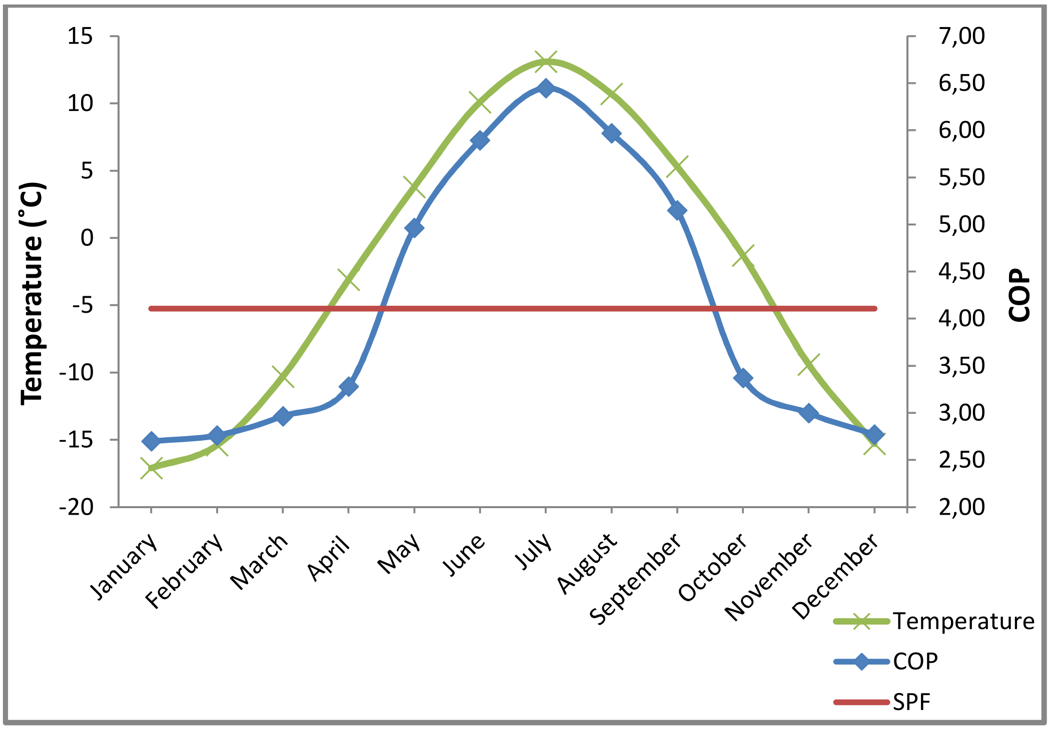

Figure 4.9: Normal outside temperature (ºC) each month, COP and SPF for R744

Figure 4.10 show that the SPF for R717 is 2.51.

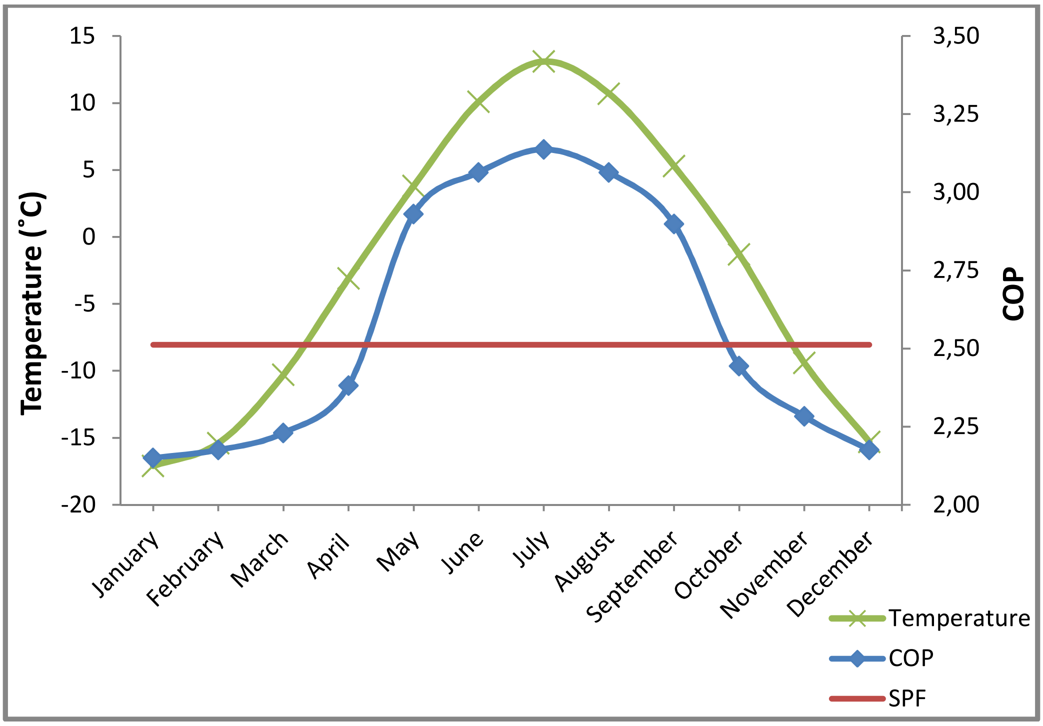

Figure 4.10: Normal outside temperature (ºC) each month, COP and SPF for R717

Figure 4.11 show that the SPF for R290 is 2.60.

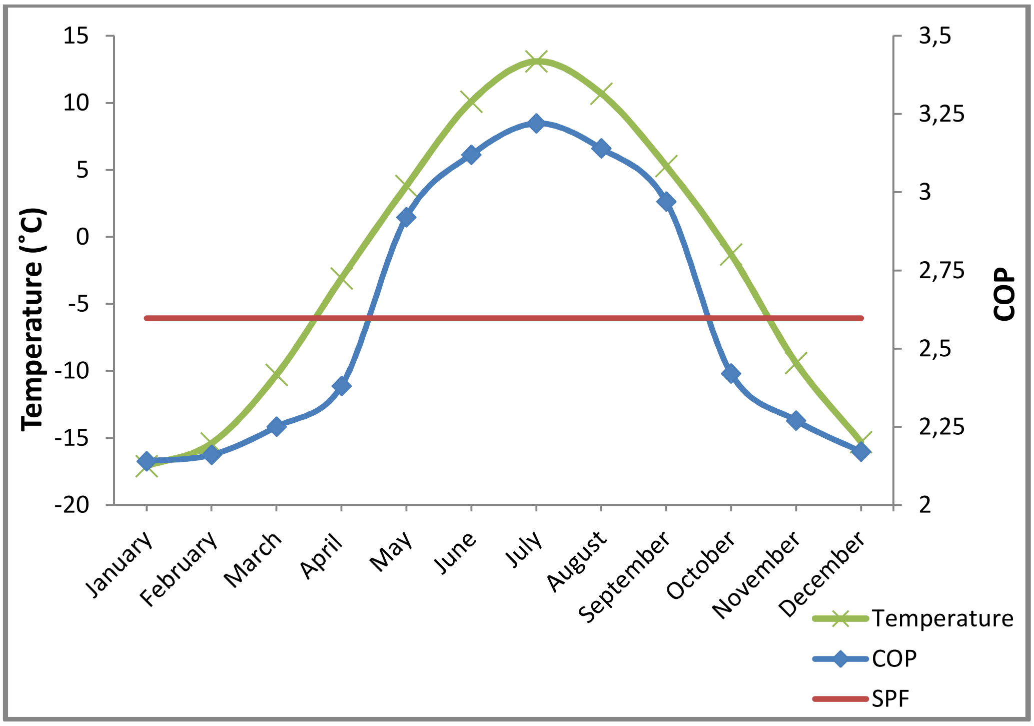

Figure 4.11: Normal outside temperature (ºC) each month, COP and SPF for R290

Figure 4.12 show that the SPF for R410A is 2.54.

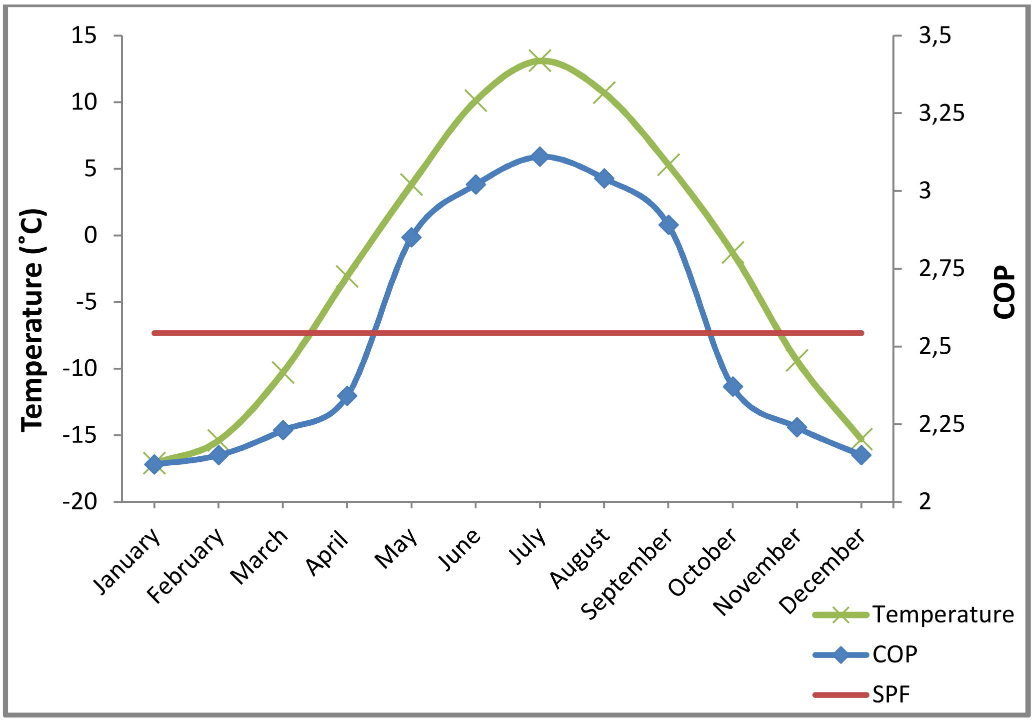

Figure 4.12: Normal outside temperature (ºC) each month, COP and SPF for R410A

4.2 Two stage compression

The difference between a one stage and a two stage cycle is that in a two stage cycle the compressor works is in two stages, one low pressure and one high pressure cycle (figure 4.13). It is also possible to achieve a higher COP with a two stage compared with a one stage cycle, because of less compressor work. Figure 4.14 illustrates how a two stage cycle can look like in a heat pump. The meanings of the different stages in a two stage compression cycle are:

- **A-B** is low pressure compression
- **B-C** the refrigerant gets cooled in the open intercooler
- **C-D** is high pressure compression
- **D-E** is the gas cooler/condenser were the heat gets realised
- **E-F** is the throttling valve, the refrigerant changes stage from liquid to liquid/gas or the gas gets cooled down
- **F-G** is where the refrigerant becomes more liquid
- **G-H** is further cooling of the refrigerant
- **H-A** is the evaporation

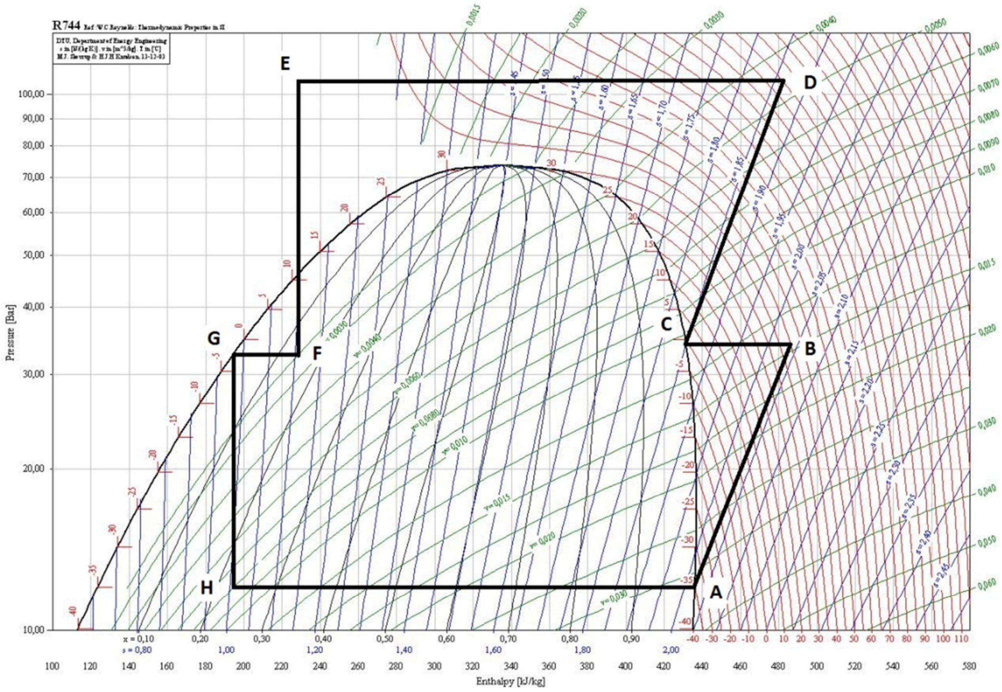

Figure 4.13: Two stage R744 heat pump cycle in a log p-h diagram

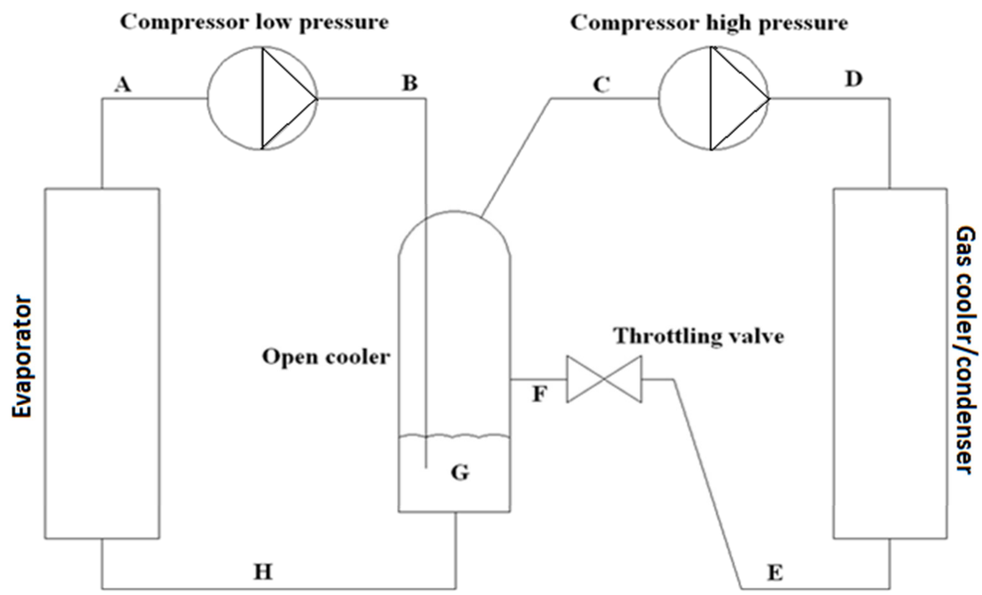

Figure 4.14: Illustration of a two stage cycle as shown in figure 4.13

Equation 5 is used for calculating the COP for a two stage cycle with an open intercooler. Values to h_A to h_H (enthalpy) is collected form a diagram like in figure 4.13. $\dot{m}_{gas\ cooler/condenser}$ and $\dot{m}_{evaporator}$ is the mass flow in the gas cooler and evaporator.

$$COP = \frac{P_{gas\ cooler/condenser}}{P_{Compressor}} = \frac{P_{gas\ cooler/condenser}}{P_{LT}+P_{HT}}$$

$$= \frac{\dot{m}_{gas\ cooler/condenser}(h_D - h_E)}{\dot{m}_{evaporator}(h_B - h_A) + \dot{m}_{gas\ cooler/condenser}(h_D - h_C)} \tag{5}$$

Equation 6 shows how mass conservation can be used for calculating the COP for a two stage cycle with the different enthalpies from a log p-h diagram (h_A to h_H) (figure 4.13). $\dot{m}_C$ is the mass flow of in state C and x_F is the liquid percent in state F in figure 4.13.

$$\dot{m}_{evaporator} = \dot{m}_{gas\ cooler/condenser} - \dot{m}_C$$

$$= \dot{m}_{gas\ cooler/condenser} - (1 - x_F)\dot{m}_{gas\ cooler}$$

$$= \dot{m}_{gas\ cooler/condenser} \cdot x_F$$

$$x_F = \frac{\dot{m}_{evaporator}}{\dot{m}_{gas\ cooler/condenser}} = \frac{h_C - h_F}{h_C - h_G} \tag{6}$$

Equation 7 is the end formula for calculating the COP for the two stage cycle.

$$COP = \frac{(h_D - h_E)}{x_F(h_B - h_A) + (h_D - h_C)} \tag{7}$$

Because of the low difference between log p-h diagram and two stage MS Excel® sheet (Kolsaker, 2013) the log p-h diagram values has not been implemented in chapter 4.2.1 – 4.2.4. The difference between the two simulations for R290 has an average of 2.46% and 3.59% (table 4.9) (file: Thesis-Two stage – Percent difference/Appendix: A). In chapter 4.2.1 it has been used log p-h diagram because MS Excel® sheet (Kolsaker, 2013) does not handle transcritical R744. The intermediate temperature for all of the refrigerants have been set to 5 ℃, at this intermediate temperature the COP was highest for the R290 and R744 (file: Intermediate and Kolsaker – Best intermediate/Appendix: A).

Table 4.9: Comparison of R290 COP between log p-h diagram and MS Excel® sheet (Kolsaker, 2013)

R290	**Log p-h diagram**		**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**COP**	**COP with DWH**	**COP**	**COP with DWH**	**COP diff. [%]**	**COP with DWH diff. [%]**
0	8.47	3.19	7.80	2.93	7.86	8.03
-5	6.78	2.99	6.54	2.80	3.49	6.22
-10	5.42	2.77	5.62	2.67	3.67	3.47
-15	4.85	2.64	4.92	2.55	1.53	3.57
-20	4.29	2.51	4.37	2.44	1.90	2.98
-25	3.87	2.41	3.93	2.34	1.44	2.99
-30	3.55	2.32	3.56	2.25	0.32	2.94
-35	3.19	2.20	3.26	2.16	2.04	1.89
-40	2.98	2.13	3.00	2.08	0.52	2.34

4.2.1 R744

The fixed values that have been used:

$\dot{m}_{R744}$ – 0.0233 kg/s

Compressor heat loss factor (f_G) – 5%

Isentropic efficiency of the compressors (η_s) – 80%

Gas cooler pressure – 85 bar

Gas cooler output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

The graph in figure 4.15 and table 4.10 show the results produced in CoolPack© and log p-h diagram. From the graph the COP is 5.63 at an evaporation temperature of 0 ºC. The total compressor effect is 3.12 kW at an evaporation temperature of -40 ºC. Table 4.10 show the produced results for the different evaporations temperatures, the difference in the COP values varies from 0.86 - 9.11%, the average difference 4.35%. The highest COP and gas cooler effect at an evaporation temperature of -40 ºC is 2.50 and 7.81 kW.

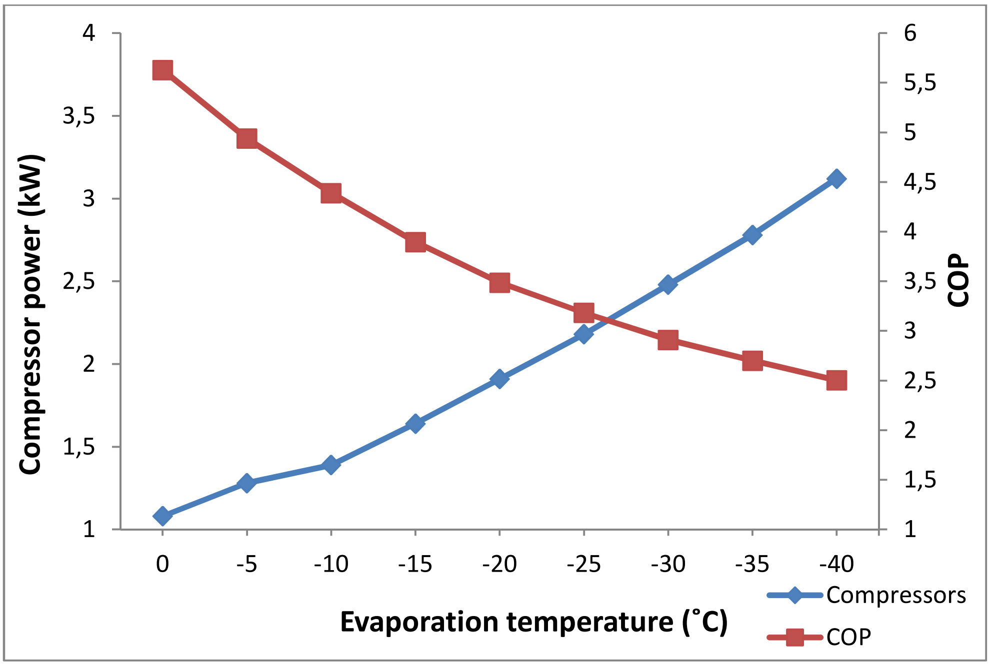

Figure 4.15: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using two stage R744 in CoolPack©

Table 4.10: R744 comparison between CoolPack© and log p-h diagram

	CoolPack©				Log p-h diagram			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Gas c. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Gas c. effect [kW]**	**COP**	**COP diff. [%]**
0	1.08	6.08	5.63		0.84	4.38	5.25	6.75
-5	1.28	6.32	4.94		0.94	4.43	4.73	4.25
-10	1.39	6.10	4.39		1.18	4.72	3.99	9.11
-15	1.64	6.39	3.90		1.40	4.99	3.58	8.21
-20	1.91	6.66	3.49		1.47	5.08	3.46	0.86
-25	2.18	6.94	3.18		1.71	5.34	3.13	1.57
-30	2.48	7.22	2.91		2.01	5.63	2.80	3.78
-35	2.78	7.51	2.70		2.07	5.71	2.76	-2.22
-40	3.12	7.81	2.50		2.47	6.02	2.44	2.40

4.2.2 R717

The fixed values that have been used:

$\dot{m}_{R717}$ – 0.0048 kg/s

Compressor heat loss factor (f_G) – 5%

Isentropic efficiency of the compressor (η_s) – 70%

Condensation temperature – 28 ºC

Condensation output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

The graph in figure 4.16 show the results produced in CoolPack©. It shows that the COP at an evaporation temperature of 0 ºC is 7.67. The total compressor effect is 3.00 kW and COP is 3.47 at an evaporation temperature of -40 ºC.

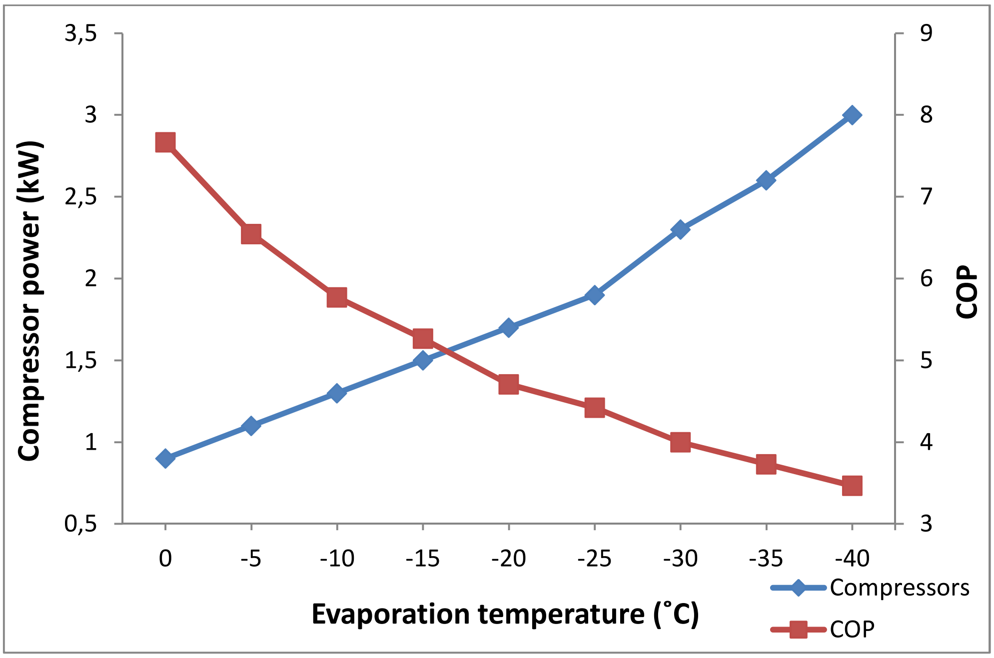

Figure 4.16: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using two stage R717 in CoolPack©

Table 4.11 and 4.12 show produced values from CoolPack© and MS Excel® sheet (Kolsaker, 2013). Table 4.11 shows the results when the heat pump only is used for heating of the house. CoolPack© has the overall (compressor, condenser and COP) highest values. It is a large difference between the results in CoolPack© and MS Excel® sheet (Kolsaker, 2013), from 1.56 – 18.44% and the average difference in COP is 11.46%. Table 4.12 shows the results when the DWH is implemented in the calculation, as shown in equation 3. The differences in COP are between 10.42 – 18.55% and the average difference is 14.25%. The highest COP and condenser effect at an evaporation temperature of -40 ºC is 2.48 and 10.40 kW.

Table 4.11: R717 comparison between CoolPack© and MS Excel® sheet (Kolsaker, 2013)

	CoolPack©			**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	0.9	6.90	7.67	0.73	5.51	7.55	1.56
-5	1.1	7.20	6.55	0.90	5.67	6.28	4.12
-10	1.3	7.50	5.77	1.09	5.83	5.37	6.93
-15	1.5	7.90	5.27	1.28	6.01	4.68	11.20

-20	1.7	8.00	4.71		1.49	6.19	4.15	11.89
-25	1.9	8.40	4.42		1.72	6.40	3.72	15.84
-30	2.3	9.20	4.00		1.96	6.61	3.37	15.75
-35	2.6	9.70	3.73		2.22	6.84	3.08	17.43
-40	3	10.40	3.47		2.50	7.10	2.83	18.44

Table 4.12: R717 comparison with DWH included

	CoolPack©				**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [°C]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	0.9	6.90	3.07		0.73	5.51	2.75	10.42
-5	1.1	7.20	2.97		0.90	5.67	2.64	11.11
-10	1.3	7.50	2.88		1.09	5.83	2.54	11.81
-15	1.5	7.90	2.83		1.28	6.01	2.44	13.78
-20	1.7	8.00	2.70		1.49	6.19	2.35	12.96
-25	1.9	8.40	2.67		1.72	6.40	2.26	15.36
-30	2.3	9.20	2.60		1.96	6.61	2.17	16.54
-35	2.6	9.70	2.54		2.22	6.84	2.09	17.72
-40	3	10.40	2.48		2.50	7.10	2.02	18.55

4.2.3 R290

The fixed values that have been used:

$\dot{m}_{R290}$ – 0.018 kg/s

Compressor heat loss factor (f_G) – 5%

Isentropic efficiency of the compressor (η_s) – 70%

Condensation temperature – 28 °C

Condensation output temperature – 10 °C

Evaporators superheat (ΔT_{SH}) – 2 K

The graph in figure 4.17 show the results produced in CoolPack©. It shows that the COP at an evaporation temperature of 0 °C is 7.50. The total compressor effect is 2.50 kW and the COP is 3.60 at an evaporation temperature of -40 °C.

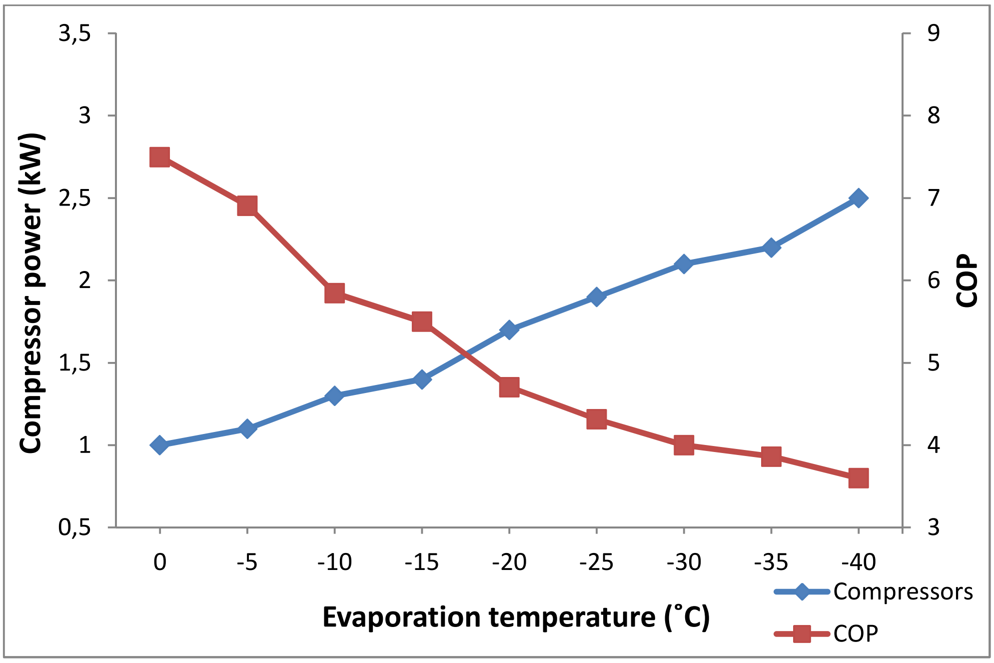

Figure 4.17: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using two stage R290 in CoolPack©

Table 4.13 and 4.14 show produced results from CoolPack© and MS Excel® sheet (Kolsaker, 2013). Table 4.13 shows the results when the heat pump only is used for heating of the house. CoolPack© has the overall (compressor, condenser and COP) highest values. It is a large difference between the results in CoolPack© and MS Excel® sheet (Kolsaker, 2013), from 3.93 – 16.67%. The average difference in COP is 9.25%. Table 4.14 shows the results when the DWH is implemented in the calculation, as shown in equation 3. The differences in COP are between 7.57 – 14.75% and the average difference is 10.68%. The highest COP and condenser effect at an evaporation temperature of -40 ºC is 2.44 and 9.00 kW.

Table 4.13: R290 comparison between CoolPack© and MS Excel® sheet (Kolsaker, 2013)

	CoolPack©				**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	1	7.50	7.50		0.80	6.20	7.80	-4.00
-5	1.1	7.60	6.91		0.96	6.28	6.54	5.35
-10	1.3	7.60	5.85		1.13	6.37	5.62	3.93
-15	1.4	7.70	5.50		1.31	6.46	4.92	10.55
-20	1.7	8.00	4.71		1.50	6.55	4.37	7.22

-25	1.9	8.20	4.32		1.70	6.66	3.93	9.03
-30	2.1	8.40	4.00		1.90	6.77	3.56	11.00
-35	2.2	8.50	3.86		2.11	6.89	3.26	15.54
-40	2.5	9.00	3.60		2.34	7.02	3.00	16.67

Table 4.14: R290 comparison with DWH included

	CoolPack©				**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	1	7.50	3.17		0.80	6.20	2.93	7.57
-5	1.1	7.60	3.10		0.96	6.28	2.80	9.68
-10	1.3	7.60	2.91		1.13	6.37	2.67	8.25
-15	1.4	7.70	2.85		1.31	6.46	2.55	10.53
-20	1.7	8.00	2.70		1.50	6.55	2.44	9.63
-25	1.9	8.20	2.62		1.70	6.66	2.34	10.69
-30	2.1	8.40	2.54		1.90	6.77	2.25	11.42
-35	2.2	8.50	2.50		2.11	6.89	2.16	13.60
-40	2.5	9.00	2.44		2.34	7.02	2.08	14.75

4.2.4 R410A

The fixed values that have been used:

$\dot{m}_{R410A}$ – 0.0293 kg/s

Compressor heat loss factor (f_G) – 5%

Isentropic efficiency of the compressor (η_s) – 70%

Condensation temperature – 28 ºC

Condensation output temperature – 10 ºC

Evaporators superheat (ΔT_{SH}) – 2 K

The graph in figure 4.18 show the results produced in CoolPack©. It shows that the COP at an evaporation temperature of 0 ºC is 8.00. The total compressor effect is 2.70 kW and the COP is 3.67 at an evaporation temperature of -40 ºC.

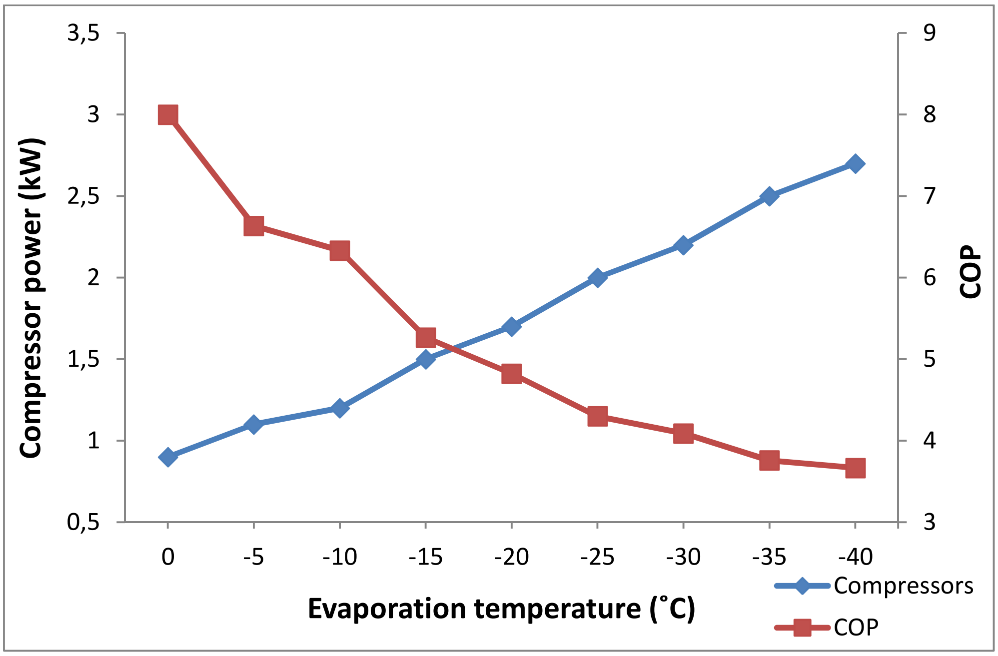

Figure 4.18: Compressor effect (kW) and COP plotted against evaporation temperature (ºC) using two stage R410A in CoolPack©

Table 4.15 and 4.16 shows produced results from CoolPack© and MS Excel® sheet (Kolsaker, 2013). Table 4.15 shows the results when the heat pump only is used for heating of the house. CoolPack© has the overall (compressor, condenser and COP) highest values. It is a large difference between the results in CoolPack© and MS Excel® sheet (Kolsaker, 2013), from 0.15 – 17.25%. The average difference in COP is 7.47%. Table 4.16 shows the results when the DWH is implemented in the calculation, as shown in equation 3. The differences in COP are between 8.30 – 17.67% and the average difference is 12.11%. The highest COP and condenser effect at an evaporation temperature of -40 ºC is 2.53 and 9.90 kW.

Table 4.15: R410A comparison between CoolPack© and MS Excel® sheet (Kolsaker, 2013)

	CoolPack©				MS Excel® (Kolsaker, 2013)			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	0.9	7.20	8.00		0.75	5.91	7.87	1.65
-5	1.1	7.30	6.64		0.91	6.06	6.65	-0.15
-10	1.2	7.60	6.33		1.08	6.22	5.75	10.09
-15	1.5	7.90	5.27		1.26	6.38	5.06	4.15
-20	1.7	8.20	4.82		1.45	6.55	4.51	6.87

-25	2	8.60	4.30		1.65	6.73	4.07	5.65
-30	2.2	9.00	4.09		1.87	6.91	3.70	10.54
-35	2.5	9.40	3.76		2.10	7.11	3.39	10.91
-40	2.7	9.90	3.67		2.34	7.32	3.13	17.25

Table 4.16: R410A comparison with DWH included

	CoolPack©				**MS Excel® (Kolsaker, 2013)**			
Evaporation temperature [ºC]	**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**		**Total comp. effect [kW]**	**Cond. effect [kW]**	**COP**	**COP diff. [%]**
0	0.9	7.20	3.17		0.75	5.91	2.87	10.45
-5	1.1	7.30	3.00		0.91	6.06	2.77	8.30
-10	1.2	7.60	3.00		1.08	6.22	2.67	12.36
-15	1.5	7.90	2.83		1.26	6.38	2.57	10.12
-20	1.7	8.20	2.76		1.45	6.55	2.48	11.29
-25	2	8.60	2.65		1.65	6.73	2.39	10.88
-30	2.2	9.00	2.62		1.87	6.91	2.30	13.91
-35	2.5	9.40	2.53		2.10	7.11	2.22	13.96
-40	2.7	9.90	2.53		2.34	7.32	2.15	17.67

4.2.5 SPF

To calculate the SPF, the values are the same as in chapter 4.1.5. This is done to make it easier to compare the results from the different graphs. To calculate the SPF equation 4 is used.

The figures 4.19 – 4.22 shows the SPF for R744, R717, R290 and R410A (file: Thesis-Two stage – SPF/Appendix: A). Figure 4.19 show that the SPF for R744 is 4.08.

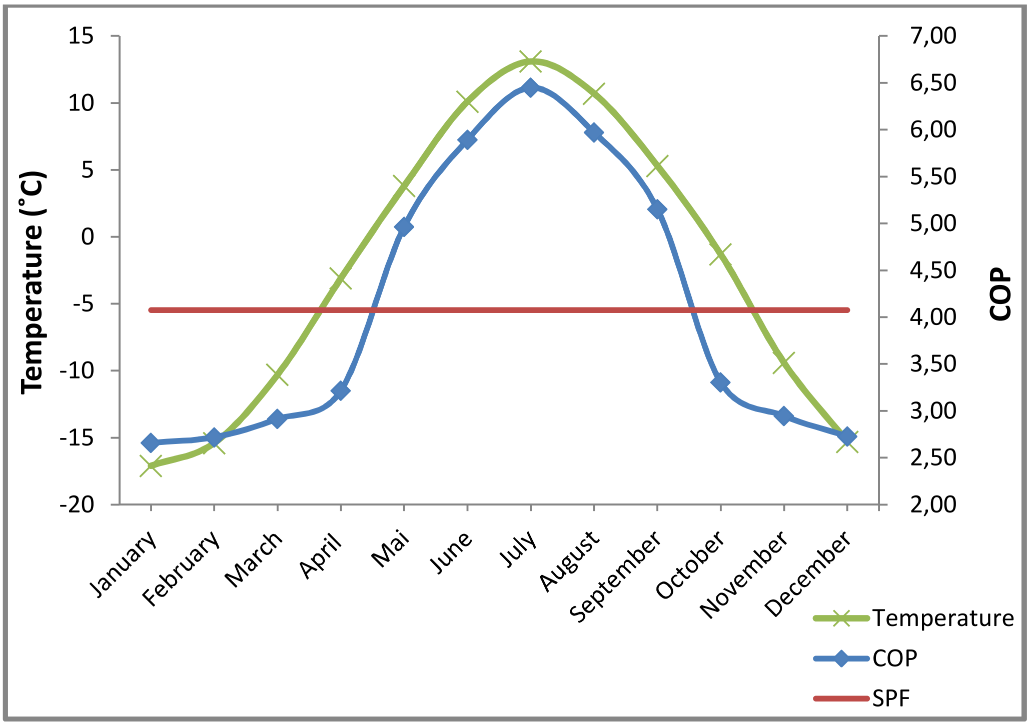

Figure 4.19: Normal outside temperature (ºC) each month, COP and SPF for two stage R744

Figure 4.20 show that the SPF for R717 is 2.58.

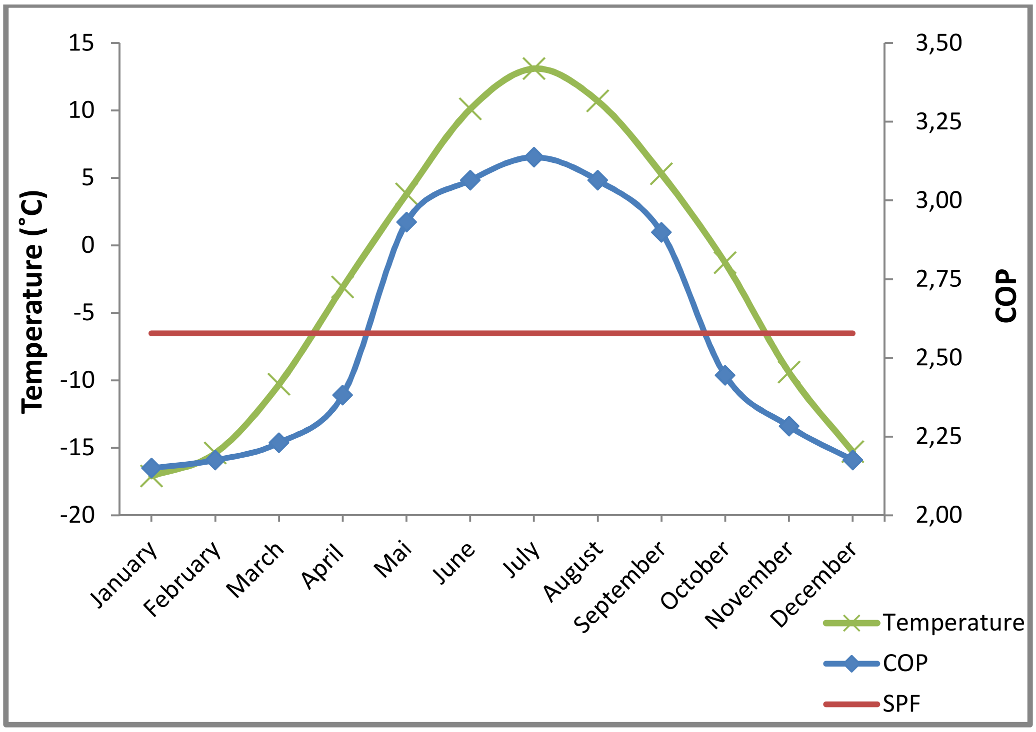

Figure 4.20: Normal outside temperature (ºC) each month, COP and SPF for two stage R717

Figure 4.21 show that the SPF for R290 is 2.67.

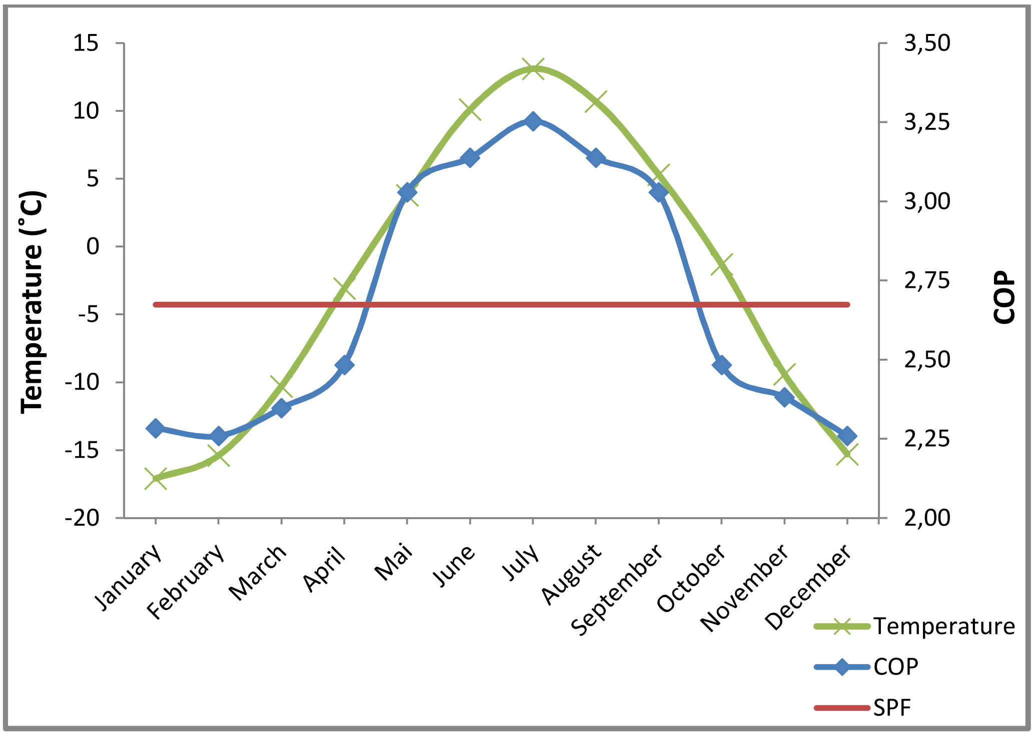

Figure 4.21: Normal outside temperature (ºC) each month, COP and SPF for two stage R290

Figure 4.22 show that the SPF for R410A is 2.69.

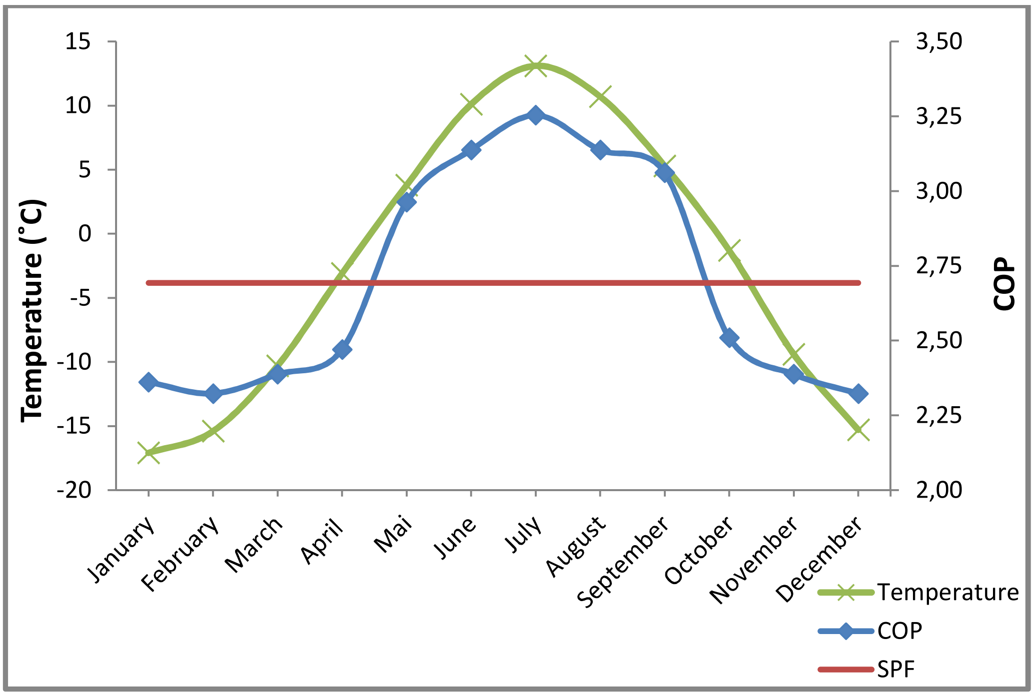

Figure 4.22: Normal outside temperature (ºC) each month, COP and SPF for two stage R410A

4.3 Electricity saving in one stage compression

By using a heat pump instead of electric heating panels and/or fossil heating (wood) it is possible to save energy. Heating demand in Karasjok is relative high in the winter months and by using a heat pump as the main heat source it should be possible to reduce the electricity consumption in the household.

The graphs (figure 4.23 – 4.26) are made by using the values from table 3.4 and implement the relevant refrigerants gas cooler/condenser value from chapter 4.1. The inside temperature is set to 22 ºC and the outside temperature is variable.
The tables (table 4.17 – 4.24) are made by using the average outside temperature each month (figure 3.3) in 2012 and 2013, the gas cooler/condenser effect and compressor effect from chapter 4.1, electricity consumption from figure 3.7 and 3.8, and heat loss value from table 3.3 (file: Electric calculation – Calc. 1 stg. and Output versus need 1 stg./Appendix: A). Assumptions made in these tables are:

- heat pump is operating 24 hours a day in the winter months (October – April)
- heat pump is operating 14 hours a day in the summer months (May – September)
- de-icing heat cable on the evaporator has an effect of 0.5 kW, it is used 70% of the time during the winter months
- efficiency of the gas cooler/condenser is 90%
- efficiency of the electric engine is 70% (ABB, 2009)

4.3.1 R744

Graph in figure 4.23 show the actual heating demand in the house and how much the R744 heat pump can deliver at different temperatures. The figure show that the R744 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -13 ºC.

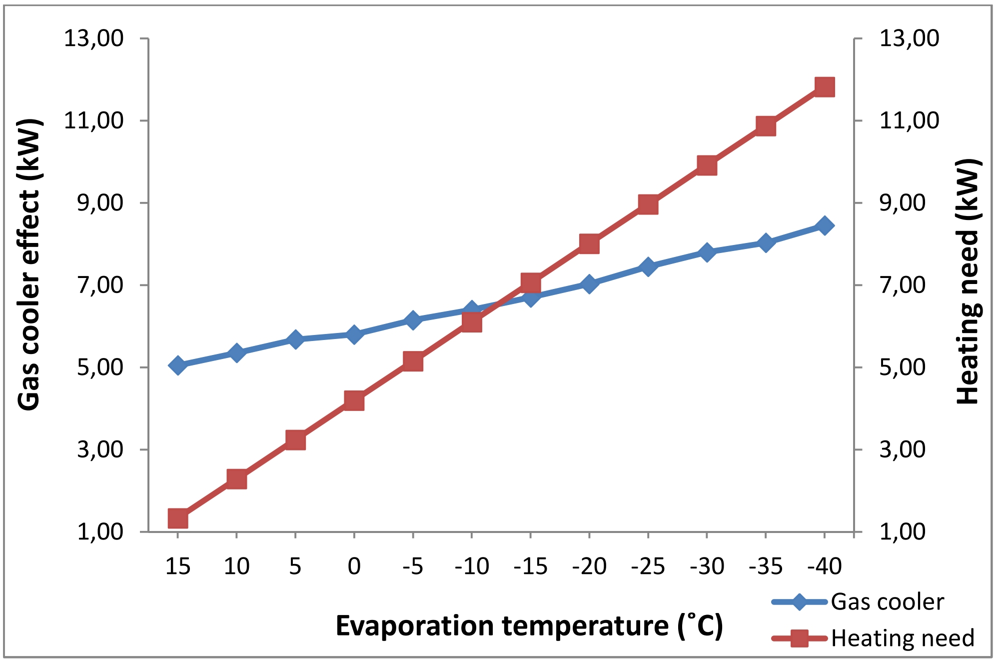

Figure 4.23: Gas cooler effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) using R744

Table 4.17 shows that by using a R744 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 19.88 – 59.72% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 54.30 – 70.87% above the demand.

The R744 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating that ranges from 17.97 – 118.62%. For the summer months it is need for additional heat in May, June and September with a range of 5.38 – 41.61%, and in July and August it is no need for additional heating.

Table 4.17: Electricity savings and heating need for R744 in 2012 (Hus og hjem, 1999)

R744 2012	**Sure plus energy from the heat pump [kWh]**	**El. used heating and DWH [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DWH [kWh]**	**Diff. between heat pump and heating need [%]**
January	2542.25	1944	23.53		5007.43	-96.97
February	2330.90	1836	21.23		5095.74	-118.62
March	2635.99	1404	46.74		3900.97	-47.99
April	2569.68	1224	52.37		3431.94	-33.56
May	1732.96	792	54.30		2454.06	-41.61

June	1693.44	612	63.86		1784.61	-5.38
July	1729.92	504	70.87		1418.53	18.00
August	1742.08	540	69.00		1617.13	7.17
September	1676.22	576	65.64		2182.71	-30.22
October	2681.38	1080	59.72		3163.33	-17.97
November	2563.20	1368	46.63		3775.13	-47.28
December	2516.21	2016	19.88		5362.06	-113.10

Table 4.18 shows that by using a R744 heat pump it is possible to reduce the electricity usage substantially. In the winter months the heat pump produces 23.28 – 62.19% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 58.70 – 77.15% above the demand.

The R744 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 20.79 – 102.40%. For the summer months it is need for additional heat in May and September with a range of 5.30 – 9.03%, and in June, July and August it is no need for additional heat.

Table 4.18: Electricity savings and heating need for R744 in 2013 (Hus og hjem, 1999)

R744 2013	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DWH [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DWH [kWh]**	**Diff. between heat pump and heating need [%]**
January	2570.52	1692	34.18		4908.13	-90.94
February	2323.10	1440	38.01		4215.33	-81.45
March	2537.04	1944	23.38		5135.09	-102.40
April	2609.28	1116	57.23		3349.57	-28.37
May	1743.38	720	58.70		1900.84	-9.03
June	1677.90	540	67.82		1139.40	32.09
July	1736.00	468	73.04		1149.01	33.81
August	1733.40	396	77.15		1347.61	22.26
September	1681.68	576	65.75		1770.88	-5.30
October	2665.75	1008	62.19		3220.07	-20.79
November	2520.00	1440	42.86		4104.60	-62.88
December	2590.61	1548	40.25		4610.24	-77.96

4.3.2 R717

Graph in figure 4.24 show the actual heating demand in the house and how much the R717 heat pump can deliver at different temperatures. The figure show that the R717 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -16 ºC.

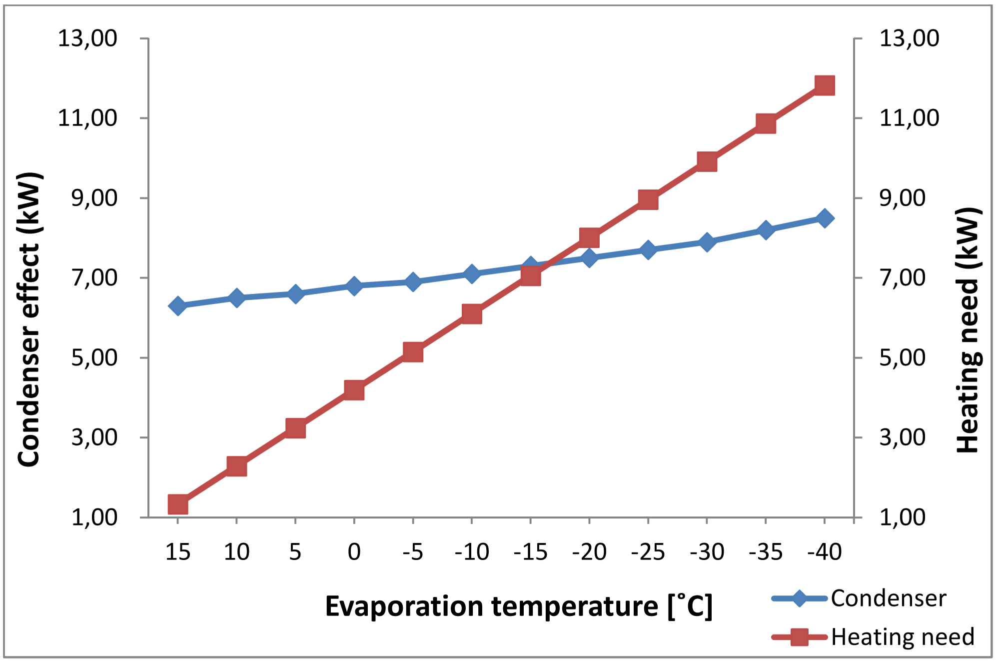

Figure 4.24: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) using R717

Table 4.19 shows that by using a R717 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 35.06 – 67.40% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 62.83 – 77.02% above the demand.

The R717 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 8.55 – 80.24%, except for October. For the summer months it is need for additional heat in May and September with a range of 5.41 – 15.16%, and in June, July and August it is no need for additional heat.

Table 4.19: Electricity savings and heating need for R717 in 2012 (Hus og hjem, 1999)

R717 2012	Sure plus energy from the heat pump [kWh]	El. used for heating and DWH [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DWH [kWh]	Diff. between heat pump and heating need [%]
January	3089.83	1944	37.08		5007.43	-62.06
February	2827.15	1836	35.06		5095.74	-80.24
March	3220.78	1404	56.41		3900.97	-21.12
April	3161.52	1224	61.28		3431.94	-8.55
May	2130.94	792	62.83		2454.06	-15.16
June	2096.64	612	70.81		1784.61	14.88
July	2193.44	504	77.02		1418.53	35.33
August	2180.85	540	75.24		1617.13	25.85
September	2070.60	576	72.18		2182.71	-5.41
October	3313.03	1080	67.40		3163.33	4.52
November	3116.88	1368	56.11		3775.13	-21.12
December	3037.75	2016	33.64		5362.06	-76.51

Table 4.20 shows that by using a R717 heat pump it is possible to reduce the electricity usage considerably. In the winter months the heat pump produces 36.64 – 69.55% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 66.65 – 81.93% above the demand.

The R717 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 5.44 – 67.36%, except for October. For the summer months it is no need for any additional heating.

Table 4.20: Electricity savings and heating need for R717 in 2013 (Hus og hjem, 1999)

R717 2013	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3098.76	1692	45.40		4908.13	-58.39
February	2826.43	1440	49.05		4215.33	-49.14
March	3068.26	1944	36.64		5135.09	-67.36
April	3176.64	1116	64.87		3349.57	-5.44

May	2159.15	720	66.65		1900.84	11.96
June	2131.08	540	74.66		1139.40	46.53
July	2203.85	468	78.76		1149.01	47.86
August	2191.27	396	81.93		1347.61	38.50
September	2096.64	576	72.53		1770.88	15.54
October	3310.06	1008	69.55		3220.07	2.72
November	3088.08	1440	53.37		4104.60	-32.92
December	3141.91	1548	50.73		4610.24	-46.73

4.3.3 R290

Graph in figure 4.25 show the actual heating demand in the house and how much the R290 heat pump can deliver at different temperatures. The figure show that the R290 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -18 ºC.

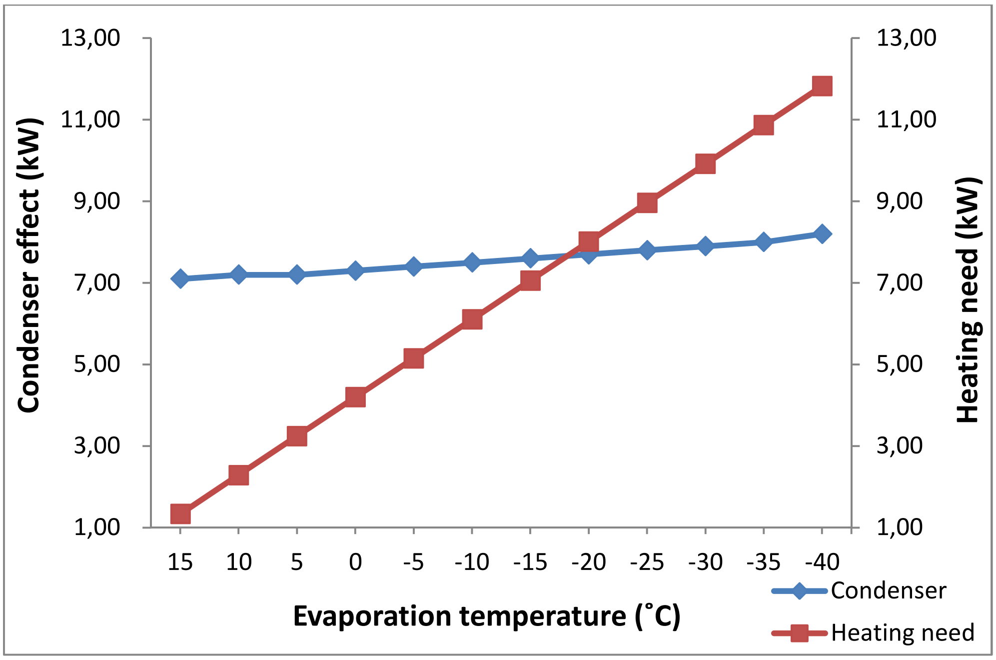

Figure 4.25: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) using R290

Table 4.21 shows that by using a R290 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 36.06 – 67.40% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 65.91 – 79.28% above the demand.

The R290 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 0.77 – 72.47%, except for October. For the summer months it is only need for additional heating in May, with a value of 5.63%.

Table 4.21: Electricity savings and heating need for R290 in 2012 (Hus og hjem, 1999)

R290 2012	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3246.82	1944	40.13		5007.43	-54.23
February	2954.52	1836	37.86		5095.74	-72.47
March	3455.88	1404	59.37		3900.97	-12.88
April	3405.60	1224	64.06		3431.94	-0.77
May	2323.20	792	65.91		2454.06	-5.63
June	2312.94	612	73.54		1784.61	22.84
July	2432.57	504	79.28		1418.53	41.69
August	2412.17	540	77.61		1617.13	32.96
September	2268.00	576	74.60		2182.71	3.76
October	3337.58	1080	67.64		3163.33	5.22
November	3344.40	1368	59.10		3775.13	-12.88
December	3180.60	2016	36.62		5362.06	-68.59

Table 4.22 shows that by using a R290 heat pump it is possible to reduce the electricity usage markedly. In the winter months the heat pump produces 39.66 – 71.84% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 69.73 – 83.73% above the demand.

The R290 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 24.91 – 59.40%, except for April and October. For the summer months it is no need for any additional heating.

Table 4.22: Electricity savings and heating need for R290 in 2013 (Hus og hjem, 1999)

R290 2013	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3269.14	1692	48.24		4908.13	-50.14
February	2989.73	1440	51.84		4215.33	-40.99
March	3221.52	1944	39.66		5135.09	-59.40
April	3427.20	1116	67.44		3349.57	2.27
May	2378.75	720	69.73		1900.84	20.09
June	2362.50	540	77.14		1139.40	51.77
July	2454.70	468	80.93		1149.01	53.19
August	2434.31	396	83.73		1347.61	44.64
September	2312.94	576	75.10		1770.88	23.44
October	3579.38	1008	71.84		3220.07	10.04
November	3286.08	1440	56.18		4104.60	-24.91
December	3329.40	1548	53.51		4610.24	-38.47

4.3.4 R410A

Graph in figure 4.26 show the actual heating demand in the house and how much the R410A heat pump can deliver at different temperatures. The figure show that the R410A heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -18 ºC.

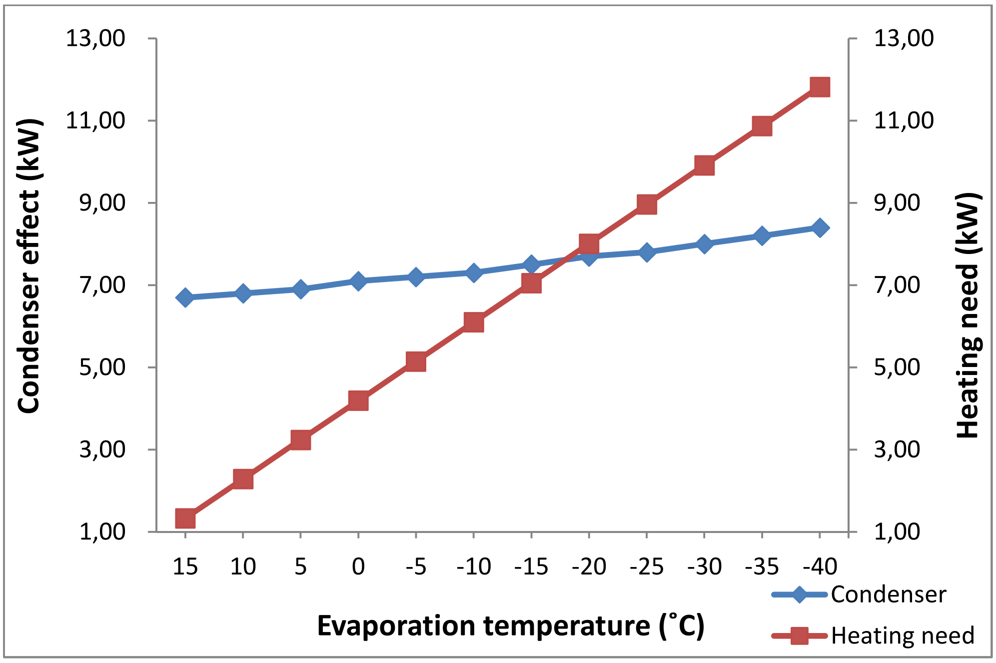

Figure 4.26: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) using R410A

Table 4.23 shows that by using a R410A heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 36.17 – 68.86% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 64.47 – 78.08% above the demand.

The R410A heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 3.64 – 73.33%, except for October. For the summer months it is need for additional heat in May and September with a range of 0.62 – 10.10%, and in June, July and August it is no need for additional heat.

Table 4.23: Electricity savings and heating need for R410A in 2012 (Hus og hjem, 1999)

R410A 2012	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DHW [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DHW [kWh]**	**Diff. between heat pump and heating need [%]**
January	3204.41	1944	39.33		5007.43	-56.27
February	2939.90	1836	37.55		5095.74	-73.33
March	3368.83	1404	58.32		3900.97	-15.80
April	3311.28	1224	63.04		3431.94	-3.64
May	2229.02	792	64.47		2454.06	-10.10

June	2085.72	612	70.66		1784.61	14.44
July	2299.33	504	78.08		1418.53	38.31
August	2286.75	540	76.39		1617.13	29.28
September	2169.30	576	73.45		2182.71	-0.62
October	3467.78	1080	68.86		3163.33	8.78
November	3260.16	1368	58.04		3775.13	-15.80
December	3158.28	2016	36.17		5362.06	-69.78

Table 4.24 shows that by using a R410A heat pump it is possible to reduce the electricity usage considerably. In the winter months the heat pump produces 36.17 – 68.86% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 64.47 – 78.08% above the demand.

The R410A heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 0.98 – 61.19%, except for October. For the summer months it is no need for additional heating.

Table 4.24: Electricity savings and heating need for R410A in 2013 (Hus og hjem, 1999)

R410A 2013	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DHW [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DHW [kWh]**	**Diff. between heat pump and heating need [%]**
January	3229.70	1692	47.61		4908.13	-51.97
February	2942.02	1440	51.05		4215.33	-43.28
March	3185.81	1944	38.98		5135.09	-61.19
April	3317.04	1116	66.36		3349.57	-0.98
May	2147.87	720	66.48		1900.84	11.50
June	2242.80	540	75.92		1139.40	49.20
July	2313.65	468	79.77		1149.01	50.34
August	2306.71	396	82.83		1347.61	41.58
September	2199.12	576	73.81		1770.88	19.47
October	3461.83	1008	70.88		3220.07	6.98
November	3218.40	1440	55.26		4104.60	-27.54
December	3263.18	1548	52.56		4610.24	-41.28

4.4 Electricity saving in two stage compression

These produced results can show if a two stage compression heat pump will be more sufficient in use during low outside temperature rather than a one stage compression heat pump. Thereby finding out which heat pump and refrigerant the electricity savings can be largest at these temperatures.

The graphs (figure 4.27 – 4.30) are made by using the values from table 3.4 and implement the relevant refrigerants gas cooler/condenser value from chapter 4.2. The inside temperature is set to 22 ºC and the outside temperature is variable.
The tables (table 4.25 – 4.32) are made by using the average outside temperature each month (figure 3.3) in 2012 and 2013, the gas cooler/condenser effect and compressor effect from chapter 4.2, electricity consumption from figure 3.7 and 3.8, and heat loss value from table 3.3 (file: Electric calculation – Calc. 2 stg. and Output versus need 2 stg./Appendix: A). Assumptions made in these tables are:

- heat pump is operating 24 hours a day in the winter months (October – April)
- heat pump is operating 14 hours a day in the summer months (May – September)
- de-icing heat cable on the evaporator has an effect of 0.5 kW, it is used 70% of the time during the winter months
- efficiency of the gas cooler/condenser is 90%
- efficiency of the electric engine is 70% (ABB, 2009)

4.4.1 R744

Graph in figure 4.27 show the actual heating demand in the house and how much the R744 heat pump can deliver at different temperatures. The graph show that the R744 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -10 ºC.

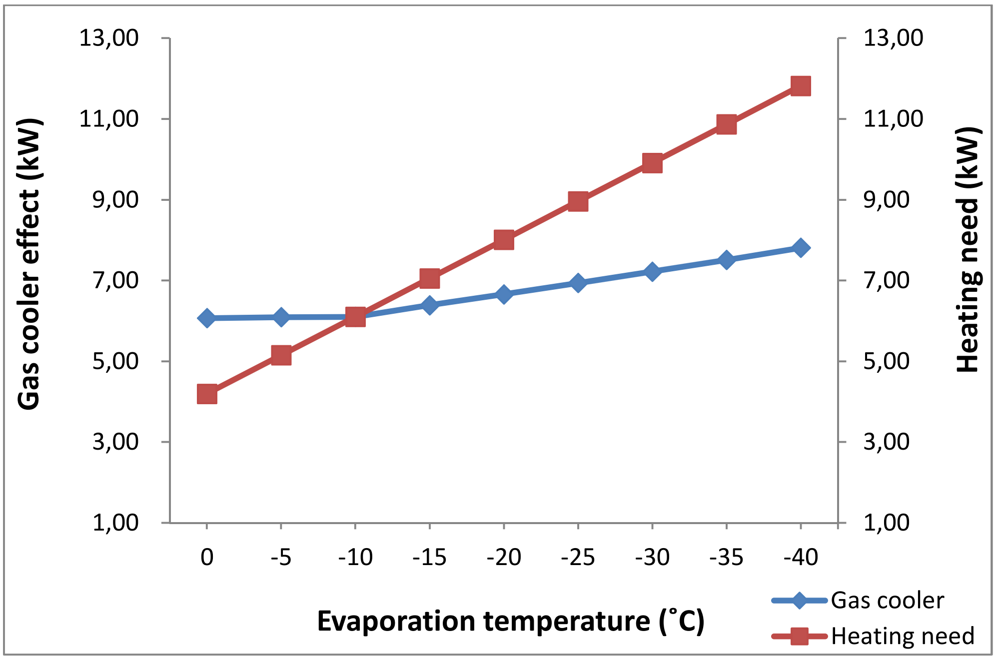

Figure 4.27: Gas cooler effect (kW) and heating need (kW) plotted against evaporation temperature (°C) for two stag R744

Table 4.25 shows that by using a R744 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 12.62 – 55.27% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 51.06 – 70.40% above the demand.

The R744 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 31.03 – 132.41%. For the summer months it is need for additional heat in May, June and September with a range of 9.17 – 51.64%, and in July and August it is no need for additional heat.

Table 4.25: Electricity savings and heating need for R744 in 2012 for two stage compression (Hus og hjem, 1999)

R744 2012	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	2339.88	1944	16.92		5007.43	-114.00
February	2162.47	1836	15.10		5095.74	-135.64
March	2416.51	1404	41.90		3900.97	-61.43
April	2360.16	1224	48.14		3431.94	-45.41

May	1618.39	792	51.06		2454.06	-51.64
June	1634.64	612	62.56		1784.61	-9.17
July	1702.58	504	70.40		1418.53	16.68
August	1666.56	540	67.60		1617.13	2.97
September	1586.34	576	63.69		2182.71	-37.59
October	2414.28	1080	55.27		3163.33	-31.03
November	2332.08	1368	41.34		3775.13	-61.88
December	2307.14	2016	12.62		5362.06	-132.41

Table 4.26 shows that by using a R744 heat pump it is possible to reduce the electricity usage considerably. In the winter months the heat pump produces 16.68 – 58.17% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 57.30 – 76.03% above the demand.

The R744 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 7.81 – 106.28%. For the summer months it is need for additional heat in May and September with a range of 7.81 – 12.74%, and in June, July and August it is no need for additional heat.

Table 4.26: Electricity savings and heating need for R744 in 2013 for two stage compression (Hus og hjem, 1999)

R744 2013	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DHW [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DHW [kWh]**	**Diff. between heat pump and heating need [%]**
January	2379.31	1692	28.89		4908.13	-106.28
February	2127.55	1440	32.32		4215.33	-98.13
March	2333.18	1944	16.68		5135.09	-120.09
April	2346.48	1116	52.44		3349.57	-42.75
May	1686.09	720	57.30		1900.84	-12.74
June	1618.68	540	66.64		1139.40	29.61
July	1674.37	468	72.05		1149.01	31.38
August	1652.24	396	76.03		1347.61	18.44
September	1642.62	576	64.93		1770.88	-7.81
October	2441.06	1008	58.71		3220.07	-31.91
November	2337.12	1440	38.39		4104.60	-75.63

December	2351.78	1548	34.18		4610.24	-96.03

4.4.2 R717

Graph in figure 4.28 show the actual heating demand in the house and how much the R717 heat pump can deliver at different temperatures. The graph show that the R717 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -20 ºC.

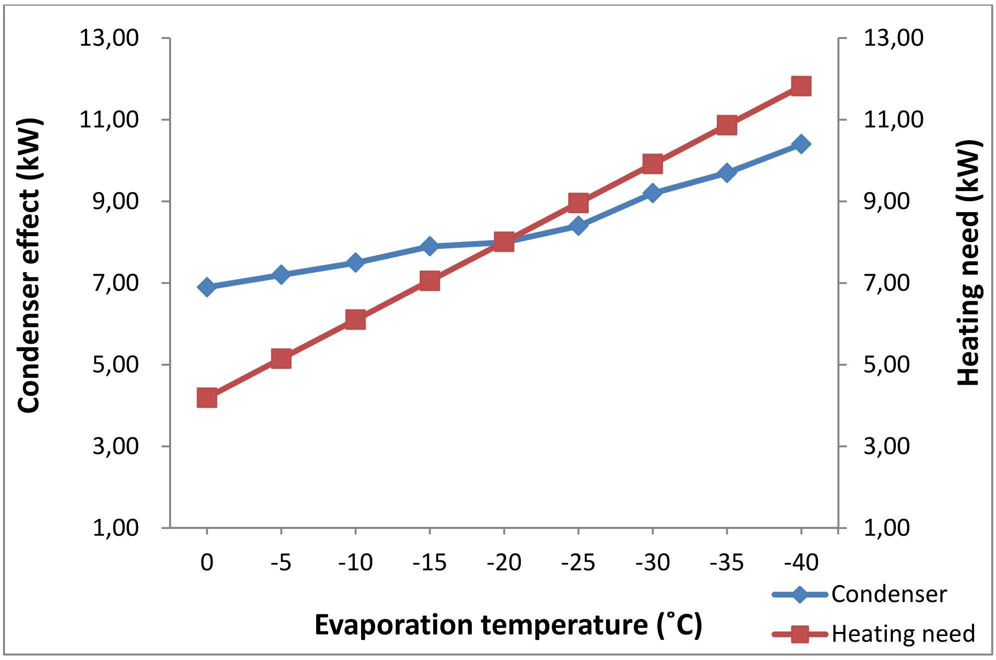

Figure 4.28: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) for two stage R717

Table 4.27 shows that by using a R717 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 32.16 – 66.62% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 61.99 – 76.96% above the demand.

The R717 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 10.72 – 84.47%, except for October. For the summer months it is need for additional heat in May and September with a range of 7.51 – 17.78%, and in June, July and August it is no need for additional heat.

Table 4.27: Electricity savings and heating need for R717 in 2012 for two stage compression (Hus og hjem, 1999) and (ABB, 2009)

R717 2012	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3013.94	1944	35.50		5007.43	-66.14
February	2762.42	1836	33.54		5095.74	-84.47
March	3153.82	1404	55.48		3900.97	-23.69
April	3099.60	1224	60.51		3431.94	-10.72
May	2083.63	792	61.99		2454.06	-17.78
June	2096.64	612	70.81		1784.61	14.88
July	2187.79	504	76.96		1418.53	35.16
August	2179.11	540	75.22		1617.13	25.79
September	2030.28	576	71.63		2182.71	-7.51
October	3235.66	1080	66.62		3163.33	2.24
November	3052.08	1368	55.18		3775.13	-23.69
December	2971.54	2016	32.16		5362.06	-80.45

Table 4.28 shows that by using a R717 heat pump it is possible to reduce the electricity usage substantially. In the winter months the heat pump produces 35.24 – 68.88% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 66.71 – 81.96% above the demand.

The R717 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 8.09 – 71.05%, except for October. For the summer months it is no need for additional heating.

Table 4.28: Electricity savings and heating need for R717 in 2013 for two stage compression (Hus og hjem, 1999)

R717 2013	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3025.85	1692	44.08		4908.13	-62.21
February	2769.31	1440	48.00		4215.33	-52.22

March	3002.04	1944	35.24		5135.09	-71.05
April	3098.88	1116	63.99		3349.57	-8.09
May	2163.06	720	66.71		1900.84	12.12
June	2131.08	540	74.66		1139.40	46.53
July	2203.85	468	78.76		1149.01	47.86
August	2195.17	396	81.96		1347.61	38.61
September	2096.64	576	72.53		1770.88	15.54
October	3239.38	1008	68.88		3220.07	0.60
November	3013.92	1440	52.22		4104.60	-36.19
December	3071.98	1548	49.61		4610.24	-50.07

4.4.3 R290

Graph in figure 4.29 show the actual heating demand in the house and how much the R290 heat pump can deliver at different temperatures. The graph show that the R290 heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -20 ºC.

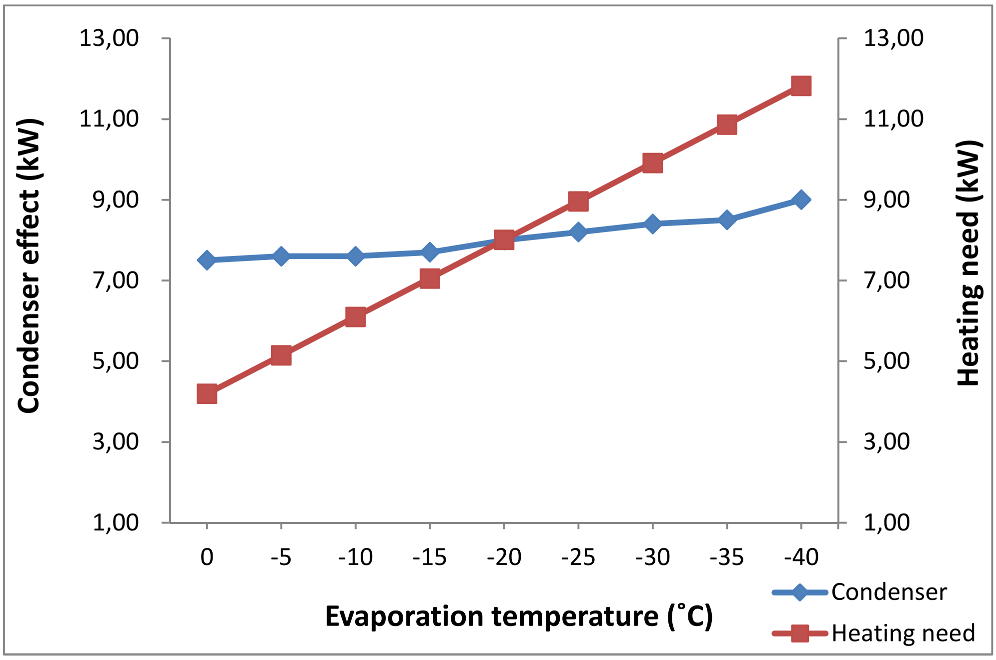

Figure 4.29: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) for two stage R290

Table 4.29 shows that by using a R290 heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 33.34 – 68.42% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 64.27 – 79.23% above the demand.

The R290 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 5.76 – 81.05%, except for October. For the summer months it is need for additional heat in May and September with a range of 0.74 – 10.70%, and in June, July and August it is no need for additional heat.

Table 4.29: Electricity savings and heating need for R290 in 2012 two stage compression (Hus og hjem, 1999)

R290 2012	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DHW [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DHW [kWh]**	**Diff. between heat pump and heating need [%]**
January	3087.60	1944	37.04		5007.43	-62.18
February	2814.62	1836	34.77		5095.74	-81.05
March	3283.27	1404	57.24		3900.97	-18.81
April	3245.04	1224	62.28		3431.94	-5.76
May	2216.87	792	64.27		2454.06	-10.70
June	2312.94	612	73.54		1784.61	22.84
July	2426.93	504	79.23		1418.53	41.55
August	2410.44	540	77.60		1617.13	32.91
September	2166.78	576	73.42		2182.71	-0.74
October	3419.42	1080	68.42		3163.33	7.49
November	3177.36	1368	56.95		3775.13	-18.81
December	3024.36	2016	33.34		5362.06	-77.30

Table 4.30 shows that by using a R290 heat pump it is possible to reduce the electricity usage substantially. In the winter months the heat pump produces 36.72 – 70.41% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 69.73 – 83.76% above the demand.

The R290 heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 2.74 – 67.16%, except for October. For the summer months it is no need for additional heating.

Table 4.30: Electricity savings and heating need for R290 in 2013 two stage compression (Hus og hjem, 1999)

R290 2013	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3103.22	1692	45.48		4908.13	-58.16
February	2848.61	1440	49.45		4215.33	-47.98
March	3071.98	1944	36.72		5135.09	-67.16
April	3260.16	1116	65.77		3349.57	-2.74
May	2378.75	720	69.73		1900.84	20.09
June	2373.84	540	77.25		1139.40	52.00
July	2458.61	468	80.96		1149.01	53.27
August	2438.21	396	83.76		1347.61	44.73
September	2312.94	576	75.10		1770.88	23.44
October	3406.78	1008	70.41		3220.07	5.48
November	3125.52	1440	53.93		4104.60	-31.33
December	3163.49	1548	51.07		4610.24	-45.73

4.4.4 R410A

Graph in figure 4.30 show the actual heating demand in the house and how much the R410A heat pump can deliver at different temperatures. The graph show that the R410A heat pump is capable of covering the DWH and heating of the house down to a temperature of approximate -22 ºC.

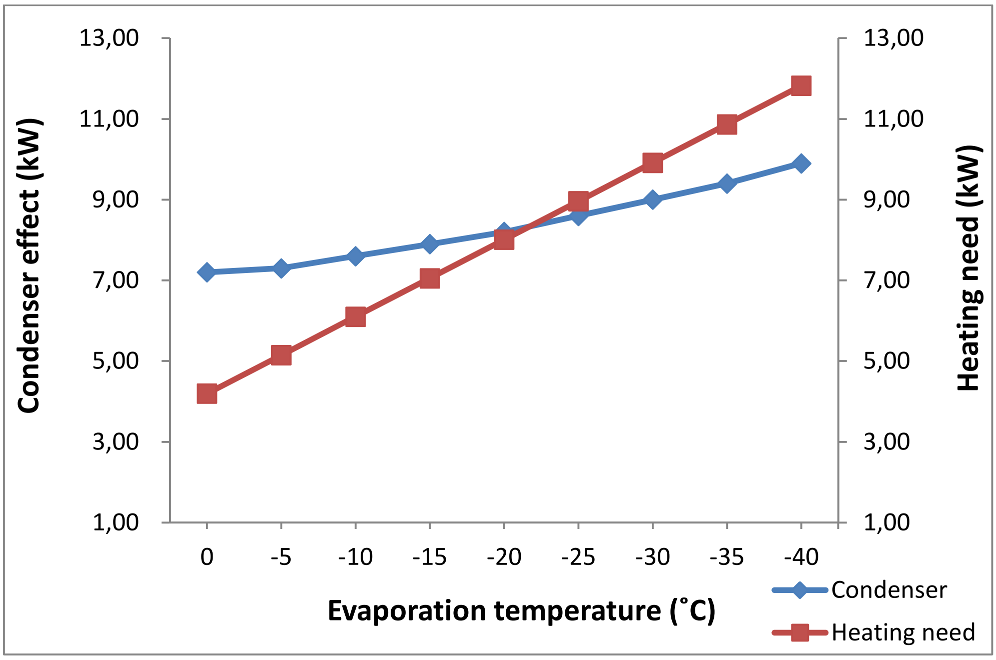

Figure 4.30: Condenser effect (kW) and heating need (kW) plotted against evaporation temperature (ºC) for two stage R410A

Table 4.31 shows that by using a R410A heat pump it is possible to reduce the electricity usage. In the winter months the heat pump produces 34.68 – 67.41% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 62.64 – 78.12% above the demand.

The R410A heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 8.21– 77.40%, except for October. For the summer months it is need for additional heat in May and September with a range of 5.95 – 15.75%, and in June, July and August it is no need for additional heat.

Table 4.31: Electricity savings and heating need for R410A in 2012 two stage compression (Hus og hjem, 1999)

R410A 2012	Sure plus energy from the heat pump [kWh]	El. used for heating and DHW [kWh]	Diff. between heat pump and el. use [%]		Heating need for house and DHW [kWh]	Diff. between heat pump and heating need [%]
January	3125.54	1944	37.80		5007.43	-60.21
February	2872.39	1836	36.08		5095.74	-77.40
March	3240.86	1404	56.68		3900.97	-20.37
April	3171.60	1224	61.41		3431.94	-8.21

May	2120.09	792	62.64		2454.06	-15.75
June	2199.12	612	72.17		1784.61	18.85
July	2303.24	504	78.12		1418.53	38.41
August	2290.65	540	76.43		1617.13	29.40
September	2060.10	576	72.04		2182.71	-5.95
October	3313.78	1080	67.41		3163.33	4.54
November	3136.32	1368	56.38		3775.13	-20.37
December	3086.11	2016	34.68		5362.06	-73.75

Table 4.32 shows that by using a R410A heat pump it is possible to reduce the electricity usage greatly. In the winter months the heat pump produces 37.57 – 69.55% above the actually demand for heating and DWH, and for the summer months the heat pump is able to produce 68.29 – 82.83% above the demand.

The R410A heat pump is not able to cover the total heating need for the house and DWH, during the winter months the need for additional heating ranges from 5.42 – 64.92%, except for October. For the summer months it is no need for additional heating.

Table 4.32: Electricity savings and heating need for R410A in 2013 for two stage compression (Hus og hjem, 1999)

R410A 2013	**Sure plus energy from the heat pump [kWh]**	**El. used for heating and DHW [kWh]**	**Diff. between heat pump and el. use [%]**		**Heating need for house and DHW [kWh]**	**Diff. between heat pump and heating need [%]**
January	3141.17	1692	46.13		4908.13	-56.25
February	2856.00	1440	49.58		4215.33	-47.60
March	3113.64	1944	37.57		5135.09	-64.92
April	3177.36	1116	64.88		3349.57	-5.42
May	2270.69	720	68.29		1900.84	16.29
June	2242.80	540	75.92		1139.40	49.20
July	2313.65	468	79.77		1149.01	50.34
August	2306.71	396	82.83		1347.61	41.58
September	2204.58	576	73.87		1770.88	19.67
October	3310.80	1008	69.55		3220.07	2.74
November	3110.40	1440	53.70		4104.60	-31.96
December	3174.65	1548	51.24		4610.24	-45.22

4.5 Electricity savings at low temperature

Produced results show how much of the heating and DWH demand the current refrigerants are able of covering at low outside temperature between -20 ℃ to -30 ℃. This is done for one stage and two stage heat pumps.

The tables (table 4.33 – 4.40) are made by using the number of days with low outside temperatures in 2012 and 2013 (figure 3.4 and 3.5), the gas cooler/condenser effect and compressor effect from chapter 4.1 and 4.2, heat loss value from table 3.3 and uses the same calculation method that is used in chapter 4.3 and 4.4. The evaporation temperature is 5 ° lower than the outside temperature that is registered in figure 3.4 and 3.5 (file: Electric calculation – Calc. 1 stg. and Calc. 2 stg./Appendix: A). The percentage of heat that the heat pump can cover is the same for both 2012 and 2013.

4.5.1 One stage compression

Table 4.33 shows that a R744 heat pump is able to cover 31.73 – 41.21% of the heating need and the DWH for the house at these low temperatures.

Table 4.33: R744 heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2059.20	41.21
-30	4301.36	1541.76	35.84
-35	3807.16	1208.06	31.73

Table 4.34 shows that a R717 heat pump is able to cover 37.09 – 49.71% of the heating need and the DWH for the house at these low temperatures.

Table 4.34: R717 heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2484.14	49.71
-30	4301.36	1839.84	42.77

-35	3807.16	1411.97	37.09

Table 4.35 shows that a R290 heat pump is able to cover 37.55 – 51.31% of the heating need and the DWH for the house at these low temperatures.

Table 4.35: R290 heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2564.02	51.31
-30	4301.36	1880.64	43.72
-35	3807.16	1429.63	37.55

Table 4.36 shows that a R410A heat pump is able to cover 38.20 – 51.40% of the heating need and the DWH for the house at these low temperatures.

Table 4.36: R410A heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2568.38	51.40
-30	4301.36	1895.52	44.07
-35	3807.16	1454.21	38.20

4.5.2 Two stage compression

Table 4.37 shows that a R744 heat pump is able to cover 21.16 – 27.89% of the heating need and the DWH for the house at these low temperatures.

Table 4.37: R744 two stage heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	1393.39	27.89
-30	4301.36	1009.92	23.48

-35	3807.16	805.63	21.16

Table 4.38 shows that a R717 heat pump is able to cover 29.45 – 39.64% of the heating need and the DWH for the house at these low temperatures.

Table 4.38: R717 two stage heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	1980.58	39.64
-30	4301.36	1464.48	34.05
-35	3807.16	1121.28	29.45

Table 4.39 shows that a R290 heat pump is able to cover 31.35 – 42.88% of the heating need and the DWH for the house at these low temperatures.

Table 4.39: R290 two stage heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2142.82	42.88
-30	4301.36	1571.04	36.52
-35	3807.16	1193.47	31.35

Table 4.40 shows that a R410A heat pump is able to cover 33.48 – 44.48% of the heating need and the DWH for the house at these low temperatures.

Table 4.40: R410A two stage heat pump is able to cover at low temperature

Evaporation temperature [ºC]	Heating need for house and domestic water [kWh]	Sure plus energy from the heat pump [kWh]	The heat pump can cover [%]
-25	4996.90	2222.69	44.48
-30	4301.36	1650.24	38.37
-35	3807.16	1274.50	33.48

5. Discussion

5.1 One stage compression

The simulations in CoolPack©, log p-h diagram and MS Excel® sheet (Kolsaker, 2013) show that there are differences between the results in these simulations. Simulations show that there are only small differences between log p-h diagram and MS Excel® sheet (Kolsaker, 2013), the average COP is 0.23% and when DWH is included the average COP is 0.27%. The average difference between the results in CoolPack© and log p-h diagram ranges from 3.89 – 4.57%, and when the DWH is included in the calculations the average difference ranges from 1.31 – 4.57%.

When comparing the results from the different refrigerants (not the DWH results), R290 has the highest COP with a value of 3.03 at an evaporation temperature of -40 ºC. R717 has the highest condenser effect at this evaporation temperature with an effect of 8.52 kW and R290 has the lowest compressor effect of 2.71 kW. R744 has the overall lowest COP at all of the different evaporation temperatures compared with the other refrigerants. The COP for R744 at an evaporation temperature of -40 ºC is 2.59, while the COP for R717, R410A and R290 are 2.91, 2.97 and 3.03, respectively. At evaporation temperature of -10 ºC R290 has the highest COP, then R410A, R717 and R744 with the values 5.64, 5.52, 5.44 and 4.45.

When comparing the results when DWH is included in the calculation of the COP, the results show that R744 has the highest COP compared with the other refrigerants. At an evaporation temperature of -40 ºC the COP for R744, R290, R410A and R717 are 2.59, 2.17, 2.15 and 2.14.

The percentage average difference between CoolPack© and log p-h diagram can be seen as within the norm of $\pm \leq 5\%$. The low difference show theses simulations are good comparisons for confirmation of the produced results. Log p-h diagram and MS Excel® sheet (Kolsaker, 2013) has an almost zero percentage difference shows that these two simulations are attuned, this can be because these two simulations do not have the same amount of variables as CoolPack©.

The results show that the refrigerant R290 has overall highest COP for all evaporation temperatures when DWH is not included in the calculation. R290 has a relative high specific

heating capacity compared with the other refrigerants, it has a low compressor effect and it is at the same time able to produce a significant amount of heat (positive for a high COP).
When the DWH is included in the calculations for the R744 has the overall highest COP. The R744 heat pump operates at transcritical pressure thereby it is good suited for heating of large quantities of water.
The SPF results with implementation of DWH for all of the refrigerants show that R744 has the highest SPF of 4.11. R290, R410A and R717 have a SPF of 2.60, 2.54 and 2.51, respectively. The produced SPF result reflects the different refrigerants COP value when DWH is included in the calculation.

5.2 Two stage compression

Simulations show that there are only small differences between log p-h diagram and MS Excel® sheet (Kolsaker, 2013), the average COP is 2.53% and when DWH is included the average COP is 3.83%, which is within the ± ≤ 5%. Simulation result differences between CoolPack© and MS Excel® sheet (Kolsaker, 2013) are overall higher than between log p-h diagram and MS Excel® sheet (Kolsaker, 2013). The average COP results between the refrigerants ranges from 4.35 – 11.46% and when DWH is included in the calculation the COP results ranges from 4.35 – 14.25%.

When comparing the results from the different refrigerants (not the DWH results) it is R410A that has the highest COP with a value of 3.67 at an evaporation temperature of -40 ºC. R717 has the highest condenser effect at this evaporation temperature with an effect of 10.40 kW and R290 has the lowest compressor effect of 2.50 kW. R744 has the lowest COP at all the different evaporation temperatures compared with the other refrigerants. The COP for R744 at an evaporation temperature of -40 ºC is 2.50, while for R717, R290 and R410A the COP is 3.47, 3.60 and 3.67, respectively. At evaporation temperature of -10 ºC R410A has the highest COP, then R290, R717 and R744 with the values 6.33, 5.85, 5.77 and 4.36.
When comparing the results when DWH is included in the calculations of the COP, the results show that R744 has the highest COP compared with the other refrigerants. At an evaporation temperature of -40 ºC the COP for R410A, R744, R717 and R290 is 2.53, 2.50, 2.48 and 2.44.

The larger percentage difference between CoolPack© and MS Excel® sheet (Kolsaker, 2013) can be a result of some errors in some parts of the MS Excel® sheet (Kolsaker, 2013) calculation or that CoolPack© takes into account several more variables than MS Excel® sheet (Kolsaker, 2013). Some of the variables that CoolPack© uses are the compressor heat loss, pressure drop in pipelines, and calculation of effect in gas cooler/condenser and compressor. The two stage compression heat pumps has several more variables than a one stage compression heat pumps, thereby this can also be a factor for why the average percentage difference are larger in the two stage compression than in the one stage compression heat pumps.

The results show that R410A has the highest COP for all of the evaporation temperatures when DWH is not included in the calculations. When DWH is included in the calculations R744 has the highest COP from an evaporation temperature of 0 to -35 ºC, although R410A has the highest COP at an evaporation temperature of -40 ºC of 2.53. This is only 0.03 higher than R744 at this evaporation temperature.

SPF with implementation of DWH for all the refrigerants shows that R744 has the highest SPF of 4.08. R410A, R290 and R717 have a SPF of 2.69, 2.67 and 2.58, respectively. The produced SPF result reflects the different refrigerants COP when DWH is include in the calculations.

5.3 Electricity saving in one stage compression

From the produced results it is possible to see that all of the refrigerants are able to cover the actual heating and DWH demand for the household that otherwise would been covered by electricity. This can indicate that by using a heat pump as the main source of heating it is possible to save a significant amount of electricity each month, but this presuppose that the temperature throughout the month stays stabile at the current temperature. Fluctuations in outside temperature can have a negative effect on the produced results. Since all of the refrigerants are able to cover actual heating and DWH demand in the household, in the winter months the heat pumps are able to cover up to 69% above the actual demand and in the summer months the heat pumps are able to cover up 84% above the actual demand. This can indicate that it is important to have a good system of how often and for how long the heat pump shall operate, this can ensure that the heat pumps only run when it is a need for heating or DWH. Another aspect can be to ensure that there always is a heating demand in the house or DHW.

For the produced results regarding heating need for the house and DHW the results show that there is a need for up to 118.62% of additional heating when the R744 heat pumps heating capacity is included in the winter months. In the summer months there can be an additional need for up to 41.61% heating and in some summer months is can be a sure plus of produced heat, up to 33.81%. The average additional heating need for R744 heat pump is 39.28% and 32.58% in 2012 and 2013.

With the R717 heat pump there is a need for additional heating up to 80.24% in the winter months and in the summer months there can be an additional heating need with up to 15.16%. In the summer months there is also a sure plus of heat, up to 47.86%. The average additional heating need for R717 heat pump is 17.46% and 8.07% in 2012 and 2013.

For the R290 heat pump there is a need for additional heating up to 72.47% in the winter months and in the summer months there can be an additional heating need for up to 5.63%. In the summer months there is a sure plus of heat, up to 53.19%. The average additional heating need for R290 heat pump is 10.08% and 0.70% in 2012 and 2013.

When the R410A heat pump is operating there is an additional need for heating up to 73.33% in the winter months and in the summer months it can be an additional need for heat, up to 10.10%. The average additional heating need for R410A heat pump is 12.87% and 3.93% in 2012 and 2013.

For the month were the heat pump is not able to cover all of the heating need (almost all of the winter months) the additional heating can come from heating with fossils (wood, oil or natural gas) or with electric heating panels. The months were there is a sure plus of heating the heat pumps should have been running for fewer hours or the compressor capacity can be reduced.

5.4 Electricity saving in two stage compression

The produced results show that all of the two stage heat pumps are able to cover the electricity consumption in the household that is used for heating and DWH, hence it is possible to reduce the electricity consumption in the house. This show that by using a heat pump as the main source of heating and DWH instead of a pure electricity usage or fossil fuel, it is possible to reduce the use of electricity significantly. Although these results are dependent on a stabile outside temperature throughout the months, so fluctuations can have an influence on the produced results. All of the refrigerants are able to cover the heating and DWH demand in the household, in the winter months there can be a sure plus of heat produced up to 70.41% in the winter months and up to 83.76% in the summer months. This shows that by regulating the operational time for the heat pumps it is possible to reduce the electricity use even more or ensure that there always is a use of the heat that the heat pumps produces.

The produced result regarding the heating need for the house and for the DWH show that the R744 heat pump needs an additional heating up to 135.64% in the winter months and this includes the use of the heat pump. In the summer months there is an additional heating need up to 51.64% and in some of the summer months the heat pump produces 31.38% above the heating and DWH need in the house. The average additional heating need for R744 heat pump is 55.05% and 42.66% in 2012 and 2013.

For the R717 heat pump there is an additional heating need in the winter months up to 66.14%. In the summer months there is an additional need for heating up to 17.78% and in some of the summer months the heat pump produces more heat than is needed, this sure plus heat can be up to 47.86% above of the actual need. The average additional heating need for R717 heat pump is 19.70% and 9.88% in 2012 and 2013.

With the R290 heat pump there is an additional need for up to 81.05% of heat in the winter months. In the summer months there can be an additional need for up to 10.70% and in some of the summer months there is a sure plus heat from the heat pump up to 53.27%. The average additional heating need for R290 heat pump is 14.21% and 4.51% in 2012 and 2013.

For the R410A heat pump there is a need for additional heating in the winter months up to 77.40%. In some of the summer months there is an additional need for up to 15.75% of heating and in some summer months there is a sure plus of heat up to 50.34% above the need. The average additional heating need for R410A heat pump is 15.90% and 5.96% in 2012 and 2013.

Ass seen there is an additional need for heating in almost all of the winter months and in some summer months. In the most of the summer months there is a sure plus amount of heat produced of the heat pumps, hence it is important that the heat pumps only are operating when it is a need for heating or DWH. If operating when it is not a need for heating or DWH, it is a waste of electricity. It is also possible to reduce the efficiency of the heat pump or to use an inverter on the compressor.

5.5 Electricity saving at low temperatures

From the produced results it can be seen that for the one stage heat pumps are able to cover the heating and DWH need for the house down from 31.73% and up to 51.40%. This shows that by using a heat pump in low temperature temperatures it is able to cover a large part of the heating and DWH need for a household.

The two stage heat pumps are able to cover the needs down from 21.16% and up to 44.48% of the heating and DWH need in the house. As seen in the relevant tables R410A and R290 are able to cover the largest amount of the heating and DWH need at low temperatures, this can be because thermodynamic propertied for these two refrigerants or that they have highest output heat from the condenser.

The two stage heat pump is not able to cover as much of the heating need at the one stage heat pumps, the reason for this can be that the two stage heat pumps have a more efficient compression cycle and thereby do not produce as much heat as a single stage heat pump. The two stage cycle is more energy efficient at lower temperature (higher COP) than a single stage heat pump.

6. Conclusion

Following conclusions can be drawn from the study:

- The use of three methodologies, CoolPack©, log p-h diagram and MS Excel® sheet (Kolsaker, 2013) for the analysis builds confidence in obtained results.
- The low percentage difference between log p-h diagram and MS Excel® sheet (Kolsaker, 2013) shows that these two analyses are close in their numerical formulation for one stage compression.
- The relatively larger percentage difference in the one stage compression between CoolPack© and other methodologies MS Excel® sheet (Kolsaker, 2013) and log p-h diagram) is due to the number of variables taken into account for the study. CoolPack© numerical methodology includes variables such as heat and pressure losses expected in the system which were not taken into account in other methodologies.
- The analysis revealed that refrigerant R290 (propane) has highest coefficient of performance (COP) without domestic water heating (DWH) in a single stage heat pump cycle. However, with the inclusion of DWH refrigerant R744 (carbon dioxide) has the highest COP. The obtained results are consistent with the scientific literature and industrial development where R744 is increasingly being used for DWH.
- The single stage heat pump cycle analysis revealed that the highest energy savings are by the use of refrigerant R290 in colder climate. In this analysis, the average outside temperature each month was considered (evaporation temperature was 5 ºC lower). The average temperature was evaluated from the climatic data of Karasjok for the years of 2012 and 2013.
- The single stage analysis also revealed that the refrigerant R410A (a 50/50 mixture of HFCs: R-32 (difluoromethane) and R-125 (pentafluoroethane)) covers the highest amount of the heating need in comparison to other refrigerants in the colder climate (outside temperature of -20 ºC to -30 ºC).
- The analysis shows that the two stage heat pumps cycles has higher COP than the single stage compression heat pumps.

- The two stage heat pump analysis shows similar result in comparison to single stage heat pump. Refrigerant R744 was found to have highest COP for heating of the house and DWH. Similarly, Refrigerant R410A was best suited to cover the heating need.

Given study concludes that the natural refrigerants R744 and R290 have potential to operate in the colder climate. This clearly shows that the natural refrigerants are capable of replacing synthetic refrigerants used nowadays.

7. Future work

Following works can be conducted based on given study:

- The study can be extended with the inclusion of other natural refrigerants such as; ethane (C_2H_6 – R170), isobutan (C_4H_{10} – R600a), water (H_2O – R718), air (R729), ethylene (C_2H_4 – R1150), propylene (C_3H_8 – R1270), etc.

- Simulation packages like Apsen HYSYS® may be used to assess the heat pump cycles in further detail.

- Experimental verification can be performed at laboratory scale to verify the theoretical results.

- Study the economical aspect by using a heat pump with natural refrigerant versus a heat pump using synthetic refrigerant.

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Appendix: A – List of related files

The CoolPack© and log p-h diagram show one example of how the simulations was conducted for each refrigerant, and for one and two stage compression.

MS Excel® (Kolsaker, 2013):

Temperatures in Karasjok 2012 – 2013

House calculation

Electric calculations

CO_2 heat pump – Pre master

Intermediate and Kolsaker

Thesis-One stage

Thesis-Two stage

CoolPack©:

Mass flow – R290

Mass flow – R410A

Mass flow – R717

Mass flow – R744

One stage – R290

One stage – R410A

One stage – R717

One stage – R744

Two stage – R290

Two stage – R410A

Two stage – R717

Two stage – R744

Log p-h diagram:

One stage Log – R290

One stage Log – R410A

One stage Log – R717

One stage Log – R744

Two stage Log – R290

Two stage Log – R410A

Two stage Log – R717

Two stage Log – R744

Appendix: B – FMEA

Table B.1: FMEA worksheet (Rausland, 2005), (Folksam, 2009) and (Stene, 2014)

System: Domestic air to air heat pump, failures in colder climate

Performed by: Nils Eivind Eriksen

Ref drawing no: Outdoor unit

Date: 16.05.14

Page: 1 of 3

Description of unit			Description of failure			Effect of failure					
Ref. no.	Function	Opera-tional mode	Failure mode	Failure cause or mechanism	Detection of failure	On the subsystem	On the system function	Failure rate	Severity ranking	Risk reducing measures	Comments
2.1	Main heat source for the house hold, air and domestic water	Operat-ing	Outside fan failure	Foreign object in/on the fan Bearing failure Failure in fan engine	Low fan speed **5** Fan does not run **2** No heat produced **1**	If one of these failures occurs in the outside unit, it may not be able to perform its tasks – (next page)	If one of these failures occurs it can lead to a reduction in the heating capability – (next page)	**5** Frequent	**3** Minor	Installing a sensor that registers if the fan is not operating as it should.	When operation a heat pump in a colder climate it is important to ensure that the heat pump is – (next page)

			Damage on compressor	Corrosion in compressor Failure in start-up procedure	Low compressor performance **6** Failure in control unit **2** Failure in boiling of refrigerant **1**	adequately as it is designed to.	of the heat pump.	**4** Probable	**2** None	Install an alarm that will be triggered if the compressor performance is lower than expected.	maintained ass instructed in the user manual and that there are taken some precursors regarding the low outside temperature that the heat pump will operate in.
			Problem with de-icing on the evaporator	Failure with de-icing De-icing occurs to often Failure in 4-way vale	Low compressor performance **6** No heat produced **1** Uses more electricity **8** Visual detection **5**	If two or more failures occur in the outside unit, it may not be able to perform its tasks as it is design to.	If two or more failures occur within the same time period the heat pump may produce some heat or not any heat at all.	**4** Probable	**6** Major	Have an alarm that will be triggered it the ice layer is to thick.	

System: Domestic air to air heat pump, failures in colder climate | Performed by: Nils Eivind Eriksen

Ref drawing no: Outdoor unit | Date: 16.05.14 | Page: 3 of 3

			Icing of sump on outdoor heat pump unit	De-icing heat cables fails	Low fan speed **5** Fan does not run **2** No heat produced **1** Visual detection **5**			**4** Probable	**3** Minor	Install a second heat cable or an alarm that triggers if the ice is not removed within a certain time.	
			Icing/snow on outdoor heat pump unit	Rain and temp. below 0 ℃ Heavy snow fall Location of the heat pump	Low effect of the heat pump **5** Uses more electricity **8** No heat produced **1** Visual detection **5**			**3** Occasi-onal	**3** Minor	Installing a “house” around the heat pump and ensure that the heat pump is installed at a sufficient height.	

Table B.2: Detection of failure (Rausland, 2005)

Rank	Description
1-2	Very high probability that the defect will be detected. Verification and/or controls will almost certainly detect the existence of a deficiency or defect.
3-4	High probability that the defect will be detected. Verification and/or controls have a good chance of detecting the existence of a deficiency/defect.
5-7	Moderate probability that the defect will be detected. Verification and/or controls are likely to detect the existence of a deficiency or defect.
8-9	Low probability that the defect will be detected. Verification and/or control not likely to detect the existence of a deficiency or defect.
10	Very low (or zero) probability that the defect will be detected. Verification and/or controls will not or cannot detect the existence of a deficiency/defect.

Table B.3: Failure rate (Rausland, 2005)

1	Very unlikely	Once per 1000 years or more seldom
2	Remote	Once per 100 years
3	Occasional	Once per 10 years
4	Probable	Once per year
5	Frequent	Once per month or more often

Table B.4: Severity ranking (Rausland, 2005) (edited)

Ranking	Description
10	Failure will result in major customer dissatisfaction and cause non-system operation or non-compliance with government regulations (Catastrophic).
8-9	Failure will result in high degree of customer dissatisfaction and cause non-functionality of system (Critical).
6-7	Failure will result in customer dissatisfaction and annoyance and/or deterioration of part of system performance (Major).
3-5	Failure will result in slight customer annoyance and/or slight deterioration of part of system performance (Minor).
1-2	Failure is of such minor nature that the customer (internal or external) will probably not detect the failure (None).

Table B.1 show the FMEA, the significance of the numbers; detection of failure, failure rate and severity ranking can be seen in table B.2, B.3 and B.4.

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